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(54) **AXIAL PISTON DEVICE**

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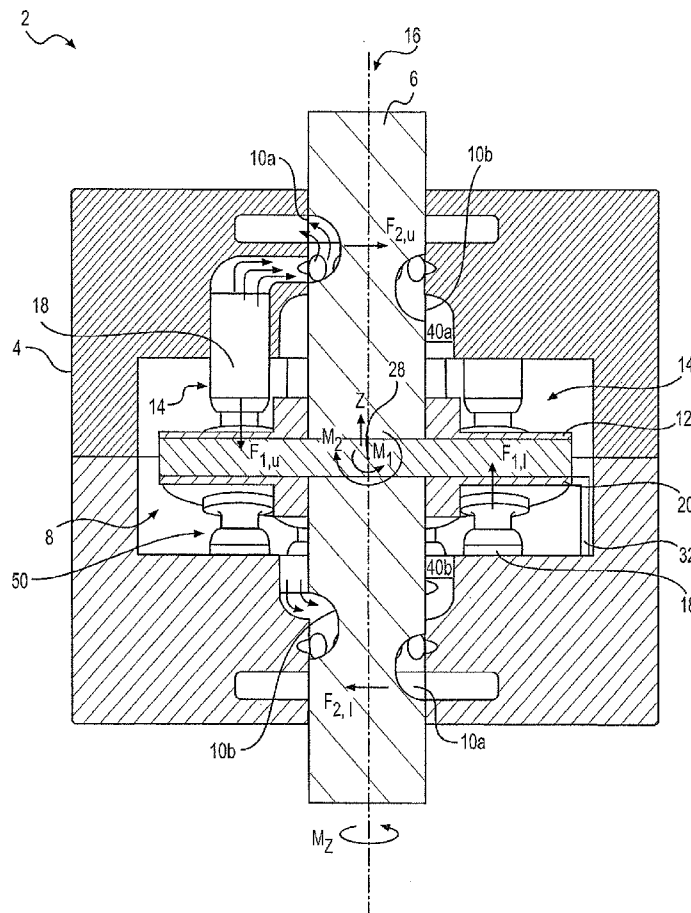(22) PCT Filed: **Feb. 5, 2015**

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(57) **ABSTRACT**

An axial piston device may be operated as a pump and includes a self-centering rotary valve. The device includes a stationary housing encompassing a shaft and the rotary valve. The rotary valve and the shaft are coupled to each other. Upon rotation, the rotary valve self-centers as a result of elimination of moments and forces within the pump. The inventive pump is a piston device. The valve is within a valve bore, which is a part of a manifold. A shaft is within the manifold and the shaft is attached at its distal end to a planar surface of the rotary valve. The shaft has a first axis of rotation and the rotary valve has a second axis of rotation. During operation of the pump, the first axis is often times offset from the second axis. The pump operates via a swashplate with reciprocating pistons while the housing remains stationary.
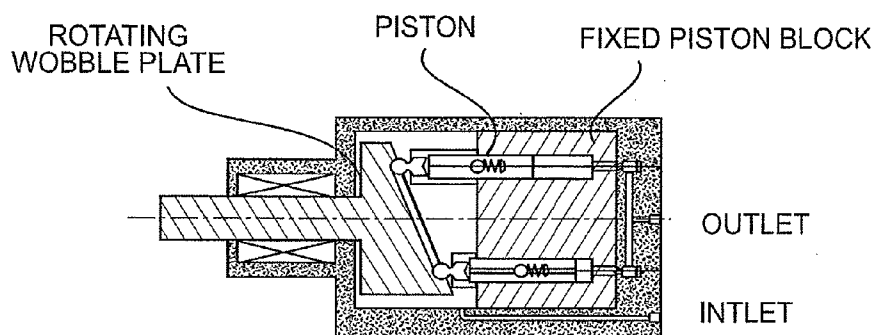
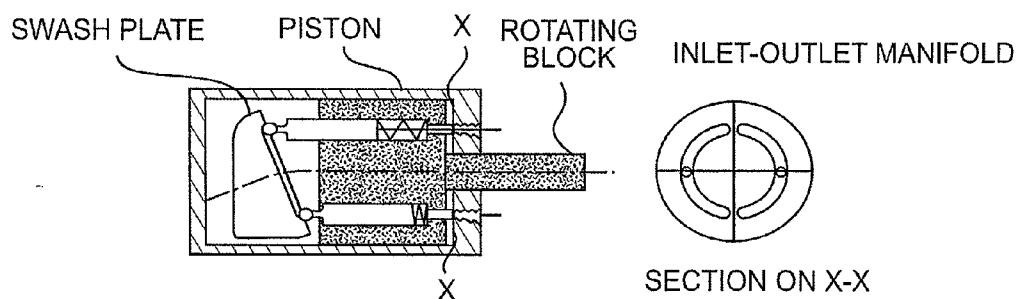ROTATING   PISTON   FIXED PISTON BLOCK
WOBBLE PLATE

OUTLET

INTLET

FIG. 1
(PRIOR ART)

SWASH PLATE   PISTON   X   ROTATING   INLET-OUTLET MANIFOLD
                                BLOCK

SECTION ON X-X

X

FIG. 2
(PRIOR ART)

LOAD   FLULD
                        WEDGE

                        PRESSURE
                        WEDGE
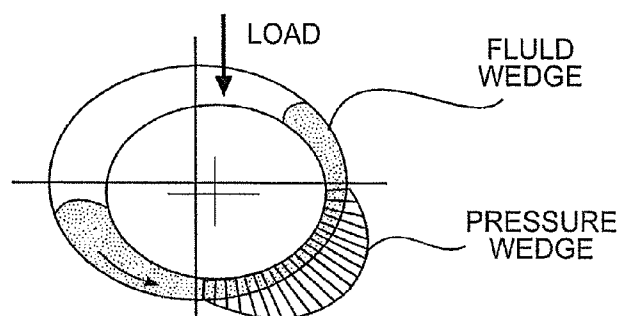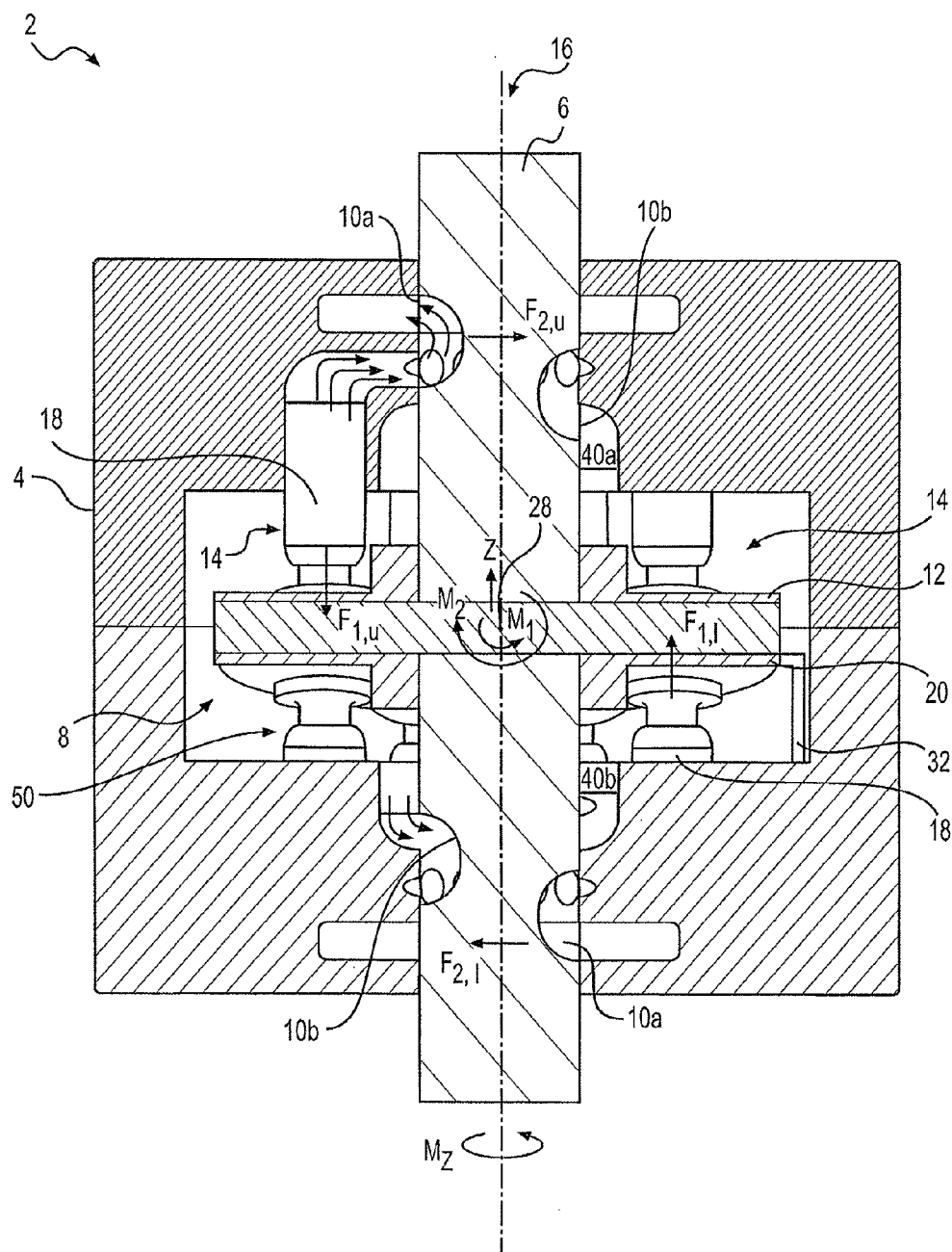
FIG. 3

FIG. 4
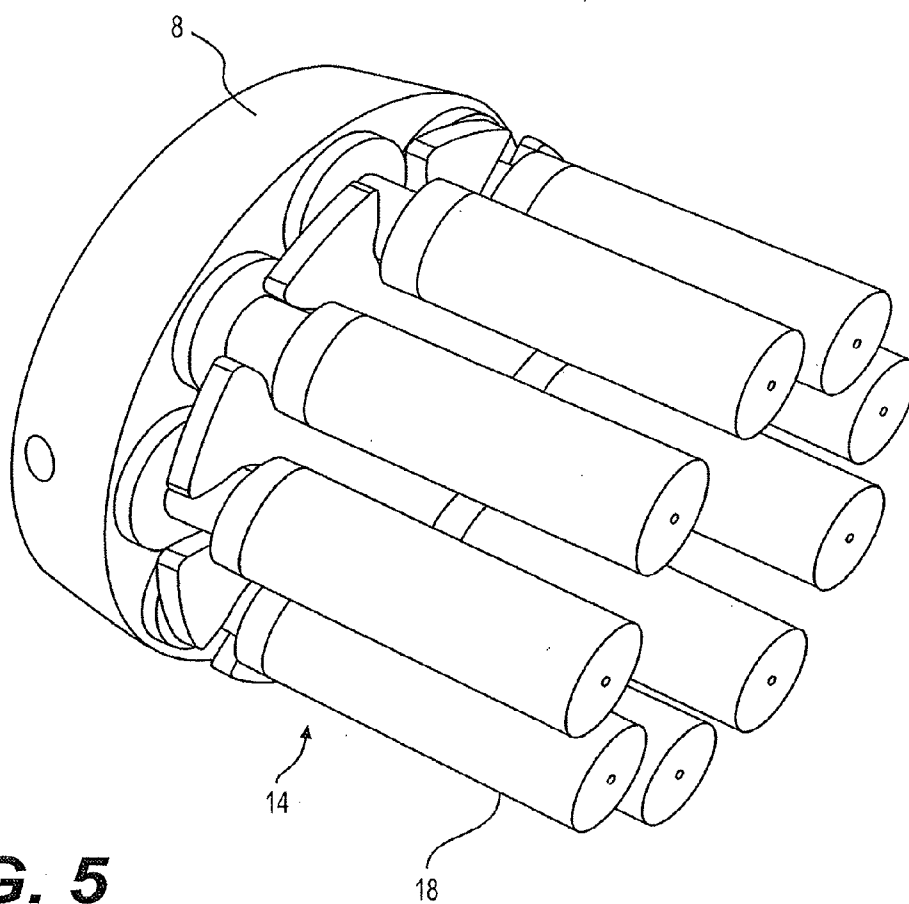
FIG. 5
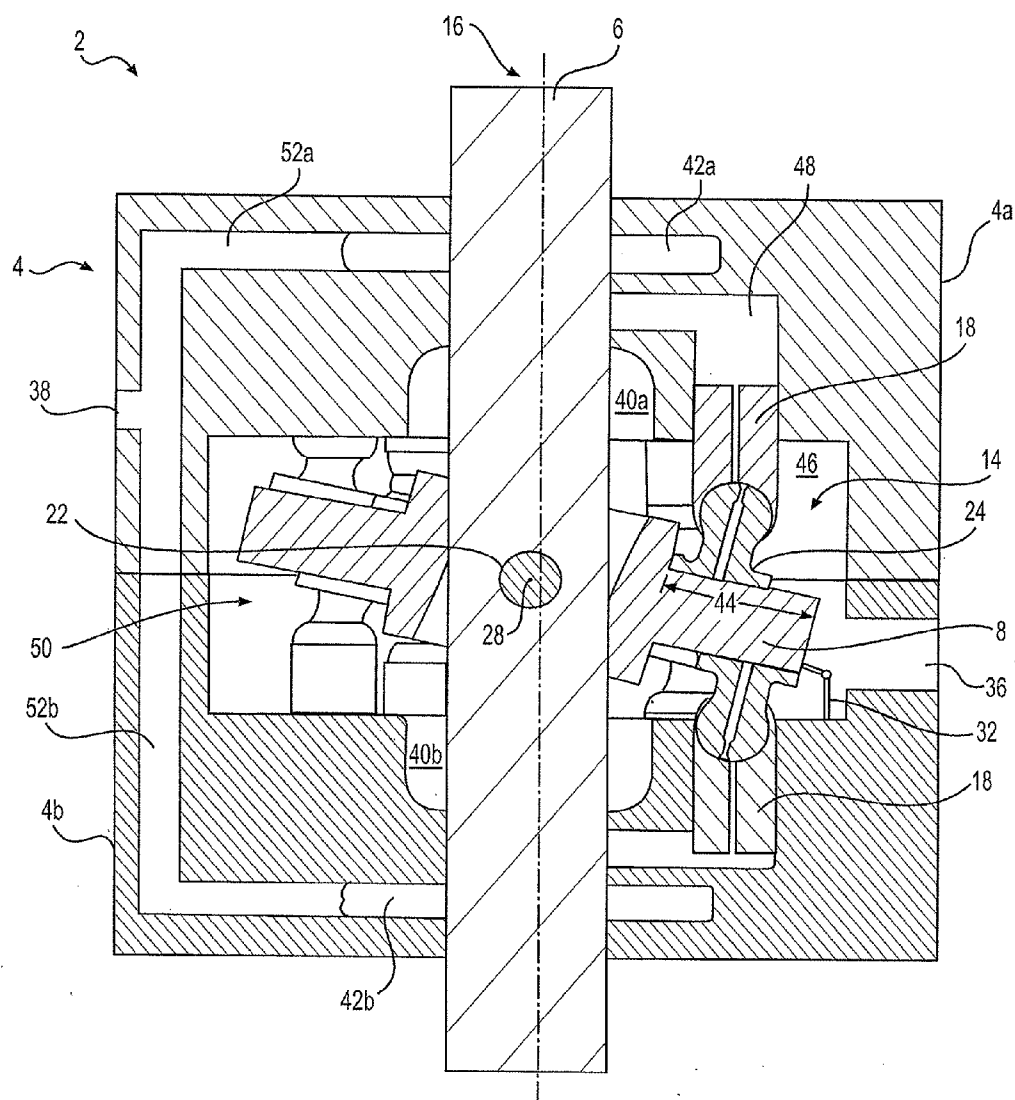
FIG. 6
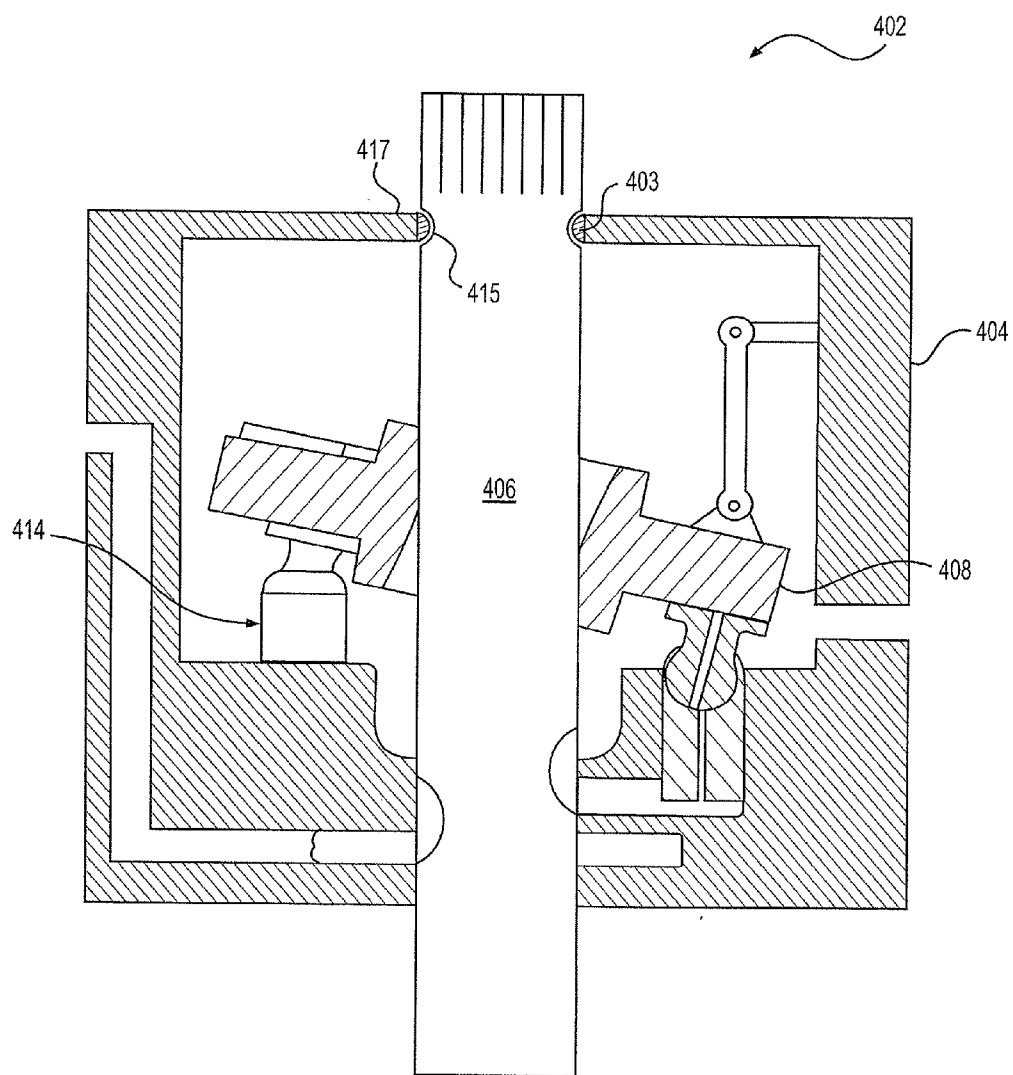
FIG. 7

