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(54) **VARIABLE CAPACITY VANE PUMP**

(57) A variable displacement vane pump 100 includes: a restrictor 37 configured to impart resistance to flow of working fluid discharged from the pump chambers 11; a control valve 27 configured to introduce the working fluid which is discharged from the pump chambers to the first fluid pressure chamber as a differential pressure between upstream and downstream of the restrictor is increased, the control valve being configured to discharge the working fluid in the first fluid pressure chamber as the differential pressure between upstream and downstream of the restrictor is reduced; a suction passage 22 configured to guide the working fluid to be sucked into the pump chambers 11, the suction passage 22 being configured to always communicate with the second fluid pressure chamber 17 ; and a guiding passage 36 configured to allow communication between the control valve 27 and the second fluid pressure chamber 17, the guiding passage 36 being configured to guide the working fluid, which is discharged from the first fluid pressure chamber 16 to the control valve 27, to the second fluid pressure chamber 17.

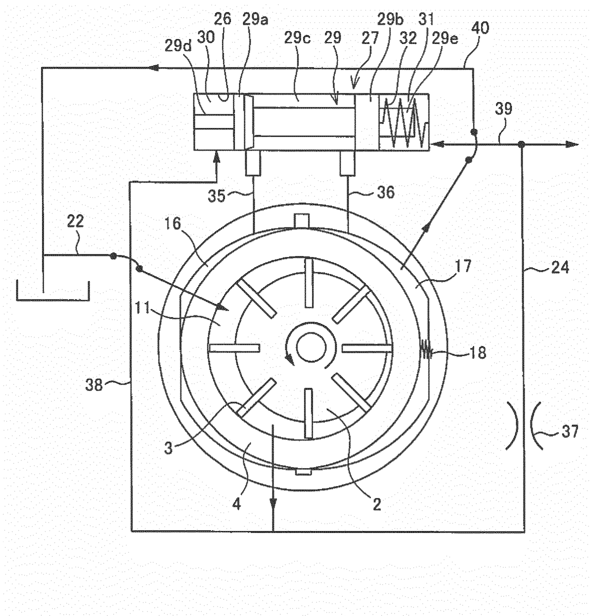

FIG.3

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Description

TECHNICAL FIELD

[0001] The present invention relates to a variable displacement vane pump used as a fluid pressure source.

BACKGROUND ART

[0002] JP2013-194692A describes a variable displacement vane pump that is capable of changing an amount of working fluid discharged by changing an amount of eccentricity of a cam ring with respect to a rotor.

[0003] In order to move the cam ring, this variable displacement vane pump includes a first fluid pressure chamber and a second fluid pressure chamber that are formed on the outer circumferential side of the cam ring; a metering orifice that is provided in a discharge passage; a control valve that introduces control pressure to the first fluid pressure chamber in accordance with movement of a spool that slides in accordance with a differential pressure between upstream and downstream of the metering orifice; and a cam spring that always biases the cam ring towards the first fluid pressure chamber side from the second fluid pressure chamber. The cam ring is provided so as to be movable between a maximum-eccentric position in which the amount of eccentricity is maximized when the cam ring is moved towards the first fluid pressure chamber side and a minimum-eccentric position in which the amount of eccentricity is minimized.

SUMMARY OF INVENTION

[0004] In the above-described conventional technique, while the control pressure is introduced to the first fluid pressure chamber from the control valve, suction pressure is always introduced to the second fluid pressure chamber. Therefore, when the cam ring is moved in the direction in which the amount of eccentricity is reduced, the cam ring is moved by the control pressure introduced to the first fluid pressure chamber. However, when the cam ring is moved in the direction in which the amount of eccentricity is increased, the cam ring is moved by biasing force exerted by the cam spring. Thus, there is a risk in that, when the cam ring is moved in the direction in which the amount of eccentricity is increased, the movement of the cam ring may be delayed, causing a follow-up delay.

[0005] The present invention has been conceived in light of such technical problems, and an object thereof is to provide a variable displacement vane pump capable of preventing the follow-up delay of a cam ring.

[0006] According to one aspect of the present invention, a variable displacement vane pump includes: a rotor that is linked to a driving shaft; a plurality of vanes provided so as to be movable in a reciprocating manner in the radial direction with respect to the rotor; a cam ring in which tip-end portions of the vanes are in sliding contact with a cam face on an inner circumference of the cam ring with rotation of the rotor arranged in the cam ring, the cam ring being capable of being made eccentric with respect to the rotor; pump chambers that are defined between the rotor and the cam ring by being partitioned by the plurality of vanes; a first fluid pressure chamber and a second fluid pressure chamber that are defined in an accommodating space on an outer circumferential side of the cam ring; a biasing member configured to always bias the cam ring in a direction in which an amount of eccentricity is increased; a restrictor configured to impart resistance to flow of working fluid discharged from the pump chambers; a control valve configured to reduce the amount of eccentricity of the cam ring by introducing the working fluid that has been discharged from the pump chambers to the first fluid pressure chamber as a differential pressure between upstream and downstream of the restrictor is increased, the control valve being configured to increase the amount of eccentricity of the cam ring by discharging the working fluid in the first fluid pressure chamber as the differential pressure between upstream and downstream of the restrictor is reduced; a suction passage configured to guide the working fluid to be sucked into the pump chambers, the suction passage being configured to always communicate with the second fluid pressure chamber; and a guiding passage configured to allow communication between the control valve and the second fluid pressure chamber, the guiding passage being configured to guide the working fluid, which is discharged from the first fluid pressure chamber to the control valve, to the second fluid pressure chamber.

BRIEF DESCRIPTION OF DRAWINGS

[0007]

[FIG. 1] FIG. 1 is a sectional view showing a cross section perpendicular to a driving shaft in a variable displacement vane pump according to a first embodiment of the present invention.

[FIG. 2] FIG. 2 is a sectional view showing a cross section parallel to the driving shaft in the variable displacement vane pump according to the first embodiment of the present invention.

[FIG. 3] FIG. 3 is a hydraulic circuit diagram of the variable displacement vane pump according to the first embodiment of the present invention.

[FIG. 4] FIG. 4 is a hydraulic circuit diagram of the variable displacement vane pump according to the first embodiment of the present invention and shows a state in which an amount of eccentricity of a cam ring with respect to a rotor is at a maximum level.

[FIG. 5] FIG. 5 is a hydraulic circuit diagram of the variable displacement vane pump according to the first embodiment of the present invention and shows a state in which the amount of eccentricity of the cam ring with respect to the rotor is at an intermediate level.

[FIG. 6] FIG. 6 is a hydraulic circuit diagram of the variable displacement vane pump according to the first embodiment of the present invention and shows a state in which the amount of eccentricity of the cam ring with respect to the rotor is at a minimum level.

[FIG. 7] FIG. 7 is a sectional view showing a cross section perpendicular to a driving shaft in a variable displacement vane pump according to a second embodiment of the present invention.

[FIG. 8] FIG. 8 is a sectional view showing a cross section parallel to the driving shaft in the variable displacement vane pump according to the second embodiment of the present invention.

DESCRIPTION OF EMBODIMENTS

[0008] The following describes embodiments of the present invention with reference to the drawings.

<First Embodiment>

[0009] A variable displacement vane pump 100 according to a first embodiment of the present invention will be described with reference to FIGs. 1 to 3.

[0010] The variable displacement vane pump 100 (hereinafter, simply referred to as "the vane pump 100") is used as a hydraulic pressure source for a hydraulic apparatus mounted on a vehicle, such as, for example, a power steering apparatus, a continuously variable transmission, and the like.

[0011] As shown in FIG. 1, in the vane pump 100, motive force from a driving source (not shown) is transmitted to a driving shaft 1, and a rotor 2 that is linked to the driving shaft 1 is rotated. In FIGs. 1 and 3, the rotor 2 is rotated in the counterclockwise direction as indicated by an arrow.

[0012] The vane pump 100 includes a plurality of vanes 3 that are provided so as to be movable in a reciprocating manner in the radial direction with respect to the rotor 2 and a cam ring 4 in which tip-end portions of the vanes 3 are in sliding contact with a cam face 4a, forming an inner circumference of the cam ring 4, by rotation of the rotor 2 arranged in the cam ring 4. The cam ring 4 is can be made eccentric with respect to the center of the rotor 2.

