United States Patent

[11] **3,585,901**

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[21] Appl. No. **800,527**
[22] Filed **Feb. 19, 1969**
[45] Patented **June 22, 1971**
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[54] **HYDRAULIC PUMP**
11 Claims, 6 Drawing Figs.

[52] U.S. Cl. 91/6.5, 417/312, 91/499
[51] Int. Cl. F04b 9/02, F01b 13/04
[50] Field of Search 103/162; 230/128; 91/198, 6.5; 92/57

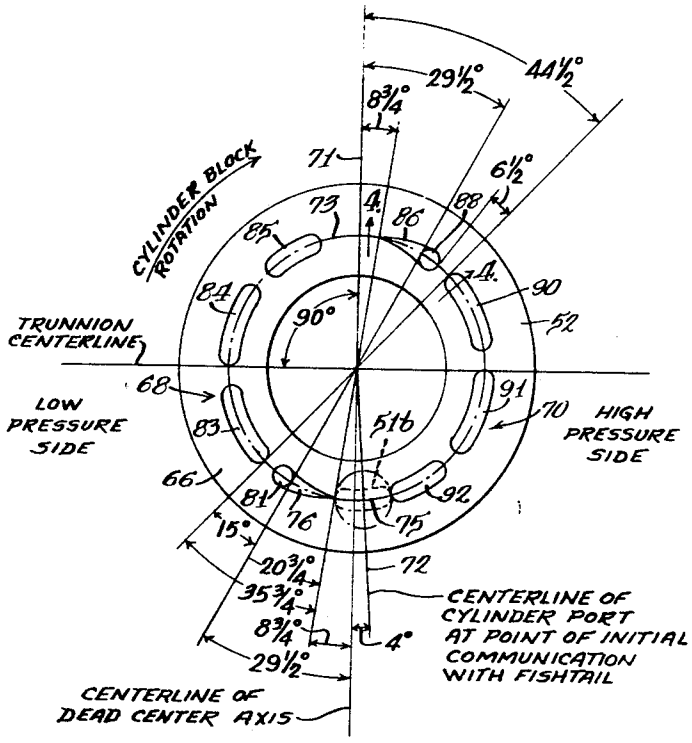
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ABSTRACT: A hydraulic pump of the multiple piston and cylinder type having inlet and outlet ports in a valve plate which serially communicate with the cylinders, with each of the ports being defined by the following, (1) a tapered fishtail groove which communicates with each cylinder prior to the time the piston in the cylinder reaches one of its dead center positions (2) a circular port extending through the valve plate and communicating with the trailing end of the fishtail groove, and (3) an elongated arcuate port separate from and trailing the circular port.



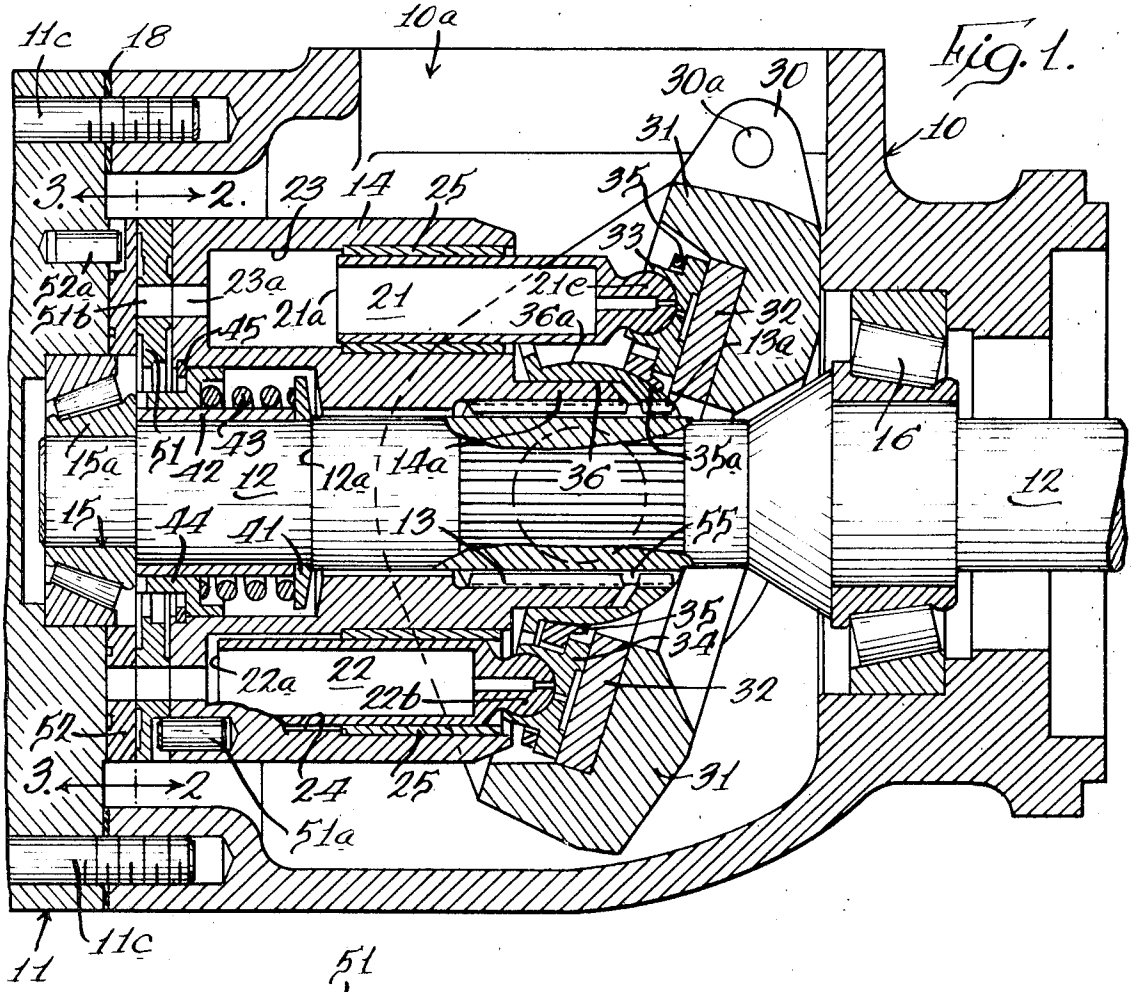


Fig. 1.

