

April 1, 1969

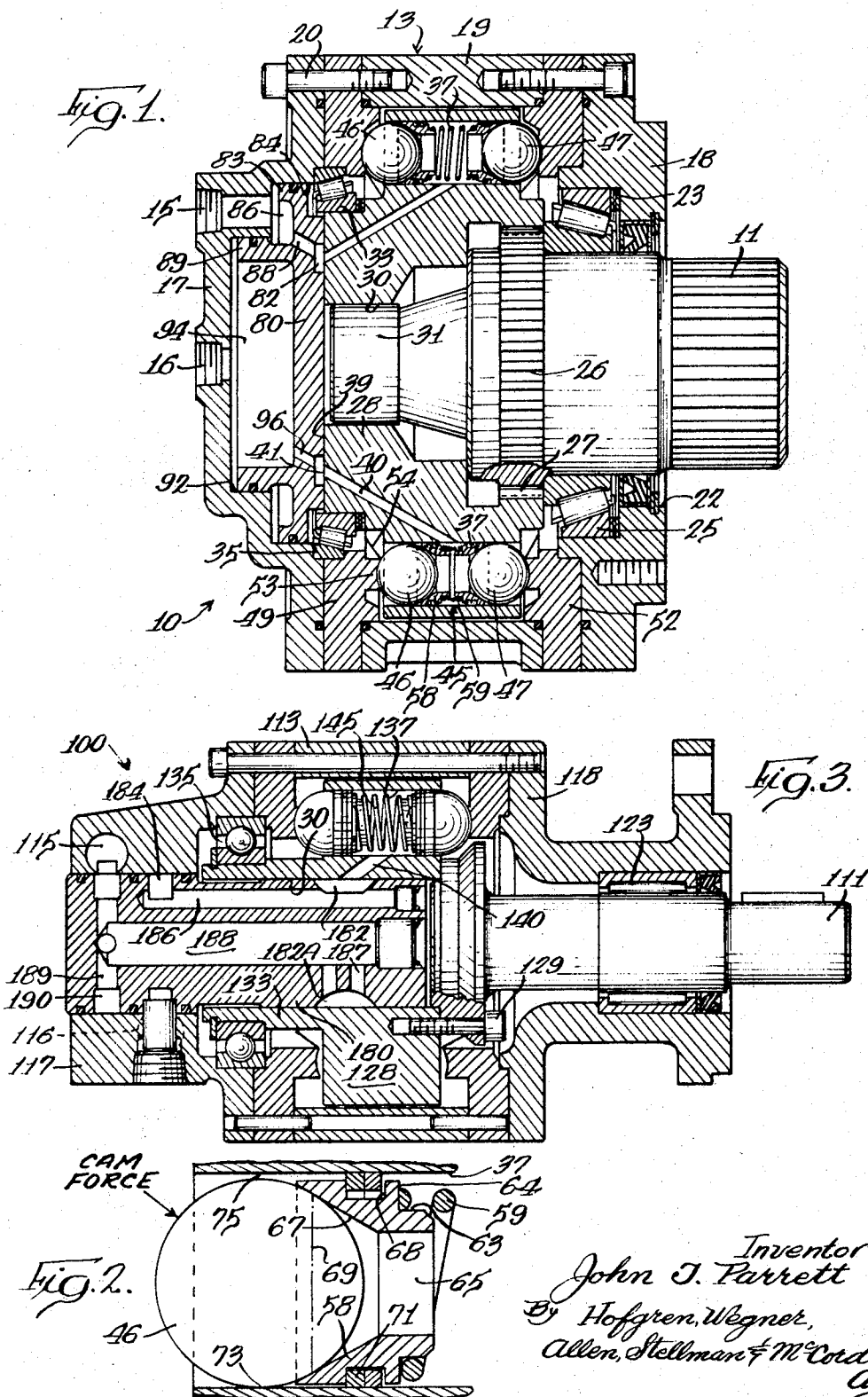
J. T. PARRETT

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HYDRAULIC PUMP OR MOTOR

Filed Dec. 1, 1966

