July 8, 1958 G. A. WAHLMARK 2,842,064

HYDRAULIC PRESSURE UNIT

Filed May 24, 1954 4 Sheets-Sheet 1

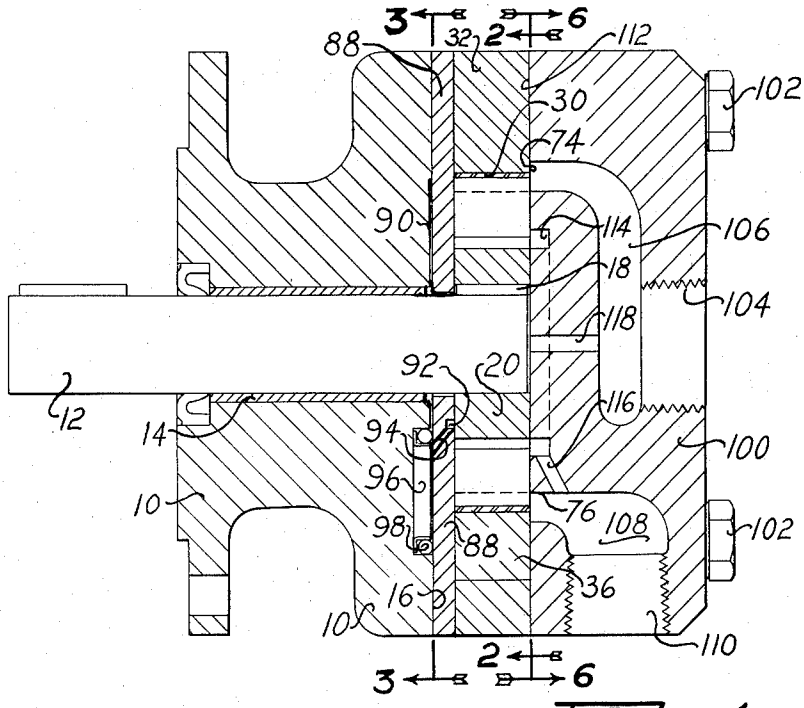


FIG. 1

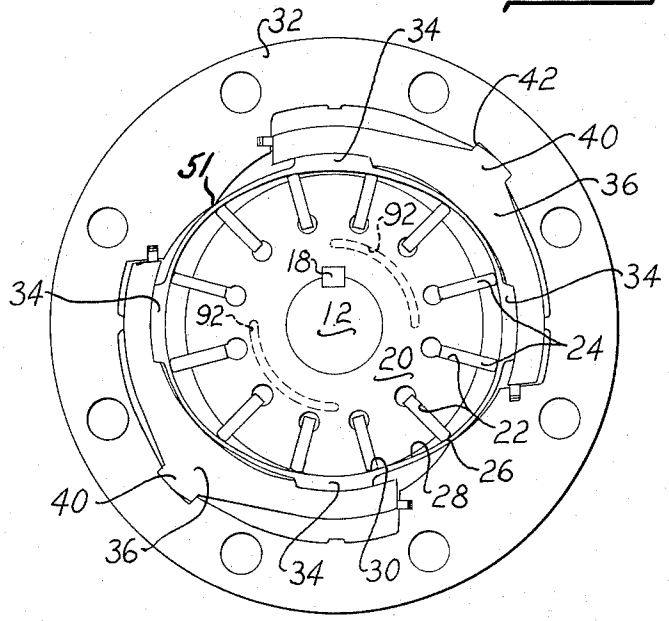


FIG. 2

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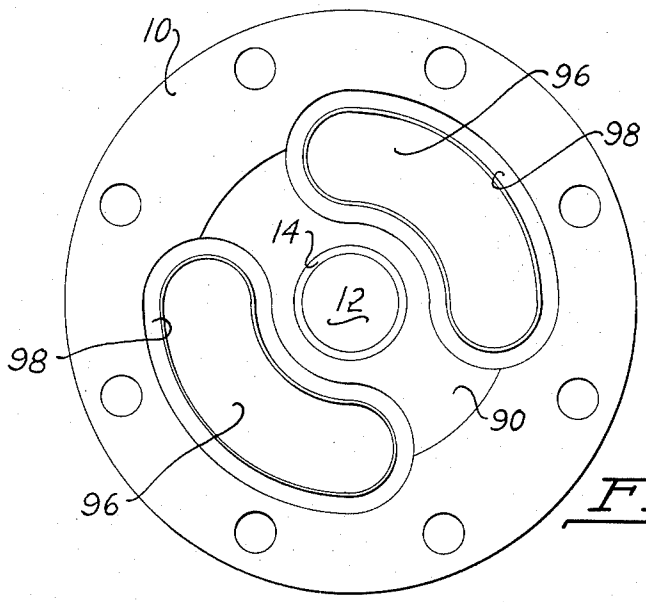


