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(54) **SEAL RING, CYLINDER ARRANGEMENT AND PUMPING/MOTOR ARRANGEMENT**

(57) An example object of the embodiments disclosed is to provide a seal ring, a cylinder arrangement and a pumping arrangement, which is low in cost and reduces buckling forces in the cylinder and pumping arrangement. The seal ring, provided for sealing between a cylinder and a piston is provided with an inner surface and an outer surface, where the outer surface is adapted to seal against an inner surface of a cylinder. The outer surface of the seal ring is provided with a ridge forming the largest outer diameter of the seal ring. The cylinder arrangement comprises a cylinder and a piston arranged to reciprocate in the cylinder, wherein the piston is provided with an upper part having a smaller diameter than a diameter of an annular edge arranged below the upper part, and wherein the seal ring is arranged upon the annular edge and wherein an inner diameter of the seal ring is larger than the diameter of the upper part such that there is a radial play between the upper part and the seal ring. The pumping arrangement comprises an input shaft and a plurality of cylinder arrangements, wherein the cylinders are provided in a circular pattern upon a cylinder plate and the pistons are provided in a corresponding circular pattern upon a piston plate. The piston plate is provided skew towards the cylinder plate such that by rotating the cylinder plate, the pistons reciprocate within their respective cylinder.

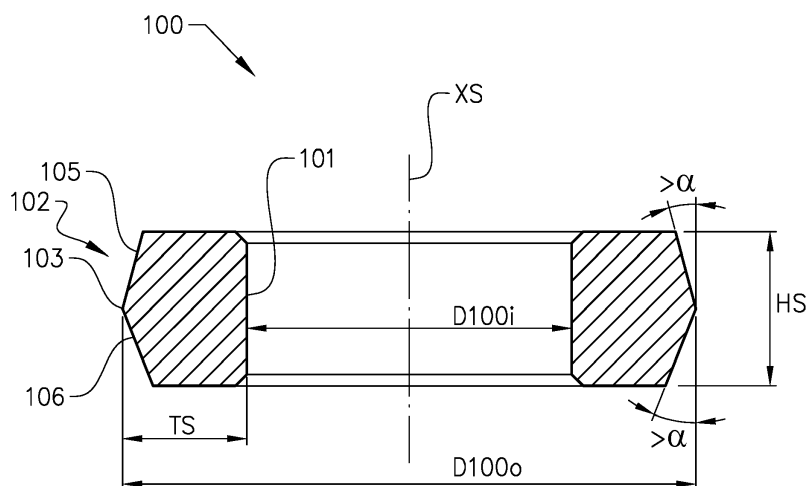


FIG. 1a

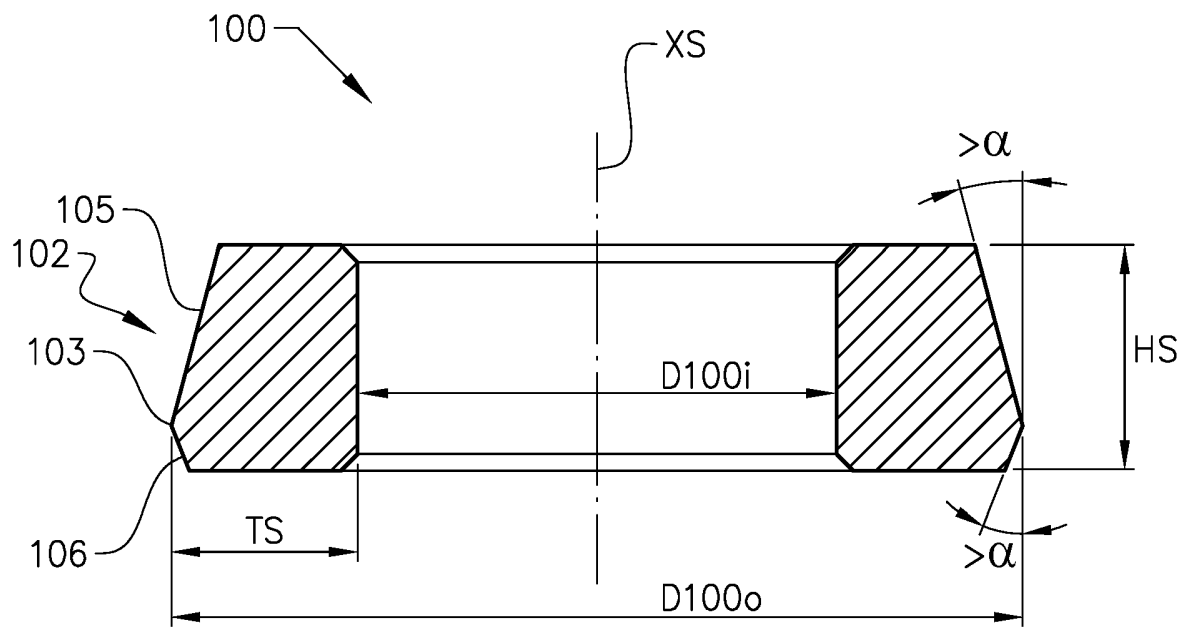


FIG. 1b

Description

TECHNICAL FIELD

[0001] The present disclosure relates to the fields of pumping/motor arrangements and especially to adjustable axial piston pumps/motor comprising a plurality of cylinder arrangements with reciprocating pistons. The disclosure further relates to the field of cylinder arrangements and seals for cylinder arrangements.

BACKGROUND ART

[0002] In conventional rotating piston pumps the pistons can be attached to a piston plate, which pistons are paired with cylinders provided upon a cylinder plate. The piston plate and the cylinder plate are arranged skew relative each other, whereby the pistons reciprocate in the cylinders when the piston plate rotates.

[0003] Through the skew arrangement of the piston plate, a rotation of the piston plate causes the reciprocating movement of the pistons. The pistons of the piston pump of this type are subjected to large bending forces, whereby the pistons and/or the cylinders must be designed to withstand these large forces. The skew arrangement also implies stringent demands in terms of manufacturing tolerances.

SUMMARY

[0004] An example object of the embodiments disclosed herein is to provide a seal ring, a cylinder arrangement and a pumping arrangement, which are efficient, low in cost and reduces bending forces in the cylinder and pumping arrangement. An exemplary embodiment of a seal ring is disclosed in claim 1, an exemplary embodiment of a cylinder arrangement is disclosed in claim 5 and an exemplary embodiment of a pumping arrangement is disclosed in claim 9. Further exemplary advantageous embodiments of seal rings, cylinder arrangements and pumping arrangements are disclosed in the dependent claims.

[0005] Conventional rotating hydraulic piston pumps generally comprise only a few pistons reciprocating in corresponding number of cylinders. The requirements regarding tolerances are stringent and in order to withstand the buckling forces generated upon the pistons and the cylinders the pump arrangements becomes bulky, heavy and need to be of high grade materials. High demands on the materials used in terms of strength and durability, and stringent tolerances, make the rotating hydraulic piston pumps expensive and bulky and further limit the number of pistons that can be used. The limited number of reciprocating pistons drives high pulsation which is problematic from a noise and vibrational perspective, generally referred to as NVH (Noise Vibration and Harshness), and also from a durability perspective.

[0006] Objects of the exemplary embodiments dis-

closed herein is reached by applying a seal ring, which is configured for sealing between an inner surface of a cylinder and an outer surface of a piston of a cylinder arrangement, and which seal ring is provided with a ridge, wherein a pumping arrangement overcoming or at least partly alleviating problems of conventional pumping arrangements can be provided.

[0007] According to an exemplary embodiment of the seal ring the outer surface of the seal ring is provided with a ridge forming the largest outer diameter of the seal ring. The seal ring has a height extending along a rotational axis of the seal ring and is provided with an inner surface and an outer surface, where the outer surface is adapted to seal against an inner surface of a cylinder. Since the ridge has the largest diameter of the seal ring the ridge defines a seal line of the seal ring. Conventional spherical seal rings are provided with a circular cross section. Due to the circular cross section, a conventional spherical seal ring, which seals against an inner surface of a cylinder, will always have a seal line perpendicular to the rotational axis of the cylinder, when the rotational axis of the spherical seal ring is set in an angle towards the rotational axis of the cylinder.

[0008] An exemplary advantage with the presented seal ring is that when the seal ring is used in a cylinder arrangement to seal between an inner surface of a cylinder and a piston arranged in the cylinder, the seal line becomes perpendicular towards an axial direction of the seal ring. Normally the seal ring is coaxial arranged upon the piston and the seal line is therefor also perpendicular towards the axial direction of the piston, whereby the piston only is subjected to axial forces, i.e. no radial and buckling forces. Because the piston is only subjected to large axial forces, which are less severe for the piston and the cylinder then large radial forces, both smaller dimensions and use of lower grade materials for both the piston and the cylinder is enabled. A more compact and less expensive cylinder arrangement can thereby be produced, due to the presented seal ring.

[0009] According to another exemplary embodiment of a seal ring the ridge is provided at half the height of the seal ring. An exemplary advantage with providing the ridge at half the height of the seal ring is that a mounting of the seal ring becomes efficient, since the seal ring can be mounted from any side, due to that the upper half of the seal ring is identical to the lower half of the seal ring.