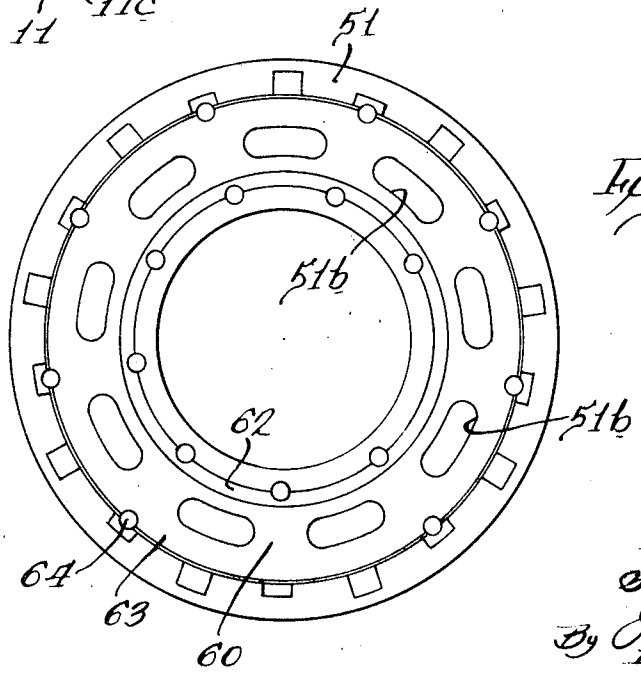


Fig. 2.

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Fig. 3.