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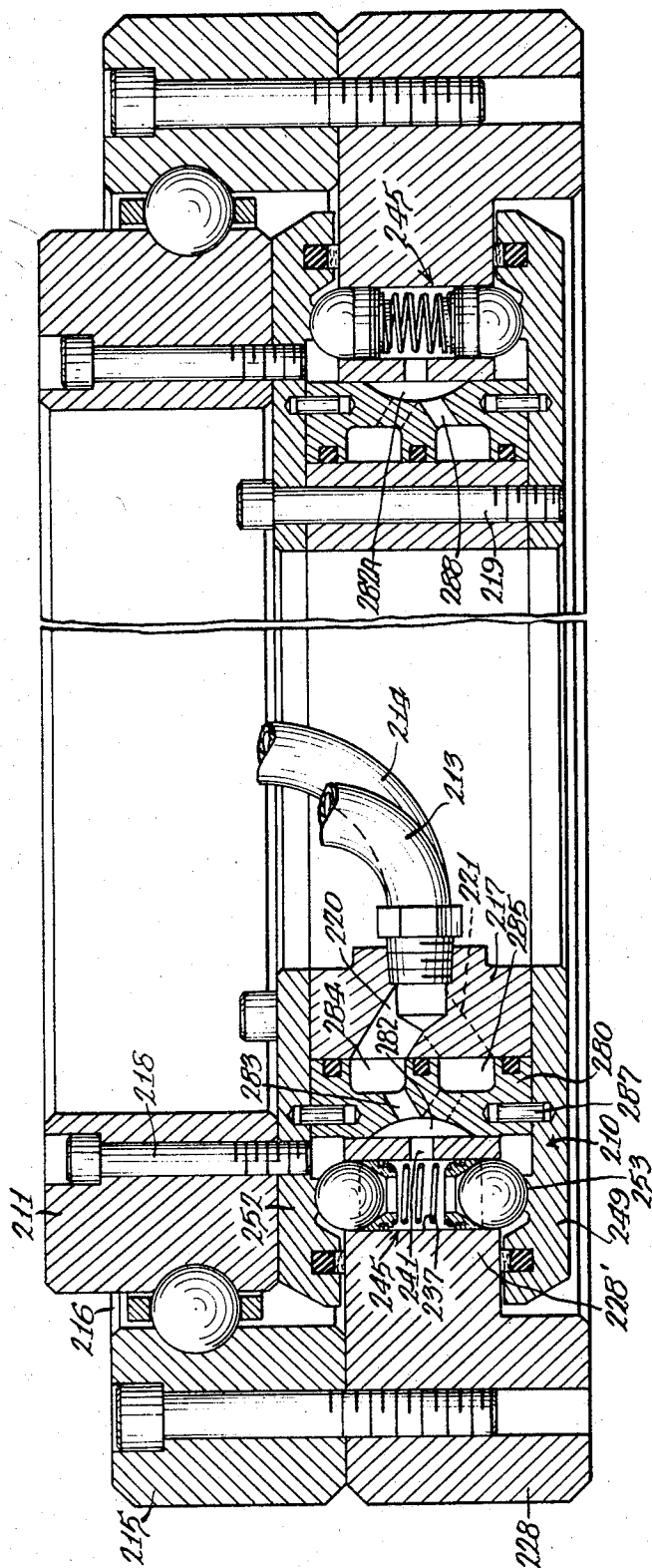
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Fig. A.