Fig. 3

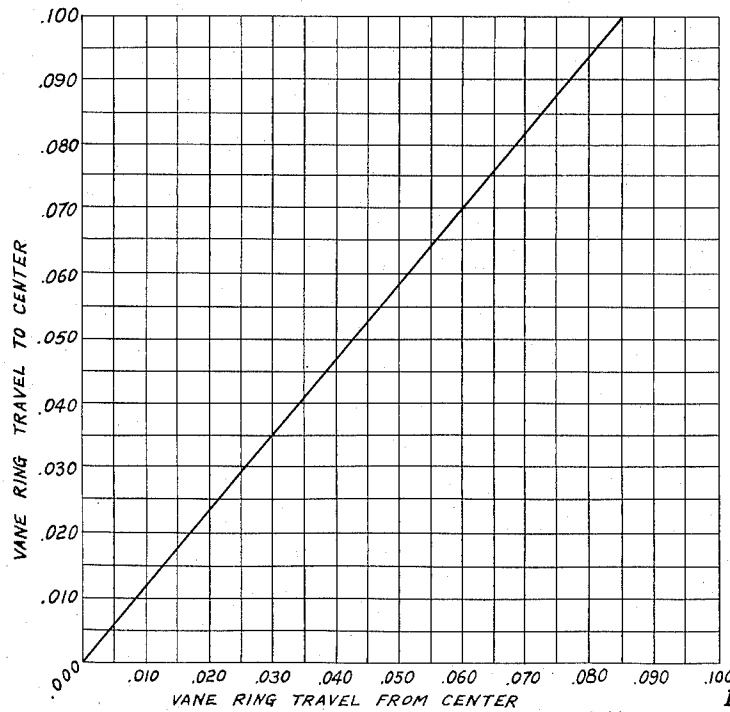


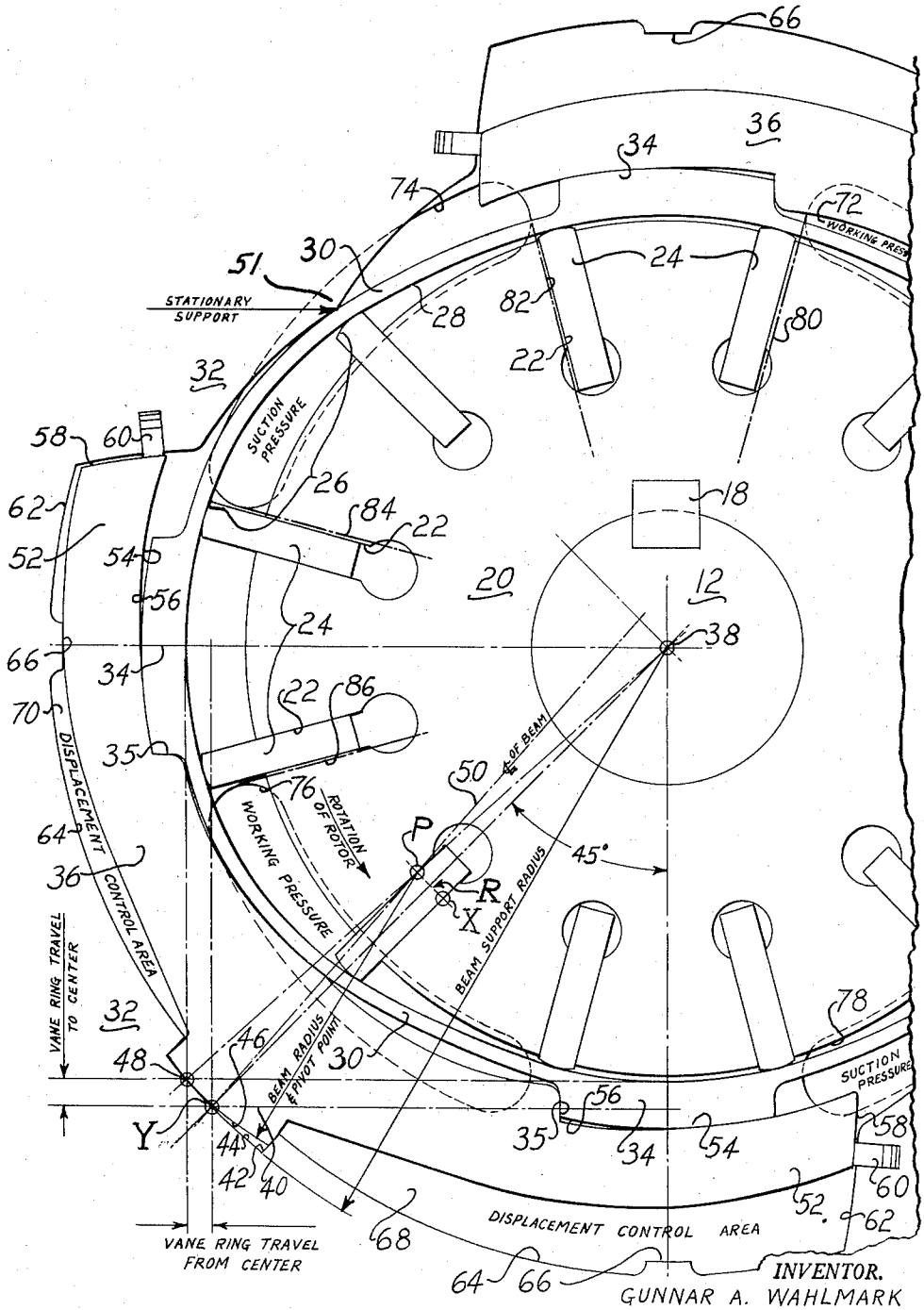
Fig. 5

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Fig. 4

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