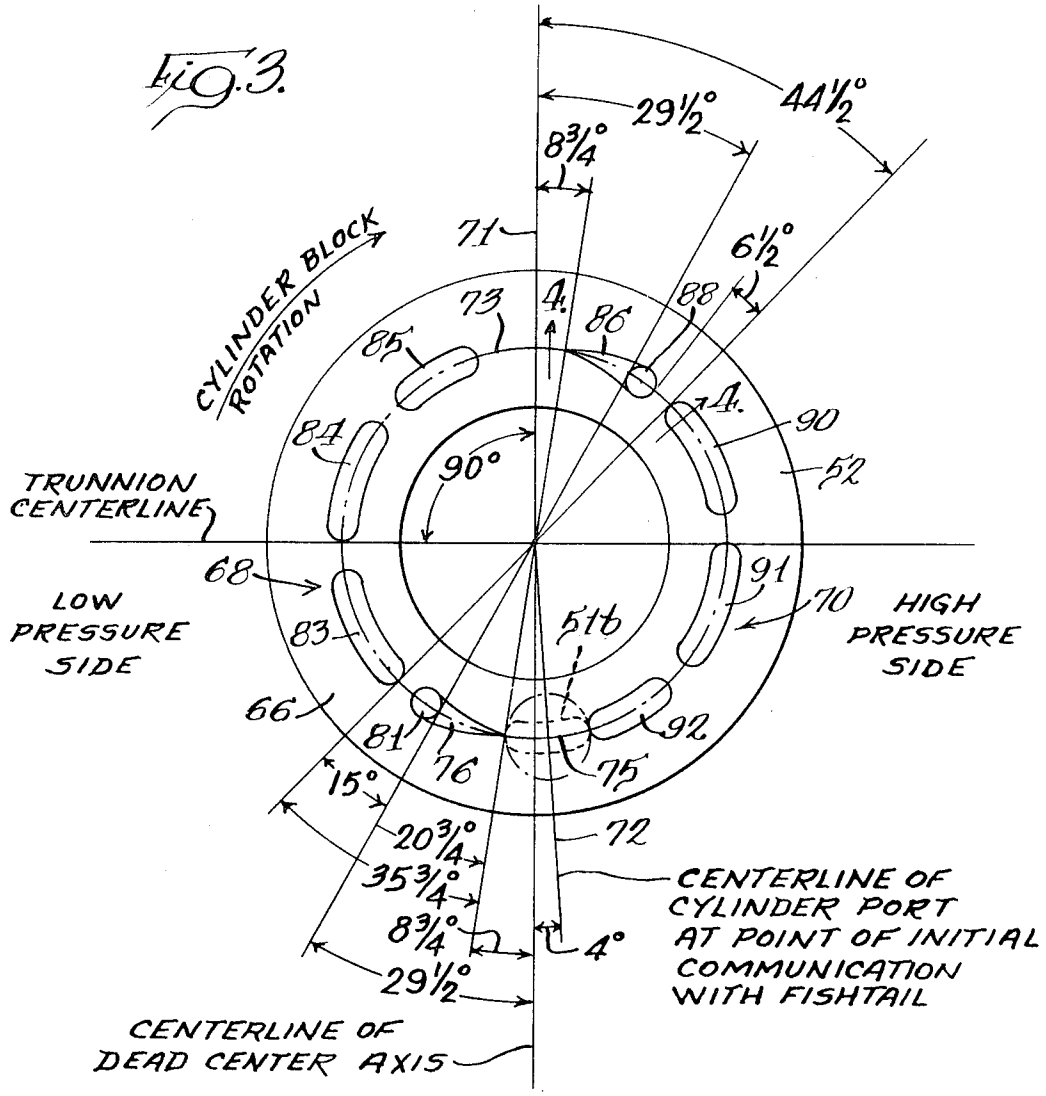

44½°
29½°
8¾°
71
6½°
73
4.
86
88
85
CYLINDER BLOCK ROTATION
4.
84
90
52
90°
TRUNNION CENTERLINE
68
91
70
LOW PRESSURE SIDE
HIGH PRESSURE SIDE
83
51b
66
81
76
75
92
15°
20¾°
72
35¾°
CENTERLINE OF CYLINDER PORT AT POINT OF INITIAL COMMUNICATION WITH FISHTAIL
8¾°
4°
29½°
CENTERLINE OF DEAD CENTER AXIS

Fig. 4.

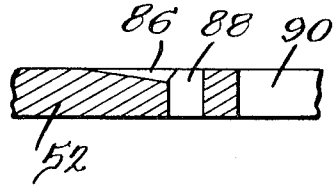

86
88
90
52

Fig. 5.

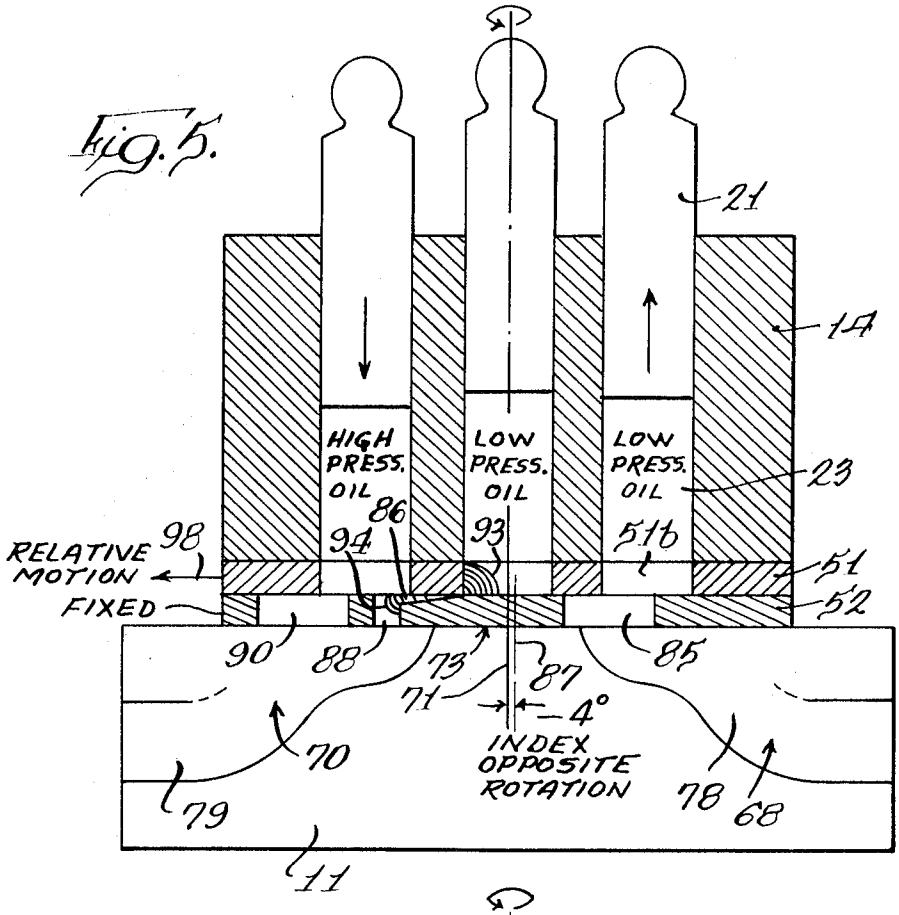
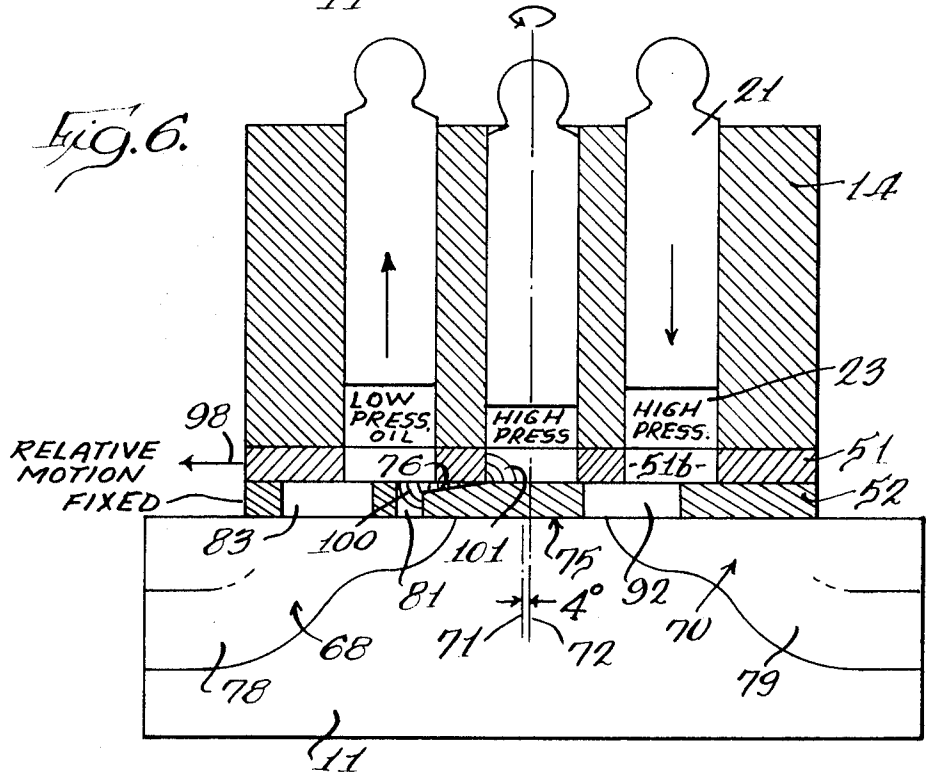

