

Oct. 15, 1963

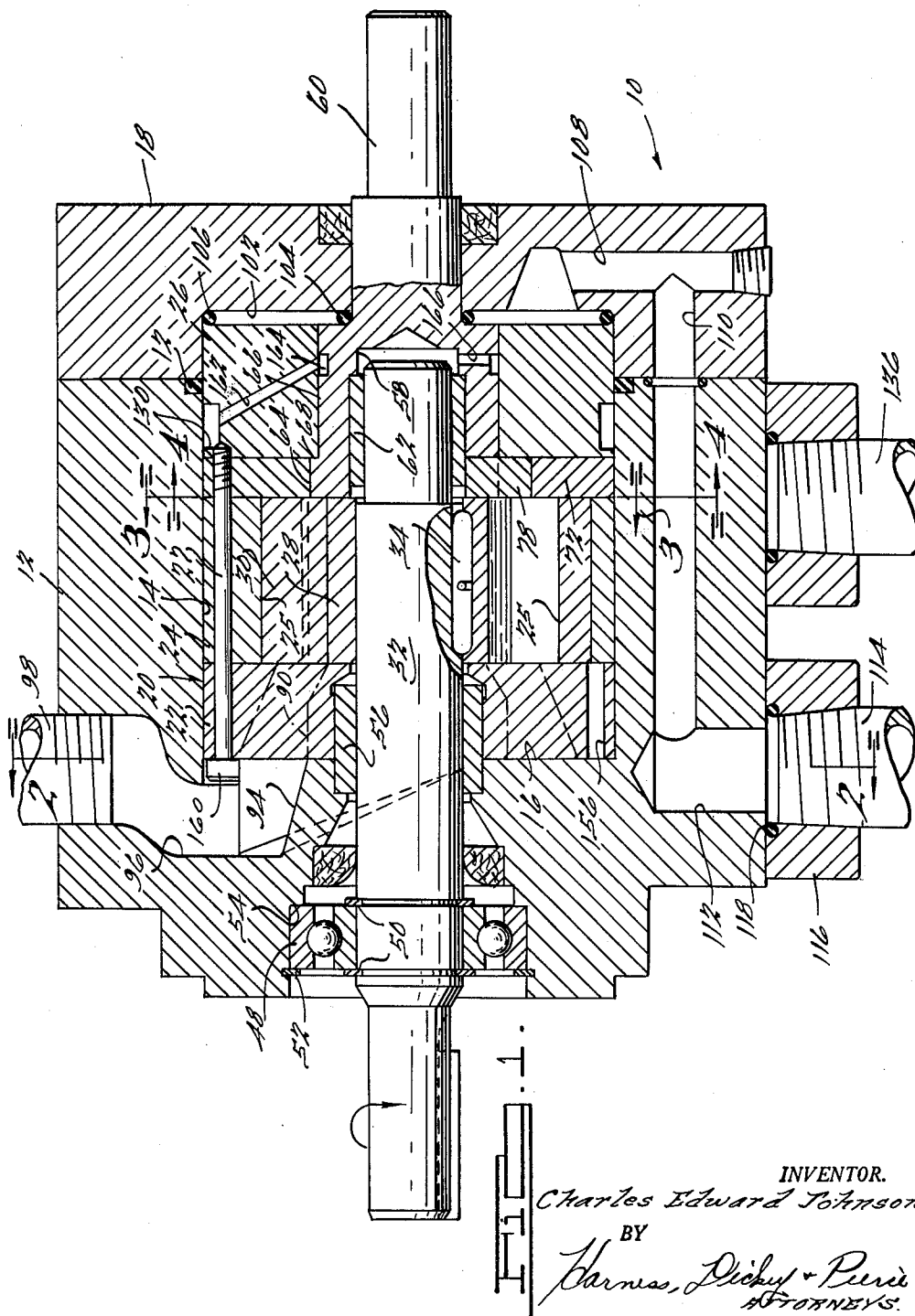
C. E. JOHNSON

3,106,897

FIXED DISPLACEMENT VARIABLE DISCHARGE PUMP

Filed May 25, 1959

3 Sheets-Sheet 1



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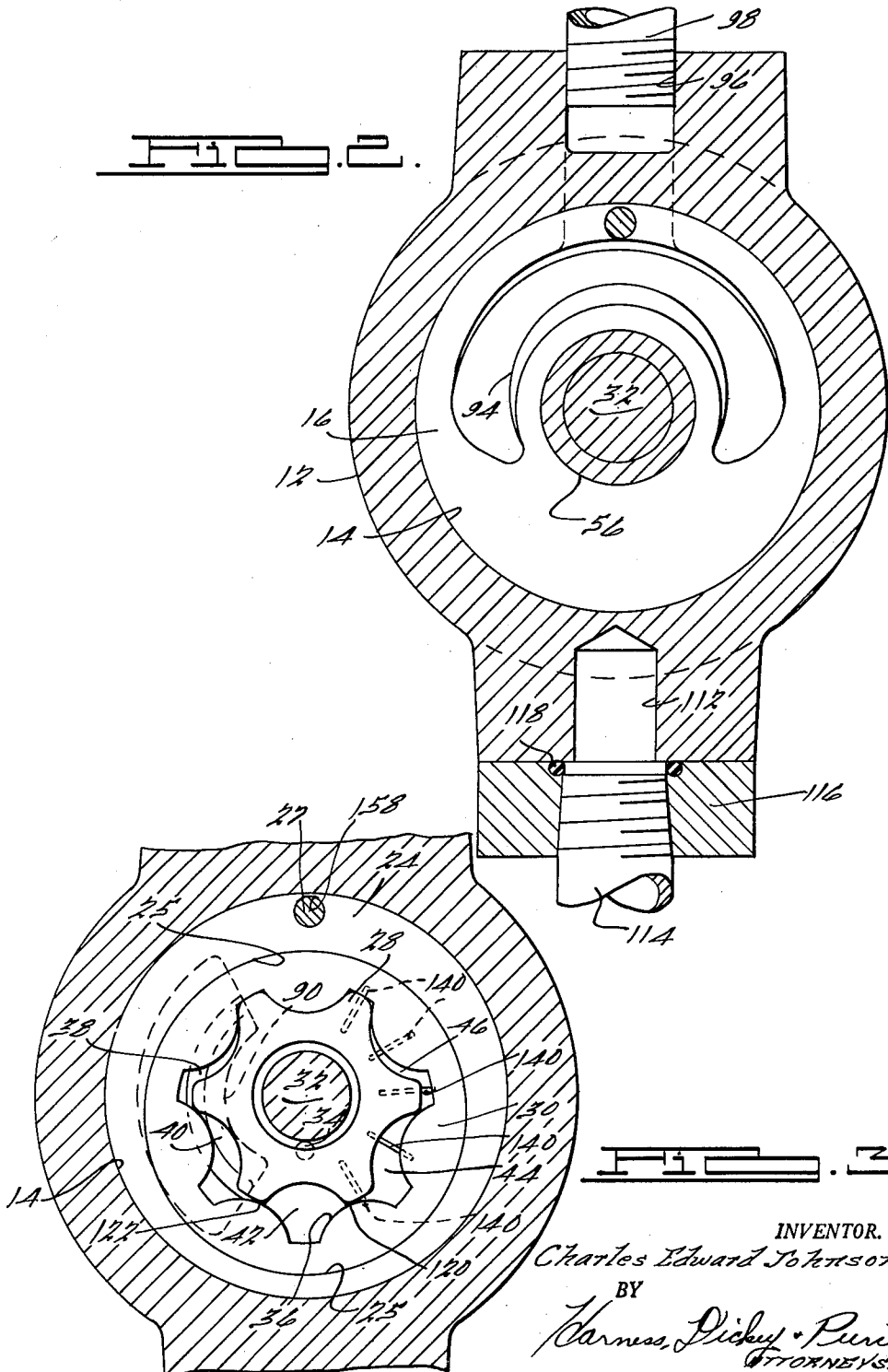
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3 Sheets-Sheet 2