[0010] According to another exemplary embodiment of a seal ring the ridge is provided below or above half the height of the seal ring. An exemplary advantage with providing the ridge below half the height of the seal ring is that in an application where the seal ring seals between a piston and a cylinder and the cylinder chamber defined by the piston and the cylinder is pressurised with a pressure medium. A total pressure force acting to press the seal ring outwards becomes smaller, whereby the friction between the seal ring and the inner wall of the cylinder is reduced. This is achieved, because of similar size of the radial areas of the inner wall of the seal ring and the

outer wall of the seal ring above the ridge, which is exposed to the pressure medium. The resulting force pressuring the seal ring radial outwards is a result of the different radial area size on the inner and outer wall of the seal ring that is exposed to the pressure in the cylinder chamber.

[0011] An exemplary advantage with providing the ridge above half the height of the seal ring is that in an application where the seal ring seals between a piston and a cylinder and the space above the piston is pressurised, a total pressure force acting to press the seal ring outwards becomes greater, because of a relative the radial outer area of the seal ring exposed to the pressure in the cylinder chamber is much smaller than the radial area of the inner wall of the seal ring.

[0012] Depending of application a lower or higher outward pressure force is desired upon the seal ring. A higher outward pressure increases the friction between an inner wall of a cylinder and the seal ring and allows the seal ring to withstand higher pressures. The higher friction also increases the friction losses. Depending on application a desired height/axial position of the ridge must be determined. According for a further exemplary embodiment of a seal ring the inner seal ring is provided with an essentially circular cylindrical inner surface, meaning that the inner surface of the circular seal ring is essentially flat in a direction extending along a rotational axis of the seal ring such that a cylinder is formed by the inner surface of the seal ring.

[0013] The seal ring is preferably made of a resilient material. The seal ring can thereby slightly expand when being exposed to forces pushing it outward. The seal ring can also thereby be provided with a pre-tension, to ensure that it is always in contact with the outer wall of the cylinder.

[0014] According to an exemplary embodiment of a cylinder arrangement the cylinder arrangement comprises a cylinder and a piston, wherein the piston is arranged to reciprocate in the cylinder. A cylinder arrangement with a cylinder and a piston is the basis for a piston pump, where the piston through its reciprocating in the cylinder builds up a low and a high pressure in order to suck in a pressure fluid in the cylinder and thereafter press it out. The pressure medium thereby exercises a pressure force upon the cylinder walls and the piston.

[0015] The piston is provided with a top portion, which is provided with an upper part having a smaller diameter than a diameter of an annular edge arranged below the upper part on the piston. A seal ring according to any previously disclosed exemplary embodiments is arranged upon the annular edge such that the seal ring surrounds upper part of the top portion. An inner diameter of the seal ring is larger than the diameter of the upper part, such that there is a radial play between the upper part of the piston and the seal ring.

[0016] The radial play provides the exemplary advantage that the seal ring can oscillate upon the upper part of the top portion, which allows the piston to reciprocate

in the cylinder with an angle between the rotational axis of the cylinder and the rotational axis of the piston. When the piston is arranged skew in relation to the cylinder ring, the upper part of piston will have different position relative the cylinder wall, depending on the pistons axial position in the cylinder. The radial play between the seal ring and the upper part takes up this radial position change of the piston during the reciprocating movement of the piston. This will be described more in detail in the detailed description and conjunction with the figures and exemplary embodiments disclosed therein.

[0017] Additionally, the fact that the seal ring is arranged with a radial play to the upper part of the top portion of the piston, wherein that the seal ring can be considered to float in relation to the top portion, has the further advantage that the piston will not be exposed to any radial forces. Since the seal ring is not fixedly arranged but floating in relation to the top portion of the piston a force acting on the seal ring will not be transferred to the piston. This has the advantage that if the shape of the seal ring deviates from the ideal shape, e.g. due to that the manufacturing tolerances would be substandard, such that the seal line formed in regards to the inner surface of the cylinder is not ideal, the piston is still not affected.

[0018] As previously disclosed the ridge of the seal ring provides a defined seal line between the inner surface of the cylinder and the piston. In combination with the radial play between the upper part of the piston and the seal ring an exemplary advantage with the cylinder arrangement is that this allows the piston to be arranged skew relative the central line of the cylinder and still only be subjected to axial forces. This is enabled in that the seal line is always perpendicular towards the axial direction of the piston, whereby the piston only is subjected to axial forces, i.e. no buckling forces, even though the piston is entering the cylinder at an angle. Some exemplary advantage of the absence of buckling forces, is that the friction between the piston and inner surface of the cylinder is minimised.

[0019] According to an exemplary advantage, in a pumping application of a rotating piston pump, where the pistons enters the cylinder with an angle and a pressure medium is pumped in and out of the cylinder chamber by means of the cylinder arrangement, the radial play between the upper part of the piston and the seal ring, enables the seal ring to be pressed against the cylinder wall by the pressure build up in the pressure medium in the cylinder chamber. When the pressure increases in the cylinder chamber, the pressure increases equally much in the space formed by the radial play between the inner surface of the seal ring and the outer surface of the upper part of the top portion. The pressure exerted by the pressure medium upon the seal ring filling the space formed by the radial play is the same as the pressure of the pressure medium in the cylinder, whereby the seal ring will be pressed out against the cylinder wall by this pressure.

[0020] In difference to a buckling force, which is direct-

ed in one radial direction, the pressure force from the pressure medium is equal in all outward radial directions of the seal ring. As described above, the resulting force pressing the seal ring outward in radial direction can be varied depending on where in axial direction the ridge is provided. The radially outward directed resulting force upon the seal ring improves the sealing between the inner surface of the cylinder and the piston provided by the seal ring. Improved sealing lowers the leakage over the seal ring and low leakage in turn gives high volumetric efficiency. The improved sealing also enables that the surface of the ridge of the seal ring sealing against the inner surface of the cylinder can be made small, or at least smaller than for conventional sealing rings, with a cylindrical cross-section, whereby according to the exemplary advantage the friction, and e.g. material wear and efficiency losses associated to friction, between the seal ring and the inner surface of the cylinder is lowered without risking leakage.

[0021] According to another exemplary embodiment of a cylinder arrangement the largest diameter of the piston is the outer circumference of the annular edge, and the outer surface of the piston below the annular edge, inclines inwards. The degree of inclination of the above referred to outer surface determines the possible inclination of the piston relative the cylinder. If the piston is inclined more than is permitted by the current inclination of the above referred to outer surface the outer surface and the inner surface of the cylinder collides. The larger inclination of the piston relative the cylinder that is desirable the more significant inclination of the above referred to outer surface is required. The play enables the seal ring to seal between the piston and the inner surface of the cylinder as the piston moves in the cylinder and a larger inclination of the piston relative the cylinder also requires a larger radial play between the seal ring and the upper part of the top portion of the piston.

[0022] According to yet an exemplary embodiment of a cylinder arrangement the top portion of the piston is provided with an upper rim. The upper rim has a diameter which is larger than the diameter of the top portion such that the upper rim protrudes out from the top portion in a direction essentially perpendicular to the axial direction of the piston. The upper rim has a diameter which is larger than the inner diameter of the seal ring. An exemplary advantage of providing the top portion with the upper rim is that the upper rim provides an axial lock for the seal ring. In a first axial direction of the piston the seal ring is locked by the annular edge of the piston and in a second axial direction the seal ring is locked by the upper rim. Thus, the upper rim secures that the seal ring is axially held in place. The second axial direction is directed opposite the first axial direction. The upper rim can be an integral part of the piston or be provided by a ring spring, screw or the like.

[0023] An exemplary advantage of any of the disclosed embodiments of the cylinder arrangement is that it is suitable to be used in a pumping arrangement such as a

rotating hydraulic piston pump, due to that the piston can enter the cylinder in an angle and still just be subjected to axial forces from the pressure medium in the cylinder.

[0024] According to an exemplary embodiment of a pumping arrangement the pumping arrangement comprises an input shaft and a plurality of cylinder arrangements according to any of the previously disclosed exemplary embodiments. The cylinders are provided in a circular pattern upon a cylinder plate and the pistons are fixedly provided in a corresponding circular pattern upon a piston plate. The input shaft is in rotational fixed connection with the cylinder plate and the piston plate is arranged about the input shaft, but not attached thereto. Additionally, the piston plate is provided skew towards the cylinder plate such that by rotating the cylinder plate, the pistons reciprocate within their respective cylinder.

[0025] By introducing a torque at the input shaft, a rotation of the cylinder plate is initiated due to that the input shaft is in rotationally fixed connection with the cylinder plate. The cylinder plate can be splined to the input shaft; other rotationally fixed connections are also possible. The cylinder plate comprises a plurality of cylinders. The cylinders provided upon the cylinder plate are aligned with the rotational axis of the cylinder plate. The piston plate comprises a plurality of parallel pistons, which are fixedly arranged upon the piston plate. The number of pistons and the number of cylinders correspond to each other. The pistons and the cylinders are arranged such that the pistons fit into the cylinders. The piston plate can thereby be arranged to the cylinder plate by that the pistons are fitted into corresponding cylinders. As the cylinder plate rotates the rotating movement is transferred to the piston plate over the pistons in the cylinders. Due to that the piston plate is arranged skew relative the cylinder plate the pistons of the piston plate reciprocate in corresponding cylinder of the cylinder plate as the cylinder plate and the piston plate rotates.

