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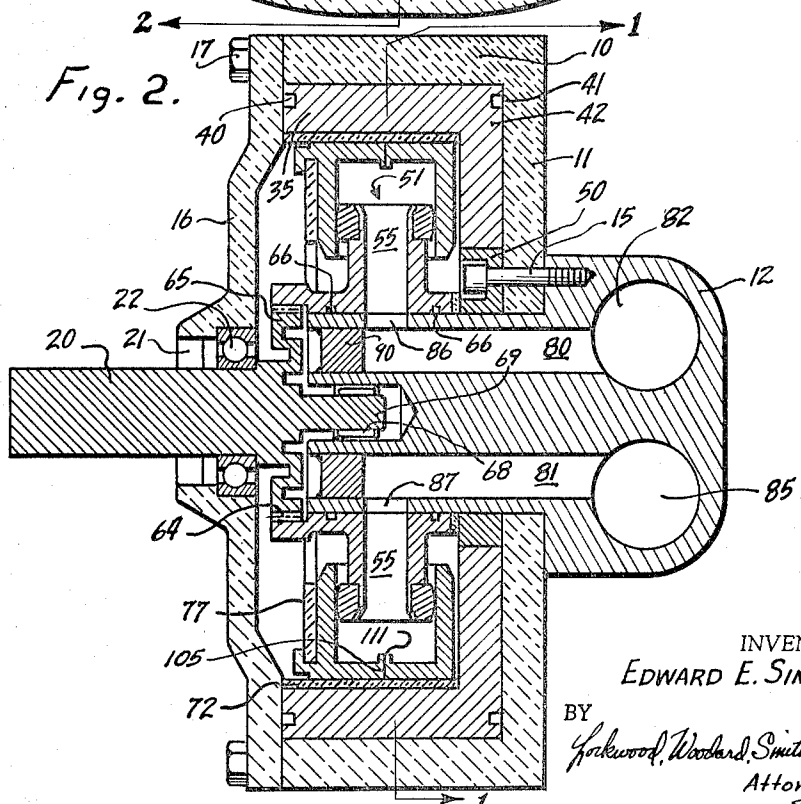
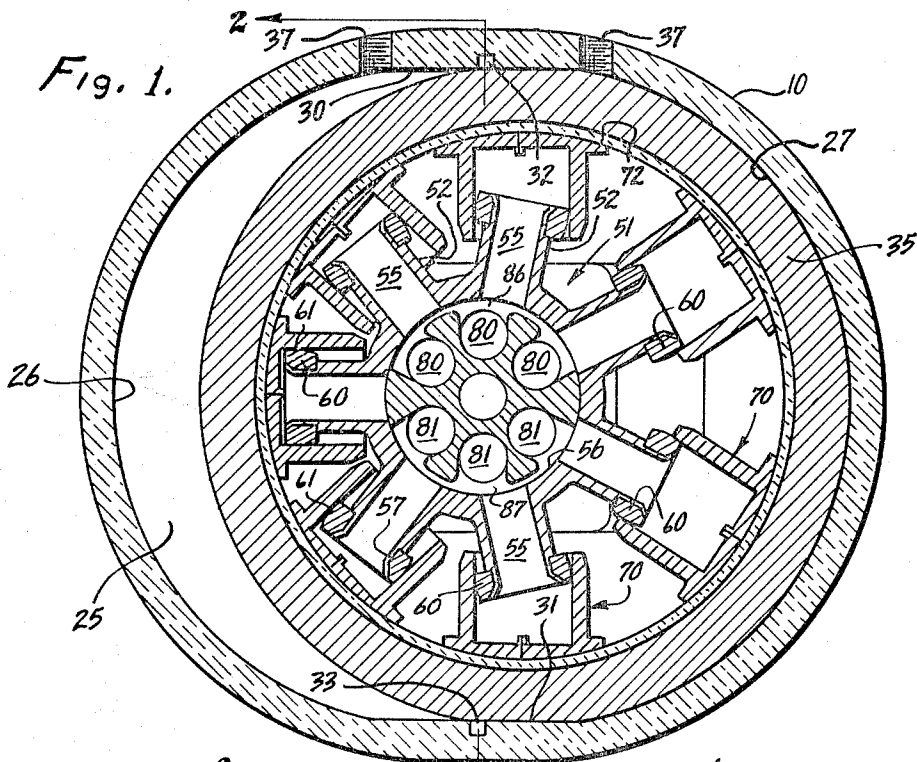
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3 Sheets-Sheet 1