[0013] As shown in FIG. 2, the driving shaft 1 is rotatably supported by a pump body 6 via a bush 5. A pump accommodating recessed portion 6a serving as a recessed portion for accommodating the cam ring 4 is formed in the pump body 6. In an end portion of the pump body 6, a seal 7 is provided for preventing leakage of lubricating oil between an outer circumference of the driving shaft 1 and an inner circumference of the bush 5.

[0014] A side plate 8 that comes into contact with first side portions of the rotor 2 and the cam ring 4 is arranged on a bottom surface 6b of the pump accommodating recessed portion 6a. An opening portion of the pump accommodating recessed portion 6a is sealed with a pump cover 9 that comes into contact with second side portions of the rotor 2 and the cam ring 4. The pump cover 9 is fastened to the pump body 6 by bolts 10 (see FIG. 1).

[0015] As described above, the pump cover 9 and the side plate 8 are arranged so as to sandwich the rotor 2 and the cam ring 4 at both side surfaces thereof. With such a configuration, pump chambers 11 are defined between the rotor 2 and the cam ring 4 by being partitioned by the respective vanes 3.

[0016] As shown in FIGs. 1 and 3, the cam ring 4 is an annular member and has a suction region in which volumes of the pump chambers 11 partitioned by and between the respective vanes 3 are expanded by the rotation of the rotor 2 and a discharge region in which the volumes of the pump chambers 11 partitioned by and between the respective vanes 3 are contracted by the rotation of the rotor 2. The pump chambers 11 suck working oil serving as working fluid in the suction region and discharge the working oil in the discharge region. In FIG. 1, an upper part of the cam ring 4 corresponds to the suction region and a lower part corresponds to the discharge region.

[0017] An annular adapter ring 12 is fitted to an inner circumferential surface of the pump accommodating recessed portion 6a so as to surround the cam ring 4. The adapter ring 12 is sandwiched by the pump cover 9 and the side plate 8 at both side surfaces thereof in the same way as the rotor 2 and the cam ring 4.

[0018] A support plate 13 that extends in parallel with the driving shaft 1 is supported on an inner circumferential surface of the adapter ring 12. The cam ring 4 is supported by the support plate 13, and the cam ring 4 swings around inside the adapter ring 12 with the support plate 13 as a supporting point.

[0019] A groove 12a extending in parallel with the driving shaft 1 is formed at an axisymmetric position to the support plate 13 in the inner circumferential surface of the adapter ring 12. A seal member 14, which is in sliding contact with the outer circumferential surface of the cam ring 4 when the cam ring 4 swings around, is fitted in the groove 12a in a state in which an elastic member 15 is compressed.

[0020] As described above, in a space between the outer circumferential surface of the cam ring 4 and the inner circumferential surface of the adapter ring 12, which is an accommodating space on the outer circumference of the cam ring 4, a first hydraulic chamber 16 serving as a first fluid pressure chamber and a second hydraulic chamber 17 serving as a second fluid pressure chamber are defined by the support plate 13 and the seal member 14.

[0021] As shown in FIG. 1, a cam spring 18 serving as a biasing member is provided on the second hydraulic chamber 17 side of the outer circumferential surface of the cam ring 4. The cam spring 18 is fitted to a spring plug 19 that is screwed into the pump body 6 from the side and always biases the cam ring 4 towards the first hydraulic chamber 16 side via a through hole 12b formed in the adapter ring 12. In other words, the cam ring 4 is always biased by the cam spring 18 in the direction in which an amount of eccentricity is increased.

[0022] The cam ring 4 swings around with the support plate 13 as the supporting point in such a manner that a differential pressure of the working oil between the first hydraulic chamber 16 and the second hydraulic chamber 17, biasing force exerted by the cam spring 18, and the internal pressure of the cam ring 4 are balanced. As the cam ring 4 swings around with the support plate 13 as the supporting point, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is changed. As the amount of eccentricity of the cam ring 4 is changed, a pump displacement volume per rotation of the rotor 2 is changed.

[0023] When the pressure in the first hydraulic chamber 16 is increased, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is reduced. In this case, the pump displacement volume per rotation of the rotor 2 is reduced. In contrast, when the pressure in the first hydraulic chamber 16 is reduced, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is increased. In this case, the pump displacement volume per rotation of the rotor 2 is increased. As described above, in the vane pump 100, the pump displacement volume is changed in accordance with the amount of eccentricity of the cam ring 4 with respect to the rotor 2.

[0024] The pump cover 9 is provided with a suction port 20 having an arc-shaped opening so as to correspond to the suction region of the pump chambers 11. In addition, the side plate 8 is provided with a discharge port 21 having an arc-shaped opening so as to correspond to the discharge region of the pump chambers 11.

[0025] As shown in FIG. 2, the suction port 20 is formed so as to communicate with a suction passage 22 formed in the pump cover 9 and guides the working oil in the suction passage 22 to the suction region of the pump chambers 11. The discharge port 21 is formed so as to communicate with a high-pressure chamber 23 formed in the pump body 6 and guides the working oil discharged from the discharge region of the pump chambers 11 to the high-pressure chamber 23.

[0026] The high-pressure chamber 23 is defined by closing a groove portion 6c, which is formed so as to open at the bottom surface 6b of the pump accommodating recessed portion 6a, with the side plate 8. The working oil in the high-pressure chamber 23 is guided to an external hydraulic apparatus of the vane pump 100 through a discharge passage 24 (see FIG. 3) formed in the pump body 6.

[0027] The pump body 6 is provided with a low-pressure chamber 25, serving as a first guiding passage, that is formed at a position corresponding to the suction region of the pump chambers 11 on the bottom surface 6b of the pump accommodating recessed portion 6a. The low-pressure chamber 25 is defined by closing a groove portion 6d, which is formed so as to open at a position corresponding to the suction region of the pump chambers 11, with the side plate 8. The low-pressure chamber 25 is formed in a straight line parallel to the driving shaft 1, and its back-most end portion communicates with a boundary between the bush 5 and the seal 7. The low-pressure chamber 25 is always connected to the second hydraulic chamber 17, and the working oil that has leaked out between the outer circumference of the driving shaft 1 and the inner circumference of the bush 5 is recovered and returned to the pump chambers 11 in the suction region.

[0028] As shown in FIGs. 1 and 2, the pump body 6 is provided with a valve accommodating hole 26 that is formed in the direction perpendicular to the axial direction of the driving shaft 1. In the valve accommodating hole 26, a control valve 27 that controls the working oil pressures in the first hydraulic chamber 16 and the second hydraulic chamber 17 is accommodated. The valve accommodating hole 26 is sealed by a plug 28.

[0029] The control valve 27 includes a spool 29 that is slidably inserted into the valve accommodating hole 26, a first pilot chamber 30 that faces one end of the spool 29, a second pilot chamber 31 that faces the other end of the spool 29, and a return spring 32 that is accommodated in the second pilot chamber 31 and biases the spool 29 in the direction in which the volume of the second pilot chamber 31 is expanded.

[0030] The spool 29 includes a first land portion 29a and a second land portion 29b that slide along an inner circumferential surface of the valve accommodating hole 26, an annular groove 29c that is formed between the first land portion 29a and the second land portion 29b, a first rod portion 29d that is connected to the first land portion 29a and extends within the first pilot chamber 30, and a second rod portion 29e that is connected to the second land portion 29b and extends within the second pilot chamber 31.

[0031] The first rod portion 29d comes into contact with the plug 28 when the spool 29 is moved in the direction in which the volume of the first pilot chamber 30 is contracted. When the spool 29 is moved in the direction in which the volume of the second pilot chamber 31 is contracted, the second rod portion 29e comes into contact with an end surface of the valve accommodating hole 26 on the opposite side from the plug 28. The return spring 32 surrounds the second rod portion 29e and is received in the second pilot chamber 31.

[0032] As shown in FIG. 3, a first passage 35 and a second passage 36, which serves as a guiding passage, that communicate with the first hydraulic chamber 16 and the second hydraulic chamber 17, respectively; a first pressure guiding passage 38 that guides to the first pilot chamber 30 the working oil that has been discharged from the high-pressure chamber 23 to the upstream side of an orifice 37 serving as a restrictor; and a second pressure guiding passage 39 that guides to the second pilot chamber 31 the working oil that has been discharged from the high-pressure chamber 23 to the downstream side of the orifice 37 are connected to the control valve 27. A drain passage 40 that is always in communication with the suction passage 22 is connected to the second hydraulic chamber 17.