[0026] A rotating hydraulic piston pump generally comprises a shaft, a piston plate, wherein the piston plate comprises a number of pistons arranged in a circular pattern, and a cylinder plate, wherein the cylinder plate comprises a corresponding number of cylinders arranged in a corresponding circular pattern. The piston plate and the cylinder plate are arranged to the shaft, wherein the piston plate is rotationally coupled to the shaft. When the piston plate and the cylinder plate are assembled the pistons and the cylinders form a number of cylinder arrangements where a piston is arranged within a cylinder. When assembled the piston plate is arranged skew in relation to the cylinder plate which causes the pistons to be more or less inserted in respective cylinder. As the piston plate is rotated in relation to the cylinder plate, due to that the piston plate is arranged skew in relation to the cylinder plate; the pistons reciprocate in respective cylinder. In the top of the cylinders a cylinder chamber is formed limited by the top of the cylinder and the top of the piston. As the pistons are reciprocating the cylinders the volume of the cylinder chambers are periodically in-

creasing and decreasing. In the top of the cylinders an opening for inlet and outlet of pressure medium is provided. When a piston reciprocates downward in a cylinder, i.e. is retracts and the volume of the cylinder chamber is increased, whereby an under pressure is formed in the cylinder chamber and pressure medium is thereby sucked into the cylinder. When a piston reciprocates upward in a cylinder, the volume of the cylinder chamber is decreased and the pressure in respective cylinder chamber is increased, whereby the pressure medium in the cylinder chamber is pressed out of the cylinder through the openings. During rotation of the piston plate and the cylinder plate half of the pistons are currently being pushed into respective cylinder and half of the pistons are currently being retracted from respective cylinder, with the exception of that if a piston is maximally pushed into a cylinder, which is referred to top dead centre, or is maximally retracted, which is referred to as bottom dead centre, the piston is momentarily essentially stationary, wherein the pressure change is momentarily essentially zero. Thereby a high pressure side, generally referred to as pumping side, and a low pressure side, generally referred to as suction side, of the rotating hydraulic piston pump is created. A piston pump can normally also be driven as motor in which the pressure in the pressure medium is used in order to provide a torque at the output shaft.

[0027] The pumping and sucking sides are stationary relative the housing of the rotating piston pump, which thereby can be controlled with a control ring. An example of a suitable control ring is disclosed in EP 14150742.

[0028] Thus, the piston plate comprising pistons that is arranged about the input shaft but is not fixed thereto. The piston plate can be referred to as being floatingly arranged within the pumping arrangement. This allows a manufacturing with high tolerances. The interacting piston plate and cylinder plate are enclosed by a housing of the pumping arrangement. In a first axial direction the piston plate is held in place by the cylinder plate with which the piston plate interacts and in a second axial direction, the piston plate is supported by the pumping housing. The second axial direction is directed essentially opposite the first axial direction. The pumping housing is provided with a flat surface to support the piston plate. The flat surface is skew in relation to the radial direction of the pumps rotational axis. The angle of the supporting surface is the same as the angle of the piston plate relative the cylinder plate.

[0029] Some exemplary advantages with the exemplary embodiment of the rotating piston pump are that; the piston plate can be manufactured with low tolerances; the pistons and the cylinders are only subjected to axial forces when pumping, due to the seal line that is always perpendicular to the pistons, high volumetric effect due to low leakage; use of lower grade material leads to reduced costs.

[0030] The pumping arrangement may also be used as a motor. Hence, a pressure, such as e.g. from a pres-

surized fluid, is driving the pistons and a torque is transferred to the input shaft, which then becomes an output shaft. According to an exemplary advantage by providing a cylinder arrangement as previously described in the rotating pumping arrangement, a high effective and low cost pump/motor can be obtained. The exemplary advantage that only axial forces are exerted on the piston allows a fix installation of the pistons to the piston plate, wherein also the piston plate is only subjected to axial forces. The axial forces are advantageously absorbed by the housing of the pumping arrangement.

[0031] This enables that according to yet exemplary embodiments of the pumping arrangement smaller and lighter pistons may be used. This saves cost, weight and is preferable from a packaging perspective.

[0032] According to further exemplary embodiments of the pumping arrangement the angle of the piston and the seal line relative the cylinders, give rise to a cylinder internal pressure area difference. The cylinder internal pressure area difference is due to that the radial pressure exerted by the pressure medium in the pump is acting on an unevenly large area, due to the skew arrangement of the piston in the cylinder. The area difference creates a side force upon the respective cylinder, which is transferred to the cylinder plate. The joint side forces from the cylinders provided on the cylinder plate creates a torque that is transferred to the input shaft and acts in the same direction as the input torque.

[0033] The cylinder plate is provided with a back and a front side. The front side is the side where the cylinders are provided and is adapted to receive the pistons. The back side is the opposite axial side. The piston plate is provided with a front and a back side. The front side is the side where the pistons are provided. The back side is the opposite axial side.

[0034] According to other exemplary embodiments of the pumping arrangement the cylinder plate is provided with at least one opening in the top of each cylinder chamber. The openings in the cylinder chamber leads to the back side of the cylinder plate.

[0035] According to yet other exemplary embodiments of the pumping arrangement the piston plate is provided with a front side and a back side, wherein the front side is the side provided with the pistons and the back side is the opposite side, wherein the backside is supported by the pumping housing of the pumping arrangement.

[0036] According to other exemplary embodiments of the pumping arrangement the pistons are provided with a duct extending through respective piston in longitudinal direction. The ducts allow pressure medium to pass from the cylinder, through the piston and to the backside of the piston plate. The backside of the piston plate is provided with a balance space where the duct orifices. The balance space is sealed off by a sealing. The balance space are defined such that they are provided with an axial surface (axial direction of the piston plate) that is about the same area as the axial area of the pistons in the cylinder chamber. An exemplary advantage thereof

is that axial pressure equilibrium is thereby obtained over the piston plate.

[0037] In an exemplary embodiment the axial area of the balance spaces is about 2-5 % smaller than the axial area in the cylinder chamber of the respective piston. An exemplary advantage thereof is that piston plate thereby always will be pressed against the support surface of the housing, since there will be a resulting pressure force directed towards the back side of the piston plate.

BRIEF DESCRIPTION OF THE DRAWINGS

[0038] Embodiments of the seal ring, the cylinder arrangement and the pumping arrangement will be described more fully hereinafter with references to the accompanying drawings. The seal ring, the cylinder arrangement and the pumping arrangement, may however be embodied in many different forms and should not be constructed as limited to the examples set forth herein. Rather, these examples are provided so that this disclosure will be thorough and complete.

Fig. 1a-b shows a schematically cross section from a side view of exemplary embodiments of a seal ring.

Fig. 2 shows a schematically exemplary embodiment of a cylinder arrangement,

Fig. 3a-c shows a schematically exemplary embodiment of a cylinder arrangement in three sequential operational stages,

Fig. 3d-f shows a schematically cross section of an exemplary embodiment of a piston in a cylinder arrangement, in three corresponding sequential operational,

Fig. 4 shows a schematically pressure distribution in an exemplary cylinder chamber,

Fig. 5 shows a schematically cross-section view of an exemplary embodiment of a pumping arrangement, and

Fig. 6 shows a schematically exploded view of an exemplary embodiment of a pumping arrangement.

DETAILED DESCRIPTION

[0039] First, exemplary embodiments of the seal ring 100 will be described. In fig. 1a a schematically discloses cross section side view of an exemplary embodiment of a seal ring 100. The seal ring 100 has a height, HS, and is axially centralized along a rotational axis, XS. The seal ring 100 comprises an inner surface 101, an outer surface 102 and a ridge 103 provided at half the height, HS, of the seal ring 100. An edge of the ridge 103 of the outer surface 102 forms the largest outer diameter, D100o, of

the seal ring 100. The seal ring 100 also has an inner diameter D100i. The seal ring has a thickness, TS, wherein the thickness is defined as a thickness extending from the edge of the ridge 103 to the inner surface 102 of the seal, in a straight line perpendicular to the rotational axis XS. The ridge 103 defines the largest outer diameter D100o of the seal ring 100. The inner surface 101 is essentially parallel to the rotational axis, XS, wherein the inner surface 101 of the seal ring 100 forms a circular cylinder having an essentially circular cylindrical inner surface 101.

[0040] The outer surface 102 further comprises a first and a second surface portion 105, 106 inclining inwards from the ridge 130. The inclination between respective first and second surface portions 105, 106 and an axis parallel to the rotational axis, XS, is always larger than an angle α . The angle α will be disclosed more in detail in relation to fig. 5.

[0041] Fig. 1b schematically shows a cross section from a side view of another exemplary embodiment of a seal ring 100. The seal ring 100 of fig. 1b corresponds to the seal ring 100 of fig. 1a except for that the ridge 103 of fig. 1 is provided below half the height HS of the seal ring 100. The ridge 103 which in the exemplary embodiment of fig. 1b is provided below half the height HS of the seal ring 100, may according to other exemplary embodiments be provided also above half the height, HS, of the seal ring 100.