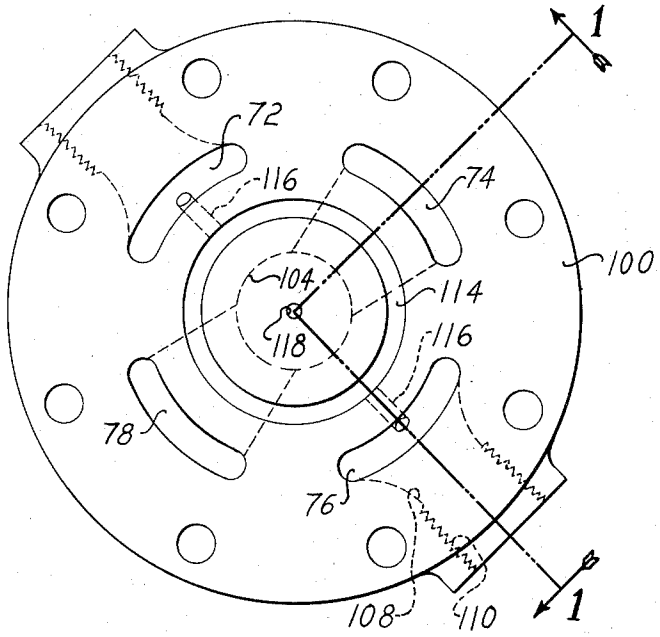


Fig. 6

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HYDRAULIC PRESSURE UNIT

Gunnar A. Wahlmark, Rockford, Ill.