[0033] The first passage 35 and the second passage 36 are formed so as to open at the valve accommodating hole 26 and to open at the first hydraulic chamber 16 and the second hydraulic chamber 17, respectively, by penetrating through the adapter ring 12.

[0034] The spool 29 slides to a position at which the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31, which face the respective ends of the spool 29, is balanced with the biasing force exerted by the return spring 32. The first passage 35 is opened/closed by the first land portion 29a, and the working oil in the first hydraulic chamber 16 is supplied/discharged depending on the position of the spool 29. The second passage 36 always opens to the annular groove 29c regardless of the position of the spool 29.

[0035] When the biasing force exerted by the return spring 32 is greater than the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31, a state in which the return spring 32 is elongated is achieved. In this state, as shown in FIGs. 1 and 3, the first passage 35 and the second passage 36 open at the annular groove 29c. With such a configuration, the communication between the first hydraulic chamber 16 and the first pilot chamber 30 is shut off.

[0036] Here, a state in which the first hydraulic chamber 16 communicates with the drain passage 40 through the first passage 35, the annular groove 29c, the second passage 36, and the second hydraulic chamber 17 is achieved. Because the cam ring 4 is always biased by the cam spring 18 in the direction in which the amount of eccentricity is increased, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is maximized.

[0037] In contrast, when the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31 is greater than the biasing force exerted by the return spring 32, the spool 29 is moved against the biasing force exerted by the return spring 32. In this case, the first passage 35 is shifted into an open state, communicates with the first pilot chamber 30, and communicates with the first pressure guiding passage 38 through the first pilot chamber 30. In addition, the second passage 36 is held in the open state and communicates with the annular groove 29c. With such a configuration, the first hydraulic chamber 16 communicates with the high-pressure chamber 23. Because the second hydraulic chamber 17 communicates with the suction passage 22 through the drain passage 40, as the pressure in the first hydraulic chamber 16 is increased, the amount of eccentricity of the cam ring 4 is reduced. In other words, when the pressure in the first hydraulic chamber 16 is increased and the force received by the cam ring 4 from the first hydraulic chamber 16 exceeds the sum of the force received by the cam ring 4 from the cam spring 18 and the force received by the cam ring 4 from the internal pressure of the cam ring 4, the cam ring 4 is moved in the direction in which the amount of eccentricity with respect to the rotor 2 is reduced.

[0038] As described above, when the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31 exceeds the biasing force exerted by the return spring 32, the spool 29 of the control valve 27 is moved so as to compress the return spring 32.

[0039] The working oil at the upstream side and the downstream side of the orifice 37 serving as the restrictor, which is interposed in the discharge passage 24 and imparts resistance to the flow of the working oil, is respectively guided to the first pilot chamber 30 and the second pilot chamber 31. In other words, the working oil in the high-pressure chamber 23 is guided directly to the first pilot chamber 30 through the first pressure guiding passage 38 without passing through the orifice 37, and is also guided to the second pilot chamber 31 through the orifice 37. Therefore, the spool 29 is moved in accordance with the differential pressure between upstream and downstream of the orifice 37.

[0040] Next, operation of the vane pump 100 will be described with reference to FIGs. 4 to 6. FIGs. 4 to 6 are hydraulic circuit diagrams of the vane pump 100 and respectively show states in which the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is at maximum, intermediate, and minimum levels.

[0041] As the rotor 2 is rotated by motive force transmitted from the driving source to the driving shaft 1, the working oil is sucked from the suction passage 22 through the suction port 20 into the pump chambers 11 whose spaces are expanded between the respective vanes 3 with the rotation of the rotor 2. In addition, the working oil is discharged through the discharge port 21 to the high-pressure chamber 23 from the pump chambers 11 whose spaces are contracted between the respective vanes 3. The working oil that has been discharged to the high-pressure chamber 23 is supplied to the hydraulic apparatus through the discharge passage 24.

[0042] When the working oil passes through the discharge passage 24, the differential pressure is generated between upstream and downstream of the orifice 37, which is interposed in the discharge passage 24, and the pressures at the upstream and downstream sides of the orifice 37 are guided to the first pilot chamber 30 and the second pilot chamber 31, respectively. The spool 29 of the control valve 27 slides to the position at which the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31 is balanced with the biasing force exerted by the return spring 32.

[0043] Because the rotation speed of the rotor 2 is low and a pump discharge flow amount is small at a pump starting time at which the rotation speed of the rotor 2 is equal to or lower than a predetermined rotation speed, the differential pressure between upstream and downstream of the orifice 37 is small, and the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31 is small. Therefore, the biasing force exerted by the return spring 32 is greater than the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31, and the return spring 32 is in an elongated state.

[0044] In this case, as shown in FIG. 4, because the first passage 35 and the second passage 36 open at the annular groove 29c, the first hydraulic chamber 16 communicates with the drain passage 40 through the annular groove 29c and the second hydraulic chamber 17. In this state, because the hydraulic pressure that makes the cam ring 4 swing around does not act on the first hydraulic chamber 16 and the second hydraulic chamber 17, the cam ring 4 is biased by the cam spring 18 in the direction in which the amount of eccentricity with respect to the rotor 2 is increased. With such a configuration, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is maximized.

[0045] In a region in which the rotation speed of the rotor 2 is equal to or lower than the predetermined rotation speed, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is maximized to cause the pump displacement volume per rotation of the rotor 2 to be maximized, and the pump discharge flow amount of the vane pump 100 becomes the flow amount substantially in proportion to the rotation speed of the rotor 2. Therefore, even when the rotation speed of the rotor 2 is low, it is possible to supply the working oil to the hydraulic apparatus at a sufficient flow amount.

[0046] As the rotation speed of the rotor 2 is increased, the differential pressure between upstream and downstream of the orifice 37 is increased, and thereby, the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31 is balanced with or becomes slightly greater than the biasing force exerted by the return spring 32. With such a configuration, the spool 29 starts to move against the biasing force exerted by the return spring 32.

[0047] Furthermore, when the rotation speed of the rotor 2 is increased and reaches the predetermined rotation speed, as shown in FIG. 5, by the movement of the spool 29, the first passage 35 is shifted into the open state and communicates with the first pilot chamber 30 and the annular groove 29c, and the second passage 36 is held in the open state. With such a configuration, because the first hydraulic chamber 16 communicates with the high-pressure chamber 23 and the second hydraulic chamber 17 communicates with the drain passage 40, as the pressure in the first hydraulic chamber 16 is increased, the cam ring 4 starts to move in the direction in which the amount of eccentricity with respect to the rotor 2 is reduced.

[0048] In a region in which the rotation speed of the rotor 2 exceeds the predetermined rotation speed, the pump discharge flow amount of the vane pump 100 becomes substantially constant. In other words, when the first passage 35 and the second passage 36 are shifted into the open state and the cam ring 4 starts to move in the direction in which the amount of eccentricity with respect to the rotor 2 is reduced, the pump discharge flow amount is reduced and the differential pressure between upstream and downstream of the orifice 37 is reduced. With such a configuration, the return spring 32 is elongated, and the first passage 35 is closed again. When the first passage 35 is closed, the cam ring 4 is moved in the direction in which the amount of eccentricity with respect to the rotor 2 is increased and the pump discharge flow amount is increased. When the pump discharge flow amount is increased, the differential pressure between upstream and downstream of the orifice 37 is increased, and the spool 29 is moved so as to compress the return spring 32, and thereby, the first passage 35 and the second passage 36 are again shifted into the open state. As described above, because a control is performed such that the first passage 35 is opened/closed to make the differential pressure between upstream and downstream of the orifice 37 constant, the pump discharge flow amount becomes substantially constant.

[0049] In a region in which the rotation speed of the rotor 2 exceeds the predetermined rotation speed, as the rotation speed of the rotor 2 is increased, because the amount of movement of the spool 29 while compressing the return spring 32 is increased and an opening degree of the first passage 35 is increased, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is reduced gradually, causing a gradual reduction in the pump displacement volume per rotation of the rotor 2.

[0050] When the rotation speed of the rotor 2 is further increased, as shown in FIG. 6, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is minimized, and the pump displacement volume per rotation of the rotor 2 is minimized.

[0051] Even in a state shown in FIG. 6 in which the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is minimized, because the amount of eccentricity does not become zero, the vane pump 100 discharges the working oil at the minimum discharge capacity.

