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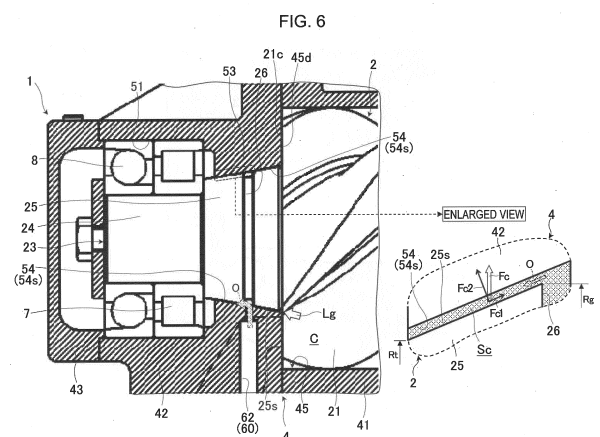
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(54) **SCREW COMPRESSOR**

(57) A screw compressor includes a screw rotor including: a rotor lobe part and a discharge-side shaft part; and a casing having a housing chamber that houses the rotor lobe part, a shaft hole through which the discharge-side shaft part passes, and a liquid feed path that introduces a liquid to the shaft hole. A second shaft section of the discharge-side shaft part disposed in the shaft hole and the shaft hole of the casing have a first opposed surface and a second opposed surface opposed to each other with a gap. The discharge-side shaft part has an annular groove on the first opposed surface, and the liquid feed path opens at a position opposed to the annular groove on the second opposed surface of the shaft hole. The second shaft section of the discharge-side shaft part and the shaft hole of the casing have, in an area remoter from the rotor lobe part than the annular groove, a tapered structure as a structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part.



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Description

Technical Field

5 **[0001]** The present invention relates to a screw compressor and more specifically relates to a screw compressor in which a shaft is sealed by supply of a liquid.

Background Art

10 **[0002]** Among screw compressors, there is an oil-flooded type screw compressor in which oil is supplied into working chambers in the compression process for the purposes of sealing of internal gaps generated between a screw rotor and a casing, cooling of a compressed gas, lubricating of sliding parts, and so forth. In the oil-flooded type screw compressor, the oil is mixed into the compressed gas to be discharged, and the oil is therefore separated from the compressed gas by using a separator. The oil separated by the separator is supplied into the working chambers again through a heat exchanger and a filter. In the oil-flooded type screw compressor, the oil separated by the separator is supplied also to bearings that rotatably support the screw rotor as lubricating oil besides the working chambers.

15 **[0003]** In gas compressors such as the screw compressor and a centrifugal compressor, compressed gas leaks through an annular gap formed between the outer circumferential surface of a shaft part of a rotor and the inner circumferential surface of a shaft hole of a casing (often referred to as shaft gap). The larger the leakage amount of the compressed gas, the lower the efficiency. Therefore, the shaft gap needs to be sealed. As one of measures for shaft sealing, there is a method in which a liquid such as lubricating oil is supplied to the shaft gap.

20 **[0004]** For example, in the oil-flooded type screw compressor, it is general to supply oil separated by a separator to the shaft gap to form an oil film, thereby suppressing the leakage of compressed gas from working chambers to a bearing chamber through a shaft gap. The oil supplied to the shaft gap is finally supplied to a bearing disposed in the bearing chamber.

25 **[0005]** Further, as a technique for shaft sealing by supplying a liquid to a shaft gap, for example, an oil film seal has been proposed in patent document 1. The oil film seal described in patent document 1 supplies seal oil at a higher pressure than the pressure in a case (casing) into a gap between the outer circumferential surface of a rotor, which penetrates the case and protrudes to the external, and the inner circumferential surfaces of an inside floating ring and an outside floating ring that surround the rotor. In this oil film seal, the outside floating is configured to be movable along the axial direction of the rotor and has an inclined surface that tapers outward in the inner circumference thereof. In the outer circumference of the rotor, an inclined surface is formed so as to be opposed to the inclined surface in the inner circumference of the outside floating and be parallel to the inclined surface. Furthermore, a hydraulic groove is formed for introducing part of the seal oil between an outside end surface of the outside floating ring and an outside seal housing that holds the outside floating ring, and a spring is arranged for pressing the outside floating ring outward. The seal oil is supplied from an oil path formed between the inside floating ring and the outside floating ring.

Prior Art Document

40 Patent Document

[0006] Patent Document 1: JP-H09-13906-A

Summary of the Invention

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Problem to be Solved by the Invention

[0007] In the screw compressor, a plurality of working chambers defined by a screw rotor having helical lobes and a casing housing the screw rotor contract while moving in the axial direction in association with rotation of the screw rotor, thereby compressing compressed gas in the working chambers. Such a principle of compression causes the pressure of the compressed gas in the vicinity of the discharge-side end face of the screw rotor to differ depending on the positions of the working chambers in the rotation direction (circumferential direction). That is, pressure distribution in the circumferential direction exists in the compressed gas in the vicinity of the discharge-side end face. Due to this pressure distribution of the compressed gas in the circumferential direction, the compressed gas to leak into an annular shaft gap has pressure distribution in the circumferential direction.

55 Thus, in a region at a relatively high pressure (for example, working chambers in the discharge process or at a late stage of the compression process), the compressed gas attempts to leak toward a bearing chamber. Meanwhile, in a region at a relatively low pressure (for example, working chambers in the suction process or at an initial stage of the compression

process), the leakage flow rate of the compressed gas is lower than that in the region at higher pressure or, depending on the case, conversely the gas in the bearing chamber attempts to flow into the working chambers.

[0008] Among oil-flooded type screw compressors, there is a compressor which circulates oil to be supplied into working chambers and bearing chambers by using the pressure difference between the discharge pressure yielded by the screw compressor itself and the pressure of the supply destination such as the working chamber or the bearing chamber in a casing (hereinafter, often referred to as self-differential pressure) without using a pressurizing apparatus such as a pump. In the oil-flooded type screw compressor with such a configuration, the oil feed pressure at the supply destination is always lower than the discharge pressure of this compressor due to pressure loss caused in the route from a separator to the supply destination.

[0009] In the case of the oil-flooded type screw compressor in which oil is supplied by use of the discharge pressure of oneself to seal a shaft, the oil supplied to the shaft gap is affected by part of the compressed gas flowing into this shaft gap, which part is the compressed gas from the working chambers at relatively high pressures in the discharge process or the compression process. Therefore, it is not easy to form a proper oil film across the whole of the annular shaft gap in the circumferential direction. Thus, improvement in the shaft seal performance is desired in the oil-flooded type screw compressor in which the shaft is sealed by supplying oil with the discharge pressure of oneself.

[0010] Furthermore, a consideration will be made about applying the technique described in patent document 1 (oil film seal) to the oil-flooded type screw compressor in which the shaft is sealed by supplying oil with use of the self-differential pressure as a driving force. In the case of supplying the seal oil from the oil path formed between the inside floating ring, opposed to a rotor outer circumferential surface with a circular cylindrical surface shape, and the outside floating ring, opposed to a rotor outer circumferential surface inclining in such a manner as to taper outward, by using the discharge pressure of the screw compressor oneself, it is considered difficult to form a proper oil film across the whole of the annular shaft gap in the circumferential direction, similarly to the above-described case. This is because the pressure of the compressed gas in the working chambers in the discharge process or the like becomes higher relative to the pressure of the seal oil. That is, in the case of shaft sealing by supplying the oil with use of the discharge pressure of the screw compressor oneself, it is difficult to obtain favorable shaft seal performance even with use of the structure of the oil film seal described in patent document 1, which is based on the premise that the seal oil at a pressure higher than the pressure in the case (casing) is supplied.

[0011] The present invention is made in order to solve the above-described problem and an object thereof is to provide a screw compressor that can improve the shaft seal performance by a liquid film in the case of supplying a liquid to a shaft gap by using the discharge pressure of the screw compressor oneself.

Means for Solving the Problem

[0012] The present application includes a plurality of means for solving the above-described problem. One example thereof is a screw compressor including: a screw rotor including a rotor lobe part having a helical lobe and a shaft part provided at a discharge-side end of the rotor lobe part in the axial direction; and a casing having a housing chamber that houses the rotor lobe part, a shaft hole through which the shaft part passes, and a liquid feed path that introduces to the shaft hole a liquid supplied from an external, the casing being configured to define working chambers together with the rotor lobe part. Further, a predetermined portion of the shaft part disposed in the shaft hole and the shaft hole of the casing have a first opposed surface and a second opposed surface opposed to each other with a gap. The shaft part has an annular groove on the first opposed surface. The liquid feed path is configured so as to open on the second opposed surface of the shaft hole of the casing and at a position opposed to the annular groove. Moreover, the predetermined portion of the shaft part and the shaft hole of the casing each have, in an area remoter from the rotor lobe part than a position of the annular groove, a structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part.

Advantages of the Invention

[0013] According to the present invention, when a gas involving pressure distribution in the circumferential direction flows into the annular groove through the gap between the shaft hole and the shaft part (shaft gap), the annular groove plays a role as an expansion chamber, thereby making the pressure distribution of the gas even. Thus, it becomes possible to form a liquid film across the whole of the annular shaft gap in the circumferential direction even in the case of supplying the liquid to the shaft gap by using the discharge pressure of the screw compressor oneself. Besides, the structure is provided whose diameter becomes smaller in the area remoter from the rotor lobe part than the annular groove, thereby generating flow resistance in the flow of the liquid film, due to action of a centrifugal force generated in association with rotation of the screw rotor, on the downstream side of the flow of the liquid film relative to the annular groove. This flow resistance suppresses the flow rate of the liquid film, and thus it becomes possible to keep the state in which the annular groove is filled with the liquid. Therefore, it becomes possible to improve the shaft seal performance

by the liquid film in the case of supplying the liquid to the shaft gap by using the discharge pressure of the screw compressor oneself.

[0014] Problems, configurations, and effects other than the above-described ones will be made apparent by the following description of embodiments.

Brief Description of the Drawings

[0015]

FIG. 1 is a cross sectional view illustrating a screw compressor according to a first embodiment of the present invention with partial omission.

FIG. 2 is a cross sectional view when the screw compressor according to the first embodiment of the present invention illustrated in FIG. 1 is viewed as indicated by the arrows II-II.

FIG. 3 is a cross sectional view when the screw compressor according to the first embodiment of the present invention illustrated in FIG. 2 is viewed as indicated by the arrows III-III.

FIG. 4 is a system diagram illustrating an external route of oil feed to the screw compressor according to the first embodiment of the present invention.

FIG. 5 is a cross sectional view illustrating the structure of a discharge-side portion in a screw compressor of a comparative example in contrast to the screw compressor according to the first embodiment of the present invention in an enlarged state.

FIG. 6 is an explanatory diagram illustrating operation and effect of shaft sealing on the discharge side in the screw compressor according to the first embodiment of the present invention.

FIG. 7 is an explanatory diagram about the amount of leakage of a compressed gas through a discharge-side shaft gap in the screw compressor of the comparative example illustrated in FIG. 5.

FIG. 8 is an explanatory diagram about the amount of leakage of the compressed gas through the discharge-side shaft gap in the screw compressor according to the first embodiment of the present invention.

FIG. 9 is a cross sectional view illustrating the structure of a discharge-side portion in a screw compressor according to a modification example of the first embodiment of the present invention in an enlarged state.

FIG. 10 is a cross sectional view illustrating a screw compressor according to a second embodiment of the present invention with partial omission.

FIG. 11 is an explanatory diagram illustrating operation and effect of shaft sealing on the discharge side in the screw compressor according to the second embodiment of the present invention.

FIG. 12 is a cross sectional view illustrating the structure of a discharge-side portion in a screw compressor according to a modification example of the second embodiment of the present invention in an enlarged state.

Modes for Carrying Out the Invention

[0016] Embodiments of the present invention will be exemplified and described below with use of the drawings. The present embodiments are examples applied to a screw compressor that is an oil-flooded type in which oil is supplied to the inside of the compressor and that is a twin-rotor type.

[First Embodiment]

[0017] The configuration of a screw compressor according to a first embodiment and the configuration of an external route of oil feed to this screw compressor will be described with use of FIG. 1 to FIG. 4. FIG. 1 is a cross sectional view illustrating the screw compressor according to the first embodiment of the present invention with partial omission. FIG. 2 is a cross sectional view when the screw compressor according to the first embodiment of the present invention illustrated in FIG. 1 is viewed as indicated by the arrows II-II. FIG. 3 is a cross sectional view when the screw compressor according to the first embodiment of the present invention illustrated in FIG. 2 is viewed as indicated by the arrows III-III. However, the outer shape part of a casing is omitted in FIG. 3. FIG. 4 is a system diagram illustrating the external route of oil feed to the screw compressor according to the first embodiment of the present invention. In FIG. 1 and FIG. 2, the right side is the suction side of the screw compressor and the left side is the discharge side. In FIG. 3, thick arrows indicate the rotation direction of screw rotors.

[0018] In FIG. 1, a screw compressor 1 is a twin-rotor type and includes a male rotor 2 (screw rotor of a male type) and a female rotor 3 (screw rotor of a female type) that mesh with each other and rotate and a casing 4 that houses both the male and female rotors 2 and 3. The male rotor 2 is rotatably supported by a suction-side bearing 6 and discharge-side bearings 7 and 8. The female rotor 3 is rotatably supported by a suction-side bearing 10 and discharge-side bearings 11 and 12.

[0019] As illustrated in FIG. 1 to FIG. 3, the male rotor 2 includes a rotor lobe part 21 having helical male lobes 21a and a suction-side shaft part 22 and a discharge-side shaft part 23 respectively disposed at one end (suction-side end) and the other end (discharge-side end) of the rotor lobe part 21 in the axial direction (in FIG. 1 and FIG. 2, left-right direction). The rotor lobe part 21 has a suction-side end face 21b and a discharge-side end face 21c at the one end and the other end, respectively, in the axial direction. For example, the suction-side shaft part 22 is coupled to a rotational drive source 70 (see FIG. 4) such as an electric motor. Details of the structure of the discharge-side shaft part 23 will be described later.

[0020] As illustrated in FIG. 1 and FIG. 3, the female rotor 3 includes a rotor lobe part 31 having helical female lobes 31a and a suction-side shaft part 32 and a discharge-side shaft part 33 respectively disposed at the one end (suction-side end) and the other end (discharge-side end) of the rotor lobe part 31 in the axial direction (in FIG. 1, left-right direction). The rotor lobe part 31 has a suction-side end face 31b and a discharge-side end face 31c at the one end and the other end, respectively, in the axial direction. Details of the structure of the discharge-side shaft part 33 will be described later.

[0021] As illustrated in FIG. 1 and FIG. 2, the casing 4 includes a main casing 41 and a discharge-side casing 42 attached to the discharge side (in FIG. 1 and FIG. 2, left side) of the main casing 41 in the axial direction. A housing chamber 45 that houses the rotor lobe part 21 of the male rotor 2 and the rotor lobe part 31 of the female rotor 3 in the state in which they mesh with each other is formed inside the casing 4 as illustrated in FIG. 1 and FIG. 3. The housing chamber 45 is formed by closing opening of two circular cylindrical spaces that are formed in the main casing 41 in a partly overlapping manner, by the discharge-side casing 42. Inner wall surfaces that define the housing chamber 45 of the casing 4 have a male-side inner circumferential surface 45a that covers the radially outside of the rotor lobe part 21 of the male rotor 2 and has a substantially circular cylindrical surface shape, a female-side inner circumferential surface 45b that covers the radially outside of the rotor lobe part 31 of the female rotor 3 and has a substantially circular cylindrical surface shape, a suction-side inner wall surface 45c opposed to the suction-side end faces 21b and 31b of the rotor lobe parts 21 and 31 of both the male and female rotors 2 and 3, and a discharge-side inner wall surface 45d opposed to the discharge-side end faces 21c and 31c of the rotor lobe parts 21 and 31 of both the male and female rotors 2 and 3. The discharge-side end faces 21c and 31c of both the male and female rotors 2 and 3 are opposed to the discharge-side inner wall surface 45d of the casing 4 across a slight gap (hereinafter, often referred to as discharge-side end face gap). A plurality of working chambers C with different pressures are formed by the rotor lobe parts 21 and 31 of both the male and female rotors 2 and 3 and the inner wall surfaces (male-side inner circumferential surface 45a, female-side inner circumferential surface 45b, suction-side inner wall surface 45c, and discharge-side inner wall surface 45d) of the housing chamber 45 of the casing 4 that surrounds them.

[0022] As illustrated in FIG. 1 and FIG. 2, an end portion of the main casing 41 on the one side in the axial direction (in FIG. 1 and FIG. 2, right side) is provided with a suction-side bearing chamber 47 in which the suction-side bearing 6 for the male rotor 2 is disposed and a suction-side bearing chamber 48 in which the suction-side bearing 10 for the female rotor 3 is disposed. Both the suction-side bearing chambers 47 and 48 are separated from the housing chamber 45 by a suction-side partition wall 49. The suction-side partition wall 49 is provided with a suction-side shaft hole 49a through which the suction-side shaft part 22 of the male rotor 2 passes and a suction-side shaft hole 49b through which the suction-side shaft part 32 of the female rotor 3 passes. In the respective suction-side shaft holes 49a and 49b, the suction-side shaft parts 22 and 32 of the male rotor 2 and the female rotor 3 are each disposed with a slight gap.

