

May 31, 1955

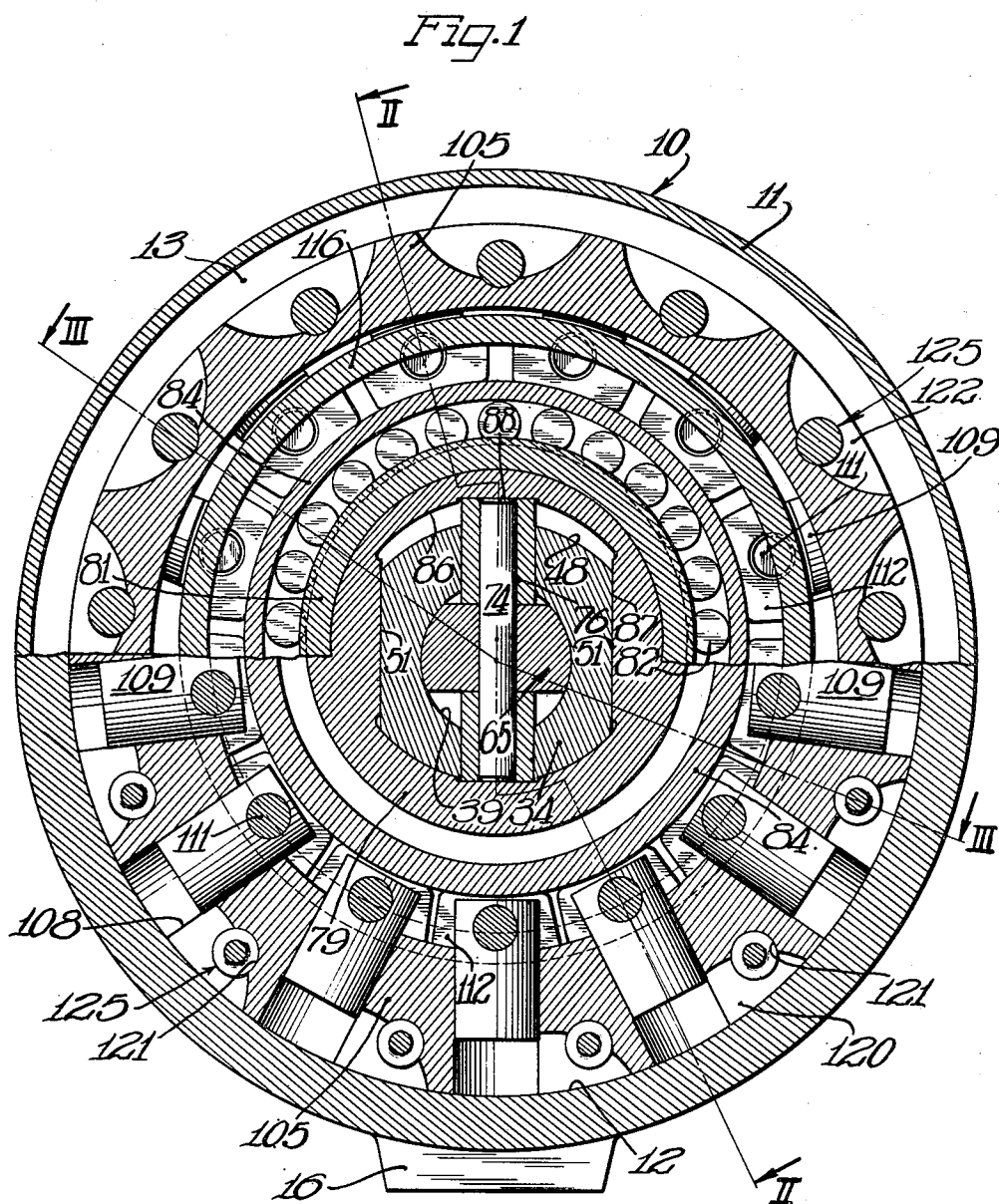
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2,709,408