[0052] As described above, the spool 29 is moved in accordance with the change in the rotation speed of the rotor 2 and the first passage 35 is opened/closed by the movement of the spool 29, and thereby, the pump discharge flow amount is adjusted. More specifically, at the pump starting time at which the rotation speed of the rotor 2 is equal to or lower than the predetermined rotation speed, because the first passage 35 is closed by the spool 29, the amount of eccentricity of the cam ring 4 with respect to the rotor 2 is maximized, and the pump discharge flow amount is increased along with the increase in the rotation speed of the rotor 2. In addition, when the rotation speed of the rotor 2 exceeds the predetermined rotation speed, because a control is performed such that the opening degree of the first passage

35 is adjusted by the movement of the spool 29 and the differential pressure between upstream and downstream of the orifice 37 becomes constant, the pump discharge flow amount becomes substantially constant.

[0053] Here, when the rotation speed of the rotor 2 is reduced from the region in which the rotation speed of the rotor 2 is greater than the predetermined rotation speed, the thrust force exerted by the differential pressure between the first pilot chamber 30 and the second pilot chamber 31 is reduced, and the spool 29 slides in the direction in which the return spring 32 is elongated. When the communication between the first passage 35 and the first pilot chamber 30 is shut off by the slide of the spool 29, the high-pressure working oil that has been guided to the first hydraulic chamber 16 is discharged to the annular groove 29c, and then, supplied to the second hydraulic chamber 17 through the second passage 36. The working oil in the second hydraulic chamber 17 is subsequently returned to the suction passage 22 through the drain passage 40 (see FIGs. 4 and 5).

[0054] With such a configuration, when the amount of eccentricity of the cam ring 4 is increased as the rotation speed of the rotor 2 is reduced, the cam ring 4 receives the force exerted, in the direction in which the amount of eccentricity is increased, by the working oil pressure that has been guided from the first hydraulic chamber 16 to the second hydraulic chamber 17 through the annular groove 29c.

[0055] Because the working oil pressure that has been guided to the second hydraulic chamber 17 is greater than the working oil pressure in the suction passage 22 that always communicates with the second hydraulic chamber 17 through the drain passage 40, it is possible to make the cam ring 4 eccentric with higher responsiveness compared to a case in which the amount of eccentricity of the cam ring 4 is increased only by the biasing force exerted by the cam spring 18 and the force exerted by the internal pressure of the cam ring 4. Thus, it is possible to prevent a follow-up delay of the cam ring 4 when the rotation speed of the rotor 2 is reduced.

[0056] With the above-mentioned first embodiment, the following effects can be afforded.

[0057] When the working oil in the first hydraulic chamber 16 is discharged to increase the amount of eccentricity of the cam ring 4 as the differential pressure between upstream and downstream of the orifice 37 is reduced, the working oil that has been discharged from the first hydraulic chamber 16 to the annular groove 29c is guided to the second hydraulic chamber 17 through the second passage 36.

[0058] With such a configuration, when the rotation speed of the rotor 2 is reduced and the amount of eccentricity of the cam ring 4 is increased, in addition to the biasing force exerted by the cam spring 18, the force exerted by the working oil pressure in the second hydraulic chamber 17 that has been guided from the first hydraulic chamber 16 through the annular groove 29c acts on the cam ring 4. Therefore, it is possible to prevent the follow-up delay of the cam ring 4.

[0059] Furthermore, because the second passage 36 opens at the valve accommodating hole 26 and opens at the inner circumferential surface of the adapter ring 12 in the second hydraulic chamber 17 by penetrating through the adapter ring 12, it is possible to shorten a distance between the control valve 27, which is arranged radially outside of the adapter ring 12 so as to be adjacent to the adapter ring 12, and the second hydraulic chamber 17.

[0060] With such a configuration, when the rotation speed of the rotor 2 is reduced and the amount of eccentricity of the cam ring 4 is increased, it is possible to reduce the time required for the working oil pressure, which has been discharged to the annular groove 29c from the first hydraulic chamber 16, to be supplied to the second hydraulic chamber 17. Thus, it is possible to improve a startup of the working oil pressure in the second hydraulic chamber 17 that biases the cam ring 4 in the direction in which the amount of eccentricity is increased and prevent the follow-up delay of the cam ring 4 more reliably.

[0061] <Second Embodiment>

[0062] A variable displacement vane pump 200 according to a second embodiment of the present invention will be described with reference to FIGs. 7 and 8.

[0063] The variable displacement vane pump 200 in this embodiment differs from that in the first embodiment in a configuration of a second passage 136, and other points are the same as those in the first embodiment. Therefore, components that are the same as those in the first embodiment are assigned the same reference signs, and descriptions thereof shall be omitted.

[0064] The second passage 36 is formed so as to open at the valve accommodating hole 26 and to open at the second hydraulic chamber 17 by penetrating through the adapter ring 12 in the first embodiment, whereas in this embodiment, the second passage 136 serving as a guiding passage is constituted of the low-pressure chamber 25 and a straight passage 101, which serves as a second guiding passage that connects the back-anost end portion of the low-pressure chamber 25 and the annular groove 29c of the control valve 27 in a straight line.

[0065] With such a configuration, the working oil that has been discharged from the first hydraulic chamber 16 to the annular groove 29c of the control valve 27 is guided to the second hydraulic chamber 17 through the straight passage 101 and the low-pressure chamber 25.

[0066] With the above-mentioned second embodiment, the following effects can be afforded.

[0067] Because the second passage 136 opens at the bottom surface 6b of the pump accommodating recessed portion 6a in the suction region in which the volumes of the pump chambers 11 are expanded, a through hole needs not be provided in the adapter ring 12, which defines an accommodating space on the outer circumferential side of the cam ring 4. Thus, there is no need to provide the through hole in the adapter ring 12, and in addition to that, there is no need to perform alignment of the through hole of the adapter ring 12 and a hole formed in the pump body 6 so as to communicate with the annular groove 29c of the control valve 27. Therefore, it is possible to prevent the follow-up delay of the cam ring 4 while reducing the manufacturing cost.

[0068] Furthermore, because the second passage 136 is constituted of the low-pressure chamber 25 that is formed in a straight line parallel to the driving shaft 1 and the straight passage 101 that connects the back-most end portion of the low-pressure chamber 25 and the annular groove 29c of the control valve 27 in a straight line, it is possible to form the second passage 136 in the pump body 6 only by providing two straight passages. Therefore, it is possible to improve the ease of processing for providing the second passage 136 and to reduce the manufacturing cost.

[0069] Furthermore, because a part of the second passage 136 is constituted of the low-pressure chamber 25, it is possible to form the second passage 136 only by providing the straight passage 101. Therefore, it is possible to further improve the ease of processing for providing the second passage 136 and to further reduce the manufacturing cost.

[0070] Embodiments of this invention were described above, but the above embodiments are merely examples of applications of this invention, and the technical scope of this invention is not limited to the specific constitutions of the above embodiments.

[0071] For example, in the above-mentioned embodiment, although a case in which the working oil is used as the working fluid has been described, other fluids than the working oil, such as water, aqueous alternative fluid, and so forth, may be used.

[0072] Furthermore, in the above-mentioned embodiment, although a case in which the low-pressure chamber 25 and the straight passage 101 are both formed in a straight line is described, the configuration is not limited thereto, and at least one of the low-pressure chamber 25 and the straight passage 101 may be formed to have a curved shape or a shape having a bent portion at an intermediate position.

[0073] This application claims priority based on Japanese Patent Application No.2014-239200 filed with the Japan Patent Office on November 26, 2014, the entire contents of which are incorporated into this specification.

Claims

1. A variable displacement vane pump comprising:

a rotor linked to a driving shaft;
a plurality of vanes provided so as to be movable in a reciprocating manner in the radial direction with respect to the rotor;
a cam ring in which tip-end portions of the vanes are in sliding contact with a cam face on an inner circumference of the cam ring with rotation of the rotor arranged in the cam ring, the cam ring being capable of being made eccentric with respect to the rotor;
pump chambers defined between the rotor and the cam ring by being partitioned by the plurality of vanes;
a first fluid pressure chamber and a second fluid pressure chamber defined in an accommodating space on an outer circumferential side of the cam ring;
a biasing member configured to always bias the cam ring in a direction in which an amount of eccentricity is increased;
a restrictor configured to impart resistance to flow of working fluid discharged from the pump chambers;
a control valve configured to reduce the amount of eccentricity of the cam ring by introducing the working fluid that has been discharged from the pump chambers to the first fluid pressure chamber as a differential pressure between upstream and downstream of the restrictor is increased, the control valve being configured to increase the amount of eccentricity of the cam ring by discharging the working fluid in the first fluid pressure chamber as the differential pressure between upstream and downstream of the restrictor is reduced;
a suction passage configured to guide the working fluid to be sucked into the pump chambers, the suction passage being configured to always communicate with the second fluid pressure chamber; and
a guiding passage configured to allow communication between the control valve and the second fluid pressure chamber, the guiding passage being configured to guide the working fluid, which is discharged from the first fluid pressure chamber to the control valve, to the second fluid pressure chamber.

