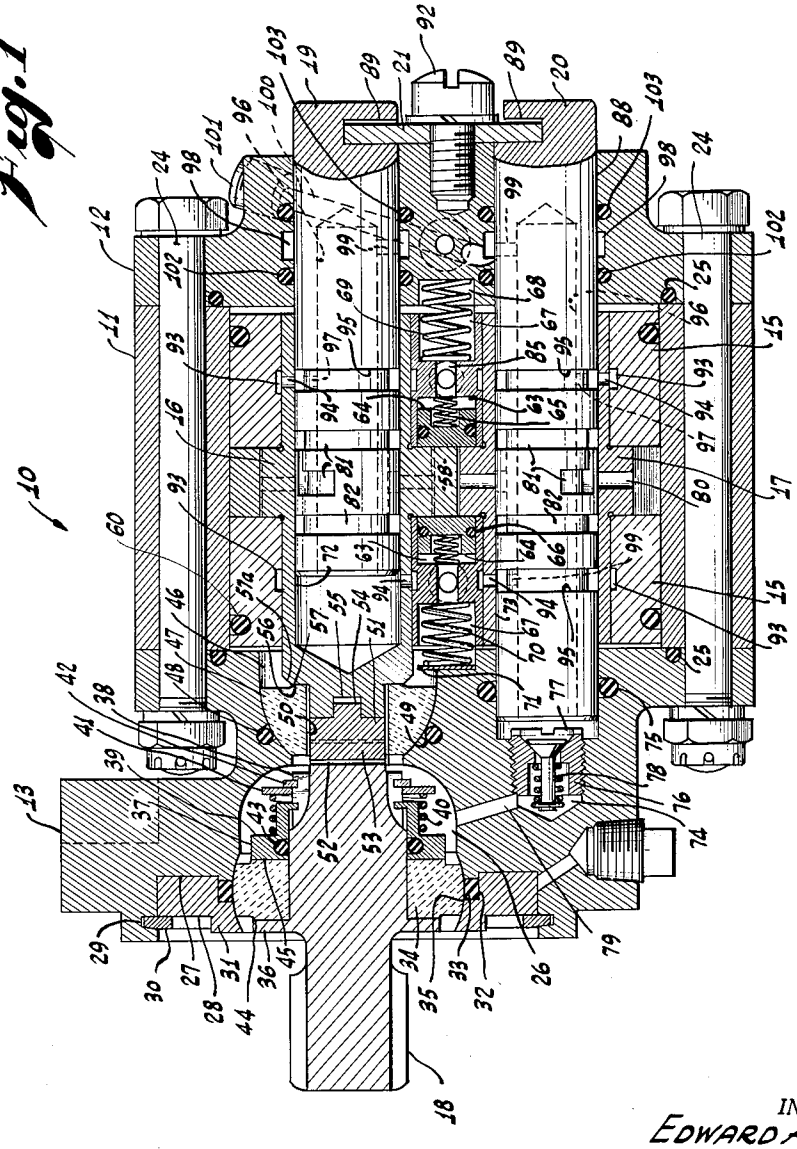
Feb. 20, 1962 E. A. ROCKWELL 3,021,791

ROTARY FLUID DISPLACEMENT DEVICE AND MECHANISM THEREFOR

Original Filed Aug. 24, 1950 4 Sheets-Sheet 1

Fig. 1



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Fig. 2

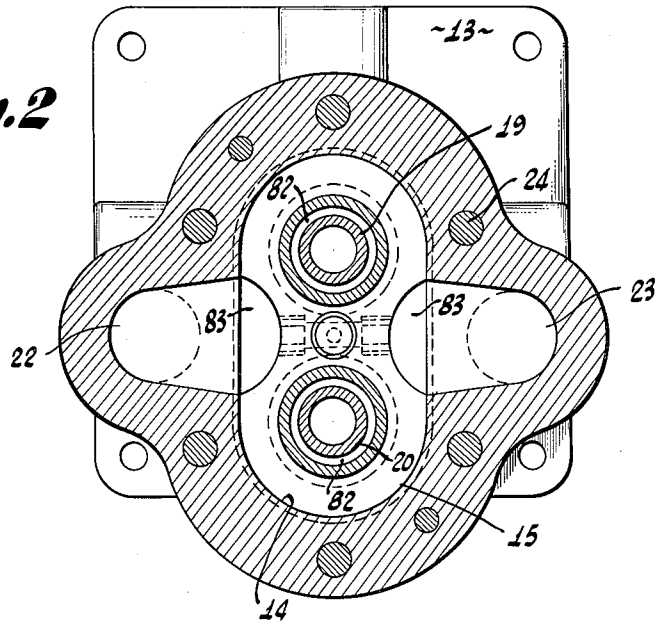
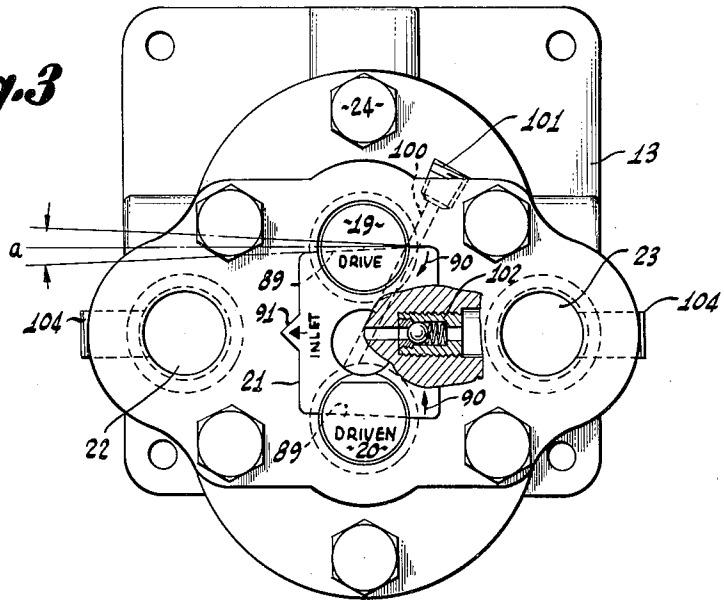