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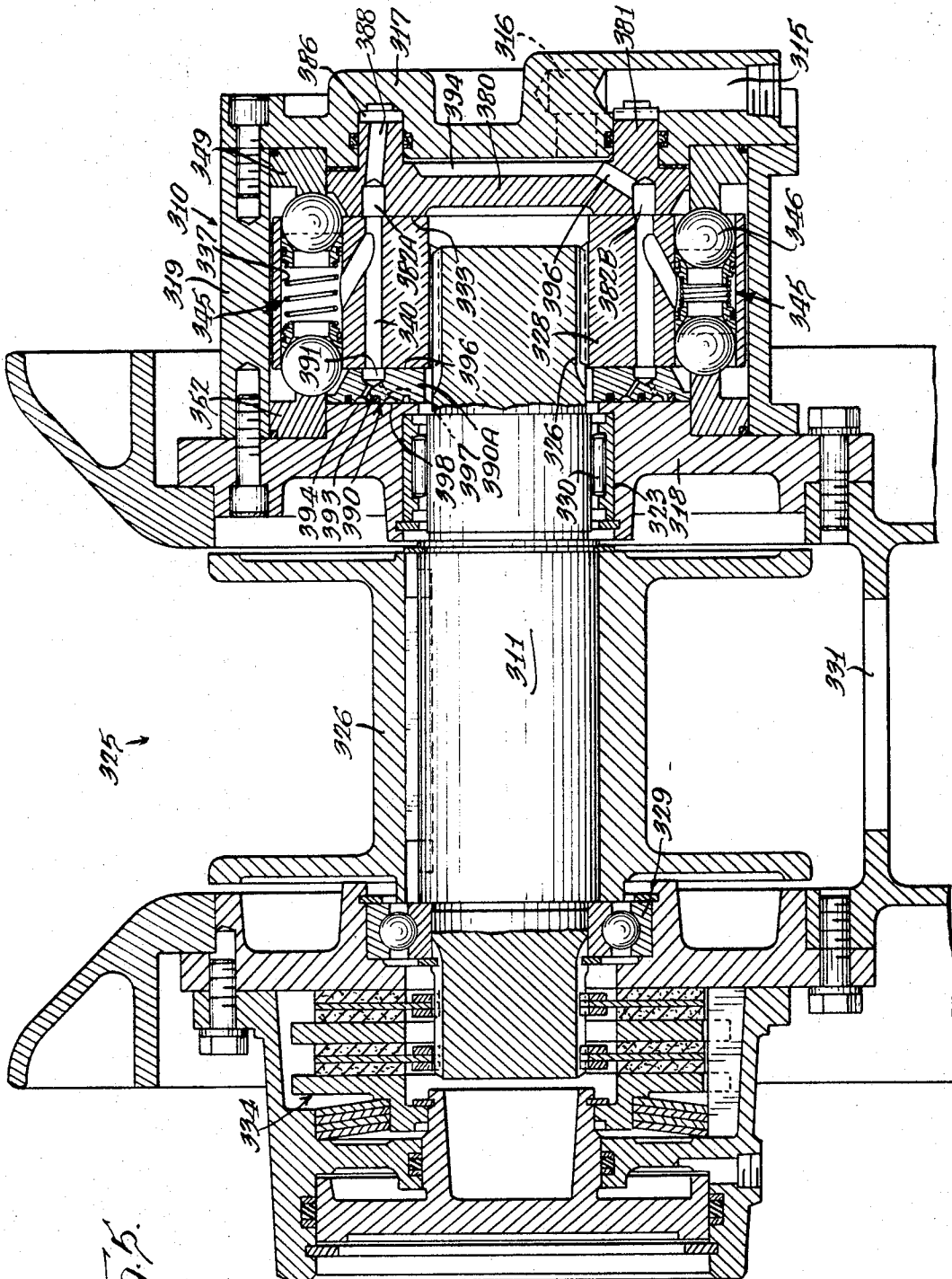
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3,435,774

HYDRAULIC PUMP OR MOTOR

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11 Claims

ABSTRACT OF THE DISCLOSURE

A multiple piston hydraulic unit in which the pistons take the form of ball members with annular sealing rings adjacent the ball members and surrounding piston rings sealingly engaging both the sealing rings and the cylinder walls with some misalignment being permitted between the sealing rings and the piston rings.

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 on the cylinder block. In one common design of this type, rigid pistons are employed which project from the cylinder block and engage an inclined cam member which serves to reciprocate the pistons. As the cylinder block rotates relative to the cam member, the pistons exert a radial load on the cylinder block tending to tilt the block and destroy the sealing engagement between the rotating cylinder block and a stationary valve member through which fluid flows to and from the block. Furthermore, the force of hydraulic fluid in the cylinders acting on the bottoms of the cylinders causes an axial load to be imposed on the cylinder block which also detracts from the performance of the device.

In accordance with the present invention, a multiple piston hydraulic unit device is provided in which all the axial and radial loads on the cylinder block are balanced. This is effected by the use of opposed spherical type pistons in each of the cylinders with separate cam members for reciprocating the pistons at one end of the cylinders and at the other end.

It is, therefore, a primary object of the present invention to provide a new and improved hydraulic fluid energy translating device.

Another object of the present invention is to provide a new and improved hydraulic fluid energy translating device in which all of the axial and radial loads on the cylinder block are balanced.

A further object of the present invention is to provide a new and improved hydraulic fluid energy translating device of the type generally described having spherical balls for pistons eliminating the radial loads on the cylinder block, and further including sealing rings around a portion of the piston for preventing the leakage of hydraulic fluid around the piston while still permitting a direct transmittal of force from the cam through the piston to the associated cylinder bore.

A still further object of the present invention is to provide a new and improved hydraulic fluid energy translating device of the type described above in which opposed spherical pistons are provided in each of the cylinders to increase the capacity of the device and eliminate axial loading of the cylinder block.

Still another object of the present invention is to provide a new and improved hydraulic energy translating device of the multiple piston type having a valve member for conveying fluid to and from the cylinders which is hydraulically urged toward the cylinder block to achieve

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a sealing relationship therewith regardless of the direction of rotation of the cylinder block.

Another object of the present invention is to provide a new and improved hydraulic fluid energy translating device of the type described above including a rotatable cylinder block mounted within a housing by peripheral bearing means surrounding the block and with a pintle type valving member mounted in the housing and cantilevered within an axial recess of the cylinder block.