FIG. 8

FIG. 9

FIG. 10

FIG. 11A

606b   618   546

614

602

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600   518

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610

612
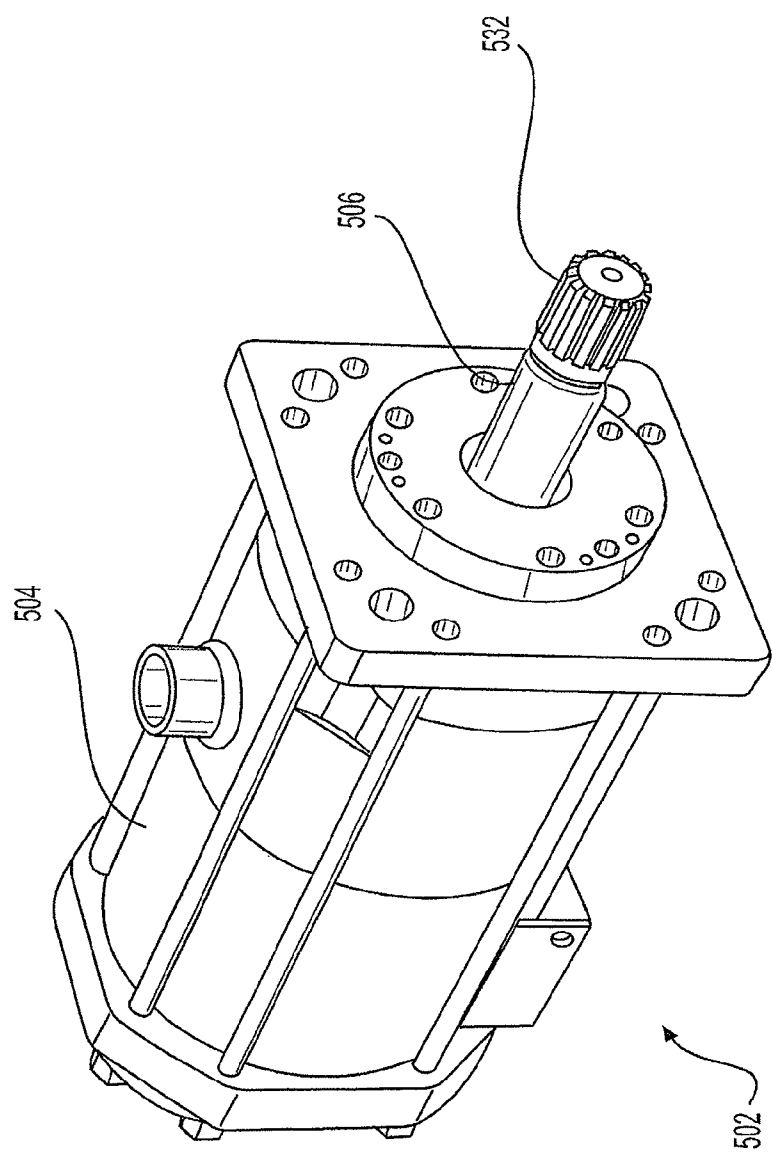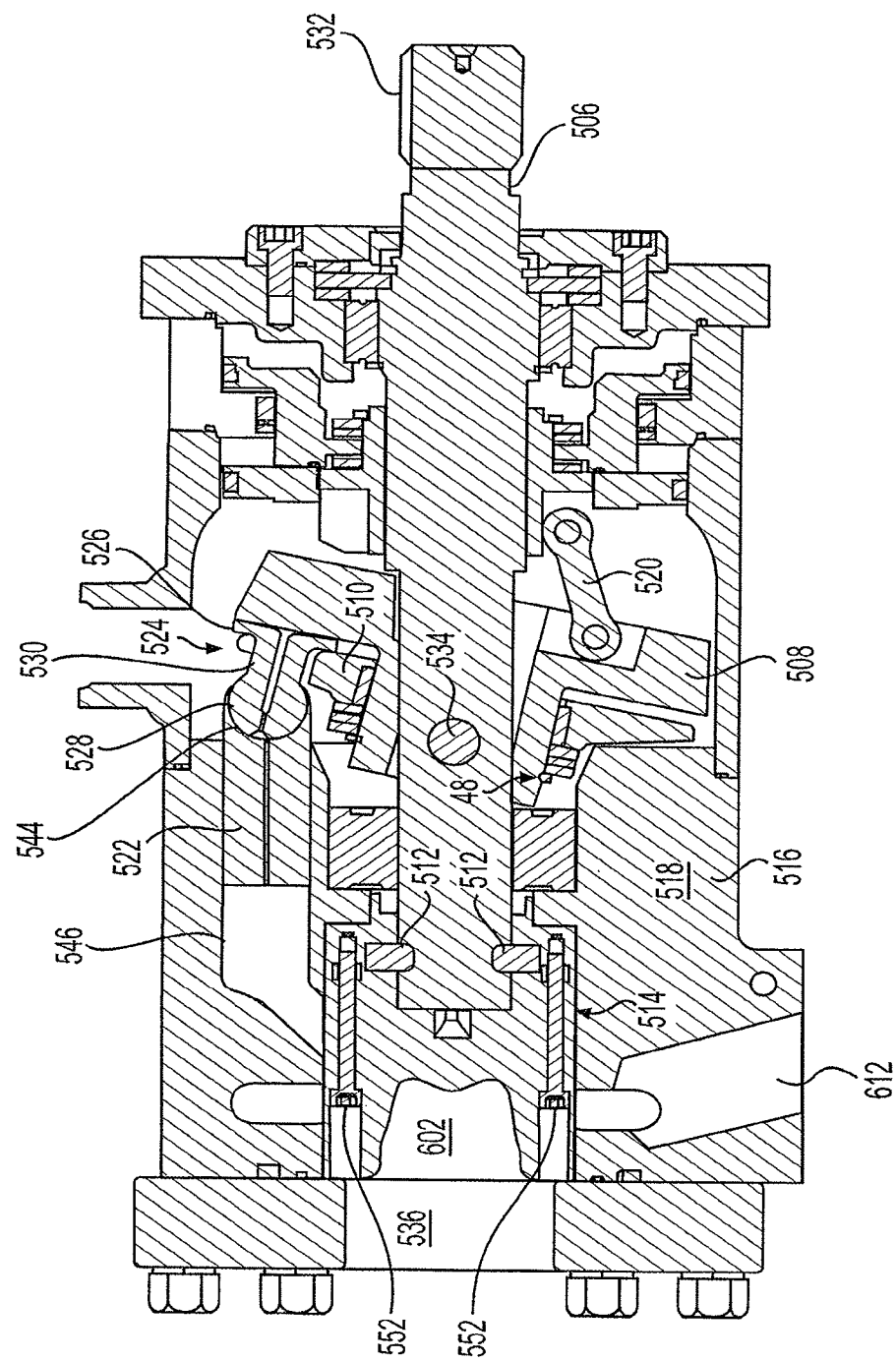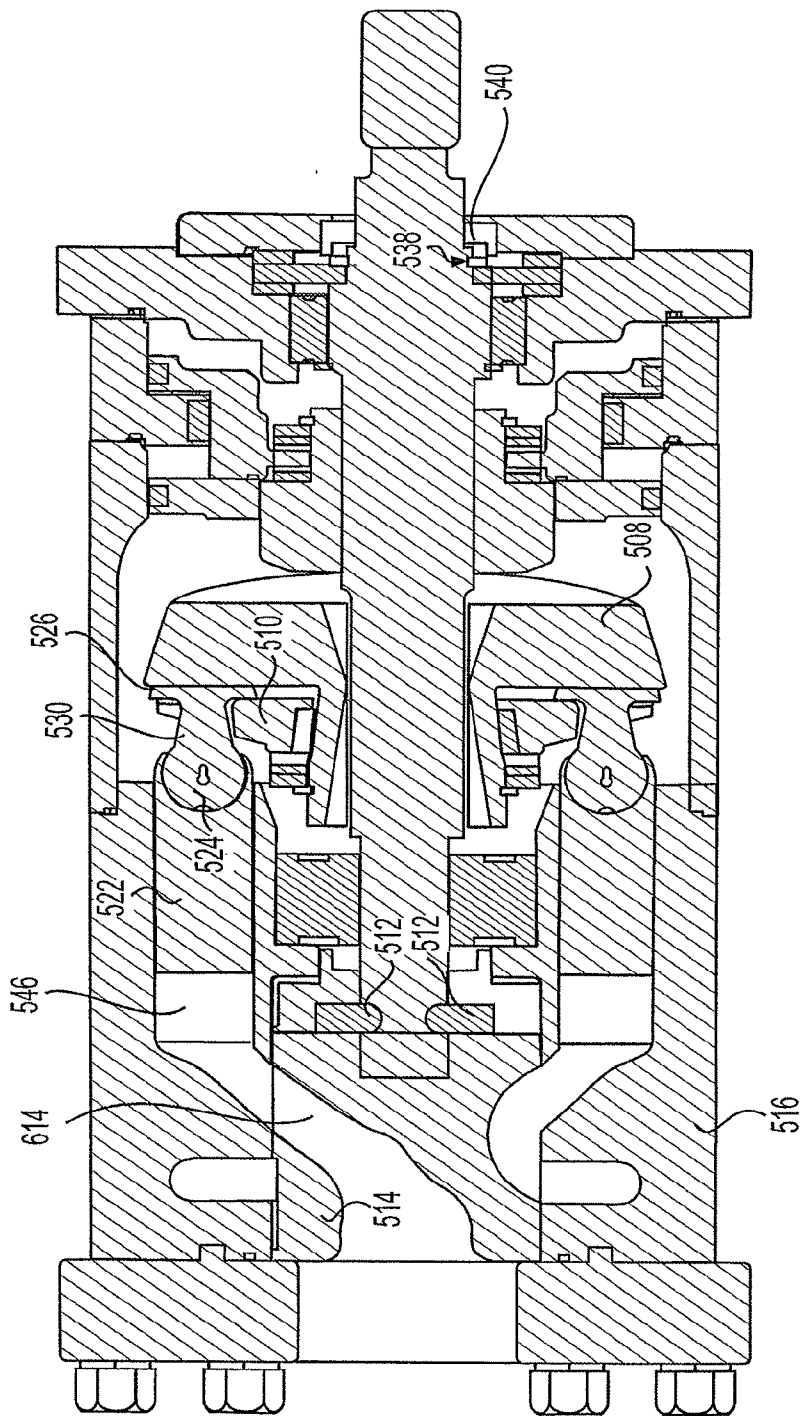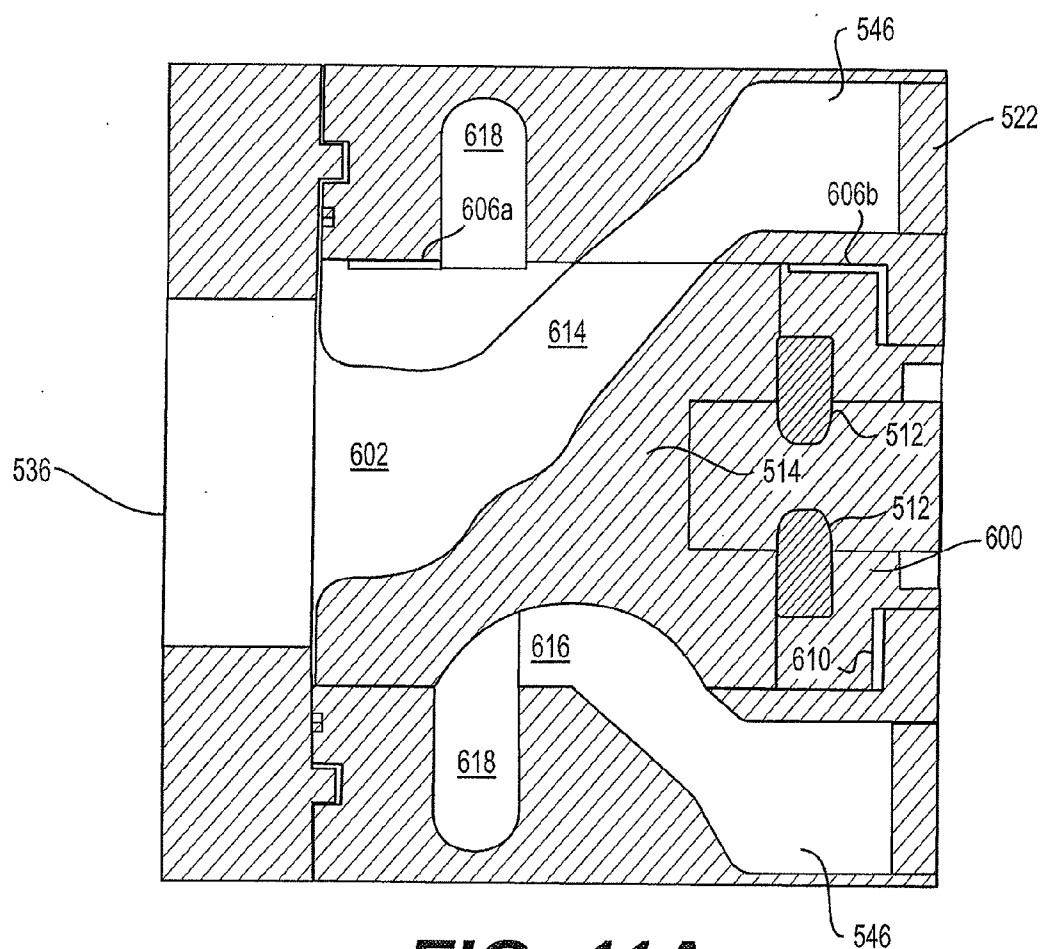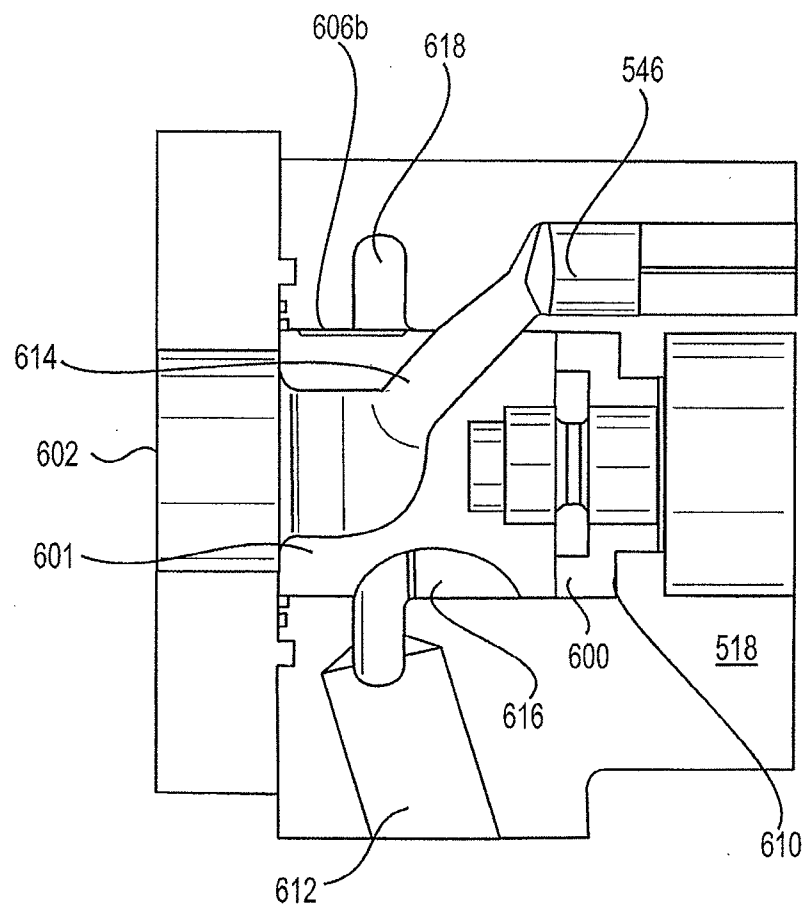
FIG. 11B

FIG. 12

FIG. 13

FIG. 14
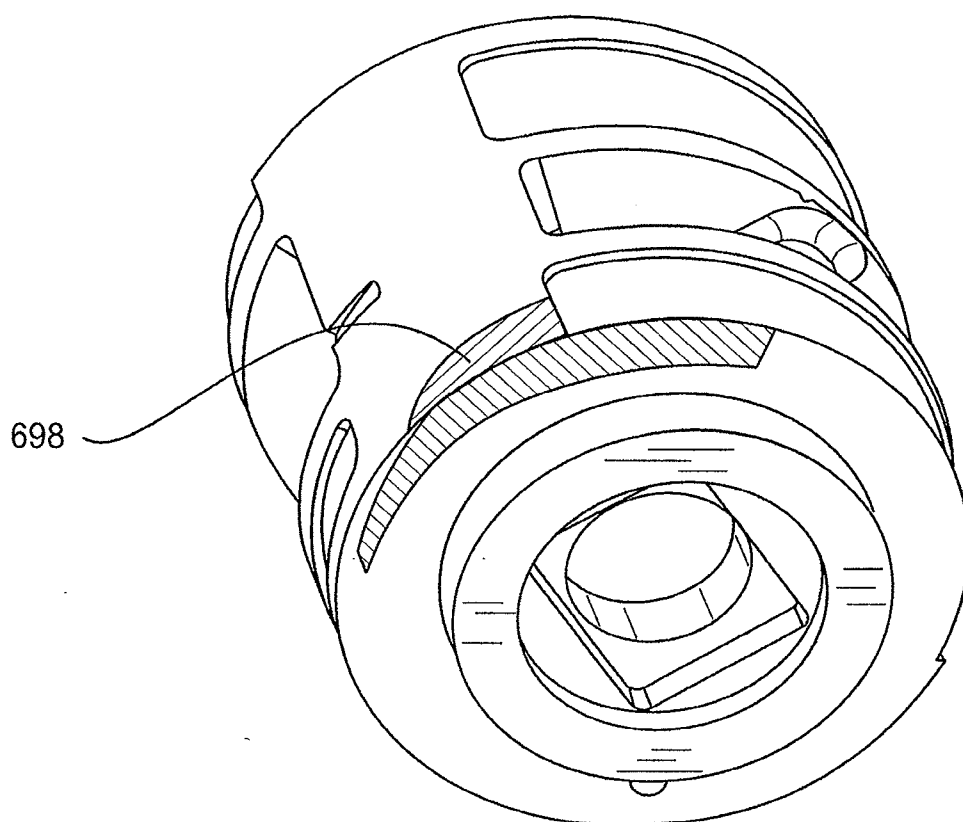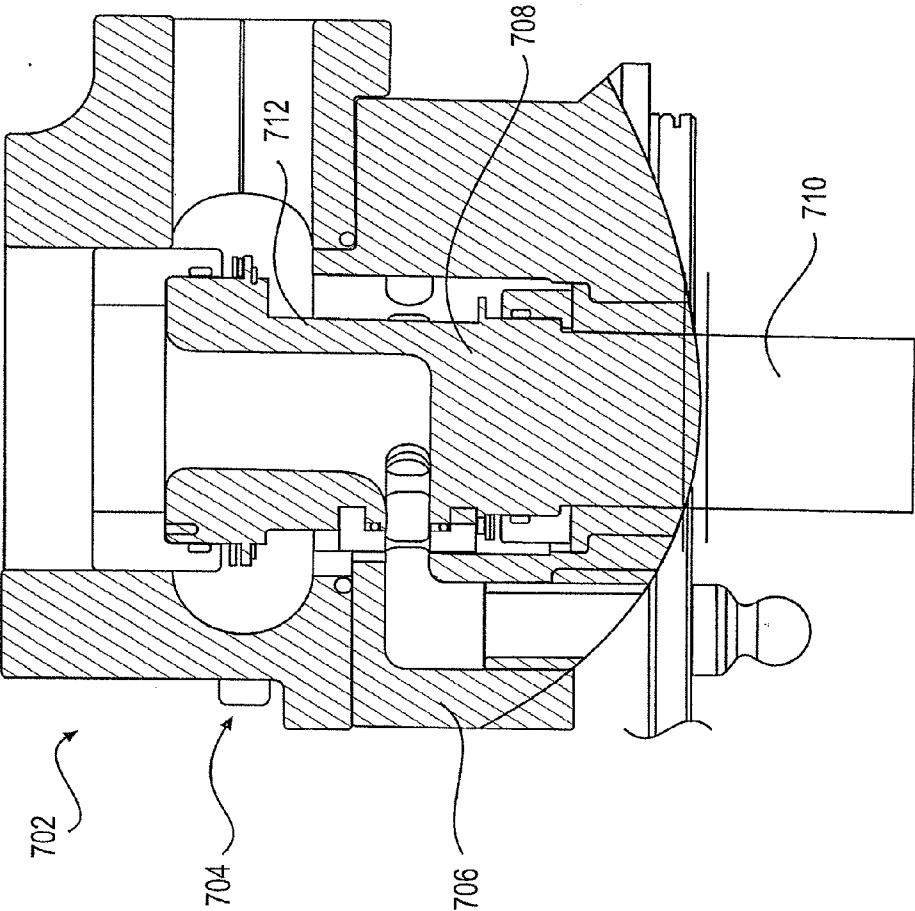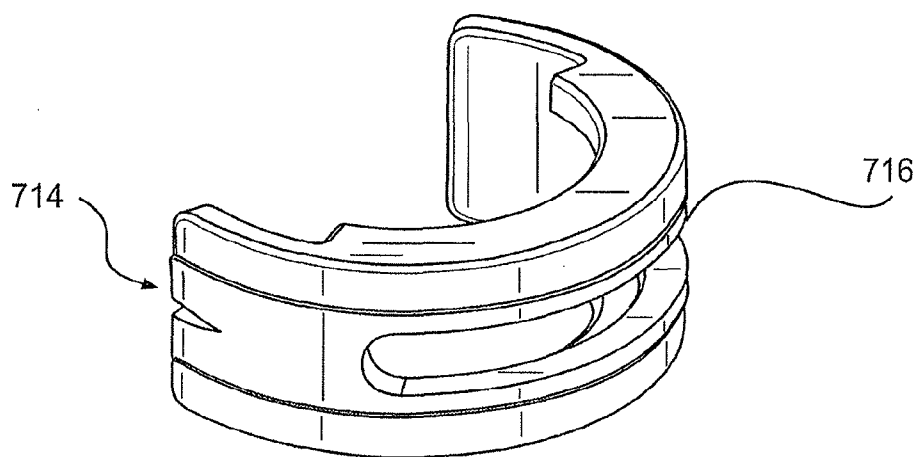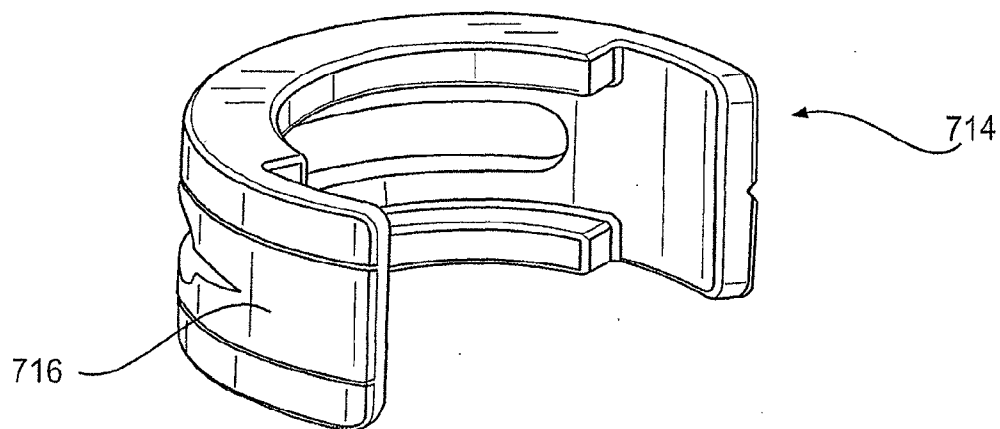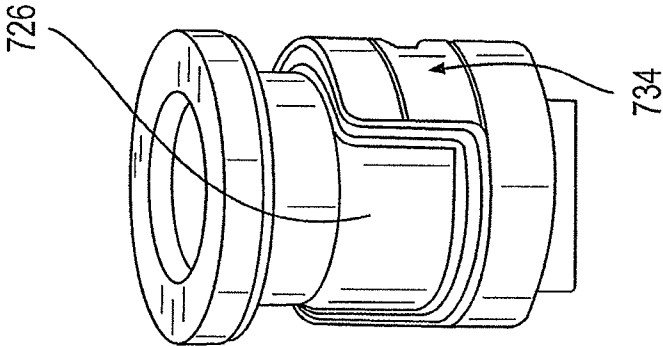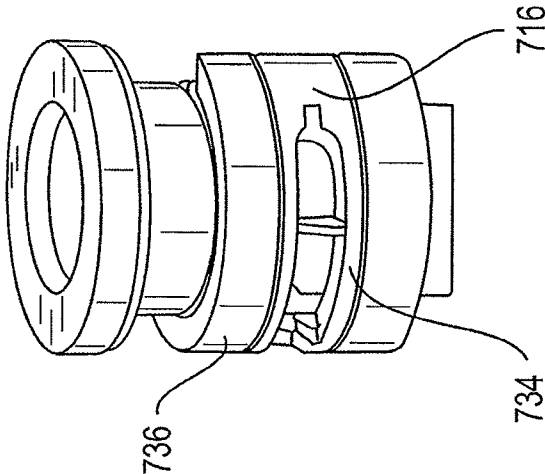
698