VARIABLE DISPLACEMENT RADIAL PISTON PUMP

Filed Jan. 25, 1951

4 Sheets-Sheet 1



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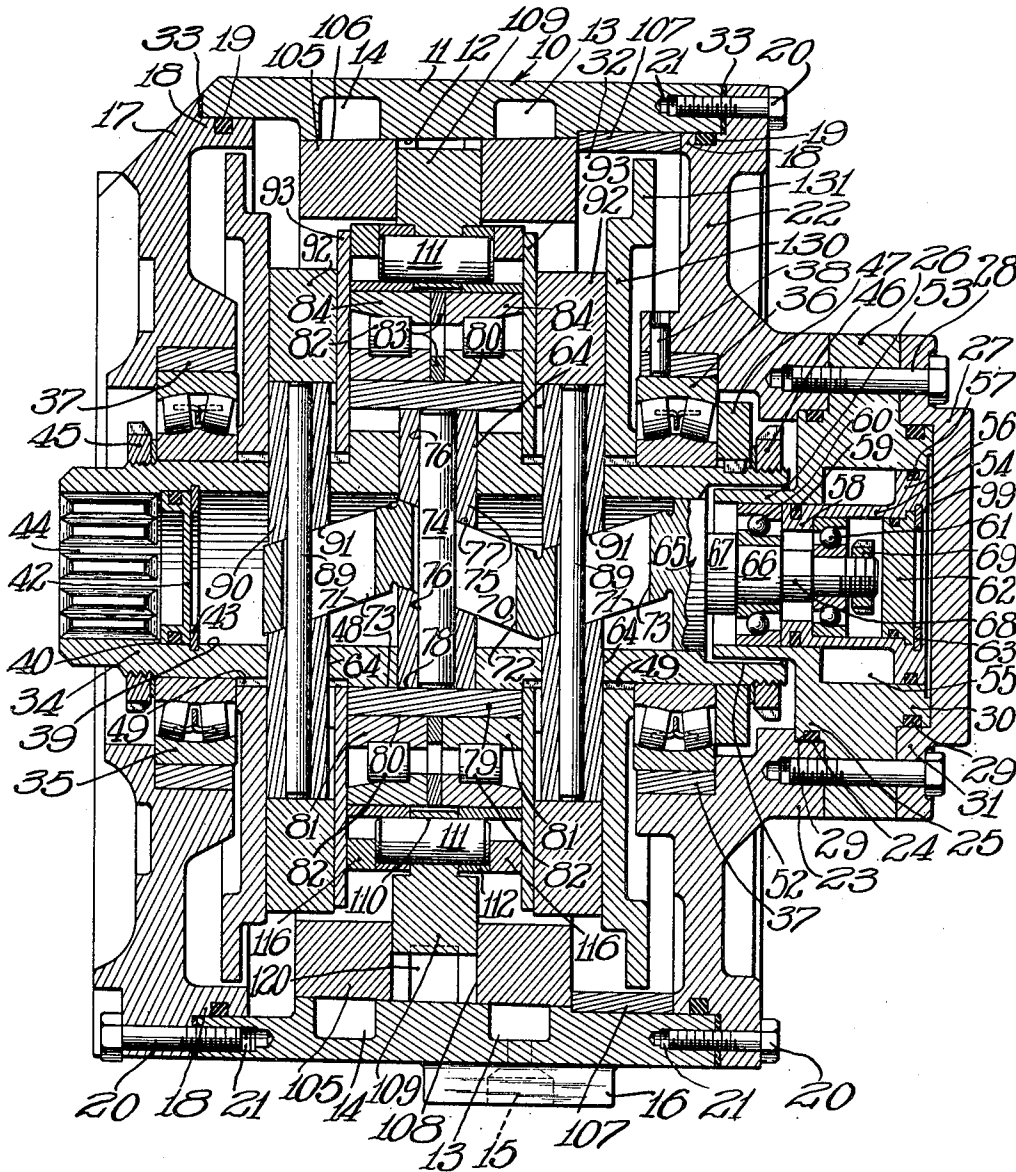
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VARIABLE DISPLACEMENT RADIAL PISTON PUMP

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4 Sheets-Sheet 2

Fig. 2



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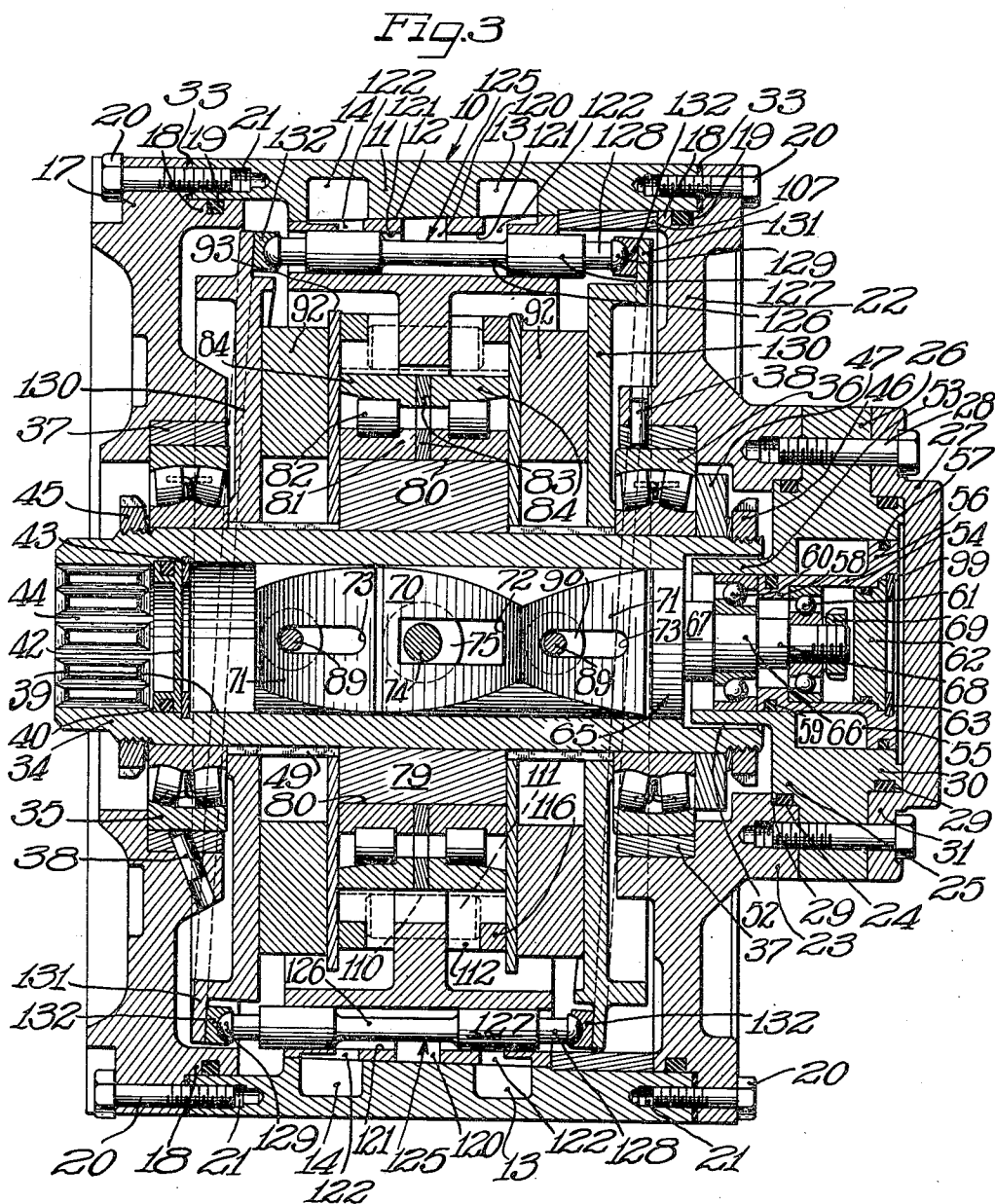
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VARIABLE DISPLACEMENT RADIAL PISTON PUMP