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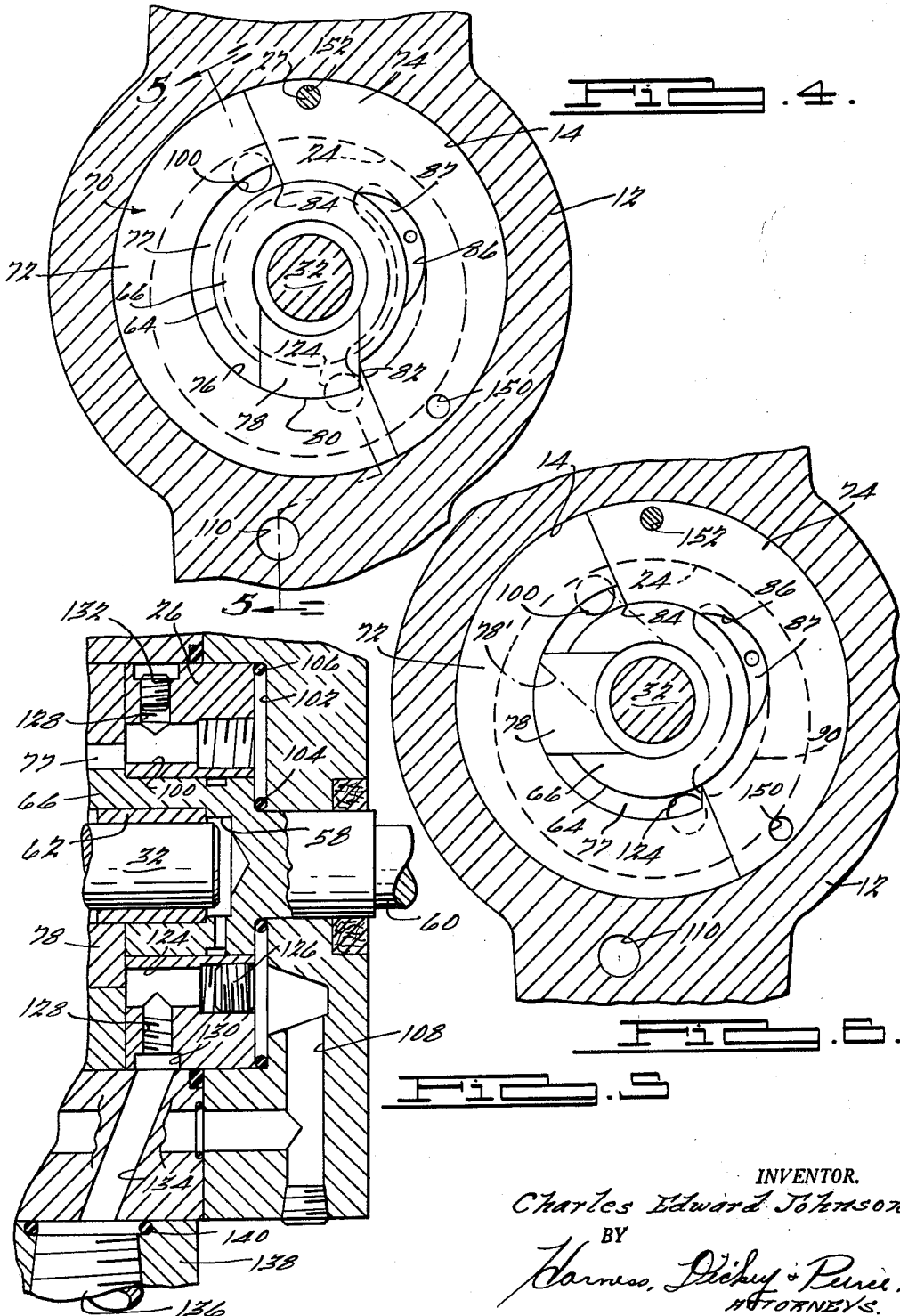
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3 Sheets-Sheet 3



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FIXED DISPLACEMENT VARIABLE
DISCHARGE PUMP

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This invention relates to hydraulic pumps and particularly to a fixed displacement pump having a variable volume pressurized discharge.