FIG. 15

712

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FIG. 16

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FIG. 17A

714

716

FIG. 17B

726

734

FIG. 17E

716

734

736

FIG. 17D

FIG. 18

FIG. 19

FIG. 17C

FIG. 20

FIG. 21

FIG. 22B

822

FIG. 22A

808

FIG. 23

F_{NET}

Z

r

836

808

FIG. 24

830

816

800

812

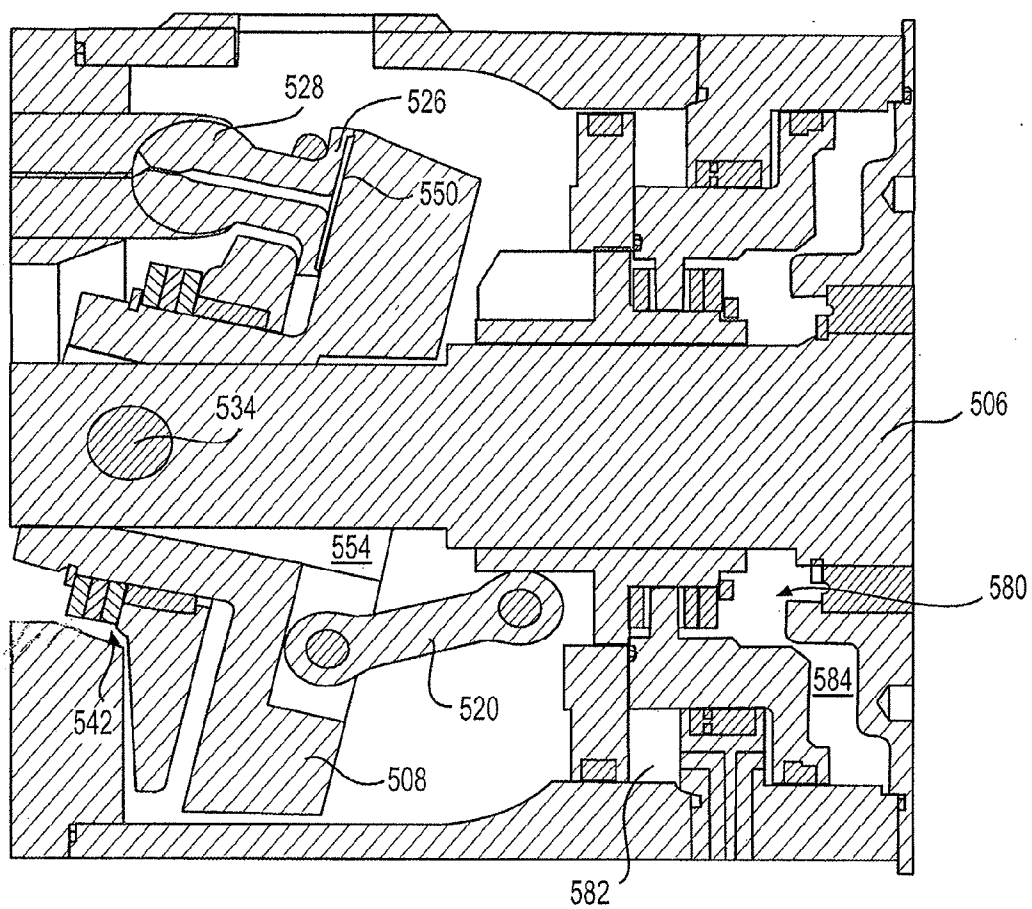h_{max}

e

r

F_{NET}

810

h_{min}

w

802

FIG. 25

FIG. 26

FIG. 29

FIG. 28

FIG. 27

FIG. 30
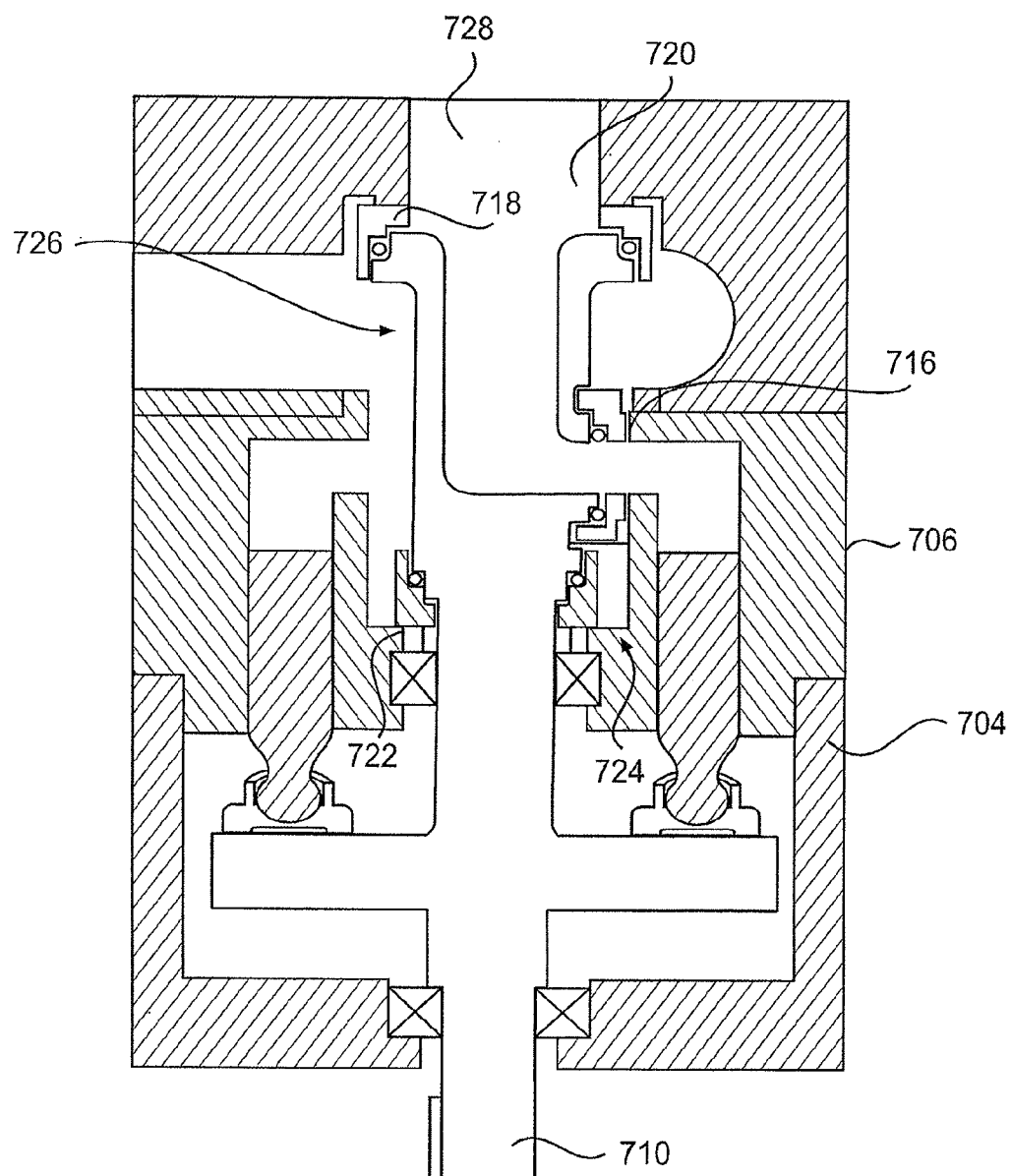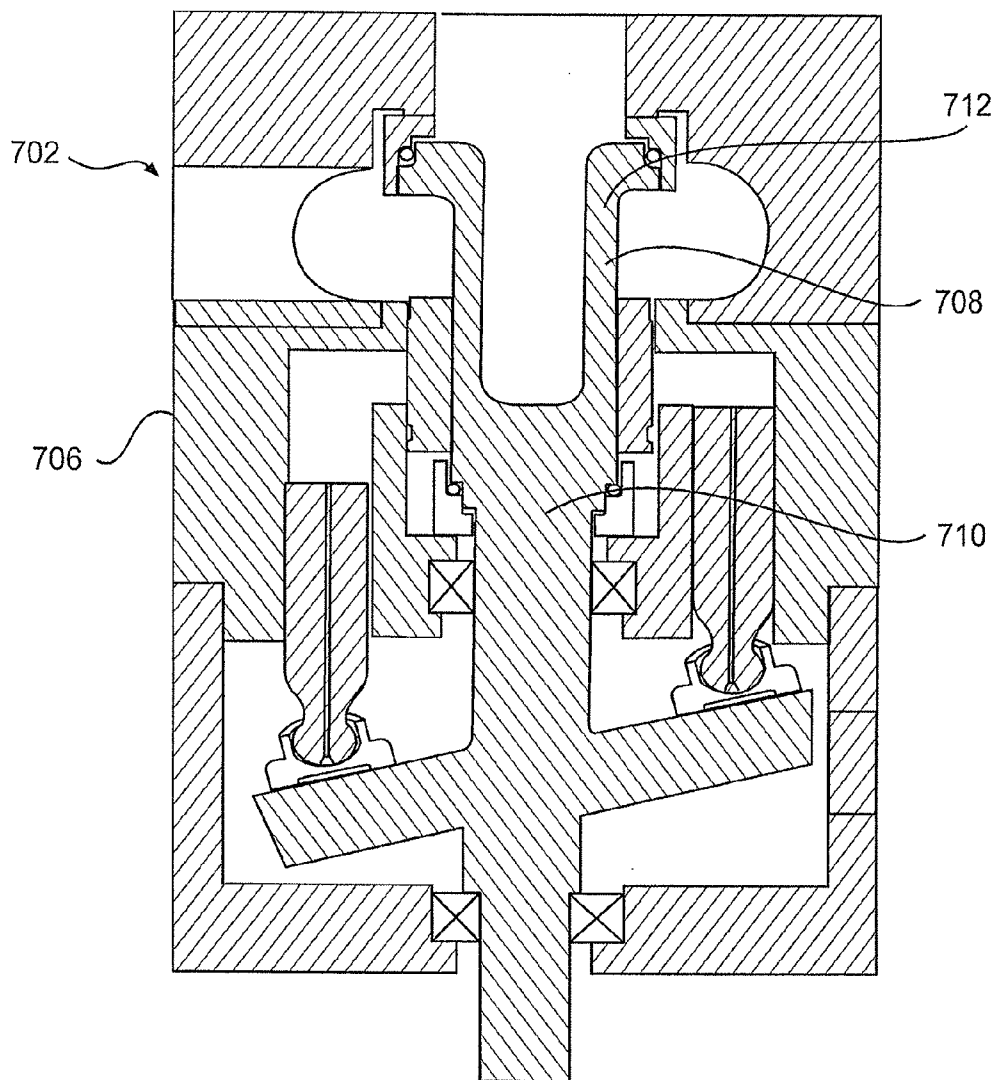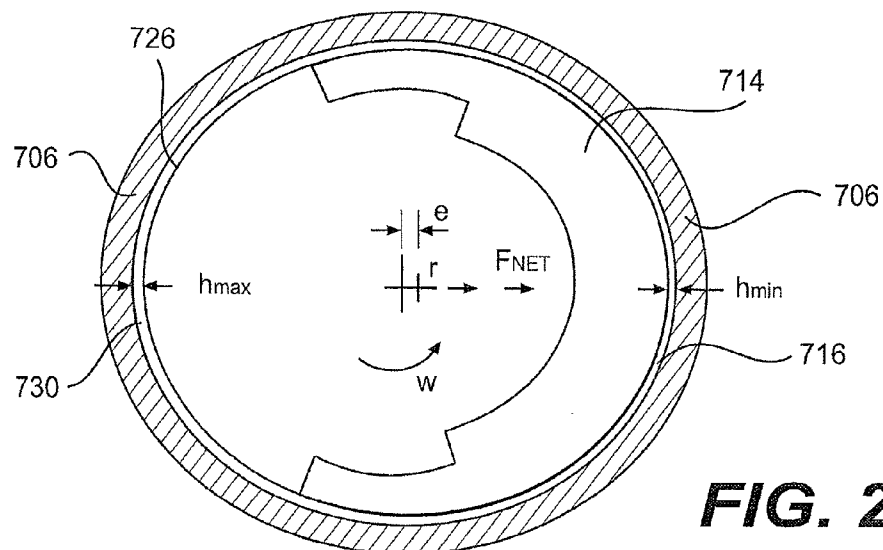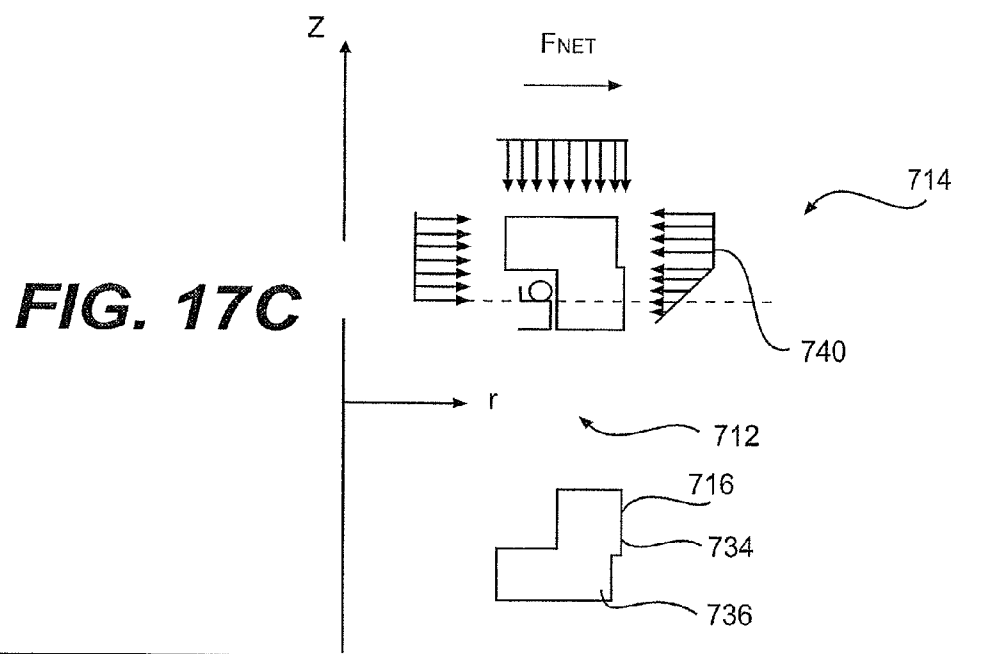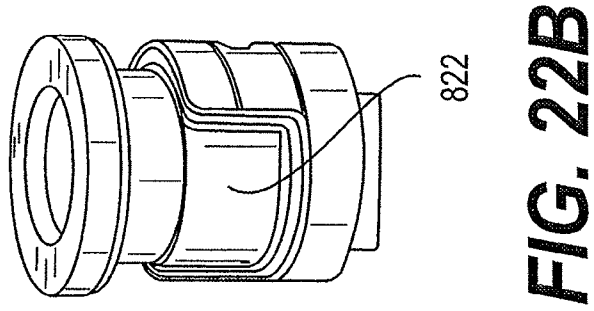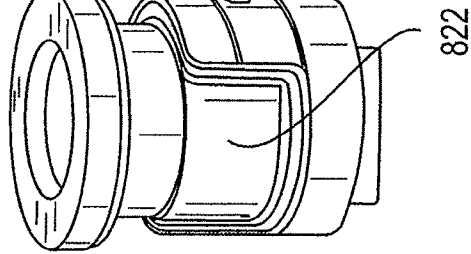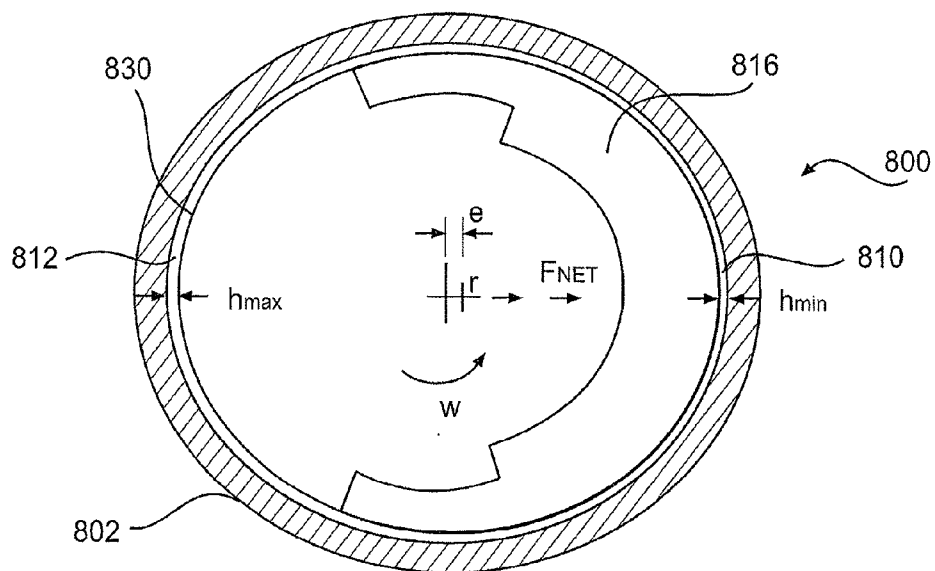
FIG. 31

FIG. 32

AXIAL PISTON DEVICE

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application is the U.S. National Phase Application of PCT International Application No. PCT/US2015/014630, filed Feb. 5, 2015, which claims the benefit of U.S. Provisional Patent Application No. 61/937,166 filed Feb. 7, 2014 and U.S. Provisional Patent Application No. 62/093,146 filed Dec. 17, 2014, the contents of such applications being incorporated by reference herein.

