

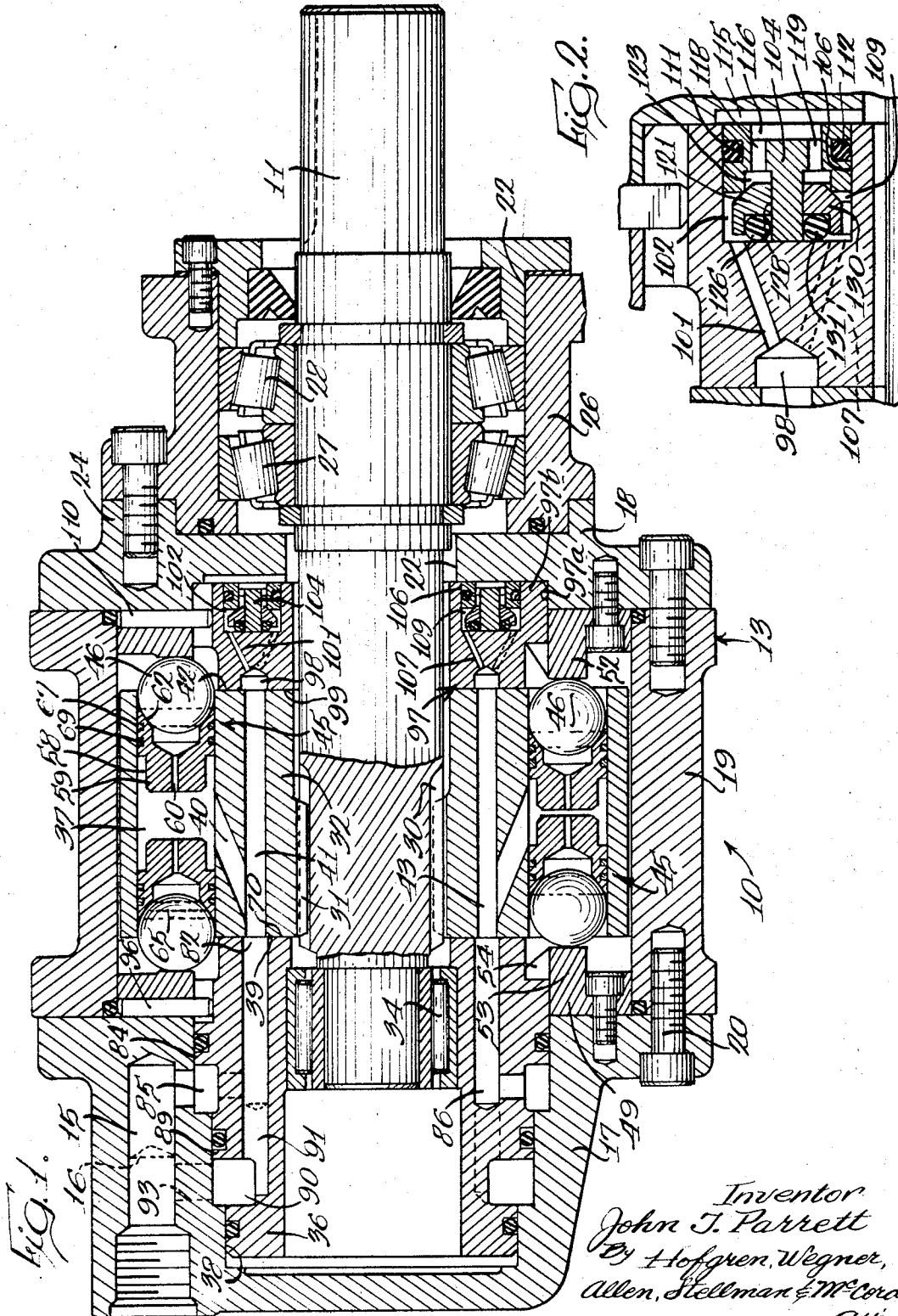
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HYDRAULIC PUMP OR MOTOR

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HYDRAULIC PUMP OR MOTOR

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ABSTRACT OF THE DISCLOSURE

A hydraulic energy translating device of the multiple piston type wherein the pistons have a plurality of power strokes during each rotation of the associated cylinder block through the provision of a plurality of inlet and outlet ports and a relatively stationary valve member engaging one end of the cylinder block, there being provided a hydrostatic thrust bearing engaging the other end of the cylinder block for balancing the axial forces on the block and serving, through the provision of check valves in the thrust bearing, the additional function of venting excess fluid in the hydraulic-unit housing to the low pressure ports.