Fig. 3



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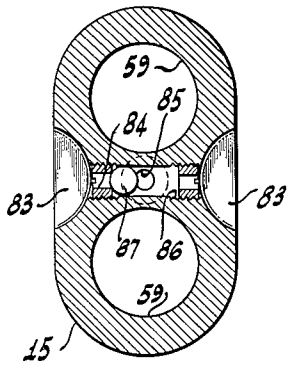


Fig. 4

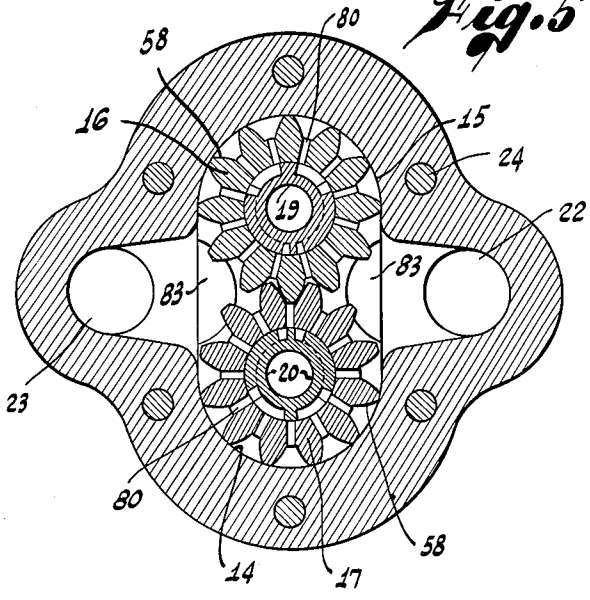


Fig. 5

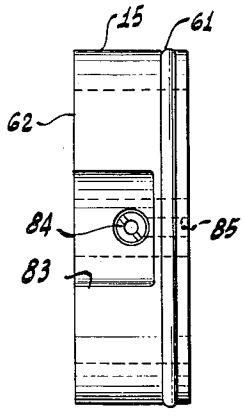


Fig. 6

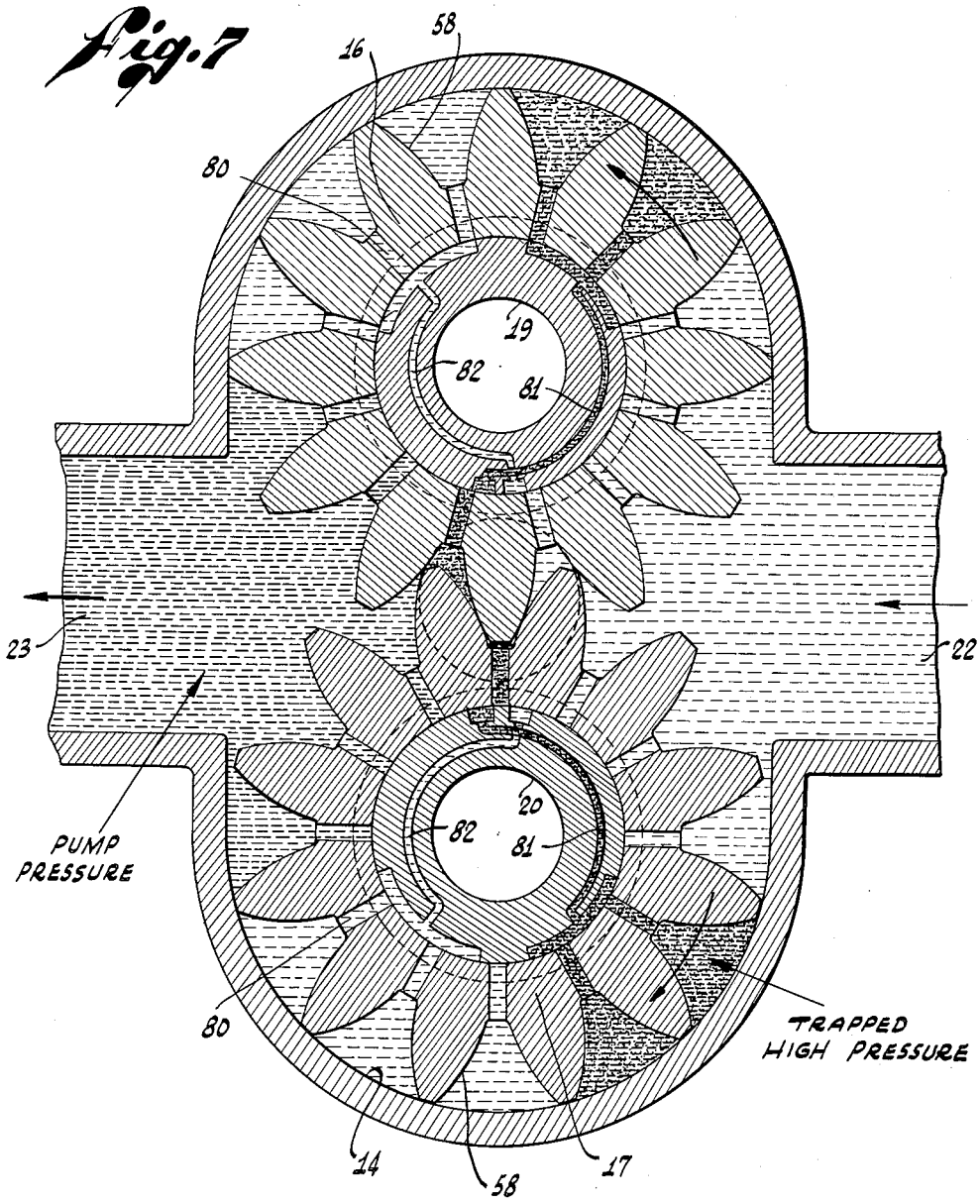
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Fig. 7