TECHNICAL FIELD

[0002] This disclosure relates to fluid power and more particularly relates to the displacement of fluid via an axial piston device.

BACKGROUND OF THE INVENTION

[0003] Axial piston technology is typically embodied in non-rotating cylinder block and rotating cylinder block devices (e.g. rotating cylinder block style hydraulic pump/motor; commonly referred to as a "rotating group"). A conventional non-rotating cylinder block pump is shown in FIG. 1 and a rotating cylinder block pump is shown in FIG. 2.

[0004] Non-rotating cylinder blocks experience problems with forces on the pump's shaft and swash plate. Those forces have to be carried by bearings. Unfortunately, forces acting on the pump's shaft and swash plate are unbalanced. If left unchecked, the unbalanced forces can cause rapid pump wear. Due to the need for large bearings to handle the unbalanced loads in a non-rotating cylinder block, pumps must be extremely large to accommodate the large bearings.

[0005] These problems do not exist in rotating cylinder blocks as the cylinder block absorbs most of the forces that are unbalanced in the non-rotating cylinder blocks. Rotating cylinder block devices can therefore be much smaller than non-rotating cylinder block designs. A rotating cylinder block design has an inherent deficiency in that it has a large rotational mass inertia. This manifests as increased power losses, especially during changes in rotational speed. Devices constructed within this technology require an additional component to house the rotating cylinder block. This requires devices that, although smaller than prior devices, are still both larger and heavier than what is desired. Both properties are detrimental to packaging and efficiency.

SUMMARY OF THE INVENTION

[0006] The present pump intakes fluid through a non-rotating cylinder block and passes it, in some embodiments, through a rotary valve. The rotary valve is in constant fluid communication with the inlet and outlet cavities, as well as a plurality of piston bores within the cylinder block.

[0007] The pump features balanced forces on a swash plate and within the pump in general, while permitting rotation of the swash plate. Distribution of fluid throughout the pump balances the unbalanced forces in the non-rotating cylinder block pump without the need for extra-large bearings. An embodiment of the present pump includes a double sided swash plate, which allows for the balance of all swash plate 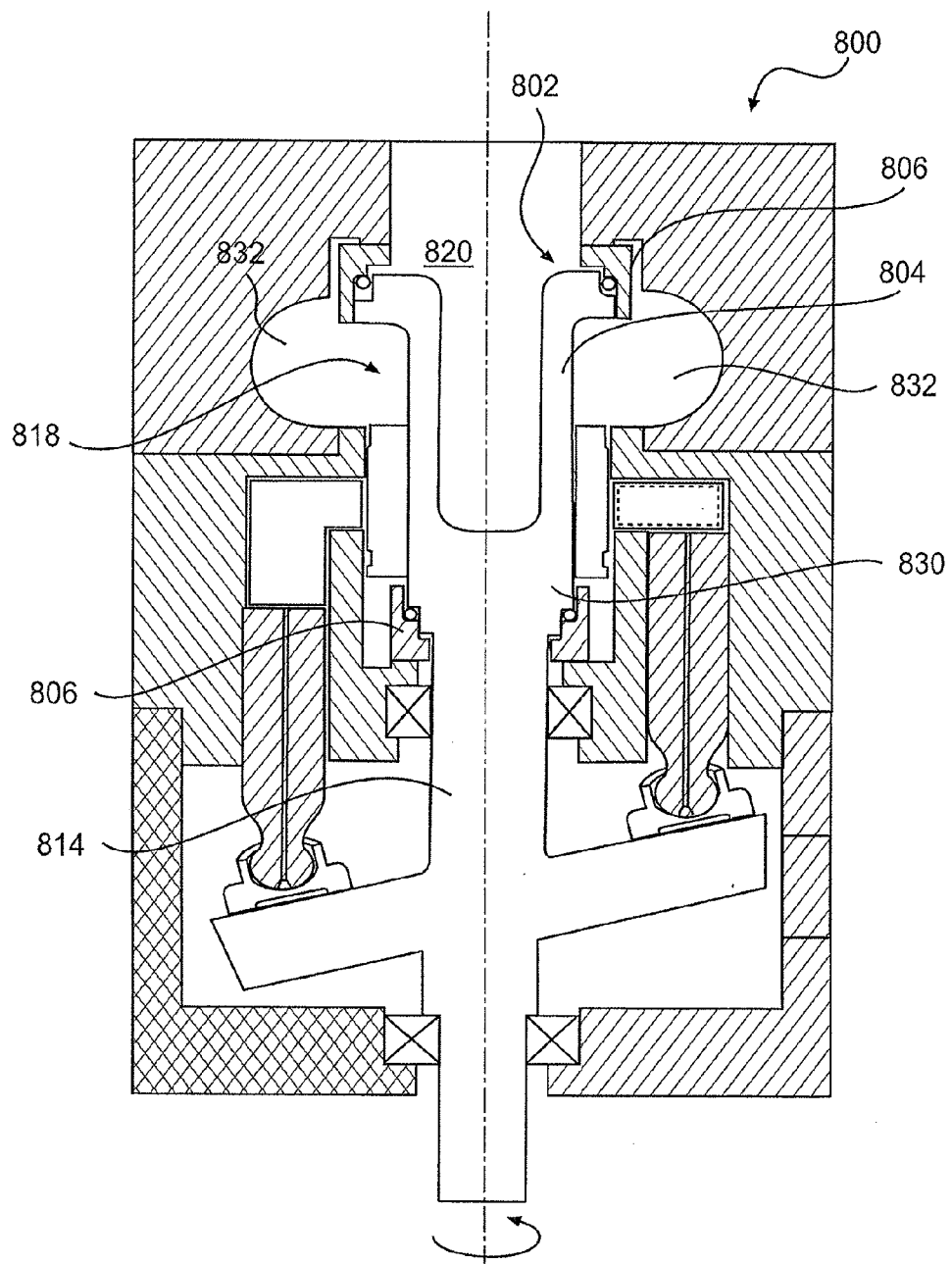forces. The size of the present pump is no longer limited by the presence of rolling element bearings during operation. The pump is capable of handling a wide range of operating conditions and shows great advantages in size reduction when used in high pressure and/or high displacement pumps. The design and function of prior art pumps is limited by the need for very large bearings. This is no longer a problem in the present design.

[0008] An exemplary embodiment of the inventive hydraulic pump is one having a plurality of passages in the shaft. Passages eliminate need for a rotary valve as the functions of the rotary valve and the shaft are performed by a single component. An embodiment of the hydraulic pump includes a swash plate, which has a first side coupled to a first set of a plurality of slipper shoes, and a self-centering shaft, w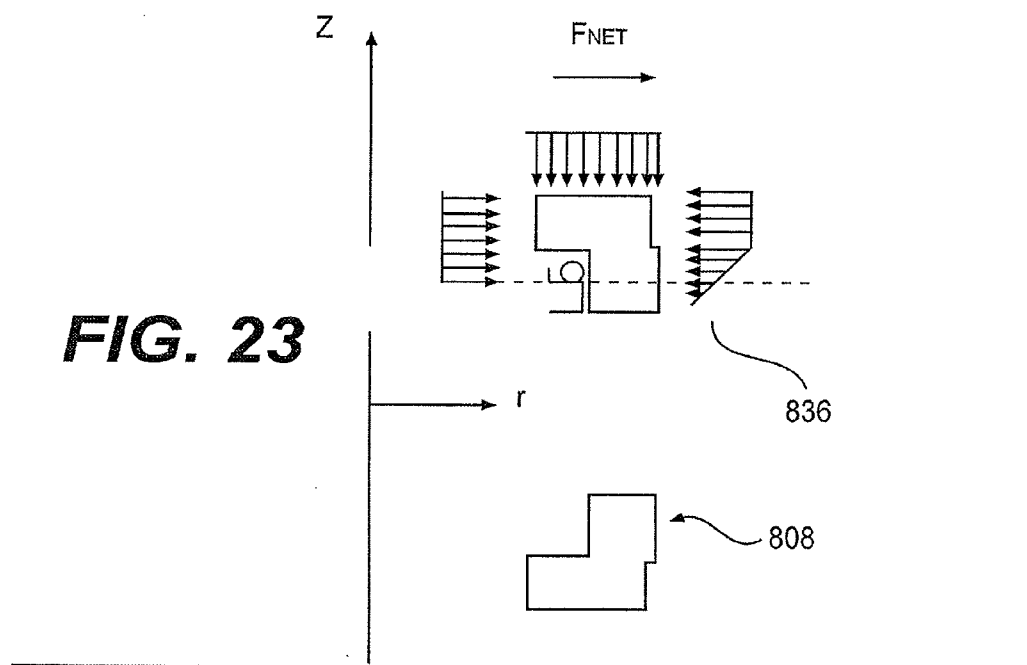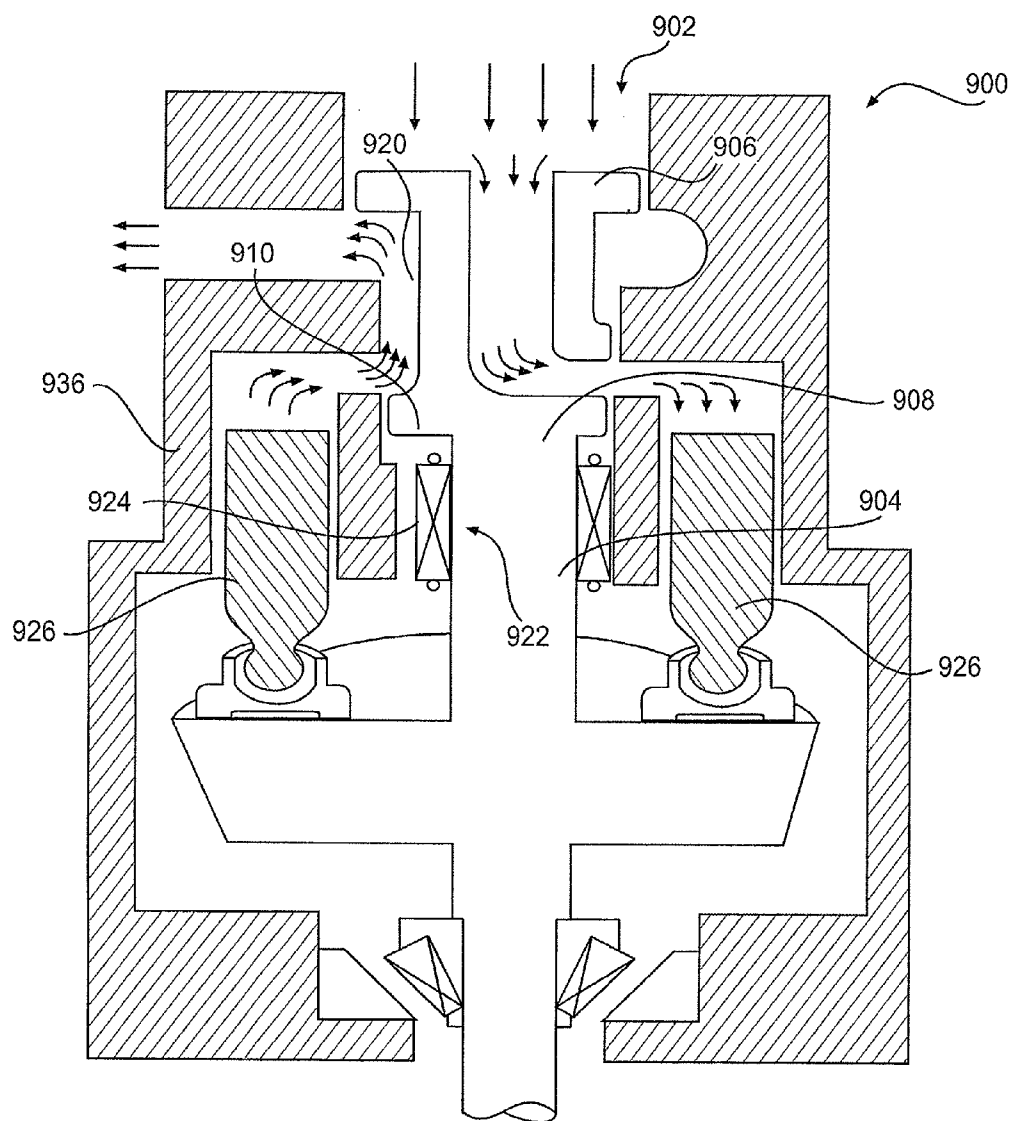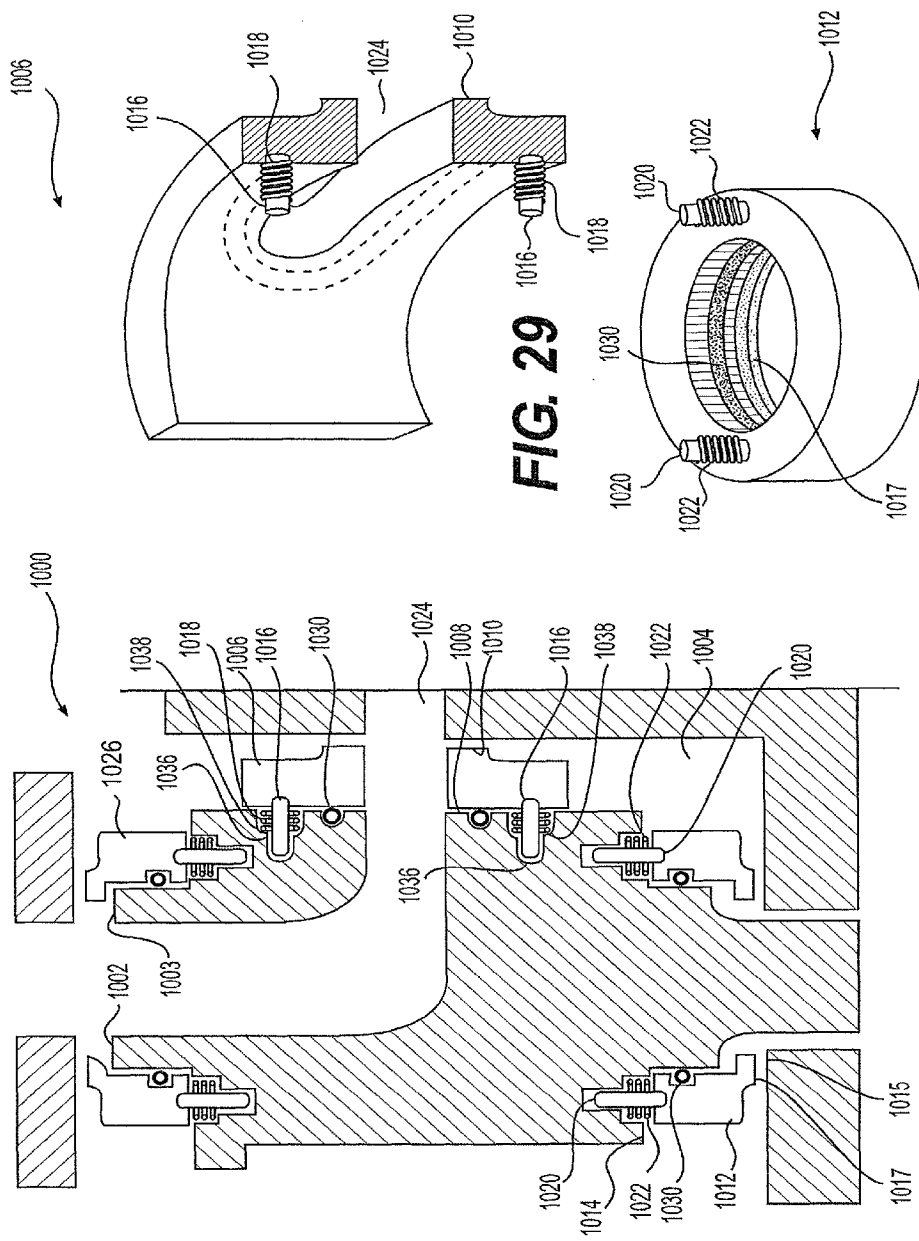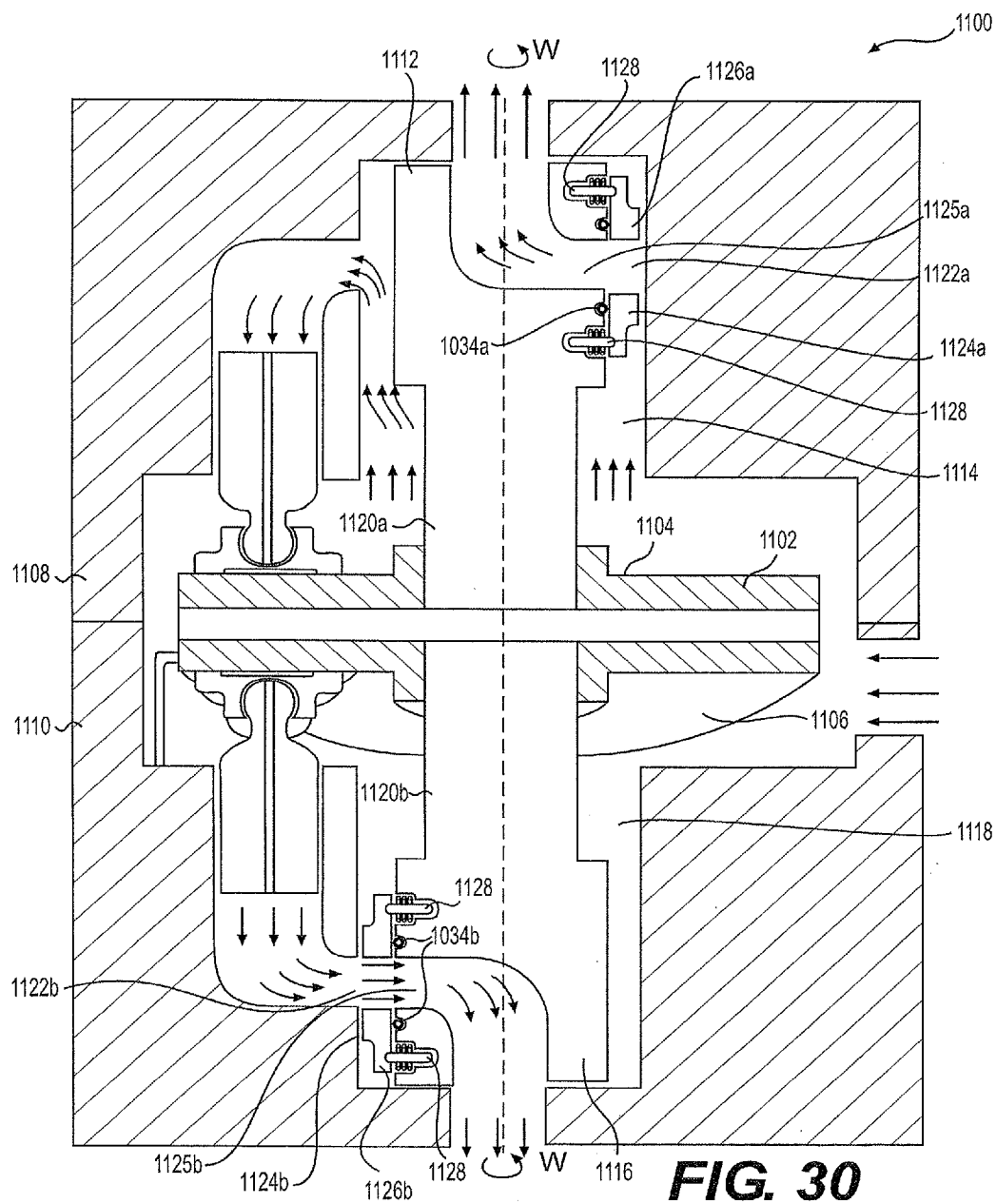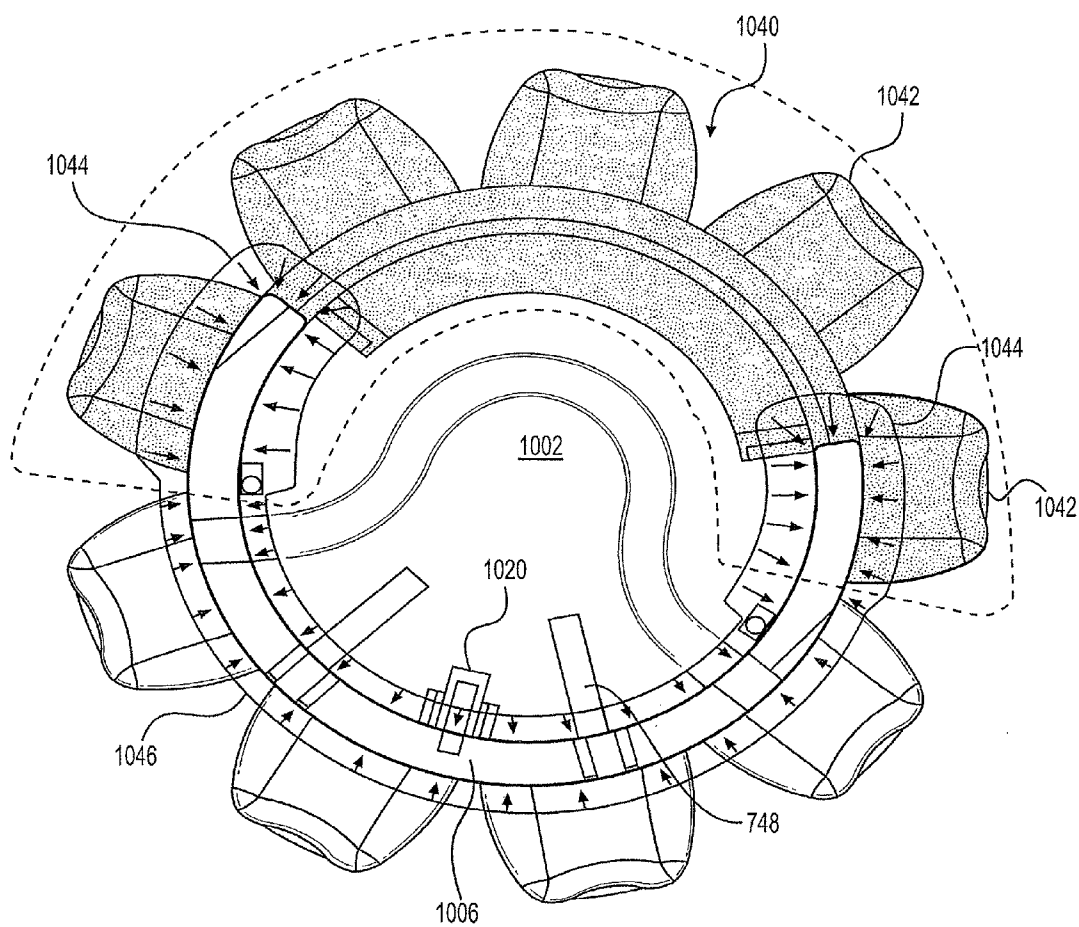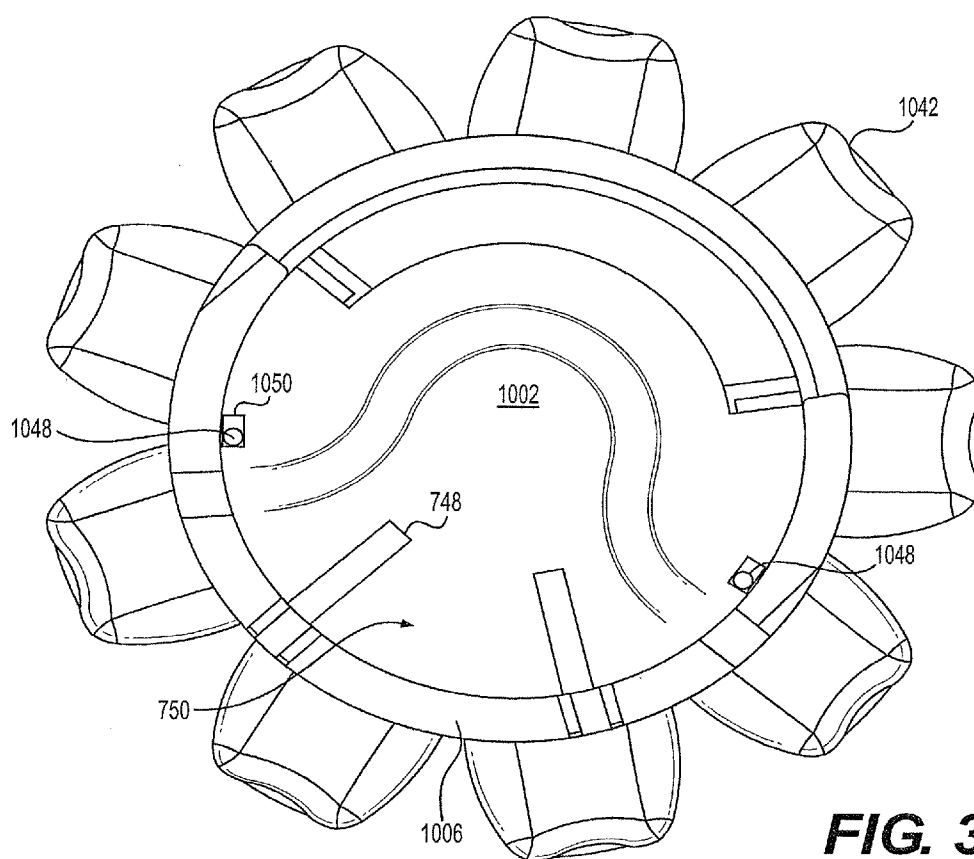hich is rotatable about its own axis within at least one stationary cylinder block. The self-centering shaft has a plurality of passages that receive, direct, and exhaust fluid. The self-centering shaft is centrically and eccentrically rotatable during operation of the pump. Fluid pressure on the passages of the shaft balance forces and/or moments on the shaft. This pressure helps to center the shaft.