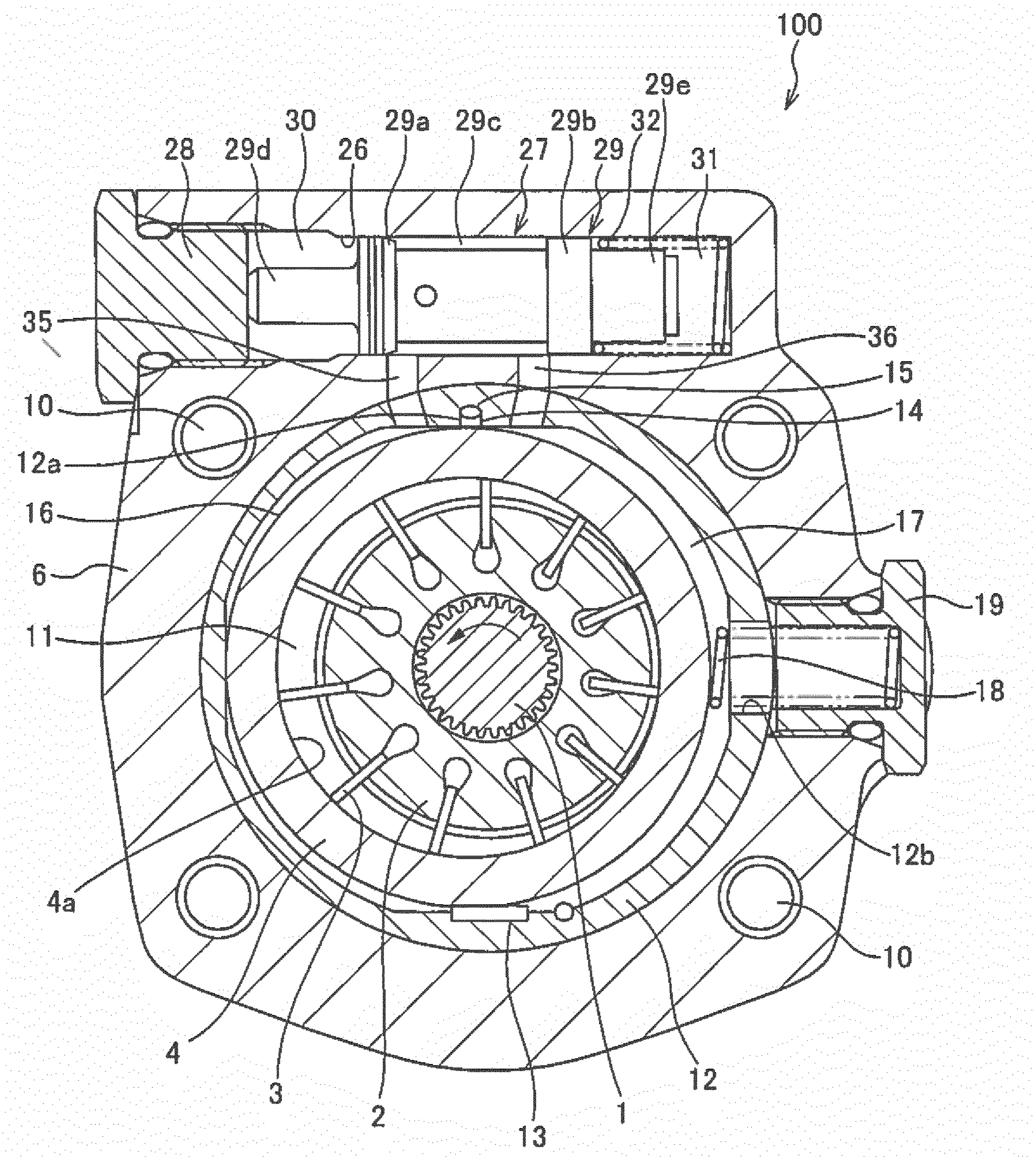
2. The variable displacement vane pump according to claim 1, further comprising:

an adapter ring formed in an annular shape so as to surround the cam ring; and
a pump body that has a recessed portion for accommodating the adapter ring, wherein
the control valve is arranged radially outside of the adapter ring so as to be adjacent to the adapter ring, and
the guiding passage opens at an inner circumferential surface of the adapter ring in the second fluid pressure chamber.

3. The variable displacement vane pump according to claim 1, further comprising
a pump body that has a recessed portion for accom-

modating the cam ring and that has the guiding passage formed therein, wherein
the guiding passage opens at a bottom surface of the recessed portion in a suction region in which volumes of the pump chambers are expanded.

4. The variable displacement vane pump according to claim 3, wherein
the guiding passage has a first guiding passage and a second guiding passage, the first guiding passage opening at the bottom surface of the recessed portion, the first guiding passage being formed in a straight line parallel to the driving shaft, the second guiding passage being configured to connect the first guiding passage and the control valve in a straight line.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12b
4a
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3
2
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12