[0023] The discharge-side casing 42 is provided with discharge-side bearing chamber 51 in which the discharge-side bearings 7 and 8 for the male rotor 2 are disposed and a discharge-side bearing chamber 52 in which the discharge-side bearings 11 and 12 for the female rotor 3 are disposed. In both the discharge-side bearing chambers 51 and 52, one side (in FIG. 1 and FIG. 2, right side) is separated from the housing chamber 45 by a discharge-side partition wall 53 whereas the other side (in FIG. 1 and FIG. 2, left side) is opened. The discharge-side partition wall 53 is provided with a discharge-side shaft hole 54 through which the discharge-side shaft part 23 of the male rotor 2 passes and a discharge-side shaft hole 55 through which the discharge-side shaft part 33 of the female rotor 3 passes. In the discharge-side shaft hole 54 and the discharge-side shaft hole 55, the discharge-side shaft part 23 of the male rotor 2 and the discharge-side shaft part 33 of the female rotor 3 are each disposed with a slight gap (hereinafter, often referred to as discharge-side shaft gap). Details of the structure of the discharge-side shaft hole 54 and the discharge-side shaft hole 55 will be described later. Moreover, a discharge-side cover 43 that closes the openings of both the discharge-side bearing chambers 51 and 52 is attached to the discharge-side casing 42.

[0024] As illustrated in FIG. 2, a suction flow path 57 for sucking a gas from the external of the casing 4 into the working chambers C is provided on the one side in the axial direction (in FIG. 2, right side) in the casing 4. A discharge flow path 58 (not illustrated in FIG. 2, see FIG. 4) for discharging a compressed gas from the working chambers C to the external of the casing 4 is provided on the other side in the axial direction in the casing 4. The discharge flow path 58 has a discharge port 58a (see FIG. 3) on the discharge-side inner wall surface 45d of the casing 4. The discharge flow path 58 is connected to an external oil feed system 100 (see FIG. 4) to be described later.

[0025] The screw compressor 1 is, for example, an oil-flooded type and has a configuration in which lubricating oil is

supplied to the working chambers C in the casing 4 for the purposes of lubricating of the male rotor 2 and the female rotor 3, cooling of the compressed gas, and sealing of gaps between the male and female rotors 2, 3 and the casing 4 and so forth. Moreover, the screw compressor 1 of the present embodiment has a configuration in which the lubricating oil is supplied also to the suction-side bearing 6 and the discharge-side bearings 7 and 8 for the male rotor 2 and the suction-side bearing 10 and the discharge-side bearings 11 and 12 for the female rotor 3 in addition to the working chambers C.

[0026] Specifically, an oil feed path 60 is provided in the casing 4 as illustrated in FIG. 2. The oil feed path 60 includes a first route 61 that introduces the lubricating oil supplied from the external of the casing 4 to the housing chamber 45 (working chambers C) in the casing 4, a second route 62 that introduces the lubricating oil to the discharge-side shaft hole 54 and the discharge-side shaft hole 55, and a third route (not illustrated) for supplying the lubricating oil to the suction-side bearing 6 and the suction-side bearing 10. The first route 61 is opened to a region in which the working chambers C are in the compression process in the housing chamber 45. The second route 62 is one that is for supplying the lubricating oil to the discharge-side bearings 7 and 8 and the discharge-side bearings 11 and 12 through the discharge-side shaft gap. Details of the structure of the second route 62 will be described later.

[0027] To the oil-flooded type screw compressor 1, the external oil feed system 100 for supplying the lubricating oil to the compressor 1 is connected as illustrated in FIG. 4. For example, the external oil feed system 100 is configured by an oil separator 101, an oil cooler 102, auxiliary equipment 103 such as an oil filter, a conduit line 104 that connects them, and so forth. The oil separator 101 is connected to the discharge flow path 58 of the casing 4 and separates the lubricating oil contained in the compressed gas discharged from the screw compressor 1. The external oil feed system 100 supplies the lubricating oil to the screw compressor 1 with use of the pressure of the compressed gas that flows into the oil separator 101 as a drive source without using a power source such as a pump. In the case of using the external oil feed system 100 with such a configuration, the supply pressure of the lubricating oil supplied to the screw compressor 1 inevitably becomes lower than the discharge pressure of the screw compressor 1. This is because, pressure loss occurs at the oil separator 101 in the process of separating the lubricating oil from the compressed gas discharged from the screw compressor 1, and so forth, and occurs due to the passing of the lubricating oil through the oil cooler 102 and the auxiliary equipment 103 such as an oil filter in the course of supplying the lubricating oil to the screw compressor 1 from the oil separator 101.

[0028] In the screw compressor 1 configured in the above-described manner, the male rotor 2 illustrated in FIG. 1 is driven by the rotational drive source 70 (see FIG. 4) to rotationally drive the female rotor 3, and the volume of the working chambers C increases and decreases while the working chambers C move in the axial direction in association with the rotation of both the male and female rotors 2 and 3. This causes a gas to be sucked into the working chambers C through the suction flow path 57 illustrated in FIG. 2 and be compressed until the pressure thereof reaches a predetermined pressure. Finally, the compressed gas is discharged to the oil separator 101 of the external oil feed system 100 through the discharge flow path 58 of the casing 4 illustrated in FIG. 4. In the oil separator 101, the compressed gas and the lubricating oil are separated. The compressed gas from which the lubricating oil has been removed is supplied to external equipment according to need and the separated lubricating oil is stored in the oil separator 101.

[0029] The lubricating oil stored in the oil separator 101 is cooled by the oil cooler 102 and impurities are removed therefrom by the oil filter 103 of auxiliary equipment. Thereafter, the lubricating oil is supplied to the screw compressor 1. The supply of the lubricating oil to the screw compressor 1 is executed by using the pressure of the compressed gas that flows into the oil separator 101 (discharge pressure of the screw compressor 1) without using a power source such as a pump.

[0030] The lubricating oil supplied to the screw compressor 1 is supplied to the working chambers C through the first route 61 of the oil feed path 60 of the casing 4 illustrated in FIG. 2. In addition, the lubricating oil is supplied to the discharge-side bearings 7 and 8 and the discharge-side bearings 11 and 12 through the second route 62 and the discharge-side shaft gaps (the gap between the outer circumferential surface of the discharge-side shaft part 23 of the male rotor 2 and the inner circumferential surface of the discharge-side shaft hole 54, and the gap between the outer circumferential surface of the discharge-side shaft part 33 of the female rotor 3 and the inner circumferential surface of the discharge-side shaft hole 55). The lubricating oil is supplied also to the suction-side bearing 6 and the suction-side bearing 10 similarly to the discharge-side bearings 7 and 8 and the discharge-side bearings 11 and 12. The lubricating oil supplied to the working chambers C cools the compressed gas, and seals the gaps between the lobe tips of both the male and female rotors 2 and 3 and the male-side inner circumferential surface 45a and the female-side inner circumferential surface 45b of the housing chamber 45 of the casing 4, the gaps between the discharge-side end faces 21c and 31c of both the male and female rotors 2 and 3 and the discharge-side inner wall surface 45d of the housing chamber 45 of the casing 4 (discharge-side end face gaps), and so forth. The lubricating oil supplied in the working chambers C is discharged from the discharge flow path 58 together with the compressed gas and flows into the oil separator 101.

[0031] By the way, the compressed gas in the plurality of working chambers C attempts to flow out from the side of the discharge-side end faces 21c and 31c of the rotor lobe parts 21 and 31 of both the male and female rotors 2 and 3 to the discharge-side bearing chambers 51 and 52 through the discharge-side end face gaps and the discharge-side

shaft gaps due to the pressure difference from the discharge-side bearing chambers 51 and 52. However, the pressure of the compressed gas in the vicinity of the discharge-side end faces 21c and 31c of the rotor lobe parts 21 and 31 of both the male and female rotors 2 and 3 differs depending on the position of the working chamber C in the rotation direction (circumferential direction). As described above, the screw compressor 1 sucks a gas into the working chambers C and compresses the gas through expansion and contraction of the plurality of working chambers C with movement of the working chambers C in the axial direction in association with rotation of both the male and female rotors 2 and 3. Such a principle of compression causes pressure distribution in the circumferential direction to exist in the compressed gas in the vicinity of the discharge-side end faces 21c and 31c of the rotor lobe parts 21 and 31. As specific one example, as illustrated in FIG. 3, for example, in regions in which the male rotor 2 and the female rotor 3 do not mesh with each other in the vicinity of the discharge-side end faces 21c and 31c, the regions (working chambers C) have higher pressure as the regions are located farther away from a start point, which is a region (working chamber C) on the opposite side (in FIG. 3, upper side) to the position of the discharge port 58a (a two-dot-dash line), toward the direction of an arrow P of a one-dot-dash line. As above, the pressure distribution in the circumferential direction exists in the compressed gas in the vicinity of the discharge-side end faces 21c and 31c of the rotor lobe parts 21 and 31. Thus, due to this pressure distribution in the circumferential direction in the gas, pressure distribution in the circumferential direction is generated also in the compressed gas to leak into the discharge-side shaft gap from the plurality of working chambers C.

[0032] In the screw compressor 1 according to the present embodiment, against the leakage of the compressed gas to the discharge-side bearing chambers 51 and 52 through the discharge-side shaft gaps from the plurality of working chambers C in which the pressure distribution is generated in the circumferential direction, the lubricating oil for lubricating the discharge-side bearings 7 and 8 and the discharge-side bearings 11 and 12 is supplied to the discharge-side shaft gaps to form oil films in the discharge-side shaft gaps, thereby sealing the discharge-side shaft gaps. Note that, the pressure of the lubricating oil supplied to the discharge-side shaft gaps is lower than the pressure of the compressed gas discharged from the screw compressor 1 because the screw compressor 1 is the configuration in which the lubricating oil is supplied to the discharge-side shaft gaps by using the discharge pressure of the screw compressor 1 oneself,.

[0033] Thus, in the screw compressor 1 according to the present embodiment, even in the case of supplying a liquid (lubricating oil) to the discharge-side shaft gap by using the discharge pressure of the screw compressor 1 oneself, improvement in the shaft seal performance on the discharge side is intended by including the following characteristic part. The characteristic part for shaft sealing on the discharge side in the screw compressor 1 according to the first embodiment will be described with use of FIG. 1 and FIG. 2.

[0034] In FIG. 1 and FIG. 2, the discharge-side shaft part 23 of the male rotor 2 has a first shaft section 24 to which the discharge-side bearings 7 and 8 are attached and a second shaft section 25 located between the first shaft section 24 and the rotor lobe part 21. The first shaft section 24 is formed into a circular column shape with a constant outer diameter. The second shaft section 25 is a portion disposed in the discharge-side shaft hole 54 of the casing 4 and has a tapered structure that tapers toward the discharge-side bearings 7 and 8 (first shaft section 24) over the entire length thereof. The discharge-side shaft hole 54 of the casing 4 (discharge-side casing 42) has a tapered structure that tapers toward the discharge-side bearing chamber 51 over the entire length thereof according to the structure of the second shaft section 25 of the discharge-side shaft part 23.

[0035] The outer circumferential surface of the second shaft section 25 and an inner circumferential surface defining the discharge-side shaft hole 54 of the casing 4 have a first opposed surface 25s and a second opposed surface 54s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers). The whole of the first opposed surface 25s of the second shaft section 25 is formed as a first tapered surface that tapers toward the direction of the discharge-side bearings 7 and 8 (such a direction as to get further away from the rotor lobe part 21) (inclined surface that gradually converges as the position gets closer to the discharge-side bearings 7 and 8). The second opposed surface 54s of the discharge-side shaft hole 54 of the casing 4 is formed as, over the entire length thereof, a second tapered surface that tapers toward the direction of the discharge-side bearing chamber 51 (such a direction as to get further away from the housing chamber 45) (inclined surface that gradually converges as the position gets closer to the discharge-side bearing chamber 51) according to the shape of the first tapered surface (first opposed surface 25s) of the second shaft section 25. Such a structure makes the gap (discharge-side shaft gap) formed by the first tapered surface (first opposed surface 25s) of the second shaft section 25 of the male rotor 2 and the second tapered surface (second opposed surface 54s) of the discharge-side shaft hole 54 of the casing 4 have a tapered surface-like shape (conical surface-like shape) that tapers in such a direction as to get further away from the housing chamber 45.

[0036] An annular groove 26 is provided on the first tapered surface (first opposed surface 25s) of the second shaft section 25. The annular groove 26 functions as an expansion chamber whose volume is relatively larger with respect to the size of the annular discharge-side shaft gap (several tens to several hundreds of micrometers). The formation position of the annular groove 26 is optional as long as it is a position with which the tapered first opposed surface 25s remains on the side of the discharge-side bearings 7 and 8 (first shaft section 24) relative to the annular groove 26. For example, as illustrated in FIG. 1 and FIG. 2, the annular groove 26 is provided on the first opposed surface 25s at a

position closer to the rotor lobe part 21 relative to the first shaft section 24, that is, a position at which the outer diameter of the tapered second shaft section 25 is relatively large. It is preferable that the formation position of the annular groove 26 be a position with which the length of the tapered first opposed surface 25s existing on the side of the discharge-side bearings 7 and 8 (first shaft section 24) relative to the annular groove 26 is longer, that is, a position closer to the rotor lobe part 21, in terms of sealing the leakage of the compressed gas in the working chambers C to the discharge-side bearing chamber 51 through the discharge-side shaft gap. Furthermore, the configuration is made in such a manner that the lubricating oil (liquid for shaft sealing) is supplied to the annular groove 26. That is, the second route 62 of the oil feed path 60 of the casing 4 is configured to open at a position opposed to the annular groove 26 of the second shaft section 25 on the second tapered surface (second opposed surface 54s) of the discharge-side shaft hole 54.

[0037] Moreover, the structure of the discharge-side shaft part 33 of the female rotor 3 and the discharge-side shaft hole 55 of the casing 4 is similar to that of the discharge-side shaft part 23 of the male rotor 2 and the discharge-side shaft hole 54 of the casing 4 as illustrated in FIG. 1. That is, the discharge-side shaft part 33 of the female rotor 3 has a first shaft section 34 to which the discharge-side bearings 11 and 12 are attached and a second shaft section 35 located between the first shaft section 34 and the rotor lobe part 31. The first shaft section 34 is formed into a circular column shape having a constant outer diameter. The second shaft section 35 is a portion disposed in the discharge-side shaft hole 55 of the casing 4 and is formed into a tapered shape (truncated cone shape) that tapers toward the discharge-side bearings 11 and 12 (first shaft section 34) over the entire length thereof. The discharge-side shaft hole 55 of the casing 4 (discharge-side casing 42) is formed into a tapered shape (truncated cone shape) that tapers toward the discharge-side bearing chamber 52 over the entire length thereof according to the shape of the second shaft section 35 of the discharge-side shaft part 33.

[0038] The outer circumferential surface of the second shaft section 35 and an inner circumferential surface defining the discharge-side shaft hole 55 of the casing 4 have a first opposed surface 35s and a second opposed surface 55s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers). The whole of the first opposed surface 35s of the second shaft section 35 is formed as a first tapered surface that tapers toward the direction of the discharge-side bearings 11 and 12 (such a direction as to get further away from the rotor lobe part 31) (inclined surface that gradually converges as the position gets closer to the discharge-side bearings 11 and 12). The second opposed surface 55s of the discharge-side shaft hole 55 of the casing 4 is formed as, over the entire length thereof, a second tapered surface that tapers toward the direction of the discharge-side bearing chamber 52 (such a direction as to get further away from the housing chamber 45) (inclined surface that gradually converges as the position gets closer to the discharge-side bearing chamber 52) according to the shape of the first tapered surface (first opposed surface 35s) of the second shaft section 35. Such a structure makes the gap (discharge-side shaft gap) formed by the first tapered surface (first opposed surface 35s) of the second shaft section 35 of the female rotor 3 and the second tapered surface (second opposed surface 55s) of the discharge-side shaft hole 55 of the casing 4 have a tapered surface-like shape (conical surface-like shape) that tapers in such a direction as to get further away from the housing chamber 45.

[0039] An annular groove 36 is provided on the first tapered surface (first opposed surface 35s) of the second shaft section 35. The annular groove 36 functions as an expansion chamber whose volume is relatively larger with respect to the size of the annular discharge-side shaft gap (several tens to several hundreds of micrometers). The formation position of the annular groove 36 is optional as long as it is a position with which the tapered first opposed surface 35s remains on the side of the discharge-side bearings 11 and 12 (first shaft section 34) relative to the annular groove 36. For example, as illustrated in FIG. 1, the annular groove 36 is provided at a position closer to the rotor lobe part 31 relative to the first shaft section 34 on the first opposed surface 35s, that is, a position at which the outer diameter of the tapered second shaft section 35 is relatively larger. It is preferable that the formation position of the annular groove 36 be a position with which the length of the tapered first opposed surface 35s existing on the side of the discharge-side bearings 11 and 12 (first shaft section 34) relative to the annular groove 36 is longer, that is, a position closer to the rotor lobe part 31, in terms of sealing the leakage of the compressed gas in the working chambers C to the discharge-side bearing chamber 52 through the discharge-side shaft gap. Furthermore, the configuration is made in such a manner that the lubricating oil (liquid for shaft sealing) is supplied to the annular groove 36. That is, the second route 62 of the oil feed path 60 is configured to open at a position opposed to the annular groove 36 of the second shaft section 35 on the second tapered surface (second opposed surface 55s) of the discharge-side shaft hole 55.

[0040] Next, operation and effect of shaft sealing in the screw compressor according to the first embodiment will be described with comparison with a screw compressor of a comparative example. Here, description will be made only about shaft sealing for the discharge-side shaft gap on the male rotor side. However, the same applies also to shaft sealing for the discharge-side shaft gap on the female rotor side.