Background of the invention

This invention relates generally to hydraulic fluid energy translating devices and more particularly, to a hydraulic pump or motor device.

One type of hydraulic fluid energy translating device is an axial piston design in which the pistons reciprocate in cylinders disposed parallel to the axis of rotation of the cylinder block. In my prior, copending application, Ser. No. 598,340, filed Dec. 1, 1966, I have disclosed a thrust plate assembly for balancing the hydraulic forces acting axially on the cylinder block of a device of this type.

Axial piston pumps and motors conventionally consist of a rotatable cylinder block drivingly connected with a shaft, with the block having a plurality of axially disposed cylinders therein. The cylinders communicate through cylinder ports with one end face of the block which slidably engages a relatively stationary valve plate having inlet and outlet ports therein. In units which have a plurality of power cycles in each cylinder per revolution of the cylinder block, a plurality of inlet ports and a plurality of outlet ports may be formed in the valve plate to accommodate the plural cycle timing. Pistons are slidably disposed in the cylinders and are reciprocated in the cylinders by a cam arrangement. As the cylinder block rotates relative to the valve plate, the pistons reciprocate, and the cylinder ports in the block serially communicate with the inlet and outlet ports alternatively arrayed in the valve plate. If the device is operating as a pump, the pistons withdraw from the cylinders as the cylinder ports communicate with the inlet ports in the valve plate, which are low pressure ports during pumping, and move into the cylinders as the cylinder ports pass over the outlet ports, which are high pressure ports during pumping. During motoring the reverse operation occurs with the pistons withdrawing from the cylinders as the cylinder ports pass over the inlet ports, which are high pressure ports during motoring, and move inwardly as the cylinder ports pass over the outlet ports, which are then low pressure ports.

In hydraulic units of this general description it is neces-

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sary to maintain a running seal between the cylinder block and the valve plate as the efficiency of these units is largely dependent upon the amount of fluid leakage between the block and the valve plate. Therefore, means are conventionally provided for urging the cylinder block in some manner against the plate. In units which have only a single piston in each cylinder, the hydraulic force acting on the bottom of this cylinder tends to urge the block into engagement with the valve plate. Moreover, suitable spring arrangements may be provided for biasing the cylinder block into engagement with the valve plate. In some units it is desirable that the axial hydraulic forces on the block be balanced, and toward this end there have been provided units with two opposed pistons in each cylinder thereby balancing the hydraulic forces within the cylinder acting on the cylinder block. Such a unit is described in my above referred to copending application, Serial No. 598,340. In such a construction it is desirable to maintain the sealing engagement between the block and the valve plate, and toward this end separate means are provided for urging the valve plate into engagement with the block.

There are, of course, other devices in the prior art designed for maintaining this sealing, sliding engagement between the cylinder block and the valve plate. Many of these produce a net axial force on the cylinder block which results in an unbalanced rotation of the cylinder block and the degradation of the operating performance of the unit. To eliminate this problem (as shown in my above referred to copending application), a hydrostatic thrust bearing has been provided which resists movement of the cylinder block away from the valve plate thus effectively balancing the axial forces acting on the cylinder block.

Another problem in hydraulic units of this general type is that the leakage fluid finds its way to the hydraulic unit casing or housing, and some means must be provided for removing this leakage fluid therefrom.

Summary of the invention

In accordance with the present invention, the problem of leakage fluid accumulation in the housing is eliminated through the provision of check valves which permit fluid flow from the interior of the housing to the low pressure side of the main hydraulic circuit associated with the hydraulic pump or motor. Thus, leakage fluid is returned directly to the system from the interior of the housing. Moreover, these check valves have been advantageously designed to be associated with the hydrostatic thrust bearing in a manner simplifying the construction of the thrust bearing as well as the check valve assemblies.