[0042] Conventional hydraulic pumps or motors are generally associated with stringent tolerances and due to that generally relatively few pistons are used, which gives high pulsation within such hydraulic pumps, NVH are generally also an issue. Further, in conventional hydraulic pumps the pistons, by means of conventional sealing rings with a cylindrical cross-section sealing between the piston and the inner surface of the cylinder, exerts forces in both axial and radial direction which in turn generates friction and buckling forces. This implies that high grade materials have to be used which together with e.g. the stringent tolerances makes hydraulic pumps or motors expensive.

[0043] By means of fig. 2-5 exemplary embodiments of applications of the proposed ring seal 100 will henceforth be disclosed. By providing the seal ring 100 in cylinder and rotating pump arrangements exemplary embodiments of cylinder arrangements 200 and exemplary embodiments of pumping arrangements 300, overcoming, or at least alleviating, the above clarified problematic areas of conventional hydraulic pumps be disclosed.

[0044] Fig. 2 schematically shows an exemplary embodiment of a cylinder arrangement 200. The cylinder arrangement 200 comprises a cylinder 210, having a cylinder inner surface 211, 212 and a piston 220. The piston 220 is arranged to reciprocate in the cylinder 210. The piston 220 is provided with a top portion 224, which is provided with an upper part 221 having a smaller diameter, D221o, than a diameter, D222o, of an annular edge 222 arranged below the upper part 221. The upper part

221 is further provided with an outer surface 225. A seal ring 100 in accordance to what has been disclosed in relation to fig. 1a is arranged upon the annular edge 222. An inner diameter, D100i, of the seal ring 100 is larger than the diameter, D221o, of the upper part 221 such that there is a radial play between the upper part 221 and the seal ring 100.

[0045] The embodiment disclosed in figure 2 may also be provided with a seal ring 100 provided with a ridge 103 either below or above half the height HS of the seal ring. The choice of seal ring is dependent of the effects desired.

[0046] The top portion 224 of the piston 220 is additionally provided with an upper rim 223. The upper rim has a diameter D223o which is larger than the diameter D221o of the top portion 221 and larger than the inner diameter D100i of the seal ring 100. The upper rim 223 has the exemplary effect that when the seal ring 100 is mounted, it is retained on the upper part by the upper rim 223 in a first axial direction of the piston 220, which in to fig. 2 is defined as the Y-direction, and by the annular edge 222 in a second axial direction of the piston 220. A top of each cylinder 210 is provided with an opening 311. The cylinder 210 is provided with an annular inner surface 211.

[0047] According to the exemplary embodiment of fig. 2 the seal ring 100 seals between the inner surface 211, 212 of the cylinder 210 and the piston 220 wherein a seal line SL is formed between the seal ring 100 and the inner surface 211, 212 of the cylinder 210. As will be discussed more in detail later on, the seal line SL is always perpendicular towards a axial direction of the piston 220.

[0048] Further, in the exemplary embodiment shown in fig. 2 the pressure duct 226 is disclosed. Through the pressure duct 226 the pressure in the cylinder chamber can be spread to the backside of the piston 220, in order to achieve axial pressure equilibrium over the piston 220. The implementation of this is further explained in conjunction with fig. 5.

[0049] In the top of the cylinder chamber is an opening 311 provided. Through the opening 311 can a pressure medium be sucked in or pressed out, depending of direction travel of the piston 220. When the piston 220 is moving up in the cylinder 210 and thereby decreasing the volume of the cylinder chamber, the pressure medium located in the cylinder chamber is pressed out of the cylinder chamber through the opening 311. When the piston 220 is moving down in the cylinder 210 and thereby increasing the volume of the cylinder chamber, pressure medium is sucked into the cylinder 210 through the opening 311.

[0050] In fig. 4 an exemplary embodiment of the cylinder arrangement 200 is schematically disclosed. The embodiment in fig. 4 the piston 220 is arranged skew in relation to the cylinder 210. The skew arrangement of the piston 220 exemplifies how the piston is arranged when the cylinder arrangement is applied in a rotating piston pump. The arrows in the cylinder chamber in fig.

4 illustrate the pressure in the cylinder chamber.

[0051] When the cylinder arrangement 200 is applied in a rotating pumping arrangement (as pumping arrangement 300 in fig. 5 and 6) a pressure is built up in the cylinder 210 as the piston 220 reciprocates and compresses the pressure medium in the cylinder 210. The built up pressure is homogenous, meaning that the pressurized fluid exerts the same pressure on all surfaces enclosing the pressurized fluid. The radial play RP between the upper part 221 of the piston and the seal ring 100 enables that a space is formed between the inner surface 101 of the seal ring 100 and the outer surface 225 of the upper part 221. Also in the space RP formed by the radial play RP, the pressure is equal to the rest of the cylinder chamber. The pressure exerted by the pressure medium on the inner surface 101 of the seal ring 100 presses the seal ring 100 against the inner cylinder wall 211. The force the seal ring 100 is pressed against the cylinder wall 211 is thereby dependent of the pressure in the cylinder chamber. When there is a high pressure the radial force on the seal ring is higher. An exemplary effect of this variable radial pressure is that the seal ring seal more efficiently with a minimum of friction losses, this because the seal ring 100 only is subjected to high forces in order to seal more efficiently when the pressure in the cylinder chamber is high and the better seal effect is needed.

[0052] The radial play also has the advantage that e. g. a force acting on the seal ring not have to be transferred to the piston since the seal ring is not fixedly arranged but floating in relation to the top portion of the piston. This e.g. implies that the piston will not be exposed to radial forces.

[0053] Improved sealing lowers the leakage over the seal ring 100, which in turn gives high volumetric efficiency. The improved sealing also enables that the surface portion of the ridge 103 of the seal ring sealing against the inner surface 211 of the cylinder 210 can be relatively small, or at least smaller then for conventional spherical sealing rings with cylindrical cross section. The friction between the relatively small ridge 103 of the seal ring 100 and the inner surface 211 of the cylinder 210 can thereby be reduced, which in turn reduces material wear and efficiency losses without risking leakage.

[0054] Conventional seal rings have circular cross sections and due to the circular cross sections conventional seal rings sealing against an inner surface of a cylinder will not have a seal line perpendicular to the rotational axis of the seal ring when the seal ring is set in an angle against the rotational axis of the cylinder. The ridge 103, which has the largest diameter of the seal ring 100, defines a seal line SL of the seal ring. This seal line SL will always be perpendicular to the longitudinal direction of the piston 220, which thereby only is subjected to axial forces.

[0055] Further in the exemplary embodiment of the cylinder arrangement 200 disclosed in fig. 4, a margin between the upper rim 223 and the seal ring 100 can be

seen. When there is an over pressure in the cylinder chamber (as it is in fig. 4) the seal ring 100 seals in axial direction between the its lower side and the upper rim 222. When there is an under pressure in the cylinder chamber (as it is when pressure medium is sucked into the cylinder chamber) the seal ring 100 will seal in axial direction between its top surface and the upper rim 223. An exemplary effect with this small axial between the seal ring and the upper rim and the annular edge, is that there will be a better flow of pressure medium into the space of the radial play RP, when there is an over pressure in the cylinder chamber. The axial play is excessive disclosed in fig. 4.

[0056] Now, when the piston 220 is skew arranged as in fig. 4, there will be a different radial pressure force in different radial direction of the cylinder 210. This is due to that the seal line SL of the seal ring always is perpendicular to the axial direction of the piston. This is illustrated by the height difference between h1 and h2 in fig. 4. An exemplary effect of this difference is that there will be a side force on the cylinder 210.

[0057] How the pressurized fluid of the cylinder arrangement 200 improves the sealing properties of the seal ring 100 is herein disclosed in relation to a cylinder arrangement 200 applied in a pumping application. However, the same principle applies if the cylinder arrangement 200 is applied in a hydraulic piston motor.

[0058] Now, in fig. 5 and 6 is a schematic cross section view and exploded view of a pumping arrangement 300 disclosed. The pumping arrangement 300 comprises a rotating piston pump using any embodiment of the cylinder arrangement 200 and seal ring 100 described above.

[0059] First, the base parts of the pumping arrangement 300 will be described. The pumping arrangement 300 comprises an input shaft 301, a cylinder plate 310, a piston plate 320 and a pumping housing 330. The pumping housing comprises three parts, the first end part 331, the middle cylinder 333 and the second end part 332. By mounting the three parts 331, 332, 333 of the housing 330 together the pumping unit can be enclosed by the housing 330. The pumping unit comprises the piston plate 320 and the cylinder plate 310. The piston plate 320 is provided with a plurality of pistons 220 and the cylinder plate 310 is provided with the same number of cylinders 210. When the pumping unit is mounted together, the pistons 220 of the piston plate 320 are provided inside the cylinders 210 of the cylinder plate 310. The input shaft 301 is in rotational fixed connection with the cylinder plate 320 and the piston plate 320 of the pumping arrangement 300 is floatingly arranged relative the cylinder plate 310 and pumping housing 330, i.e. the piston plate 320 is arranged about the input shaft 301 but is not directly connected thereto. The input shaft 301 is mounted in the first end plate 331 of the housing 330, through a bearing 342. A seal 342 seals between the input shaft 301 and the first end part 331 of the housing 330. A locking ring 344, 345 locks the seal 343 and bearing 342 to their positions on the housing 330.