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United States Patent Office 3,021,791 Patented Feb. 20, 1962

3,021,791

ROTARY FLUID DISPLACEMENT DEVICE AND MECHANISM THEREFOR

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Original application Aug. 24, 1950, Ser. No. 181,203, now Patent No. 2,758,548, dated Aug. 14, 1956. Divided and this application Aug. 9, 1956, Ser. No. 603,041

9 Claims. (Cl. 103—126)

This is a divisional application continued from my copending application Serial No. 181,203, filed August 24, 1950, which has issued as Patent No. 2,758,548 entitled Rotary Fluid Displacement Device and Mechanism therefor.

The present invention relates to rotary fluid displacement devices and particularly to a reversible pump or the like with self-aligning pressure seals for its driving gear shaft.

An object of my invention is to provide a reversible gear pump having pressure balancing means that can be adapted for a change in direction of the pump by control means on the pump exterior.

Another object of the invention is to provide a pressure balanced reversible gear pump in which the gears are ported for sequential valving registration with passages formed in valve plugs on which the gears are mounted. The valving means thus defined form part of the means for balancing the pressures on the gears and these means are synchronized with the direction of the pump by providing valve plugs that can be slightly rotated to adapt the pump for a change of direction.

It is a further object of the invention to provide a reversible pump of this character in which a key plate controls the positions of the valve plugs with reference to the direction of the pump, the key plate being adapted for registration with the plugs in such a way as to prevent any possibility of improper synchronization of the pressure balancing means of the pump with its flow direction.

It is also an object of my invention to provide a self-aligning high pressure seal for the driving gear shaft of a pump that has means for balancing the axially directed forces on the shaft associated therewith so that minimum loading pressures against the seal are maintained at all times.

A still further object of my invention is in a self-aligning secondary low pressure seal having a sealing face acting on a flat surface exterior of the pump body to effectively seal any negative pressure on the inlet side of the pump.

A further object of the invention is to provide a reversible pump in which the seepage of fluid therethrough is automatically diverted to the low pressure side of the pump.

These and other objects and advantages of the invention will be apparent from the following description of a presently preferred embodiment thereof and from the drawings annexed hereto in which:

FIGURE 1 is a longitudinal section of a gear pump embodying my invention, taken through the axes of the gears;

FIGURE 2 is a transverse sectional view taken substantially on the line 2—2 of FIGURE 1;

FIGURE 3 is an end view taken in the direction 3—3 of FIGURE 1 and partially cut away to show one of the check valves for automatically sealing off the high pressure side of the pump from the seepage leakdown circuit of the device;

FIGURE 4 is a transverse sectional view taken on the line 4—4 of FIGURE 1 showing one of the two bushings for supporting the gears, each of which is provided with

shutttle valve means for automatically communicating with the port of the pump carrying fluid at outlet pressure;

FIGURE 5 is a transverse sectional view taken along the line 5—5 of FIGURE 1;

FIGURE 6 is a side elevation of the bushing shown in FIGURE 4; and

FIGURE 7 is an enlarged schematic view similar to FIGURE 5.

Referring to the drawings for the general arrangement of the invention, and in particular to FIGURE 1, the pump will be seen to have a housing, designated generally by the numeral 10, comprising a body member 11 closed at one end by a cover 12 and at the other end by a pump mounting member 13. A chamber 14 is thus defined in housing 10 having a configuration such as is seen in transverse section in FIGURES 2, 5, and 7, and generally resembling a pair of confronting semi-cylindrical cavities tangentially joined by opposite side walls. Slidably disposed in chamber 10 are two bushings 15 such as are shown in FIGURES 4 and 6 wherein it can be seen that the configuration of bushings 15 is complementary to that of chamber 14. A drive gear member 16 and a driven gear member 17 are rotatably supported in bushings 15, power being supplied to member 16 through a drive shaft 18 that is rotatably mounted in pump mounting member 13. It will be noted that the two gear members 16 and 17 are internally bored to receive plugs 19 and 20, respectively. These plugs are mounted at their outer ends in the pump cover 12 and are immovably held in selected position by a key plate 21 which is best seen in FIGURE 3. Extending longitudinally of housing 10 and on opposite sides of the common axial plane passing through the axes of rotation of the gears 16 and 17 are inlet passage 22 and outlet passage 23 which can be seen in FIGURES 2, 3, and 5. As will appear from the following detailed description of the device, the pump includes pressure balancing means that can be synchronized with the direction of fluid therethrough by means of key plate 21 and as will also appear a highly effective fluid seal is provided between the driving gear member and its drive shaft.