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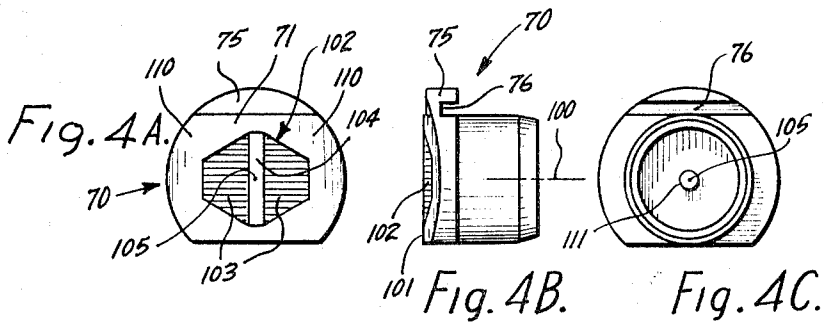
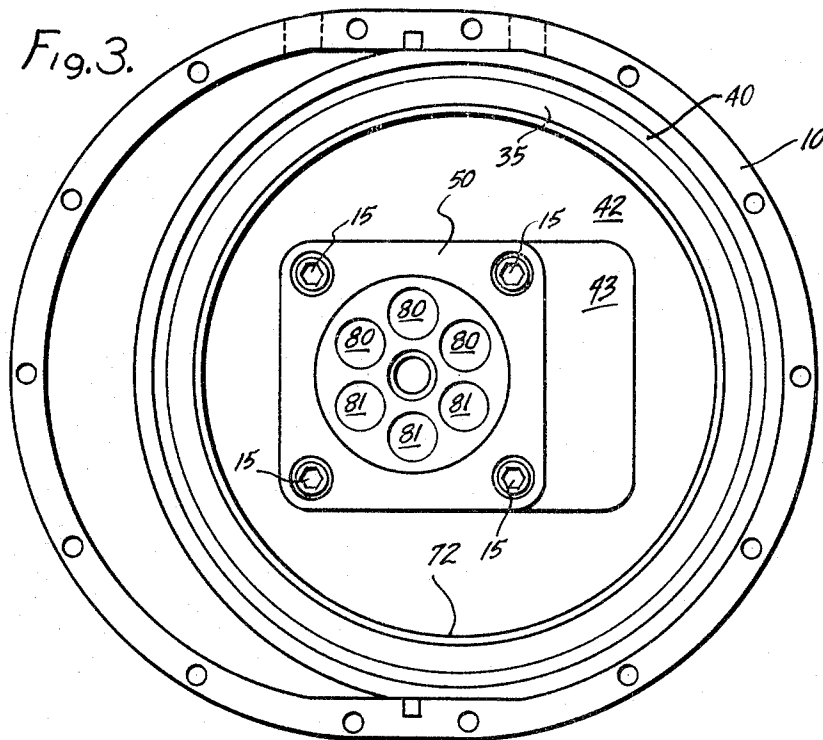
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Fig. 5.