[0041] First, the structure of the screw compressor of the comparative example and a problem of the shaft sealing on the discharge side attributed to the structure will be described with use of FIG. 5. FIG. 5 is a cross sectional view illustrating the structure of a discharge-side portion in the screw compressor of the comparative example in contrast to the screw compressor according to the first embodiment of the present invention in an enlarged state.

[0042] A main different point of a screw compressor 201 of the comparative example illustrated in FIG. 5 from the screw compressor 1 according to the present embodiment is that the shape of a discharge-side shaft part 223 of a male rotor 202 and the shape of a discharge-side shaft hole 254 of a casing 204 are different. Specifically, the male rotor 202 of the comparative example includes the discharge-side shaft part 223 having the circular columnar first shaft section 24 to which the discharge-side bearings 7 and 8 are attached and a circular columnar second shaft section 225 located between the first shaft section 24 and the rotor lobe part 21. The second shaft section 225 is a configuration with no annular groove. The discharge-side shaft hole 254 of the casing 204 (discharge-side casing 242) of the comparative example is formed into a circular column shape according to the shape of the second shaft section 225 of the discharge-side shaft part 223.

[0043] The outer circumferential surface of the second shaft section 225 of the male rotor 202 and an inner circumferential surface defining the discharge-side shaft hole 254 of the casing 204 have a first opposed surface 225s and a second opposed surface 254s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers). The whole of the first opposed surface 225s of the second shaft section 225 is formed as a circular cylindrical surface. The second opposed surface 254s of the discharge-side shaft hole 254 of the casing 204 is formed as, over the entire length thereof, a circular cylindrical surface according to the shape of the first opposed surface 225s, which is the circular cylindrical surface, of the second shaft section 225. Such a structure makes the gap (discharge-side shaft gap) formed by the first opposed surface 225s, which is the circular cylindrical surface, of the second shaft section 225 of the male rotor 202, and the second opposed surface 254s, which is the circular cylindrical surface, of the discharge-side shaft hole 254 of the casing 204 have a circular cylindrical surface shape.

[0044] As above, the discharge-side shaft part 223 and the discharge-side shaft hole 254 of the casing 204 in the screw compressor 201 of the comparative example with the above-described structure are different from the discharge-side shaft part 23 and the discharge-side shaft hole 54 of the casing 4 in the screw compressor 1 according to the present embodiment in that the first opposed surface 225s of the second shaft section 225 and the second opposed surface 254s of the discharge-side shaft hole 254 are the circular cylindrical surfaces and, in association with this, the discharge-side shaft gap has the circular cylindrical surface shape, and in that the no annular groove is provided on the second shaft section 225.

[0045] In the screw compressor 201 of the comparative example with the above-described structure, lubricating oil is supplied from the second route 62 of the oil feed path 60 of the casing 204 to the annular gap (discharge-side shaft gap) between the first opposed surface 225s of the second shaft section 225 of the discharge-side shaft part 223 and the second opposed surface 254s of the discharge-side shaft hole 254 of the casing 204. A compressed gas flows into the annular discharge-side shaft gap from a plurality of working chambers C (see FIG. 3) that line up in the circumferential direction at the discharge-side end face 21c of the rotor lobe part 21 of the male rotor 202. The plurality of working chambers C lining up in the circumferential direction have pressure distribution. Therefore, as described above, pressure distribution in the circumferential direction is generated also in the compressed gas flowing into the discharge-side shaft gap. When the pressure of the lubricating oil of the oil feed path 60 is lower than the discharge pressure of the screw compressor 201 of the comparative example, the lubricating oil supplied to the discharge-side shaft gap is affected by part of the compressed gas flowing into the discharge-side shaft gap, which part is the compressed gas from the working chambers C at relatively high pressures in the discharge process and at a late stage in the compression process. Due to this, it is difficult to form an oil film (oil seal) over the entire annular discharge-side shaft gap. Thus, the compressed gas at a relatively high pressure tends to leak to the discharge-side bearing chamber 51 through the discharge-side shaft gap and the effect of suppression of leakage Lg of the compressed gas by the oil film (oil seal) of the discharge-side shaft gap becomes restrictive.

[0046] Next, the operation and effect of the shaft sealing on the discharge side in the screw compressor according to the first embodiment will be described with use of FIG. 6. FIG. 6 is an explanatory diagram illustrating the operation and effect of the shaft sealing on the discharge side in the screw compressor according to the first embodiment of the present invention.

[0047] In the screw compressor 1 of the present embodiment, as illustrated in FIG. 6, the first opposed surface 25s of the second shaft section 25 in the discharge-side shaft part 23 of the male rotor 2 is formed as the first tapered surface that tapers toward the direction of the discharge-side bearings 7 and 8. In addition, the second opposed surface 54s of the discharge-side shaft hole 54 of the casing 4 is formed as the second tapered surface that tapers toward the direction of the discharge-side bearing chamber 51 according to the shape of the first tapered surface (first opposed surface 25s) of the second shaft section 25. This makes a discharge-side shaft gap Sc (see also enlarged view) have a tapered surface-like shape (conical surface-like shape) that tapers in the direction of the discharge-side bearing chamber 51. Besides, the annular groove 26 is provided at a midway position in the axial direction in the first opposed surface 25s of the second shaft section 25. In addition, the second route 62 of the oil feed path 60 opens on the second opposed surface 54s of the discharge-side shaft hole 54 and at a position opposed to the annular groove 26.

[0048] In such a structure, the lubricating oil for the discharge-side bearings 7 and 8 is supplied from the second route

62 of the oil feed path 60 to the discharge-side shaft gap Sc. The lubricating oil of the second route 62 of the oil feed path 60 first flows into the annular groove 26. As described also in the explanation for the screw compressor 201 of the comparative example, the plurality of working chambers C (see FIG. 3) that line up in the circumferential direction at the discharge-side end face 21c of the rotor lobe part 21 of the male rotor 2 have pressure distribution, and thus pressure distribution in the circumferential direction is generated also in the compressed gas that flows into the discharge-side shaft gap Sc. When the compressed gas involving the pressure distribution in the circumferential direction flows into the annular groove 26, the annular groove 26 having a larger volume than the annular discharge-side shaft gap Sc plays a role as an expansion chamber. That is, the compressed gas that has flown into the annular groove 26 from the working chambers C at relatively high pressures in the discharge process and at a late stage of the compression process and the compressed gas that has flown into the annular groove 26 from the working chambers C at relatively low pressures are mixed, so that the pressure distribution in the circumferential direction is made even. Therefore, the compressed gas that has flown into the annular groove 26 becomes the state in which the pressure thereof is lower than the pressure of the working chambers C in the compression process and at a late stage of the discharge process. Thus, it becomes possible to form an oil film (oil seal) across the whole of the annular discharge-side shaft gap Sc in the circumferential direction even when the pressure of the lubricating oil of the oil feed path 60 is lower than the discharge pressure of the screw compressor 1.

[0049] The lubricating oil that has flown into the annular groove 26 flows into the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc and lubricates the discharge-side bearings 7 and 8. At this time, it is desirable that the annular groove 26 is filled with the lubricating oil across the whole thereof in the circumferential direction. When the whole of the annular groove 26 in the circumferential direction is not filled with the lubricating oil, there is a concern that the compressed gas that has flown into the discharge-side shaft gap Sc gets caught up into the annular groove 26 and leaks to the discharge-side bearing chamber 51. To fill the annular groove 26 with the lubricating oil across the whole thereof in the circumferential direction, the flow rate of the lubricating oil that flows into the annular groove 26 from the second route 62 of the oil feed path 60 needs to become higher than that of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51.

[0050] The present embodiment has a configuration to generate flow resistance in a region of the discharge-side shaft gap Sc on the side of the discharge-side bearings 7 and 8 relative to the annular groove 26 (flow path on the downstream side), thereby limiting the flow rate of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51. Specifically, as illustrated in the enlarged view in FIG. 6, lubricating oil O that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc flows toward the circumferential direction in association with rotation of the second shaft section 25 of the discharge-side shaft part 23 due to the viscosity thereof. At this time, a centrifugal force F_c toward the radially outside acts on the lubricating oil. The centrifugal force F_c that acts on the lubricating oil O can be resolved into a first component force F_{c1} in the direction along the second opposed surface 54s of the discharge-side shaft hole 54 of the casing 4 and a second component force F_{c2} in the direction orthogonal to it. Because the second opposed surface 54s of the casing 4 is the second tapered surface with an inner diameter gradually larger toward the side of the housing chamber 45, the first component force F_{c1} is a force toward the side of the housing chamber 45. Therefore, the force toward the upstream side in the flow direction of the lubricating oil acts on the lubricating oil that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc with the tapered surface-like shape that tapers in the direction of the discharge-side bearing chamber 51. That is, flow resistance is generated in the lubricating oil that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc. This flow resistance suppresses the flow rate of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51. As a result, the state in which the annular groove 26 is filled with the lubricating oil across the whole thereof in the circumferential direction can be kept. The magnitude of the flow resistance is defined by the magnitude of a radius R_t on the tip side (side of the discharge-side bearing chamber 51) of the discharge-side shaft gap Sc with the tapered surface-like shape and a radius R_g on the side of the annular groove 26.

[0051] As above, the lubricating oil supplied from the second route 62 of the oil feed path 60 to the annular groove 26 flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc with the tapered surface-like shape while the state in which the whole of the annular groove 26 in the circumferential direction is filled with the lubricating oil is kept. That is, an oil film O is formed in the discharge-side shaft gap Sc on the side of the discharge-side bearing chamber 51 relative to the annular groove 26. Therefore, even when the pressure of the lubricating oil O supplied to the discharge-side shaft gap Sc is lower than the discharge pressure of the screw compressor 1, the amount of leakage of the compressed gas from the plurality of working chambers C to the discharge-side shaft gap can be suppressed by the lubricating oil with which the annular groove 26 is filled across the whole thereof in the circumferential direction and the oil film O formed on the side of the discharge-side bearing chamber 51 relative to the annular groove 26, and thus the shaft seal performance on the discharge side improves. In terms of the effect of the shaft seal performance on the discharge side, it is advantageous for improving the shaft seal performance that the route of the oil film O that flows from the annular groove 26 toward the side of the discharge-side bearing chamber 51

(first shaft section 24) is long. Therefore, when improvement in the shaft seal performance is considered, it is preferable to form the annular groove 26 and the opening of the second route 62 of the oil feed path 60 at positions closer to the rotor lobe part 21 (housing chamber 45) relative to the first shaft section 24 (discharge-side bearing chamber 51).

[0052] Furthermore, the structure of the characteristic portion of the screw compressor 1 according to the first embodiment makes the amount of leakage of the compressed gas to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc smaller than in the case of the structure of the screw compressor 201 of the comparative example. The reason for this will be described with use of FIG. 7 and FIG. 8. FIG. 7 is an explanatory diagram about the amount of leakage of the compressed gas through the discharge-side shaft gap in the screw compressor of the comparative example illustrated in FIG. 5. FIG. 8 is an explanatory diagram about the amount of leakage of the compressed gas through the discharge-side shaft gap in the screw compressor according to the first embodiment of the present invention.

[0053] The case in which the lubricating oil is not supplied to the discharge-side shaft gap will be assumed. At this time, the amounts of leakage of the compressed gas in the working chambers C to the discharge-side bearing chamber 51 through the discharge-side shaft gap are compared between the screw compressor 1 according to the present embodiment and the screw compressor 201 of the comparative example. Note that the influence of rotation of the male rotor 2 will be ignored. That is, in a situation in which the male rotor 2 remains still, leakage due to a pressure difference ΔP between a pressure P_1 of the working chamber C in the discharge process and a pressure P_2 of the discharge-side bearing chamber 51 will be considered. As illustrated in FIG. 7 and FIG. 8, a size h of the discharge-side shaft gap, that is, the diameter difference h between the inner diameter of the discharge-side shaft hole 54 or 254 of the casing 4 or 204 and the outer diameter of the second shaft section 25 or 225 of the discharge-side shaft part 23 or 223 of the male rotor 2 or 202, is set to the same value between the screw compressor 1 of the present embodiment and the screw compressor 201 of the comparative example.

[0054] At this time, an amount Q_e of leakage of the compressed gas through a discharge-side shaft gap See (double circular cylindrical portion) in the screw compressor 201 of the comparative example is obtained by the following Formula (1). In Formula (1), R_1 denotes the hole diameter (inner diameter) of the discharge-side shaft hole 254 of the casing 204 illustrated in FIG. 5 and FIG. 7. L_e denotes the length of the discharge-side shaft gap See along the inner circumferential surface of the discharge-side shaft hole 254 illustrated in FIG. 5 and FIG. 7. μ denotes the viscosity of the gas. [Math. 1]

$$Q_e = \frac{\pi \Delta P}{6 \mu L_e} h^3 R_1 \quad \dots \text{Formula (1)}$$

[0055] Meanwhile, an amount Q_i of leakage of the compressed gas through the discharge-side shaft gap Sc (double circular cylindrical portion) in the screw compressor 1 according to the present embodiment is obtained by the following Formula (2). In Formula (2), R_1 denotes the hole diameter (inner diameter) on the side of the housing chamber 45 (larger-diameter side) of the tapered discharge-side shaft hole 54 of the casing 4 illustrated in FIG. 6 and FIG. 8. R_2 denotes the hole diameter (inner diameter) on the side of the discharge-side bearing chamber 51 (smaller-diameter side) of the tapered discharge-side shaft hole 54 of the casing 4 illustrated in FIG. 5 and FIG. 7. L_i denotes the length of the discharge-side shaft gap along the inner circumferential surface of the discharge-side shaft hole 54 illustrated in FIG. 6 and FIG. 8. μ denotes the viscosity of the gas. [Math. 2]

$$Q_i = \frac{\pi \Delta P}{6 \mu L_i} h^3 \frac{R_1 - R_2}{\log(R_1/R_2)} \quad \dots \text{Formula (2)}$$

[0056] Here, the following magnitude relations hold.

[Math. 3]

$$\frac{R_1 - R_2}{\log(R_1/R_2)} < R_1, \quad L_e < L_i$$

[0057] Therefore, when the amount Q_e of leakage obtained from Formula (1) and the amount Q_i of leakage obtained from Formula (2) are compared with each other, the amount Q_e of leakage in the comparative example is larger than the amount Q_i of leakage in the present embodiment ($Q_e > Q_i$). The relation of $Q_e > Q_i$ is kept even when the male rotors 2 and 202 are rotating.

[0058] As above, in the present embodiment, the discharge-side shaft gap Sc formed between the first opposed surface 25s (first tapered surface) of the second shaft section 25 in the discharge-side shaft part 23 of the male rotor 2 and the second opposed surface 54s (second tapered surface) of the discharge-side shaft hole 54 of the casing 4 is made into the tapered surface-like shape (conical surface-like shape). This can reduce the amount of leakage of the compressed gas relative to the discharge-side shaft gap See with the circular cylindrical surface-like shape in the screw compressor 201 of the comparative example. Thus, the amount of supply of the lubricating oil to the discharge-side shaft gap Sc with the tapered surface-like shape (conical surface-like shape) can be reduced, and it becomes possible to reduce mechanical loss correspondingly.

[0059] The screw compressor 1 of the above-described present embodiment includes the male rotor 2 and the female rotor 3 (screw rotors) including the rotor lobe parts 21 and 31 having the helical lobes 21a and 31a and the discharge-side shaft parts 23 and 33 (shaft parts) provided at the discharge-side end of the rotor lobe part 21 or 31 in the axial direction. The screw compressor 1 includes also the casing 4 having the housing chamber 45 that houses the rotor lobe parts 21 and 31, the discharge-side shaft holes 54 and 55 (shaft holes) through which the discharge-side shaft part 23 or 33 (shaft part) pass, and the second route 62 of the oil feed path 60 (liquid feed path) that introduces to the discharge-side shaft holes 54 and 55 (shaft holes) the lubricating oil (liquid) supplied from the external, and the casing 4 is configured to define the working chambers C together with the rotor lobe parts 21 and 31. The second shaft sections 25 and 35 (predetermined portions) of the discharge-side shaft part 23 or 33 (shaft part) disposed in the discharge-side shaft hole 54 or 55 (shaft hole) and the discharge-side shaft holes 54 and 55 (shaft holes) of the casing 4 have the first opposed surface 25s or 35s and the second opposed surface 54s or 55s opposed to each other with the gap Sc . The discharge-side shaft parts 23 and 33 (shaft parts) have the annular groove 26 or 36 on the first opposed surface 25s or 35s, and the second route 62 (liquid feed path) is configured so as to open on the second opposed surface 54s or 55s of the discharge-side shaft hole 54 or 55 (shaft hole) of the casing 4 and at positions opposed to the annular groove 26 or 36. The second shaft sections 25 and 35 (predetermined portions) of the discharge-side shaft parts 23 and 33 (shaft parts) and the discharge-side shaft holes 54 and 55 (shaft holes) of the casing 4 each have, in an area remoter from the rotor lobe part 21 or 31 than the position of the annular groove 26 or 36, the tapered structure as a structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part 21 or 31.

[0060] According to this configuration, when a gas involving pressure distribution in the circumferential direction flows into the annular groove 26 or 36 through the shaft gap Sc (gap) between the discharge-side shaft hole 54 or 55 (shaft hole) and the second shaft section 25 or 35 (predetermined portion) of the discharge-side shaft part 23 or 33 (shaft part), the annular groove 26 or 36 plays a role as an expansion chamber, thereby making the pressure distribution of the gas even. Thus, it becomes possible to form a liquid film across the whole of the annular shaft gap Sc in the circumferential direction even in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1 oneself. Besides, the tapered structures are provided as the structures whose diameters become smaller in the areas remoter from the rotor lobe part 21 or 31 than the annular groove 26 or 36, thereby generating flow resistance in the flow of the liquid film, due to action of a centrifugal force generated in association with rotation of the male rotor 2 and the female rotor 3 (screw rotors), on the downstream side of the flow of the liquid film relative to the annular groove 26 or 36. This flow resistance suppresses the flow rate of the liquid film, and thus it becomes possible to keep the state in which the annular grooves 26 and 36 are filled with the liquid. Therefore, it becomes possible to improve the shaft seal performance by the liquid film in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1 oneself.