Application May 24, 1954, Serial No. 431,634

9 Claims. (Cl. 103—120)

This invention relates to hydraulic units of the hydrostatic type and has particular reference to a device of this character having variable displacement and being capable of variation in displacement from a maximum positive displacement through neutral to a maximum negative displacement.

As illustrated, the hydraulic unit consists of a pump of the vane type in which the vanes are slidable in a rotor and in contact with a flexible vane ring adapted to be distorted from its neutral circular position into an ellipse, such distortion being arranged to take place selectively upon axes at right angles to one another through the center of the ring. The distortion of the vane ring is accomplished by means of walking beams or rocker arms adapted to be actuated by means of hydraulic pressure to various positions for varying the displacement of the pump.

In the illustrated example of the pump, two such rocker arms are provided approximately 180° apart, although additional rocker arms may be employed in the event additional pumping chambers are desired. In the example illustrated, all hydraulic loads are balanced about the periphery of the pump and no radial load is imposed upon the shaft, thus permitting the use of simple inexpensive bearings while producing extremely high pressures.

An important object is to employ a flexible vane ring which in positions of extreme distortion causes the vane to contact the vane ring over a radius in the sealing areas between ports which represents a true radius from the center of the pump. Moreover, at the areas of sealing the vane ring is arranged so that no sliding action of the vanes takes place and the vane ring is most heavily backed and reinforced for resistance against movement at such points.

It is a further object to arrange the flexible vane ring and the inlet and outlet ports in such a manner that the vane ring is exposed both on its inside and outside diameters to identical pressures in the port areas.

It is a still further object to employ a rocker arm which with the housing constitutes a multiple chamber control member, the rocker arm being pivoted within the housing with the pivot point separating the control chambers. The central portion of the rocker arm adjacent the vane ring is exposed to working pressure, tending to seat said rocker arm upon its pivot point and to balance the rocker arms, thus reducing to a minimum the force required to actuate the rocker and control distortion of the vane ring.

It is a further object of the invention to provide a means for compensating for heavy pressures which is self-adjusting and which compensates only to the extent required by the amount of pump chamber distortion taking place at a given instant.

In the drawings, of which there are four sheets:

Figure 1 is a longitudinal, sectional view through an embodiment of the invention taken on the radial planes of broken line 1—1 of Figure 6 so as to illustrate a pressure port and a suction port.

Figure 2 is a vertical, transverse, sectional view taken substantially upon a plane as indicated by the line 2—2 of Figure 1.

Figure 3 is a transverse, vertical, sectional view taken substantially upon a plane as indicated by the line 3—3 of Figure 1 and showing the compensating pockets.

Figure 4 is an enlarged detailed view similar to Figure 2 and illustrating in greater detail the operation of the pump and the means for adjusting displacement.

Figure 5 is a graph illustrating the ratio of vane ring displacement as between minor and major axes of an ellipse.

Figure 6 is a transverse vertical sectional view taken substantially on a plane as indicated by the line 6—6 of Figure 1 and looking in the direction of the arrows.

In the drawings, the pump is illustrated as comprising a body portion 10, which supports a drive shaft 12 in a plain or sleeve bearing 14, shaft 12 extending past the face 16 of the body portion 10 and being keyed at 18 to a rotor 20. Rotor 20 is provided with a series of radial slots 22 which break through the periphery of the rotor and in which vanes 24 are fiitted for radial sliding movement. The outer ends of vanes 24 are rounded as at 26 for sliding contact with the inside diameter 28 of a vane ring 30.

Vane ring 30 comprises a metallic cylindrical element of the same dimension along the direction of the shaft as the rotor 20 and is housed within a pump housing 32, having approximately the same thickness as the thickness of the rotor 20. Manufacturing operations and good running clearance considerations require that the pump housing 32 be slightly thicker in a matter of a few thousandths of an inch than are the rotor 20 or the vane ring 30.