Specifically, the pump body 11, cover 12, and mounting member 13 are held in assembled relationship by a plurality of longitudinally extending bolts 24, suitably provided with nuts. If desired, a pair of dowel pins, such as are shown in FIGURE 2, may be provided to aid in accurately aligning the parts of housing 10. A fluid sealed housing assembly is provided by a pair of circumferential seals 25 retained in grooves formed on the inner faces of cover 12 and member 13. Mounting member 13 has a suitable flange formed integrally therewith so that the pump can be mounted to a power source for driving connection with the splined drive shaft 18.

Member 13 is formed with a cavity 26 opening into its outer face that is counterbored as at 27, thus providing a seat for a retaining collar 28. An annular groove 29 in the counterbore wall wedgingly receives the tapered edge of a snap ring 30 whereby the collar 28 is axially immovably held in place. At its axially outermost end the collar 28 has a radially inwardly extending lip 31 that defines an annular groove 32 with the shoulder of counterbore 27. An elastic seal ring 33 is disposed in grove 32 to effect a fluid seal with the radially outward wall of a bushing 24. The bushing 34 is preferably of a self-lubricating material such as the carbon-graphite compounds. It should particularly be noted that the radially outwardly facing wall of bushing 34 and the contacting surfaces of collar 28 and cavity 26 are complementarily arcuate as at 35 so that the bushing 34 is self-aligning.

The drive shaft 18 is rotatably mounted in bushing 34 and is formed with a continuous flange 36 abutting the

exterior end face of the bushing. Abutting the interior end of bushing 34 is a sealing collar 37 that is keyed to the inner end of shaft 18, as at 38. This collar 37 has an inwardly extending portion of reduced diameter to define an annular shoulder 39 against which an end of a compression spring 40 bears. The other end of spring 40 bears against an annular spring retainer 41 that is held by a snap ring 42 mounted in a circumferentially extending groove formed in the inner end of the shaft 18.

Leakage of fluid between shaft 18 and sealing collar 37 is prevented by an elastic seal ring 43 disposed in an annular groove formed in the bore of the collar. It should be observed that further sealing means are provided by virtue of the construction just described. The spring 40 causes the bushing 34 to be tightly embraced between drive shaft flange 36 and collar 37, in effect providing a seal 44 at the outer end of the bushing and a seal 45 at the inner end of the bushing. As will later appear, the cavity 26 receives seepage fluid at low pressure and it can readily be appreciated that the drive shaft mounting means just described effectively seals off such fluid from the exterior of the pump.

Referring to FIGURE 1, it will be seen that the cavity 26 of mounting member 13 develops into a socket-like depression 46 that opens into the inner end of the member 13. The depression 46 is spherically concave to matingly receive the rounded outer surface of a self-aligning sealing member 47. An elastic seal ring 48 is disposed in an annular groove 49 formed in the depression 46 to provide a fluid seal with member 47. An axial bore 50 extends through member 47 that is adapted to receive the inner end of drive shaft 18.

Drive gear 16 is connected to drive shaft 18 by a self-aligning coupling member 51. Shaft 18 at its inner end has a transverse slot 52 that is adapted to receive a tongue 53 formed on the adjacent end of member 51. At its other end the member 51 has a tongue 54 90° apart from the tongue 53 that is adapted to engage a slot 55 formed in the adjacent end of drive gear 16. It will be noted that clearance space is provided between the coupling means just described and the bore 50 of member 47.

The inner end of sealing member 47 has a circular raised flat surface 56 with a high polish against which a similarly polished shoulder 57 of drive gear 16 bears. The aforementioned clearance space in bore 50 of sealing member 47 permits the surface 56 and shoulder 57 to bear squarely against one another, thus providing a self aligning high pressure seal. A chamfer 57*a* is formed on shoulder 57 whose annular area approaches, but does not exceed the annular area of the opposite end surface of the shaft of drive gear 16. With this arrangement, axially directed fluid pressure forces on the drive gear shaft are counterbalanced so that minimum loading pressures on the high pressure seal are maintained. As will appear, spring pressure and residual fluid pressure maintain shoulder 57 in running and sealing engagement with surface 56 of member 47.

Both of the gears 16 and 17 have teeth 58 integrally formed thereon. The teeth 58 are preferably machined to involute tooth form having a suitable pressure angle such as 28 degrees. Referring to FIGURES 5 and 7, it will be seen that this tooth form provides a trapping effect or secondary pumping means in the meshing area of the gears for a purpose which will later be set forth. The gears 16 and 17 are rotatably supported in the two identical bushings 15 (FIGS. 4 and 6). Each bushing 15 has a pair of bores 59 adapted to receive a shaft portion of one of the gears in a close running fit. Each bushing 15 also has a peripheral groove 60 formed therein that receives a seal ring 61 for sealing engagement with the interior surface of pump body 11. The inner end faces 62 of the bushings are perfectly flat, the teeth 58 of the gears 16 and 17 being adapted to run in the space therebetween.