A further object of the present invention is to provide a new and improved hydraulic fluid energy translating device of the type described generally above, mounted directly to rotate a turret in a movable crane or the like without the necessity of employing any gearing between the device and the rotatable turret.

Other features and advantages of the invention will be apparent from the following description of certain embodiments thereof taken in conjunction with the accompanying drawings. Of the drawings:

FIG. 1 is a longitudinal elevation of one embodiment of the present invention;

FIG. 2 is a fragmentary view of a portion of the piston assembly used in all of the embodiments illustrated;

FIG. 3 is a longitudinal elevation of another embodiment of the present hydraulic fluid energy translating device;

FIG. 4 is a cross sectional elevation of a mobile crane turret incorporating the hydraulic fluid energy translating device of the present invention; and

FIG. 5 is a sectional view of boom winch actuated by a motor generally similar in construction to the motor shown in FIG. 1, except that an axial thrust bearing is provided for the cylinder block.

Referring now to the drawings 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. It should be understood that while the illustrated embodiments shown in the drawings will be described as motors, they may operate as well as pumps when a suitable prime mover is connected to drive shaft 11. Shaft 11 is supported within a housing assembly 13 having inlet and outlet hydraulic fluid passages 15 and 16 extending therethrough. One of these passages 15, 16 is adapted to 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 15, 16 the output shaft 11 will be driven in rotation in one direction. If hydraulic fluid under pressure is delivered to the other passage the shaft 11 will be driven in the opposite direction of rotation.

The housing assembly 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 therethrough for receiving shaft 11 and a counterbore 23 for receiving a roller bearing 25 which supports the projecting end of the shaft 11.

Formed approximately centrally on the shaft 11 are splines 26 which interengage mating splines 27 formed internally on a rotatable cylinder block 28. Cylinder block 28 has a central opening 30 therethrough which receives a pilot 31 on the end of shaft 11 to support this end in the block.

Cylinder block 28 is generally annular in configuration and has a reduced axially extending portion 33 at one end thereof supported within a bearing 35 in turn mounted within housing 13. Bearing 35 peripherally supports one end of the cylinder block while the other end is supported on the splines 26 of shaft 11.

Formed axially in the cylinder block 28 in annular ar-

ray about the axis of rotation of the block 28 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 cylinder block through angled passages 40, it being understood that there is one passage 40 provided for each cylinder 37. The intersections of the passages 40 with the port face 39 define a plurality of cylinder ports 41 which are annularly arrayed in the port face 39.

Piston assemblies 45 are slidably received in each of the cylinders 37, and each include opposed spherical pistons 46 and 47. A first cam member 49 mounted between housing member 19 and end plate 17 is provided for reciprocating pistons 46 thus permitting the high pressure fluid in the cylinders 37 to be translated into rotary motion of the cylinder block 28 and output shaft 11. Similarly, an identical cam 52 is provided between housing member 19 and end cap 18 for reciprocating the pistons 47. 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 cam lobes 54. Lobes 54 are defined by generally circular arcs. There are a plurality of lobes 54 in each of the cam members 49 and 52 so that the pistons 46 and 47 reciprocate through a plurality of power strokes during each revolution of the cylinder block 28. In one exemplary construction of the device shown in FIG. 1 twenty cylinders 37 and twelve cam lobes 54, are provided so that for each revolution of the cylinder block 28 each piston 46 has twelve power cycles.

Referring now in more detail to the piston assemblies 45 which include spherical pistons 46 and 47, annular sealing rings 58 are provided adjacent each piston, both biased against the pistons by a compression spring 59. Spring 59 maintains a continuous engagement between the pistons and the cam track 53 during rotation of the cylinder block.

As shown more clearly in FIG. 2, seal ring 58 is seen to be a generally annular member slidably received in cylinder 37, having a reduced portion 63 defining a radial shoulder 64 serving as a seat for spring 59. The seal ring 58 has a bore 65 therethrough so that one side of the spherical piston freely communicates with the interior of the cylinder 37. Bore 65 has an outwardly tapered conical portion 67 which has line contact as shown at 69 with the spherical piston. Carried in an annular recess 68 in the periphery of the seal ring 58 are piston rings 71 which serve to prevent the leakage of hydraulic fluid within the cylinder 37 around the periphery of the seal ring 58. The diameter of the spherical pistons 46, 47 is only slightly smaller than the cylinders 37, and the seal rings 58 have a diameter smaller than the diameter of the spherical pistons so that they do not directly engage the cylinders 37.