[0060] In fig. 6, the control ring 400 controlling is also disclosed. The functioning of the control ring is disclosed in EP 14150742. However, the control ring 400 connects to the openings 311 of the cylinders 220, 220' of the cylinder plate 310. Through the screw 406, the control ring 400 can be turned and the flow in and out from the pumping unit can be regulated. The control ring 400 is provided with a first and a second channel 401, 402, which are mouthed to a mouthed area in the second end plate 332 of the housing 330. The respective mouthed area in the second end plate 332 are separated and isolated from each other through three seals 403, 404, 405.

[0061] The piston plate 320 comprises a plurality of pistons 220, 220' arranged in a circular pattern. The piston plate 320 is provided with a front side 321 and a back side 322, wherein the front side 321 is the side provided with the pistons 220, 220' and the back side 322 is the opposite side of the piston plate 320. The pistons 220, 220' are fixedly arranged in the piston plate 220, 220'. The backside of the piston plate 320 is supported by a supporting surface provided on the first end plate 331. The supporting surface is skew arranged, in order to achieve the skew arrangement of the piston plate 320. The rotational axis xp of the piston plate 320 thereby becomes an angle α to the rotational axis xi of the input shaft, the cylinder plate 310.

[0062] The cylinder plate 310 comprises a plurality of cylinders 210, 210' arranged in a circular pattern, wherein the number of, and position of, the cylinders 210, 210' correspond to a number of pistons 220, 220' of the piston plate 320. When the pumping arrangement 300 is assembled the pistons 220, 220' of the piston plate 320 are arranged to reciprocate within corresponding cylinders 210, 210' of the cylinder plate 310.

[0063] Thus, the piston plate 320 is freely/floatingly arranged in relation to the input shaft 301 and is radial positioned through that the pistons 220, 220' of the piston plate 210 are arranged in the corresponding cylinders 210, 210' and is axial positioned through the pressure from the spring 340 and the support at the support surface of the first end plate 331.

[0064] The skew arrangement of the piston plate 320 in relation to the cylinder plate 310 enables the pistons 220, 220' to reciprocate in the cylinders as a rotational movement is introduced in the pumping unit over the input shaft 301 and the cylinder plate 310. According to an exemplary embodiment the piston plate 320 is angularly offset in relation to the cylinder plate 310 with an angle α , wherein α in an exemplary embodiment is about 2 - 15 degrees and in another exemplary embodiment about 8 degrees.

[0065] In the piston 220' the pressure duct 226' connecting the cylinder chamber of the cylinder 201' with the balance space 323' can be seen. The axial area of the balance space 323' is in one embodiment about 1-10 % smaller than the axial area of the piston 320' in the cylinder 210' in order to always have a resulting force pressing the piston plate 320 against the support surface of

the first end plate 331. In another exemplary embodiment the axial area of the support space 323' is 3 % smaller than the axial area of the piston 320' in the cylinder 210'. All the balance spaces of the piston pump is provided with an equally large axial surface. The example of the balance space 312' is applicable on all balance spaces 323 of the piston plate 320.

[0066] Now, by introducing a torque at the input shaft 301, the torque is transferred to the cylinder plate 310 over the spline connection 341. The pistons 220, 220' provided upon the piston plate 320 will rotate with their respective cylinder 210, 210', whereby the pistons 220, 220' will reciprocate in the cylinders 210, 210' due to the skew arrangement of the pistons plate 320. A pressure medium can thereby be pumped from a high pressure side to a low pressure side of the cylinders. The pumping arrangement can easily be controlled by a control ring 400 as described in EP 14150742.

[0067] Now, to allow this configuration of the pumping arrangement disclosed in fig. 5 and 6, the pumping arrangement 300 is provided with a cylinder arrangement 200 according to fig. 2, 3 and 4 and a seal ring according to fig. 1. Fig. 3a-f schematically illustrates how the piston 220 and the seal ring 100 moves within the cylinder 210, during operation of the pumping arrangement.

[0068] Fig. 3a to 3c schematically shows a side view of an exemplary embodiment of a cylinder arrangement 200. The cylinder arrangement 200 is provided with a seal ring 100 according to fig. 1a. Any variation of the suggested seal ring would however do. Fig. 3a - c shows a piston 220 reciprocating within a cylinder 210, in three sequential operational stages S1, S2, S3 during operation. Fig. 3a to 3c shows the cylinder arrangement 200 in an X-Y plane. The cylinder 210 is provided with a cylinder wall 211. The sequence of operational stages illustrates how a piston 220 moves within a cylinder 210, when the cylinder arrangement 200 is provided in a rotating piston pump as disclosed in fig. 5 and 6.

[0069] Fig. 3d to 3f schematically shows a cross section view of a piston 220 of the cylinder arrangement 200. The cross section view is from directly beneath the upper rim 223 and is simplified and the movements are exaggerated and simplified in order to facilitate the explanation. Fig. 3d-f disclose the three sequential operational stages S1, S2, S3 during operation in order to illustrate how the seal ring 100 oscillates upon the piston 220. Fig. 3d to 3f shows the piston 220 of the cylinder arrangement 200 in an X-Z plane.

[0070] During operation the piston 220 is defined as reciprocating essentially in the Y-direction, wherein in order to improve the clarity a movement in positive Y-direction is referred to as upwards whereas a movement in negative Y-direction is referred to as downwards, and performing a swirling movement in the X-Z plane. The movements in the Y-X plane and in the X-Z plane of the piston 220 in the cylinder 210 during the sequential operational stages S1 to S3 are disclosed in fig. 3a to 3c and 3d to 3f respectively.

[0071] In fig. 3a and 3d the cylinder arrangement 200 is in the first operational stage S1, the piston 220 reciprocates upwards within a cylinder 210 after a bottom dead centre of the piston 220 is passed. Thus, at the position of the piston 220 disclosed in fig. 3a the piston 220 has recently performed a reciprocating movement downwards in the cylinder 210 and is now reciprocating upwards. As required in the exemplary embodiments, the seal ring 100 is provided with a radial play RP upon the piston 220. The radial play RP allows the movement of the piston 220 in the cylinder 210. When the piston 220 reciprocates in the cylinder 210 as disclosed in figure 3a-c the seal ring 100 will oscillate upon the upper part 221 of the piston 220 as disclosed in figure 3d-f.

[0072] Due to that the seal line SL always is perpendicular to the axial direction of the piston 220, the piston will only be subjected to axial forces. This gives the exemplary effects that the piston can be provided fixedly in the piston plate 320 and also be made of a lower grade material.

[0073] It should be noted that for clarification purposes the movements of the piston 220 in the cylinder 210, as the piston 220 reciprocates in the cylinder 210, and the radial play RP formed as the seal ring 100 oscillates as the piston 220 reciprocates, is exaggerated in fig. 3a to 3f.

[0074] A pumping arrangement designed according to any of the exemplary embodiments disclosed herein, will have the exemplary advantage of: Low tolerances, due to the use of pistons seals 100 with a seal line always perpendicular to the axial direction of the pistons 220, whereby the piston plate 320 can be arranged floating in the pumping arrangement 300. The pistons 220 can be fixedly arranged to the floating piston plate 320. An easy control of the pumping arrangement can be provided by a turntable control ring. High volumetric efficiency due to the low leakage of the piston seal 100. The piston 220 never experience any side load, which means the friction between the piston 220 and the cylinder wall 211 can be neglected.

[0075] As will be realised, the embodiments of seal ring, the cylinder arrangement and the pumping arrangement disclosed herein can be subjected of modification in various obvious respects, all without departing from the scope of the appended claims. Accordingly, the drawings and the description thereto are to be regarded as illustrative in nature and not restrictive.

Claims

1. Seal ring (100) for sealing between a cylinder and a piston, wherein the seal ring (100);
 - having a height (HS) extending along a rotational axis (XS) of the seal ring (100),
 - is provided with an inner surface (101) and an outer surface (102), where the outer surface (102) is adapted to seal against an inner surface

() of a cylinder (),

characterised in that the outer surface (102) is provided with a ridge (103) forming the largest outer diameter (D100o) of the seal ring (100).