[0009] The swash plate is connected to the shaft and is configured to tilt about an axis perpendicular to the axis of the shaft. The tilting of the swash plate can be achieved by a hydraulic means, a mechanical dog-bone assembly, an electric means, a gearing system, etc.

[0010] Another embodiment involves a pump that has a self-balanced shaft. The swash plate has first and second substantially planar surfaces opposing each other. First and second pluralities of slipper assemblies are coupled to the first and second planar surfaces. Fluid pressure on the passages on the shaft balances forces in all directions. A housing encases the shaft, swash plate and slipper assemblies. The housing has at least one inlet for receiving fluid in the pump and at least one outlet for expelling fluid from the pump.

[0011] The shaft of the hydraulic pump is self-centering. An exemplary hydraulic pump would include at least one stationary one piece cylinder block and a shaft that is rotatable about its own axis within the stationary cylinder block. This embodiment of the hydraulic pump also includes a first and second plurality of pistons. Each of the plurality of pistons is contained within the cylinder block and is arcuately positioned coaxially with the shaft. The pluralities of pistons are on opposite ends of the swashplate. The at least one (or first) cylinder block encompasses the first plurality of pistons and a second stationary cylinder block encompasses the second plurality of pistons.

[0012] Fluid flow throughout the pump, which is produced by the swashplate driving the reciprocation of pistons, creates a force and/or moment on the shaft. The plurality of passages are configured on the shaft such that the forces and/or moments on the shaft are balanced.

[0013] The pump can be configured such that the shaft of the hydraulic pump has a valve integrated into it and the passages along the shaft do more than the minimal function of ensuring that the shaft is centered. The passages are also provided to receive, direct and exhaust fluid flow at a plurality of advantageous locations along the shaft. In other words, passages in addition to the minimum number of passages necessary to center the shaft can be added to the shaft based on the fluid flow requirements of the pump. An embodiment configured in this manner would include at least one stationary cylinder block and a valve enclosed within the cylinder block. The valve would include a shaft

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that extends through the stationary cylinder block. The shaft would have a plurality of passages along the length of the shaft.

[0014] An embodiment of the hydraulic pump has a double sided swash plate. A swash plate has a first side coupled to a first set of a plurality of slipper shoes and a second side coupled to a second set of a plurality of slipper shoes. The first side of the swash plate is generally parallel to the second side of the swash plate. The components that are coupled to the first side of the swash plate are substantially symmetric about the swash plate with respect to the components that are coupled to the second side of the swash plate.

[0015] A housing encloses the swash plate. The swash plate is coupled to a plurality of slipper assemblies via a fluid bearing. The slipper assemblies are on either side of the swash plate. A shaft anchors the swash plate at a fixed distance relative to the end of the shaft. The swash plate might be anchored in place via a pin. Regardless of the means used to anchor the swash plate, the swash plate is configured to be adjustable about an axis that is askew or perpendicular to the axis of the shaft. Alternatively, it is possible for the swashplate, in cases of fixed volumetric displacement, to be fixed at a particular angle. I.e., the swash plate does not necessarily have to be adjustable.

[0016] Another embodiment of the pump has a double sided swash plate in which a plurality of pumps are contained within a single housing. This embodiment has a swash plate that has a first swash plate surface. A first pump has a first plurality of components including the first swash plate surface, which is coupled to a first plurality of slipper shoes. The slipper shoes are coupled to a plurality of pistons. Each piston is contained within a respective cylinder bore. A second pump includes a second plurality of components that are substantially identical to the components of the first pump. The second pump components are substantially symmetrical with the first pump components. A mid-plane of the swash plate is a boundary line between the pumps.

[0017] A single housing surrounds the first and second pumps and the cylinder bores are integral with the housing. This pump includes an integral means for balancing internal forces and/or moments within the first and second pumps. It is conceivable that at least one additional pump is included in the single housing and configured substantially similar to the first and/or second pumps. The shafts of the additional pumps would be situated parallel to the first and second pumps within the housing.

[0018] In a further embodiment, a rotary valve is heterogeneously coupled to a shaft in such a way that only thrust and torque is transferred between the rotary valve and the shaft. The shaft is also coupled to a swash plate. The swash angle, i.e., the angle of the sliding interface of the swash plate and slipper relative to a plane perpendicular to the shaft axis of rotation, is set to an angle of tilt, i.e., fixedly coupled to the shaft or variably controlled by a swash control mechanism. The rotating swash plate forces the reciprocation of the pistons through slipper assemblies. On one half of the swash plate, the pistons are extracted or pulled out of their respective cylinder bores, whereas on the other half, the pistons are extended or pushed into their respective cylinder bores.

[0019] The rotary valve is generally cylindrical with two separate fluid passages—suction and discharge i.e., inlet and outlet, respectively. Both passages communicate with one or more piston bore(s) on a curved surface. The curved surfaces of the two passages are opposite each other and each take up less than half of the rotary valve circumference. The rotary valve is coupled to the shaft in such a way that a pump inlet passage is open to piston bores of those pistons being retracted from a piston bore and a pump discharge passage is open to piston bores of those pistons being extended into a piston bore. Therefore, as the pistons are pulled from the bores, the rotary valve allows fluid from the valve's suction cavity to be transferred into the growing volume of the respective piston bores. Whereas in piston bores on the other diametrical side of the pump axis, fluid is forced out of the piston bores through the discharge passage and into the valve's discharge cavity.

[0020] Variations on the design of the rotary valve are possible. An embodiment of the disclosed pump includes a housing with an integral manifold that encompasses a rotary valve. The rotary valve includes a shaft, a passaged section and a sealing section. The sealing section has a semicircular sealing ridge. The rotary valve has first and second axial face seals on a first and second planar end of the rotary valve as well as a high pressure discharge section on a side of the rotary valve opposite the semicircular sealing ridge and an inlet section on the first end of the rotary valve. The rotary valve is generally cylindrical and the semicircular sealing ridge encompasses less than three hundred and sixty degrees of the rotary valve. The rotary valve has a manifold engaging portion (i.e., the sealing ridge) and a recessed portion.

[0021] An operating gap is between the high pressure discharge section and the manifold. The width of the operating gap is dependent upon a force applied in the direction of the semicircular sealing ridge at the high pressure discharge section. The semicircular sealing ridge remains at a generally constant operating distance from the manifold during operation. The width of the operating gap also depends on shaft speed and the circumference of the rotary valve.

[0022] A further embodiment of the pump includes a valve bore that encompasses a rotary valve. The rotary valve has at least one axial face seal and at least one radial face seal. An operating sealing clearance gap is between the radial face seal and the valve bore. The operating sealing clearance gap maintains a generally constant thickness during operation of the rotary valve. A discharge gap is between the rotary valve and the bore. The rotary valve is biased by a force applied in the direction of the semi-circular sealing ridge and the force determines a width of the operating sealing clearance gap. The rotary valve includes a shaft. The radial face seal has a semicircular sealing component and is generally coaxial with the shaft, which has a an inlet passage and a discharge passage. Notably, the radial face seal is allowed a freedom to move relative to the shaft so that it is not always coaxial with the shaft.

[0023] A yet further embodiment of the pump includes a housing that has a valve bore and a manifold. A shaft is within the manifold and a rotary valve is within the valve bore. The shaft is attached at a distal end to a planar surface of the rotary valve. The shaft has a first and a second axis that are offset from each other. During operation, the second axis is coincident with an axis of the valve bore and the first axis is offset from a centerline of the manifold.

[0024] The rotary valve has a high pressure outlet and a low pressure inlet. The shaft can be offset in a direction

opposite the high pressure outlet of the rotary valve and it can also be offset in a direction toward the high pressure outlet of the rotary valve.

[0025] The shaft contacts the manifold via a manifold contact surface on the shaft. A bearing is positioned in the manifold and is in contact with the shaft at the manifold contact surface.

[0026] A plurality of cylinders are arranged within the manifold parallel and in a circle around and coaxial with the valve bore. The rotary valve is generally coaxial with the valve bore and the shaft is not coaxial with the valve bore. The shaft is monolithic with the rotary valve.

[0027] The rotary valve balances high pressure loads. The valve can be configured to eliminate unbalanced forces, which cause the valve to move off its rotational axis. Any unbalanced force on the valve could break the lubricant fluid film barrier, which would cause metal to metal contact between the rotary valve and the valve bore. This could cause the valve to stick to the valve bore, thereby disrupting rotating members.

[0028] The rotary valve transports or directs high pressure fluid to a thrust cavity between the valve and a manifold thrust shelf to balance the axial load on the shaft. In one embodiment, the area of the valve/fluid thrust cavity interface is equal to about half of the sum of all piston/fluid interfaces. The majority of axial loads on the shaft are contained within the cylinder block. A roller element thrust bearing is adequate to control a fraction of the total axial load between the shaft and the exterior housing.

[0029] In this embodiment, low pressure fluid enters the pump through an inlet port into an inlet cavity in the valve. This fluid is passed on to a plurality of piston bores by way of a valve inlet passage. Each piston in communication with the valve inlet passage is being forced out of the piston bore. As the shaft-valve continues to rotate, the piston is forced into the piston bore, thereby compressing the fluid. At this time, the piston bore begins to open up to a valve discharge passage. The valve discharge passage allows for the high pressure fluid from the piston bore to be passed on to a discharge cavity where it then exits the pump through a discharge port. The force required to move the pistons is provided by an input torque to the shaft-valve. This torque is transferred to the swash plate. The swash plate is forced into the slippers on the high pressure half of the swash plate due to the tilt of the swash plate, relative to the shaft-valve (known as the swash angle). The slipper assembly is then forced into the piston, which communicates with the fluid inside the piston bore. This process creates a set of different forces/moments. An axial load is carried by the swash plate, which is transferred to the valve shaft. An equal and opposite axial load is carried by the valve shaft by means of the thrust cavity, thereby resulting in zero net axial loads. A moment is carried by the swash plate, which is transferred to the valve-shaft. An equal and opposite moment is carried by the valve-shaft by means of a pressure differential between the valve discharge passage and valve inlet passage. The final force/moment is the moment about the axis of rotation, which is reacted by the input torque required to drive the shaft-valve. Thus, the shaft-valve and all other rotating members are balanced. The balance of the rotating members is independent of the swash angle control and/or slipper hold down mechanism.

[0030] In prior art pumps, the displacement of high pressure fluid results in a large thrust, or axial load on the swash plate and manifold; which is then transferred to the pump exterior housing. Also, in prior art pumps, the thrust load creates large separation forces between the exterior housing components, thereby increasing the required clamp load. Sharing this thrust load between multiple components is disadvantageous because it increases the overall size and weight of the pump, as opposed to the present invention utilizing a one-piece cylinder block that contains all axial loads within the body and inherently reduces the clamp load requirements, which, in turn, permits reducing the pump size and weight.

[0031] The pump incorporates a non-rotating cylinder block. Reciprocation is achieved by affixing a swash plate to a rotating shaft, and the plate rotates about a single axis at an angle relative to the single axis to effect reciprocation of pistons within the pump. The generally cylindrical rotary valve has a suction or intake area at a planar surface in communication with an external supply of low pressure fluid. The intake communicates through the valve with an inlet passage on a curved surface of the rotary valve configured to alternately communicate with one or more cylinder bores. On an opposing curved surface of the rotary valve is a recessed output or discharge area of the rotary valve. The output area is alternately in communication with one or more cylinder bores, and is a conduit for output of high pressure fluid from the pump.

[0032] The present pump intakes fluid through a centerline of the rotary valve, and the fluid follows a more direct and consistent path, which reduces the likelihood of increased fluid velocity and thus reduces pressure losses between the inlet passage and the piston bore. However, the present valve is consistently open to cylinder block conduits to the piston bore.

[0033] The rotary valve balances high pressure loads. This eliminates forces that might otherwise throw the valve off-center. The rotary valve can preserve the lubricating fluid film barrier, preventing metal-metal contact between rotary valve and the cylinder block.

[0034] A conventional slipper shoe may be mounted directly to the spherical head of each piston, and is maintained in effective sliding contact with the flat face portion of the swash-plate by means of a hold down plate.

[0035] Previous iterations of non-rotating cylinder block axial pumps required large bearings. Resulting pumps were large and heavy. The embodiments disclosed herein reduce size and weight by as much as one-third, and possibly more with various further pump modifications, without loss in capacity or efficiency. The present pump permits larger diameter pistons with a shorter stroke. The pistons can be shorter without sacrificing capacity and this permits a shorter pump. Without balancing the loads of the pistons with working fluid, the loads would have to be balanced with a mechanical bearing (i.e., a thrust bearing). In a prior art pump, such a mechanical bearing with short and wide pistons would have to be very large. However, because the present pump employs a fluid bearing, short and wide pistons are usable without increasing the size of the pump.

[0036] The swash plate is configured to adjust a stroke length of the plurality of pistons. A means for adjusting a tilt angle of the swash plate can be mechanical or hydraulic or a combination of the two. If hydraulic, it can include a plurality of hydraulically filled volumes that are configured to adjust the tilt angle of the swash plate in response to a variance of hydraulic fluid in the volumes. The pump's

rotary valve also preferably includes a moment zeroing means. The moment zeroing means has balanced high pressure passages provided to accept high pressure fluid during rotation.

[0037] Each piston of the plurality of pistons has a working fluid end. The effective (net) surface area of the planar end of the rotary valve that is exposed to high pressure fluid is equal to (or nearly equal to) a sum of surfaces areas of the high pressure ends of each of the plurality of pistons that is exposed to high pressure fluid. A plurality of slipper assemblies connects the swash plate to a respective piston. A fluid bearing is positioned between each slipper assembly and the swash plate. The fluid bearing is fluidly coupled to the high pressure ends of a respective piston.

[0038] The shaft and rotary valve are designed to move freely about at least four degrees of freedom. This hydraulic pump has a stationary cylinder block and a valve within the stationary cylinder block. The valve is connected to and is substantially coaxial with a shaft within the stationary cylinder block. The valve is rotatable about an axis of the shaft and the valve is configured to be offsetable relative to an axis of the shaft about four degrees of freedom.

[0039] The connection between the valve and the shaft is configured to limit transfer of force, from the shaft to the valve, to a torque about an axis of rotation of the shaft and a thrust load along the axis of rotation. The valve is balanced within the cylinder block by controlled leakage rates as a function of force and/or moment on the valve.

[0040] Another embodiment of the hydraulic pump is one designed with a drive key. This embodiment has a shaft-valve assembly. A valve is connected to a shaft via interaction between a drive key and a keyway. The connection between the valve and the shaft is maintained via any of a plurality of types of fasteners. The interaction is configured to provide at least one degree of freedom for the valve relative to the shaft. The degree (or degrees of freedom, if there is more than one degree of freedom) is in a linear direction that is configured perpendicular to an axis of rotation of the shaft. A swash plate is connected to the shaft. The swash plate rotates with the shaft and is configured to tilt about an axis perpendicular to the axis of the shaft via a means for tilting the swash plate.

[0041] The rotary valve is an elongated cylinder having a height that is preferably larger than the rotary valve's diameter. High pressure cross passages are juxtaposed in a side surface of the rotary valve. A low pressure transition zone is in the side surface of the rotary valve and is in contact with and extends a predetermined distance from the high pressure passages.

[0042] A is integral with a planar end of the rotary valve. A surface area of the hydrodynamic thrust bearing discharge thrust cavity is equal to about one half of the total area of end surfaces of the plurality of pistons.

[0043] A plurality of slipper assemblies is fluidly coupled to the swash plate. The slipper assemblies each include a slipper ball, a slipper neck and a slipper shoe. A hold-down plate maintains a fluid coupling between the slipper assemblies and the swash plate. The hold-down plate comprises a plurality of recesses each configured to engage a respective slipper assembly. The hold-down plate is prevented from rotating about the shaft.