When cylinder block 28 rotates, the cam lobes 54 exert forces on the pistons 46, 47 in a tangential direction (in a plane tangent to the cylinder defined by the revolution of the axes of the cylinders 37) as indicated in FIG. 2. This cam force is transmitted from the spherical pistons directly to the cylinder block at point 73 so that none of the driving load passes through the sealing ring 58. The cam force causes the spherical pistons to have some clearance with cylinder 37 opposite point 73 as shown at 75, and this would, in the absence of seal ring 58, cause a leakage around the pistons detracting from the efficiency of the device. This misalignment of the pistons 46, 47 causes the seal ring 58 to be shifted slightly in the cylinder 37 so that the line contact 69 is maintained, and fluid leakage is prevented between the conical portion 67 of the seal ring and the spherical piston. The piston rings 71, however, remain centered in the cylinder 37 and by engagement with the sides of recess 68 prevent leakage around the ring 58. The axial location of line contact

69 is selected, with respect to the spherical pistons 46, 47 so that a sufficient area of each piston is exposed to fluid pressure in cylinder 37 so that the hydraulic force on the piston approximately equals that of the cam force thereon. This assures proper rolling contact of the spherical pistons on the cam tracks 53.

A nonrotatable valve plate 80 is provided for conveying fluid between passages 15, 16 and the cylinder ports 41. The port face 39 on the cylinder block slidably engages valve plate 80 during rotation of the cylinder block. Formed in annular array in the valve plate 80 about the axis of cylinder block 28 are a plurality of valve ports 82 corresponding in number to twice the number of lobes 54, so that if each cam had twelve lobes there would be twenty-four ports 82. Ports 82 are arranged so that alternate ones communicate with passage 15 and remaining ones communicate with passage 16 so that there will be alternate high and low pressure ports 82. Toward this end the valve plate 80 has an annular flange 83 slidably received in a counterbore 84 in end cap 17, defining therein a chamber 86 continuously communicating with passage 15. Alternate ports 82 communicate with chamber 86 through passages 88.

Further, a smaller axially extending flange 89 on valve plate 80 defines a second piston slidably received in a complementary counterbore 92 in end plate 17. Flange 89 defines in the end plate 17 a second chamber 94 which continuously communicates with passage 16. The other alternate ports 82 continuously communicate with chamber 94 through passages 96.

The chambers 86 and 94 form fluid pressure chambers which serve depending upon the direction of flow of hydraulic fluid through the device, i.e., depending upon which of the passages 15 or 16 carries high pressure fluid to the motor 10, to urge the valve plate 80 into sealing engagement with the port face 39 on the cylinder block, thus insuring a minimal loss of hydraulic fluid through leakage.

The effective fluid areas in each of these chambers defined by the flanges 83 and 89 respectively, are equal so that for a given inlet fluid pressure the axial force on the valve plate 80 against the port face 39 will be substantially equal regardless of the direction of rotation of the device. When passage 15 carries high pressure fluid to the cylinders, chamber 86 will be pressurized acting against flange 83 to urge the valve plate 80 to the right in sealing engagement. Conversely, when high pressure fluid is ported to passage 16, chamber 94 will be pressurized acting against piston 89 and the central portion of the valve plate 80 urging the plate against the port face 39.

While the operation of the embodiment shown in FIG. 1 is believed obvious from the above description, it will be described briefly for a better understanding. Assuming that high pressure fluid is ported to passage 15 from a suitable source, alternative valve ports 82 will be pressurized. The passages 40 communicating with the high pressure ones of ports 82 will convey hydraulic fluid to their associated cylinders and the pistons in these cylinders will be forced apart causing them to ride down the cam lobe 54 adjacent thereto causing rotation of the cylinder block 28. As these pistons pass the bottom of the cam lobe 54 (bottom dead center) and begin moving up to the adjacent cam lobe the associated pistons are forced into the cylinder 37 expelling fluid from the cylinder. At this time the passages 40 associated with these pistons have serially moved out of communication with the high pressure ports and moved serially into communication with the low pressure port 82 so that fluid is expelled from the cylinders through passage 16. While the cams 49 and 52 have their lobes slightly out of phase, e.g., six degrees, the pistons 46, 47 move toward and away from one another simultaneously in the cylinder 37. It is apparent that for each revolution of the cylinder block 28 that each piston will

have a plurality of power cycles corresponding in number to the number of lobes 54 and that since there are two such pistons in each cylinder that the present device has a very high displacement per revolution.