2. Seal ring (100) according to claim 1, wherein the ridge (103) is provided on half the height (HS) of the seal ring (100).
3. Seal ring (100) according to claim 1, wherein the ridge (103) is provided below or above half the height (HS) of the seal ring (100).
4. Seal ring (100) according to any of the preceding claims 1-3, wherein the seal ring (100) is provided with essentially circular cylindrical inner surface.
5. Cylinder arrangement (200) comprising a cylinder (210) and a piston (220) arranged to reciprocate in the cylinder (210), wherein the piston (220) is provided with a top portion (221), which is provided with an upper part (221) having a smaller diameter (D221) than a diameter (D222) of an annular edge (222) arranged below the upper part (221), wherein a seal ring (100) according to any of the claims 1-4 is arranged upon the annular edge (222), wherein an inner diameter (D100i) of the seal ring (100) is larger than the diameter (D221) of the upper part (221), such that there is a radial play (RP) between the upper part (221) and the seal ring (100).
6. Cylinder arrangement (200) according to claim 4, wherein the largest diameter of the piston (220) is the outer circumference of the annular edge (222), and the outer surface of the piston (220) below the annular edge (222) inclines inwards.
7. Cylinder arrangement (200) according to claim 6, wherein the top portion (221) of the piston (220) is provided with an upper rim (223) with a diameter (D223) larger than the diameter of the top portion (221) and larger than the inner diameter of the seal ring (100).
8. Pumping arrangement (300) comprising;

an input shaft (301) and
a plurality of cylinder arrangements (200) according to any of the preceding claims 1-7,

wherein the cylinders (210) are provided in a circular pattern upon a cylinder plate (310) and the pistons (220) are provided in a corresponding circular pattern upon a piston plate (320) and the input shaft (301) is in rotational fixed connection with the cylinder plate (310) and the piston plate (320) is rotatable arranged upon the input shaft (301), **characterised**

in that the piston plate (320) is provided skew towards the cylinder plate (310) such that by rotating the cylinder plate (310), the pistons (220) reciprocate within their respective cylinder (210).

9. Pumping arrangement (300) according to claim 8, wherein the cylinder plate (310) is provided with openings (311) in the top of each cylinder (210), wherein the openings (311) mouth to the backside (315) of the cylinder plate (310).
10. Pumping arrangement (300) according to claim 9, wherein the piston plate (310) is provided with a front side (321) and a back side (322), wherein the front side (321) is the side provided with the pistons (220) and the back side (322) is the opposed side, wherein the backside (322) is supported by the a pumping housing (330).
11. Pumping arrangement (300) according to any of the claims 9-10, wherein each piston (220) is provided with a pressure duct (226) forming a pressure passage through respective piston (220), such that a pressure difference over a piston (220) is at least partially equalized.