[0044] Another embodiment is a segregated rotary valve/shaft embodiment. The rotary valve of the segregated embodiment is a rotating barrel valve. One of the advantages

of this embodiment is the elimination of cantilevered forces by using a valve that emulates a barrel as opposed to a valve that is more disk-shaped. This is because a disk is susceptible to cantilever forces. This embodiment includes a shaft having a swash plate. The shaft and the swash plate rotate within a housing. A plurality of pistons oscillate in response to rotation of the shaft and the swash plate. The rotary valve is coaxial with the shaft and is fixedly attached at a planar end of the rotary valve to the shaft. The rotary valve is configured to rotate about an axis common to the shaft.

[0045] A beneficial feature of this embodiment is that an inlet port is centered along a main axis of the pump. One advantage to this configuration is that the number of turns of the inlet passageway within the rotary valve can be reduced. Thus, the possibility of cavitation is reduced. This embodiment includes a shaft and a rotary valve within a pump housing. The rotary valve is coaxial with the shaft and is connected to the shaft. A passageway is connected to the inlet port and has a path of travel with bends or turns of no less than ninety degrees. At least one high pressure passage is located in the rotary valve.

[0046] The pump housing has an integrated manifold and a plurality of cylinders. Cylinder bores are fluidly connected to the rotary valve high pressure passages. The bores alternately fluidly connect with the pump inlet and passageway and pump outlet upon rotation of the rotary valve.

[0047] This embodiment includes a manifold housing that has a plurality of piston bores. A rotary valve is rotatably housed within the manifold housing and includes an inlet orifice and an inlet passage that are serially and fluidly connected to the cylinder bores. The inlet passage has an axis through the rotary valve. The inlet passage has a maximum bend angle of about ninety degrees to avoid excessive fluid separation, thus increased fluid velocity through the passage.

[0048] The swash plate at least partially governs the flow rate through the inlet and the flow rate is further governed by rotational speed of the shaft. The rotary valve includes a plurality of high pressure cross passages juxtaposed in a side surface of the rotary valve. High pressure passages are fluidly connected to the high pressure cross passages and extend a predetermined distance from the high pressure cross passages. The planar end of the rotary valve provides for the presence of a discharge thrust cavity that is configured to balance axial forces on the pump.

[0049] A connection between the rotary valve and a drive element (shaft) can be via a rotary valve keyway and one or more c-clips. The advantage to this construction is that planar thrust loads are carried by the interface between the keyway, c-clips and the shaft and the interface provides a means for transmitting torque from the drive element to the rotary valve. The connection between the keyway of the rotary valve and the key of the shaft transfers torque from the shaft to the rotary valve.

[0050] The connection between the shaft and the rotary valve is locked by at least one and preferably by a pair of c-clips that are coaxial with the shaft. They engage a recess in the shaft and a corresponding recess in the rotary valve. The keyway can have at least one degree of freedom in the planar surface of the rotary valve

[0051] The rotary valve has components integrated in the surface of the rotary valve that are configured to balance the forces acting on the rotary valve. These components can also be configured to help balance the moments about the entire

pump. At a minimum, the components are configured to balance rotational and axial forces within the pump.

[0052] The segregated shaft-valve pump (and the integrated shaft valve pump) can include pre-compression and decompression notches within the surface of the rotary valve. The advantage to this is the reduction of pump noise. Pump noise results from, among other things, fast changes of fluid pressure.

[0053] This embodiment of the pump includes a manifold that has a main bore and a rotary valve within the manifold-housing. The rotary valve includes a high pressure outlet and low pressure external surfaces. The rotary valve has an axis of rotation along a centerline of the main bore. At least one pre-compression notch is fluidly connected to an entry side of the high pressure outlet.

[0054] The rotary valve also includes an inlet in one of its planar surfaces. The inlet is centered about the rotary valve's axis of rotation. The rotary valve also includes cross passages in its curved side surfaces. The forces acting on the rotary valve due to high pressure fluid flow are balanced by the cross passages. At least one high pressure zone and at least one low pressure zone are provided by the cross passages. The high pressure zones and low pressure zones are configured about the rotary valve to eliminate moment forces on the rotary valve, placement of the zones being determined experimentally or by an equation in which all moments and forces on the valve are balanced.

[0055] An advantage to the one piece cylinder block described above is that forces acting on seams within multiple part housings of prior art cylinder blocks are eliminated. The cylinder block contains a plurality of cylinder bores. A piston is slidingly positioned within each cylinder bore. The cylinder block is monolithic and the cylinder bores are integral with the cylinder block.

[0056] A hydraulic slipper assembly fluid supply channel is present through the entire length of the slipper assembly. Also, each of the plurality of pistons has a piston fluid supply channel that is fluidly coupled to the slipper assembly fluid supply channel and further fluidly coupled to the piston bore.

[0057] The disclosed subject matter employs a fluid bearing that allows relative motion between the pump's valve/shaft/swashplate and the cylinder block/manifold/housing. This configuration reduces the loads on all bearings within the device, which in turn reduces the parasitic bearing losses during operation.

[0058] The reduction of bearing loads is a function of the balance between the fluid forces and the structural response to those loads. Fluid forces occur in the pump when mechanical energy is transferred from the drive motor to the fluid as pressure and/or energy is transferred from the fluid into mechanical force or work into a load. By design there is a balance between the piston axial force, which is created by the reaction of the fluid pressure resisting the motion of the piston, and a second force which is the reactive force generated by the high pressure fluid acting on a planar surface of the valve/shaft (said planar surface, having a normal axial component). The net force (i.e. the difference in magnitude of the aforementioned first and second forces) varies continually during the operation of the device. The time varying imbalance of these forces requires the structure of the device to support the net residual forces and moments.

[0059] Non-rotating cylinder block technology advantages include lower mass and smaller device diameter, which affords lower rotational mass and thus inertia, which

improves overall efficiencies when operating in non-steady-state conditions. Also, the cylinder block can be used as the device housing, thus obviating the need for the addition of a separate component, namely the housing as a structural member of the device.

[0060] The present pump uses the working fluid and the inevitable leakage from the working fluid (i.e. the fluid being displaced at high-pressure, that would otherwise have leaked between the various components within the pump only to be drained to a tank/reservoir, thus representing a 100% parasitic loss) to balance the aforementioned internal forces and/or moments on the rotating members (i.e. the shaft, swash plate, and valve; as well as, the mechanisms used to couple and/or control them). The present pump device minimizes the use of a predetermined (by design) controlled leakage of the working fluid to balance the internal forces on mechanical components as generated by the working fluid while it is displaced at high-pressure in a manner that does not require additional parasitic porting of said working fluid to develop the load resistant bearing (i.e. to support a hydrostatic bearing). This has the effect of improving the overall efficiency of the device. Thus, the present pump does not require the use of active element bearing (i.e. roller element type bearing). Typical axial piston devices, for the applied loads and rotational speeds, require large bearings in terms of diameter, weight and cost.

[0061] Embodiments of the disclosed pump have a valve geometry that is cylindrical, having a curved sealing/bearing surface. Conventional technology uses the valve plate as a planar bearing (i.e. two flat sealing surfaces separated by the working fluid acting as a fluid film). The advantage of the cylindrical shape is that, when operating under off-axis loaded conditions (i.e. eccentric position of cylinder relative to the bore), the clearance between the cylinder and its bore creates a variable gap size. This in turn creates a wedge like effect of the working fluid during rotation. This phenomenon, which is known as the wedge effect (see FIG. 3), produces a non-uniform pressure distribution within the load region which performs two functions. First and foremost, it provides a reaction force, equal to and opposite the applied load. Second, the wedge effect works to pull new fluid into the loaded region as the cylinder (or shaft) rotates, which replenishes the fluid lost due to pressure differential between the loaded region and the rest of the fluid cavity.

[0062] The disclosed pump combines the valve and journal bearing mechanisms into one component. The axial leakage flow from discharge to suction (across the bearing land) provides a means of heat transfer and thereby eliminates the need to port pressurized fluid (lubricant) through a dedicated port to the center of the journal land area.

BRIEF DESCRIPTION OF THE DRAWINGS

[0063] FIG. 1 shows a conventional pump having a non-rotating cylinder block;

[0064] FIG. 2 shows a conventional pump having a rotating cylinder block;

[0065] FIG. 3 shows a rotary valve experiencing a wedge effect;

[0066] FIG. 4 shows a first embodiment of a self-balancing hydraulic pump;

[0067] FIG. 5 shows a plurality of pistons in the inventive hydraulic pump arranged arcuately;

[0068] FIG. 6 shows the first embodiment of the hydraulic pump with a tilted swash plate;

[0069] FIG. 7 shows a second embodiment of the hydraulic pump;

[0070] FIG. 8 shows a third embodiment of the hydraulic pump;

[0071] FIG. 9 is a cross-sectional view of the hydraulic pump shown in FIG. 5;

[0072] FIG. 10 is a further cross-sectional view of the hydraulic pump shown in FIG. 5;

[0073] FIGS. 11A-11B are detailed views of a rotary valve of the hydraulic pump of FIG. 5;

[0074] FIG. 12 is a detailed view of a swash plate;

[0075] FIG. 13 is a detailed view of a slipper assembly;

[0076] FIG. 14 is a view of a two-piece rotary valve;

[0077] FIG. 15 is a rotary valve showing significant wear;

[0078] FIG. 16 is a cross sectional view of a pump showing a first version of a rotary valve;

[0079] FIG. 17A is a first view of a sealing section of the rotary valve shown in FIG. 16;

[0080] FIG. 17B is a second view of the sealing section shown in FIG. 17A;

[0081] FIG. 17C is a cross sectional view of the sealing section of FIG. 17A showing pressure drop across surfaces of the sealing section;

[0082] FIG. 17D is a first view of the sealing section of FIG. 17A attached to the rotary valve;

[0083] FIG. 17E is a second view of the sealing section of FIG. 17A attached to the rotary valve;

[0084] FIG. 18 is a first cross sectional view of the entire pump using the rotary valve of FIG. 16;

[0085] FIG. 19 is a second cross sectional view of the entire pump using the rotary valve of FIG. 16 with the rotary valve having been rotated ninety degrees from its position in FIG. 18;

[0086] FIG. 20 is a top view of the rotary valve and sealing section of FIG. 16 inside a valve bore;

[0087] FIG. 21 is a cross sectional view of a pump showing a second version of a rotary valve;

[0088] FIG. 22A is a first view of a sealing section attached to the rotary valve of FIG. 21;

[0089] FIG. 22B is a second view of a sealing section attached to the rotary valve of FIG. 21;

[0090] FIG. 23 is a cross sectional view of the sealing section of FIG. 21 showing pressure drop across surfaces of the sealing section;

[0091] FIG. 24 is a top view of the rotary valve and sealing section of FIG. 21 inside the valve bore;

[0092] FIG. 25 is a cross sectional view of a pump showing a third version of a rotary valve;

[0093] FIG. 26 is a cross sectional view of the rotary valve and shaft of the pump shown in FIG. 25;

[0094] FIG. 27 is an axial face seal embodiment and a radial face seal embodiment;

[0095] FIG. 28 is a partial view of an rotary face seal;

[0096] FIG. 29 is a perspective view of an axial face seal;

[0097] FIG. 30 shows a double sided pump having the radial face seal embodiment and rotary face seal embodiment of FIG. 27;

[0098] FIG. 31 is a cross sectional view of the valve and radial face seal and shows a pressure gradient at the valve's radial face seal; and

[0099] FIG. 32 is a cross-sectional view showing an 0-ring relative to a passage in the valve.

DETAILED DESCRIPTION OF THE INVENTION

[0100] FIG. 4 shows a hydraulic pump 2. The pump 2 includes a housing 4. A shaft-valve 6 extends along an entire length of the housing 4. The shaft-valve 6 allows for balancing of forces and/or moments within the pump 2. As a result of the force and/or moment balancing, the necessity for mechanical bearings in the pump is reduced. Therefore, construction and capability of the pump 2 are not limited by the capabilities of any rolling element bearings during operation as roller element bearings are not necessary in the pump 2.

[0101] This embodiment of the pump 2 has a plurality of passages 10a and 10b in the shaft-valve 6. Fluid flow throughout the pump 2 and rotation of a swash plate 8 creates a force and/or moment on the shaft-valve 6. Fluid flow is a result of the rotating swash plate. Pistons cause the fluid flow via their reciprocation that results from engagement with the rotating swash plate. Thus, the rotating swash plate (and fluid flow) creates the forces on the pump. The plurality of passages 10a and 10b are configured in the shaft-valve 6 such that the forces and/or moments on the shaft-valve 6 are balanced.

[0102] A rotary valve that is separate from the shaft is beneficial in certain circumstances and is discussed below in detail. In the present embodiment, however, the passages 10a and 10b are integral with the shaft-valve 6 and eliminate any need for a separate rotary valve as the functions of a rotary valve and the shaft are performed by a single component—the shaft-valve 6. A conventional non-rotating cylinder block pump that does not have a rotary valve as disclosed herein is shown in FIG. 1.

[0103] The swash plate 8 is connected to the shaft-valve 6 and is preferably configured to tilt about an axis 28, perpendicular to the axis of the shaft-valve 6. The swash plate 8 has a first side 12 coupled to a first plurality of slipper assemblies 14. Contact between the slipper assemblies 14 and the swash plate 8 is shown in FIG. 5. The shaft-valve 6 is self-centering within the housing 4 about its own physical axis 16, i.e., the shaft-valve's 6 axis of rotation is aligned as much as possible with the shaft-valve's axis of symmetry. The swash plate 8 is a double sided swash plate. The swash plate 8 is disk shaped and symmetrical about a plane through the middle of the disk. The first side 12 is preferably a planar surface. The swash plate 8 has a second side 20, which is also a preferably planar surface. The first side 12 and the second side 20 of the swash plate 8 oppose each other. A second plurality of slipper assemblies 50 is coupled to the second side 20 of the swash plate 8.

[0104] The use of wedge shaped swash plates in double sided pumps dominated the prior art; however, in the present pump 2, the first side 12 of the swash plate and the second side of the swash plate 20 are as close to parallel to each other as possible. It is not necessary that the first side 12 and the second side 20 be absolutely parallel to each other. However, the closer to parallel the first side 12 and the second side 20 are, the closer to fully balanced the forces and/or moments on the shaft-valve 6 will be.

[0105] Each of the slipper assemblies 14 includes a piston 18. The housing 4 acts as a cylinder block for each piston 18. In the pump 2, the housing 4 is non-rotating and each piston 18 remains circumferentially stationary while the shaft-valve 6 and swash plate 8 rotate. This is in contrast to prior art pumps in which the cylinder block rotates.

[0106] With reference to FIG. 6, the first plurality of slipper assemblies 14 is coupled to the first side 12 of the swash plate 8 and a second plurality of slipper assemblies 50 is coupled to the second side 20. At the end of each slipper assembly 14 is a slipper shoe 24. The slipper shoe 24 can slide a distance 44 along a surface of the swash plate 8. Coupling between the slipper shoes 24 and the sides 12 and 20 of the swash plate 8 is via a fluid bearing. Thus, the slipper shoes 24 slide freely along the surface of the swash plate 8.

[0107] A pin 22 anchors the swash plate 8 at a fixed distance relative to the end of the shaft-valve 6. Regardless of the means used to anchor the swash plate 8, the swash plate 8 is configured to be adjustable about the axis 28 that is askew or perpendicular to the axis of the shaft-valve 6. Increasing or decreasing a swash angle of the swash plate 8 adjusts the volumetric displacement of the hydraulic pump 2 i.e., fluid volume displacement per shaft revolution. The greater the swash angle of the swash plate 8, the higher the volumetric displacement per revolution of the pump 2. As the swash angle of the swash plate 8 is increased, the stroke of each of the pistons 18 in the slipper assemblies 14 and 50 is increased. Thus, the piston bore 48 that houses each piston 18 can accommodate more fluid to thereby increase the volumetric displacement.

[0108] To actuate the pistons 18, the shaft-valve 6 rotates on its axis 16. Rotation of the shaft-valve 6 causes the swash plate 8 to rotate about axis 16. When the swash plate 8 is tilted, interaction of the swash plate 8 with the slipper assemblies 14 and 50 causes the pistons 18 to reciprocate along each individual axis of each piston 18. Interaction between the each slipper assembly 14 and 50 and the swash plate 8 is enhanced by the presence of a fluid bearing between the swash plate 8 and slipper assemblies 14 and 50. The fluid bearing is provided by high pressure fluid that is fed through the center of the piston and slipper.

[0109] The swash plate 8 is tiltable about an axis 28 of the pin 22. Operatively connected to the swash plate 8 is a means 32 for tilting the swash plate 8 (i.e. angular position), relative to the shaft-valve's 6 rotational axis 16. The means 32 for tilting the swash plate 8 can be any of a hydraulic means such as a hydraulic jack or cylinder, a mechanical dog-bone assembly responsive to mechanical input, an electric means, a gearing system, or any combination thereof. The shaft-valve 6 is capable of unintended but sometimes unavoidable eccentric rotation (and centric rotation) during operation of the pump 2. Hence, it is advantageous to have the shaft-valve 6 be self-centering. As shown in FIG. 5, the plurality of passages 10a and 10b receive, direct and exhaust fluid into, around and from the shaft-valve 6 throughout the pump 2. Each of the plurality of pistons 18 is arcuately positioned coaxially with the shaft-valve 6. The plurality of pistons 18 are grouped in two sets and are on opposite ends of the housing 4 and engage with sides 12 and 20 of the swash plate 8, respectively.

[0110] The housing 4 has the at least one inlet passage 36 for receiving fluid in the pump and at least one outlet passage 38 for expelling fluid from the pump 2. The passages 10a and 10b on each end of the shaft are generally one-hundred and eighty degrees apart.

[0111] As can be seen in FIG. 6, fluid enters the housing 4 at inlet passage 36 and proceeds to low pressure cavity 46. Fluid enters a plurality of piston bores 48 via low pressure ports 40a and 40b and low pressure passages 10b (see FIG.

4). As the shaft-valve 6 rotates, low pressure passages 10b are continuously aligned with low pressure ports 40a and 40b. High pressure passages 10a are continuously aligned with high pressure ports 42a and 42b. High pressure ports 42a and 42b connect with outlet passage 38 via lines 52a and 52b.

[0112] The forces acting on the shaft-valve 6 (primarily caused by the moment generated by high pressure fluid forces on the pistons) are balanced via the geometry and location of the passages 10a and 10b. With further reference to FIG. 4, when passage 10a is engaged with high pressure port 42a, passage 10b, which dynamically opposes passage 10a, i.e., which balances forces on the pump, engages with low pressure port 40b. This is because the actions of the slipper assemblies are in opposition to each other due to their generally opposing positions on the swash plate 8 (i.e., when the swash plate 8 is tilted and rotating, one slipper assembly is sliding out of its bore thereby creating a vacuum, while an opposing slipper assembly is sliding into its bore helping to create high pressure on the fluid situated inside the bore). The effect is for the forces on the pump 2 that are caused by fluid flow within the pump 2 balance the forces of the slipper assemblies 14 and 50 on the swash plate 8 on.

[0113] The sides 12 and 20 of the swash plate 8 apply pressure on the slipper assemblies 14 on the high pressure half of the swash plate 8 due to the tilt of the swash plate 8, relative to the shaft-valve 6 (known as the swash angle). Each slipper shoe 24 of each slipper assembly 14 and 50 is then forced to apply pressure on each respective piston 18, which communicates with the fluid inside the piston bore 48. As shown in FIG. 4, this process creates four different forces/moments ($F_{1,u}$, $F_{1,l}$, M_1, M_2). An axial load (F_1) is carried by the swash plate 8. There are two components to the axial load (F1u and F1l) that are equal and opposite. There are trade-offs in back to back piston alignment. Full balance of axial loads can be achieved with zero indexing (i.e. misalignment), whereas indexed pistons results in non-zero net axial load. However, the indexing of back to back pistons provides an adequate balance of axial loads, the advantage of indexing is that the flow ripple (i.e. vibration, noise, pressure/flow pulsation, etc.) can be greatly reduced (i.e. nearly half the amplitude of zero indexing).

[0114] A radial load (F_2) is carried by the shaft-valve 6. There are two components to the radial load ($F_{2,u}$ and $F_{2,l}$). These components $F_{2,u}$ and $F_{2,l}$ are equal and opposite to each other, thereby resulting in zero net radial loads. A moment (M_1) is carried by the swash plate 8, which is transferred through the pivot pin 22 to the shaft-valve 6. An equal and opposite moment (M_2) is carried by the shaft-valve 6 by means of a pressure differential between the high pressure passages 10a and the low pressure passages 10b (M2 is created by F2,u & F2,l). Another force/moment is the moment (M_z) about the axis of rotation, which is reacted by the input torque required to drive the shaft-valve 6. Thus, the shaft-valve 6 and all other rotating members are balanced. The balance of the rotating members is independent of the swash angle.