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4 Sheets-Sheet 3



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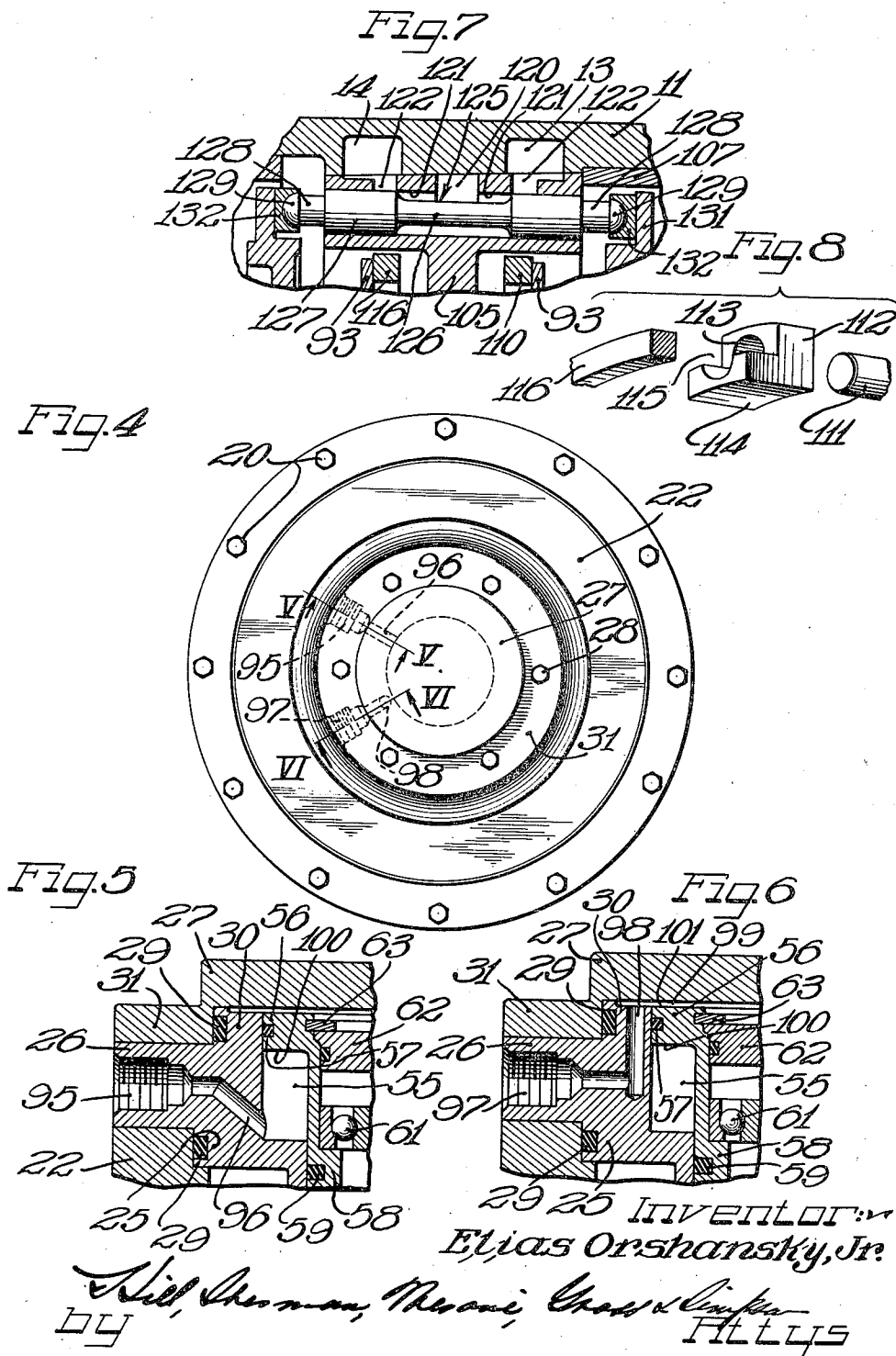
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VARIABLE DISPLACEMENT RADIAL PISTON PUMP

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2,709,408

VARIABLE DISPLACEMENT RADIAL PISTON PUMP

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Application January 25, 1951, Serial No. 207,697

4 Claims. (Cl. 103—38)

The present invention relates to a variable displacement radial piston pump and more particularly to a multi-piston radial pump wherein the magnitude of piston displacement is varied through a relatively wide range without disturbing the position of the pump driving means and without disturbing the dynamically balanced condition of the pump.

Various types of variable displacement radial piston pumps have been proposed in the prior art. Such prior pumps have generally effected a displacement variation by shifting the center of eccentric rotation, either by movement of a pintle or a drive shaft, or by movement of a rotatable cylinder block. This displacement of rotary parts has seriously upset the dynamic balancing of the pump structure, causing a loss in pumping efficiency, and efforts to prevent such a loss of efficiency have resulted in the provision of very complicated shifting mechanisms.

The present invention now provides an improved type of variable displacement radial piston pump wherein an extremely simple, automatically compensating means is utilized for maintaining a constant dynamic balance within the pump throughout the entire range of displacement variation. Further, the present invention effects the displacement variation without shifting the position of either the pump driving means or the pump cylinder block, and the variation may be governed by fluid pressure means responsive to either the output pressure of the pump or to an external source of pressure.