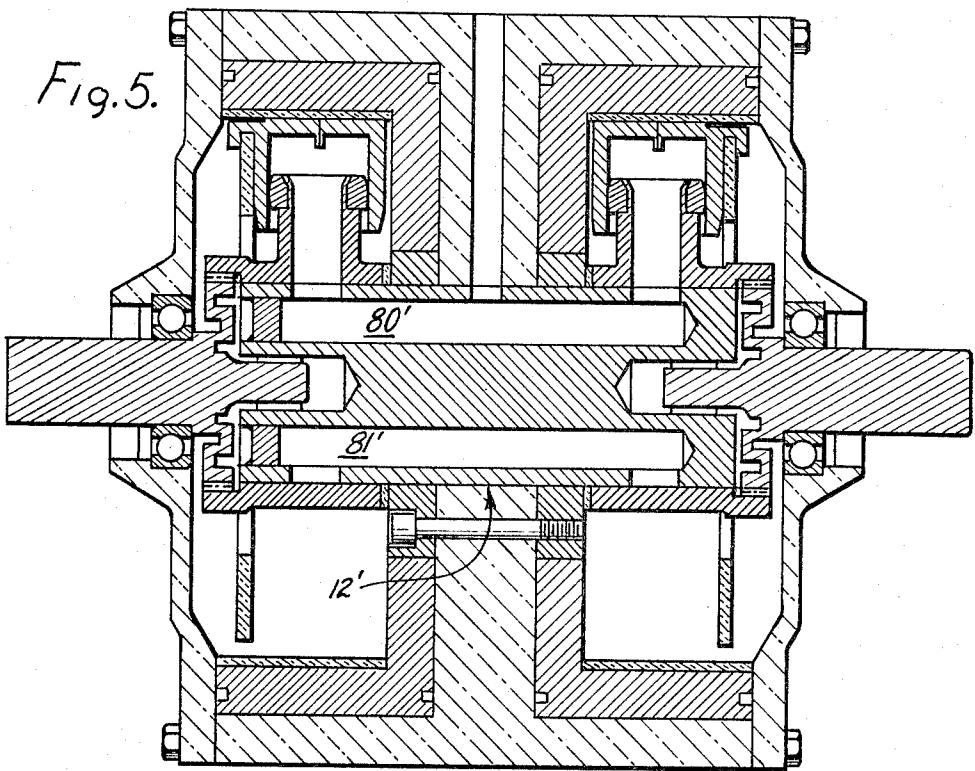
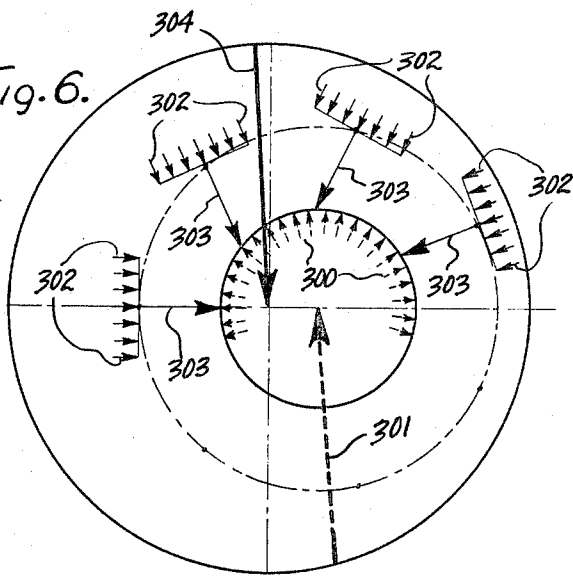


Fig. 6.



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This invention relates to an improved power transfer device which may be used as a pump or motor.

One object of this invention is to provide a power transfer device having exceptional capacity, ruggedness, and simplicity and incorporating parts readily manufactured and serviced at nominal cost. A further object of this invention is to provide a power transfer device having improved efficiency. According to the invention, a contributing feature to the solution of these objects is the converting of hydraulic pressure to mechanical torque or mechanical torque to hydraulic pressure with fluid pressure balanced components whereby the need for heavily loaded rolling or sliding bearings is eliminated.

Another object of this invention is to provide a power transfer device in which wear is eliminated or reduced as compared to conventional such devices. One feature of the invention is the wedging of a dynamic oil film between ball pistons and cylinders to reduce wear. Wear is also reduced in the present invention by eliminating unbalanced side thrust loads between ball pistons and cylinder walls.

A further element of the invention contributing to the solution of the objects thereof is the use of free body separate floating cylinders and pistons free from thermal or strain distortions of adjacent cylinders or pistons.

Another object of this invention is to provide a power transfer device having high displacement.

Still another object of this invention is to provide a power transfer device having superior thermal stability preventing seizures of moving parts.

A further object of this invention is to provide a power transfer device incorporating means for preventing air from entering the working fluid of the device.

A further object of this invention is to provide a power transfer device having a minimum of pressure lines and junctions.

The full nature of the invention will be understood from the accompanying drawings and the following description and claims.

FIG. 1 is a section taken through the power transfer device of the present invention, said section being taken along the line 1-1 of FIG. 2 in the direction of the arrows.

FIG. 2 is a section taken along the line 2-2 of FIG. 1 in the direction of the arrows.

FIG. 3 is a front elevation of the structure shown in FIG. 1 but with the rotor and cylinders of the present device removed to show certain internal structure of the device.

FIGS. 4A, 4B and 4C are end elevational, side elevational and opposite end elevational views, respectively, of a cylinder and slipper shoe bearing forming a part of the structure of FIGS. 1 and 2.