FIG.1

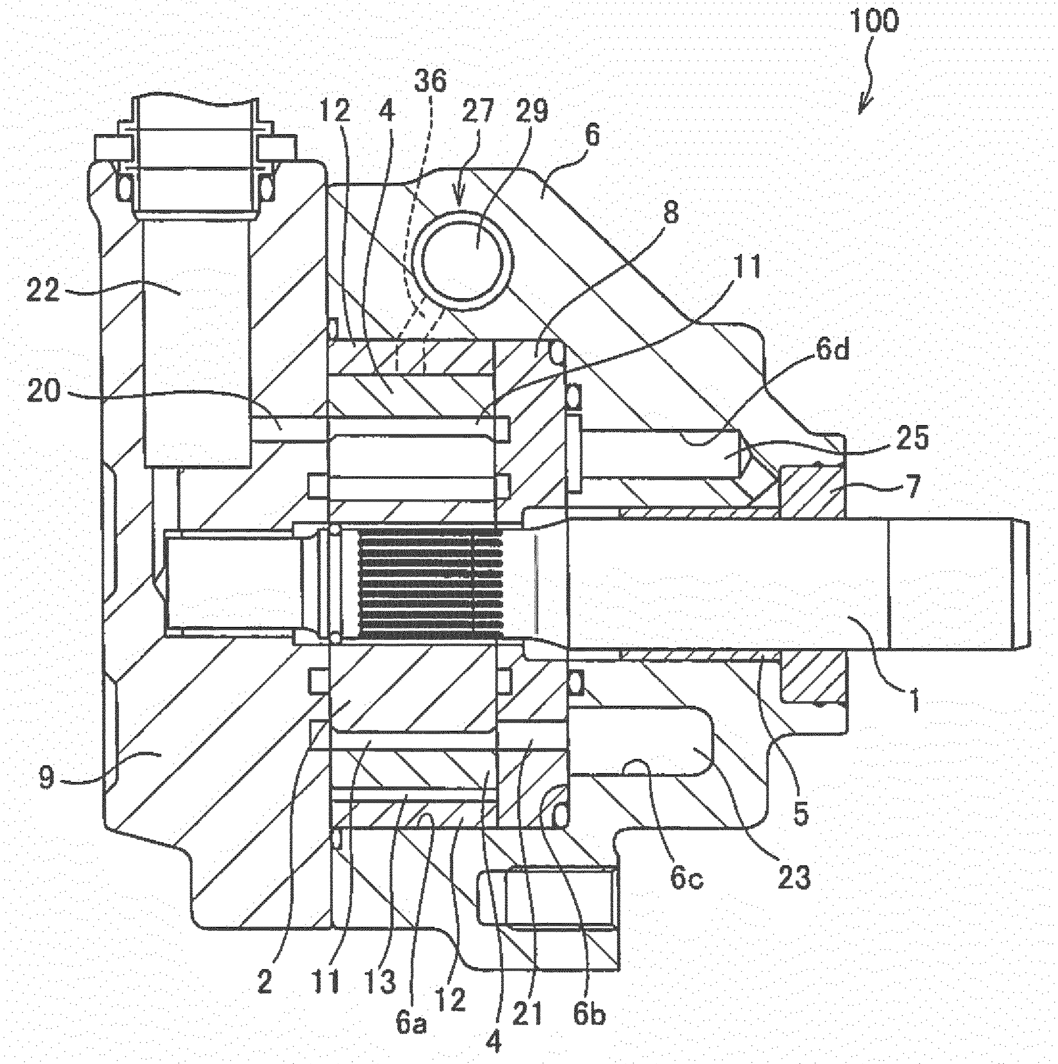

FIG.2

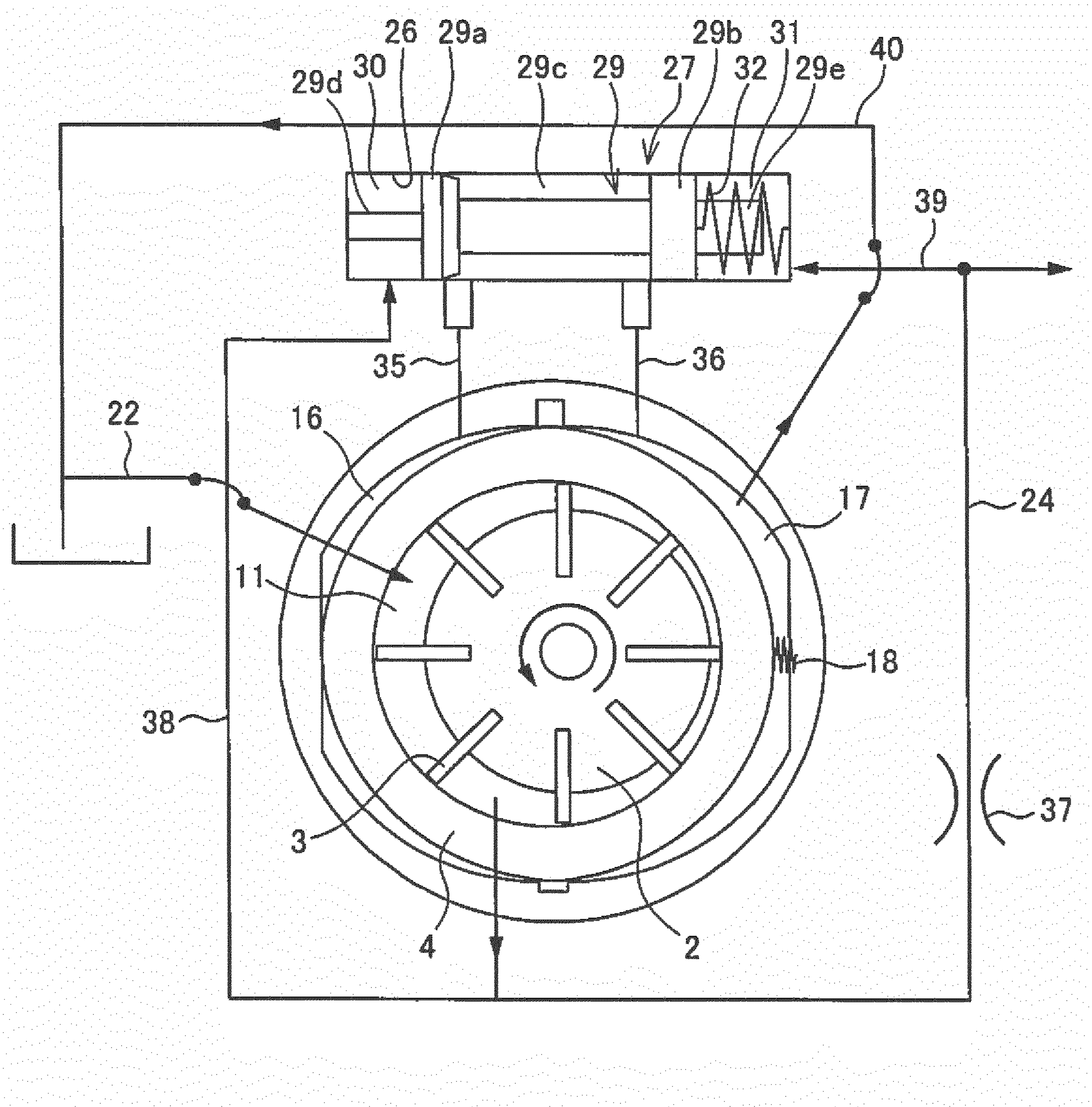
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FIG.3

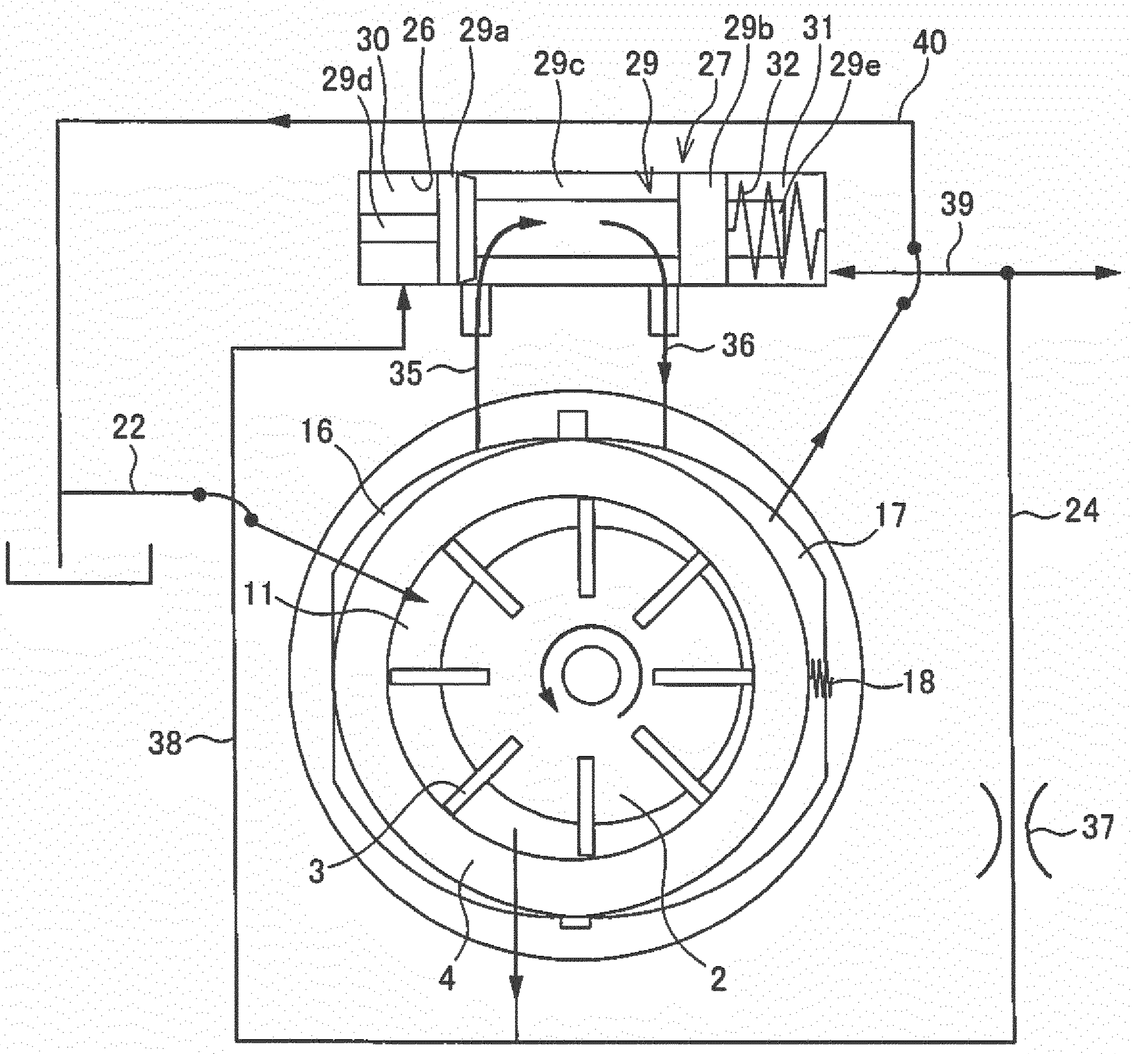
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FIG.4