Structurally, the pump of the present invention preferably includes a casing having a fixed cylinder block mounted therein to provide radially extending piston chambers. Each chamber accommodates reciprocation of a piston which is actuated by a centrally located, radially fixed rotatable shaft or hub journaled in the casing. The shaft is axially bored to receive a wedge member having an inclined face which is movable axially of the shaft under the influence of fluid pressures, the wedge member being operably connected to a radially shiftable rotatable eccentric means which is interposed between the cylinder block and the wedge means, the eccentric means being shifted radially by means of the inclined wedge surface upon axial movement of the wedge. The wedge is also preferably provided with oppositely directed inclined surfaces which control movement of counterweight means so that counterweight movement is effected in a direction opposite to movement of the eccentric means to thereby maintain a dynamically balanced condition of the pump assembly. In this manner the eccentric means may be radially shifted in response to fluid pressures which control axial movement of the wedge means, and coordinate control of the counterbalanced means insures a constantly balanced assembly throughout the entire range of piston displacement variation.

The control of fluid flow through the pump is effected by double-acting valve means reciprocably carried by the cylinder block and actuated by the shaft for valving movement in synchronization with radial movement of the pis-

2

tons, the valve means controlling ingress and egress of working fluid to the cylinder block piston chambers.

It is, therefore, an important object of the present invention to provide an improved radial piston pump having means for varying the displacement thereof.

Another important object of the present invention is to provide an improved radial piston pump of variable displacement, displacement variations being effected by actuation of a wedge means having inclined surfaces controlling the eccentricity of the piston reciprocation means.

It is a further important object of the present invention to provide a variable displacement radial multi-piston pump provided with eccentric means movable to vary pump displacement and coordinately movable counterweight means for maintaining a dynamically balanced condition within the pump.

Still another important object of the present invention is to provide a variable displacement radial piston pump provided with novel means for effecting a displacement variation by simultaneous actuation of radially movable eccentric means and counterbalance means, the displacement variation being accomplished without disturbing the dynamic balance of the pump.

A more specific object of the present invention is to provide a radial piston pump including a casing having a cylinder block mounted therein receiving radial pistons reciprocable in the block upon rotation of the radially movable eccentric means, radial movement of the eccentric means being accomplished by actuation of a wedge means having inclined surfaces operably connected to the wedge means.

An additional object of the present invention is to provide an improved variable displacement multi-piston radial pump provided with a rotatable shaft, a fixed cylinder block carrying reciprocable pistons movable in response to rotation of an eccentric means actuated by the shaft, double-acting valve means synchronized for valving action with rotation of the shaft, and means for shifting the radial position of the eccentric means relative to the cylinder block to vary the pump displacement.

Other and further important objects of this invention will be apparent from the disclosures in the specification and the accompanying drawings.

On the drawings:

Figure 1 is a radial sectional view, with parts shown in elevation, taken along a pair of parallel axially spaced planes of a radial piston pump of the present invention;

Figure 2 is a vertical sectional view taken along the planes II—II of Figure 1;

Figure 3 is a horizontal sectional view taken along the planes III—III of Figure 1;

Figure 4 is a fragmentary end elevational view of a pump of the present invention;

Figure 5 is a view taken along the plane V—V of Figure 4;

Figure 6 is a view taken along the plane VI—VI of Figure 4;

Figure 7 is a fragmentary sectional view, with parts shown in elevation, similar to Figure 3 and illustrating a double-acting valve element in neutral position; and

Figure 8 is an exploded fragmentary perspective view, partly in section, illustrating a portion of the piston actuating mechanism of a pump of the present invention.

As shown on the drawings:

Casing and shaft structure

The pump 10 includes a central, open-ended, generally cylindrical casing section or barrel 11. The barrel 11 has an inner generally frusto-conical bore 12 extending between a pair of spaced, annular, inwardly opening fluid conduits 13 and 14 providing means for accommodating the ingress and egress, respectively, of fluid to and from

3

the pump 10. The inlet passage 13 communicates with an inlet opening 15 provided in a generally radially extending embossment 16 formed integrally with the housing (Figure 2), and a similar embossment is provided for the discharge of fluid under pressure from the passage 14. One open end of the barrel 11 is closed by a front cover plate 17 having an annular shoulder 18 snugly bottomed within the corresponding open end of the barrel 11 to confine an annular sealing ring or O ring 19 therebetween. The front cover plate 17 is secured to the barrel 11 by suitable means, as by screws 20 threadedly retained by apertures 21 formed in the annular end faces of the barrel.

The other open end of the barrel 11 is closed by a rear cover plate 22 having a shoulder 18 similar to the shoulder 18 of the front plate 17 for confining a sealing ring 19 in fluid pressure-tight relation with the barrel, and the rear cover plate 22 is secured to the annular end face of the barrel by means of screws 20 received by threaded barrel apertures 21, as hereinbefore described in connection with the front plate 17.

The rear cover plate 22 is provided with an axially outwardly extending boss 23 having an internal shoulder 24 against which is bottomed a similar inner shoulder 25 formed in a cylinder control sleeve 26 which is interposed between the end plate boss 23 and a piston cover 27. The piston cover 27 and the sleeve 26 are each secured to the end plate 22 by suitable means, as by screws 28 threadedly retained by the boss and extending completely through the sleeve 26 as shown in Figures 2 and 3.

An annular seal ring 29 is interposed between the shoulders 24 and 25 of the boss 23 and the sleeve 26, respectively, and a similar seal ring 29 is interposed between corresponding shoulders 30 and 31 formed on the sleeve 26 and the cover 27, respectively. Thus, it will be seen that a sealed interior space 32 is provided in the interior of the barrel 12 for the reception of the pressure-generating portions of the pump 10. Annular shims 33 are interposed between each of the end plates 17-22 and the barrel 11, respectively.