21
14
HIGH PRESS. OIL
LOW PRESS. OIL
LOW PRESS. OIL
23
98
RELATIVE MOTION
FIXED
86
94
93
51b
51
52
90
88
73
87
85
71
4°
70
INDEX OPPOSITE ROTATION
79
78
68
11

Fig. 6.


21
14
23
LOW PRESS. OIL
HIGH PRESS
HIGH PRESS.
98
RELATIVE MOTION
FIXED
76
51b
51
52
83
100
101
75
81
4°
92
68
71
72
70
78
79
11

HYDRAULIC PUMP

BACKGROUND OF THE PRESENT INVENTION

Multiple piston hydraulic units have gained considerable acceptance both in the hydrostatic transmission field and even more generally as hydraulic units for supplying fluid under pressure. One disadvantage in hydraulic units of this character, which has limited their desirability in some applications, is the noise created by the units which is objectionable to the human ear and the initiating vibrations which may cause damage or reduction in life to the associated parts of the unit.

One of the main sources of noise generated by these multiple piston pumps is the generation of pulsing wave fronts which result from the sudden release of high-pressure oil both from the cylinders into the ports and from the ports into the cylinders. For example, when the device is operating as a pump, when the cylinders pass from the low-pressure port to the high-pressure port and communicate with the latter port, there is a sudden release of high-pressure fluid into the cylinder. Conversely, as the cylinders pass from the high-pressure port to the low-pressure port there is a sudden release of fluid from the cylinders to the low-pressure port.

These wave fronts consist of a compressive wave front traveling through the hydraulic fluid in a direction from the high-pressure oil (which may be either in the port or the cylinder) to the low-pressure oil. An additional source of noise is a rarefactive wave front which travels through the hydraulic fluid in a direction from the low-pressure oil to the high-pressure oil, which of course is in a direction opposite to the compressive wave front described above. As the high pressure compressive wave front moves through the low-pressure oil, an area of lower pressure exists behind the wave front, and the tendency of the oil to flow backward from the compressive front to the lower pressure area creates the rarefactive wave front.

Both of these wave fronts are characterized by impulse motions transmitted to the structure of the pump and all fluid passageways interconnecting the pump and auxiliary components. The fronts travel through the structure of the pump and auxiliary components in a manner similar to the way they travel through the hydraulic fluid itself. As the fronts approach the outer surface of the hydraulic unit, the waves are transmitted to the air much like a loud speaker transmits sound to the air. Then the front propagates through the air and at this point it may be classified as an airborne noise disturbance which may be objectionable to the human ear.

As the cylinders pass from the low to the high-pressure ports, and as they come into contact with the high-pressure port in conventional units, the compressibility of the oil in the high-pressure port and the associated kidney accelerates fluid particles into the cylinder and generates a compressive wave front. This front of energy travels through the fluid in the cylinder until it contacts the end of the piston in the cylinder. Some of the energy of this front is transmitted through the piston and into the driving swashplate, and from the swashplate through the supporting trunnion and into the air where it forms a noise disturbance. The remaining energy from the front not transmitted through the piston is reflected back into the fluid in the bore and propagates throughout the rest of the hydraulic structure and it eventually also is transmitted to the air.

The rarefactive wave front, described above generally, is generated at the same time, propagates into, and through the high-pressure oil in the high-pressure port.

An opposite wave front effect occurs as the cylinders move from the high-pressure port to the low-pressure port and come into communication with the low-pressure port. The compressibility of the oil in the cylinders accelerates fluid particles into the low-pressure kidney and generates a compressive wave front in the kidney. The front of energy travels through the fluid in the port of the associated end cap and excites the structure surrounding the fluid, and eventually is transmitted to the air as an unwanted noise disturbance. The rarefactive wave front is generated in the opposite direction and propagates from the low-pressure kidney into and through the high-pressure oil in the cylinder bore, through the piston and swashplate, and again results in an unwanted noise disturbance.

The objective of the present invention is to reduce the magnitude of these wave fronts and the resulting vibration and noise disturbances.

SUMMARY OF THE PRESENT INVENTION

In accordance with the present invention, the energy of the wave fronts generated in a hydraulic unit at both the low to high and high to low pressure crossovers in the valve plate is reduced or minimized by reducing the rate of pressure rise in the low-pressure chamber. In this sense the chamber may be either the port, if it be the low-pressure port, or the cylinders, if the cylinders be low-pressure cylinders. Thus, the magnitude of the disturbances is dependent upon this rate of pressure rise.

The rate of pressure rise according to the present invention is controlled and minimized by reducing the rate of bulk modulus flow of oil between the ports and the cylinders and by communicating the cylinders with the approached port prior to the piston and the associated cylinder reaching their dead center position. Bulk modulus flow may be defined as flow of oil due to compression or expansion of the oil when exposed to a change of pressure.