The hydraulic motor 100 shown in FIG. 3 is generally similar to that shown in FIG. 1 except that pintle type valving is employed rather than a valve plate engaging the end of the cylinder block as described above. A housing assembly 113 supports an output shaft 111 rigidly connected to a cylinder block 128 by suitable fasteners 129. The cylinder block 128 has an axially extending portion 133 seated in a bearing 135 mounted in end plate 117. The other end of the cylinder block 128 is supported by shaft 111 which is in turn mounted in a bearing 123 seated in a boss on end plate 118. Cylinders 137 have piston assemblies 145 therein identical with assemblies 45 shown in FIG. 1.

The end plate 117 has bores 115 and 116 defining inlet or outlet passages and they are adapted to be alternatively connected either to a source of hydraulic fluid under pressure or a suitable tank, as in the embodiment shown in FIG. 1.

For the purpose of conveying hydraulic fluid from one of the passages 15, 16 to the cylinders 137, and from the cylinders to the other passages 15, 16 a cylindrical pintle 180 is provided mounted within end plate 117 and cantilevered within a central bore 30 in the cylinder block 128. Each of the cylinders 137 has a passage 140 extending generally radially therefrom and communicating with the surface of pintle 180. A first plurality of ports 182 are provided in the periphery of pintle 180 axially adjacent the passages 140. Ports 182 each communicate with an annular recess 184 through axial passages 186. Recess 184 communicates with passage 116 so that port 182 continuously remain low or high pressure ports depending upon whether passage 116 is under low or high pressure. A second set of ports 182A are provided in the periphery of the pintle 180 and are alternately disposed with respect to ports 182. Ports 182A continuously communicate with passage 115 through radial passages 187, a single central axial passage 188, radial passages 189 and annular recess 190. It is apparent that when ports 182 are high pressure ports that ports 182A will be low pressure ports and vice versa, in a manner similar to the ports 82 in FIG. 1.

The operation of the FIG. 3 embodiment is the same as that described above with respect to FIG. 1 except that fluid flows to and from the cylinders 137 through the pintle 180. If passage 115 conveys high pressure fluid to the device, the cylinders communicating with the ports 182A will be pressurized causing the pistons therein to move apart riding down the associated cam lobes resulting in rotation of the device, and the pistons moving up the associated cam lobes will expel fluid through the low pressure ports 182 and out passage 116. Conversely, when passage 116 conveys high pressure fluid to the device ports 182 will be high pressure ports and ports 182A will be low pressure ports causing the cylinder block 128 to rotate in a reverse direction.

In another embodiment as shown in FIG. 4, a hydraulic pump or motor device 210, generally similar to those in FIGS. 1 to 3, is shown driving a rotary turntable 211. Turntable 211 is adapted to support a mast carrying an articulated beam derrick on a mobile crane. The motor 210 is controlled by an operator through a manual control valve (not shown) on the rotatable mast which delivers fluid under pressure to and from conduits 213 and 214 so that the mast may be rotated as desired.

The turntable 211 is rotatably mounted in an annular support 215 by a heavy duty bearing 216. Support 215 is fixed to an annular base 228 which is adapted to be fixed to the frame of the mobile crane (not shown). The base 228 has an inwardly projecting flange 228' which defines the cylinder block for the motor 210 and has formed therein an annular series of cylinders 237

which receive piston assemblies 245 identical with those described above with reference to FIGS. 1 to 3.

For reciprocating the piston assemblies 245 rotatable cam plates 249 and 252 are provided each having multiple lobe cam tracks 253. Plate 252 is rigidly fixed to turntable 211 by threaded fasteners 218. Fasteners 219 fix the lower cam plate 249 to the upper cam plate 252 so that both rotate together with the turntable 211.

For conveying fluid to and from the cylinders 237 an annular valve member 217 is provided fixed between the cam plates 249 and 252. Valve member 217 has passages 220 and 221 therein communicating respectively with conduits 213 and 214, and opening at their other ends to axially spaced areas on the periphery of the valve 217.

Mounted between the valve member 217 and the cylinder block 228' is an annular port member 280 fixed to the valve plates by suitable pins 287. Spaced annular manifold passages 284 and 285 are provided on the inner surface of port member 280 and continuously communicate with passages 220 and 221, respectively. Manifold passage 284 communicates with ports 282 through passages 283, there being one of these passages for each of the ports 282. Further, manifold 285 continuously communicates with ports 282A through passages 288. The ports 282A are alternately disposed with respect to the ports 282 defining alternate high and low pressure ports disposed about the periphery of valve member 280 so that they serially communicate with cylinders 237 through cylinder ports 241 as the valve member rotates.

In one exemplary construction of the turntable drive shown in FIG. 4, one hundred cylinders 137 were provided, each cam 249, 252 had sixty-four cam lobes, and the port member 280 had sixty-four ports 282 and sixty-four ports 282A.