Vane ring 30, while of a uniform wall thickness, has thickened areas 34 in four locations spaced 90° apart circumferentially of the vane ring. These thickened areas 34 serve a dual purpose; first, to provide a land which may be used as a pressure point for generally distorting the vane ring without producing a local deformation of the ring, and, second, to provide a land which seats against shoulders 35 upon rocker arms 36 to prevent rotation of said vane ring. Thickened areas 34 also insure that such areas upon the inside diameter of the vane ring as lie beneath such thickened areas will not partake of general distortion of the vane ring upon deflection thereof. This insures that no sliding motion of the vanes 24 within the vane slots 22 will take place while the vanes are traveling across the inside diameter of the vane ring in the vicinity of the thickened areas 34, the vane ring being ground so that the inside diameter of the vane ring in such areas represents a true radius from the center 38 of the pump shaft.

General distortion of the vane ring, as previously noted, is accomplished by means of the rocker arms 36, each of which has a lug 40 positioned in a ground slot 42 in the pump housing 32.

Since the distortion of a cylinder into an ellipse by imposing a load upon a diameter of the cylinder will result in a greater displacement along the axis of the load than along the axis at right angles thereto, accommodation has to be made in the unit for backing up the vane ring upon the suction side or at right angles to the loaded axis.

Reference should be had to Figure 5, which comprises a graph showing on the vertical axis the amount of travel of a vane ring toward its center as it is loaded. On the horizontal ordinate the amount of vane ring travel away from the center is indicated. It will be noted that with a deflection toward the center of 0.10 inch, the vane ring travel upon the unstressed axis will be 0.085

inch. The vane ring travel toward the center of 0.10 inch is the maximum for which this particular pump is designed and the vane ring travel away from the center of 0.085 inch is the maximum of which the pump is capable. These travels are determined experimentally by distorting a tubular vane ring of this type and measuring the amount of distortion along both axes of the ellipse. To accommodate the disparity of movements upon the minor and major axes of the ellipse, bearing in mind that a reversal of the pump through a reversal of the major and minor axes of the ellipse requires different degrees of movement along such axes, it is necessary to construct the rocker arm 36 in such a fashion that these variances in movement can be accommodated.

To this end the inwardly facing portion 44 of the slots 42 in the pump housing 32 are ground upon a radius from the center 38 of the pump shaft 12. The outside radius of the rocker arm 36 upon lug 40 is ground at 46 on a true radius from point P, determined geometrically as follows: measuring from the inside diameter of the vane ring in the normal or at rest position, when the vane ring is undistorted, toward the center in the bottom segment of the ring, the amount of desired vane ring travel toward the center, as illustrated in the lower portion of Figure 4, and measuring then again from the normal position of the vane ring in the right hand segment to the desired amount of vane ring travel away from the center, produces a set of lines which intersect at point 48. A radius is drawn from point 48 to the center point 38 of the shaft and this radius intersects at point P at center line 50 of the rocker arm 36. The point P will, of course, move from the location shown to a point X along the arc R as the rocker arm 36 is activated from the position shown to the reverse position of the pump. Again, in so moving from the point P to the point X the vane ring will be returned to neutral or cylindrical condition and then distorted in the reverse direction about the opposite axes to cause the pump to reverse its suction and pressure sides.

Stationary supports 51 upon housing 32 contact the outside diameter of the vane ring intermediate the rocker arms 36, such points on the vane ring being neutral and not partaking of the distortion of the vane ring when the displacement is altered, and serving to maintain the proper configuration and function of the vane ring.

The ends 52 of the rocker arms 36 are adapted to contact the outside diameter of the heavy sections 34 of the vane ring 30 and both are ground in the contacting areas so as to maintain at least 50% of the facing area 54 and the external surface 56 of thickened lands 34 in contact in any position of displacement of the vane ring. The extreme end surfaces 58 of the rocker arms 36 are ground on a radius from the point Y and are in constant contact with sealing blades 60, which are spring pressed into contact with the surface 58 to provide, with the pump housing, control chambers 68 and 70 separated by lugs 40.

In addition to the slot 42, the pump housing 32 is cut away generally as at 62 and 64 to receive the rocker arm 36, the areas 64 being provided with lands 66 which serve to limit the rocking movement of the arm 36. Arm 36 is accurately controlled in thickness so as to be precisely the same thickness as the thickness of the rotor 20 and vane ring 30, thus preventing or inhibiting the leakage of hydraulic fluid past the arm.