FIG. 5 is a section similar to FIG. 1 of a variable displacement fully reversible hydrostatic transmission embodying the present invention.

FIG. 6 is a force vector diagram showing the various forces involved in the present invention wherein hydraulic thrust is converted to torque by a pressure balanced rotor.

For the purposes of promoting an understanding of the principles of the invention, reference will now be made to the embodiment illustrated in the drawing and

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specific language will be used to describe the same. It will nevertheless be understood that no limitation of the scope of the invention is thereby intended, such alterations and further modifications in the illustrated device, and such further applications of the principles of the invention as illustrated therein being contemplated as would normally occur to one skilled in the art to which the invention relates.

It should be understood that the present invention may operate as a pump or as a motor or as both in the hydrostatic transmission of FIG. 5. In the description following, however, the invention is described principally as a hydraulic pump which has been found to be capable of operating at pressures in the neighborhood of 5,000 p.s.i.

Referring to the drawings, there is illustrated a housing 10 which is closed at one end by an integral end portion 11 and a pintle valve 12 fixed to the end portion 11 by screws 15. The opposite end of the housing 10 is closed by end cover 16 which is fixed to the housing 10 by screws 17. The end cover 16 provides journal mounting for an input shaft 20 which projects through the central opening 21 in the end cover 16 and is rotatably received within ball bearings 22 mounted within the opening 21.

The housing 10 and end cover 16 define therewithin a chamber 25 which is bounded by two semi-circular surfaces 26 and 27 joined by two straight or flat surfaces 30 and 31. Elongated resilient seals 32 and 33 are received within suitable grooves in the surfaces 30 and 31 and act to prevent hydraulic fluid from moving from one side to the other of a cylindrical race 35 received within the housing 10. The cylindrical race 35 may be moved from one side to the other of the chamber 25 by pumping hydraulic fluid into the passage 36 or the passage 37 in the wall of the housing 10 and by removing hydraulic fluid from the other of the two passages 36 and 37. As will become evident below, the adjustable positioning of the race 35 makes possible the adjustability of the eccentricity of the pump and the adjustment of the displacement and direction of flow of the pumped fluid. It should also be mentioned that the mounting or reception of the race 35 within the fluid at the chamber 25 acts to dampen vibration and to reduce operational noise of the present device.

Referring more particularly to FIG. 3, the race 35 is provided with annular resilient seals 40 and 41 also functioning to prevent leakage of the race positioning hydraulic fluid into the central portion of the chamber 25 and retaining said fluid generally outside or on either side of the race 35. The race 35 further includes a back portion 42 which has a rectangular shaped opening 43 therethrough. The screws 15 function not only to fix the pintle valve member 12 in position but also function to secure a race support plate 50 to the rear 11 of the housing 10. It can be appreciated that the race 35 is retained against rotation by the plate 50 but can be adjusted from one side to the other of the chamber 25 by riding upon the plate 50.

The pintle valve 12 also functions as a journal for a rotor 51. The rotor 51 includes a plurality of radially extending spokes 52 each of said spokes having a radial passage 53 therethrough. Each of the passages 53 extends from a central cylindrical opening 56 in the rotor to the distal end 57 of the respective spoke.

A plurality of annular bearings 60 are each swaged upon the distal end 57 of a respective one of the spokes 52. Each of the bearings 60 is formed of a suitable hard bearing material and has an external peripheral surface 61 which is part-spherical in shape. Each of the annular bearings 60 has a constant sized cross sectional configuration which is cut at and just adjacent to the great circular

portion of a sphere. Each of the members 60 is coaxial with its spoke 52 with all of said spoke and bearing axes being radial of the journal mounting of the rotor 51 on the valve member 12.

The input shaft 20 is coupled to the rotor 51 by means of a flexible coupling 65 having splines 64 around its outer periphery which engage complementary inwardly directed splines on the rotor. It has been found that at high pressure operation of the pump, it is desirable that the rotor, journal 12 and the rotor 51 establish an oil film clearance therebetween to control leakage of fluid between the rotor and the member 12. Annular grooves 66 are provided in the opening 56 of the rotor for a purpose described below. The flexible coupling 65 helps to prevent excess leakage by permitting a certain amount of radial as well as axial freedom of the rotor 51 so that said oil film clearance can be established. Thus, the ball bearings 22 and also roller bearings 68 receiving the end 69 of the shaft primarily locate the input shaft relative to the member 12 and the cover 16.