A rotatable, axially elongated shaft 34 is provided to extend axially into the space 32 for rotation therein. The shaft 34 is disposed for rotation upon spaced bearing elements 35 and 36. More specifically, the bearing elements 35 and 36 are carried by the front cover plate 17 and the rear cover plate 22, respectively, the bearings being of the double-row thrust roller bearing variety and having their inner races press-fitted or otherwise secured to axially spaced portions of the shaft 34 and their outer races snugly fitting within annular spacing members 37 carried by the cover plates. The spacer members 37 are fixed against rotation by generally radially extending stop pins 38 press-fitted into registering apertures formed in the cover plate and the bearing member respectively. Thus, it will be seen that the shaft 34 is fixed by the thrust bearings 35 and 36 against axial movement, while free rotation of the shaft is accommodated by the same bearing members.

The shims 33 provide means for varying the spacing of each of the shaft thrust bearings 35-36 within the space 32. By varying the thickness of the shims, the shaft may be effectively moved axially within the barrel 11.

The shaft 34 is provided with an axial bore 39 which is open at each end; that end adjacent the front cover plate 17 being provided with an internal shoulder 40 against which is bottomed a circular seal ring 42 held against the shoulder 40 by means of a snap ring 43. That end of the shaft axially outwardly of the shoulder 40 is provided with internal splines 44 for the reception of a similarly splined driving member, such as a drive shaft of a motor or the like. The splined open end of the shaft 34 is peripherally toothed to threadedly receive a lock nut 45, and a similar lock nut 46 is carried by

4

the other threaded end of the shaft to thereby prevent unwarranted axial movement of the shaft. A spacer member 47 is interposed between the lock nut 46 and the adjacent bearing 36, the spacer member 47 being keyed to the shaft 34 for rotation therewith.

The shaft 34 is provided with a central eccentric peripheral surface 48 projecting radially beyond the periphery of the shaft, and adjacent shaft portions are longitudinally splined, as at 49, on either side of the eccentric surface 48.

Eccentric control structure

An eccentric control structure for the pump actuating mechanism is provided so that the displacement of the pump may be varied throughout a relatively wide range without upsetting the dynamic balancing of the pump structure as hereinbefore explained.

Initially, it will be noted that that end of the shaft bore 39 remote from the splined end 44 is open, and from Figure 1 it will also be noted that diametrically opposed portions 51 of the shaft 34 are flat, while the shaft bore 39 has a truly cylindrical inner peripheral surface. The open end of the shaft bore is provided with a radially enlarged counterbore 52 receiving an axially inwardly projecting, axially bored boss 53 formed on the sleeve 26. The sleeve 26 is fixed against rotation by the screws 28 joining the sleeve to the rear cover plate 22 and, correspondingly, the boss 53 is non-rotatable within the rotatable shaft.

An axially movable cylinder sleeve piston 54 is mounted within an enlarged inwardly extending counterbore 55 formed in the sleeve 26 for communication with the bore of the sleeve boss 53, the piston 54 having an outer peripheral flange 56 snugly fitting within the sleeve counterbore 55 for axial movement therein, and a piston ring 57 is confined by the flange against the inner peripheral surface of the counterbore 55. The piston 54 extends axially into the sleeve boss 53, and the piston terminates at its inner end in a radially inwardly extending shoulder 58 having an additional peripheral piston ring 59 snugly engaging the inner wall of the sleeve boss bore. A thrust-type ball bearing 60 is slidably journaled within the bore of the sleeve boss 53 to abut the inner end face of the boss 53, and a similar, oppositely directed thrust-type ball bearing 61 is confined within the piston 54 against the shoulder 58 thereof. The bore of the piston 54 is closed by a generally circular closure plate 62 bottomed against an internal shoulder formed in the piston by means of an annular lock ring 63 retained by the piston.

Journaled in the shaft bore 39 for axial reciprocation therein is a wedge member 65 which is generally cylindrical in contour and which snugly mates with the shaft bore. The wedge member 65 is provided with a reduced diameter axially extending shaft 66 journaled within the bearings 60 and 61 which have their outer races confined within the sleeve 26 and the piston 54, respectively. More particularly, the shaft 66 is provided with a first radial shoulder 67 bottomed against the thrust bearing 60 and a second reduced diameter radial shoulder 68 bottomed against the thrust bearing 61. The extreme end of the wedge shaft 66 is threaded for the reception of a lock nut 69 adjacent the thrust bearing 61.

It will thus be seen that the wedge member 65 is piloted at one end by the shaft bore 39 and journaled at its other end by the shaft 66 positioned within spaced thrust bearings 60 and 61 carried by the sleeve 26 and the piston 54, respectively, so that rotation of the eccentric member is accommodated.

Diametrically opposed peripheral portions of that section of the wedge member 65 extending into the shaft bore 39 are radially inwardly recessed in planes inclined with respect to the axis of the member 65 to define flat parallel wedge surfaces inclined axially and radially with respect to the shaft. More specifically, a plurality of axially spaced inclined surfaces are provided as best shown in Figures 2 and 3. For example, a central por-

tion of the wedge member 65 extending into the shaft bore 39 is provided with parallel inclined wedge surfaces 70, while oppositely directed inclined surfaces 71 are provided on each side of the surfaces 70. It will be seen that the surfaces 71 are of substantially the same angle of inclination as the surfaces 70, the surfaces 71 being oppositely directed to the surfaces 70 for a reason to be hereinafter more fully explained. Inasmuch as the surfaces are formed by milling or otherwise cutting or removing peripheral portions of the wedge member, the surfaces 70 and 71 are substantially flat and have a discrete width which increases as the inclined surfaces extend toward the center of rotation of the wedge member 65.

Intermediate portions of each of the inclined sections of the wedge member are provided with radial slots. For instance, the slot 72 is provided intermediate the inclined wedge surfaces 70 at the central portion of the wedge member, while similar slots 73 are provided to join each of the surfaces 71 on either side of the surfaces 70. Non-slotted portions of the wedge member serve to separate the slots from one another so that isolated individual slots are provided.