Opening into the inner end faces 62 of the bushings 15 at the meshing point of the gears 16 and 17 each of the bushings is formed with a cylindrical cavity indicated at 63. Each of the cavities 63 is adapted to receive a sealing plunger 64. Light compression springs 65 are interposed in the cavities 63 behind plungers 64 that are adapted to at all times urge the plungers into contact with the end faces of the gears 16 and 17 at the meshing point of the gears. It will be observed that each of the plungers 64 carries a seal ring 66 to define a fluid seal with the wall of the cavity 63. As has already been pointed out, the gears 16 and 17 at the meshing point provide a secondary pumping means which produce very high pressure entrapped fluid. It is the purpose of the seal rings 66 to prevent any seepage of the entrapped fluid into these cavities 63.

Referring again to FIGURE 1, it will be seen that each of the bushings 15 on its outer end face is provided with a cavity 67 directly opposite to but separate from the cavities 63. The inner face of the cover 12 at a point directly opposite to the cavity 67 in the right hand bushing 15 is formed with a blind hole 68 in which a relatively heavy compression spring 69 is seated. It will be seen that there is a certain small clearance at both ends of the pair of bushings 15 between the bushings and the interior end faces of the cover 12 and the pump mounting member 13. Accordingly, the bushings 15 are slidable within this clearance area. The heavy spring 69 is of sufficient force to urge all of the parts enclosed in the pump chamber 14 to the position shown in FIGURE 1 overcoming the friction of the seals 61. The left hand bushing 15 is provided with a lighter spring 70 held in place in cavity 67 by a retainer plate 71. Spring 70 is only of sufficient force to hold the left hand bushing 15 normally against the left end faces of the gears 16 and 17.

It will be observed that the drive gear 16 is formed with an axial bore 72 that is blind at the left hand end. The driven gear 17 is formed with an axial bore 73 that extends from end to end. The drive gear 16 is adapted to slidably receive the plug 19 and the driven gear 17 adapted to receive the longer plug 20. As shown in FIGURE 1, the longer plug 20 extends through gear 17 into a counterbored hole 74 formed in the inner end face of the pump mounting member 13. The hole 74 is formed with an annular groove which receives a seal ring 75 to prevent any seepage of the liquid therethrough. A valve seat plug 76 is threaded into the reduced diameter portion of the hole 74 to carry a check valve 77 that is provided with compression spring means 78 on its stem to close the valve. A passage 79 extends between the cavity 26 of mounting member 13 and check valve 77. The structure just described provides a low pressure return from the cavity 26 between the low pressure seal means disposed in said cavity and the high pressure seal defined at the shoulder 57 of driving gear 16. When the accumulation of pressure in the cavity 26 exceeds the spring loading on the check valve 77 and the pressure of fluid in hole 74, the valve will open into the hole 74. At this point the fluid is returned to the low pressure side of the pump by means to be described later.

It will be observed that since the bushings 15 are slidable relative to the gears 16 and 17, there is a variable clearance space therebetween in which the gears rotate. In a fixed clearance pump, as a result of the wear on the end faces of the gears, excess by-passage of fluid along the end faces of the gears occurs, greatly reducing the volumetric efficiency of the pump. In the present invention means are provided utilizing fluid at the outlet pressure of the pump to force the bushings 15 against the end faces of the gears, whereby excessive by-passage of fluid therebetween is prevented. The greater volumetric efficiency thus achieved does not increase friction nor lower the torque efficiency of the pump, because the pressures urging the bushings 15 against the end faces of gears 16 and 17 are counterbalanced by opposing

higher pressures acting on the inner end faces 62 of the bushings. The bushings are thus floated away from the gears, and wear on them is greately reduced. This action takes place regardless of the direction of rotation of the gears, the bushings 15 being provided with suitably valved passages for this purpose, as will presently appear.

Referring to FIGURES 4 and 6, it will be seen that each of the bushings 15 is formed with two oppositely disposed semi-circular cavities 83 that open into the inner end face 62 of the bushing. In FIGURE 5, a bushing 15 is shown in its assembled position, and it will be noted that one of the cavities 83 communicates with the inlet 22 of the pump and the other cavity 83 communicates with the pump outlet 23. A valve passage 86 extends between the two cavities 83. Both ends of valve passage 86 have a screw fitted valve seat 84 adapted to be closed by a ball check valve 87 in the passage 86. An axially drilled passageway 85 communicates with the valve passage 86.

In FIGURE 5, the drive gear 16 is being turned in a counter-clockwise direction to draw fluid from inlet passage 22 to discharge it at higher pressure into outlet passage 23. The cavities 83 of the two bushings 15 that are in communication with outlet passage 23 also receive high pressure fluid. The ball check valves 87 are thus forced to close the low pressure side of valve passage 86. High pressure fluid is then admitted through passageways 85 to the outer faces of both bushings (FIG. 1) to press them inwardly against the end faces of gears 16 and 17. As is apparent, when the pump is reversed so that passage 23 becomes the inlet and passage 22 the outlet, the ball check valves 87 will close to passage 23 to achieve the same result of applying fluid pressures to the outer faces of the bushings 15.