A tapered groove, sometimes referred to as a fishtail, is provided adjacent the leading end of each of the high- and low-pressure ports. These provide orifices that change and control the rate of bulk modulus flow. By providing a gradually increasing orifice the rate of pressure rise in the cylinders is decreased since it is proportional to the rate of change of flow between the ports of the cylinders. Moreover, the positioning and shape of the present fishtail or groove limits the rate of bulk modulus flow so as to decrease the magnitude of the disturbance generated but at the same time allows complete cylinder filling so as not to produce pump starvation.

The present indexing of the ports, or more specifically the angular relationship of the leading edge of the fishtail and the cylinders when they communicate therewith as compared to the position of the cylinders when the pistons therein are at their dead center positions, further contributes to reducing noise disturbance. For example, at the low- to high-pressure crossover the cylinders communicate with the fishtails while the pistons are still in their suction stroke. Because at this time the bulk modulus flow of oil tends to increase pressure in the cylinders as they communicate with the high-pressure port through the fishtail at a certain pressure rise rate, the outward movement of the piston tends to decrease this pressure rise rate and the net effect is a lower pressure rise rate into the low-pressure port.

A similar phenomenon occurs at the high- to low-pressure crossover. As the cylinders communicate with the low-pressure port through the fishtail, the pistons are still moving in their compressive stroke so that while the bulk modulus flow of oil tends to decrease the pressure in the cylinder at a certain rate, this decrease is offset by the compressive stroke of the piston and again the net effect is a lower pressure change rate in the cylinders.

To further decrease the energy of the wave fronts generated, the waves are baffled before they propagate out into the manifold associated with the valve plate by a discrete aperture through the valve plate associated with each port and with the communicating trailing ends of each of the fishtail grooves. These apertures reflect the wave fronts back toward the fishtail openings so that each wave must make a series of reflections, each of which decreases the wave energy.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal cross section taken through a hydraulic pump embodying the principles of the present invention;

FIG. 2 is a subassembly view taken generally along line 2–2 of FIG. 1 showing the port plate;

FIG. 3 is a subassembly view taken generally along line 3–3 of FIG. 1 showing the valve plate;

FIG. 4 is a fragmentary section taken generally along line 4–4 of FIG. 3 showing one of the tapered fishtail grooves;

FIG. 5 is a schematic illustration of several of the cylinders passing from the low-pressure port to the high-pressure port during pumping; and

FIG. 6 is a schematic illustration of several of the cylinders passing from the high-pressure port to the low-pressure port during pumping.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Turning now to FIG. 1, the embodiment illustrated is incorporated in a pump of the axial piston type. The pump includes a housing indicated generally at 10, having an end cap plate 11 removably secured thereto by suitable screw or bolt means 11*c*. A drive shaft 12 is rotatably supported at one end of housing 10 by mounting bearings 16 and extends through cylinder block 14 to end plate 11 where the drive shaft is secured by press fit within race 15*a* of mounting bearing 15 abutting end plate 11. If desired, the press fit may be omitted and hearing 15 retained solely by contact with end plate 11. Bearings 15 and 16 secure drive shaft 12 against axial movement.

Drive shaft 12 is drivingly connected as by splines 13 to cylinder block 14 for rotation of cylinder block 14 and drive shaft 12 together. The splined connection 13 includes cooperating splines on the drive shaft and cylinder block through which the drive shaft passes, and the splines are slightly crowned longitudinally to permit tilting of the cylinder block at the splined connection, which allows for irregularities in the rotation of the cylinder block.

Cylinder block 14 includes pistons, two of which are shown at 21 and 22 having inner ends 21*a* and 22*a* reciprocating within bores or cylinders 23 and 24 in cylinder block 14. Cylinders 23 and 24 are provided with bearing inserts or bushings 25 within which the pistons reciprocate. Although only two pistons are shown in FIG. 1, it is to be understood that cylinder block 14 includes an annular plurality of axially disposed cylinders within the cylinder block, each cylinder having an inner end of a piston reciprocating therein. Although it will be apparent from the discussions below that there are nine pistons in the annular series in the illustrated embodiment, any number of pistons and cylinders may be used, as is known to the art.

Cam or swashplate 31 is mounted within housing 10 at one end of cylinder block 14 for pivotal movement about an axis transverse to and intersecting the axis of the drive shaft. The pivotal mounting of cam plate 31 is provided by trunnions (not shown) secured to housing 10. The cam plate is adapted to be pivoted about its axis at an incline in either direction with respect to a neutral central position for adjustment of displacement of the pistons as at 21 and 22 within the cylinder block cylinders. For this purpose projection 30 on cam plate 31 is provided to pivot the cam plate on its axis. The projection is accessible through opening 10*a* in the wall of housing 10 and may be linked to suitable actuating means connectable by means of bore 30*a* to control the inclination of cam plate 31 as disclosed, for example, in the Hann et al. U.S. Pat. No. 3,126,707, assigned to the assignee of this application.

The outer ends of pistons 21 and 22 are of spherical configuration and are universally connected to bearing members such as bearing shoes 33 and 34 which are adapted to slide upon an annular cam plate, thrust bearing member or surface 32 which is supported on cam plate 31. Of course, other

pistons within the pump are equipped with similar bearing shoes and wherever pistons 21 and 22 are described in association with other members, it is to be understood that the remainder of the pistons are in association with similar or the same members, as will be obvious to those in the art from the descriptions herein.

A holddown member, such as ring 35, engages the bearing shoes 33 and 34 and carries the bearing shoes in slidable contact with bearing surface 32. Holddown ring 35 is provided with a spherical bore 35*a* which is engaged by and receives the spherical outer surface 36*a* of a spherical collar 36. Spherical collar 36 is splined at 13*a* for rotation with and axial movement relative to shaft 12. The collar is slidable on and piloted by cylindrical projection 14*a* of cylinder block 14 for axial movement of collar 36 with respect to the block 14 and shaft 12. Collar 36 is adapted to urge holddown ring 35 toward the bearing surface of cam plate 31, thereby urging the bearing shoes against the bearing surface.