The check valve assemblies each include an annular movable valve member seated within one of two annular galleries interconnecting alternate ones of fluid bearing pockets in the hydrostatic thrust bearing. The annular galleries are required for the thrust bearing so that alternate ones of the hydrostatic bearing pockets are all at high pressure or all at low pressure. A similar check valve assembly is provided for the other annular gallery. Since alternate ones of these pockets continuously communicate with the low pressure side of the circuit, one of the galleries is at low pressure and the check valve associated with this gallery will open when the fluid pressure in the housing exceeds that in the low pressure side of the circuit permitting leakage fluid from within the pump housing

to pass through the check valve and the low pressure gallery to the low pressure side of the hydraulic circuit.

Brief description of the drawing

FIG. 1 is a longitudinal section of a hydraulic energy translating device according to the present invention.

FIG. 2 is a fragmentary section of a portion of the hydrostatic thrust bearing shown in FIG. 1.

Description of the preferred embodiment

Referring now to the drawing and particularly FIGS. 1 and 2 therein, a hydraulic motor 10 is shown having an output shaft 11 connected to drive a suitable load such as the turntable of a mobile crane. Shaft 11 is supported within a housing assembly 13 having inlet and outlet passages 15 and 16 extending therethrough. Either of the passages 15 and 16 may be connected to a suitable source of hydraulic fluid under pressure, such as a pump, and the other connected to return fluid to the source inlet or to a suitable tank (not shown). With hydraulic fluid being supplied to one of the passages the output shaft 11 will be driven in rotation in one direction and if fluid under pressure is delivered to the other passage the shaft 11 will be driven in the reverse direction.

The housing 13 consists of a central annular member 19 with end plates 17 and 18 fixed thereto by suitable threaded fasteners 20. The end plate 18 has a central opening 22 and an annular flange 24 for receiving a bearing support 26. Support 26 receives tapered roller bearings 27 and 28 which support the right end of shaft 11.

Formed on the shaft 11 within housing member 19 are splines 30 which interengage mating splines 31 formed internally on a rotatable cylinder block 32. The inner end of shaft 11 is supported within a needle bearing 34 seated within an annular valve member 36 in turn seated in a stepped bore 38 in end plate 17.

Cylinder block 32 is generally annular in configuration and is supported on the spline portion 31 of shaft 11.

Formed axially in the cylinder block 32 in annular array about the axis of rotation thereof and shaft 11 are a plurality of cylinders 37 extending completely therethrough. Each of the cylinders 37 communicates with a port face 39 at one end of the cylinders 37 communicates with a port face 39 at one end of the cylinder block through angled passages 40 and intersecting axial passages 41 which open to face 39. Passages 41 also open to opposite face 42 of the cylinder block 32 for a purpose described in more detail hereinbelow.

It should be understood that there is one passage 40 and one passage 41 for each of the cylinders 37. The intersection of the passages 41 with the port face 39 define a plurality of cylinder ports 43 which are annularly arrayed on the port face 39.

Two piston assemblies 45 are slidably received in each of the cylinders 37, and each includes a spherical ball piston 46. A first annular cam 49 is mounted on the inner end of end cap 17 for reciprocating the pistons 46 in the left ends of the cylinders 37, thus permitting the high pressure fluid in the cylinders 37 to be translated into rotary motion of the cylinder block 32 and output shaft 11. Similarly, an identical cam 52 is mounted on an inward face of end cap or plate 18 for reciprocating the pistons 46 adjacent thereto. The cams 49 and 52 are slightly out of phase with respect to one another to achieve a more uniform flow of hydraulic fluid.

Each of the cams has an upstanding annular cam track 53 defining a plurality of lobes 54 in each of the cam members 49 and 52 so that the pistons 46 reciprocate through a plurality of power strokes during each revolution of the cylinder block 32. In one exemplary construction of the device shown in FIG. 1 twenty cylinders 37 and twelve cam lobes 54 were provided so that for each revolution of the cylinder block 32 each piston 46 has twelve power cycles.

Referring now in more detail to the piston assemblies

45, cylindrical sealing rings 58 are provided adjacent and engaging each piston 46.