[0115] The housing 4 shown in FIG. 4 and FIG. 6 is a two part housing including parts 4a and 4b. However, the housing can be monolithic or it can be of many components. If the housing 4 is of many components, it is beneficial to provide a static seal, e.g., o-ring or gasket between the components. The components can be held together via a flange or socket type connection. With the two component

configuration, a first stationary housing/cylinder block **4a** encompasses the first plurality of pistons and a second stationary housing/cylinder block **4b** encompasses the second plurality of pistons.

[0116] As shown in FIG. **4** and FIG. **6**, the components that are coupled to the first side **12** of the swash plate **8** are substantially symmetric about the swash plate **8** with respect to the components that are coupled to the second side **20** of the swash plate **8**. However, the pump disclosed herein is not limited to the double sided configuration shown in FIG. **4** and FIG. **6**. As shown in FIG. **7**, a single sided pump **402** embodiment is contemplated. In the single sided pump **402**, the swash plate **408** is configured at one axial end of the pump **402**. In this case, fluid bearings **403** are more necessary as opposing sets of slipper assemblies are not present in this embodiment. Rather, a single set of slipper assemblies **414** are used. Therefore, moments and forces that are created by rotation of the swash plate and fluid forces can be zeroed out. To compensate for axial forces, a notch **415** or similar relief can be added to the shaft-valve **406** and engage with edge **417** of housing **404**.

[0117] It is conceivable that at least one additional pump (not shown) is included in the single housing and configured substantially similar to the first and/or second pumps. The additional pumps would be situated parallel to the first and second pumps within the housing. Thus, a plurality of pumps is contained within a single housing such that two or more shaft-valves are within the housing and in parallel with each other. A parallel configuration is not necessary. Operation of each pump within the housing can be completely independent of operation of another pump within the housing.

[0118] A further embodiment of the present hydraulic pump is one in which the shaft-valve **6** is designed for motion about at least three degrees of freedom and as many as four degrees of freedom. There are two linear degrees of freedom and at least one rotatable degree of freedom about an axis perpendicular to the linear degree of freedom.

[0119] An additional embodiment of an inventive pump **502** is shown in FIG. **8**-FIG. **12** and is one in which a shaft and a valve are not integrated as in the previous embodiments. With particular reference to FIG. **8** and FIG. **9**, components of the pump **502** include a pump housing **504** (shown in FIG. **8**), which houses a shaft **506**, a swash plate **508**, a hold down plate **510**, one or more c-clips **512**, a rotary valve **514**, a cylinder block **516**, a manifold **518**, a swash plate control link **520** for setting an angle of the swash plate **508**, a plurality of pistons **522**, and a plurality of slipper assemblies **524**. Each of the slipper assemblies **524** contains a slipper shoe **526**, a slipper ball **528** and a slipper neck **530**, each slipper neck **530** connects a slipper shoe **526** to a respective slipper ball **528**.

[0120] The shaft **506** is positioned along a center axis of the pump housing **504**. One end of the shaft **506** extends outside of the pump housing **504** and includes a spline **532**. The spline **532** is toothed for attachment to a gear of a motor, crank, flywheel or some other motion transferring mechanism. The shaft **506** is held in place by multiple bearings at each end of the pump **502**.

[0121] The connection between the valve **514** and the shaft **506** is configured to limit transfer of force, from the shaft to the valve, to a torque about an axis of rotation of the shaft and a thrust load along the axis of rotation. The valve

514 is balanced within the cylinder block **516** by controlled leakage rates as a function of force and/or moment on the valve **514**.

[0122] The hydraulic pump **502** intakes fluid through a non-rotating cylinder block and passes it through the rotary valve **514**. The rotary valve **514** is in constant communication with an inlet port **536** and an discharge port **612** as well as a plurality of piston bores **546** within the cylinder block **516**.

[0123] In an embodiment, and with reference to FIG. **10** and FIG. **11A**-**11B**, low pressure fluid enters the pump through an inlet port **536** into the inlet cavity **602**. This fluid is passed on to the plurality of piston bores **546** by way of the valve inlet passage **614**. Each piston **522** in communication with the valve inlet passage **614** is being forced out of the piston bore **546**. When a piston bore **546** is closed off from the valve inlet passage **614**, the low pressure fluid becomes trapped. As the valve **514** continues to rotate, the piston **522** is forced into the piston bore **546**, thereby compressing the fluid inside the bore **546**. At this time, the piston bore **546** begins to open up to a valve discharge passage **616**. The valve discharge passage **616** allows for the high pressure fluid from the piston bore **546** to be passed on to a discharge cavity **618** where it then exits the pump through a discharge port **612** at an end of outlet cavity **612**. The force required to move the pistons is provided by an input torque into the shaft **506** (see FIG. **9**). This torque is transferred to the pivot pin **534** (see FIG. **9**), where it is then transferred to the swash plate **508**.

[0124] At an opposite end of the shaft **506** from the spline **532**, the rotary valve **514** is attached and held in place via the c-clips **512**. Whenever the shaft **506** rotates, the rotary valve **514** rotates at the same rotational velocity. Similarly, between each end of the shaft **506**, a swash plate **508** is connected to the shaft **506** via a pin **534**. Whenever the shaft **506** rotates, the swash plate **508** rotates at the same rotational velocity as the shaft **506**. The shaft **506** functions as a rotational motion transmission component in that it accepts rotational motion from an external motor to ultimately cause the rotary valve **514** and swash plate **508** to rotate thereby creating a pumping force.

[0125] Alternatively, the shaft **506** receives a rotational motion from the swash plate from fluid passing through the rotary valve which are translated into rotational motion to turn the spline **532** (i.e., to cause the pump to act in reverse as a motor). To reverse flow (alternatively referred to hereinafter as "going over center"), additional cross passages must be added on the opposite side of the valve from where they are currently. The reason for adding additional cross passages is because high pressure cross-passages become low pressure and vice versa.

[0126] With reference to FIG. **12**, the swash plate **508** is attached to the shaft **506** via the pin **534**. The pin **534** in the shaft **506** allows the swash plate **508** to tilt, thereby altering the axis of the swash plate **508** relative to the axis of the shaft **506**. A hollow stem **554** extends from a bottom surface of the swash plate **508**. The pin **534**, which engages with the shaft **506** is positioned within the stem **554**. Other than where the stem **554** extends from the swash plate **508**, the bottom surface of the swash plate **508** is flat. Such a flat surface of the swash plate **508** allows for a sliding interaction between the flat bottom of the swash plate **508** and the slipper shoes **526**, which are explained in more detail later.

[0127] The pin **534** facilitates rotation of the swash plate **508** about the axis of the shaft **506**. The pin **534** provides for torque and thrust loads to be transferred between the swash plate **508** and the shaft **506**. The advantage of the pin **534** affords a one-piece swash plate, which affords structural stability due to the forces acting on opposite sides of the swash plate **508**.

[0128] In previous pumps, a rotating cylinder block was used. As such, it was necessary to have bolts holding the housing to the block. An axial pump's housing was a combination of two components where the pump was held together by all of the bolts. It was necessary to contain the separation force of multiple piston areas whereas with the present design, it is only necessary to consider the weight of the components and the forces that the bearings are imposing (due to the one piece housing). This is advantageous in that the bolts have a much longer life because tension forces are not acting on the bolts. Also, seals would be necessary to maintain the hydraulic fluid within the junction between the block and the housing.

[0129] Radial loads on the pump are limited by limiting the tilt of the swash plate. There is less of side load or moment (less of a radial load) with smaller tilt angles. Thus, the importance of a radial bearing is reduced and a lighter duty bearing can be used. Further, the present pump allows for a reduced stroke. The pistons are shorter but wider than conventional pistons. Therefore, the present pump can displace the same amount of volume as conventional pumps via a reduced swash plate tilt angle. The cylinder block is integral with the pump housing. Reciprocating pumps with fixed cylinder blocks have structural advantages. By removing rotation of the cylinder block, centripetal forces are eliminated and a structural unit of reduced size and mass is possible.

[0130] Also, as there is a shorter tilt angle of the swash plate (no greater than about twelve degrees is necessary for operation of this pump), there can be an increased bore/piston diameter without a decrease in volume flow rate. Increased diameter, and decreased swash plate tilt angle result in a majority of the load being axial rather than radial. This results in less friction and wear on pistons, which in turn produces higher mechanical efficiency. A series of bearings **542** are meant to accommodate a moment load about an axis perpendicular to the pin **534**. The bearings **542** are above and below a center point of the pivot pin **534** to balance out the high pressure and low pressure sides of the swash plate **508**.

[0131] The swash plate's axis of tilt and the center of the slipper ball **528** are in line along the same plane, which is parallel to the sliding face of the swash plate **508**. This configuration limits the size of the path travelled by the slipper shoe **526** along the sliding face of the swash plate **508**.

[0132] A lemniscate resembles the shape of an infinity sign; it may also be described as a Figure-**8**. This shape represents the path of which the slipper shoe **526** travels about the undersurface of the rotating swash plate **508** during displacement. In addition to the lemniscate path, the slipper shoe **526** also travels three hundred and sixty degrees about the swash plate **508** (more precisely, the swash plate rotates 360° about the non-rotating slipper shoes **526**). Effectively, the slipper shoe **526** moves radially, inwards and outwards, as the swash plate **508** is rotated. During this

rotation, the fluid within a slipper pressure pocket **550** (shown in FIGS. **12** and **13**) is constantly being sheared.

[0133] Each slipper shoe's slipper shoe pressure pocket **550** is centered on the flat surface of shoe that contacts the flat undersurface of the swash plate **508**. Each respective slipper shoe pressure pocket **550** is connected to a fluid supply to assure that fluid pressure present at the shoe/swash plate interface is proportional at all times with fluid pressure at the head of each piston **522**.

[0134] With reference to FIG. **13**, the slipper ball **528** engages with the piston **522** in slipper ball receiving portion **544**. Because the slipper ball **528** can actually go into the piston's bore **546**, one can limit the length of the piston **522**, which in turn limits the exposed surface (length) of the piston **522** at full extension, which in turn reduces unwanted moment loads. During operation of the pump **522**, axial motion of the piston **522** is translated to the slipper ball **528**. The slipper ball **528** also rotates about its center and therefore generates lateral motion and resulting forces. The lateral forces cause an unwanted moment about an axis orthogonal to the shaft's axis of rotation. To reduce this moment, lateral motion (and resulting forces) in the is reduced. One way to reduce lateral motion and forces is to reduce the stroke of the piston **522** as described elsewhere herein.

[0135] The swash plate **508** rotates with the shaft; but the slipper assembly **514** does not. There is a fluid film bearing **576** between swash plate **508** and the slipper shoe **526**. The slipper shoe **526** maintains alignment with the non-rotating pistons **522** and bore **546** notwithstanding the constantly varying tilt of the slipper shoe **526**. As such, the slipper shoe **526** moves in a lemniscate path within its slipper ball receiving recess **572**. The path of the slipper shoe **526** around the underside of the swash plate **508** is elliptical (not circular). Further, only the slipper shoe and hold down plate nutates (i.e. oscillation of an axis or revolution of a tilted axis about a central axis).

[0136] The hold down plate **510** holds the slipper assembly **524** in place against the swash plate **508**. The hold down plate **510** is kept in place on the swash plate **508** by the series of bearings **542** that fixedly engage the swash plate **508**. To assemble the hold down plate **510** on the swash plate **508**, the hold down plate **510** has a through hole in its center. The swash plate **508** slides through the center hole of the hold down plate **510** and the series of bearings **542** are slid on the swash plate **508**.

[0137] The hold-down plate **510** is provided with a plurality of openings, each of which surrounds a neck **530** of a respective slipper assembly **524**. A respective special washer **578** is fixed to integral with the slipper shoe **526**. Each washer **578** maintains the shoe in contact, via the fluid bearing **576**, with the flat undersurface of the swash plate **508** at all times. The hold down plate **510** is prevented from rotating independent of the slipper shoe **526** while at same time the hold down plate **510** does not restrict the movement of the slipper shoe **526**. The hold down plate **510** holds slipper shoe **526** flush with the swash plate **508**, maintaining pressure between slipper shoe **526** and the swash plate **508**.

[0138] The angle of the swash plate **508** is adjusted by pumping fluid into one of two volumes **582** and **584**. Fluid is pumped from an external source into the first volume **582** to raise a shift piston **580** or it is pumped into a second volume **584** to lower the shift piston **580**. The shift piston **580** is hydraulically actuated up and down the shaft **506** to change the angle of the swash plate **508**. Shift piston

displacement is controlled externally. Hydraulic control is affected through an external hydraulic control mechanism (not shown). Displacement of the shift piston **580** results in swash angle displacement, which results in displacement of pistons **522**, which produces fluid displacement.

[0139] With further reference to FIG. **13**, hydraulic fluid is allowed access to slipper shoe pressure pocket **550** via a through-hole **573** in the slipper ball **528**. The hydraulic fluid acts as both a seal and a bearing for the interaction of the slipper shoe **526** with the flat surface of the swash plate **508**. The neck **530** connects the slipper shoe **526** to the slipper ball **528**. The slipper neck **530** has a through-hole **574**, which is a continuation of the through-hole **573** of the slipper ball **528**. Slipper ball fluid entry **572** is in constant communication with piston through-hole **571** to enable access to high pressure fluid pumped by piston **522**. To this end, fluid entry **572** is preferably conical to account for the constant rotation of the slipper ball **528**.

[0140] With reference to FIG. **14**, an embodiment of the rotary valve **514** is a two piece assembly: thrust plate **600** and valve body **601** are held together by screws **552** (shown in FIG. **9**), adhesives or any other suitable fastener. Together they act as one valve **514**. Thrust plate **600** carries the thrust load created and carried through the shaft **506**. The shaft **506** drives the rotary valve **514** through a torque component only. The shaft **506** and the rotary valve **514** are mated solely to affect a torque or rotational movement of the valve. The rotary valve **514** includes the inlet **602** (FIG. **9**), an discharge passage **616** and balancing recesses **606a** and **606b**. The balancing recesses **606a** and **606b** are configured to receive hydraulic fluid such that, as the shaft **6** and the rotary valve **514** rotate and as the pistons **522** oscillate, the moments caused by the motion of these components is counterbalanced by the pressure within the balancing recesses **606a** and **606b**. A pre-compression notch **620** is provided on the leading edge of the discharge passage **616** directly on the curved surfaces of the rotary valve. This pre-compression notch **620** is provided to reduce damage to the valve resulting from a rapid increase in pressure at the discharge passage **616**. The pre-compression notch provides control of the rate of pressure increase within individual pumping chambers that minimizes the resulting impact forces on the pistons. By minimizing the impact forces on the pistons, the attendant noise that results from the contact between the slipper shoe and swashplate and eventually radiated to the local environment is minimized.

[0141] The thrust plate **600** is located on an upper portion of the rotary valve **514** opposite the inlet passage **602**. The surface area of the thrust plate **600** is equal to half of the total surface areas of the pistons (as half of the surface area will be under high pressure at any one time). A major obstacle for any axial piston pump is dealing with the large thrust loads, which are created by the continuously shifting high pressure force on the piston from the piston bore **546**. Half of the pistons will have a high pressure flow while the other half will have a low pressure flow, depending on the position of the piston within the bore. Lubrication happens in high or low pressure. The load is transferred from the piston **522**, to slipper shoe **526**, to swash plate **508**, to shaft pin **534**, and finally to the shaft **506**. The thrust plate **600** is used to counter act the thrust load, which otherwise would force the shaft **506** away from the cylinder block **516**.

[0142] It is not necessary to place low pressure recesses on the rotary valve as the film strength along the surface of the

rotary valve is enough to counter the smaller inlet pressure values. Further, adding additional cross passages would result in too much fluid loss.

[0143] As can be seen in FIGS. **11**A and **11**B, the recesses in the rotary valve **514** form one volume with the cylinder bores **546**. Thus, the pressure is equal throughout that volume. The recesses **606a** and **606b** help to counteract the moment caused by pressure rotation around the plurality of pistons **522**. The pistons **522** do not rotate around the axis of the shaft **506**; however, the reciprocal motion of the pistons **522** within their respective bores **546** causes a moment about an axis perpendicular to the axis of the shaft **506**. The recesses also allow for the inlet passage **536** to be centered along the axis of the rotary valve (and the axis of the pump). Due to the non-rotating pistons, centripetal forces are eliminated. The rotary valve **514** is therefore automatically self-centered with the use of cross porting hydrodynamic balance.

[0144] The recesses **606a** and **606b** of FIGS. **11**A-**11**B represent areas of high pressure; whereas the inlet **602** represents low pressure. The inlet pressure is close to atmospheric conditions unless otherwise provided with pressure. The outlet pressure can reach **6000** psi and more. The two recesses **606a** and **606b** are split apart by the high pressure outlet passage **612**. Recess **606a** is larger than recess **606b** to balance the moment about an axis perpendicular to the axis of rotation. The surface between the passages **606a** and **606b** is long enough to ensure a proper seal of the valve in its bore. The length of each of the passages **606a** and **606b** is based on fluid pressure. The following equation is used to determine a ratio of **606a** to **606b**:

[0145] Hydrodynamic thrust bearing **610** is a fluid filled volume between the thrust plate **600** and the manifold **518**.

[0146] The projected areas of the high pressure outlet **612** and valve discharge passage **616** are equal in size; thereby balancing the lateral forces on the rotary valve **514** in the. Additionally; these areas are positioned on the y-axis in such a way that the sum of the moments about the center of mass is zero. The thrust bearing at the upper end of the pump can handle one piston worth of area at operating pressure.

[0147] The rotary valve's inlet flow is through the inlet passage **602** in the center of the valve along the valve's axis (and accordingly along the axis of the pump **502**) and the rotary valve **514**'s outlet flow is ported around the periphery of the pump frame. The location of the outlet **612** has the additional benefit that it helps to cool the pump **502**. The inlet passage **602** through the center of the rotary valve **514** allows for a more direct flow path to the piston bore **546**. This also decreases the volume of the suction cavity as it affords a more direct flow path and lower surface area, which also reduces fluid friction and thereby reducing parasitic losses.

[0148] The present pump **502** combines a piston housing and a discharge cavity (manifold) in one unit (component). By combining high pressure forces into one housing, forces of separation that would normally present in prior art pumps are eliminated. Without the balancing effect of rotary valve **514**, the valve would tip or tilt, and potentially break the fluid barrier, and potentially seize the rotating parts of the pump **502**. With the forces, the valve will seek its own center within its bore.

[0149] The following embodiments are intended to avoid wear **698** on a rotary valve as show in FIG. **15**. Such wear

is typically caused by eccentric rotation and tipping of the valve within the pump's valve bore.

[0150] With specific reference to FIG. 16, a first disclosed embodiment is a pump 702 that includes a housing 704 and an integrated manifold 706. A rotary valve 708 is encompassed by the manifold 706. The rotary valve 708 includes a shaft 710, a ported section 712 and a sealing section 714 (shown in FIGS. 17A-17E). The sealing section 714 includes a semicircular sealing ridge 716. With particular reference to FIG. 17C, the pressure drop across the sealing ridge 716 is shown. Fluid passing through the ported section is at a much lower pressure relative to the pressure outside the ported section as shown by the pressure gradient 740.

[0151] With reference to FIGS. 18 and 19, the rotary valve 708 includes a first axial face seal 718 on a first end 720 of the rotary valve 708 and a second axial face seal 722 on a second end 724 of the rotary valve 708. A high pressure discharge section 726 is on a side of the rotary valve opposite the semicircular sealing ridge 716. An intake section 728 is on the first end 720 of the rotary valve 708.

[0152] As shown in FIG. 20, an operating gap 730 is between the high pressure discharge section 726 and the manifold 706. The operating gap 730 has a width that is dependent upon a force applied in the direction of the semi-circular sealing ridge at the high pressure discharge section 726. The side of the bore that engages with the sealing section 714 experiences a much lower pressure than the opposite side of the valve. That is because the opposite side of the valve experiences an extremely elevated pressure from discharge of the fluid from cylinders within the pump. The sealing section 714 remains at a generally constant operating distance from the manifold 706 throughout the life of the pump because the higher pressure on the discharge side of the valve pushes the valve toward the bore on the low pressure side of the valve. No matter how much the low pressure side (i.e. the sealing ridge 716 of sealing section 714) of the valve wears down, it will always push the valve toward the bore so that the sealing section 714 engages the bore surface (via a bearing/fluid bearing/mechanical bearing, etc.). As such, the valve always maintains a constant distance from the bore surface even as the valve experiences surface wear. The width of the operating gap is also dependent upon shaft speed and a circumference of the valve.

[0153] The rotary valve 708 is generally cylindrical and the semicircular sealing ridge 716 encompasses less than three hundred and sixty degrees of the rotary valve 708. The semicircular sealing ridge 716 comprises a manifold engaging portion 734 and a recessed portion 736. Ribs 748 are included for reinforcement of passage 750.