An auxiliary piston return is provided by annular shim 55, illustrated in FIG. 1 as a crushed or bent washer. Spherical collar 36 is spaced longitudinally along shaft 12 from cylinder block 14 by means of shim 55 which is interposed between spherical collar 36 and cylinder block 14. Shim 55 acts as a stop to prevent longitudinal abutment of the cylinder block and spherical collar.

In order to maintain bearing shoes 33 and 34 against bearing surface 32, a spring means, such as an annular series of compression springs (not shown) are interposed between the cylinder block cylinders. As shown in U.S. Pat. No. 3,191,543, these compression springs are grounded to drive shaft 12 through annular collar 41 and bias the spherical collar 36, carrying holddown ring 35 in the direction of cam plate 31, thereby urging shoes 33 and 34 against the bearing surface 32.

More particularly, with respect to shim 55, the shim functions to maintain bearing members 33 and 34 in close proximity with the bearing surface 32 of cam plate 31 without respect to the urging of the compression springs. Thus, if the compression springs should become overpowered during operation of the device, e.g., by sticking of pistons within cylinder block cylinders or by momentary sticking delay of the pistons being urged from the cylinders upon a rapid increase in torque, shim 55 prevents the withdrawal of the piston bearing members from the cam plate a greater distance than a minimum operating clearance. The minimum operating clearance is preselected and is provided during assembly of the device. Accordingly, as the elements are assembled from the left side of housing 10 with housing end plate 11 removed, the spherical collar 36 is positioned, followed by an annular washer and then cylinder block 14, etc. The annular washer is sufficient to cause the valve plate 52 to abut housing end plate 11 before housing end plate 11 is peripherally secured in sealing relation with housing 10.

Annular collar 41 abuts annular shoulder 12*a* on drive shaft 12 and is held in abutment with the shoulder by means of annular sleeve 42 around drive shaft 12. Annular sleeve 42 in turn abuts race 15*a* of mounting bearing 15. Either annular sleeve 42 or race 15*a* is press fitted on shaft 12 to in effect anchor annular sleeve 42 against axial movement with respect to shaft 12.

A second spring means such as coil spring 43 is provided, reacting against annular collar 41. Coil spring 43 biases an annular collar 44, which is axially slidable on sleeve 42 or in the cylinder block toward a hardened port plate 52. Annular collar 44 abuts snap ring 45, which is secured to cylinder block 14, and thereby carries the snap ring and cylinder block toward valve plate 52 under the urging of spring 43. A hardened port plate 51 pinned to the cylinder block at 51*a* rotates with the cylinder block and is carried by the cylinder block under the urging of spring 43 into slidable contact with valve plate 52 which is pinned at 52*a* to end plate 11.

Cylinders 23 are provided with arcuate end ports 23*a* and port plate 51 includes complementary extensions 51*b* of ports 23*a* therethrough. The passage configuration, as viewed from

either side of plate 51 is the same as the passage configuration on the end of cylinder 14 as illustrated in FIG. 2. Plate 51 may include wear-resistant materials on its opposing surfaces and particularly on the surface facing valve plate 52, or may be entirely of suitable bearing material such as bronze.

Valve plate 52 includes arcuate inlet and outlet port means 68 and 70 for conducting fluid to and from passages 51*b* as cylinder block 14, including plate 51, rotates. Valve plate 52 may also have wear-resistant surfaces, such as hardened steel, especially facing cylinder block end plate 51.

The inlet and outlet are connectable to fluid lines (not shown) for operation of the device as a pump as will be readily apparent to those skilled in the art.

The spring means biasing the piston bearing members against the cam plate is adapted to exert a force upon the spherical collar approximately double the force exerted by the spring means acting to bias the cylinder block against the port plate, since it has been found that the forces tending to separate the piston bearing members from the cam plate are approximately double those tending to separate the cylinder block from the port plate.

Returning to the construction of the port plate 51 in somewhat more detail and as shown in FIG. 2, the ports 51*b* are seen to be arcuate in configuration, extend completely through the port plate 51 and open at a continuous sealing land 60 flanked by annular recess grooves 62 and 63. Grooves 62 and 63 collect leakage oil from sealing surface 60 and deliver this fluid to the rear side of the bearing plate through axial bores 64 communicating with the recesses.

The valve plate 52 as shown clearly in FIG. 3 is seen to include a sealing surface 66 which slidably engages sealing surface 60 on the port plate 51. Formed in the valve plate 52 and opening to the sealing surface 66 is an inlet port means 68 and an outlet port means 70. Since the present porting configuration is primarily adapted for use in a hydraulic pump, the inlet port 68 may be considered the low-pressure port and the outlet port means 70 may be considered the high-pressure port.

Since the position of the port means with respect to piston position is important in the present invention, it will be helpful to reference the port means with respect to a piston dead center axis 71 as shown in FIG. 3. Axis 71 is defined by a line through the centers of the pistons when the pistons are at their top dead center and bottom dead center positions. Assuming the cylinder block 14 to be traveling in a clockwise direction of rotation as indicated in FIG. 3, the portion of the sealing surface 66 between the low-pressure port means and the high-pressure port means may be defined as a low- to high-pressure crossover 73, while the portion of the surface 66 opposite thereto may be defined as the high- to low-pressure crossover 75.

As seen in FIG. 5, the pistons 21 pass through their bottom dead center positions at crossover 73 and as seen in FIG. 6, the pistons pass through their top dead center positions as they pass crossover 75.