When no displacement demands are imposed upon the pump, equal pressures will exist within the area 68 and the area 70 on opposite sides of the lug 46 upon rocker arm 36. Consequently, the outer extremities 52 of the arm 36 will stand at equal distances from the center 38 of the pump shaft and no sliding movement of vanes 24 will take place within the slots 22 of rotor 20 when rotated. Upon the demand for displacement or pressure, a valve mechanism (not shown) will induce the flow of control pressure to the area 68 and will induce the flow of suction pressure to the area 70, causing the rocker arm 36 to move to the position of deflection shown in Figure 4, in which the vertical axis of the vane ring is compressed and the horizontal axis of the vane ring is essentially released from pressure. Under conditions of maximum displacement the outer surface of the rocker arm 36 will contact stop 66 upon the horizontal axis. It should be understood that this same condition prevails upon the rocker arm 36 upon the opposite side of the pump rotor.

It will be seen that pressure ports are provided at 72 and 76, whereas suction ports are provided at 74 and 78. While the vane 24 is moving from the position indicated by center line 80 to the position indicated by the center line 82, it is being subjected to the maximum working pressure. Also across this area, porting between the pressure and suction ports takes place with at least one vane providing a seal against the escape of suction pressure across between the ports. It can consequently be understood that the maximum lateral loading exists upon the vane as it is in the porting or sealing position and is moving across the area between center lines 80 and 82.

From an examination of the drawings it will be noted that the vane is not compelled to slide within the slot 22 while it is moving between these positions and a minimum of vane wear hence occurs at maximum loading. As each vane passes the center line 82, both sides of the vane are exposed to suction pressure from the port 74 and as the vane moves outwardly in the slot 22, following the internal diameter 28 of the vane ring in the distorted position shown in Figure 4, hydraulic fluid will be drawn from port 74 into the pumping chamber formed between the inside diameter of the vane ring, the outside diameter of the rotor 20, the side walls of the pump elements, and the vanes 24.

Again, as each vane leaves the suction port 74, approximately as shown by the center line 84, it is sealed against working pressure from the pressure port 76 by the next adjacent vane 24. When such next adjacent vane 24 crosses the edge of the working pressure port 76 at approximately the center line 86, the following vane will be exposed to working pressure and will in turn take up the function of sealing between working and suction pressure in the area between center lines 84 and 86.

Also, it should be noted that the internal diameter 28 of the vane ring between center lines 84 and 86 forms a true radius from the center 38 of the pump and, consequently, while performing the sealing function and while exposed to maximum pressure differentials, the vanes 24 are not required to perform any sliding movement within slots 22. It should also be noted each of the suction and pressure ports is vented both to the exterior and interior of the vane ring so as to equalize pressures upon opposite sides of the vane ring and to avoid local undesirable distortions from this cause.

This equalization of pressures on inside and outside diameters of the vane ring creates an area between the shoulders 35 of rocker arm 36 which is exposed to working pressure. Since this area extends equally on both sides of the center point Y of the lug 40, the working pressure tends to seat the rocker arm 36 in the slot 42. This balanced arrangement also makes it possible to impart control movement to the rocker arm with a minimum force and eliminates the necessity for using sufficient force to overcome the working pressure in accomplishing a change in displacement.

Referring again to Figure 1, it will be seen that the support housing 10 is provided with a plate 88 which is secured at its outside edge between pump housing 32 and the support housing 10, the support housing being relieved toward the center of the shaft 12, as at 90, to permit the plate 88 to flex in the direction of the axis of

the shaft. Plate **88** is provided with segmental grooves **92** in its surface next to the rotor **20**, each such segmental groove having a circumferential extent equal to the corresponding working pressure segment of the pump under normal conditions of displacement of the pump. Segmental grooves **92** are each joined by a drilled channel **94** to a kidney-shaped compensating area **96** in the support **10** in which a seal **98**, comprising an O ring, is located.