Hydraulic pumps can broadly be classified as either fixed displacement or variable displacement pumps. A good example of a fixed displacement pump is the well known gear pump wherein for a given speed of rotation, the pump will take in and discharge the same volume of fluid per unit of time. A good example of a variable displacement pump is a vane pump wherein the eccentricity of the rotating element carrying the radial vanes can be varied to increase or decrease the volume intake of the pump which, in turn, directly increases or decreases the volume discharge of the pump. Of course, the variable displacement pump derives its name from the fact that its volume intake is variable whereas the fixed displacement pump derives its name from the fact that its volume intake is fixed for a given rotational speed of the pump shaft.

Although a variable displacement pump is less efficient than a fixed displacement pump, there are many hydraulic power applications, such as providing pressurized fluid for operating machine tools and hydraulic presses, which can best be performed with a variable displacement pump since these applications initially require fluid at a high volume and low pressure, and thereafter require fluid at a low volume and high pressure. When a variable displacement pump supplies fluid at a low volume and high pressure to a hydraulic press, for example, after it has engaged the work and is applying pressure thereto with very little movement, the condition of the pump is commonly referred to as "dead heading." As is well known in the art, when a variable displacement pump, such as a vane pump, is "dead heading" it generates a great deal of heat and thus wastes a significant portion of its energy input in an extremely undesirable manner. In fact, the heat generated by a "dead heading" vane pump is so detrimental that some manufacturers intentionally manufacture the pump with greater tolerances to increase the volume leakage of the pump in an attempt to combat the heating problem. It is apparent that increasing the volume leakage further decreases the volumetric efficiency of the variable displacement pump, which in the case of the vane-type pump is already poor.

Of course, a fixed displacement pump can also be used in applications requiring a variable volume of pressurized discharge by combining suitable external hydraulic circuitry with the pump to divert part of the output volume thereof back to tank after it has been pressurized. Obviously, however this completely wastes the energy used to pressurize the output volume diverted back to tank. This arrangement has a further major disadvantage in that the horsepower required to operate the pump is directly proportional to the volume output at the discharge multiplied by the fluid pressure and, therefore, when the pump is "dead heading," the horsepower necessary to provide the required maximum pressure for a given application will be determined by the total output volume pressurized which includes the wasted portion diverted back to tank by the external hydraulic circuitry. Consequently, the horsepower required is far greater than that required by the pressurized volume of fluid necessary for the particular application.

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These major shortcomings of hydraulic pumps have seriously limited the use of hydraulic power for many applications, such as for driving machine tools, where hydraulic power would otherwise be ideally suited. Instead, electric motor driven power equipment has been turned to as a more suitable power source for these applications, since the disadvantages of the above-mentioned pumps outweigh the many advantages of hydraulic power for these applications, such as: quick stop and go movement, immediate reversal of direction, precision control of the rate of movement, and the ability to provide extremely high working forces. Admittedly, there are piston-type variable delivery pumps which eliminate the aforementioned disadvantages, but their complexity and cost have priced them out of the bulk of the market enjoyed by the electric motor driven power equipment.

It is an object of the present invention to greatly expand the field of application of hydraulic power by providing a fixed displacement pump having a variable, pressurized discharge that retains the advantages of a fixed displacement pump and provides the variable discharge advantages of a variable displacement pump without incorporating the attendant disadvantages thereof.

It is another object of the invention to modify a fixed displacement pump of the type marketed under the well known trademark Gerotor to provide a variable volume pressurized discharge.

It is a further object of the invention to modify a fixed displacement pump of the type marketed under the well known trademark Gerotor to provide a pressurized discharge that can be varied from a full volume position with a pressure equivalent to that obtainable with the above mentioned pump before modification, to a low volume, high pressure position wherein the pressure can be increased up to approximately three times the maximum pressure heretofore obtainable by such a pump.

It is a still further object of the invention to provide a fixed displacement variable discharge pump having a reduced slippage area at a high pressure, low volume condition, which minimizes the volume leakage at this condition.

It is a still further object of the invention to provide a fixed displacement variable delivery pump wherein the area of the rotating pump elements upon which the pressurized fluid acts is reduced as the pressure of the fluid increases to thereby reduce the unbalanced loading on the pump elements.

It is a still further object of the invention to provide a fixed displacement variable discharge pump having a mechanical efficiency and overall efficiency substantially equal to the efficiency of present fixed displacement pumps and wherein internal components of the pump can be re-oriented to enable the direction of rotation of the pump to be reversed without requiring the external connections to the pump to be changed.

It is a still further object of the invention to provide a fixed displacement variable discharge pump having fluid pressure balance grooves therein such that changes in the volume output of the pump automatically causes the balance area provided by the balance grooves to vary directly in proportion to the change in volume output.

It is a still further object of the invention to provide a fixed displacement variable discharge pump wherein the orientation of the port plates to the rotating pump elements determines the direction the pump shaft is to be rotated.

It is a still further object of the invention to provide a fixed displacement variable discharge pump which is efficient in operation, economical to manufacture, and versatile and rugged in operation.

Other objects and features of novelty of the invention will be specifically pointed out or otherwise become

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apparent when referring, for a better understanding of the invention, to the following description taken in conjunction with the accompanying drawings wherein:

FIG. 1 is a sectional view of a pump embodying features of the present invention;

FIG. 2 is a reduced sectional view taken along the line 2—2 of FIG. 1;

FIG. 3 is a reduced sectional view taken along the line 3—3 of FIG. 1;

FIG. 4 is a reduced sectional view taken along the line 4—4 of FIG. 1;

FIG. 5 is a sectional view taken along the line 5—5 of FIG. 4, and

FIG. 6 is a sectional view similar to that of FIG. 4 illustrating another position of the seal member of the pump to provide a reduced volume of pressurized fluid.

Referring to FIG. 1, a fixed displacement variable discharge pump 10 embodying features of the invention is illustrated. The pump 10 is comprised of a cylindrical body 12 having a cylindrical chamber 14 therein terminating in a bottom wall 16. The open end of the chamber 14 is closed by a cover plate 18 suitably secured to the body 12 with an O-ring 20, or the like, disposed therebetween to provide a seal against leakage of hydraulic fluid from the chamber 14.