Over each of the annular bearing elements 60, there is received a cylinder 70. Each of the cylinders 70 is reciprocated relative to its respective spoke and bearing 60 so that the bearing 60 acts as a piston within the cylinder. The details of the cylinders 70 are shown in FIGS. 4A, 4B and 4C as including a cylinder head 71 formed integrally with the cylinder 70.

Cylinder head 71 forms what is herein termed a slipper shoe bearing which is free to follow the eccentrically located path of the race 35. It should be mentioned that a suitable cylindrical bearing 72 is received within the race 35 and is the member against which the cylinders 70 act. Each of the cylinders 70 has a projection 75 on one side thereof and within which there is formed a groove 76. A ring 77 is received within each of the grooves 76 and forms a mechanical guide element for the cylinders causing them to maintain or follow a circular path around the outer race bearing 72. The ring 77 also keeps the cylinders in proper position when the pump is stopped so that the cylinders cannot strike or dent the race bearing 72 during a quick start.

The pintle valve 12 has six longitudinally extending passages 80 and 81 formed therein. Three of the passages 80 join within the passage 82 while the other three passages 81 join within the passage 85. Either the passage 82 or 85 may be the inlet passage for the pump while the other passage will be the outlet passage.

All of the passages 80 and 81 open outwardly through the wall of the pintle valve at 86 and 87. That is, all of the passages 80 again join at 86 and all of the passages 81 again join at 87. The inner end of the pintle valve 12 is closed by six individual plugs 90 which are brazed to the member 12. It will be noted that the passage portion 86 extends approximately 140° around the member 12 while the passage portion 87 extends approximately 140° around the opposite side of the member 12.

When the input shaft 20 is rotated, the rotor 51 is also rotated causing the cylinders 70 to slide around the bearing 72 and to reciprocate and oscillate relative to the annular bearings 60. As each cylinder 70 moves outwardly relative to the spoke 52, oil is drawn through the passage 80 or 81 and through the passage 55 in the respective spoke into the respective cylinder. As the rotor turns past the maximum eccentricity, each respective cylinder moves radially inwardly relative to the respective spoke and displaces oil through the spoke and into the opposite passage 80 or 81 in the pintle valve 12. Of course, the direction of pumping can be reversed by reversing the direction of rotation of the input shaft 20 or by reversing the eccentricity of the race 35 by moving it from one side to the other side of the chamber 25.

It can be seen that each piston 60 acts both as a friction-free oscillating wrist pin and also as a reciprocating piston as long as the spherical surface of the bearing has adequate width to completely seal the cylinder diameter at

maximum tilt angle. No engagement overlap is required for alignment stability and, therefore, the device is designed so that the piston sweeps the full cylinder length.

The cylinder axis 100 is normal to the surface 101 of the slipper shoe 71. Formed within the surface 101 is a hydrostatic bearing pocket 102 which has two sides 103 connected by a flat groove 104. During operation, the fluid pressure within the pocket 102 rises causing the slipper shoe 71 to rise out of contact with the bearing 72 and thus causes the cylinder 70 to float. A small restricted sized orifice 105 extends from the pocket 102 into the inside of the cylinder. Because the pressure within the pocket 102 is less than the pressure inside the cylinder during operation of the pump, some of the oil will flow through the restricted orifice 105 into the pocket 102. It can be appreciated that this supply of oil will increase when the internal pump pressure is greater so that there is always provided a proper oil film between the cylinder 70 and the bearing 72.

When the cylinder is in the above mentioned floating condition, there are no unbalanced side loads from hydraulic or centrifugal force acting upon the cylinder. Also, because the cylinder is closed at the bearing piston end and with a right circular seal normal to the cylinder axis, there are no side thrust forces on the cylinder regardless of oil pressure.

The fact that the cylinders 70 float permits the dynamic wedge oil film between the piston bearing and the cylinder to keep these two members out of contact at the seal line. Consequently, wear without load is nominal.

Each of the cylinders 70 is obviously separate and apart from the remaining cylinders and consequently, cooling of the cylinders provides no problem. Preferably, the cylinders 70 and the bearings or pistons 60 are formed from an alloy steel having a similar coefficient of thermal expansion so that the clearance between these members remains the same even though the pump structure becomes heated. Noise in the present device is reduced because the slipper shoe 71 for each cylinder adjusts its own film clearance whereby there is no mechanical looseness to produce noise. For normal operation, the centrifugal force and oil pressure hold the cylinders smoothly against the race bearing 72.