The seal rings 58 are seen to be generally cylindrical and slidably received in the cylinders 37 and each have a reduced portion extending inwardly defining a projection 59.

The projections 59 are sufficiently long with respect to the length of the cylinders 37 so that they serve as stops engaging one another when the device is freewheeling and not under pressure preventing the cylinder engaging portions of the sealing rings from covering the passages 40 and preventing, if permitted, the flow of hydraulic fluid under pressure to the cylinders. However, as may be seen by the relationship between the sealing rings in the lower cylinders in FIG. 1, the projections 59 are not so long that they will engage during any portion of the stroke of the piston assemblies when the cylinders are under pressure.

The sealing rings 58 each have an orifice 60 therethrough so that one side of the spherical pistons freely communicates with the interior of the cylinders 37. An outwardly tapered conical portion 62 is formed within the sealing rings having line contact as shown at 65 with the spherical piston thereby sealing any fluid tending to discharge from the cylinders through the orifice 60.

Carried in annular recesses 67 in the periphery of the sealing rings 58 are piston rings 69 which serve to prevent the leakage of hydraulic fluid from within the cylinders 37 around the periphery of the sealing rings 58. The diameter of the spherical pistons 46 is only slightly smaller than the cylinders 37. The sealing rings 58 have a diameter smaller than the diameter of the spherical pistons so that the former do not directly engage the cylinders 37.

When the cylinder block 32 rotates the cam lobes 54 exert forces on the pistons 46 in a tangential direction (in a plane tangent to the cylinder defined by the revolution of the axes of the cylinders 37). This cam force is transmitted from the spherical pistons directly to the cylinder block so that none of the driving load passes through the sealing rings 58. The cam force causes the spherical pistons to have some clearance with the cylinder 37 on one side thereof and this would in the absence of the sealing ring 58, cause leakage around the pistons detracting from the efficiency of the device. This misalignment of pistons 46 causes the sealing rings to be shifted slightly in the cylinders 37 assuring that line contact 65 is maintained and fluid leakage is prevented between the conical portion 62 of the seal rings and the spherical pistons. The piston rings 69, however, remain centered in the cylinders 37 and by engagement with the sides of the recesses 67 prevent leakage around the rings 58. The axial location of line contact 65 is selected with respect to the spherical pistons 46, to expose a sufficient area of each piston to fluid pressure in the cylinder 37 so that the hydraulic force on the piston, as distinct from the mechanical force transmitted to the piston by the line of contact 65, is large enough to assure proper rolling contact of the spherical pistons on the cam tracks 53.

The proper sizing of the orifice 60 assures continuous sealing engagement between ring 58 and piston 46 and between piston 46 and cam track 53 when the device is operating as a motor. As described above in the present device no springs are necessary to prevent chattering and proper sealing of the piston assemblies when the device is operating as a motor. A further function of the orifice 60 is to assure return movement of the sealing rings 58 toward the pistons 46 when the cylinders 37 are pressurized after the device has been freewheeling. During freewheeling when the drive shaft 11 is rotated without the delivery of fluid under pressure to the cylinders the seal rings 58 will be driven together toward the center of the cylinders and in this position they will remain until hydraulic fluid is again delivered to passages 40. When the delivery of hydraulic fluid under pressure is resumed it is of course desirable that the seal rings 58 be immediately returned to sealing engage-

ment with the pistons 46 and that the pistons 46 return to rolling engagement with the cam tracks 53.

To achieve these objectives the orifice 60 is sufficiently small so that a net hydraulic force is applied to the seal ring 58 when separated from piston 46 in a direction toward the spherical piston 46, with this net hydraulic force being at least as great as the frictional force between the seal ring and the cylinder walls resisting such movement of the seal ring. A complete development for the proper sizing of orifice 60 is contained in my copending application Ser. No. 674,203, filed Oct. 10, 1967, and reference should be made thereto for this subject matter.