Counterbalancing pressures are applied to the inner faces 62 of bushings 15 by porting and valving means defined in the gear members 16 and 17 and the plugs 19 and 20. As has already been noted, the gears 16 and 17 are adapted to entrap fluid in the meshing area thereof in the manner shown in FIGURE 7. The very high pressure of this entrapped fluid is utilized as the aforesaid counterbalancing pressure. The entrapped fluid is further used to counterbalance the gears against the reactive force of the pump outlet pressure. It is also desirable to balance the gears against the low pressure side of the pump and, therefore, the porting and valving arrangement of my invention is adapted for this purpose as well. By adjustment of the key plate 21 the valving action can be synchronized with the direction of the pump.

Referring now to FIGURES 5 and 7, it will be seen that both gear members 16 and 17 have a plurality of radial ports 80, each of which opens into the inter-tooth space between a pair of teeth 58. Each of the plugs 19 and 20 has a circumferential channel groove 81 with offset end portions that align with gear ports 80 (FIG. 1). The grooves 81 serve as a high pressure balance by intermittently communicating the previously mentioned very high pressure entrapped fluid to selected inter-tooth spaces between gear teeth 58. This arrangement is shown diagrammatically in FIGURE 7 by the dark shading, wherein it will be noted that the selected spaces are on the low pressure side of the pump and thus oppose the force on the gears from the outlet pressure of the pump. I have estimated that pressure of the entrapped fluid at three times the pump outlet pressure. It will be noticed from FIGURE 7 that this very high pressure is distributed to two, or for a short period, three selected inter-tooth spaces. Distributed in this way, this very high pressure is sufficient not only for the purpose of keeping the gear teeth 58 from being driven against the inner walls of pump body 11, but also for the purpose of counterbalancing the fluid pressure on the outer end faces of bushings 15. As the bushings are separated, a reduction in the trapped high pressure will occur due to the leakdown to lower pressure areas. In this connection it will be noted that the two pressure loaded sealing plungers 64 are also pressure counterbalanced and permit leakdown of very high pressure fluid to the low pressure side of the pump.

Each of the plugs 19 and 20 also has a circumferential channel groove 82 with offset end portions aligned with gear ports 80 (FIGURE 1). The grooves 82 are oppositely disposed to grooves 81 and serve as a low pressure balancing system indicated in light shading in FIGURE 7. By this arrangement selected inter-tooth spaces on the high pressure side of the pump are intermittently communicated to the low pressure side of the pump.

As examination of FIGURES 5 and 7 will show, in order to utilize the very high pressure generated by the entrapment of fluid at the meshing area of the gears, the entrance to the high pressure channel 81 must be accurately located so as to align with a certain gear port 80 during the interval of entrapment. This alignment, or valving action, should occur at a point on the high pressure side of a line drawn between the axes of the gears 16 and 17 in close proximity to the line. The valving action for the low pressure channel 82 should occur on the low pressure side of the line at a point more remote from said line than the high pressure valve action. A certain gear port 80 will then be open to low pressure from inlet 22, rather than in communication with an enclosed inter-tooth space. With this arrangement, if the direction of gear rotation is reversed without changing the valve timing, the desired pressure balancing will not take place; but if plug 19 is rotated through a small arc in a clockwise direction, and plug 20 rotated through the same arc in a counter-clockwise direction, the valve events will be properly synchronized with the pump direction. The entrances to the channels 82 will then be in a position to align with gear ports 80 containing entrapped fluid. The entrances to channels 81 will be in position to align with gear ports 80 that open to passage 23, which now serves as the inlet of the pump.

Referring now to FIGURE 1, it will be seen that the plugs 19 and 20 protrude outwardly from the pump housing through a pair of bores 88 in the cover 12. Each of plugs 19 and 20 in the outwardly protruding portion thereof is provided with a key-way 89 that is adapted to receive the key plate 21 in order to control reversing of the pump. Key plate 21 is shown in FIGURE 3 to have a pair of arrow markings 90 indicating direction of gear rotation and an arrow projection 91 indicating the inlet side of the pump. Duplicate inlet and outlet markings and rotation arrows are marked on the opposite side of the key plate 21. It will be observed that the key plate 21 is slightly wedge shaped, i.e., the upper and lower edges thereof are tapered to define a slight angle "*a*" on both of said edges. A retainer screw 92 holds the key plate to the cover 12, the screw being located in the common axial plane passing through the axes of the plugs 19 and 20 but being slightly offset vertically. It is apparent that this arrangement is such as to make it impossible to connect the plate 21 to the plugs by simply reversing the plate without turning it over. The previously described high pressure balancing and low pressure balancing means can be correctly related to the direction of rotation of the gears merely by rotating the plugs 19 and 20 through the slight angle shown. In FIGURE 3 this angle is roughly 2 degrees. When key plate 21 is reversed in the manner just described from the position shown in FIGURE 3, plug 19 and the plug 20 will be rotated through a total of approximately 4 degrees each.

It has already been remarked that seepage fluid accumulating in the cavity 26 is relieved through the check valve 77. As an examination of FIGURE 1 will show, seepage of high pressure fluid will also occur at various other points in the pump. Accordingly, my invention includes means to relieve the device of this seeped down

fluid and conduct it automatically to one of the passages 22 and 23. If the passage 22 is the inlet side of the pump for the moment, the seeped down fluid will automatically be returned to the passage 22.