[0061] In addition, according to this configuration, only the characteristics of the shape of the discharge-side shaft parts 23 and 33 (shaft parts) of the male rotor 2 and the female rotor 3 (screw rotors) and the shape of the discharge-side shaft holes 54 and 55 (shaft holes) of the casing 4 corresponding to it allow the shaft sealing on the discharge side without using another shaft seal member such as a seal ring. Thus, the configuration for the shaft sealing on the discharge side can be simplified.

[0062] Furthermore, in the screw compressor 1 of the present embodiment, the second shaft sections 25 and 35 (predetermined portions) of the discharge-side shaft parts 23 and 33 (shaft parts) and the discharge-side shaft holes 54 and 55 (shaft holes) of the casing 4 may be each configured to have the whole of the area remoter from the rotor lobe part 21 or 31 than the position of the annular groove 26 or 36 as the tapered structure.

[0063] According to this configuration, the gap (shaft gap Sc) formed between the second shaft section 25 or 35 (predetermined portion) of the discharge-side shaft part 23 or 33 (shaft part) and the discharge-side shaft hole 54 or 55 (shaft hole) has the tapered surface-like shape (conical surface-like shape) in the whole of the downstream side relative to the annular groove 26 or 36. Therefore, the centrifugal force can be caused to act on the oil film O over the entire length of the oil film O that flows on the downstream side relative to the annular groove 26 or 36.

[0064] Moreover, in the screw compressor 1 of the present embodiment, the second shaft sections 25 and 35 (predetermined portions) of the discharge-side shaft parts 23 and 33 (shaft parts) and the discharge-side shaft holes 54 and 55 (shaft holes) of the casing 4 each have, also in the area closer to the rotor lobe part 21 or 31 relative to the position of the annular groove 26 or 36, the tapered structure with the diameter smaller toward such a direction as to get further away from the rotor lobe part 21 or 31.

[0065] According to this configuration, the whole of the shaft gap Sc formed between the second shaft section 25 or 35 (predetermined portion) of the discharge-side shaft part 23 or 33 (shaft part) and the discharge-side shaft hole 54 or 55 (shaft hole) is formed into the tapered surface-like shape (conical surface-like shape). Therefore, it is possible to reduce the amount of leakage of the compressed gas compared with the shaft gap with a circular cylindrical surface-like shape.

[Modification Example of First Embodiment]

[0066] Next, a screw compressor according to a modification example of the first embodiment of the present invention will be exemplified and described with use of FIG. 9. FIG. 9 is a cross sectional view illustrating the structure of a discharge-side part in the screw compressor according to the modification example of the first embodiment of the present invention in an enlarged state. In FIG. 9, one that has the same numeral as a numeral indicated in FIG. 1 to FIG. 8 is a similar part. Therefore, detailed description thereof is omitted.

[0067] A difference of a screw compressor 1A according to the modification example of the first embodiment illustrated in FIG. 9 from the screw compressor 1 according to the first embodiment is that the structure of a second shaft section 25A in a discharge-side shaft part 23A of a male rotor 2A and the structure of a discharge-side shaft hole 54A of a casing 4A (discharge-side casing 42A) in which the second shaft section 25A of the discharge-side shaft part 23A is disposed are different. Although only the discharge-side shaft part 23A of the male rotor 2A and the discharge-side shaft hole 54A of the casing 4A corresponding to it are illustrated in FIG. 9, it is possible that the discharge-side shaft part of the female rotor and the discharge-side shaft hole of the casing 4A corresponding to it also employ a similar structure.

[0068] Specifically, the second shaft section 25A in the discharge-side shaft part 23A of the male rotor 2A has a circular columnar shaft section 27 that is connected to the first shaft section 24 and a tapered shaft section 28 that is located on the side of the rotor lobe part 21 relative to the circular columnar shaft section 27 and is continuous with the circular columnar shaft section 27. The outer diameter of the circular columnar shaft section 27 is set larger than that of the first shaft section 24, for example. The tapered shaft section 28 is formed into a tapered shape (truncated cone shape) that tapers toward the direction of the circular columnar shaft section 27 (such a direction as to get further away from the rotor lobe part 21). The discharge-side shaft hole 54A of the casing 4A (discharge-side casing 42A) has a first hole section 541 in which the circular columnar shaft section 27 of the second shaft section 25A is disposed and a second hole section 542 in which the tapered shaft section 28 of the second shaft section 25A is disposed, according to the shape of the second shaft section 25A of the discharge-side shaft part 23A. One side of the first hole section 541 opens to the discharge-side bearing chamber 51 and the other side thereof is continuous with the second hole section 542. One side of the second hole section 542 is continuous with the first hole section 541 and the other side thereof opens to the housing chamber 45.

[0069] The outer circumferential surface of the circular columnar shaft section 27 in the second shaft section 25A and an inner circumferential surface that defines the first hole section 541 in the discharge-side shaft hole 54A of the casing 4A have a first opposed surface 27s and a second opposed surface 541s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers). The first opposed surface 27s of the circular columnar shaft section 27 and the second opposed surface 541s of the first hole section 541 are formed as circular cylindrical surfaces whose outer diameter and inner diameter, respectively, are substantially constant. The outer circumferential surface of the tapered shaft section 28 in the second shaft section 25A and an inner circumferential surface that defines the second hole section 542 in the discharge-side shaft hole 54A of the casing 4A have a first opposed surface 28s and a second opposed surface 542s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers). The first opposed surface 28s of the tapered shaft section 28 is formed as a first tapered surface that tapers toward the direction of the discharge-side bearings 7 and 8 (such a direction as to get further away from the rotor lobe part 21) (inclined surface that gradually converges as the position gets closer to the discharge-side bearings 7 and 8). The second opposed surface 542s of the second hole section 542 of the casing 4A is formed as a second tapered surface that tapers toward the direction of the discharge-side bearing chamber 51 (such a direction as to get further away from the housing chamber 45) (inclined surface that gradually converges as the position gets closer to the discharge-side bearing chamber 51) according to the shape of the first tapered surface (first opposed surface 28s) of the tapered shaft section 28.

[0070] That is, a first opposed surface of the second shaft section 25A has a first outer circumferential opposed surface that is the first opposed surface 27s of the circular columnar shaft section 27 configured as the circular cylindrical surface and a second outer circumferential opposed surface that is located closer to the rotor lobe part 21 relative to the first

outer circumferential opposed surface 27s and is the first opposed surface 28s of the tapered shaft section 28 configured as the first tapered surface that tapers toward such a direction as to get further away from the rotor lobe part 21. A second opposed surface of the discharge-side shaft hole 54A of the casing 4A has a first inner circumferential opposed surface that is the second opposed surface 541s of the first hole section 541 configured as the circular cylindrical surface opposed to the first opposed surface 27s of the circular columnar shaft section 27, which is the first outer circumferential opposed surface of the second shaft section 25A, and a second inner circumferential opposed surface that is located closer to the housing chamber 45 relative to the first inner circumferential opposed surface 541s, that is opposed to the first opposed surface 28s of the tapered shaft section 28, which is the second outer circumferential opposed surface of the second shaft section 25A, and that is the second opposed surface 542s of the second hole section 542 configured as the second tapered surface that tapers in such a direction as to get further away from the housing chamber 45.

[0071] In the present modification example, in the gap (discharge-side shaft gap) formed between the first opposed surfaces 27s and 28s of the second shaft section 25A of the male rotor 2A and the second opposed surfaces 541s and 542s of the discharge-side shaft hole 54A of the casing 4A, the discharge-side shaft gap formed between the first opposed surface 27s of the circular columnar shaft section 27 and the second opposed surface 541s of the first hole section 541 has a circular cylindrical surface-like shape. In addition, the discharge-side shaft gap formed between the first opposed surface 28s of the tapered shaft section 28 and the second opposed surface 542s of the second hole section 542 has a tapered surface-like shape (conical surface-like shape) that tapers toward such a direction as to get further away from the housing chamber 45.

[0072] The first opposed surface 28s (first tapered surface) of the tapered shaft section 28 of the second shaft section 25A is provided with the annular groove 26 that functions as an expansion chamber. The formation position of the annular groove 26 is optional as long as it is a position with which the tapered first opposed surface 28s remains on the side of the discharge-side bearings 7 and 8 (first shaft section 24) relative to the annular groove 26. The annular groove 26 is provided at a midway position between the circular columnar shaft section 27 and the rotor lobe part 21 in the first opposed surface 28s as illustrated in FIG. 9 for example. It is preferable that the formation position of the annular groove 26 be a position with which the length of the tapered first opposed surface 28s lying on the side of the discharge-side bearings 7 and 8 (first shaft section 24) relative to the annular groove 26 is longer, that is, a position closer to the rotor lobe part 21, in terms of sealing the leakage of the compressed gas in the working chambers C to the discharge-side bearing chamber 51 through the discharge-side shaft gap. Furthermore, the configuration is made in such a manner that the lubricating oil (liquid for the shaft sealing) is supplied to the annular groove 26. That is, the second route 62 of the oil feed path 60 of the casing 4A is configured to open on the second opposed surface 542s (second tapered surface) of the second hole section 542 of the discharge-side shaft hole 54A and at a position opposed to the annular groove 26 of the tapered shaft section 28 of the second shaft section 25A.

[0073] In such a structure, when the compressed gas involving pressure distribution in the circumferential direction flows into the annular groove 26 through the discharge-side shaft gap, the pressure of the compressed gas that has leaked from the working chamber C in the discharge process or at a late stage of the compression process or lowers by the annular groove 26 playing a role as an expansion chamber. Therefore, the lubricating oil supplied from the oil feed path 60 to the annular groove 26 is allowed to flow into the annular groove 26 across the whole thereof in the circumferential direction even when the pressure thereof is lower than the discharge pressure of the screw compressor 1, and flows into the discharge-side bearing chamber 51 through the discharge-side shaft gap to lubricate the discharge-side bearings 7 and 8. That is, it becomes possible to form an oil film (oil seal) across the whole of the annular discharge-side shaft gap in the circumferential direction on the side of the discharge-side bearing chamber 51 relative to the annular groove 26.

[0074] Moreover, in the present modification example, the one portion 28s of the first opposed surface of the second shaft section 25A in the discharge-side shaft part 23A of the male rotor 2A is formed as the first tapered surface that tapers toward the direction of the discharge-side bearings 7 and 8. In addition, the one portion 542s of the second opposed surface of the discharge-side shaft hole 54A of the casing 4A is formed as the second tapered surface that tapers toward the direction of the discharge-side bearing chamber 51 according to the shape of the first tapered surface (one section 28s of the first opposed surface) of the second shaft section 25A. This makes a portion of the discharge-side shaft gap on the side closer to the housing chamber 45 have a tapered surface-like shape (conical surface-like shape) that tapers in the direction of the discharge-side bearing chamber 51.

[0075] In such a structure, flow resistance is generated in a region of the discharge-side shaft gap on the side of the discharge-side bearings 7 and 8 relative to the annular groove 26 (flow path on the downstream side). That is, in the lubricating oil that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap with the tapered surface-like shape that tapers in the direction of the discharge-side bearing chamber 51a, a force toward the upstream side in the flow direction of the lubricating oil acts due to a centrifugal force. This flow resistance suppresses the flow rate of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51, and thus the state in which the annular groove 26 is filled with the lubricating oil across the whole thereof in the circumferential direction can be kept.

[0076] As above, the lubricating oil supplied from the second route 62 of the oil feed path 60 to the annular groove 26 flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap with the tapered surface-like shape while the state in which the whole of the annular groove 26 in the circumferential direction is filled with the lubricating oil is kept. That is, an oil film is formed on the side of the discharge-side bearing chamber 51 relative to the annular groove 26 in the discharge-side shaft gap. Therefore, even when the pressure of the lubricating oil supplied to the discharge-side shaft gap is lower than the discharge pressure of the screw compressor 1, the amount of leakage of the compressed gas from the plurality of working chambers C to the discharge-side shaft gap can be suppressed by the lubricating oil with which the annular groove 26 is filled across the whole thereof in the circumferential direction and the oil film formed on the side of the discharge-side bearing chamber 51 relative to the annular groove 26, and thus the shaft seal performance on the discharge side improves.

[0077] According to the above-described modification example of the first embodiment, similarly to the first embodiment, when a gas involving pressure distribution in the circumferential direction flows into the annular groove 26 through the shaft gap Sc (gap) between the discharge-side shaft hole 54A (shaft hole) and the second shaft section 25A (predetermined portion) of the discharge-side shaft part 23A (shaft part), the annular groove 26 plays a role as an expansion chamber, thereby making the pressure distribution of the gas even. Thus, it becomes possible to form a liquid film across the whole of the annular shaft gap Sc in the circumferential direction even in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1A oneself. Besides, the tapered structures are provided as the structures whose diameters become smaller in the areas remoter from the rotor lobe part 21 than the annular groove 26, thereby generating flow resistance in the flow of the liquid film, due to action of a centrifugal force generated in association with rotation of the male rotor 2A (screw rotor), on the downstream side of the flow of the liquid film relative to the annular groove 26. This flow resistance suppresses the flow rate of the liquid film, and thus it becomes possible to keep the state in which the annular groove 26 is filled with the liquid. Therefore, it becomes possible to improve the shaft seal performance by the liquid film in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1A oneself.

[0078] Furthermore, in the screw compressor 1A according to the present modification example, the second shaft section 25A (predetermined portion) of the discharge-side shaft part 23A (shaft part) and the discharge-side shaft hole 54A (shaft hole) of the casing 4A are each configured to have only the part 28 of the area remoter from the rotor lobe part 21 than the position of the annular groove 26 as the tapered structure. This configuration can enhance the rigidity of the discharge-side shaft part 23A (shaft part) relative to the discharge-side shaft part 23 of the first embodiment.

[Second Embodiment]

[0079] Next, the structure of a screw compressor according to a second embodiment of the present invention will be exemplified and described with use of FIG. 10. FIG. 10 is a cross sectional view illustrating the screw compressor according to the second embodiment of the present invention with partial omission. In FIG. 10, one that has the same numeral as a numeral indicated in FIG. 1 to FIG. 9 is a similar part. Therefore, detailed description thereof is omitted.

[0080] Differences of a screw compressor 1B according to the second embodiment illustrated in FIG. 10 from the screw compressor 1 according to the first embodiment are that a second shaft section 25B in a discharge-side shaft part 23B of a male rotor 2B has not a tapered structure but a stepped structure, and that a discharge-side shaft hole 54B of a casing 4B (discharge-side casing 42B) in which the second shaft section 25B of the discharge-side shaft part 23B is disposed has not a tapered structure but a stepped structure according to the shape of the second shaft section 25B. Although only the discharge-side shaft part 23B of the male rotor 2B and the discharge-side shaft hole 54B of the casing 4B corresponding to it are illustrated in FIG. 10, it is possible that the discharge-side shaft part of the female rotor and the discharge-side shaft hole of the casing 4B corresponding to it also employ a similar structure.

[0081] Specifically, the second shaft section 25B in the discharge-side shaft part 23B of the male rotor 2B has, in an area remoter from the rotor lobe part 21 than the position of the annular groove 26, the stepped structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part 21. The stepped structure of the second shaft section 25B is configured by only one step portion. Specifically, the second shaft section 25B has a first circular columnar shaft section 251 that is connected to the first shaft section 24 and a second circular columnar shaft section 252 that is located on the side of the rotor lobe part 21 relative to the first circular columnar shaft section 251 and is connected to the first circular columnar shaft section 251. The second circular columnar shaft section 252 is configured to have a larger diameter than the first circular columnar shaft section 251. The second shaft section 25B has an annular step portion 29 at the connection position between the second circular columnar shaft section 252 of the larger-diameter side and the first circular columnar shaft section 251 of the smaller-diameter side.

[0082] The discharge-side shaft hole 54B of the casing 4B (discharge-side casing 42B) has, in an area remoter from the rotor lobe part 21 than the position of the annular groove, the stepped structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part 21 26 according to the structure of the second shaft section 25B of the discharge-side shaft part 23B. The stepped structure of the discharge-side shaft hole 54B is configured by

only one step portion. Specifically, the discharge-side shaft hole 54B has a first hole section 541B in which the first circular columnar shaft section 251 of the second shaft section 25B is disposed and a second hole section 542B in which the second circular columnar shaft section 252 of the second shaft section 25B is disposed. One side of the first hole section 541B opens to the discharge-side bearing chamber 51 and the other side thereof opens to the second hole section 542B. One side of the second hole section 542B opens to the first hole section 541B and the other side thereof opens to the housing chamber 45. The second hole section 542B is configured to have a larger hole diameter than the first hole section 541B. The discharge-side shaft hole 54B has an annular step portion 56 at the connection position between the second hole section 542B of the larger-diameter side and the first hole section 541B of the smaller-diameter side.