[0154] A further disclosed embodiment is shown in FIGS. 21-24. In FIGS. 21 and 22A, a pump 800 having a valve bore 802 encompasses a rotary valve 804. The rotary valve 804 has at least one axial face seal 806 on an end of the rotary valve 804 and at least one radial face seal 808 on the curved surface of the rotary valve 804. An operating sealing clearance gap 810 (FIG. 24) is between the radial face seal 808 and the valve bore 802. The operating sealing clearance gap 810 maintains a generally constant thickness during operation of the rotary valve 804.

[0155] A discharge gap 812 is located between the rotary valve 804 and the bore 802. The rotary valve 804 is biased in the direction of the radial face seal 808. The bias is caused by an applied force. The amount of the applied force determines a width of the operating sealing clearance gap 810. The applied force is determined by high pressure fluid exiting the pump 800 through the valve 804.

[0156] With particular reference to FIG. 23, the pressure drop 836 across the sealing ridge 716 is shown. Fluid passing through the inlet passage 820 is at a much lower pressure relative to the pressure outside of the passaged section 820 as shown by the pressure gradient 836.

[0157] The rotary valve 804 is monolithic and is made up of two portions, the passaged portion 830 and the shaft 814. The radial face seal 808 has a semicircular sealing component 816 and is generally coaxial with the shaft 814. The passaged portion 830 in this embodiment is coaxial with the shaft 814. The passaged portion 830 includes an inlet passage 820 and a discharge passage 822 (FIG. 22B). The pump 800 intakes fluid through the inlet passage 820 and enters piston bores that contain pistons as described above with respect to the previous embodiments, which are actuated by the rotating swash plate as also described above. Fluid is forced out of the pump by the same pistons at a much higher pressure than when the fluid entered the pump. The fluid exits the pump 800 via the discharge passage 822 in the passaged portion 830. As each piston pushes fluid toward the passaged section 830, the passaged section 830 rotates with the shaft 814 so that the discharge passage 822 receives the high pressure fluid from the pistons. The high pressure fluid is then directed toward an outlet 832 of the pump 800.

[0158] In a yet further embodiment as shown in FIGS. 25 and 26, a pump 900 has a rotary valve 906 within a valve bore 902, the hydraulic pump 900 includes a manifold 936. A shaft 904 is located within the manifold and is attached at a distal end 908 to a planar surface 910 of the rotary valve 906. With reference to FIG. 26 the axis of the shaft, a first axis, 912 and the axis of the rotary valve, a second axis, 914 are offset from each other so that they are not coaxial. Accordingly, at least one of the axes is not coaxial with the valve bore 902 when the shaft 904 and the rotary valve 906 are joined to each other and stationary.

[0159] The second axis 914 is coincident with an axis 916 of the valve bore 902. At times, the second axis 914 is coaxial with the axis 916 of the valve bore 902; however, there are times when the second axis 914 is not completely coaxial with the axis 916 of the valve bore 902. During these times, the second axis 914 and the axis 916 of the valve bore are considered merely coincident rather than coaxial due to their proximity, likelihood of becoming coaxial and coaxial rotation being optimum.

[0160] The first axis 912 is offset from a centerline 918 of the manifold 936. The rotary valve 906 has a high pressure outlet 920. The shaft 904 is offset in a direction opposite the high pressure outlet 920 of the rotary valve 906. However, the shaft 904 can also be offset in a direction toward the high pressure outlet 920 of the rotary valve 906.

[0161] A plurality of cylinders 926 within the manifold 936 are arranged in parallel and in a circle around and coaxial with the valve bore 902. The rotary valve 906 is generally coaxial with the valve bore 902. However, it is possible for the rotary valve 906 not to be coaxial with the valve bore 902. Also, the shaft 904 can be monolithic with the rotary valve 906 but it is not required.

[0162] The shaft 904 comprises a manifold contact surface 922. A bearing 924 is positioned in the manifold 936 and is in contact with the shaft 904. When the shaft 904 is offset in a direction away from the high pressure outlet 920, pressure

fluid flowing through the high pressure outlet **920** urges the second axis **914** of the rotary valve **906** toward the axis **916** of the valve bore **902**. Thus, the offset of the rotary valve **906** from the shaft **904** helps to account for the high pressure fluid flowing through the rotary valve **906** and therefore, helps to eliminate any eccentricity caused thereby.

[0163] Alternatively, moving the shaft **904** away from the high pressure outlet **920** urges the rotary valve away from the high pressure outlet **920**. This helps in securing the seal between a low pressure side of the rotary valve **906**, which is a side of the valve opposite the high pressure outlet **920**, and the valve bore **902**.

[0164] FIG. **27** shows a portion of a hydraulic pump **1000** that includes a valve **1002** in a bore **1004**. A radial face seal **1006** is on a radial surface **1008** of the valve **1002** and is positioned between the valve **1002** and the bore **1004**. The radial face seal **1006** at least partially circumferences the valve **1002**. A sealing ridge **1010** is on an outer circumference of the radial face seal **1006**. And a passage **1024** is located at a midway distance along the height of the radial face seal **1006**.

[0165] The sealing ridge **1010** protrudes toward the bore **1004** from the radial face seal **1006**. A minimally sized gap between the sealing ridge **1010** and the bore **1004** allows a fluid bearing to form between the sealing ridge **1010** and the bore **1004**. The fluid bearing provides for smoother rotation of the valve **1002** within the bore **1004**.

[0166] The radial face seal **1006** is C-shaped to fit coaxially with the valve **1002**. As shown in FIG. **32**, an O-ring **1048** is placed between the radial face seal **1006** and the valve **1002**. The O-4int **1048** is generally oval to encompass the passage **750**. A relief **1050** can be cut into the valve for the O-ring (or anywhere else for any other O-ring) but it is not necessary. With additional reference to FIG. **17***c*, there is a pressure drop toward the passaged section **712**, which is analogous with passage **1024** of the present embodiment as shown in FIG. **29**, from the recessed section **734**, which is analogized to the portion of the radial face seal above and/or below the sealing ridge **1010** in the present embodiment. This is because the fluid entering the pump enters through the passage **1024** at a lower pressure relative to the portion of the radial face seal above and/or below the sealing ridge **1010**. The upper and lower edges of the sealing ridge **1010** are at a greater pressure than the part of the sealing ridge **1010** closest to the bore. Further, as in select previous embodiments, higher pressure on a discharge side of the valve **1002** pushes the valve **1002** toward the bore **1004** on a low pressure side of the valve **1002**, i.e., on a sealing ridge side of the valve **1002**. The high pressure side will always push the radial face seal **1006** toward the bore **1004** so that the sealing ridge **1010** engages the bore surface (via a bearing/fluid bearing/mechanical bearing, etc.). The sealing ridge **1010** completely encompasses the passage **1024** so that the passage is shielded as much as possible from a pressure differential within the pump.

[0167] The radial face seal **1006** is connected to the valve **1002** via at least one radial face seal pin **1016** (alternatively referred to as a "torque translator"). Each radial face seal **1006** is biased away from the valve **1002** via a radial face seal spring **1018**. A torque translator **1016** translates rotational force from the valve **1002** to the rotary face seal **1006**. The torque translator can be a pin, a protrusion, an high friction surface such as an O-ring, or any other means capable of translating torque from one component to

another. As one of the purposes of the radial face seal **1006** is to provide a seal between the sealing ride **1010** and the bore **1004**, the radial face seal **1006** is biased toward the bore **1004** so that the seal is maintained whether the pump operates without a pressure differential or whether the pump is not in operation at all. Each radial face seal pin **1016** sits inside a recess **1036** of valve **1002** and each radial face seal spring **1018** is generally coaxial with the pin and sits on a shoulder **1038**.

[0168] An axial face seal **1012** is on an axial surface **1014** of the valve **1002**. The axial face seal **1012** is between the axial surface **1014** of the valve and an axial surface **1015** of the bore **1004**. A ridge **1017** is in contact with the axial surface **1015** of the bore via a fluid bearing. The axial face seal **1012** is also connected to the valve **1002** via at least one axial face seal pin **1020**. A second axial face seal **1026**, which is configured similarly to the axial face seal **1012**, is located at a second axial surface **1003** of the valve **1002**. Sealing of the second axial surface would be via a ridge in contact with the bore as with the axial surface **1015**.

[0169] Each axial face seal **1012** is biased away from the valve **1002** via an axial face seal spring **1022**. As can be seen in FIG. **27** and FIG. **28**, an O-ring **1030** is provided between the radial face seal **1006** and the radial surface **1008** and between the axial face seal **1012** and the shaft. And as can be seen in FIG. **29**, the sealing ridge **1010** has a passage **1024**, which is an oval through-hole providing fluid access to the valve **1002**. The O-ring **1030** provides a static seal between the radial seal and the valve while allowing small radial movement between the two components and still providing the seal. The axial face seal **1012** provides a similar seal via an O-ring while allowing movement between the axial face seal and the valve while still providing the seal.

[0170] FIG. **30** shows a double sided hydraulic pump **1100**. The hydraulic pump **1100** includes a swash plate **1102** that has a first side **1104** and a second side **1106**. The first side of the swash plate **1104** is generally parallel to the second side **1106**.

[0171] A first portion **1108** of the pump is on the first side **1104** of the swash plate. And a second portion **1110** of the pump is on the second side **1106** of the swash plate. The first portion **1108** has a first portion rotary valve **1112** and is encompassed by a first portion bore **1114**. And the second portion **1110** has a second portion valve **1116** and is encompassed by a second portion bore **1118**. Both valves have a shaft **1120***a*, **1120***b*, and a radial seal **1126***a*, **1126***b*. Each radial seal has a sealing ridge **1124***a*, **1124***b* and a passaged section **1122***a*, **1122***b* that is in communication with a respective passage **1125***a*, **1125***b* in the respective valve **1112**, **1116**.

[0172] At least one of the portions **1108**, **1110** has a respective radial face seal **1126***a*, **1126***b* connected to the respective valve **1112**, **1116**, which provides a seal between the respective valve **1112**, **1116** and the respective bore **1114**, **1118**. A torque translator **1128** connects the respective radial face seal **1126***a*, **1126***b* with the respective valve **1112**, **1116** for transmitting rotational force from the valve **1112**, **1116** to the respective radial face seal **1126***a*, **1126***b*. O-rings **1034***a*, **1034***b* are provided between both of the radial face seals **1126***a*.

[0173] To seal off the fluid flow through the pump through the rest of the pump, an axial face seal **1026** should be on the side of the valve opposite the opposing portion of the valve.

As can be seen in FIG. **31**, the radial face seal **1006** is subjected to a pressure gradient. The pressure gradient is a result of a higher pressure section **1040** applying pressure to the valve **1002** from fluid in the cylinders **1042** being pressed into the valve by pistons. The pressure at the lateral extremities **1044** of the radial face seal **1006** is higher than the pressure at the mid portion **1046** of the radial face seal **1006**.

[0174] The aforementioned disclosure is described as a pump. However, a person having ordinary skill in the art would recognize that the disclosed device can function as a hydraulic motor, engine, etc.

[0175] While the foregoing has described what are considered to be the best mode and/or other examples, it is understood that various modifications may be made therein and that the subject matter disclosed herein may be implemented in various forms and examples, and that the teachings may be applied in numerous applications, only some of which have been described herein. It is intended by the following claims to claim any and all applications, modifications and variations that fall within the true scope of the present teachings.

What is claimed is:

1. A piston device comprising:

a stationary cylinder block,

a shaft that is rotatable about its own axis within the stationary cylinder block, and the shaft comprising a plurality of passages configured to receive, direct and exhaust fluid, the shaft centrically and eccentrically rotatable during operation of the pump, and

a swash plate coupled to the shaft, the swash plate having a first side coupled to a first plurality of pistons.

2. The piston device as recited in claim **1** wherein the swash plate is configured to tilt about an axis perpendicular to the axis of the shaft.

3. The piston device as recited in claim **2** further comprising a means for tilting the swash plate.

4. The piston device as recited in claim **1** wherein fluid pressure on the passages in the shaft balances at least one force and/or moment.

5. The piston device as recited in claim **1** further comprising a respective plurality of slipper shoes each connected to a piston, and the swash plate comprising at least one active surface for interacting with the slipper shoes.

6. The piston device as recited in claim **1** wherein fluid flow throughout the pump creates at least one of a force and a moment on the shaft and the plurality of passages are configured on the shaft such that at least one of the forces or the moments on the shaft are balanced.

7. The piston device as recited in claim **6** further comprising a second plurality of pistons, each plurality of pistons contained within the cylinder block and arcuately positioned coaxial with the shaft on opposite ends of the cylinder block.

8. The piston device as recited in claim **7** further comprising a first active surface and a second active surface, the first and second active surfaces are on opposing sides of the swash plate.

9. A piston device comprising:

a stationary cylinder block;

a shaft extending through the stationary cylinder block and having a plurality of passages along a portion of the shaft; and

a valve enclosed within the stationary cylinder block, the valve comprising the shaft integrally coupled to the valve.

10. The piston device as recited in claim **9** further comprising a first plurality of pistons and a second plurality of pistons, each plurality of pistons arcuatley positioned coaxial with the shaft on opposite sides of the shaft.

11. The piston device as recited in claim **9** wherein the plurality of passages are configured to receive, direct and exhaust fluid flow throughout the pump.

12. The piston device as recited in claim **11** wherein the fluid flow throughout the pump creates at least one of a plurality of forces and a plurality of moments on the shaft and the plurality of passages are configured on the shaft such that the at least one of the plurality of forces and the plurality of moments is balanced.

13. The piston device as recited in claim **12** further comprising a plurality of slipper assemblies, each slipper assembly coupled to one piston of the plurality of pistons; and

a swash plate coupled to the shaft and comprising at least one active surface for interacting with a respective slipper assembly.

14. The piston device as recited in claim **13** further comprising a first active surface and a second active surface, the first and second active surfaces are on opposing sides of the swash plate.

15. A piston device comprising:

a swash plate having a first side coupled to a first plurality of slipper shoes and a second side coupled to a second plurality of slipper shoes, the first side of the swash plate is generally parallel to the second side of the swash plate; and

a first portion of the pump on the first side of the swash plate and a second portion of the pump on the second side of the swash plate, the first portion of the pump on the first side of the swash plate is substantially symmetric about the swash plate with the second portion of the pump.

16. The piston device as recited in claim **15** further comprising a shaft, the swash plate coupled to the shaft

17. The piston device as recited in claim **16** configured to be adjustable about an axis askew or perpendicular to an axis of the shaft.

18. The piston device as recited in claim **16** further comprising a means for balancing at least one of a plurality of forces and a plurality of moments acting on the shaft.

19. The piston device as recited in claim **17** further comprising a means for tilting the swash plate with respect to the shaft.

20. The piston device as recited in claim **16** further comprising a housing encasing the shaft, swash plate and slipper assemblies.

21. A piston device comprising:

a stationary cylinder block,

a valve-shaft assembly encompassed by the stationary cylinder block, the valve-shaft assembly comprising a valve connected to a shaft of the valve shaft assembly via interaction between a drive key and a valve keyway in the valve and a shaft keyway in the shaft, and wherein the interaction is configured to provide at least one degree of freedom for the valve relative to the shaft.

22. The piston device as recited in claim **21** wherein the valve is configured to balance within the cylinder block by controlled leakage rates of fluid within the hydraulic pump as a function of at least one of a force and a moment on the valve

23. The piston device as recited in claim **22** further comprising a swash plate connected to the shaft.

24. The piston device as recited in claim **23** further comprising a means for tilting the swash plate about an axis perpendicular to the axis of the shaft.

25. The piston device as recited in claim **24** wherein the swash plate comprises at least one active surface.

26. The piston device as recited in claim **25** wherein the swash plate comprises a plurality of slipper assemblies coupled to the at least one active surfaces of the swash plate.

27. A piston device comprising:

a housing and a manifold integral with the housing, and

a rotary valve encompassed by the manifold, the rotary valve comprising a shaft, a passaged section and a sealing section,

wherein the sealing section comprises a semicircular sealing ridge.

28. The piston device as recited in claim **27**, the rotary valve further comprising a first axial face seal on a first end of the rotary valve and a second axial face seal on a second end of the rotary valve.

29. The piston device as recited in claim **28**, the rotary valve further comprising a high pressure discharge section on a side of the rotary valve opposite the semicircular sealing ridge.

30. The piston device as recited in claim **29**, the rotary valve further comprising an intake section on the first end of the rotary valve.

31. The piston device as recited in claim **29** further comprising an operating gap formed between the high pressure discharge section and the manifold, the operating gap having a width that is dependent upon a force applied in the direction of the semi-circular sealing ridge at the high pressure discharge section.

32. The piston device as recited in claim **31** wherein the semicircular sealing ridge is configured to remain at a generally constant operating distance from the manifold.

33. The piston device as recited in claim **31** wherein the width of the operating gap is further dependent upon shaft speed and a circumference of the valve.

34. The piston device as recited in claim **27** wherein the rotary valve is generally cylindrical and the semicircular sealing ridge encompasses less than three hundred and sixty degrees of the rotary valve.

35. The piston device as recited in claim **34** wherein the semicircular sealing ridge comprises a manifold engaging portion and a recessed portion.

36. The piston device as recited in claim **28**, the rotary valve further comprising a low pressure suction section on a side of the rotary valve opposite the semicircular sealing ridge.

37. A piston device comprising:

a valve bore,

a rotary valve encompassed by the valve bore, the rotary valve comprising at least one axial face seal and at least one radial face seal,

an operating sealing clearance gap between the radial face seal and the valve bore,

wherein the operating sealing clearance gap maintains a generally constant thickness during operation of the rotary valve.

38. The piston device as recited in claim **37** further comprising a discharge gap between the rotary valve and the bore, the rotary valve being biased by a force applied in the direction of the semi-circular sealing ridge, wherein the force determines a width of the operating sealing clearance gap.

39. The piston device as recited in claim **38**, the rotary valve further comprising a shaft, wherein the radial face seal comprises a semicircular sealing component and is generally coaxial about the shaft.

40. The piston device as recited in claim **39** further comprising a passaged section on the shaft.

41. The piston device as recited in claim **40** wherein the shaft comprises an inlet passage and a discharge passage.

42. A piston device comprising:

a housing having a valve bore and manifold,

a shaft within the manifold and a rotary valve within the valve bore, the shaft attached at a distal end to a planar surface of the rotary valve; and

the shaft having a first axis and the rotary valve having a second axis, the first axis being offset from the second axis.

43. The piston device as recited in claim **42**, wherein the second axis is coincident with an axis of the valve bore.

44. The piston device as recited in claim **43** wherein the first axis is offset from a centerline of the manifold.

45. The piston device of claim **42** wherein the rotary valve comprises a high pressure outlet.

46. The piston device as recited in claim **45** wherein the shaft is offset in a direction of the high pressure outlet of the rotary valve.

47. The piston device as recited in claim **46** wherein the shaft comprises a manifold contact surface.

48. The piston device as recited in claim **45** wherein the shaft is offset in a direction opposite the high pressure outlet of the rotary valve.

49. The piston device as recited in claim **42** further comprising a plurality of cylinders within the manifold arranged in parallel and in a circle around and coaxial with the valve bore.

50. The piston device as recited in claim **49** wherein the rotary valve is generally coaxial with the valve bore.

51. The piston device as recited in claim **50** wherein the shaft is not coaxial with the valve bore.

52. The piston device as recited in claim **42** the shaft being monolithic with the rotary valve.

53. A piston device comprising

a valve in a bore,

a radial face seal on a radial surface of the valve positioned between the valve and the bore and at least partially circumferencing the valve, the radial face seal comprising a sealing ridge and

an axial face seal on an axial surface of the valve, the axial face seal between the axial surface of the valve and the axial surface of the bore.

54. The piston device as recited in claim **53** wherein the sealing ridge protrudes toward

the bore away from the radial surface of the valve.

55. The piston device as recited in claim **53** wherein the radial face seal is connected to

the valve via at least one radial face seal torque transmitting device.

56. The piston device as recited in claim 55 wherein the radial face seal is biased away

from the valve via at least one radial face seal spring.

57. The piston device as recited in claim 53 wherein the axial face seal is connected to

the valve via at least one axial face seal torque transmitting device.

58. The piston device as recited in claim 57 wherein the at least one axial face seal is

biased away from the valve via an axial face seal spring.

59. The piston device as recited in claim 53 wherein the sealing ridge further comprises

a passage.

60. A piston device comprising:

a swash plate having a first side and a second side, the first side is generally parallel to the second side;

a first portion of the pump on the first side and a second portion of the pump on the second side,

the first portion comprising a first portion rotary valve encompassed by a first portion bore and the second portion comprising a second portion valve encompassed by a second portion bore,

wherein both valves comprise a shaft, a passaged section and a sealing ridge.

61. The piston device as recited in claim 60 wherein at least one of the portions further

comprises a respective radial face seal connected to the respective valve and

providing a seal between the respective valve and the respective bore.

62. The piston device as recited in claim 61 further comprising torque translator

connecting the respective radial face seal with the respective valve for transmitting rotational force from the valve to the respective radial face seal.

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