These compensating areas operate as follows: Upon the occurrence of heavy pressures within the pumping chambers the plate **88** will deflect toward the left, as viewed in Figure 1. Such deflection will allow a portion of the fluid under working pressure to escape toward the axis of the shaft **12** from the pumping chamber and into the segmental grooves **92**, whence such pressure will be conducted along the channels **94** into the compensating pockets **96**. The more such leakage occurs, the greater the amount of fluid that will be trapped in the chambers **96** until they become filled and in turn produce a compensating pressure upon the plate **88**, whereupon plate **88** will be deflected toward the right sufficiently to compensate for the initial deflection and to reduce the leakage which otherwise might occur.

It will be understood that this compensating action requires a very slight amount of fluid and results in reduction of running clearances between plate **88** and rotor **20** and, consequently, between rotor **20** and end cap **100**.

End cap **100** is secured by means of cap screws **102** to the shaft support **10**, such cap screws passing through pump housing **32** and plate **88** to secure them firmly in position. End cap **100** is provided with an axial aperture **104** which is internally threaded to receive a suction line and then in turn opens into radially disposed suction channels **106** to conduct the fluid to suction ports **74** and **78** (see Figure 6).

Each pressure port **72** and **76** is connected through a channel **108** to an internally threaded outlet **110**. These two outlets may be used for supplying fluid to different instrumentalities or may be joined externally of the pump. The surface **112** of the end cap **100** adjacent to pump housing **32** and rotor **20** is provided with an annular groove **114** concentric with the center of the pump shaft **12**. This groove is located on the circumference formed by the inner end of the vane slots **22** and is adapted to carry away to the pressure exit ports via the channel **116** fluid subjected to pressure behind each vane within the vane slot.

Since no compensating action occurs at the inside diameter of the plate **88**, any leakage of the pump will result in leakage toward the pump shaft where it will serve to lubricate bearing **14**. Such leakage as does occur will be withdrawn toward the right along the pump shaft between the rotor and pump shaft by means of channel **118**, which is vented into suction inlet chamber **106**.

It will thus be seen that an hydraulic unit has been provided which accomplishes the objects set forth above.

I claim:

1. In a fluid pressure unit having a flexible normally cylindrical vane ring mounted in a working chamber of a housing and having a rotor coaxial with the ring and rotatably mounted in the chamber with a plurality of pumping vanes radially shiftably retained by the rotor and operatively engaging the ring, the improvement comprising a pair of rocker members disposed in said housing on opposite sides of the axis of the rotor and each having a pair of engagement areas engaging said ring at positions 90° apart, means oscillatably mounting said members in said housing, and means for actuating said members for shifting the respective engagement areas in opposite radial directions to deflect said ring elliptically along axes defined between diametrically opposite pairs of engagement areas whereby either of the axes becomes a major elliptical axis while the other becomes a minor elliptical axis.

2. A fluid pressure unit according to claim 1 wherein said means oscillatably mounting said members comprise a lug on each of said members intermediate the respective engagement areas and a pair of diametrically opposite fulcrums formed on said housing intermediate said elliptical axes and supporting the respective lugs.

3. A fluid pressure unit according to claim 1 wherein said means oscillatably mounting said members comprise a pair of diametrically opposite fulcrums formed in said housing intermediate said elliptical axes and having opposed support faces formed as segments of cylinders coaxial with said rotor, and a lug on each of said members intermediate the respective areas, each of said lugs having a segmental cylindrical surface engaging the respective fulcrum support faces with the axis of each of said surfaces being parallel to the axis of the rotor and passing through a point determined by the intersection of the center line of the respective member with a radius of the rotor passing through a point determined by the intersection of tangents to the vane engaging surface of said ring taken at the major and minor elliptical axis points on the vane engaging surface.

4. A fluid pressure unit according to claim 1 wherein integral support ridges are formed on said housing and engage and support said ring at four points equally spaced between said elliptical axes whereby said vane ring is supported at each of said four points throughout all deflection of the ring.