[0083] The outer circumferential surface of the first circular columnar shaft section 251 in the second shaft section 25B and an inner circumferential surface defining the first hole section 541B in the discharge-side shaft hole 54B of the casing 4B have a first opposed surface 251s (see FIG. 11) and a second opposed surface 541s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers) in the radial direction. The first opposed surface 251s of the first circular columnar shaft section 251 and the second opposed surface 541s of the first hole section 541B are formed as circular cylindrical surfaces whose outer diameter and inner diameter, respectively, are constant. The outer circumferential surface of the second circular columnar shaft section 252 in the second shaft section 25B and an inner circumferential surface defining the second hole section 542B in the discharge-side shaft hole 54B of the casing 4B have a first opposed surface 252s (see FIG. 11) and a second opposed surface 542s opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers) in the radial direction. The first opposed surface 252s of the second circular columnar shaft section 252 and the second opposed surface 542s of the second hole section 542B are formed as circular cylindrical surfaces whose outer diameter and inner diameter, respectively, are constant. Moreover, the step portion 29 of the second shaft section 25B and the step portion 56 of the discharge-side shaft hole 54B are configured to be opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers) in the axial direction.

[0084] That is, a first opposed surface of the second shaft section 25B has a first outer circumferential opposed surface of the circular cylindrical surface that is the first opposed surface 251s of the first circular columnar shaft section 251, a second outer circumferential opposed surface of the circular cylindrical surface that is located closer to the rotor lobe part 21 relative to the first outer circumferential opposed surface part 251s and is the first opposed surface 252s of the second circular columnar shaft section 252 with a larger diameter than the first circular columnar shaft section 251, and the step portion 29 formed at the connection position between the first outer circumferential opposed surface 251s with the smaller diameter and the second outer circumferential opposed surface 252s with the larger diameter. A second opposed surface of the discharge-side shaft hole 54B of the casing 4B has a first inner circumferential opposed surface that is the second opposed surface 541s of the first hole section 541B configured as the circular cylindrical surface opposed to the first opposed surface 251s of the first circular columnar shaft section 251 (first outer circumferential opposed surface of the second shaft section 25B), a second inner circumferential opposed surface that is located closer to the housing chamber 45 relative to the first inner circumferential opposed surface 541s and is opposed to the first opposed surface 252s of the second circular columnar shaft section 252 (second outer circumferential opposed surface of the second shaft section 25B), and that is the second opposed surface 542s of the second hole section 542B with a larger diameter than the first inner circumferential opposed surface 541s, and the step portion 56 formed at the connection position between the first inner circumferential opposed surface 541s with the smaller diameter and the second inner circumferential opposed surface 542s with the larger diameter.

[0085] In the present embodiment, in the gap (discharge-side shaft gap) formed between the first opposed surfaces 251s, 252s, and 29 of the second shaft section 25B of the male rotor 2B and the second opposed surfaces 541s, 542s, and 56 of the discharge-side shaft hole 54B of the casing 4B, the discharge-side shaft gap formed between the first opposed surface 251s of the first circular columnar shaft section 251 and the second opposed surface 541s of the first hole section 541B has a circular cylindrical surface-like shape. In addition, the discharge-side shaft gap formed between the first opposed surface 252s of the second circular columnar shaft section 252 and the second opposed surface 542s of the second hole section 542B has a circular cylindrical surface-like shape. The discharge-side shaft gap formed between the step portion 29 of the second shaft section 25B and the step portion 56 of the discharge-side shaft hole 54B has a circular annular shape that spreads in the radial direction.

[0086] In the present embodiment, the annular groove 26 is provided on the first opposed surface 252s of the second circular columnar shaft section 252 of the larger-diameter side in the second shaft section 25B. The annular groove 26 needs to be provided in an area closer to the rotor lobe part 21 relative to the step portion 29 in the second shaft section 25B. The annular groove 26 functions as an expansion chamber whose volume is relatively larger with respect to the size of the discharge-side shaft gap (several tens to several hundreds of micrometers). It is preferable that the formation position of the annular groove 26 be a position with which the length of the first opposed surface 252s lying on the side of the discharge-side bearings 7 and 8 (first shaft section 24) relative to the annular groove 26 is longer, that is, a position

closer to the rotor lobe part 21, in terms of sealing the leakage of the compressed gas in the working chambers C to the discharge-side bearing chamber 51 through the discharge-side shaft gap.

[0087] Next, operation and effect of shaft sealing on the discharge side in the screw compressor according to the second embodiment will be described with use of FIG. 11. FIG. 11 is an explanatory diagram illustrating the operation and effect of the shaft sealing on the discharge side in the screw compressor according to the second embodiment of the present invention.

[0088] Pressure distribution exists in the plurality of working chambers C (see FIG. 3) that line up in the circumferential direction at the discharge-side end face 21c of the rotor lobe part 21 of the male rotor 2B, and therefore pressure distribution in the circumferential direction is generated also in the compressed gas that flows into the discharge-side shaft gap Sc. When the compressed gas involving the pressure distribution in the circumferential direction flows into the annular groove 26, the annular groove 26 plays a role as an expansion chamber to make the pressure distribution in the circumferential direction in the compressed gas even. Thus, it becomes possible to form an oil film (oil seal) across the whole of the annular discharge-side shaft gap Sc in the circumferential direction even when the pressure of the lubricating oil of the oil feed path 60 is lower than the discharge pressure of the screw compressor 1B.

[0089] The lubricating oil that has flown into the annular groove 26 flows into the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc to lubricate the discharge-side bearings 7 and 8. At this time, it is desirable that the annular groove 26 be filled with the lubricating oil across the whole thereof in the circumferential direction, and the flow rate of the lubricating oil that flows into the annular groove 26 from the second route 62 of the oil feed path 60 therefore needs to become higher than that of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51.

[0090] In the present embodiment, a configuration is made to generate flow resistance in a region of the discharge-side shaft gap Sc on the side of the discharge-side bearings 7 and 8 relative to the annular groove 26 (flow path on the downstream side), thereby limiting the flow rate of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51. Specifically, the lubricating oil O that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc flows toward the circumferential direction in association with rotation of the second shaft section 25B of the discharge-side shaft part 23B due to the viscosity thereof. At this time, as illustrated in an enlarged view in FIG. 11, the centrifugal force Fc toward the radially outside acts on the lubricating oil O existing in the discharge-side shaft gap Sc formed between the step portion 29 of the second shaft section 25B and the step portion 56 of the discharge-side shaft hole 54B. Therefore, a force toward the upstream side in the flow direction of the lubricating oil acts on the lubricating oil that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc. That is, flow resistance is generated in the lubricating oil that flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc. This flow resistance suppresses the flow rate of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51.

[0091] Furthermore, the second shaft section 25B and the discharge-side shaft hole 54B have the stepped structures, and thus a portion that bends at a steep angle exists in the discharge-side shaft gap Sc as the flow path in which the oil film flows. The bent portion of the discharge-side shaft gap Sc attributed to the stepped structures become flow resistance against the oil film that flows.

[0092] As above, in the region of the discharge-side shaft gap Sc on the side of the discharge-side bearings 7 and 8 relative to the annular groove 26 (flow path on the downstream side), the flow resistance is generated due to the action of the centrifugal force, and the flow resistance is generated due to the bent portion of the discharge-side shaft gap Sc. These limits the flow rate of the lubricating oil that flows out from the annular groove 26 to the discharge-side bearing chamber 51, and therefore it is possible to keep the state in which the annular groove 26 is filled with the lubricating oil across the whole thereof in the circumferential direction. The magnitude of the flow resistance due to the action of the centrifugal force is defined by the magnitude of the radius Rt of the first circular columnar shaft section 251 of the smaller-diameter side and the radius Rg of the second circular columnar shaft section 252 of the larger-diameter side.

[0093] Accordingly, the lubricating oil supplied from the second route 62 of the oil feed path 60 to the annular groove 26 flows from the annular groove 26 to the discharge-side bearing chamber 51 through the discharge-side shaft gap Sc while keeping the state in which the whole of the annular groove 26 in the circumferential direction is filled with the lubricating oil. That is, the oil film O is formed on the side of the discharge-side bearing chamber 51 relative to the annular groove 26 in the discharge-side shaft gap Sc. Even when the pressure of the lubricating oil O supplied to the discharge-side shaft gap Sc is lower than the discharge pressure of the screw compressor 1, the amount of leakage of the compressed gas from the plurality of working chambers C to the discharge-side shaft gap Sc can be suppressed by the lubricating oil with which the annular groove 26 is filled across the whole thereof in the circumferential direction and the oil film O formed on the side of the discharge-side bearing chamber 51 relative to the annular groove 26, and thus the shaft seal performance on the discharge side improves. In terms of the effect of the shaft seal performance on the discharge side, it is advantageous for improving the shaft seal performance that the route of the oil film O that flows from the annular groove 26 toward the side of the discharge-side bearing chamber 51 (first shaft section 24) is longer.

Therefore, when improvement in the shaft seal performance is considered, it is preferable to form the annular groove 26 and the opening of the second route 62 of the oil feed path 60 at positions closer to the rotor lobe part 21 (housing chamber 45).

[0094] The screw compressor 1B of the above-described second embodiment includes the male rotor 2B (screw rotor) including the rotor lobe part 21 having the helical lobes 21a and the discharge-side shaft part 23B (shaft part) provided at the discharge-side end of the rotor lobe part 21 in the axial direction. The screw compressor 1B includes also the casing 4B having the housing chamber 45 that houses the rotor lobe part 21, the discharge-side shaft hole 54B (shaft hole) through which the discharge-side shaft part 23B (shaft part) passes, and the second route 62 of the oil feed path 60 (liquid feed path) that introduces to the discharge-side shaft hole 54B (shaft hole) the lubricating oil (liquid) supplied from the external, and the casing 4 is configured to define the working chambers C together with the rotor lobe part 21. The second shaft section 25B (predetermined portion) of the discharge-side shaft part 23B (shaft part) disposed in the discharge-side shaft hole 54B (shaft hole) and the discharge-side shaft hole 54B (shaft hole) of the casing 4B have the first opposed surfaces 251s, 252s, and 29 and the second opposed surfaces 541s, 542s, and 56 opposed to each other with the gap Sc. The discharge-side shaft part 23B (shaft part) has the annular groove 26 on the first opposed surface 252s, and the second route 62 (liquid feed path) is configured to open on the second opposed surface 542s of the discharge-side shaft hole 54B (shaft hole) of the casing 4B and at a position opposed to the annular groove 26. The second shaft section 25B (predetermined portion) of the discharge-side shaft part 23B (shaft part) and the discharge-side shaft hole 54B (shaft hole) of the casing 4B each have, in an area remoter from the rotor lobe part 21 than the position of the annular groove 26, the stepped structure as a structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part 21.

[0095] According to this configuration, when a gas involving pressure distribution in the circumferential direction flows into the annular groove 26 through the shaft gap Sc (gap) between the discharge-side shaft hole 54B (shaft hole) and the second shaft section 25B (predetermined portion) of the discharge-side shaft part 23B (shaft part), the annular groove 26 plays a role as an expansion chamber, thereby making the pressure distribution of the gas even. Thus, it becomes possible to form a liquid film across the whole of the annular shaft gap Sc in the circumferential direction even in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1B oneself. Besides, the stepped structures are provided as the structures with the diameters smaller in the areas remoter from the rotor lobe part 21 than the annular groove 26, thereby generating flow resistance in the flow of the liquid film, due to action of a centrifugal force generated in association with rotation of the male rotor 2B (screw rotor), on the downstream side of the flow of the liquid film relative to the annular groove 26. This flow resistance suppresses the flow rate of the liquid film, and thus it becomes possible to keep the state in which the annular groove 26 is filled with the liquid. Therefore, it becomes possible to improve the shaft seal performance by the liquid film in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1B oneself.

[0096] Furthermore, in the second shaft section 25B (predetermined portion) of the discharge-side shaft part 23B (shaft part) and the discharge-side shaft hole 54B (shaft hole) of the casing 4B in the screw compressor 1B of the present embodiment, the structures with the diameters smaller toward such a direction as to get further away from the rotor lobe part 21 are the stepped structures.

[0097] According to this configuration, the bent portion exist in the discharge-side shaft gap Sc as the flow path in which the oil film flows, and therefore the flow resistance against the flow of the oil film further increases. This allows further improvement in the shaft seal performance by the liquid film.

[0098] Moreover, in the screw compressor 1B of the present embodiment, the stepped structures of the second shaft section 25B (predetermined portion) of the discharge-side shaft part 23B (shaft part) and the discharge-side shaft hole 54B (shaft hole) of the casing 4B are each configured to have only one step portion. According to this configuration, it is possible to keep also easiness of processing while allowing improvement in the shaft seal performance by the liquid film.

[Modification Example of Second Embodiment]

[0099] Next, a screw compressor according to a modification example of the second embodiment of the present invention will be exemplified and described with use of FIG. 12. FIG. 12 is a cross sectional view illustrating the structure of a discharge-side part in the screw compressor according to the modification example of the second embodiment of the present invention in an enlarged state. In FIG. 12, one that has the same numeral as a numeral indicated in FIG. 1 to FIG. 12 is a similar part. Therefore, detailed description thereof is omitted.

[0100] Differences of a screw compressor 1C according to the modification example of the second embodiment illustrated in FIG. 12 from the screw compressor 1B according to the second embodiment are that a second shaft section 25C in a discharge-side shaft part 23C of a male rotor 2C has a stepped structure with five step portions, and that a discharge-side shaft hole 54C of a casing 4C (discharge-side casing 42C) in which the second shaft section 25C of the discharge-side shaft part 23C is disposed has a stepped structure with five step portions according to the shape of the second shaft section 25C.

[0101] Specifically, the second shaft section 25C in the discharge-side shaft part 23C of the male rotor 2C has, in an area remoter from the rotor lobe part 21 than the position of the annular groove 26, the stepped structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part 21. The stepped structure of the second shaft section 25C is configured by five step portions. Specifically, the second shaft section 25C has a first circular columnar shaft section 251C, a second circular columnar shaft section 252C, a third circular columnar shaft section 253C, a fourth circular columnar shaft section 254C, a fifth circular columnar shaft section 255C, and a sixth circular columnar shaft section 256C sequentially from the side of the first shaft section 24 toward the side of the rotor lobe part 21. The second shaft section 25C is configured in such a manner that the diameter becomes smaller sequentially from the sixth circular columnar shaft section 256C toward the first circular columnar shaft section 251C. The second shaft section 25C has a first step portion 291, a second step portion 292, a third step portion 293, a fourth step portion 294, and a fifth step portion 295 sequentially from the sixth circular columnar shaft section 256C of the larger-diameter side toward the first circular columnar shaft section 251C of the smaller-diameter side.

[0102] The discharge-side shaft hole 54C of the casing 4C (discharge-side casing 42C) has, in an area remoter from the rotor lobe part 21 than the position of the annular groove 26, the stepped structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part 21 according to the structure of the second shaft section 25C of the discharge-side shaft part 23C. The stepped structure of the discharge-side shaft hole 54C is configured by five step portions. Specifically, the discharge-side shaft hole 54C has a first hole section 541C, a second hole section 542C, a third hole section 543C, a fourth hole section 544C, a fifth hole section 545C, and a sixth hole section 546C in such a manner that the first circular columnar shaft section 251C to the sixth circular columnar shaft section 256C in the second shaft section 25C are sequentially disposed therein. The discharge-side shaft hole 54C is configured in such a manner that the diameter becomes smaller sequentially from the sixth hole section 546C toward the first hole section 541C. The discharge-side shaft hole 54C has a first step portion 561, a second step portion 562, a third step portion 563, a fourth step portion 564, and a fifth step portion 565 sequentially from the sixth hole section 546C of the larger-diameter side toward the first hole section 541C of the smaller-diameter side.

[0103] The outer circumferential surfaces of the first circular columnar shaft section 251C to the sixth circular columnar shaft section 256C in the second shaft section 25C and inner circumferential surfaces defining the first hole section 541C to the sixth hole section 546C in the discharge-side shaft hole 54C of the casing 4C corresponding to them have first opposed surfaces and second opposed surfaces opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers) in the radial direction. Moreover, the first step portion 291 to the fifth step portion 295 in the second shaft section 25C and the first step portion 561 to the fifth step portion 565 in the discharge-side shaft hole 54C corresponding to them are configured to be opposed to each other with a gap having a size in a predetermined range (for example, several tens to several hundreds of micrometers) in the axial direction.

[0104] In the present embodiment, the annular groove 26 is provided on the outer circumferential surface of the sixth circular columnar shaft section 256C with the maximum diameter in the second shaft section 25C. The annular groove 26 functions as an expansion chamber whose volume is relatively larger with respect to the size of the discharge-side shaft gap (several tens to several hundreds of micrometers). The annular groove 26 needs to be provided in an area of the second shaft section 25C closer to the rotor lobe part 21 relative to the fifth step portion 295. It is preferable that the formation position of the annular groove 26 be a position closer to the rotor lobe part 21 relative to the first step portion 291 in terms of sealing the leakage of the compressed gas in the working chambers C to the discharge-side bearing chamber 51 through the discharge-side shaft gap. In this case, the action of the centrifugal force generated on the oil film that flows in the discharge-side shaft gap becomes the largest, and the length of the flow path of the discharge-side shaft gap becomes the longest, and therefore the flow resistance becomes high correspondingly.

[0105] According to the above-described modification example of the second embodiment, similarly to the second embodiment, when a gas involving pressure distribution in the circumferential direction flows into the annular groove 26 through the shaft gap Sc (gap) between the discharge-side shaft hole 54C (shaft hole) and the second shaft section 25C (predetermined portion) of the discharge-side shaft part 23C (shaft part), the annular groove 26 plays a role as an expansion chamber, thereby making the pressure distribution of the gas even. Thus, it becomes possible to form a liquid film across the whole of the annular shaft gap Sc in the circumferential direction even in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1C oneself. Besides, the stepped structures are provided as the structures whose diameters become smaller in the areas remoter from the rotor lobe part 21 than the annular groove 26, thereby generating flow resistance in the flow of the liquid film, due to action of a centrifugal force generated in association with rotation of the male rotor 2C (screw rotor), on the downstream side of the flow of the liquid film relative to the annular groove 26. This flow resistance suppresses the flow rate of the liquid film, and thus it becomes possible to keep the state in which the annular groove 26 is filled with the liquid. Therefore, it becomes possible to improve the shaft seal performance by the liquid film in the case of supplying the liquid to the shaft gap Sc by using the discharge pressure of the screw compressor 1C oneself.