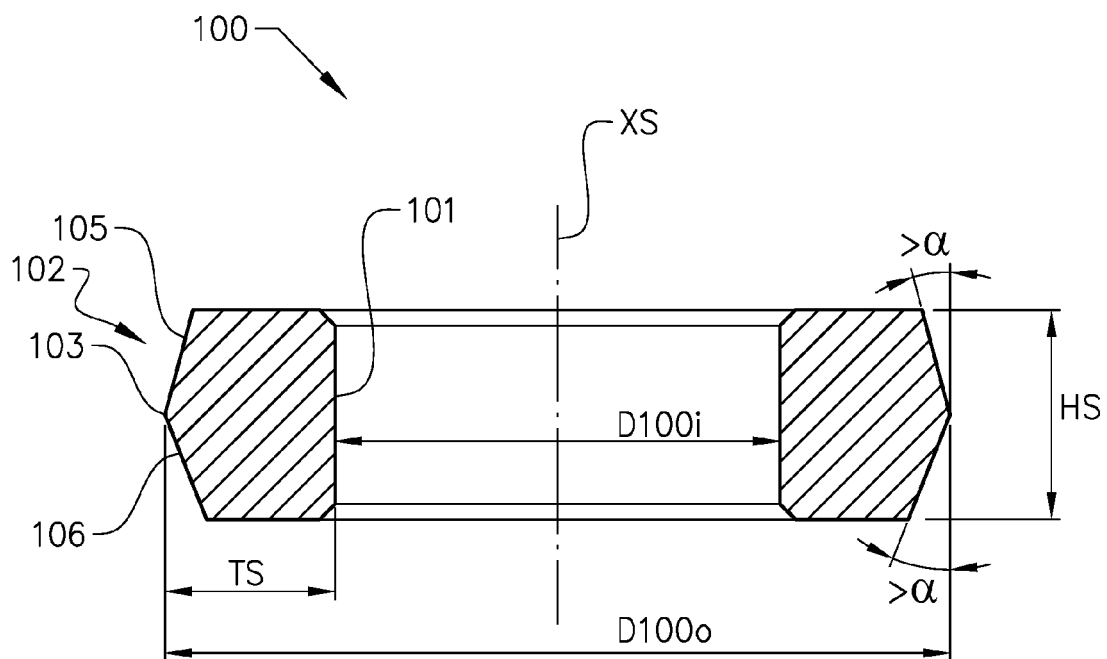


FIG. 1a

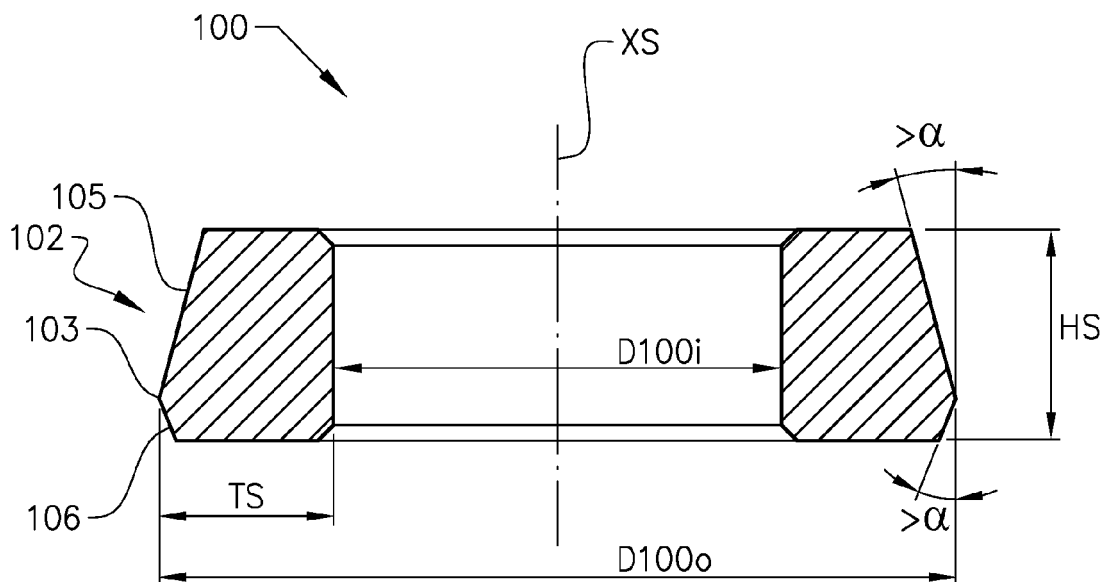


FIG. 1b

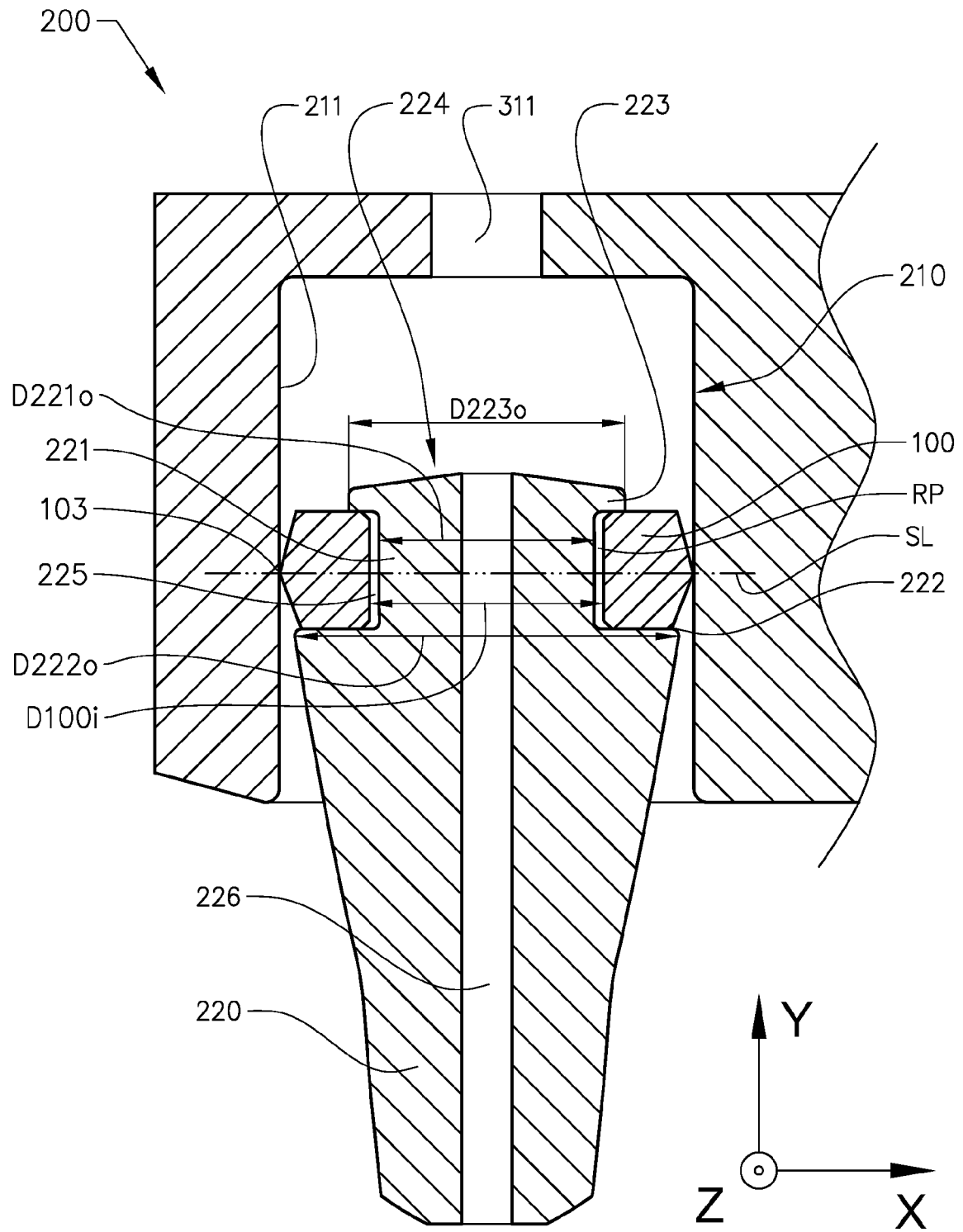
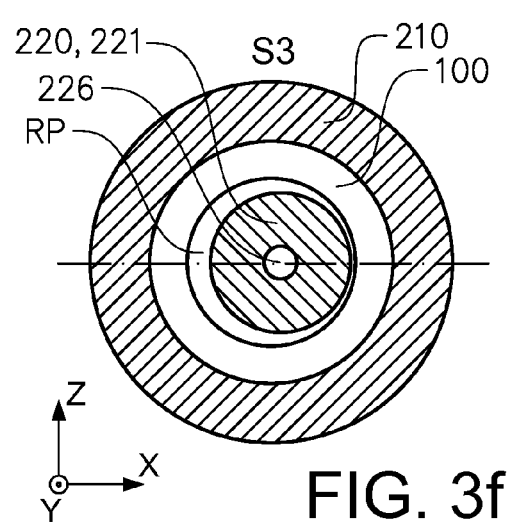
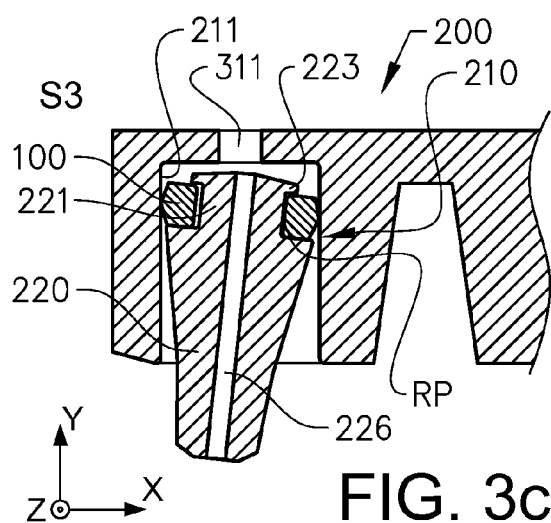
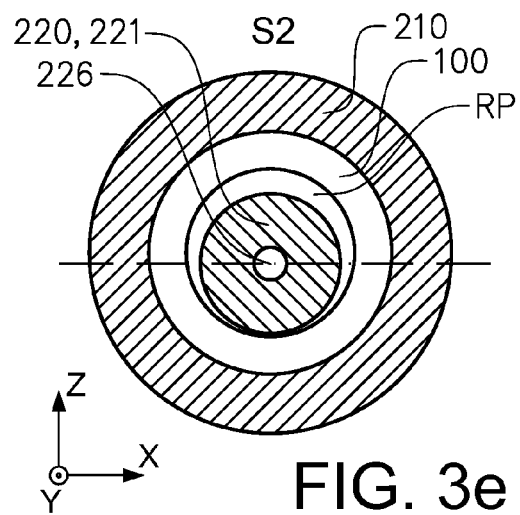
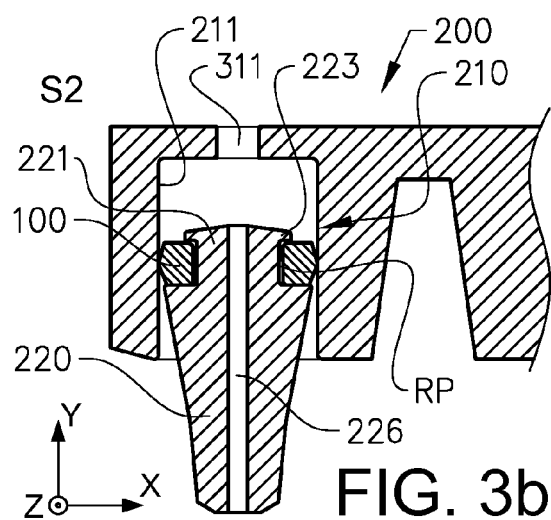
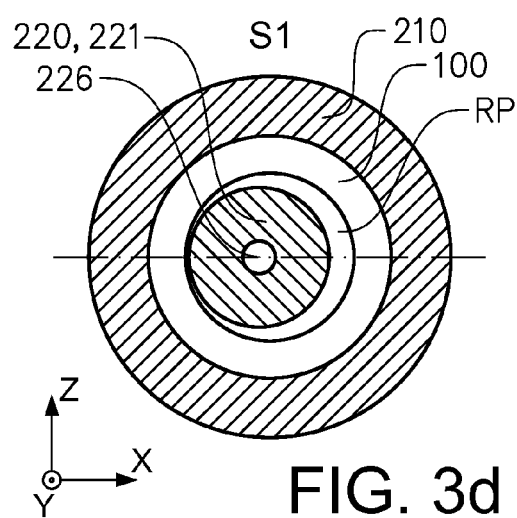
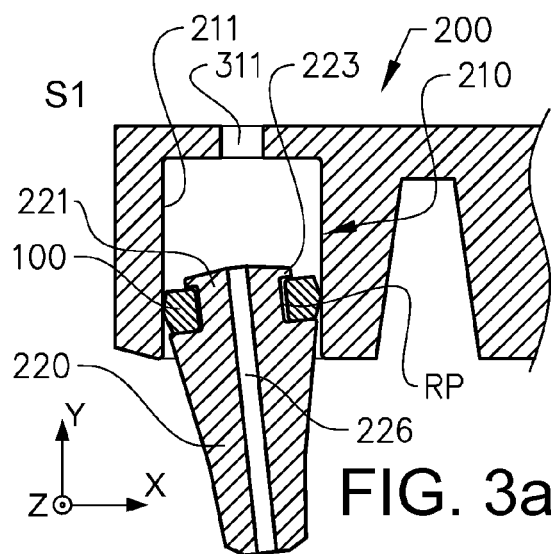