5. In a fluid pressure unit having a flexible normally cylindrical vane ring mounted in a working chamber of a housing and having a rotor coaxial with the ring and rotatably mounted in the chamber with a plurality of pumping vanes radially shiftably retained by the rotor and operatively engaging the ring, the improvement comprising ring deflecting means engaging said ring at four positions 90° apart, and means continuously supporting said ring at four neutral points respectively spaced between said four positions whereby said vane ring is supported at each of said neutral points throughout all deflection of the ring.

6. A fluid pressure unit according to claim 5 wherein said supporting means comprise integral support ridges formed on said housing and engaging and supporting said ring at said four neutral points equally spaced between said four positions.

7. In a fluid pressure unit having a flexible normally cylindrical vane ring mounted in a working chamber of a housing and having a rotor coaxial with the ring and rotatably mounted in the chamber with a plurality of pumping vanes radially shiftably retained by the rotor and operatively engaging the ring, the improvement comprising a pair of rocker members disposed in said housing on opposite sides of the axis of said rotor and each having a pair of arms with engagement areas engaging said ring at positions 90° apart, means oscillatably mounting said members in said housing, means for actuating said members for shifting the respective engagement areas in opposite radial directions to deflect said ring elliptically, fluid pressure inlet and outlet means communicating with said chamber between said rotor and said ring, said fluid pressure inlet and outlet means being located to apply equal pressure force against each arm of each of said members to reduce the force required for actuating the members.

8. In a fluid pressure unit having a flexible normally cylindrical vane ring mounted in a working chamber of a housing and having a rotor coaxial with the ring and rotatably mounted in the chamber with a plurality of pumping vanes radially shiftably retained by the rotor and operatively engaging the ring, the improvement comprising a pair of rocker members disposed in said housing on opposite sides of the axis of said rotor and each having a pair of arms with engagement areas en-

gaging said ring at positions 90° apart, means oscillatably mounting said members in said housing, fluid pressure inlet and outlet means communicating with said chamber between said rotor and said ring, and means for applying pressure force behind either pair of diametrically opposite arms of the respective members for moving the opposite pairs of engagement areas in opposite radial directions to deflect said ring elliptically along axes defined between diametrically opposite pairs of engagement areas whereby either of the axes becomes a major elliptical axis while the other of the axes becomes a minor elliptical axis.

9. In a fluid pressure unit having a flexible normally cylindrical vane ring mounted in a working chamber of a housing and having a rotor coaxial with the ring and rotatably mounted in the chamber with a plurality of pumping vanes radially shiftably retained by the rotor and operatively engaging the ring, the improvement comprising ring deflecting means engaging said ring at four positions 90° apart, said housing having a first end cap portion engaging one face of said rotor to close one side of said chamber, angularly spaced fluid pressure inlet and outlet ports formed in said one end cap portion and communicating with said chamber between said rotor and said ring, a deflectible pressure plate engaging the other face of said rotor to close the other side of said chamber, said housing having a second end cap portion engaging said pressure plate in the side opposite to said rotor, a segmental channel formed in said pressure plate radially adjacent said pressure outlet port and normally blocked by said pressure plate from communicating with the outlet port, a recess formed in said second end cap adjacent said pressure plate axially opposite said outlet port, and a passage connecting said channel and said recess, whereby said pressure plate is deflected away from said rotor by the fluid pressure adjacent said outlet port to permit fluid pressure leakage to said channel and through said passage into said recess to deflect said pressure plate toward said rotor to reduce leakage.

References Cited in the file of this patent

UNITED STATES PATENTS

1,190,139	Ford	July 4, 1916
1,972,632	Patton	Sept. 4, 1934
2,016,315	Calzoni	Oct. 8, 1935
2,176,322	Barrett	Oct. 17, 1939
2,535,267	Cline	Dec. 26, 1950
2,538,193	Ferris	Jan. 16, 1951
2,538,194	Ferris	Jan. 16, 1951
2,646,003	Ferris	July 21, 1953
2,673,448	Wheeler	Mar. 30, 1954
2,691,482	Ungar	Oct. 12, 1954

FOREIGN PATENTS

814,240	Germany	Sept. 20, 1951