[0106] Furthermore, in the screw compressor 1C of the present modification example, the stepped structures of the

second shaft section 25C (predetermined portion) of the discharge-side shaft part 23C (shaft part) and the discharge-side shaft hole 54C (shaft hole) of the casing 4C are each configured to have a plurality of step portions. According to this configuration, the discharge-side shaft gap Sc has a large number of bent portions depending on the number of the step portions, thereby increasing the flow resistance and allowing the shaft seal performance by the liquid film to be further improved.

[Other Embodiments]

[0107] The present invention is not limited to the above-described embodiments and various modification examples are included therein. The above-described embodiments are described in detail in order to explain the present invention in an easy-to-understand manner, and are not necessarily limited to that including all configurations described. That is, it is possible to replace part of a configuration of a certain embodiment by a configuration of another embodiment. Furthermore, it is also possible to add a configuration of another embodiment to a configuration of a certain embodiment. Moreover, it is also possible to execute addition, deletion, or substitution of another configuration regarding part of a configuration of each embodiment.

[0108] For example, in the above-described embodiments, description has been made by taking the oil-flooded type screw compressors 1, 1A, 1B, and 1C as examples. However, the present invention can be applied to a liquid-flooded screw compressor to which a liquid such as water is supplied. Furthermore, although description has been made by taking the screw compressors 1, 1A, 1B, and 1C of the twin-rotor type as examples, the present invention can be applied also to screw compressors other than the twin-rotor type, such as a single-rotor type and a triple-rotor type.

[0109] Moreover, in the above-described first embodiment, the example of the configuration in which both the discharge-side shaft parts 23 and 33 of the male rotor 2 and the female rotor 3 have the second shaft sections 25 and 35 with the tapered shape, and the configuration in which the discharge-side shaft holes 54 and 55 of the casing 4 corresponding to them are each formed into the tapered shape has been illustrated. However, a configuration is also possible in which only either one of the discharge-side shaft part 23 of the male rotor 2 and the discharge-side shaft part 33 of the female rotor 3 has the second shaft section with the tapered shape. In this case, only the discharge-side shaft hole of the casing 4 corresponding to the discharge-side shaft part having the second shaft section with the tapered shape is formed into the tapered shape according to the tapered shape of the second shaft section.

Description of Reference Characters

[0110]

1, 1A, 1B, 1C: Screw compressor
 2, 2A, 2B, 2C: Male rotor (screw rotor)
 3: Female rotor (screw rotor)
 4, 4A, 4B, 4C: Casing
 21: Rotor lobe part
 21a: Male lobe (lobe)
 23, 23A, 23B, 23C: Discharge-side shaft part (shaft part)
 25s: First opposed surface
 251s: First opposed surface
 252s: First opposed surface
 26: Annular groove
 27s: First opposed surface
 28s: First opposed surface
 29: Step portion
 31: Rotor lobe part
 31a: Female lobe (lobe)
 33: Discharge-side shaft part (shaft part)
 35s: First opposed surface
 45: Housing chamber
 54, 54A, 54B, 54C: Discharge-side shaft hole (shaft hole)
 54s: Second opposed surface
 541s: Second opposed surface
 542s: Second opposed surface
 55: Discharge-side shaft hole (shaft hole)
 55s: Second opposed surface

56: Step portion
 60: Oil feed path (liquid feed path)
 62: Second route (liquid feed path)
 C: Working chamber
 Sc: Shaft gap (gap)

Claims

1. A screw compressor comprising:

a screw rotor including a rotor lobe part having a helical lobe and a shaft part provided at a discharge-side end of the rotor lobe part in an axial direction; and
 a casing having a housing chamber that houses the rotor lobe part, a shaft hole through which the shaft part passes, and a liquid feed path that introduces to the shaft hole a liquid supplied from an external, the casing being configured to define working chambers together with the rotor lobe part, wherein
 a predetermined portion of the shaft part disposed in the shaft hole and the shaft hole of the casing have a first opposed surface and a second opposed surface opposed to each other with a gap,
 the shaft part has an annular groove on the first opposed surface,
 the liquid feed path is configured so as to open on the second opposed surface of the shaft hole of the casing and at a position opposed to the annular groove, and
 the predetermined portion of the shaft part and the shaft hole of the casing each have, in an area remoter from the rotor lobe part than a position of the annular groove, a structure with a diameter smaller toward such a direction as to get further away from the rotor lobe part.

2. The screw compressor according to claim 1, wherein
 the structure is a stepped structure whose diameter smaller becomes toward such a direction as to get further away from the rotor lobe part.

3. The screw compressor according to claim 2, wherein
 the stepped structure is configured to have only one step portion.

4. The screw compressor according to claim 2, wherein
 the stepped structure is configured to have a plurality of step portions.

5. The screw compressor according to claim 1, wherein
 the structure is a tapered structure whose diameter smaller becomes toward such a direction as to get further away from the rotor lobe part.

6. The screw compressor according to claim 5, wherein
 the predetermined portion of the shaft part and the shaft hole of the casing are each configured to have whole of the area remoter from the rotor lobe part than the position of the annular groove as the tapered structure.

7. The screw compressor according to claim 5, wherein
 the predetermined portion of the shaft part and the shaft hole of the casing are each configured to have only a part of the area remoter from the rotor lobe part than the position of the annular groove as the tapered structure.

8. The screw compressor according to claim 6, wherein
 the predetermined portion of the shaft part and the shaft hole of the casing each have, also in an area closer to the rotor lobe part than the position of the annular groove, a tapered structure whose diameter becomes smaller toward such a direction as to get further away from the rotor lobe part.

FIG. 1

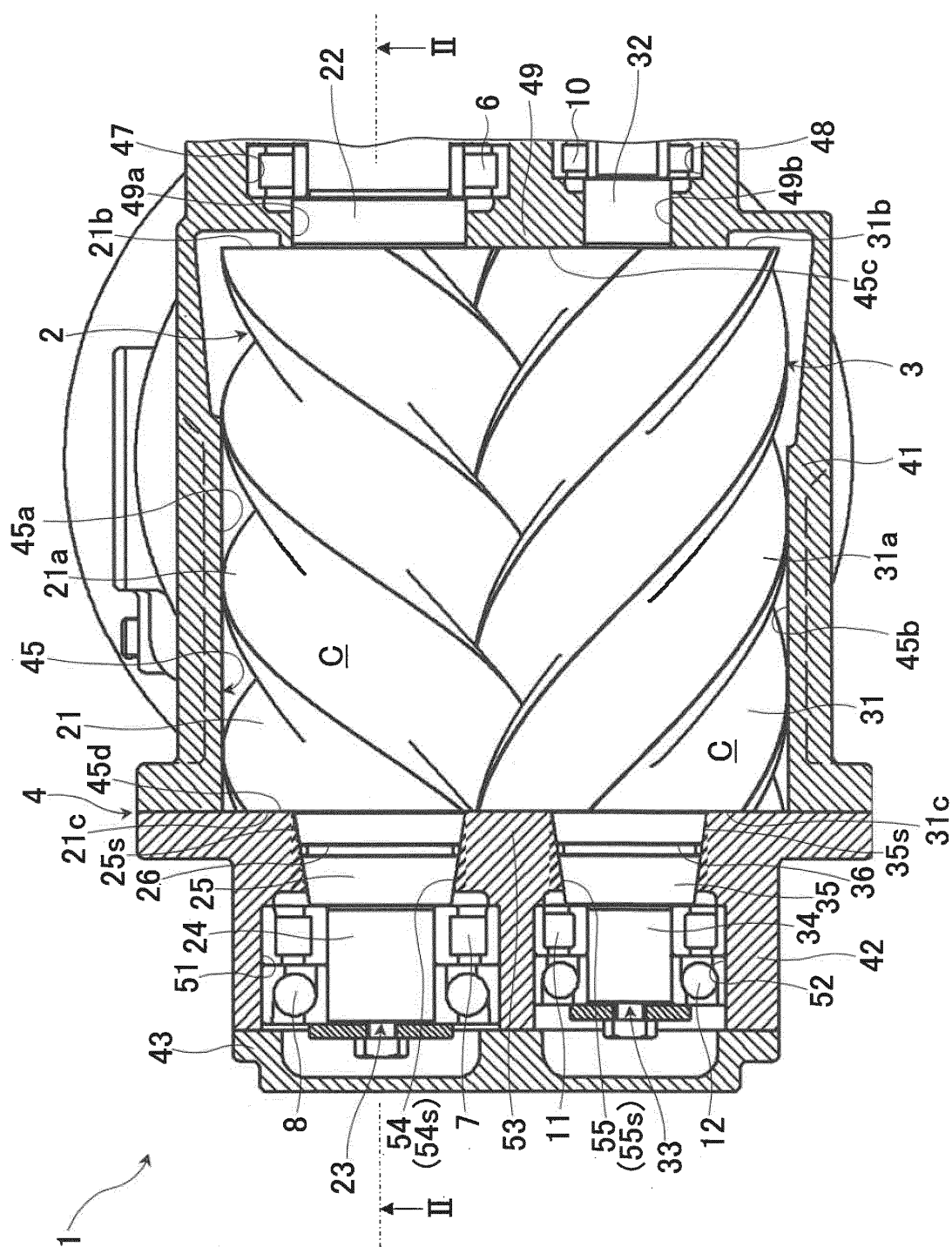
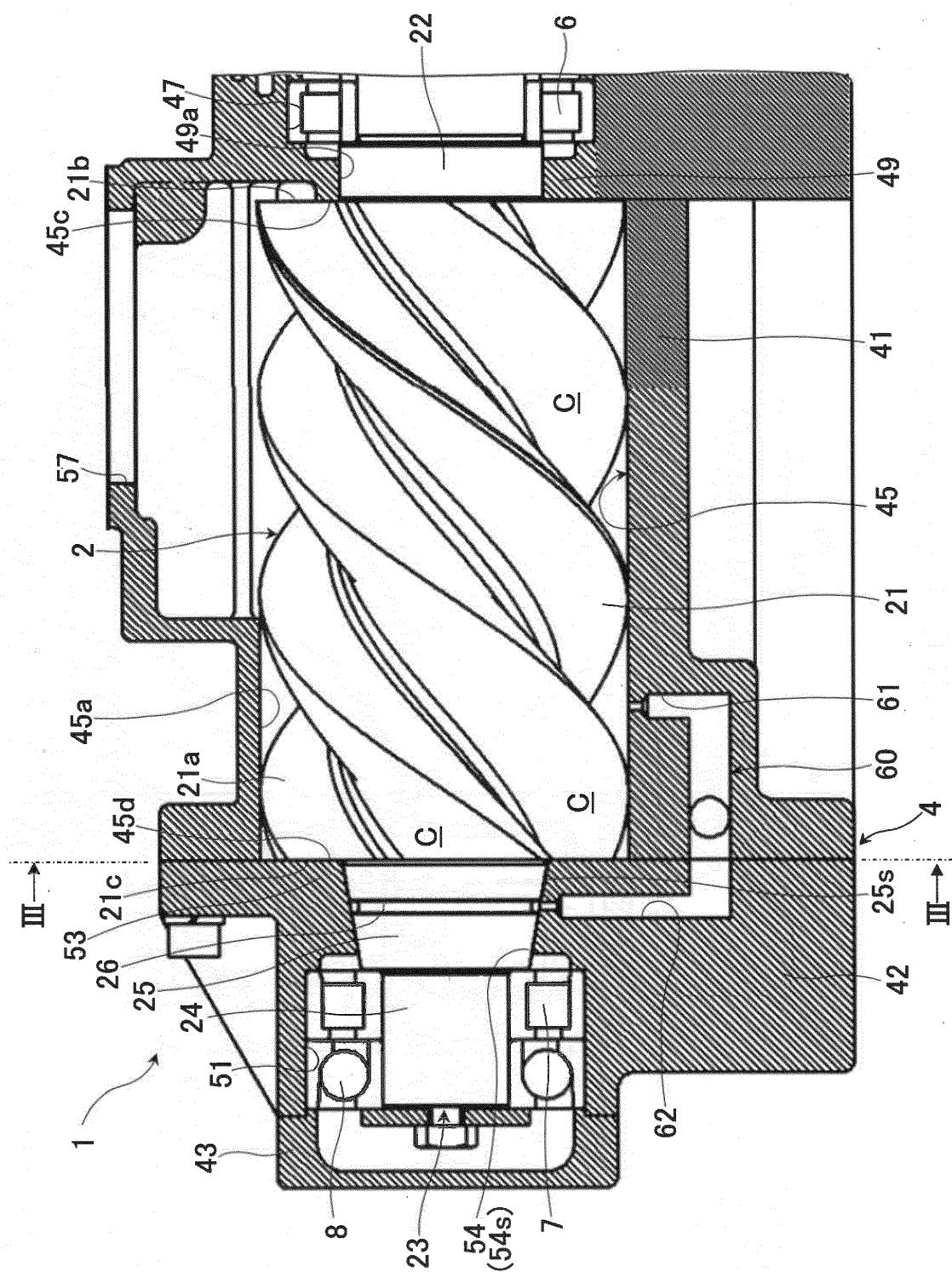