The slot 72 receives therethrough a wedge arm 74 which takes the form of a generally cylindrical rod or pin extending through and projecting radially beyond the wedge member 65. A slidable wedge block 75 is positioned on each side of the wedge member, the blocks each having a central bore 76 receiving the rod 74 therethrough, and each block is provided with an inclined inner face 77 mating with one of the inclined surfaces 70 and slidable therealong. The blocks 75 are each generally cylindrical in contour and the outer annular end face 78 of each block is flush with the corresponding end of the associated rod 74. The rod 74 and the blocks 75 project radially outwardly through apertures 64 formed in the shaft 34.

The end faces 78 of the blocks and the corresponding ends of the rods each contact the inner periphery of a generally cylindrical eccentric strap 79 having an eccentric outer peripheral surface 80 contacting the inner races 81 of a pair of axially spaced thrust-type roller bearings 82. The bearing races 81 are separated by annular spacer members 83, and similar outer races 84 of the bearings 82 are also separated by concentric members 83. The outer races 84 form an eccentric actuating peripheral surface for the pump pistons, as will be hereinafter explained in greater detail.

The eccentric strap 79 is provided with a central bore 86 receiving the shaft eccentric surface 48 therein. It will be noted that the bore 86 is provided with diametrically opposed flattened surfaces 87 which are in extended flush surface contact with corresponding flattened surfaces 51 of the shaft 34, so that the strap is rotatable with the shaft. In addition, the bore 86 is provided with diametrically opposed recesses 88 for receiving the outer cylindrical surfaces of the wedge blocks 75, thus insuring rotation of the blocks 75, the rods 74, and the slotted wedge member with the shaft and the eccentric strap 79.

The slots 73 on each axial side of the slot 72 receive therethrough a generally cylindrical rod 89 extending radially beyond the wedge member 65 through shaft apertures 64 aligned with the slots, as hereinbefore explained in connection with the rod 74. The rods 89 extend through wedge blocks 90 having inner surfaces 91 mating with the inclined wedge surfaces 71 and movable therealong as hereinbefore explained in connection with the blocks 78. The outer ends of the blocks 90 and the corresponding ends of the rods 89 contact annular relatively massive counterweights 92. The counterweights 92 are radially aligned with the slots 73 and these counterweights 92 are disposed on either side of the bearings 82 in axially spaced relation relative thereto. Annular spacer members 93 are interposed between the counterweights 92 and the races of the bearings 82, the spacer members 93 being splined to the shaft 34 for rotation therewith by means of the splines 49. It will be ap-

preciated that the shaft 34, the wedge member 65, the eccentric strap 79, and the counterweights 92 rotate as a unit in synchronism with rotation of the shaft as caused by a connection by means of the shaft splines 41 to a suitable source of power. Further, it will be seen that the oppositely directed surface 70 and surfaces 71 will cause concurrent oppositely directed movement in parallel radial planes of the counterweights 92 and the eccentric strap 79.

Movement of the wedge member 65 axially of the shaft is preferably accomplished by fluid pressure exerted upon the piston 54 to cause sliding movement of the piston within the counterbore 55 of the sleeve 26. As illustrated in Figures 4-6, inclusive, the cap 27, carried by the end plate 22, is provided with an internally threaded passage 95 which is joined, as at 96, to the counterbore 55 formed in the sleeve 26. A second passage 97, also internally threaded, is provided in the sleeve 26, this passage 97 communicating through a cross-passage 98 with a pressure chamber 99 formed by a circular end recess in the sleeve 26, or more particularly by a recess jointly defined by the sleeve 26 and the cover 27. It will be seen that fluid pressure in the counterbore 55 will be exerted upon the inner face 100 of the piston flange 56, while fluid pressure in the chamber 99 will be exerted upon the opposite face 101 of the piston flange and also upon the piston closure plate 62.

Thus, fluid pressure on the face 100 will urge the wedge member toward its extreme position illustrated in Figure 2, while fluid pressure on the face 101 will urge the wedge member in its opposite direction to change the relative radial positions of the eccentric strap 79 and the counterweights 92, thus changing the eccentricity of the pump actuating elements and therefore the pump displacement.

Actually, the area of the right-hand piston face, provided by the piston face 101 and the plate 62, is greater than the area of the left-hand piston face 100. This greater area is required since a greater total force is required to move the piston inwardly against the action of centrifugal force which resists inward movement of the counterweights 92 and the strap 79.

Cylinder block and piston structure

The pump operating structure which is actuated by rotation of the eccentric mechanism hereinbefore described includes a generally cylindrical cylinder block 105 best seen in Figures 1 and 2. More particularly, the cylinder block is seated within the barrel 13 by engagement of a frusto-conical external peripheral cylinder block surface 106 in mating engagement with the correspondingly tapered surface 12 of the barrel. The cylinder block 105 is held in seated relation by means of a spacing sleeve 107 interposed between the rear cover plate 22 and an adjacent side surface of a cylinder block.

The cylinder block 105 also provides radially extending, generally cylindrical piston chambers 108 (Figures 1 and 2) which are positioned axially of the pump 10 intermediate the inlet and outlet fluid flow passages 13 and 14 formed in the barrel. It will be seen that those portions of the cylinder block 105 defining the piston chambers 108 bridge the gap between the inlet and outlet passages. A plurality of radially extending piston chambers is thus provided by the cylinder block, and each chamber slidably receives a reciprocable piston 109 mounted therein for radial movement with respect to the shaft 34.