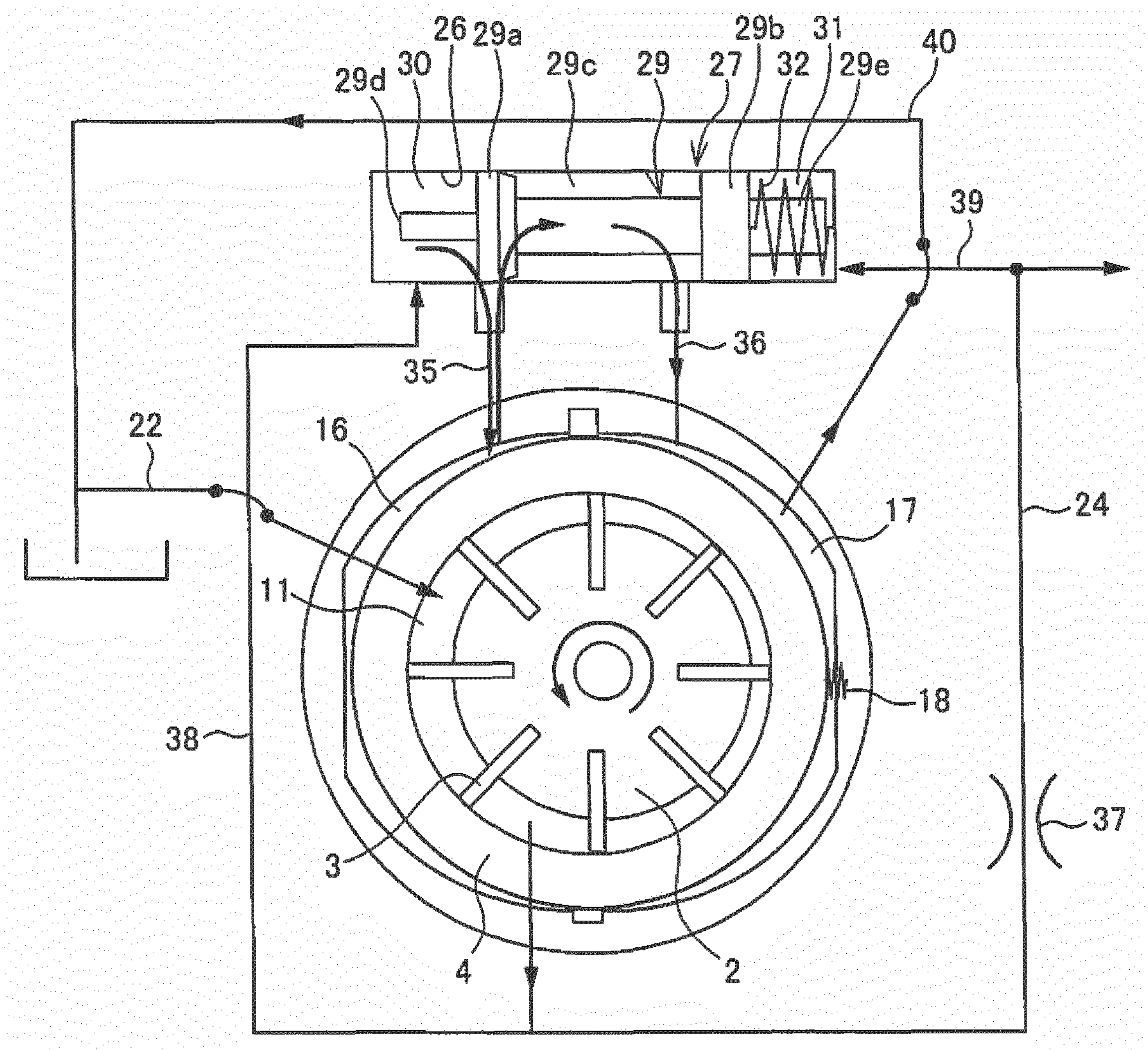
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FIG.5

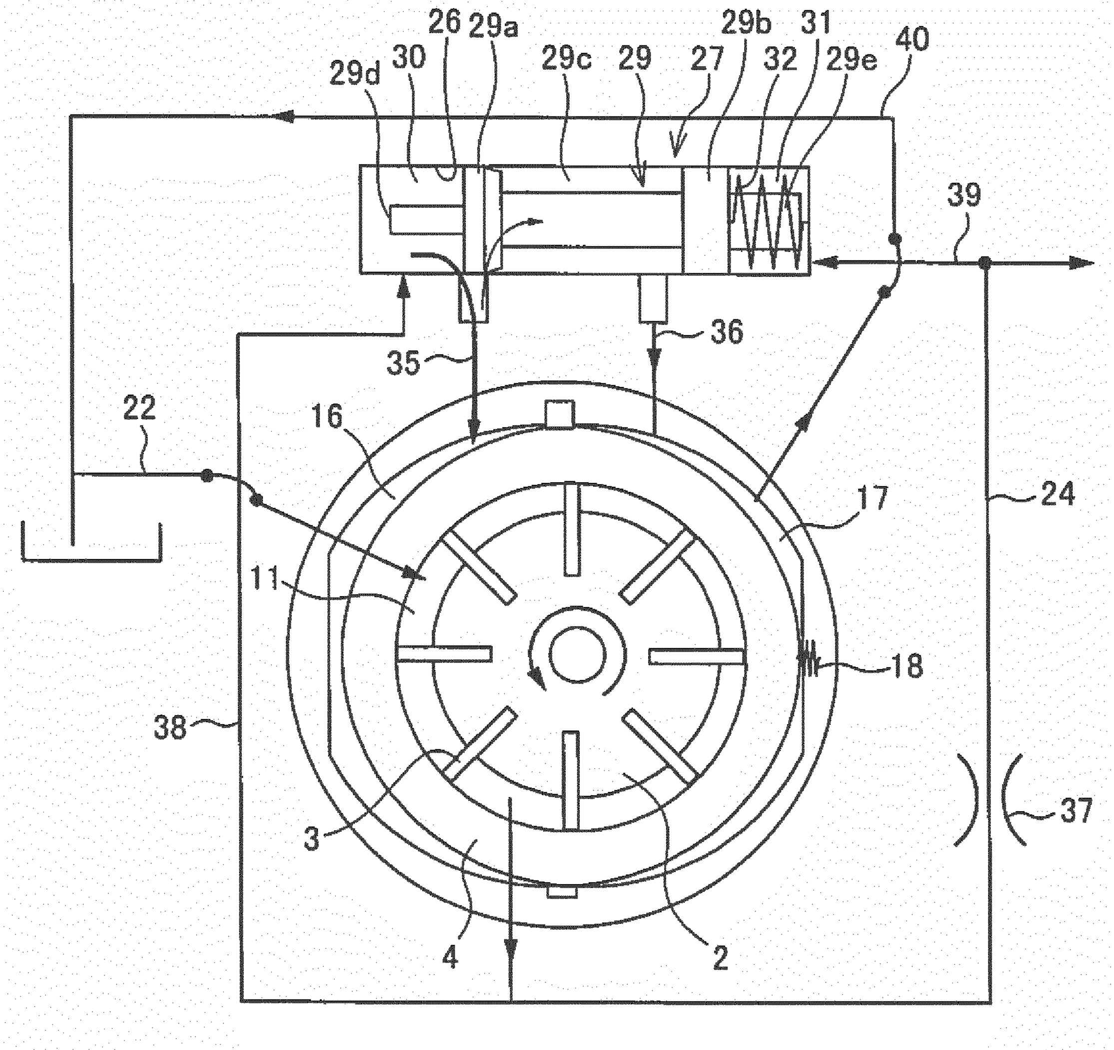
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FIG.6