Referring again more particularly to FIG. 4, the slipper shoe 71 is a combination bidirectional hydrostatic and hydrodynamic film design. The hydrostatic portion incorporates the feed orifice 105 whereby oil flows into the pocket 102. Hydrostatic bearing capacity is approximately equal to the pocket pressure times the pocket area plus one-half the pocket pressure times the pocket seal area (slipper shoe area outside of the pocket 102). Preferably, the hydrostatic bearing capacity is about one and one-half times the cylinder load. When the pocket pressure reaches two-thirds of the cylinder internal pressure, the slipper shoe 71 lifts away from the bearing 72 to a floating regulating condition. In such a condition, the pressure drop across the orifice 105 from inside to outside the cylinder is about one-third the cylinder pressure.

The hydrodynamic feature of the present device is provided by the fact that as the shoe 71 travels in one of two directions, oil or another lubricating fluid is wedged between either one or the other of the toes 110 of the slipper shoe and the bearing 72. This dynamic wedge is provided because of the fact that the curvature of the surface 71 is to a radius which is less than and in one preferred embodiment is .002 inch less than the radius of curvature of the bearing 72. It can be appreciated, therefore, that the viscous lubricating fluid creates a dynamic wedge action lifting the toe of the cylinder.

Of course, when one of the portions 110 acts as the toe of the shoe, then the other of the portions 110 acts as the heel thereof. The heel of the shoe is caused to ride out of contact with the race bearing 72 by reason of the increased pressure of the trapped fluid within the pocket 102 and between the pocket and the race bearing. This trapped fluid raises the pressure at the heel portion

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of the shoe so that the shoe has a toe and heel support facilitating stable operation.

Of course, one feature greatly adding to the success of the above described slipper bearing is the universal type connection between the cylinder and the annular bearing 60 which permits maximum freedom for the shoe and permits it to align with the race bearing regardless of deflection or minor geometry machining errors in the parts.

A further feature of the present invention is the ability of the device to operate in the presence of dirt in the pumped fluid. The cup-like cylinders 70 have a centrifuge action which causes the dirt particles to collect at the inside base of the cylinder 70. Large dirt particles are prevented from entering the orifice 105 by a radially inwardly projecting tube 111. The extent of projection of this tube is optional depending upon the protection desired. The fact that the centrifugal action on the dirt is radially outwardly away from the seal between the bearing 60 and the cylinder keeps dirt particles away from the cylinder seal. It has been found that the hardened steel members 60 and 70 have considerable ability to crush and digest a certain amount of dirt without problems. However, molybdenum or Carboloy surfaces are advantageous in certain applications. It can be appreciated that one feature of the present pump is the capability of servicing individual pistons and cylinders independently of the remaining pistons and cylinders.

A further feature of the present invention is its ability to exclude air from the pumped fluid. At the race bearing 72, air could be drawn back through the orifice 111, but the continuous centrifugal shoe contact apparently maintains this orifice submerged in fluid at all times. At the junction between each piston 60 and its cylinder 70, the oil that leaks past the piston on the pressure stroke is centrifugally held around in the slipper shoe 71 to substantially flood this junction of the slipper shoe and bearing race so that air does not move between two members on the suction stroke.

At the junction between the rotor and the pintle 12, oil leaking from the pressure port between the rotor and pintle fills the seal grooves 66 with a certain amount of pressure oil, which, in turn, keeps the rotor to pintle junction around the suction port flooded with oil excluding air.

Torque is developed in the present device by a true hydrostatic thrust couple. Referring to FIG. 6, to convert hydraulic thrust to torque this rotor has a high outward thrust exerted thereon from the full hydraulic pressure on the pintle port area plus a lesser hydraulic pressure on the pintle seal area around each port. These forces are represented by the arrows 300 and resultant 301. Each element of the outward thrust is normal to the rotor journal surface with a common or resultant vector point at the center of the pintle. Inward thrust, represented by arrows 302 and resultants 303 and composite resultant 304 is exerted on each ball piston in line with each cylinder axis. Each cylinder axis is pointed directly at the geometric center of the outer race. The thrust or vector sum of the forces from all the cylinders form the total resultant force or vector 304 through the axis of the outer race.