The annular stepped valve member 36 is provided for conveying fluid between passages 15, 16 and the cylinder ports 43. The port face 39 on the cylinder block slidably engages valve surface 70 on valve member 36 during rotation of the cylinder block. Formed in annular array in the valve surface 70 about the axis of cylinder block 32 are a plurality of valve ports 82 corresponding in number to twice the number of lobes 54, so that if each cam has twelve lobes there would be twenty-four ports 82. Ports 82 are arranged so that alternate ones communicate with passage 15 and the remaining ones communicate with passage 16 so that there will be alternate high and low pressure ports 82. Toward this end valve member 36 has a first annular piston 84 defining in the stepped bore 38 an annular chamber 85 continuously communicating with alternate ones of ports 82 through axial annularly arrayed passages 86. Annular chamber 85 continuously communicates with port 15. A second smaller piston 89 is formed on the valve member 36 and defines a second annular chamber 90 continuously communicating with the remaining ports 82 through axial annularly arrayed passages 91. Chamber 90 continuously communicates with port 16 through radial passage 93.

The chamber 85 and 90 form fluid pressure chambers which serve as high or low pressure chambers depending upon the direction of flow of hydraulic fluid through the device, i.e. depending upon which of the passages 15 or 16 receives high pressure fluid to the motor 10. It should be understood that valve member 36, while prevented from rotation by radial pin 96 is permitted limited axial movement within the stepped bore 38. Fluid pressure acting in one of the chambers 85 or 90 urges the valve member 36 to the right so that proper sealing engagement is effected between valve surface 70 and the cylinder block port face 39.

For resisting any tendency of the cylinder block 32 to move away from the valve member 36, a hydrostatic thrust bearing 97 is provided in accordance with the present invention. The thrust bearing is annular in configuration and mounted in a counterbore 97a within housing plate 18 and held against rotation by an axial pin 110.

A plurality of pockets 98 are formed in flat bearing face 99 for the purpose of providing a hydrostatic bearing film between cylinder block surface 42 and bearing surface 99. Alternate ones of the pockets 98 are pressurized by hydraulic fluid in the high pressure ones of the axial porting passages 41 in the cylinder block.

Toward this end pockets 98 are disposed in annular array in the thrust bearing surface 99 and are equal in number and aligned with the inlet and outlet ports 82 in the valve member 36. By pressurizing the pockets 98 that are aligned with the high pressure ones of the ports 82, the hydrostatic bearing effect provided by bearing 97 will be equal and opposite to that provided between the valve surface 70 and the other end of the cylinder block 32. To contribute to this result, the bearing areas between the valve surface 70 and the adjacent end of the cylinder block, and between the bearing surface 99 and cylinder block surface 42 are substantially equal.

For the purpose of providing continuous communica-

tion between alternate ones of the pockets 98 so that the hydrostatic bearing effect is the same as that effected by ports 82, a gallery arrangement is provided in the thrust bearing 97 connecting alternate ports. Toward this end an annular recess 106 is provided in the rear surface of the thrust bearing member 97b. Seated within this recess is an annular T-shaped separator defining annular passages 102 and 109. Seal rings 111 and 112 prevent the escape of fluid between the inner and outer sides of the recess 106 and the T-shaped member 104. Alternate ones of the pockets 98 communicate with the common annular passage 102 through discrete passages 101, thus providing continuous communication between alternate pockets 98. These, for example, may be the pockets 98 aligned with the ports 82 in continuous communication with main port 15.

The remaining alternate pockets 98 communicate with the other annular passage 109 through discrete passages 107 so that these pockets continuously communicate with each other but not with the pockets 98 in communication with passage 102. The former alternate pockets may, for example, be aligned with the ports 82 in continuous communication with the main port 16. Thus, one set of alternate pockets 98 receive high pressure fluid when port 15 is pressurized, and the other set of alternate pockets 98 receive high pressure fluid when port 16 is pressurized.

Since the thrust member 97b is held against rotation, the pockets 98 serially communicate with the passages 41 as the cylinder block 42 rotates. Fluid under pressure from passages 41 in balancing ports of pockets 98 passes between the surfaces 42 and 99 providing a fluid film defining a hydrostatic thrust bearing. It is apparent that each time one (or more) of the pockets 98 communicates with a cylinder under pressure (through the associated passage 41), one of the annular chambers 102 or 109 communicating therewith provides pressurization of all the associated alternate ports communicating with that chamber to effectively balance the opposite hydrostatic effect resulting from simultaneous pressurization of the aligned alternate flow ports 82. Of course, at this time the pockets 98 communicating with the other chamber would all be at low pressure. The gallery arrangement, including chambers 102 and 109, is necessary since all of the alternate pockets are not simultaneously in communication with the high pressure ones of the passages 41 as the number of ports and pockets differs from the number of passages 41.