In operation of FIG. 4 embodiment, if the operator ports fluid to conduit 213, ports 282A will be high pressure ports pressurizing the cylinders 237 communicating therewith, and ports 282 will be low pressure ports receiving discharge fluid from the cylinders and conveying it to conduit 214. Rotation of the turntable 211 then begins and the pistons withdrawing from the cylinders and riding down the cam lobes and cam tracks 253 cause rotation of the cam plates along with port member 280, valve member 217 and turntable 211. The operation of the motor 210 is similar to that described above with reference to FIGS. 1 to 3, except that instead of the cylinder block rotating, the cam plates and valve members rotate.

Another embodiment as shown in FIG. 5 includes a hydraulic motor 310 generally similar in construction and operation to the motor 10 shown in FIG. 1, except that a hydrostatic thrust bearing 390 is provided for resisting axial movement of the cylinder block from valve plate 380.

The motor 310 is shown driving a boom winch 325 adapted to move various implements on the derrick of a mobile crane. The winch 325 includes a pulley 326 driven by output shaft 311 which is in turn seated in bearings 329 and 330 supported on a frame 331. A suitable brake assembly 334 is provided for holding the shaft 311 so that the motor 310 is not under load while stationary. As the details of the winch 325 form no part of the present invention, it is not believed necessary that it be described in any more detail.

The motor 310 includes a rotary cylinder block keyed at 326 to drive the shaft 311. Opposed piston assemblies 345 and cams 349 and 352 cooperate in a manner identical to that shown in FIG. 1 to drive the cylinder block in rotation.

Valve plate 380 is provided for conveying fluid to and from cylinders 337 and inlet and outlet ports 315 and 316. Valve plate 380 has an annular projection 381 seated in an annular recess 386 in cover member 317. Chamber 386 continuously communicates with port 315,

so that when port 315 receives hydraulic fluid under pressure, chamber 386 will be pressurized urging valve plate 380 to the left toward the cylinder block 328. Valve plate 380 also defines another chamber 394 adjacent cover 317 which is in continuous communication with the port 316. Thus, when port 316 is the pressurized one of the ports, chamber 394 will be pressurized urging the valve plate 380 toward the cylinder block 328.

High and low pressure ports 382a and 382b are defined by bores annularly arrayed in the valve plate 380 communicating with ports 316 and 315, respectively, through passages 388 and 396.

In fact, the valve plate 380 defines a hydrostatic thrust bearing with one end of the cylinder block 328. Toward this end, valve plate 380 has a flat generally annular surface 333 to which the ports 382 open. High pressure leakage fluid from the ports 382 across surface 333 provide a hydraulic film between the valve plate and the cylinder block.

For resisting the tendency of the cylinder block 328 to move away from the valve plate 380 a hydrostatic thrust bearing 390 is provided. Thrust bearing 390 includes a generally stationary annular plate 390a having a plurality of pockets 391 therein corresponding in size and number with the ports 382 in valve plate 380. Pockets 391 serially communicate with the cylinders 337 through passages 340 at the same time as the corresponding aligned ports 382 in the valve plate 380.

Fluid under pressure in pockets 391 passes between flat annular surface 396 on the plate and the cylinder block thereby providing a fluid film for the hydrostatic thrust bearing effect. It is apparent that each time one of the ports 382 communicates with a cylinder under pressure the pocket 391 aligned therewith will also be pressurized producing an equal and opposite thrust effect on the cylinder block 328, particularly since surface area 396 is equal to surface area 333 on valve plate 380.

As noted above with respect to the embodiment of FIG. 1 all of the ports 382a are either high or low pressure ports and the alternate ports 382b are all of the opposite state. Therefore, to provide the thrust bearing 390 with an equal and opposite effect, all of the corresponding alternate ones of the pockets 391 must be pressurized as well. As thus described, so long as the pockets 391 communicate through one of the passages 340 with a high pressure cylinder, this objective will be accomplished. However, since the passage 340 serially communicate with pockets 391, there are times when one or more of the pockets are out of communication with cylinders 337 when they should be pressurized in order to effectively provide an equal and opposite force to the thrust bearing provided by valve plate 380. Toward this end, alternate ones of the pockets 391 are interconnected by passages 393 and an annular groove 394 formed in the back of plate 390a. The other alternate pockets 391 are interconnected by passages 397 and annular groove 398. The galleries defined by these grooves and passages maintain all of the alternate pockets 391 at high pressure to maintain an equal thrust bearing force on the cylinder block 328.