If the pintle center and the outer race center are common, there is no torque, however if the outer race is moved eccentrically relative to the pintle center, the torque will be the vector sum of all the piston forces times the race eccentricity. To complete the rotor balance, the pintle pressure port area is proportioned by seal design to exactly equal the piston thrust. Therefore, theoretically the rotor is in pressure balance all the time with no mechanical bearing loads, and the torque couple is delivered through the splined coupling 65 to the shaft. With floating cylinders on slipper shoes, this pump has balanced hydrostatic thrust bearings at all junctions.

The limit of capacity of hydrostatic bearings is un-

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known, but final capacity should only be limited by mechanical deflections which permit mechanical friction due to poor seal pressure distribution, or surface erosion caused by high velocity flow and cavitation in seal areas. High speed and heat load may eventually limit the capacity of slipper shoe bearings, however additional cooling oil may be easily applied to race bearing 72.

The mechanical strength of this pump is exceptionally high since the steel pistons, cylinder, and rotor exposed to cyclic pressure loading have unit stresses only about four times the hydraulic pressure, so they would appear to withstand peak fluid pressures as high as 10,000 pounds per square inch with no distress. The cylinder thrust loads cancel directly through the race flange 42, through support member 50 to the pintle 12. The race 35 is of relatively heavy section to keep deflections from upsetting the geometry of the slipper shoe 71 on the race bearing 72. The housing 10 carries only nominal control pressure and support loads.

The balancing and cancelling of load is carried even further in the combination pump and motor of FIG. 5. The structure of FIG. 5 is identical to that above described in connection with FIGS. 1-4 with the exception that the pintle 12' is double in nature with passages 82 and 85 eliminated and with passages 80' and 81' extending completely through the pintle from the pump to the motor. In the device of FIG. 5, the pressure load is balanced from the pump to the motor in the one piece pintle valve. There is minimum exposure of high pressure lines and maximum interchangeability of parts between units.

In hydrostatic transmission applications, the motor displacement should be as large as possible to get adequate torque at a reasonable pressure, however at maximum speed the maximum flow rate may cause excessive flow losses with reasonable size valve and channel areas. Larger pintle valve area would increase mechanical friction and leakage losses with some sacrifice in space for cylinder capacity. For high speed operation this design uses a fully variable pump and motor, so the flow rate may be adjusted with speed and power to obtain optimum efficiency over a wide range. The simplicity of moving parts (rotor assembly plus seven cylinders) and the relatively light polar moment mass of the rotor assembly compared to its displacement torque capacity make this a very agile unit well adapted to shock loads and full torque reversals encountered in power applications. Displacement control may be servo pressure or mechanical follower servo types readily adapted to outside linkage.

From the above description, it will be evident that the present invention provides a power transfer device which permits the locating of cylinders at a maximum radius next to the outer race instead of in a rotor next to the pintle whereby the cylinders can be of much larger diameter and greater displacement. It will also be evident that the present invention provides a power transfer device so configured that the pistons and cylinders thereof can be thoroughly lubricated and flushed on all working surfaces thereof for reducing wear and removing heat.

It will further be evident that the present invention provides a power transfer device incorporating means for keeping all seal functions flooded with oil. From the above description, it will also be appreciated that the preferred configuration of the present invention exposes only the cylinder rotor assembly and pintle to high pressure making easier and less expensive the design of the pump for high pressure.

While the invention has been illustrated and described in detail in the drawings and foregoing description, the same is to be considered as illustrative and not restrictive in character, it being understood that only the preferred embodiment has been shown and described and that all changes and modifications that come within the spirit of the invention and the scope of the claims are also desired to be protected.

The invention claimed is:

1. A hydraulic pump and motor comprising a housing, a cylindrical race mounted within said housing, a spindle fixed to said housing, a rotor rotatably received on said spindle for rotation within said race about an axis located eccentrically relative to the axis of said cylindrical race, said rotor having a plurality of radial spokes with a radial passage through each of said spokes each of said spokes having a rounded ball shape at its distal end, said spindle having a pair of passages therethrough one of which opens on one side of said spindle and the other of which opens on the other side of said spindle and over both of which said rotor rotates, and cup-like elements each received over a respective spoke and having an outer surface which rides against the inside of said race whereby rotation of said rotor occurs simultaneously with the passage of fluid from one of said passages through said rotor into the other of said passages, said cup-like elements being free to expand away from their respective spokes when friction between said race and cup-like elements causes said elements to be heated.