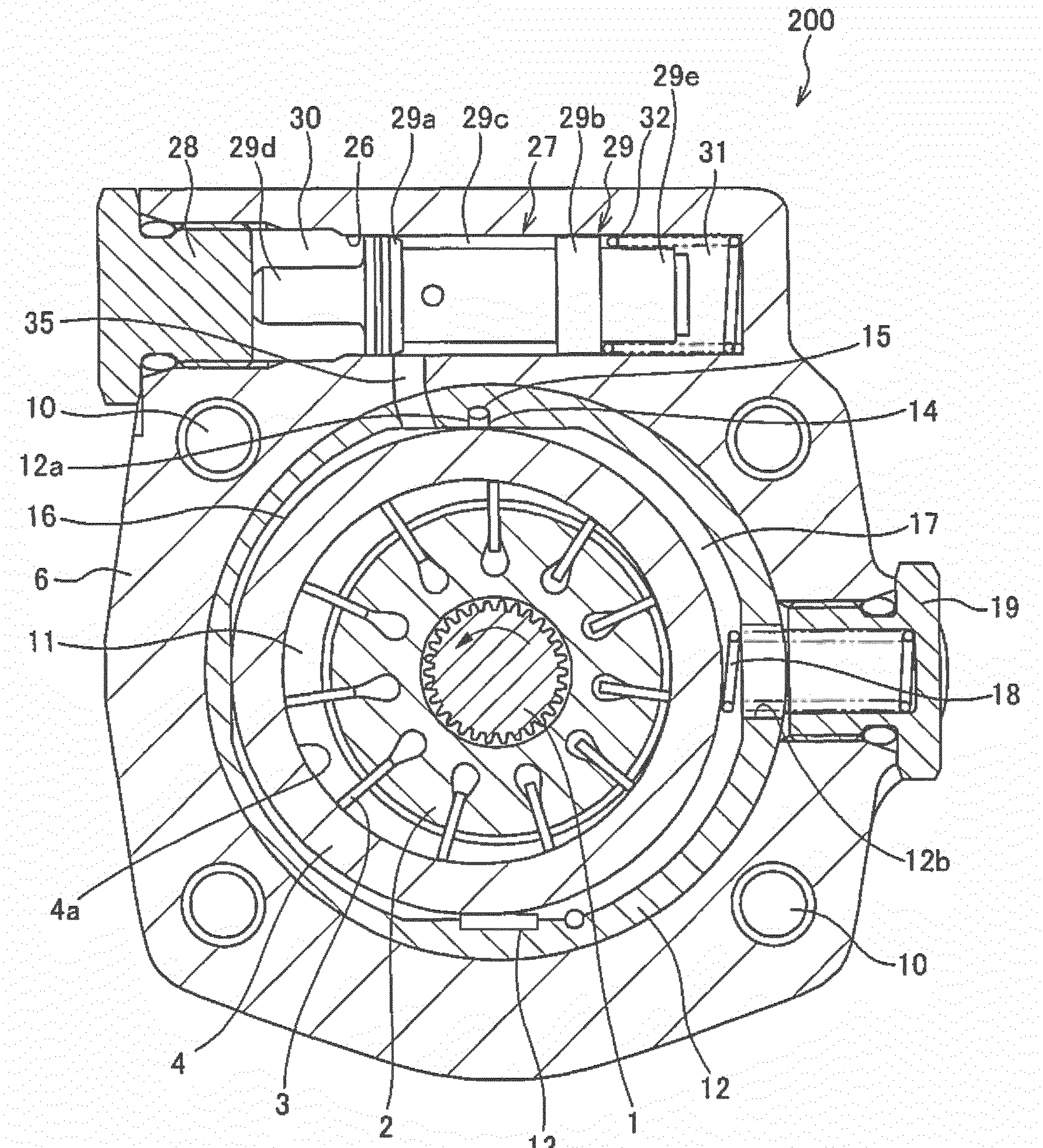
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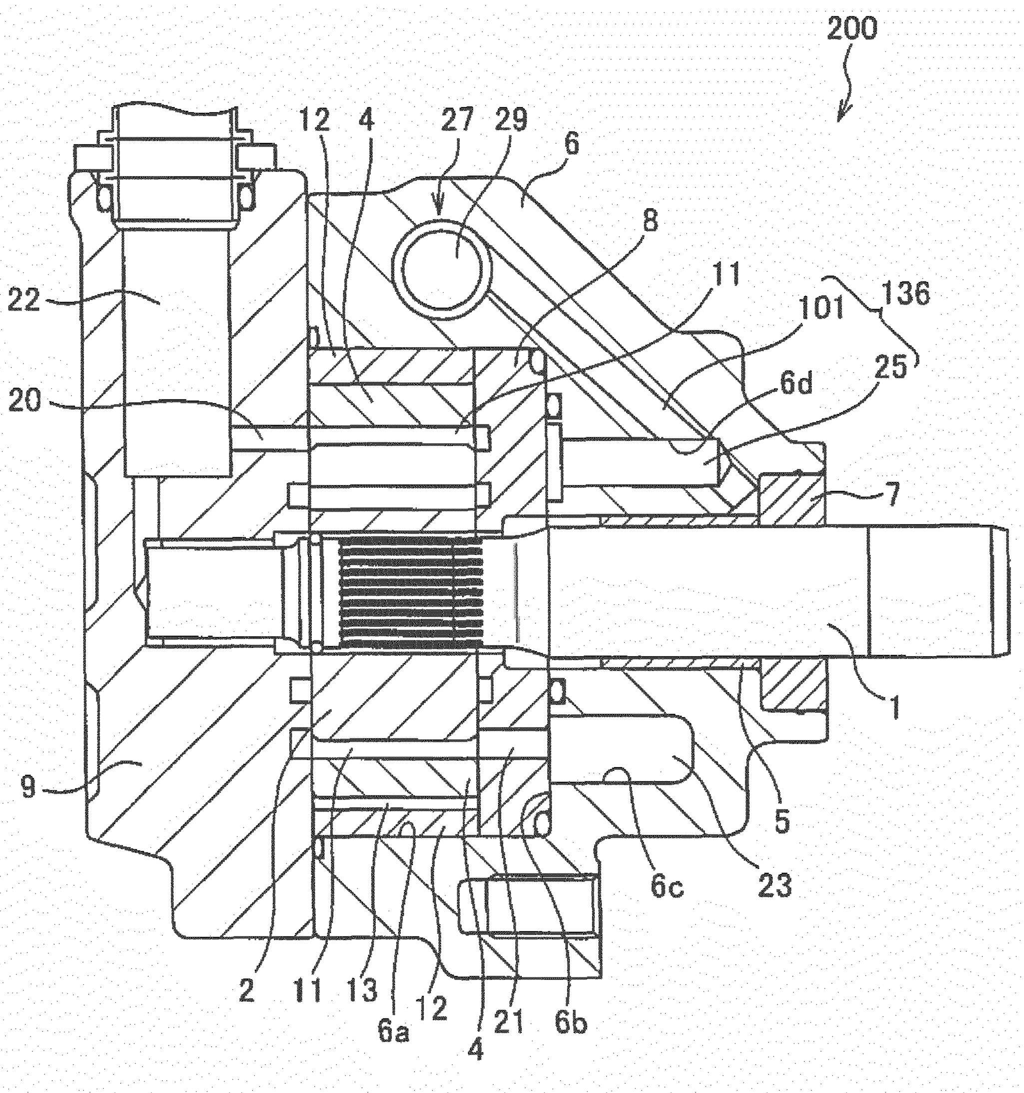
FIG.7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25
6d
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11
13
6a
12
4
21
6b

FIG.8

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2015/082937

A. CLASSIFICATION OF SUBJECT MATTER

F04C14/22(2006.01)i, *F04C2/344*(2006.01)i

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F04C14/22, F04C2/344

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2016
Kokai Jitsuyo Shinan Koho	1971-2016	Toroku Jitsuyo Shinan Koho	1994-2016

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 5116546 B2 (Kayaba Industry Co., Ltd.), 09 January 2013 (09.01.2013), entire text & US 2009/0269233 A1 & EP 2112378 A2	1-4
A	JP 2010-255552 A (Kayaba Industry Co., Ltd.), 11 November 2010 (11.11.2010), entire text (Family: none)	1-4
A	US 2013/0251570 A1 (TANASUCA, Cezar), 26 September 2013 (26.09.2013), entire text & US 2010/0221126 A1 & WO 2007/087704 A1 & KR 10-2008-0094902 A	1-4

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

* Special categories of cited documents:
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"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
17 February 2016 (17.02.16)	01 March 2016 (01.03.16)

Name and mailing address of the ISA/	Authorized officer
Japan Patent Office 3-4-3,Kasumigaseki,Chiyoda-ku, Tokyo 100-8915,Japan	Telephone No.

Form PCT/ISA/210 (second sheet) (January 2015)

REFERENCES CITED IN THE DESCRIPTION

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- JP 2013194692 A **[0002]**
- JP 2014239200 A **[0073]**