While the construction of the inlet port means 68 is identical to the outlet port means 70, they function somewhat differently as described above and as further described below.

Referring to FIG. 3, inlet port 68 includes a tapered V-shaped fishtail groove 76 providing gradual communication between cylinder ports 51*b* leaving crossover 75 and low-pressure port 78 in end cap 11. A high-pressure port 79 is also provided in end cap 11 and continuously communicates with the high-pressure port means 70 in valve plate 52 in a similar fashion. The fishtail groove 76 communicates with the low-pressure port 78 through a bore 81 extending completely through the valve plate 52. In an exemplary construction, the leading end of the fishtailed groove 76 is 8¾° from the piston dead center axis 71 and the leading edge of bore 81 is 29½° from the dead center axis. The important positioning of the fishtail groove 76, however, is that it be located with respect to the dead center axis 71 such that the approaching cylinder port 51*b* begins communication therewith when the center of the cylinder port is 4° ahead of the piston dead center axis 71 as shown at 72 in both FIG. 3 and FIG. 6.

The remaining portion of the inlet port means 68 is defined by a plurality of elongated arcuate apertures 83, 84 and 85 extending completely through the valve plate 52. If desired, these latter apertures may be combined into a single elongated aperture, so long as they remain separate and spaced from the bore 81. The leading edge of port 83 is 44½° from the dead center axis.

The outlet port means 70 is identical in construction to the inlet port means 68 and is seen to include a tapered fishtail groove 86 communicating with port 79 through bore 88, and a plurality of separate elongated apertures 90, 91 and 92.

As shown in FIG. 4, the fishtail grooves 76 and 86 have a gradually increasing depth, as well as width. While the operation of the above device is believed obvious from the detailed description, reference will be made to FIGS. 5 and 6 for a brief description of the manner in which the present device reduces vibration and noise. Referring to FIG. 5 which illustrates the low- to high-pressure crossover movement of the cylinders 23, the right cylinder is in communication with low-pressure aperture 85 and the piston 21 therein is withdrawing from the cylinder so that the cylinder is filling with low-pressure oil. When the cylinder arrives at the middle position 87 between port 85 and fishtail 86, it is 4° short of the dead center axis 71 (i.e. with continued rotation of the cylinder block 14 in the direction of arrow 98, the piston in the middle cylinder will continue its outward movement for 4° after position 87 before it begins its inward movement at position 71). When the pistons pass 87, they begin communication with the tapered fishtail groove 86. Because of the construction of this groove, the bulk modulus flow of oil from the high-pressure port 79 will increase gradually, reducing the otherwise high rate of pressure rise in the cylinder. At the same time the piston in this cylinder is still moving outwardly so that the rate of pressure rise in the cylinder is decreased even further. The compressive wave fronts are indicated by lines 93 in FIG. 5, and are shown traveling from the high-pressure port 79 into the middle cylinder as communication therebetween is initiated. The rarefactive wave fronts described above as moving in the opposite direction are indicated by lines 94 in FIG. 5. The rarefactive wave fronts are reflected and diminished in energy by the sides of the bore 88.

As the middle cylinder continues movement from the position shown in FIG. 5, gradually increasing communication is provided with high-pressure port 79 until the cylinder port fully communicates with bore 88 and aperture 90 as shown by the left cylinder illustrated in FIG. 5.

The porting operates in a similar manner as the cylinders pass from the high-pressure port to the low-pressure port as shown in FIG. 6 although the direction and propagation of the wave fronts are somewhat different. In this case, as the cylinder block moves in the direction of arrow 98, the right cylinder port 51*b* is in communication with the high-pressure port 92 so that the associated cylinder 23 is under high pressure, and the piston 21 is moving inwardly toward its top dead center position. As the cylinder 23 reaches the position 72, the piston therein is still moving inwardly and at this time the associated port 51*b* begins communication with the fishtail groove 76. Since the center of the middle cylinder is at this time 4° short of the dead center axis 71, the piston therein is still moving inwardly. Communication with the low-pressure port 78 causes a compressive wave front indicated at 100 from the cylinder to the low-pressure port. This wave front is minimized, however, because of the gradual communication provided by the fishtail groove 76 so that the rate of pressure rise in the port 78 is minimized. At the same time, since the piston continues movement inwardly it tends to offset by its compressive stroke the decrease in pressure in the cylinder caused by oil escaping through the fishtail groove 76, thus reducing the rate of pressure decrease in the cylinder as it moves from the piston 72 to the dead center position 71. The cylinder continues to move toward a position in full communication with the low-pressure port 78 through bore 81 and aperture 83 in the valve plate. In this case the bore 81 serves to deflect and reduce the energy of the compressive wave

front 100. The rarefactive wave front is indicated by lines 101 in FIG. 6 and is seen to move into the cylinders.

We claim:

1. A hydraulic energy translating device, comprising: housing means having inlet and outlet ports therein, a cylinder block having cylinders therein and being rotatable relative to said housing means so that the cylinders serially communicate with said inlet and outlet ports, pistons slidable in said cylinders, cam means for reciprocating said pistons between bottom and top dead center positions, the positions of the pistons at said dead center positions defining a dead center axis substantially intersecting the axis of rotation of said cylinder block, valve means including a valve plate, outlet port means in said valve plate including means in said valve plate for gradually increasing communication between the approaching cylinders and the outlet port so that the pressure rise in the approaching cylinder will be gradual, said means for gradually increasing communication between the approaching cylinders and the outlet port means being positioned so that the approaching cylinder communicates with the outlet port prior the time when the associated piston reaches its dead center position so that the pressure rise in the cylinder is reduced by the movement of the associated piston toward its dead center position including a tapered groove in said valve plate increasing in depth and width toward the outlet port means to provide uniformly increasing communication between the approaching cylinder and the outlet port means.