A cartridge assembly 20 is disposed within the chamber 14 in sealing relationship therewith and comprises a suction port plate 22, a ring 24 and a discharge port plate 26 all suitably secured together in sandwich relationship by an elongated bolt 27. As most clearly illustrated in FIG. 3 a pair of rotating pumping elements, namely, an inner gearlike element 28 and an outer internally generated element 30, are disposed within an eccentric opening 25 of the ring 24 with the opposite faces thereof slidably engaging the faces of the suction port plate 22 and discharge port plate 26.

The inner and outer elements 28 and 30 and the ring 24 illustrated in FIG. 3 are of the type used in the well-known pump sold under the trademark Gerotor, and in this arrangement the outer element 30 fits closely within the eccentric opening 25 of the ring 24 but is free to rotate relative thereto. The inner element 28 is suitably keyed to a pump shaft 32 by a key 34, and the axis of the shaft is concentric with the axis of the ring 24 so that the inner element 28 is eccentrically disposed relative to the outer element 30. As is well known in the art the teeth of the inner element 28 drivingly engage projections 36 of the outer element and the teeth not drivingly engaged with the projections are each so closely disposed to, or contact one of, the projections 36 so that at all times each of the teeth cooperates with one of the projections to provide a seal on either side of openings 38, 40, 42, 44 and 46 as most clearly illustrated in FIG. 3.

Referring again to FIG. 1 in particular, the pump shaft 32 extends through the suction port plate 22 and the left end of the body 12, and is journaled in the body 12 by a suitable ball bearing 48, or the like, which is fixed against axial movement relative to the shaft by suitable snap rings 50 fitted in grooves on the shaft. An additional snap ring 52 may also be provided which fits within a groove in the body 12 to retain the bearing 48 against a shoulder 54. The pump shaft 32 may also be supported intermediate its ends by a bushing 56, or the like, and the right end thereof may be suitably journaled within a counterbore 58 in the left end of a control shaft 60 by a bushing 62. The control shaft 60 is rotatable relative to the discharge port plate 26 and the shaft 32, and has an annular flange 64 projecting radially from the enlarged left end 66 thereof to provide an annular shoulder 68 which overlaps the discharge port plate 26 a sufficient distance to prevent axial movement of the control shaft 60 to the right as viewed in FIG. 1.

Referring to FIG. 4 as well as FIG. 1, a split ring 70 forms part of the discharge port plate 26 and is made up of substantially semi-circular portions 72 and 74 secured

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to the discharge port plate 26 in any suitable manner, such as by bolting. The inner periphery of the semi-circular portion 74 has the same radius of curvature as the flange 64 of the control shaft 60 and the inner periphery 76 of the semi-circular portion 72 has a larger radius of curvature than the flange 64 so as to be spaced therefrom and cooperate therewith to define a semi-circular discharge groove 77 between the discharge port plate 26 and the right face of the inner and outer rotating elements 28 and 30 as viewed in FIG. 1. In addition, a seal member 78 is keyed to the enlarged left end 66 of the control shaft 60 and projects therefrom radially outwardly into and completely fills a portion of the discharge groove 77. It will be observed that the seal member 78 has the same thickness as the semi-circular portion 72 as illustrated in FIG. 1 and that outer end 80 thereof has the same radius of curvature as the inner periphery 76. A small integral projection 82 is also provided on the corner of the semi-circular portion 72 so as to completely fill the space between the edge of the seal member 78 and the adjacent edge of the semi-circular portion 74.

As stated previously, the seal member 78 completely fills a portion of the discharge groove 77 and can be rotated in a clockwise direction as viewed in FIG. 4 to the position illustrated in FIG. 6 and further to the phantom position indicated by the numeral 78' wherein the edge thereof abuts the corner 84 of the semi-circular portion 74. The seal member 78 is of course rotated by rotating the control shaft 60 as will be described in greater detail hereinafter. It will also be observed in FIG. 4 that the semi-circular portion 74 is relieved as at 86 so as to cooperate with the annular flange 64 to define a chamber 87 for a purpose to be described.

With the rotating pump elements 28 and 30 arranged as illustrated in FIG. 3, the pump shaft 32 would be rotating in a counterclockwise direction which, of course, also rotates the inner element 28 and the outer element 30 in a counterclockwise direction, and the left side of the elements is conventionally referred to as the suction side and the right side thereof the discharge side. With the elements in the particular relative position illustrated in FIG. 3, it is apparent that as the inner and outer elements 28 and 30 rotate, the openings 38 and 40 will progressively increase in volume and the openings 44 and 46 will progressively decrease in volume whereas the opening 42 is at a transition point. That is, prior to the position illustrated in FIG. 3, the opening 42 will have been increasing in volume and after passing through the position illustrated in FIG. 3, the opening 42 will decrease in volume.

In the conventional pump sold under the trademark Gerotor, an elongated curved suction port is placed in communication with the openings 38 and 40 so that as they increase in volume, fluid will be sucked into the openings. A similar elongated curved discharge port is placed in communication with the openings 44 and 46 so that as they decrease in volume the fluid in the openings will be pressurized and forced out the discharge port. Of course, the fluid in the opening 42 is not under pressure at the particular position illustrated in FIG. 3 and is not in communication with either the suction or discharge ports since the points of contact between the teeth of the inner element, and the projections 36 on the outer element provide a leading seal 120 on one side of the opening 42 and a trailing seal 122 on the other side thereof to prevent communication with the suction and discharge ports. However, just prior to the elements reaching the position illustrated in FIG. 3, the opening 42 was in communication with the suction port since it was increasing in volume at the time and the trailing seal 122 overlapped the suction port. Similarly, when the elements rotate beyond the position illustrated in FIG. 3, the leading seal 120 overlaps the discharge port to place the opening 42 in communication with the discharge port so

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that as it progressively decreases in volume, it will force the fluid therein out the discharge port.