I claim:

1. A hydraulic fluid energy translating device comprising: a valve member having inlet and outlet ports therein; a cylinder block rotatable relative to said valve member and having a plurality of cylinders therein serially communicable with said ports; a piston means slidable in each of said cylinders; cam means for reciprocating said piston means in said cylinders; said piston means each including a generally spherical member slidable in said cylinders and engaging said cam means, said spherical member being directly engageable with said cylinders; a sealing ring sealingly engaging one side of said spherical member, and seal means between said sealing ring said cylinders, said seal means being shiftable radially with respect to said seal ring to permit slight misalignment

therebetween as the spherical member drivingly engages the cylinders.

2. A hydraulic fluid energy translating device as defined in claim 1, wherein each of said sealing rings engage the side of the spherical member opposite said cam means.

3. A hydraulic fluid energy translating device as defined in claim 2, wherein each of said seal rings has an opening therethrough communicating the side of the spherical member opposite the cam means with fluid under pressure in said cylinder, whereby the spherical member rolls freely on said cam means.

4. A hydraulic fluid energy translating device as defined in claim 2, wherein each of said spherical members has clearance with said cylinders, said seal rings also having clearance with said cylinders so that the force of the cam means on the spherical members will cause contact between one side of the spherical members and the associated cylinders and will shift the seal rings somewhat from the centerline of the cylinders, said seal means being self-adjusting so that the clearance between the seal rings and the cylinders remains sealed even with shifting of the seal rings.

5. A hydraulic fluid energy translating device as defined in claim 4, wherein said seal means includes piston rings surrounding and carried by each of said seal rings.

6. A hydraulic fluid energy translating device as defined in claim 3, wherein said openings in the seal rings have a conical portion, said conical portions having line contact with said spherical members to effect proper sealing of said spherical members.

7. A hydraulic fluid energy translating device as defined in claim 6, wherein said conical portions diverge at a rate to expose a sufficient portion of the spherical members to fluid under pressure in the cylinders to substantially balance the force of the cam means on the spherical members to assure proper rolling contact with the cam means.

8. A hydraulic fluid energy translating device as defined in claim 2, wherein said cylinders are axially disposed in said cylinder block with respect to the axis of rotation thereof, said cam means including a cam member adjacent each end of said cylinder block having a plurality of lobes thereon, said cam members being constructed to exert only axial and tangential loads on said spherical members, there being two spherical pistons in each cylinder with each cooperating with one of said cam members, whereby the axial loads on the cylinder block will be balanced.

9. A hydraulic energy translating device comprising: a valve plate having inlet and outlet ports therein, a cylinder block slidably engaging said valve plate at one end and having a plurality of cylinders therein, piston means slidable in said cylinders, cam means adjacent said cylinder block for reciprocating the piston means in said cylinders, said valve plate defining a first hydrostatic thrust bearing resisting axial movement of said cylinder block in one direction, a second hydrostatic thrust bearing for resisting axial movement of said cylinder block in the other direction including plate means slidably engaging the end of the cylinder block opposite said valve plate, and means for supplying hydraulic fluid between said plate means and said cylinder block, said plate means having a flat generally annular bearing surface, said surface having a plurality of pockets therein corresponding in number and alignment with said inlet and outlet ports in said valve plate, said supply means including means for connecting alternate ones of said pockets with fluid under pressure.

10. A hydraulic energy translating device as defined in claim 9 wherein said supply means includes a first annular passage on the side of said plate means opposite the cylinder block, passage means interconnecting each alternate pocket with said first annular passage, a second annular passage on the side of said plate means opposite

said cylinder block, passage means interconnecting the remaining pockets with said second annular passage.

11. A hydraulic energy translating device comprising: a valve plate having inlet and outlet ports therein, a cylinder block slidably engaging said valve plate at one end and having a plurality of cylinders therein, piston means slidable in said cylinders, cam means adjacent said cylinder block for reciprocating the piston means in said cylinders, said valve plate defining a first hydrostatic thrust bearing resisting axial movement of said cylinder block in one direction, a second hydrostatic thrust bearing for resisting axial movement of said cylinder block in the other direction including plate means slidably engaging the end of the cylinder block opposite said valve plate, means for supplying hydraulic fluid between said plate means and said cylinder, said plate means having a flat generally annular bearing surface, said surface having a plurality of pockets therein corresponding in number and alignment with said inlet and outlet ports in said valve plate, said supply means including means for connecting alternate one of said pockets with fluid under pressure, said supply means including passage means in said cylinder block communicating each of said cylinders with said valve plate and with said plate means, said passage means being arranged to serially communicate with said inlet and outlet

ports and with said pockets, and means for interconnecting alternate ones of said pockets whereby each pocket will communicate with one of the cylinders even though said passage means is not in communication with all of said pockets at any instant.

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U.S. Cl. X.R.

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