Means are provided for the purpose of venting or draining the interior of the pump housing 13 to the low pressure side of the hydraulic circuit, or more specifically to the low pressure ones of the ports 82 or to the low pressure cylinders 37. This returns the leakage fluid directly to the hydraulic circuit without the necessity of any conduits or valves outside the hydraulic unit itself.

Toward this end selective communication is provided between the interior of the pump casing and both of the annular chambers 102 and 109 through radial slots 115 in the end cap 18, an annular recess 116 in the T-shaped member 104, and passages 118 and 119 extending through the T-shaped member into the chambers 102 and 109.

For the purpose of selectively blocking communication between the chamber 102 and the interior of the pump housing when that chamber is at high pressure, an annular check valve member 121 is provided biased against an annular seat defining shoulder 123 on the T-shaped member by an annular "O" sealing ring 126. The O-ring 126 serves not only to bias the check valve 121 to its closed position, but also since it engages both the bottom of annular recess 106, and the side 128 of the T-shaped member 104, as well as the two adjacent surfaces of the valve member 121, it functions as a sealing member preventing the escape of fluid from chamber 102 to either chamber 109 or between the valve member 121 and surface 128 into the passages 118.

A similar check valve member 130 biased by a resilient O sealing ring 131 is provided in the same manner for chamber 109, although its shape is reversed.

When the ports or pockets 98 associated with chamber 102 are at high pressure, i.e. communicating with the high pressure side of the hydraulic circuit, valve member 121 is biased to its closed position shown by the force of hydraulic fluid acting on the valve member as well as the force of resilient O-ring 126. Under these conditions, the pockets 98 associated with annular chamber 109 will be at low pressure, as will be the chamber itself, but the valve member 130 will also remain closed until the pressure of leakage fluid within the housing 13 exceeds that in the low pressure pockets 98. If the fluid pressure in the housing reaches this value, (which it will as the housing becomes filled) valve member 130 will open, permitting leakage fluid to pass from the housing through slot 119, chamber 109, passages 107, alternate pockets 98, into passages 41 communicating with the low pressure ones of the pockets 98. If the device is operating as a pump, this leakage fluid will flow directly into the cylinders 37 since the pistons 45 will be proceeding on their intake strokes. If the device is operating as a motor, the leakage fluid reentering the system through the check valves will pass with the other hydraulic fluid being expelled from the cylinders 37 into the low pressure ones of the ports 82 and out the low pressure one of the ports 15, 16.

If the hydraulic device is reversed so that the other set of alternate ports 82 becomes the high pressure set, check valve 130 will remain closed under the influence of high pressure in the associated pockets 98, while check valve 121 will serve to vent excessive fluid pressure within the interior of housing 13 in the same manner as that described above.

I claim:

1. A hydraulic energy translating device, comprising: housing means, said housing acting as a reservoir for leakage fluid, valve means in said housing means having inlet and outlet ports therein, a cylinder block rotatably mounted in said housing means forming a plurality of cylinders therein serially communicable with said ports, pistons reciprocally mounted in said cylinder block, cam means for reciprocating said pistons as the cylinder block rotates relative to said valve means, means urging the cylinder block into engagement with said valve means, hydrostatic bearing means engaging the side of the cylinder block opposite said valve means, passage means selectively communicating said bearing means with said inlet and outlet port means, and valve means in said bearing means permitting fluid flow from within the reservoir to the passage means in communication with the low pressure one of said ports.

2. A hydraulic energy translating device as defined in claim 1 wherein said valve means includes a first check valve communicating the interior of the housing means with the passage means when the fluid pressure in said housing means exceeds the fluid pressure in one of said port means, and a second check valve communicating the interior of the housing means with the passage means when the fluid pressure in said housing means exceeds the fluid pressure in the other of said port means.