The pistons 109 are provided with reduced inwardly extending portions which are transversely apertured, as at 110, for the reception of piston pins 111. The opposing ends of the piston pins 111 are received within piston slipper blocks 112 (Figure 8). The slipper blocks 112 are provided in pairs for assembly adjacent each lateral edge of the apertured piston inward portion, the blocks being apertured as at 113 for reception of the pins and

the blocks each having an arcuate inner surface 114 mating with and slidable relative to the outer race of the bearings 82. Actually, sliding movement between the slippers and the outer race need not occur inasmuch as relative rotation between the eccentric strap 79 and the piston slipper blocks is accommodated by the thrust bearings 82 interposed therebetween. However, some sliding movement occurs between these members due to lesser spacing of the slipper blocks from one another as the piston approaches the pump center.

The outer ends of each of the slipper blocks are recessed, as at 115, for the reception of annular retaining rings 116 which are snugly interposed between the spacer rings 93 carried by the shaft 34 and the adjacent ends of the pins 111 and the slipper blocks 112. It will be seen that the retaining rings 116 prevent transverse or axial displacement of the pins 111 from their snugly fitted position within both the pistons 109 and the slipper blocks 112, and each of the pair of retaining rings 116 contacts each of the pins 111.

The cylinder block is also provided with means for establishing communication between the piston chambers 108 and the inlet and outlet grooves 13 and 14, respectively, formed in the barrel 11. This means is best illustrated in Figure 1 in which it will be seen that the cylinder block 105 is provided with an arcuate valving recess 120 adjacent to and communicating with each individual piston chamber 108. The valving recesses 120 for the various piston chambers 108 communicate individually with the piston chambers, and there is no communication between the peripherally adjacent valving recesses 120 or the adjacent piston chambers 108. The valving recesses 120 are joined by generally cylindrical cross-passages 121 with individual peripherally arcuately bottomed valving ports 122 which are radially aligned with the inlet and outlet passages 13 and 14, respectively. More specifically, a valving port 122 is positioned on each side of each valving recess 120, the valving recesses 122 for each valving recess being isolated from the other valving ports of the cylinder block, and each of the valving ports 122 is radially aligned with its corresponding barrel fluid passage.

Thus, fluid communication from a source of fluid takes place through the fluid inlet 15 to the fluid inlet passage 13 formed in the barrel and then to the valving ports 122 aligned with the inlet passage 13. From the valving ports 122 fluid passes through the cross-passage 121 into the valving recesses 120 and thence into the piston chambers 108 associated with the respective valving passages. A similar outlet fluid flow may be traced from the piston chamber 108 through the valving recess 120 associated therewith, then through the cross-passage 121 to the other of the associated valving ports 122 and thence to the fluid pressure discharge passage 14 provided by the barrel for flow to the pump outlet.

Valving structure

It will be seen that in order for effective pumping action to be obtained, some control over fluid flow through the cross-passages 121 must be obtained. In this manner, controlled communication between the valving recesses 120 and the inlet and outlet passages 13 and 14, respectively, is effected, and upon correlation of the valving action with rotation of the pump eccentric driving means, a desirable synchronized valving action is obtained.

Specifically, the valving structure of the present invention includes dumbbell-shaped double-acting valve elements 125, one valving element 125 being provided for each piston 109. The valving elements 125 each includes a central reduced diameter shank portion 126 joining a pair of radially enlarged generally cylindrical valving heads 127 journaled in the cross-passages 121. The valving heads are provided with axially extending projections 128 which extend axially beyond the cylinder block and which are provided with round terminal por-

tions 129. It will be seen that the length of the shank 126 separating the heads 127 is substantially equal to the distance between the inlet and outlet valving ports 122 so that when the valving structure 125 is centered (Figure 7), the heads are effective to close both of the ports 122 and no communication whatsoever between the inlet and outlet passages and the piston chambers is possible.

The radial alignment of the valve heads 127 with the ports 122 to effect exact zero registry, as shown in Figure 7, is assured by means of the shims 33. Any variance in registry can be adjusted by increasing the shims 33 on one side of the pump and decreasing the shim thickness on the opposing side. Thus, an axial shifting of the shaft 34 and the pressure generating elements carried thereby, including the valving elements 125, can be effected.

The actuating means for the valving elements includes a pair of valving plates 130 having central apertures received by and splined to the splined portions 49 of the shaft so that the plates 130 are rotatable with the shaft. The plates 130 are provided with offset radially extending outer peripheral flanges 131 which are inclined with respect to a radial plane on the shaft 34 (Figure 2). The valve plate flanges 131 thus defines a continuous, annular camming surface which, being inclined with respect to the radial plane of the shaft, goes through an axial movement of "wobble plate" action during each rotation of the shaft 34. This wobbling action of the valve plate flanges 131 is utilized to control actuation of the valving elements 125. It will be seen that a pair of the plates 130 are provided having parallel in-phase camming flanges 131 so that a constant distance is maintained between the flanges 131 of the plates 130 during all rotating positions of the same.

The rounded terminal portions 129 of the valving elements 125 are received by hemispherical recesses formed in valve slippers 132, that face of the slipper opposite the hemispherical seating face contacting the valve plate flanges 131, and it will be seen that upon rotation of the shaft a corresponding rotation of the plates 130 is obtained with the resulting wobble action heretofore described, this wobbling action causing a complete reciprocation of each of the valving elements 125 through its entire range of operation, including the extreme positions illustrated in Figure 3. In Figure 3 it will be seen that this wobbling action establishes communication between the inlet passages 13 and the piston chambers and between the piston chambers and the outlet passage 14, and the resultant valving action is synchronized with rotation of the shaft by virtue of the splined shaft-plate construction.

Operation

The operation of the variable displacement pump of the present invention as hereinbefore explained will be readily evident to those skilled in the art, and the following operational summary may be best understood by reference to the foregoing structural description.