FIG. 2



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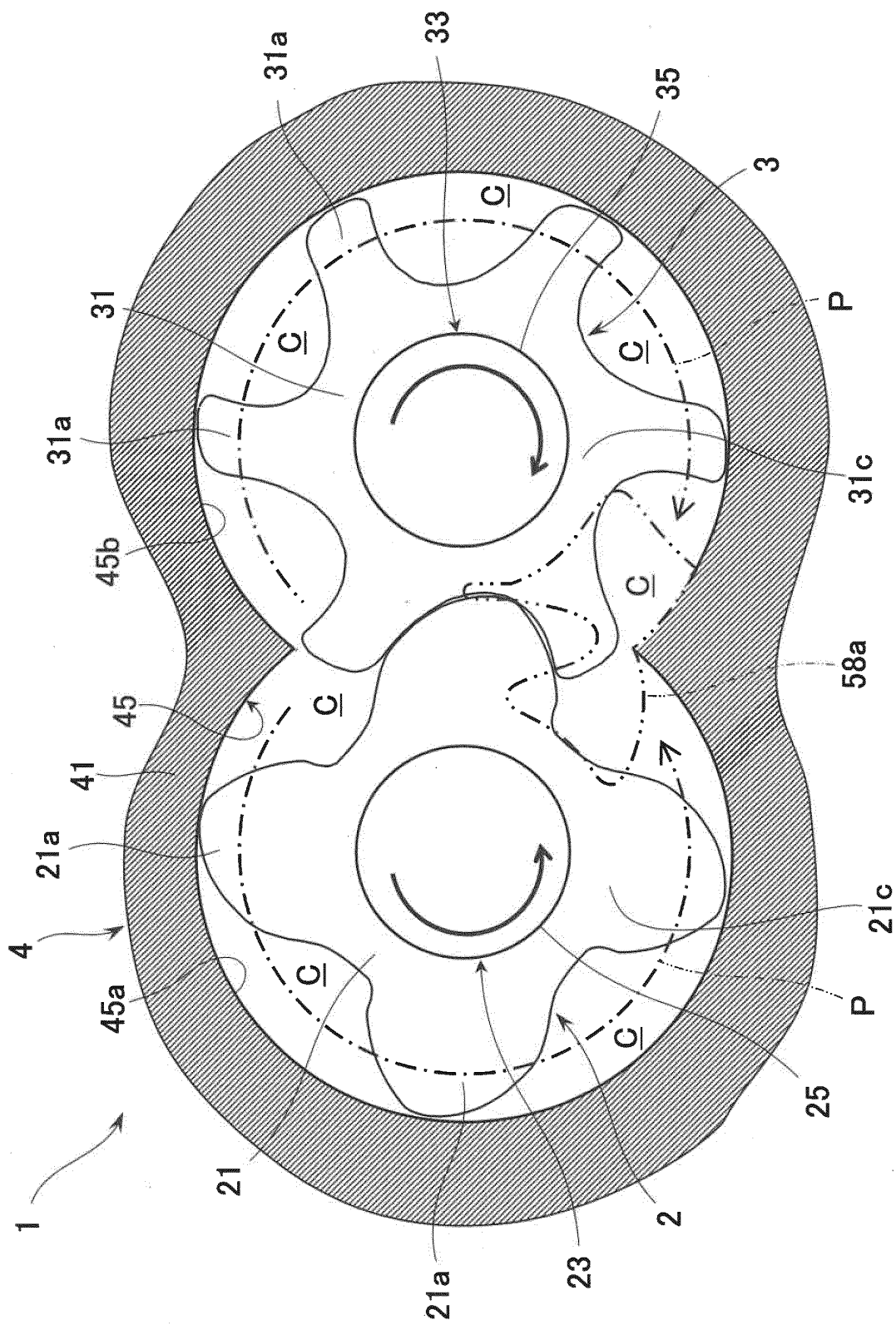
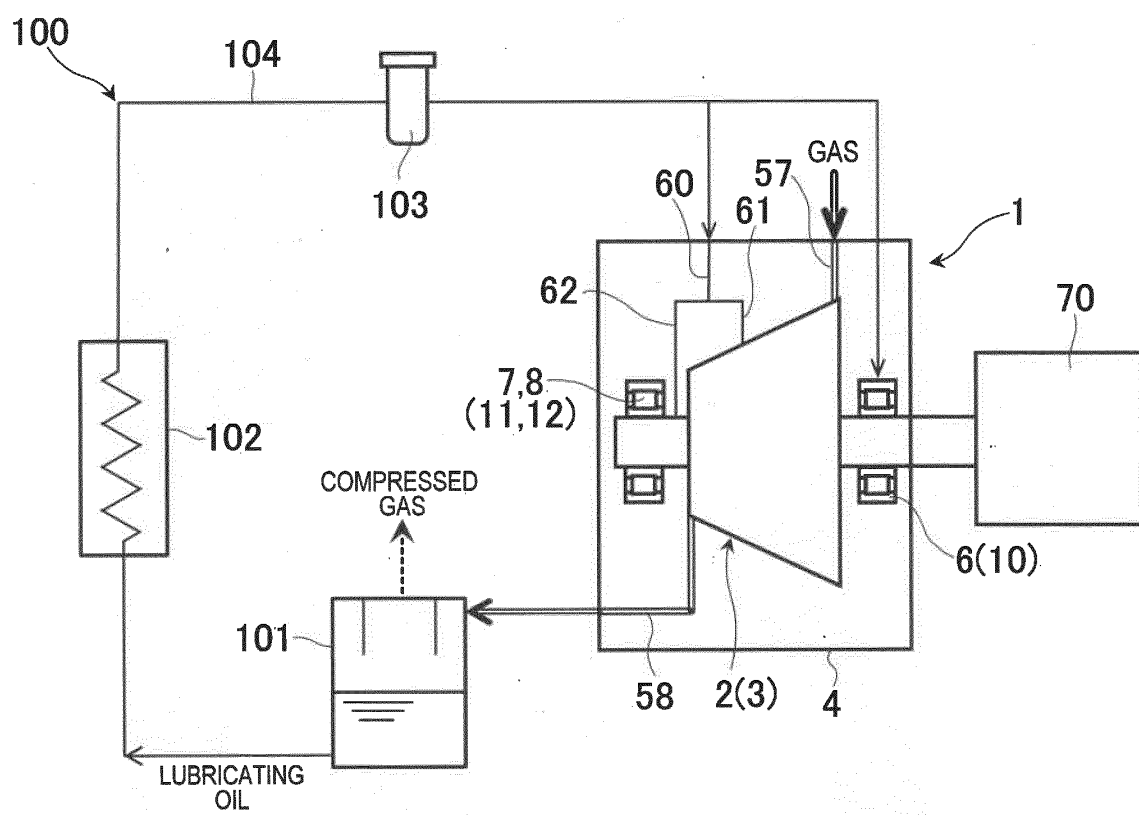
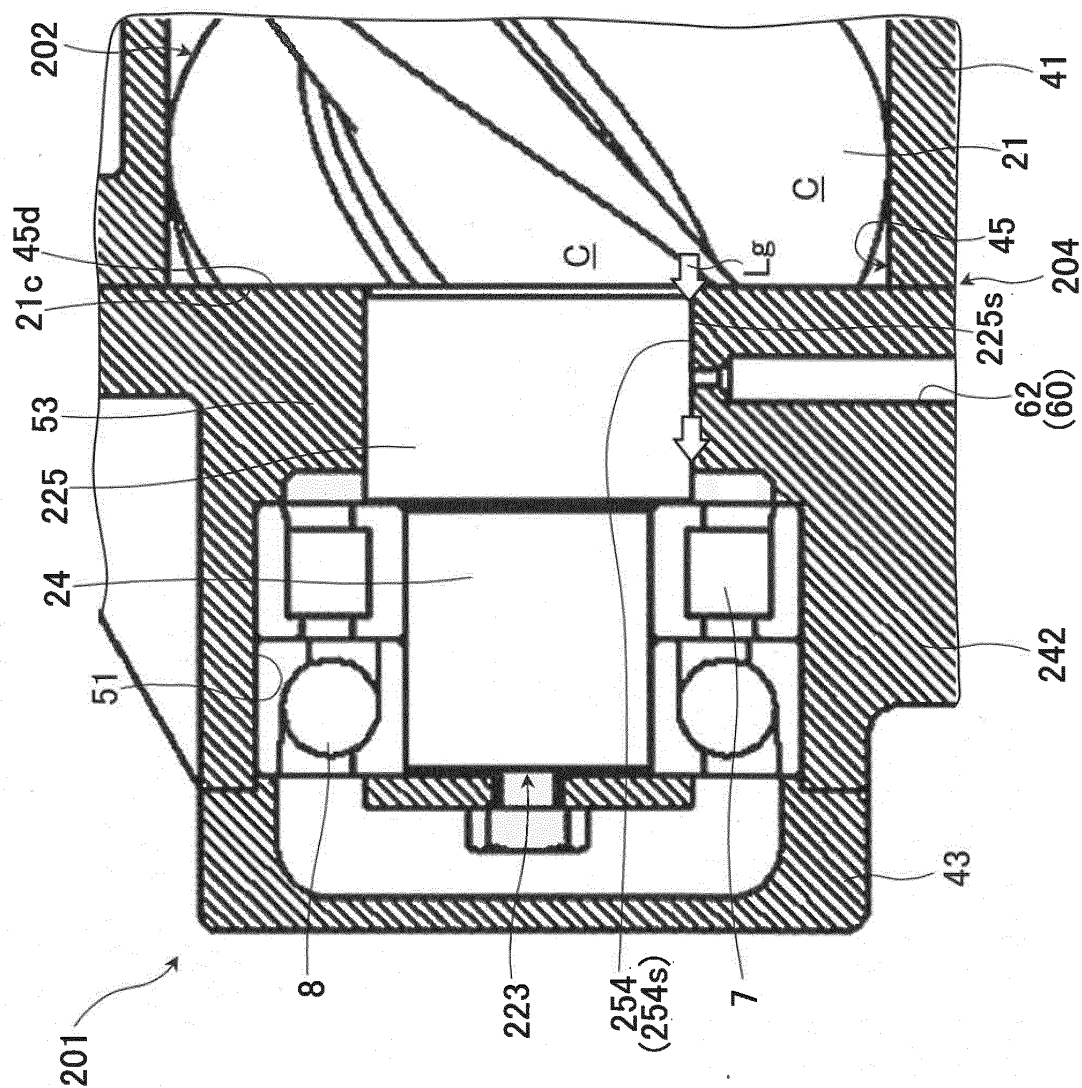


FIG. 4



F/G/5

COMPARATIVE EXAMPLE



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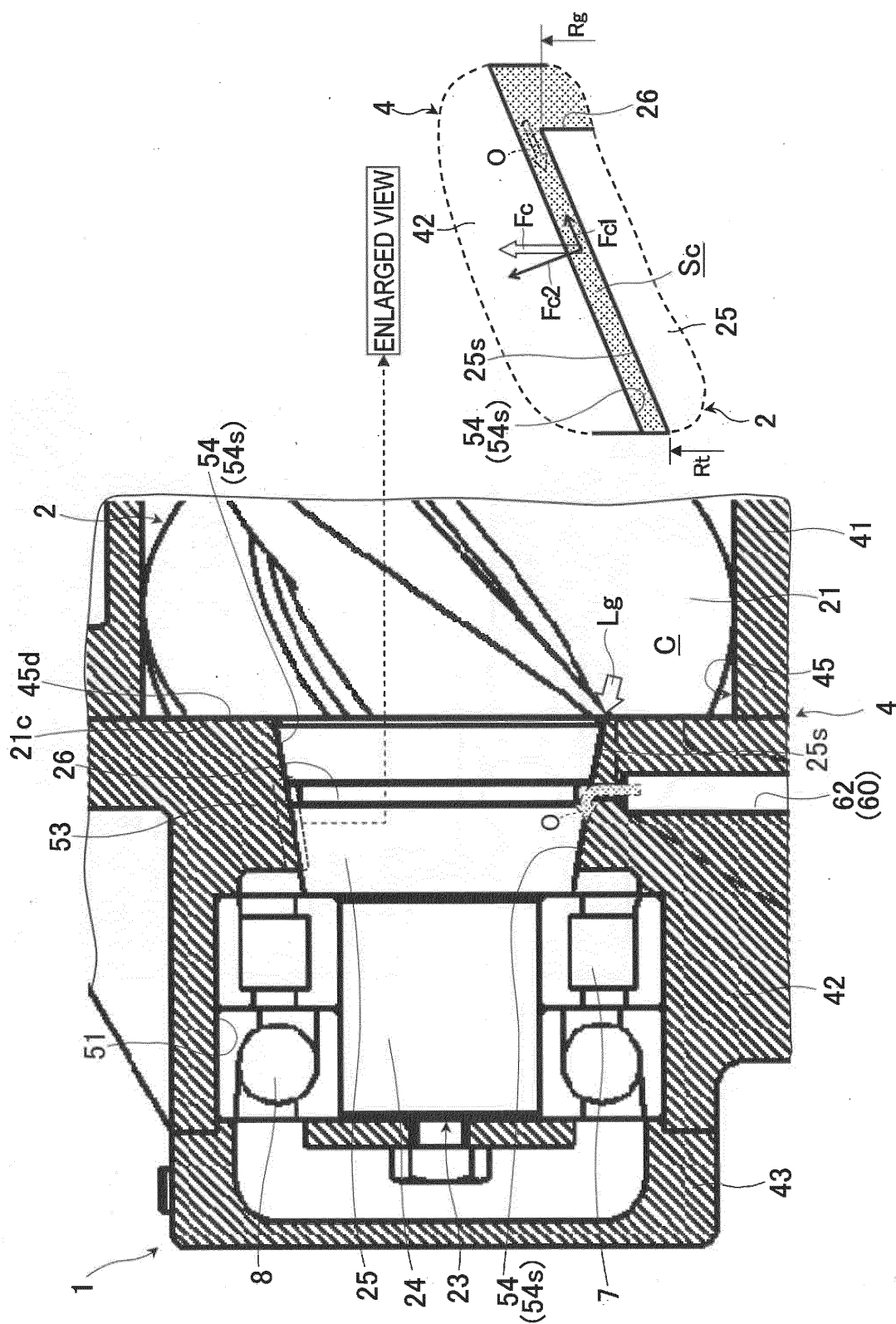
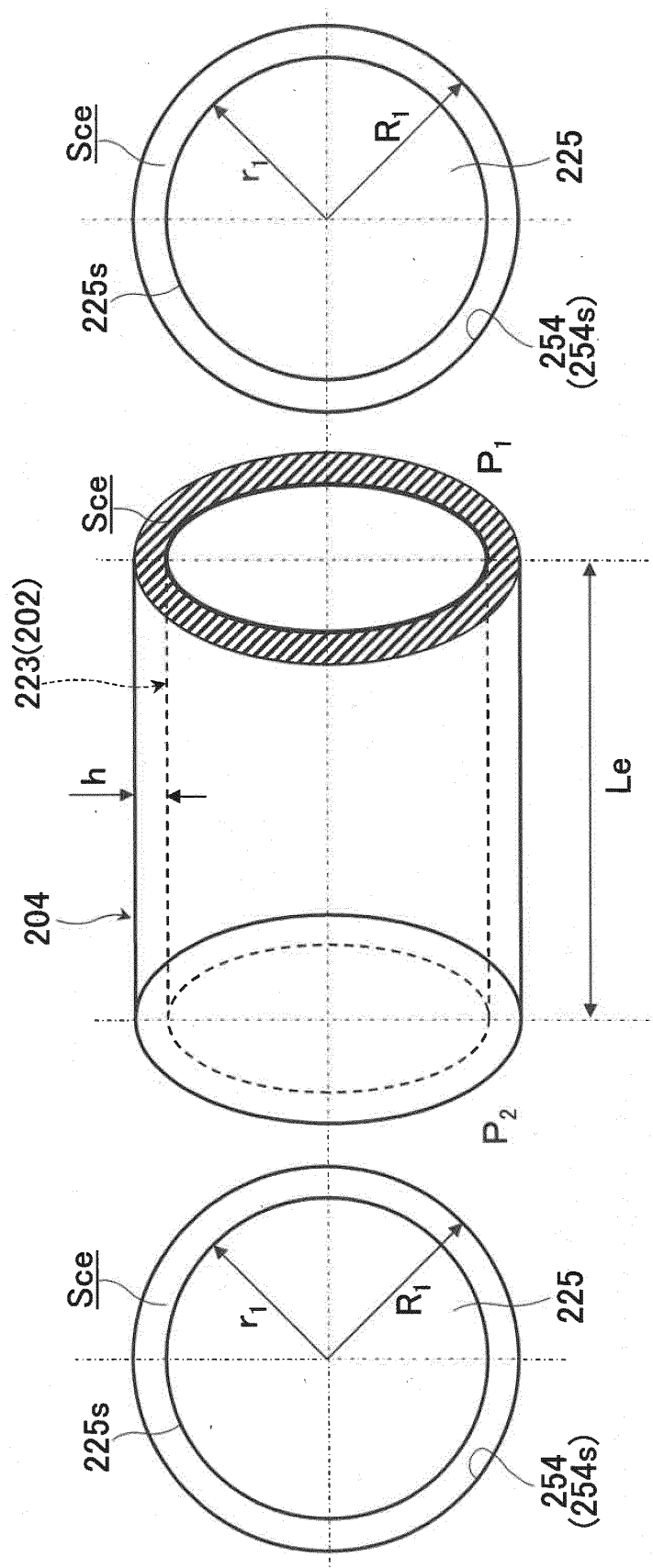
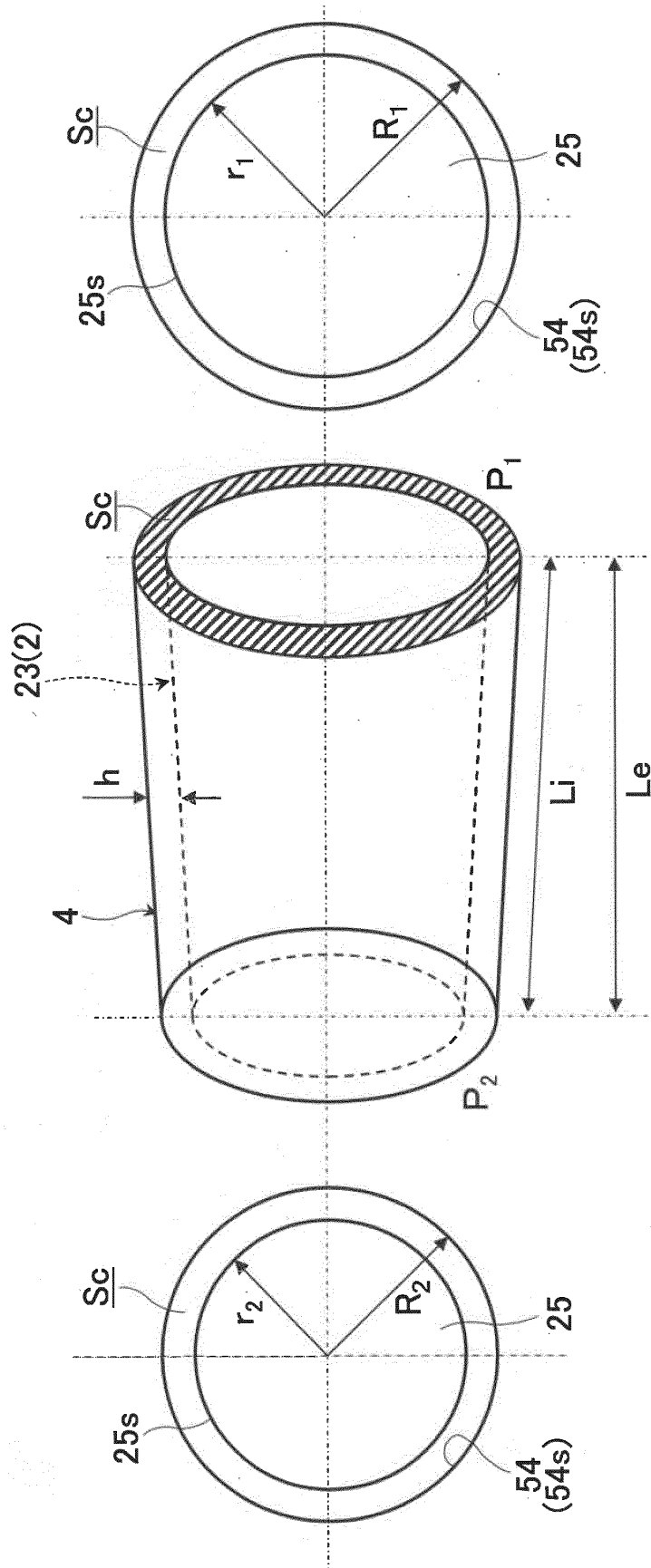