However, it is apparent that when the pumping elements operate in this conventional manner, the same amount of fluid is sucked into the suction side of the rotating elements as is forced out the discharge side. As will be seen from the following description, the present invention greatly improves upon this conventional fixed displacement operation by enabling the pressurized discharge volume to be varied while the volume intake on the suction port side remains constant.

Referring to FIGS. 1 and 3, an elongated curved suction port 90 is provided in the suction port plate 22. The suction port 90 preferably has a narrow opening on the face thereof overlying the openings 38 and 40 between the rotating inner and outer elements 28 and 30 and a wider opening on the other face thereof communicating with the left portion of a horseshoe-shaped passageway 94 most clearly illustrated in FIG. 2. The passageway 94 is formed in the bottom wall 16 of the body 12 and communicates with a radially extending inlet port 96 having an internally threaded portion for receiving the threaded end of an inlet conduit 98. Of course, the remainder of the horseshoe-shaped passageway 94 that does not communicate with the suction port 90 seals against the flat surface of the suction port plate 22. The novel purpose and function of the horseshoe-shaped passageway 94 will be described in greater detail hereinafter.

Referring also to FIG. 4, the discharge groove 77 defined by the annular flange 64 and the semi-circular portion 72 overlies the openings 44 and 46, and the seal member 78 completely overlies the opening 42. Therefore, as the openings 44 and 46 progressively decrease in size in response to the rotation of the inner and outer elements, the pressurized fluid therein will be forced into the discharge groove 77 under pressure, and into a high pressure port 100 which is formed in the discharge port plate 26 and communicates with one end of the discharge groove 77.

The high pressure port 100 is most clearly illustrated in FIG. 5 where it will be seen that it extends parallel to the axis of the pump shaft 32 with the right end thereof communicating with an annular chamber 102 between the discharge port plate 26 and the cover plate 18. The annular chamber 102 is sealed by an inner O-ring 104 and an outer O-ring 106, and communicates with a radially extending passageway 108 in the valve body 12 which, in turn, communicates with a passageway 110. The passageway 110 communicates at its other end with a passageway 112 (FIG. 1) and a suitable externally threaded high pressure conduit 114 may be threaded into an internally threaded high pressure fitting 116 suitably secured to the valve body 12 with an O-ring 118 therebetween to prevent leakage.

Referring particularly to FIGS. 3 and 4 when the seal member 78 is in the position illustrated, it completely covers the opening 42 as previously mentioned and, therefore, prevents communication thereof with either the suction port 90 or the discharge groove 77. However, as soon as the elements rotate a little further in a counter-clockwise direction, as viewed in FIG. 3, which would be a clockwise direction of rotation of the pump shaft 32 as viewed in FIG. 4, the leading seal 120 immediately passes beyond the edge of the seal member 78 whereas the trailing seal 122 continues to overlie the seal member. This immediately places the opening 42, which is now decreasing in volume, in communication with the discharge groove 77 and the high pressure port 100 and, of course, the trailing seal 122 continues to prevent communication between the discharge and suction side of the rotating elements.

At this point it may be well to note that in a hydraulic pump such as the one being described, all surfaces which engage each other or slide upon each other are precision machined with suitably close tolerances to ensure a good

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high pressure seal therebetween. Therefore, when the inner and outer rotating elements rotate relative to the surfaces of the suction port plate 22 and the split ring 70, including the portion of the seal member 78 projecting within the discharge groove 77, a sealing relationship obtains therebetween. It is also noted that the distance between the leading seal 120 and the trailing seal 122 is equal to or slightly less than the width of the seal member 78 to ensure that the opening 42 is not in communication with either the discharge or suction side of the rotating elements 28 and 30 when they are in the position illustrated in FIG. 3, and that the opening 42 will be in communication with the suction port 90 just before the rotating elements reach this position and in communication with the discharge groove 77 just after the rotating elements move past this position.

Before describing the manner in which the volume of pressurized discharge can be reduced relative to the volume intake of the pump, it will be observed in FIG. 4 that a bypass port 124 is also formed in the discharge port plate 26 which communicates with the other end of the discharge groove 77 but is sealed off by the seal member 78. As most clearly illustrated in FIG. 5, the bypass port 124 extends parallel to the axis of the pump shaft 32 and has a plug 126 screwed into one end thereof to prevent communication with the annular chamber 102 with which the high pressure port 100 communicates. The discharge port plate 26 is also provided with a radially extending passageway 128 which communicates with the bypass port 124 and an annular groove 130 in the outer wall of the discharge port plate. It will be observed that the high pressure port 100 has a similar radially extending passageway 128 communicating therewith and with the annular groove 130 and that it is sealed off by a plug 132. With this construction, the high pressure port 100 and the bypass port 124 are identical and the plug 132 can be removed and used to seal off the other passageway 128 to prevent communication of the bypass port 124 with the annular groove 130, and the plug 126 can be removed and used in the internally threaded end of the high pressure port 100 to prevent communication thereof with the annular chamber 102. In this way, the high pressure port 100 can be made to function as the bypass port 124 and vice versa as will be described. A slanted passageway 134 is formed in the valve body 12 with one end thereof communicating with the groove 130 and the other end thereof communicating with an external bypass conduit 136 screwed into an internally threaded bypass fitting 138 suitably secured to the valve body 12 with an O-ring 140 provided therebetween to prevent leakage.

In operation, with the seal member 78 in the position illustrated in FIG. 4, the volume of pressurized fluid discharged will be equal to the volume of fluid taken in through the suction port 90. However, when the seal member 78 is moved in a clockwise direction, as viewed in FIG. 4, toward the high pressure port 100, the volume of pressurized fluid will be reduced and the difference between this volume and the intake volume will pass directly through the bypass port 124 and thence out through the external bypass conduit 136 without being pressurized. For example, when the seal member 78 is in the position illustrated in FIG. 6, one of the teeth of the rotating inner element 28 will be engaged with one of the projections 136 of the outer element to provide a sealing point overlying the seal member 78 to prevent communication between the high pressure port 100 and the bypass port 124. This is so because the width of the seal member 78 is substantially equal to the spacing between two adjacent sealing points, as previously described. Therefore, the fluid in the openings between the rotating elements 28 and 30 which are on the pressurized side of the rotating elements and which are also on the bypass port side of the sealing point overlying the seal member 78, will be pumped directly out the bypass port 124 and into the bypass conduit 136 without being pressurized. Similarly, the fluid in the

openings on the high pressure port side of the seal point overlying the seal member 78 will be pumped directly into the high pressure port 100, and then out the high pressure conduit 114. Of course, the pressure developed by the pump at the high pressure port will be determined by the hydraulic device being operated by the pump.