As is shown in FIGURE 1, each of the bores 59 in the bushings 15 is formed with an annular groove 93. The gear members 16 and 17 are formed with radial bores 94 adapted to register with one of the grooves 93. The plug 19 is formed with a single circumferential groove 95 that registers with the right hand bore 94 of the gear 16. The plug 19 also has an axial bore 96 that is open at the left hand end of the plug. A radial orifice 97 communicates the bore 96 and the groove 95 of the plug 19. With this arrangement high pressure fluid will seep down towards the bushing groove 93 and then into the axal bore 96 of the plug 19 via the groove 95 and the orifice 97. The left hand bore 94 of the gear 16 communicates directly with the open end of the plug 19.

A somewhat similar arrangement is shown for the gear 17 except that the plug 20 is formed with a pair of grooves 95 because of its greater length. The grooves 95 register with a pair of radial bores 94 formed in the gear 17. Fluid seeping into the bushing grooves 93 is conducted into the axial bore 96 of the plug 20 via the orifices 97 and grooves 95 of the plug 20.

Both of the bores 88 in the cover 12 are formed with annular grooves 98. The grooves 98 register with radial bores 99 formed in each of the plugs 19 and 20. The radial bores 99 communicate with the axial bores 96 of the plugs to conduct seepage fluid therefrom into the grooves 98.

The cover 12 has a diagonally drilled passage 100 therein that intercommunicates the upper and lower grooves 98. The passage 100 can be seen in FIGURES 3 and 1, wherein it will be noted that the passage is sealed by a pipe plug 101. The cover 12 has two return check valves, one of which is indicated at 102 in FIGURE 3. The check valve to the inlet 22 is not shown; however, it is located symetrically and in precisely the same manner as the one shown in FIGURE 3. Both of these valves 102 communicate with the diagonally drilled passage 100. These spring pressed valves retain a slight residual pressure leading to the inlet, but will check off any high pressure acting on the valves from the outlet. It will be noted that each of the bores 88 in the cover 12 is adapted to receive a pair of seal rings 102, 103 disposed on opposite sides of the annular grooves 98.

In the operation of the device, seeped down fluid will enter the axial bores 96 of the plugs 19 and 20 through the previously described ports, orifices, grooves, and passages. As viewed in FIGURE 1, the seepage fluid will leave the plugs 19 and 20 through the bores 99 to enter the annular grooves 98. As will be observed in FIGURE 3, the return check valves 102 communicate with the passages 22 and 23. If the passage 23 is assumed to be the outlet passage, it will contain fluid at pump outlet pressure. Accordingly, the check valve 102 associated therewith will be closed thereby preventing any entry of high pressure fluid into the valve. The other of the check valves 102 will open into the inlet passage 22 whenever the accumulated pressures of the seepage fluid in the annular grooves 98 exceeds the inlet pressure in the passage 22. As is apparent from the foregoing description, this structure provides automatic means for communicating the seepage fluid with the inlet passage of the pump. It will be noted that each of the passages 22 and 23 is provided with a plugged opening 104 whereby access can be had to the check valves 102.

While I have disclosed my invention in connection with certain specific constructions, it is to be understood that these are by way of example rather than by limitation, as it is intended that the invention defined by the following claims should be given as broad a scope as permitted by the state of the art.

I claim:

1. A reversible fluid displacement device which may be operated in either of two directions comprising: a housing with a work performing chamber having inlet and outlet openings, one of which is high pressure and the other low pressure for a given operative direction of the device; a pair of meshed gears rotatably mounted in a close running fit in said chamber, said gears and housing being adapted to cooperate when the gears are rotated to pressurize the fluid trapped in closed pockets at the point of meshing to a higher pressure than the high pressure of the device; a pair of normally stationary cylindrical plugs coaxially positioned in said gears and having a close running fit therein; a plurality of radical passages in each of said gears and each having a central port; passage means in each of said plugs in one position thereof being adapted for successive alignment with said ports, said passage means at one end aligning only with the ports leading to said closed pockets and at the other end only with selected inter-tooth spaces of said gears, all of said selected inter-tooth spaces being closed and located on the low pressure side of the common axial plane of the gears; and means mounting said plugs for selective rotation to either of two positions, one for each operative direction of said device; and additional passage means in said plug interconnecting said closed pockets and selected inter-tooth spaces on the opposite side of said axial plane in the second positions of said plugs whereby all of said selected inter-tooth spaces are always located on the side of said axial plane which is opposite said high pressure chamber opening.

2. A reversible liquid displacement device comprising: a housing having wall means forming a work performing chamber having an inlet and an outlet; a pair of meshed gears each centrally bored and rotatably mounted in close running fit with said chamber, said gears and housinging being adapted to cooperate to pressurize liquid trapped in closed pockets at the area of meshing to a higher pressure than the outlet pressure of the device when the gears are rotated, each of said gears having a plurality of radial passages formed therein each leading from the bore of such gear to the one of the inter-tooth spaces therein; a pair of cylindrical plugs coaxially positioned in first positions in said gears, respectively, and having a close running fit therein, each of said plugs being formed with distribution means therein adapted for successive alignment with the passages of its respective gear, one end of said distribution means aligning only with said closed pockets and the other end thereof aligning only with certain inter-tooth spaces thereof, all of said last named spaces being between teeth having their radially outward end surfaces contacted by the portion of said wall means located on the inlet side of the axial plane through the axes of said gears; and means on said housing engaging both of said plugs to normally hold them stationary in said first positions and that can be removed to permit rotation of both said plugs through the same arc to second positions thereof, wherein said means can be reengaged with said plugs to normally hold them stationary in said second position, and additional distribution means in said plugs complementary to said first distribution means and in said second positions communicating said closed pockets only to certain other gear inter-tooth spaces on the opposite side of said axial plane, whereby the direction of fluid pressure distribution through said device is reversed.