Initially, assuming the pump to be in its operational condition as illustrated in the drawings, it will be seen that rotation of the shaft 34 effects simultaneous rotation of the eccentric strap 79. Rotation of the strap is effected through the flattened side surfaces 87 thereof in extended engagement with the corresponding surfaces 51 of the shaft 34. Rotation of the wedge member 65 is effected by virtue of the strap pin 74 and the associated blocks 75 projecting through the central radial shaft aperture 64. Simultaneous rotation of the counterweights 92 is effected by means of the counterweight pins 89 also carried by the wedge member 65 and extending through the additional radial bores 64 of the shaft. Also, the counterweights 92 are in extended side surface contact with the valve plates 130 and the spacer plates 93 splined to the shaft 34.

Rotation of the eccentric strap 79 accomplishes radial

reciprocation of the pistons 109 through the medium of the bearings 82, the piston slippers 112 and the piston pins 111.

Concurrent valving action is also obtained inasmuch as rotation of the shaft causes concurrent rotation of the valving plates 130 splined thereto with the camming flanges 131 of the valving plates effecting reciprocation of the double-acting valve elements 125. By virtue of the reciprocation of the valving elements, communication of the piston chambers 108 with the inlet and outlet passages 13 and 14, respectively, of the barrel is accomplished as hereinbefore explained.

To vary the displacement of the pump, it is only necessary to shift the center of rotation of the eccentric strap 79. This may be easily accomplished by axial shifting movement of the wedge member 65. Such movement of the wedge member is accomplished by the utilization of fluid pressure exerted upon the piston 56 as hereinbefore explained, namely, by the admission of fluid pressure either into the counterbore 55 of the sleeve 26 or into the chamber 99 confined between the piston 56 and the end cap 27. The fluid pressure may be supplied from either the outlet pressure of the pump 10 or from an external source of fluid pressure actuated by either manual or automatic valve means. In case the outlet pressure of the pump 10 is utilized as a control factor, one of the ports 95 and 97 may be connected to the outlet pressure and the other port connected to a control pressure, so that the balancing of these pressures on either side of the piston 56 will determine the axial positioning of the wedge member 65, so that a constant delivery pressure is obtained.

Another novel operational feature of the present invention is the utilization of counterweights 92 which are radially movable in a direction opposite to the direction of movement of the eccentric strap 79, so that the device is in constant dynamic balance about the center of rotation, thus avoiding wear and tear on the pump and further avoiding any unevenness or roughness of running, even during those periods in which the eccentricity is being shifted.

It will, of course, be appreciated that the pump 10 may also be used as a motor by merely reversing the path of fluid flow through the device and the utilization of the shaft 34 as a source of power.

It will be understood that modifications and variations may be effected without departing from the scope of the novel concepts of the present invention.

I claim as my invention:

1. In a radial piston type pump apparatus, a wedge having a first inclined central portion and second oppositely inclined portions disposed on each side of said first portion, a cylinder block surrounding said wedge first portion and having radially extending ported piston chambers, radially shiftable rotatable eccentric means interposed between said cylinder block and said wedge first portion, radially shiftable massive counterweights, pistons reciprocable in said chambers upon rotation of said eccentric means, means slidable along the wedge first portion and projecting therefrom into contact with said shiftable eccentric means to radially shift the same upon axial movement of said wedge first surface, and means slidable along the wedge second portions and projecting therefrom into contact with said counterweights for radially shifting the same upon axial movement of the wedge, the radial movements of said eccentric means and said counterweights being concurrent and oppositely directed to maintain a dynamically balanced pump structure despite variations in the displacement thereof.

2. In a radial piston pump, a casing and a cylinder block mounted therein having an annular row of radially extending piston chambers receiving therein reciprocable pistons, a movable wedge projecting into said cylinder

block and having two separate camming surfaces, a radially shiftable and rotatable eccentric means interposed between said wedge and said block for reciprocating said pistons, a radially shiftable and rotatable counterweight means also interposed between said wedge and said block, means contacting one camming surface of said wedge and said eccentric means and arranged for shifting the radial position of the eccentric means, and means contacting the other camming surface of said wedge and said counterweight means and arranged for concurrently shifting the radial position of the counterweight means in a direction opposite to the shifting of the eccentric means upon movement of said wedge member to preserve dynamic balancing of the assembly despite a variation in the eccentricity of the eccentric means.

3. In a radial piston pump, a casing and a cylinder block mounted therein having radially extending piston chambers receiving reciprocable pistons therein, a wedge member projecting into said cylinder block and movable relative thereto, said wedge member having a central wedging surface inclined with respect to a radial plane of the cylinder block and oppositely inclined wedging surfaces on either side of said central wedging surface, a radially shiftable and rotatable eccentric means interposed between said wedge member and said block for reciprocating said pistons, said eccentric means being radially aligned with the first wedging surface, axially spaced massive counterweight means adjacent said eccentric means and radially aligned with said oppositely inclined surfaces of the wedge, means contacting said central wedging surface and said eccentric means for shifting the radial position of the eccentric means upon movement of the wedge relative thereto, and means contacting said oppositely inclined surfaces and said counterweight means for concurrently shifting the radial position of said counterweights in a direction opposite to radial shifting of said eccentric means upon said movement of said wedge to preserve dynamic balancing of the assembly despite the variation of the eccentricity of said eccentric means.

4. In a radial piston type pump, the improvement of a hollow shaft, a wedging member in said shaft and being co-rotatable therewith, said wedging member having a central inclined wedging surface and two additional inclined wedging surfaces on axially opposite sides of said central inclined wedging surface, means providing an annular row of ported cylinders around said shaft, a piston in each cylinder, wedging means extending through said shaft engaging each piston and said central wedging surface, means providing annular counterweight guideways adjacent each side of said cylinders, counterweight means in each guideway, wedging means extending through said shaft and engaging each counterweight means and the respective additional wedging surfaces, and means to selectively axially shift said wedging member in said hollow shaft relative to said wedging means to concurrently shift the pistons and the counterweight means in opposite radial directions.

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