From the foregoing, it is readily apparent that the volume of the pressurized discharge can be varied by simply adjusting the position of the seal member 78 between the high volume position illustrated in FIG. 4 to the low volume position indicated by the number 78' of FIG. 6. Thus, the present invention provides, for the first time, a fixed displacement pump having a variable volume discharge which embodies substantially all of the advantages of both fixed and variable displacement pumps, plus some additional advantages. For example, as the seal member 78 is rotated to decrease the volume of the pressurized discharge, it is apparent that the unbalanced loading on the inner rotating element 28 proportionately decreases and, therefore, the pressure that can be developed by the pump can be proportionately increased. Theoretically, the pump of the present invention can be operated at up to three times the pressure at which a conventional fixed displacement Gerotor pump can be operated.

Similarly, as the volume of the pressurized discharge is reduced, the slippage areas are proportionately reduced and, consequently, the volume leakage of the pump is greatly reduced. Still further, when the seal member 78 is rotated to the dead heading position illustrated by the number 78' of FIG. 6, the fluid being circulated directly through the pump and out the bypass port 124 without being pressurized is at a maximum volume and provides an excellent coolant that offsets the tendency of the pump to heat up. As described at the outset of this application, the tendency of a conventional variable displacement pump to heat up when dead heading is a serious problem.

Another important feature of the pump of the present invention resides in the plurality of substantially radially extending balance grooves 140 provided in the face of the suction port plate 22 in position to overlie the pressurized side of the rotating elements 28 and 30 as viewed in FIG. 3. The balance grooves 140 retain pressurized fluid to balance or offset the tendency of the pressurized fluid in the discharge groove 77 to press the rotating elements against the suction port plate 22. In fact, the balance grooves pressurize a film of fluid between the port plate 22 and the rotating elements over separated areas surrounding each balance groove, which areas are spaced far enough apart so as not to communicate with one another. This pressurized film further helps balance the axial loading on the rotating elements 28 and 30.

Further, as the seal member 78 is moved in a clockwise direction, as viewed in FIGS. 4 and 6, to reduce the volume of the pressurized discharge, it is apparent that the force tending to press the rotating elements 28 and 30 against the suction port plate 22 reduces since the area of the rotating elements exposed to the pressurized fluid in the discharge groove 77 is being reduced. However, since the balance grooves 140 and the surrounding areas of pressurized film are all separated from one another, the number of grooves which act to offset the unbalancing force is proportionately reduced as the seal member 78 moves in a clockwise direction. Another advantage of the balance grooves 140 is that they also introduce the oil film between the suction port plate 22 and the rotating elements which film acts as a lubricant to further minimize the frictional resistance to the free rotation of the elements 28 and 30.

Still another feature of the present invention resides in the provision of the relieved portion 86 on the inner periphery of the semi-circular portion 74 which defines a chamber 87 in the discharge port plate 26 on the suction side of the rotating elements 28 and 30. By positioning the chamber 87 at the point of maximum rate of change

of the displacement of the openings 38 and 40, cavitation within the pump, particularly when it is operating at higher speeds, can be greatly reduced if not eliminated. The chamber 87 accomplishes this by providing an additional reservoir for fluid so that the suction openings between the rotating elements 28 and 30 will at all times have an adequate supply of fluid to draw upon to ensure that they are completely filled without creating voids.

Still another important feature of the present invention resides in the U-shaped passageway 94 in the bottom wall 16 of the pump body 12, and the manner in which the fluid is introduced thereto intermediate the ends thereof by the passageway 96. As previously described, the pump shaft 32 rotates in a counterclockwise direction as viewed in FIG. 2, and the suction port 90 overlies the left half of the U-shaped passageway 94. Therefore, as the openings on the suction side of the rotating elements 28 and 30 increase in size, the fluid will be drawn from the inlet conduit 98 into the left side of the U-shaped passageway 94 in a counterclockwise direction. This in turn introduces the fluid to the suction port 90 in a counterclockwise direction so that it is delivered to the openings between the elements 28 and 30 in the same direction in which they are rotating. By introducing the fluid in the same direction as the direction of rotation of the elements, unnecessary turbulence is avoided and the likelihood of cavitation developing within the pump is greatly reduced.

Further, by making the passageway 94 U-shaped, the cartridge assembly 20 can be repositioned within the chamber 14 of body 12 so that the suction port 90 overlies the right side of the U-shaped passageway 94. With this arrangement, the direction of rotation of the pump shaft 32 can be reversed to a clockwise direction, and the fluid will be introduced from the passageway 96 into the right side of the U-shaped passageway 94 in a clockwise direction to reduce the likelihood of cavitation in the same manner as previously described. Thus, by a very simple reorientation of the cartridge assembly 20, the fluid can always be introduced into the openings 38, 40 and 42 in the same direction that pump shaft 32 is rotating regardless of whether the shaft is rotating in a clockwise or counterclockwise direction.

When it is desired to reverse the direction of rotation of the shaft 32 in the manner just described, the entire cartridge assembly 20 is removed from the chamber 14 and disassembled by removing the bolt 27. The cartridge is then reassembled with the suction port plate 22 rotated 180° relative to the ring 24 and the discharge port plate (with the control shaft 60 still assembled thereto) rotated through somewhat less than 180°. When reoriented in this manner, the aperture 150 (FIG. 4) in the discharge port plate 26 and the aperture 156 (FIG. 1) in the suction port plate 22 are aligned with the aperture 158 (FIG. 3) in the ring 24 and the bolt 27 is then inserted through the newly aligned apertures to reassemble the cartridge assembly 20. The cartridge assembly is then reinserted within the chamber 14 with the suction port 90 overlying the right side of the U-shaped passageway 94 as previously described. It will be observed in FIG. 1 that head 160 of the bolt 27 projects into a recess in the bottom wall 16 of the valve body 12 so that the cartridge assembly 20 can be positioned within the valve body 12 in only one position. Therefore, when the bolt head 160 is fitted within the suitable recess provided in the bottom wall, the suction port 90 will automatically overlie the desired side of the U-shaped passageway 94 depending upon the manner in which the elements of the cartridge assembly 20 were oriented relative to one another prior to inserting the cartridge assembly within the chamber 14.