2. A hydraulic pump and motor comprising a housing, a cylindrical outer race received within said housing and eccentrically positioned relative thereto, a pintle valve fixed to said housing and having a pair of passages, a rotor having a plurality of radial spokes with a radial passage through each of said spokes, each of said spokes having a part-spherical outer surface at the distal end of each spoke and coaxial with said spoke, said rotor having a cylindrical opening through the center thereof which is received upon said pintle valve and into which said radial passages open, said pintle valve passages opening on opposite sides of said pintle valve outwardly toward said radial passages, cup-like cylinders slidably received on each of said spokes for reciprocation on said rotor spokes as said rotor rotates eccentrically relative to said race, said cup-like elements being free to expand away from their respective spokes when friction between said race and cup-like elements causes said elements to be heated, and a ring received within said housing and race, each of said cylinders having an inwardly opening groove which receives said ring, said ring acting to maintain said cylinders closely adjacent to said race as said cylinders rotate with said rotor.

3. A hydraulic pump and motor comprising a housing, a cylindrical outer race received within said housing and eccentrically positioned relative thereto, a pintle valve fixed to said housing, a rotor having a plurality of radial spokes with a radial passage through each of said spokes, said rotor having a cylindrical opening through the center thereof which is received upon said pintle valve, said pintle valve having a pair of passages for conducting fluid to and from said rotor, said pintle valve passages opening on opposite sides of said pintle valve outwardly toward said rotor radial passages, annular piston each fixedly received on the distal end of a respective spoke and each having a part-spherical outer surface coaxial with its respective spoke, and cup-like cylinders slidably received on each of said spokes and about said pistons for reciprocation on said rotor spokes as said rotor rotates eccentrically relative to said race, said cup-like elements being free to expand away from their respective spokes when friction between said race and cup-like elements causes said elements to be heated.

4. The hydraulic pump and motor of claim 3 additionally comprising a ring received within said housing and race, each of said cylinders having an inwardly opening groove which receives said ring, said ring acting to maintain said cylinders closely adjacent to said race as said cylinders rotate with said rotor, said ring having a radially extending surface which engages the external surface of

said cylinders maintaining the cylinders accurately positioned in an upright non-canted position.

5. The hydraulic pump and motor of claim 3 in which each of said cup-like cylinders and spokes are separate and isolated from the others of said cylinders and spokes whereby said cylinders and spokes can be separately cooled by the fluid passing through said device and each cylinder is free from pressure and thermal distortion of adjacent cylinders.

6. A power transfer device comprising a housing, a cylindrical outer race received within said housing and eccentrically positioned relative thereto, a pintle valve fixed to said housing and having a pair of passages, a rotor having a plurality of radial spokes with a radial passage through each of said spokes, each of said spokes having a part-spherical outer surface at the distal end of each spoke and coaxial with said spoke, said rotor having a cylindrical opening through the center thereof which is received upon said pintle valve and into which said radial passages open, said pintle valve passages opening on opposite sides of said pintle valve outwardly toward said radial passages, cup-like cylinders slidably received on each of said spokes for reciprocation on said rotor spokes as said rotor rotates eccentrically relative to said race, said cylinders having a slipper shoe bearing integral therewith, said slipper shoe bearing having an outer peripheral surface curved to the same radius as the inside of said race, said surface being normal to the axis of the cylindrical shape of said cup-like cylinders, said shoe bearing surfaces each having a pocket therein, each of said cylinders having a restricted orifice leading from said pocket to the inside of the respective cylinder whereby oil pressure may be supplied to the pockets through the orifices from the inside of the respective cylinders, and a plurality of tubes each formed integrally with a respective one of said cylinders on the inside thereof, each of said tubes projecting inwardly and forming an extension of a respective orifice and preventing the entry of heavy dirt particles into the respective bearing pockets from centrifuging of dirt into the cylinders.

7. The hydraulic pump and motor of claim 1 wherein each cup shaped element has an axis and in which said cup-like elements are oscillated about said ball shape on said rotor spokes in such a manner that a hydraulic thrust is produced in line with each element axis and imparts unidirectional thrust with no unbalanced hydraulic side load between each element wall and spoke; the aimed direction of the hydraulic pressure on the spokes and hydraulic pressure load at the pintle valve producing a hydrostatic torque couple which is balanced with no mechanical loads on the rotor.

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