3. A hydraulic energy translating device as defined in claim 1, wherein said hydrostatic bearing includes a substantially flat bearing plate having a flat bearing surface engaging said other side of the cylinder block, a plurality of fluid pressure pockets in said bearing surface positioned to serially communicate with said cylinders corresponding in number to and aligned with said inlet and outlet port means, said pockets exceeding two in number, first passage means in said bearing plate interconnecting first alternate ones of said pockets so that said first alternate pockets are all either at high pressure or at low pressure, second passage means in said bearing plate interconnecting second alternate ones of said pockets so that said second alternate pockets are all either at high pressure or at low

pressure, said valve means including a first check valve in said first passage means and a second check valve in said second passage means.

4. A hydraulic energy translating device as defined in claim 3, wherein said first and second passage means each include an annular passage in said bearing plate centered about the axis of said cylinder block, said first and second check valves each including an annular valve member in the associated annular passage.

5. A hydraulic energy translating device, comprising: housing means, valve means in said housing means having inlet and outlet ports therein, a cylinder block rotatably mounted in said housing means forming a plurality of cylinders therein serially communicable with said ports, pistons reciprocally mounted in said cylinder block, cam means for reciprocating said pistons as the cylinder block rotates relative to said valve means, means urging the cylinder block into engagement with said valve means, hydrostatic bearing means engaging the side of the cylinder block opposite said valve means, passage means selectively communicating said bearing means with said inlet and outlet port means, valve means permitting fluid flow from within the housing means to the passage means in communication with the low pressure one of said ports, said valve means including a first check valve communicating the interior of the housing means with the passage means when the fluid pressure in said housing means exceeds the fluid pressure in one of said port means, a second check valve communicating the interior of the housing means with the passage means when the fluid pressure in said housing means exceeds the fluid pressure in the other of said port means, said hydrostatic bearing including a substantially flat bearing plate having a flat bearing surface engaging said other side of the cylinder block, a plurality of fluid pressure pockets in said bearing surface positioned to serially communicate with said cylinders corresponding in number to and aligned with said inlet and outlet port means, said pockets exceeding two in number, first passage means in said bearing plate interconnecting first alternate ones of said pockets so that said first alternate ones of said pockets are all either at high pressure or at low pressure, second passage means in said bearing plate interconnecting second alternate ones of said pockets so that said second alternate pockets are all either at high pressure or at low pressure, said valve means including a first check valve in said first passage means and a second check valve in said second passage means, said first and second passage means each including an annular passage in said bearing plate centered about the axis of said cylinder block, said first and second check valves each including an annular valve member in the associated annular passage, and annular sealing rings associated with each annular valve member for urging the valve members to a closed position in opposition to the fluid pressure in said housing means.

6. A hydraulic fluid energy translating device, comprising: a housing, a valve plate in said housing having a plurality of inlet ports and a plurality of outlet ports alternately disposed in annular array, a cylinder block slidably engaging said valve plate and having cylinders therein differing in number from the inlet and outlet ports, said cylinders serially communicating with said ports as the cylinder block rotates relative to said valve plate, pistons slidable in said cylinders, cam means for reciprocating said piston, means urging the cylinder block into engagement with the valve plate, and a hydrostatic bearing at the end of the cylinder block opposite the valve plate for balancing the axial forces acting on the cylinder block including a bearing plate having a flat bearing surface slidably engaging the cylinder block, annularly arrayed pockets in said bearing surface corresponding in number with the inlet and outlet ports, an annular recess in said bearing plate in the side thereof opposite said pockets, a generally T-shaped member in said recess defining two annular passages therein, a plurality of first

passages connecting first alternate pockets with one of said annular passages, a plurality of second passages connecting second alternate pockets with the other of said annular passages, and check valve means in each of said annular recesses permitting venting of said housing each including passage means in said T-shaped member providing communication between the interior of said housing and the annular passages, an annular valve member in said annular passage means blocking the passage means in the T-shaped member when closed, and a resilient O-ring engaging said movable valve member and said bearing plate for urging the valve member closed and for sealing the T-shaped member.

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