Normally when the direction of rotation of a conventional pump shaft is reversed the external conduits connected to the pump must also be reversed. However, when the cartridge assembly 20 is reoriented in the previously described manner to enable the shaft 32 to be ro-

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tated in the reverse direction, the external conduits 96, 114 and 136 do not have to be changed. This very important and desirable feature is accomplished by enabling the high pressure port 100 to act as the bypass port 124 by merely removing the plug 126 (FIG. 5) from the bypass port and screwing it into the corresponding position in the high pressure port 100 and removing the plug 132 and screwing it into the passageway 128 communicating with the bypass port 124 as previously described. With this change it is apparent that the high pressure port 100 will now serve as the bypass port since it communicates with the annular groove 130 which in turn communicates with the passageway 134 and the bypass conduit 136. Likewise, the bypass port 124 now becomes the high pressure port since it communicates with the annular chamber 102 which in turn communicates with the high pressure conduit 114 as previously described. Thus, by merely reversing the plugs 126 and 132 the high pressure port 100 becomes the bypass port and the bypass port 124 becomes the high pressure port to enable the pump shaft to be rotated in the opposite direction without changing a single external connection to the pump.

As most clearly illustrated in FIG. 1 a slanting passageway 162 may also be provided in the discharge port plate 26 to communicate the annular groove 130 with an annular groove 164 on the enlarged left end portion 66 of the control shaft 60. A radially extending passageway 166 may then be provided for communicating the annular groove 164 with the counterbore 58 in the left end of the control shaft to drain any fluid that finds its way past the bushing 62 into the counterbore 58.

To summarize the operation of the pump, when it is assembled as illustrated in the drawings and with the seal member in the position illustrated in FIG. 4, the volume of the pressurized discharge will be the same as the volume intake of the pump on the suction side thereof. This may be referred to as a high volume, low pressure condition. When it is desired to vary the volume of the pressurized discharge, the control shaft 60 is rotated to vary the position of the seal member 78 within the discharge groove 77 as previously described. Of course the seal member can be moved to the extreme position illustrated in phantom in FIG. 6 and indicated by the numeral 78 wherein the pump would be operating at its lowest volume and can safely develop the maximum pressure to be supplied to the hydraulic device being operated by the pump. As the seal member is rotated to vary the volume delivery of the pump, the number of separated balance grooves 140 (FIG. 3) exposed to the pressurized fluid automatically changes in proportion to the change in area of the discharge grooves 77 of the pump. This tends to balance the rotating elements 28 and 30 axially as previously described, so that they are not pressed against the suction port plate 22 or the split ring 70 with sufficient force to create excessive frictional forces that would interfere with the rotation of the elements. Of course, the rotation of the pump shaft is determined by the relationship of the suction and discharge port plates to the rotating elements at the time of assembly of the cartridge assembly 20 by the bolt 27, and the position of the cartridge within the chamber 14 is automatically determined by fitting the bolt head 160 into the recess in the valve body 10 as previously described.

The control shaft 60 can be either manually controlled, or automatically controlled by suitable external control mechanisms to automatically vary the position of the seal member 78 in response to the conditions of the external hydraulic device and/or hydraulic circuitry associated therewith. For example, the seal member 78 can be automatically controlled to move toward its low volume position as the fluid pressure required by the external hydraulic circuitry increases in order to maintain the horsepower needed to rotate the pump shaft 32 substantially constant. Likewise, if desired, suitable control mecha-

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nism may be associated with the control shaft 60 to automatically vary the volume delivery of the pump to a hydraulic device, such as a hydraulic motor, to run the motor at a constant speed. Of course, many other variations in the control of the pump are possible since the pressurized volume delivery of the pump can be regulated so easily. The only requirement is that the control shaft 60 be actuated in a manner to counteract the tendency of the pressurized hydraulic fluid in the discharge groove 77 to move the seal member 78 in a counterclockwise direction as viewed in FIG. 4.

In the preferred embodiment of the invention described above, the discharge groove 77 and seal member 78 are disposed on the opposite side of the ring 24 from the suction port plate 22. Although this arrangement is preferred, it is to be understood that the present invention is not limited to this arrangement since the discharge groove and seal member could be disposed on the same side as the suction port plate, if desired, and still function to vary the volume of the pressurized discharge in accordance with the teaching of the present invention. Of course, if this were done some of the advantages of the preferred arrangement would be sacrificed, but the basic teaching of the invention could still be utilized. Further, it will be apparent to those skilled in the art that several of the features of the invention, such as the relief chamber 87 and the U-shaped passageway 94, can be advantageously employed in a conventional fixed displacement pump wherein the output volume is not variable.

Therefore, while it will be apparent that the embodiment of the invention herein disclosed is well calculated to fulfill the objects of the invention, it will be appreciated that the invention is susceptible to modification, variation and change without departing from the proper scope or fair meaning of the subjoined claims.