FIG. 7



COMPARATIVE EXAMPLE

FIG. 8



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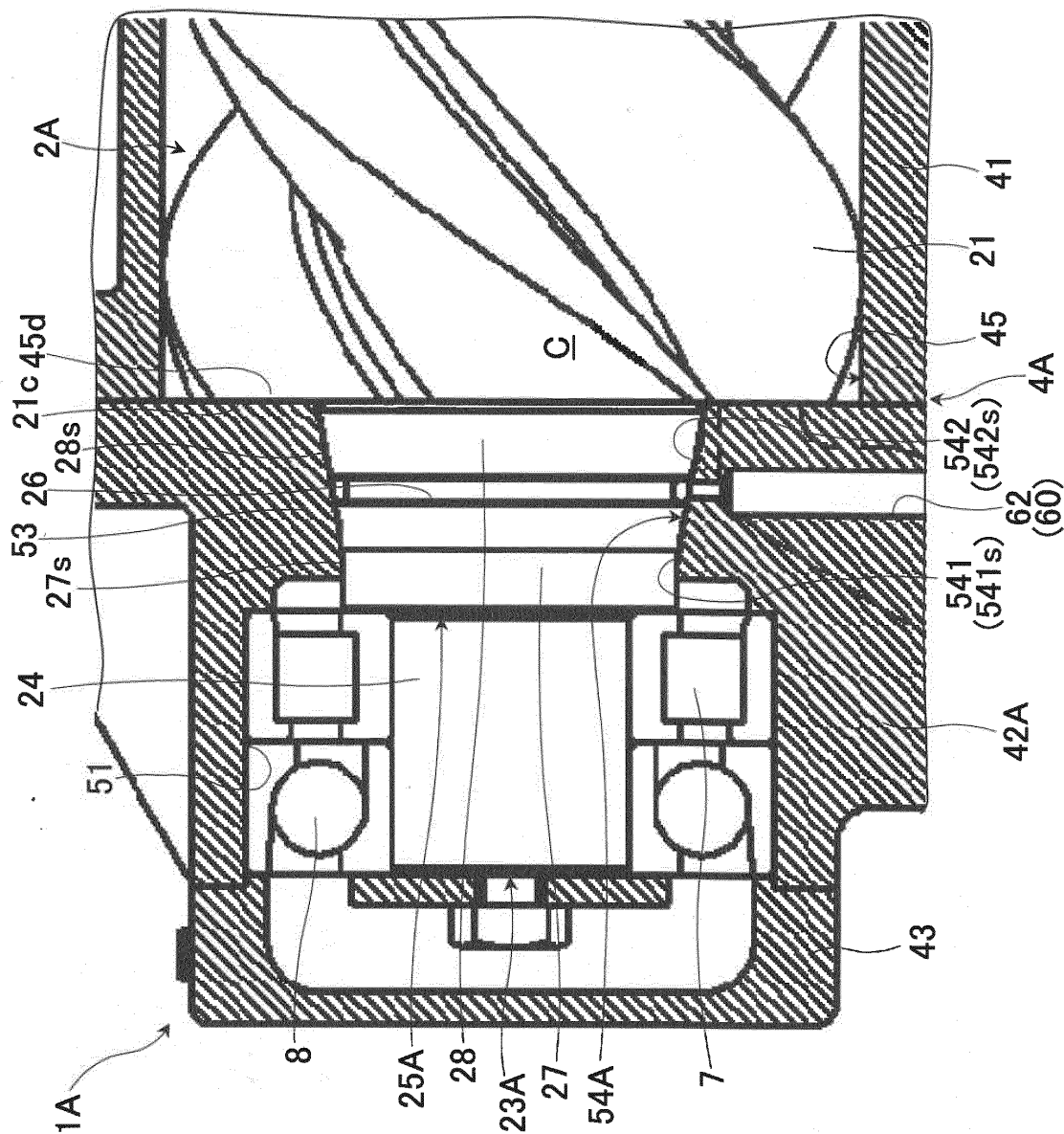


FIG. 10

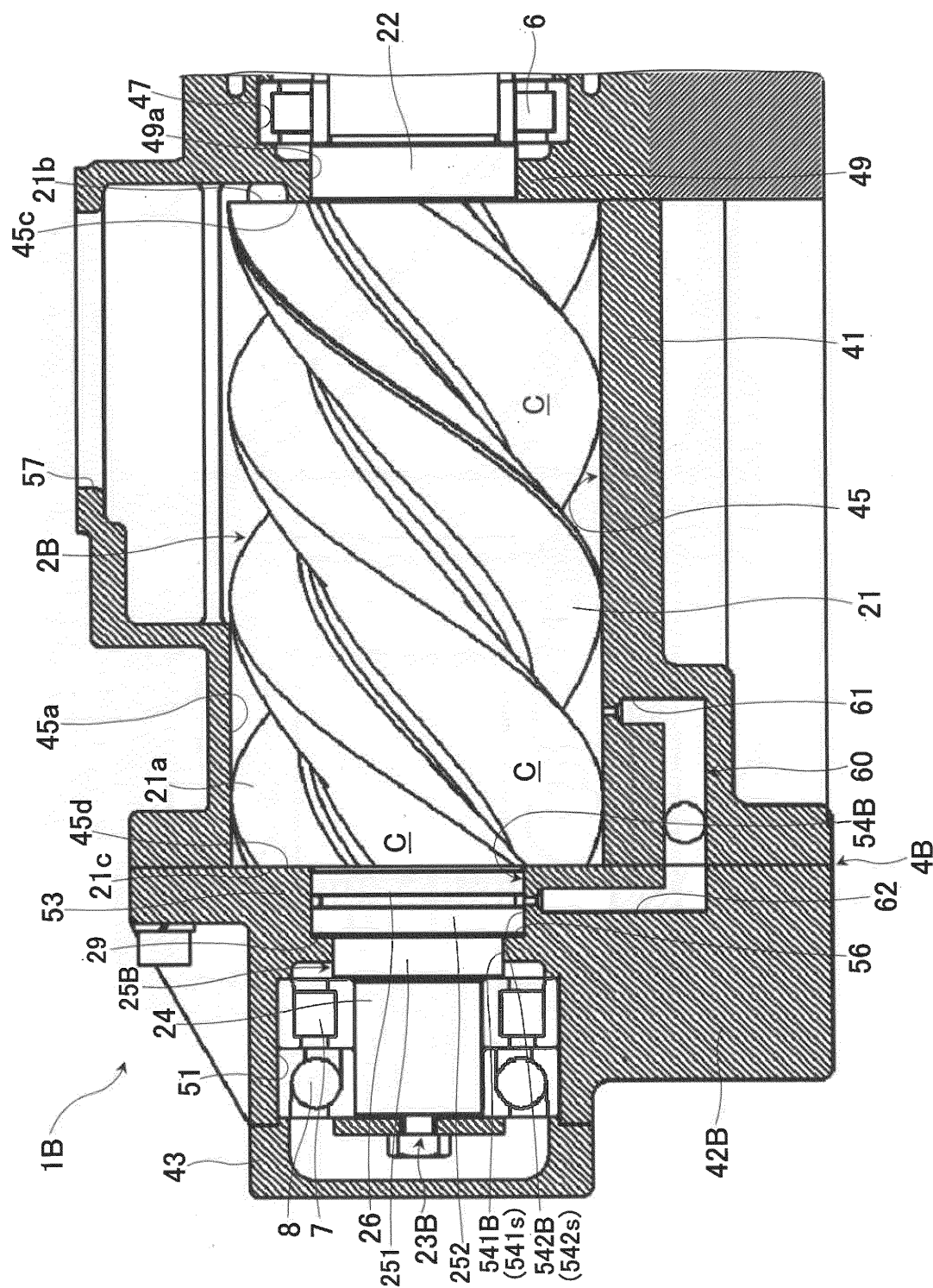


FIG. 11

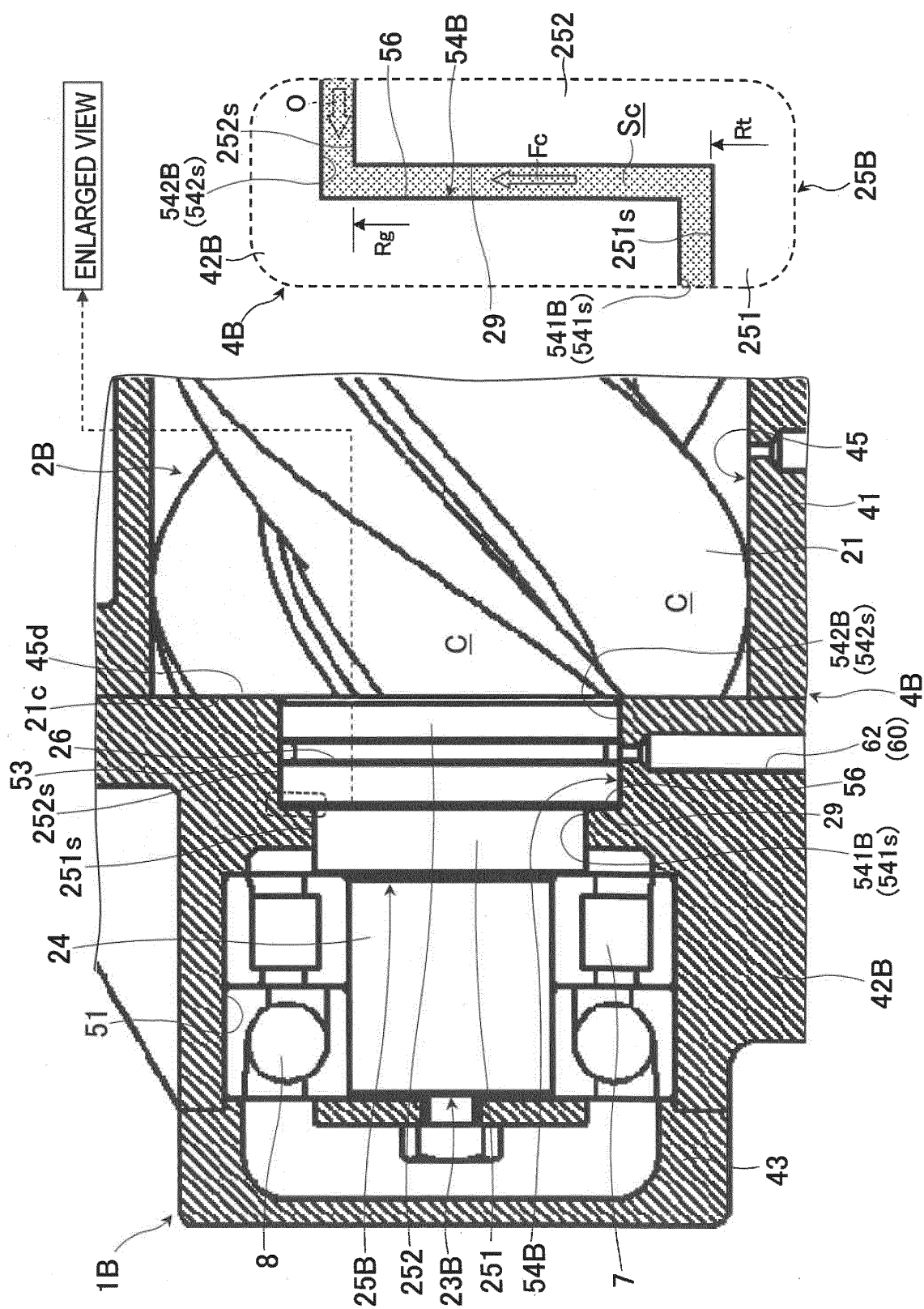
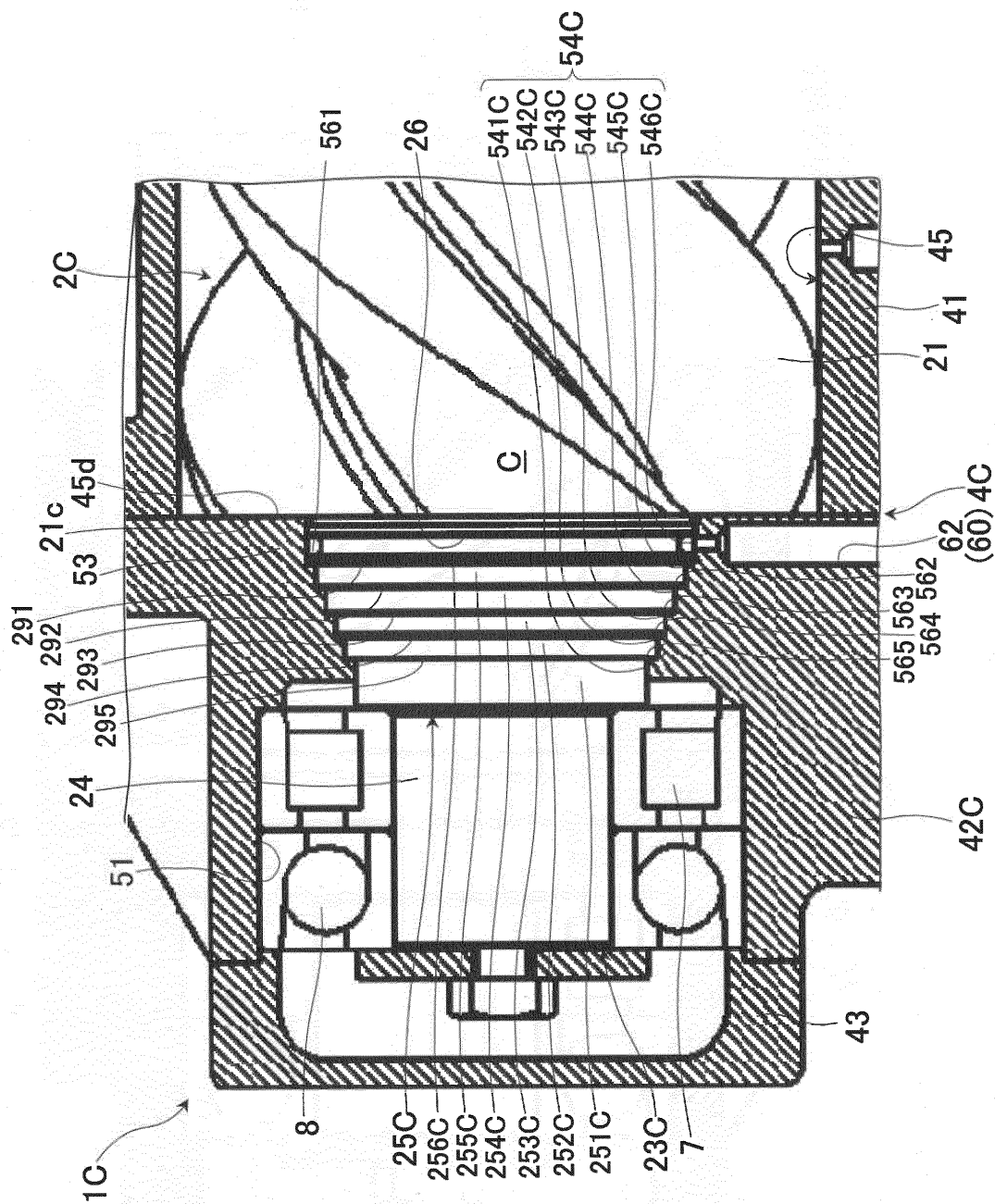


FIG. 12



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2022/036619

A. CLASSIFICATION OF SUBJECT MATTER F04C 18/16 (2006.01)i; F04C 27/00 (2006.01)i FI: F04C27/00 331; F04C18/16 Z According to International Patent Classification (IPC) or to both national classification and IPC															
B. FIELDS SEARCHED Minimum documentation searched (classification system followed by classification symbols) F04C18/16; F04C27/00; F16C17/00; F16J15/40 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 1971-2022 Registered utility model specifications of Japan 1996-2022 Published registered utility model applications of Japan 1994-2022 Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)															
C. DOCUMENTS CONSIDERED TO BE RELEVANT <table border="1"> <thead> <tr> <th>Category*</th> <th>Citation of document, with indication, where appropriate, of the relevant passages</th> <th>Relevant to claim No.</th> </tr> </thead> <tbody> <tr> <td>Y</td> <td>JP 2015-4306 A (KOBE STEEL, LTD.) 08 January 2015 (2015-01-08) paragraphs [0026]-[0045], fig. 1, 2</td> <td>1-8</td> </tr> <tr> <td>Y</td> <td>JP 7-55017 A (MITSUBISHI HEAVY IND., LTD.) 03 March 1995 (1995-03-03) paragraphs [0003]-[0011], fig. 1</td> <td>1-4</td> </tr> <tr> <td>Y</td> <td>JP 9-13906 A (MITSUBISHI HEAVY IND., LTD.) 14 January 1997 (1997-01-14) paragraphs [0009]-[0017], fig. 1</td> <td>1, 5-8</td> </tr> <tr> <td>Y</td> <td>JP 2010-242656 A (DAIKIN IND., LTD.) 28 October 2010 (2010-10-28) paragraphs [0022]-[0032], [0039], [0040], fig. 1, 5</td> <td>1-4</td> </tr> </tbody> </table>	Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.	Y	JP 2015-4306 A (KOBE STEEL, LTD.) 08 January 2015 (2015-01-08) paragraphs [0026]-[0045], fig. 1, 2	1-8	Y	JP 7-55017 A (MITSUBISHI HEAVY IND., LTD.) 03 March 1995 (1995-03-03) paragraphs [0003]-[0011], fig. 1	1-4	Y	JP 9-13906 A (MITSUBISHI HEAVY IND., LTD.) 14 January 1997 (1997-01-14) paragraphs [0009]-[0017], fig. 1	1, 5-8	Y	JP 2010-242656 A (DAIKIN IND., LTD.) 28 October 2010 (2010-10-28) paragraphs [0022]-[0032], [0039], [0040], fig. 1, 5	1-4
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Date of the actual completion of the international search 07 December 2022	Date of mailing of the international search report 20 December 2022														
Name and mailing address of the ISA/JP Japan Patent Office (ISA/JP) 3-4-3 Kasumigaseki, Chiyoda-ku, Tokyo 100-8915 Japan	Authorized officer Telephone No.														

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INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/JP2022/036619

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JP	7-55017	A	03 March 1995	(Family: none)	
JP	9-13906	A	14 January 1997	(Family: none)	
JP	2010-242656	A	28 October 2010	(Family: none)	

REFERENCES CITED IN THE DESCRIPTION

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