2. A hydraulic energy translating device, comprising: housing means having inlet and outlet ports therein, a cylinder block having cylinders therein and being rotatable relative to said housing means so that the cylinders serially communicate with said inlet and outlet ports, pistons slidable in said cylinders, cam means for reciprocating said pistons between bottom and top dead center positions, the positions of the pistons at said dead center positions defining a dead center axis substantially intersecting the axis of rotation of said cylinder block, valve means including a valve plate, said valve plate having inlet and outlet port means therein, at least one of said port means including a tapered groove at the leading edge thereof providing gradually increasing communication between the approaching cylinder and the outlet port, said one port means including a discrete aperture through the valve plate at the trailing end of the tapered groove, said one port means also including an elongated aperture separate from and trailing said discrete aperture.

3. A hydraulic energy translating device as defined in claim 2, wherein both the inlet and the outlet port means include said tapered groove, said discrete aperture and said separate elongated aperture.

4. A hydraulic energy translating device as defined in claim 2, wherein said tapered groove is positioned with respect to said dead center axis so that the approaching cylinder communicates with said one port means prior to the time the associated piston reaches its dead center position.

5. A hydraulic pump, comprising: housing means having inlet and outlet ports therein, a cylinder block having cylinders therein and being rotatable relative to said housing means so that the cylinders serially communicate with said inlet and outlet ports, pistons slidable in said cylinders, cam means for reciprocating said pistons between bottom and top dead center positions, the positions of the pistons at said dead center positions defining a dead center axis substantially intersecting the axis of rotation of said cylinder block, a valve plate having low-pressure inlet port means and high-pressure outlet port means therein, at least one of said port means including means providing gradually increasing communication between the approaching cylinder and the associated port, said means providing gradually increasing communication between the approaching cylinder and the associated port being positioned with respect to said dead center axis so that the approaching cylinder communicates with the associated port prior to the piston in the cylinder arriving at its dead center position, including a tapered groove in said valve plate increasing in depth and width toward the outlet port means to provide uniformly increasing communication between the approaching cylinder and the outlet port means.

6. A hydraulic pump, comprising: housing means having inlet and outlet ports therein, a cylinder block having cylinders therein and being rotatable relative to said housing means so that the cylinders serially communicate with said inlet and outlet ports, pistons slidable in said cylinders, cam means for reciprocating said pistons between bottom and top dead center positions, the positions of the pistons at said dead center positions defining a dead center axis substantially intersecting the axis of rotation of said cylinder block, a valve plate having low-pressure inlet port means and high-pressure outlet port means therein, at least one of said port means including means providing gradually increasing communication between the approaching cylinder and the associated port, said means providing gradually increasing communication between the approaching cylinder and the associated port being positioned with respect to said dead center axis so that the approaching cylinder communicates with the associated port prior to the piston in the cylinder arriving at its dead center position, said one port means being the outlet port and the means providing gradual communication being a tapered groove in the valve plate positioned so that the approaching cylinder communicates with the outlet port prior to when the piston in the cylinder reaches its bottom dead center position.

7. A hydraulic pump, comprising: housing means having inlet and outlet ports therein, a cylinder block having cylinders therein and being rotatable relative to said housing means so that the cylinders serially communicate with said inlet and outlet ports, pistons slidable in said cylinders, cam means for reciprocating said pistons between bottom and top dead center positions, the positions of the pistons at said dead center positions defining a dead center axis substantially intersecting the axis of rotation of said cylinder block, a valve plate having low-pressure inlet port means and high-pressure outlet port means therein, at least one of said port means including means providing gradually increasing communication between the approaching cylinder and the associated port, said means providing gradually increasing communication between the approaching cylinder and the associated port being positioned with respect to said dead center axis so that the approaching cylinder communicates with the associated port prior to the piston in the cylinder arriving at its dead center position, said one port means being the inlet port means and the means providing gradually increasing communication being a tapered groove in the valve plate positioned to communicate the approaching cylinder with the inlet port prior to when the piston in the cylinder reaches its top dead center position.

8. A hydraulic pump, comprising: housing means having inlet and outlet ports therein, a cylinder block having cylinders therein and being rotatable relative to said housing means so that the cylinders serially communicate with said inlet and outlet ports, pistons slidable in said cylinders, cam means for reciprocating said pistons between bottom and top dead center positions, the positions of the pistons at said dead center positions defining a dead center axis substantially intersecting the axis of rotation of said cylinder block, a valve plate having low-pressure inlet port means and high-pressure outlet port means therein, at least one of said port means including means providing gradually increasing communication between the approaching cylinder and the associated port, said means providing gradually increasing communication between the approaching cylinder and the associated port being positioned with respect to said dead center axis so that the approaching cylinder communicates with the associated port prior to the piston in the cylinder arriving at its dead center position, each of the port means in the said valve plate including a fishtail tapered groove leading the port means, a generally cylindrical aperture through said plate trailing and communicating with said tapered groove, and at least one separate elongated arcuate aperture through said plate trailing the cylindrical aperture.

9. A hydraulic pump as defined in claim 8, wherein the tapered grooves are positioned so that the approaching cylinder communicates with the approached port when the center of the cylinder is substantially 4° behind the dead center axis.

10. A hydraulic pump as defined in claim 8 wherein the cylindrical aperture is positioned so that the approaching cylinders communicate therewith substantially 30° after the dead center axis.

11. A hydraulic pump as defined in claim 8 wherein said elongated arcuate aperture is positioned so that the approaching cylinder communicates therewith substantially 44° after the dead center axis.