What is claimed is:

1. A fixed displacement variable discharge pump comprising a pump body having parallel spaced-apart suction and discharge port plates therein, means defining a pumping chamber, rotatable pumping elements in said pumping chamber, said chamber being sealingly disposed between the surfaces of said port plates, suction port means formed in said body and said suction port plate and opening into said chamber on one side of said pump elements in position to overlie a predetermined portion of one-half of the periphery of said pumping elements, a rotatable shaft having one end extending through said discharge port plate and abuttingly engaging said pumping means with the other end thereof projecting from said pump body, a split ring interposed between said discharge port plate and the other side of said pumping means and fixed to the discharge port plate, the inner periphery of one part of said split ring sealingly engaging said shaft and the inner periphery of the other part of the split ring having a greater radius of curvature than the radius of said shaft so as to cooperate therewith to provide an annular semi-circular groove on the face of said discharge port plate having one axial end thereof in communication with the other half of the periphery of said pumping means, a portion of the other axial end of said semi-circular groove being closed by said discharge port plate, bypass port means formed in said body and discharge port plate communicating with a low pressure source and opening on one end of said semi-circular groove, said bypass and discharge ports being circumferentially spaced from one another, high pressure port means formed in said body and said discharge port plate and opening on the other end of said semi-circular groove, a seal finger keyed to said shaft and projecting therefrom into said semi-circular groove, said seal finger having a thickness substantially equal to the thickness of said split ring and a curved end having substantially the same radius of curvature as the inner periphery of said other part of the split ring, said seal finger being movable in response to the rotation of said shaft wherein it overlies and seals off said bypass

port means at said one end of the semi-circular groove toward the other end of the semi-circular groove, whereby as the seal finger moves from said one end of the groove to the other it progressively decreases the length of the semi-circular groove communicating with the high pressure port and increases the length thereof communicating with the bypass port.

2. In a fixed displacement variable discharge pump, a cartridge assembly adapted to be positioned within said pump comprising a ring having an eccentric opening, an internally generated outer element rotatably positioned within said eccentric opening and defining a pumping chamber, a gear-like inner element eccentrically positioned within and drivingly engaging said outer element in a manner to provide a suction side and a pressure side between the elements, said sides extending one hundred eighty degrees on opposite sides of a plane containing the axis of said inner element, a suction port plate secured to one face of said ring and slidably engaging said inner and outer elements, a discharge port plate secured to the other face of said ring and slidably engaging said inner and outer elements, an elongated curved suction port formed in said suction port plate in position to overlie said suction side of the inner and outer elements, a split ring secured between said ring and discharge port plate, a control shaft rotatably extending through and projecting from said discharge port plate with the inner end thereof slidably engaging said inner element and the axis thereof co-axially aligned with the axis of said inner element, means for retaining the control shaft against axial movement, the inner periphery of one part of said split ring slidably engaging the periphery of said control shaft adjacent said one end thereof, the inner periphery of the other part of said split ring being spaced from the adjacent periphery of the control shaft to define a discharge groove having one axial end communicating with said pressure side of the inner and outer elements, a portion of the other axial end of said groove being closed by said discharge port plate, bypass port means in said discharge port plate communicating with one end of said discharge groove and a low pressure source, said bypass and discharge ports being circumferentially spaced from one another, high pressure port means in said discharge port plate communicating with the other end of said discharge groove, and a seal member disposed within said discharge groove and movable from a position wherein it seals off said bypass port means toward said high pressure port means, said seal member providing a seal between said bypass and high pressure port means at all positions therebetween.

3. In a fixed displacement variable discharge pump, a cartridge assembly adapted to be positioned within said pump comprising a ring having an eccentric opening, an internally generated outer element rotatably positioned within said eccentric opening and defining a pumping chamber, a gear-like inner element eccentrically positioned within and drivingly engaging said outer element in a manner to provide a suction side and a pressure side between the elements, said sides extending one hundred eighty degrees on opposite side of a plane containing the axis of said inner element, a suction port plate secured to one face of said ring and slidably engaging said inner and outer elements, a discharge port plate secured to the other face

of said ring and slidably engaging said inner and outer elements, an elongated curved suction port formed in said suction port plate in position to overlie said suction side of the inner and outer elements, a split ring secured between said ring and discharge plate, a control shaft rotatably extending through and projecting from said discharge port plate with the inner end thereof slidably engaging said inner element and the axis thereof co-axially aligned with the axis of said inner element, means for retaining the control shaft against axial movement, the inner periphery of one part of said split ring slidably engaging the periphery of said control shaft adjacent said one end thereof, the inner periphery of the other part of said split ring being spaced from the adjacent periphery of the control shaft to define a discharge groove having one axial end communicating with said pressure side of the inner and outer elements, a portion of the other axial end of said discharge groove being closed by said discharge port plate, a bypass port in said discharge port plate communicating with one end of said discharge groove and opening on the outer face of the discharge port plate and communicating with a low pressure source, said bypass and discharge ports being circumferentially spaced from one another, a high pressure port communicating with the other end of said discharge groove and opening on the outer face of the discharge port plate, an annular groove formed on the outer periphery of said discharge port plate, a first passageway in said discharge port plate communicating said bypass port with said annular groove, a second passageway communicating said high pressure port with said annular groove, a seal member keyed to said one end of the control shaft and projecting into said discharge groove, the outer end of said seal member slidably engaging the inner periphery of said other part of the split ring and the thickness of said seal member being substantially equal to the thickness of said split ring whereby the seal member completely fills a portion of the discharge groove and provides a movable seal which at all times prevents communication between said bypass high pressure ports.

References Cited in the file of this patent

UNITED STATES PATENTS

45	Re. 24,064	Welch	Sept. 20, 1955
	1,482,807	Newberg	Feb. 5, 1924
	2,277,160	Shaw	Mar. 24, 1942
	2,416,987	Fleischer	Mar. 4, 1947
	2,426,491	Dillon	Aug. 26, 1947
50	2,728,297	Cilley	Dec. 27, 1955
	2,872,872	Quintilian	Feb. 10, 1959
	2,898,862	Brundage	Aug. 11, 1959
	2,899,903	Ryder	Aug. 18, 1959
	2,915,017	Whitney	Dec. 1, 1959
55	2,937,599	Rosaen	May 24, 1960
	2,938,663	Luck	May 31, 1960
	2,948,229	Brundage	Aug. 9, 1960

FOREIGN PATENTS

60	81,457	Netherlands	May 15, 1956
	470,699	Germany	Jan. 25, 1929
	499,585	Great Britain	Jan. 25, 1939
	1,035,238	France	Apr. 15, 1953
	1,209,036	France	Sept. 21, 1959