FIG. 2



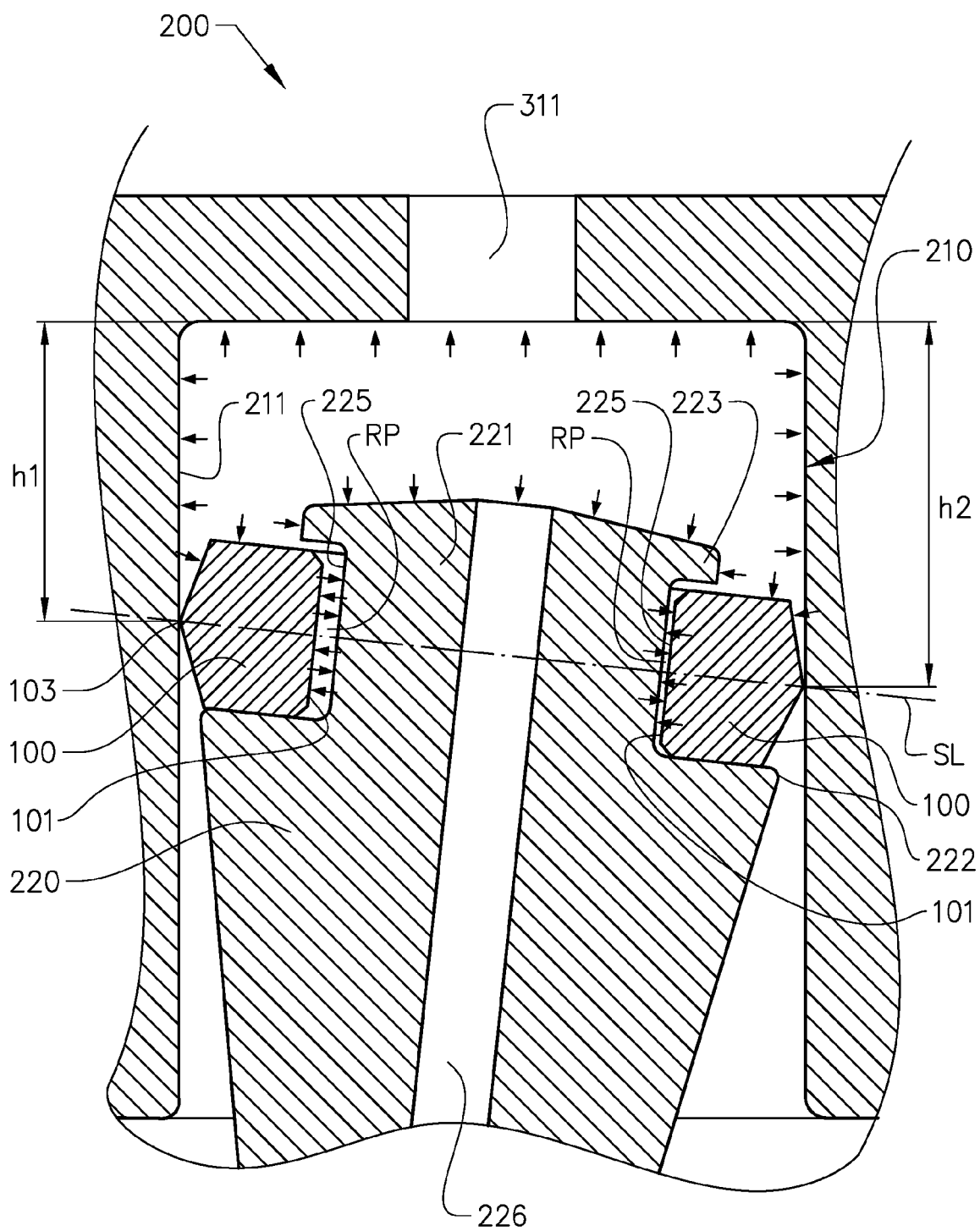


FIG. 4

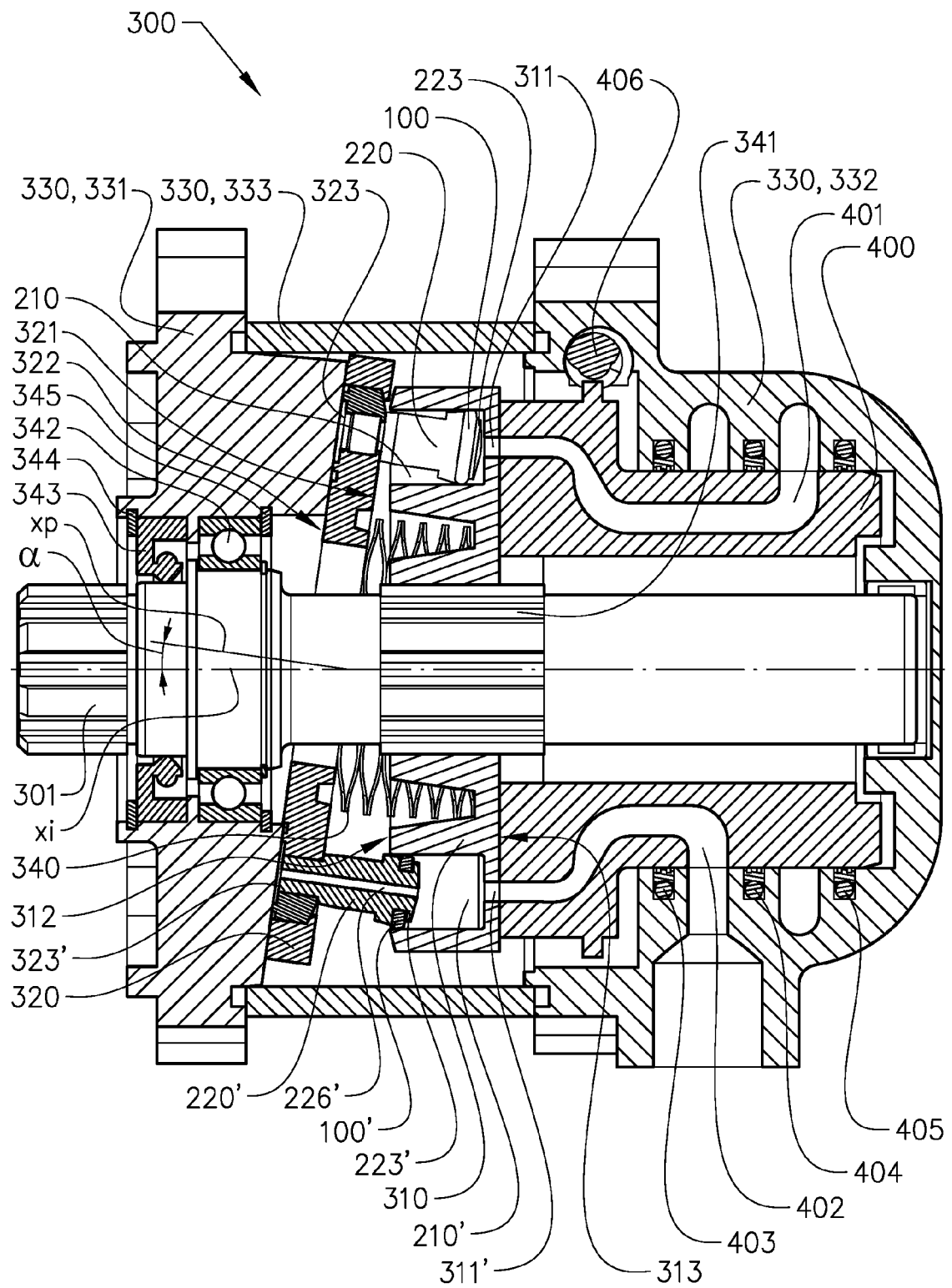


FIG. 5

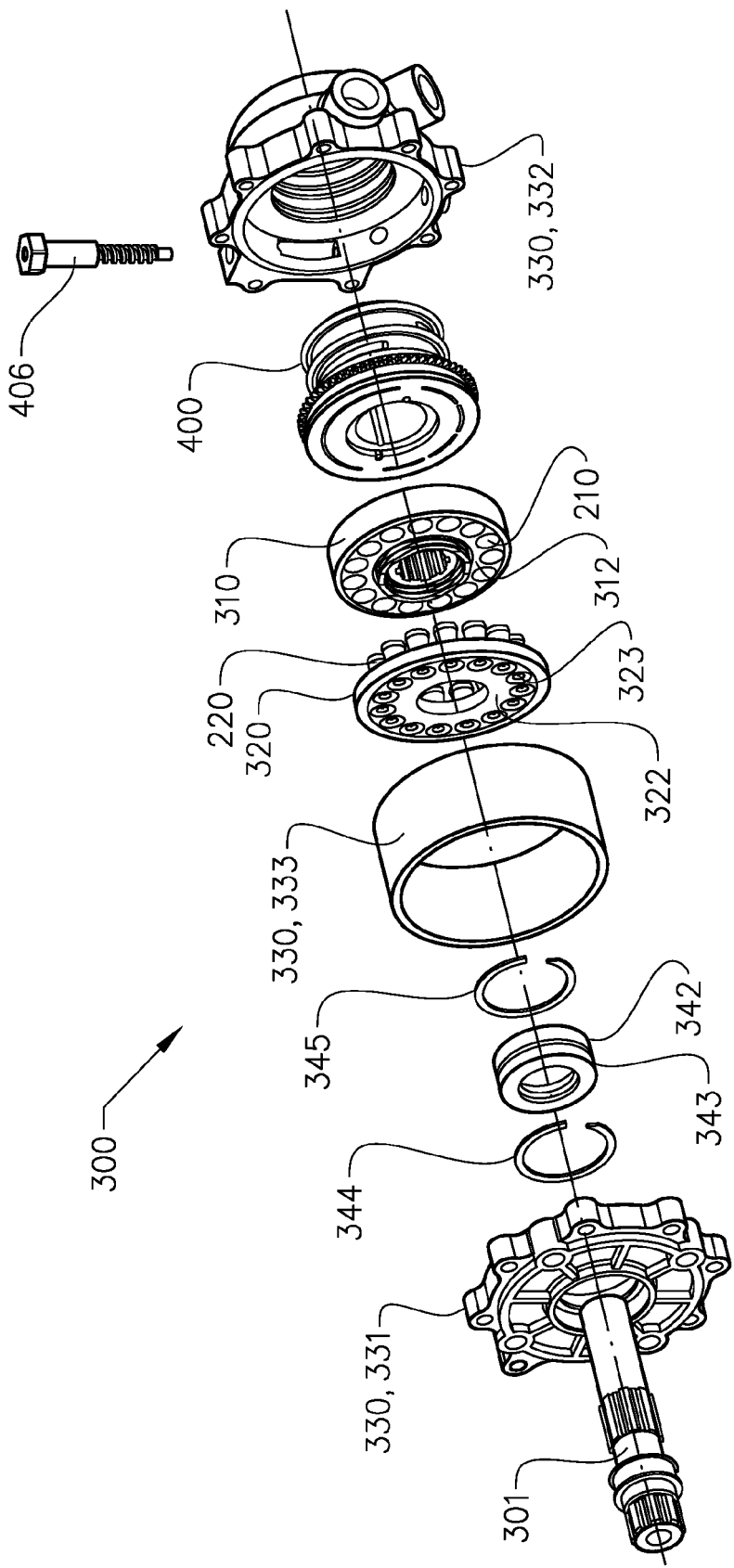


FIG. 6



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Application Number
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Place of search Munich		Date of completion of the search 3 March 2016	Examiner Pinna, Stefano
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