3. The device as claimed in claim 2 in which said additional distribution means aligns with said gear radial passages in the first positions of the plugs to communicate said gear inter-tooth spaces on said opposite side of said axial plane with the inlet side of the pump to relieve the increased pressure within said last-mentioned inter-tooth spaces and in which said first distribution means performs a similar pressure-relieving function for the inter-tooth

spaces on the first side of said axial plane when said plugs are in the second positions.

4. A reversible liquid displacement device comprising: a housing having a work performing chamber therein; a pair of axially bored and meshed gears rotatably mounted in a close running fit in said chamber; fluid inlet and outlet means on opposite sides of the axial plane through the axes of said gears; a pair of cylindrical plugs coaxially positioned in the respective bores of said gears and having a close running fit therein; circumferentially spaced openings in the exterior surface of each of said plugs; passage means in each of said plugs interconnecting said openings; passages leading from the root spaces of said gears to said bores and adapted for successive alignment wtih said openings, said gears trapping fluid in closed pockets between the meshing teeth thereof and compressing said trapped fluid to a pressure greater than the outlet pressure of the device, certain of said openings communicating with said closed pockets and other of said openings communicating with fluid spaces between the gear teeth closed to the inlet and located only on the inlet side of said axial plane whereby to feed said high-pressure trapped fluid into said fluid spaces to increase the pressure therein only on the inlet side of said plane to counter balance radial forces on said gears exerted by the outlet pressure of the device; and a key plate wedgingly engaging confronting keyways formed in said plugs to normally hold them stationary in first arcuate positions for one direction of fluid distribution through the device, said key plate being reversible to wedgingly engage said keyways with said plugs when rotated into a second position to change the point of engagement of the openings with said closed pockets when the direction of fluid distribution through the device is reversed.

5. A device as set forth in claim 4 therein said key plate is connected to the exterior of said housing by screw means centered in said plane through the axes of said plugs at a point closer to one of said plugs than to the other of said plugs.

6. A reversible liquid displacement device comprising: a housing having a work performing chamber therein; a pair of axially bored and meshed gears rotatably mounted in a close running fit in said chamber and having the tooth tips thereof in sliding contact with cylindrical walls of said chamber; a fluid inlet and a fluid outlet on opposite sides of a common axial plane through the axes of said gears, said gears trapping fluid in closed pockets between the meshing teeth thereof and pressurizing said trapped fluid to a higher pressure than the outlet pressure of the device, each of said gears having passages communicating the root spaces of said gears with the bores of the gears; a normally nonrotatable cylindrical plug coaxially positioned in a close running fit within the bore of each of said gears in a first position, each of said plugs having circumferentially spaced openings therein and passageways interconnecting said openings, one of said spaced openings communicating with the gear passages leading to said closed pockets, and the other of said openings communicating only with selected gear passages leading to fluid spaces between gear teeth closed from the inlet and located on the inlet side of said axial plane whereby to communicate said high-pressure trapped fluid in the said pockets only with closed fluid spaces on the inlet side of said axial plane whereby to exert radial forces tending to counterbalance forces exerted on said gears by the outlet pressure of the device; and means mounting said plugs for selective movement to second positions thereof to move said openings for proper cooperation with said pockets when the direction of fluid flow through the device is reversed.

7. A device as set forth in claim 6 in which each of said plugs has two pairs of circumferentially spaced openings and passageways interconnecting said openings, said second pairs of openings communicating with gear passages leading to the fluid spaces between gear teeth located on the outlet side of said axial plane to relieve the increased pressure within fluid spaces on the outlet side of said axial plane.

8. In a reversible gear pump of the type in which fluid is transmitted in inter-tooth spaces between the gear teeth and the walls of a housing with which the gear teeth make sliding contact and which generates a pressure higher than the outlet pressure of the pump in fluid trapped in closed pockets between the meshing teeth of the gears, distribution means to lead off said higher pressure to inter-tooth spaces closed to the pump inlet and located only on the inlet side of the axial plane through the axes of said gears, comprising: normally stationary cylindrical plugs coaxially positioned in said gears and having a close running fit therein; passageways in said plugs communicating spaced circumferential openings therein, said gears having ports therein leading from their bores to the inter-tooth spaces therein, said ports successively aligning with said plug openings as said gears rotate, one opening on each of said plugs aligning only with gear ports leading to said closed pockets between the meshing teeth of the gears and the other opening on each plug aligning only with gear ports leading to selected inter-tooth spaces on the gears on the inlet side of said axial plane whereby said higher pressure exerts a radial force on said gears tending to counterbalance the force exerted on said gears by the outlet pressure of the pump; and means mounting said plugs for selective movement to either of two positions to provide for selective alignment of plug openings with said pockets in each of the operative directions of rotation of said pump.

9. A pump as set forth in claim 8 in which said plugs are provided with two pairs of openings and passageways interconnecting said openings, with said second pairs of openings communicating inter-tooth spaces on the outlet side of said axial plane with the inlet of the pump to relieve the increased pressure within said inter-tooth spaces on the outlet side of said axial plane.

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