

[54] **PRESSURE BALANCING SYSTEM FOR GEAR PUMPS OR MOTORS**

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[21] Appl. No.: **472,371**

[52] U.S. Cl. **418/132; 418/189**

[51] Int. Cl.² **F04C 15/00**

[58] Field of Search **418/131, 132, 133, 189**

[56] **References Cited**

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Assistant Examiner—Leonard Smith

Attorney, Agent, or Firm—McNenny, Farrington, Pearne & Gordon

[57] **ABSTRACT**

A hydraulic gear pump or motor is illustrated in which pressure plates are positioned adjacent to the ends of the gears to provide a dynamic seal therewith. Each pressure plate is provided with a 3-shaped groove on the pressure plate side in which is positioned a U-shaped rubber seal. The seal separates the pressure side of the pressure plate into a high pressure zone in communication with the high pressure port of the device and a low pressure zone in communication with the low pressure port of the device. The seal is pressurized during gear rotation with liquid that is entrapped and pressurized in the zone of intermeshing of the gears. Two orifices communicate between the pressure seal and the high pressure zone of the device to controllably bleed liquid from the seal and prevent excessive pressures from occurring.

14 Claims, 11 Drawing Figures

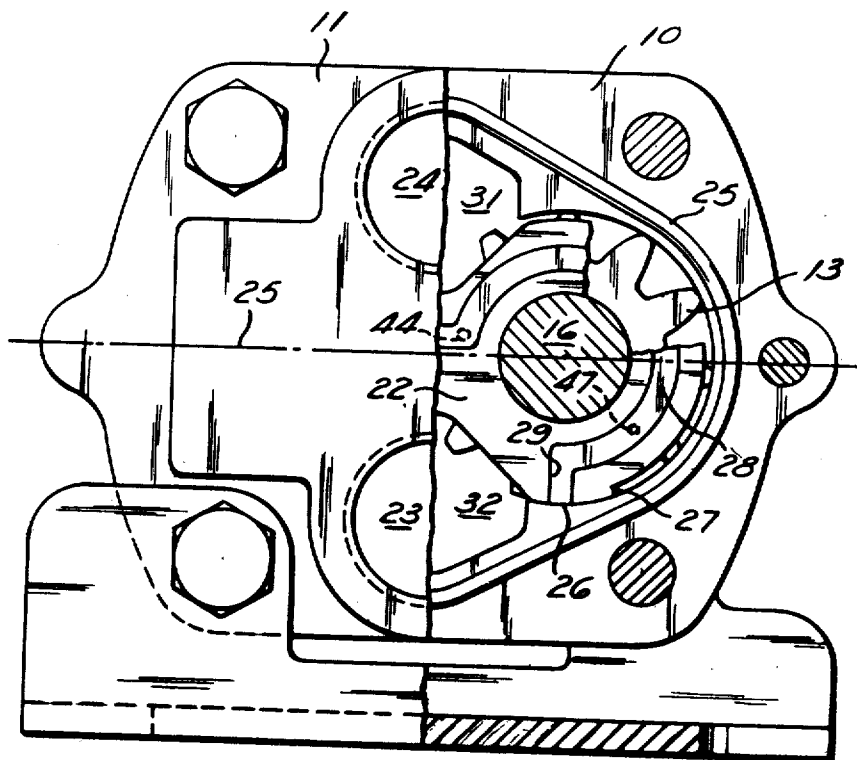


Fig. 1

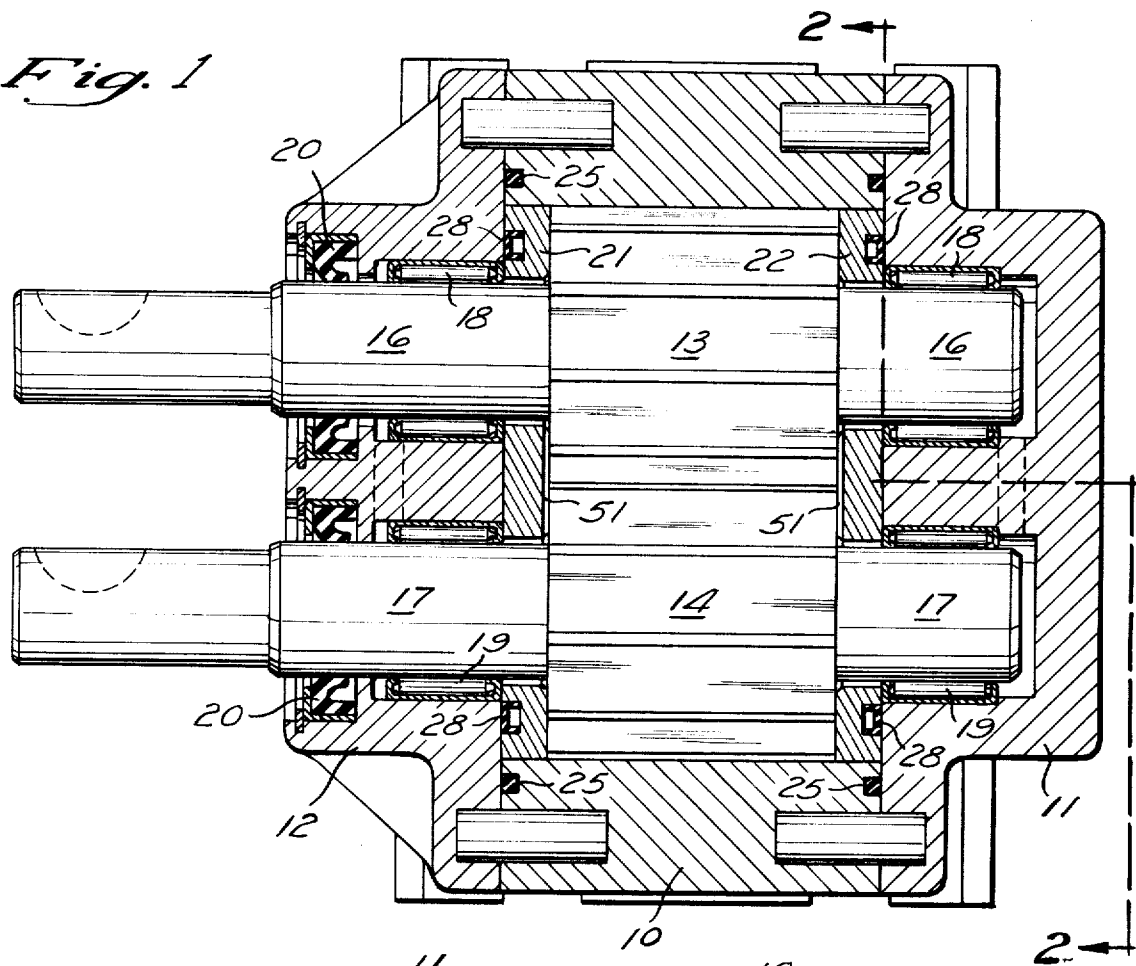


Fig. 2

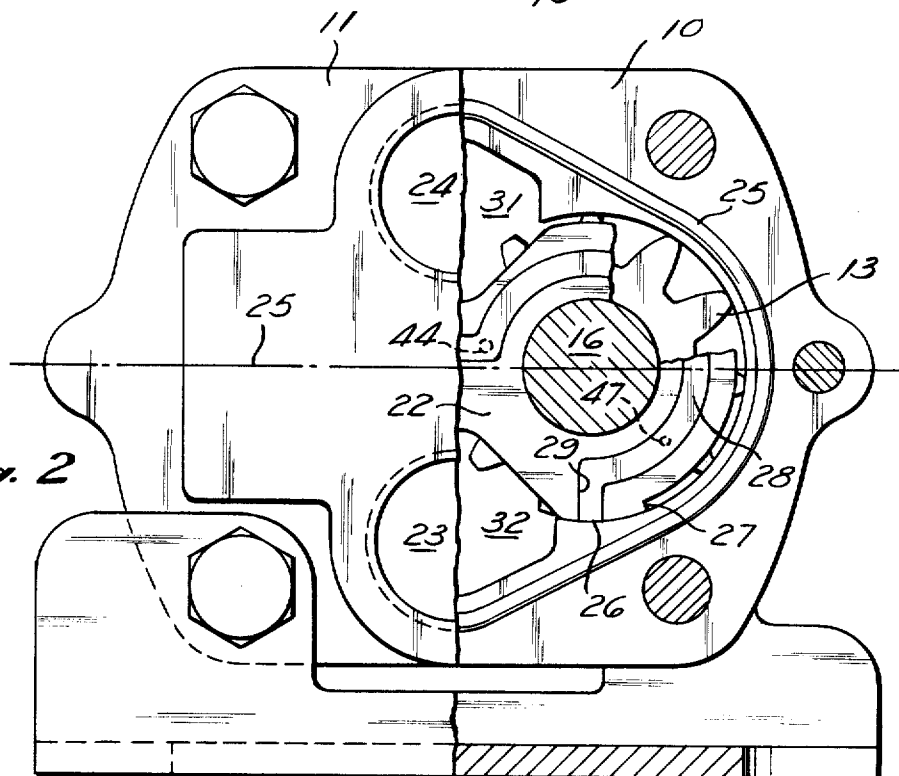


Fig. 3

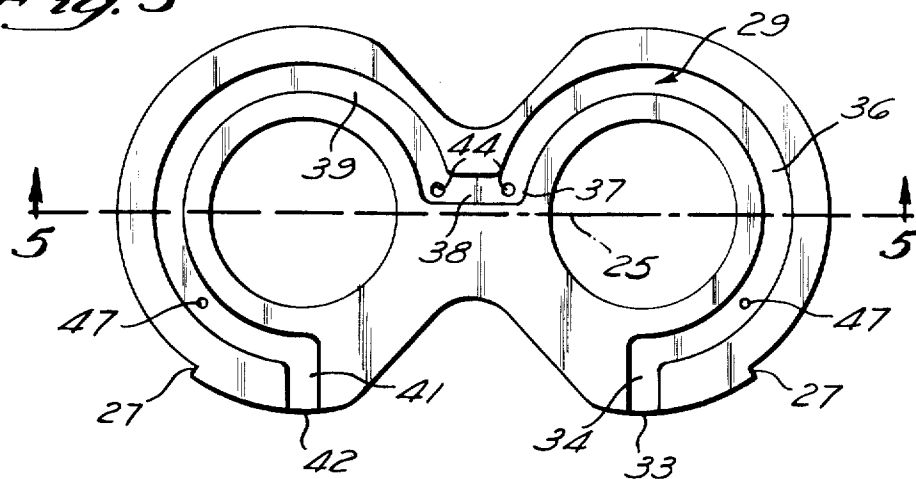


Fig. 5

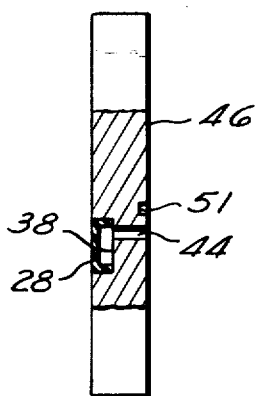
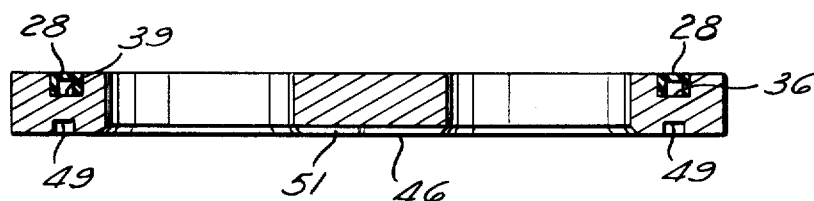


Fig. 6

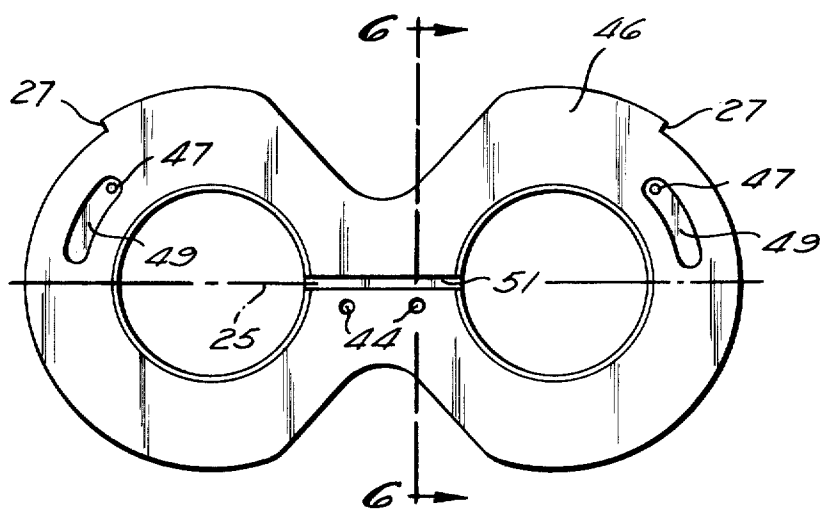


Fig. 4

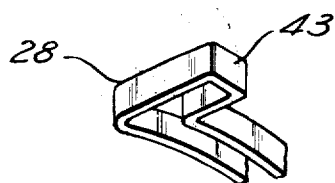


Fig. 7

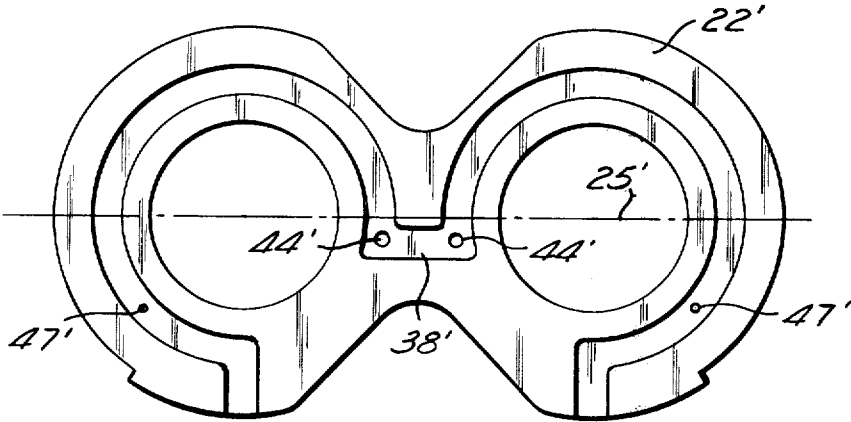
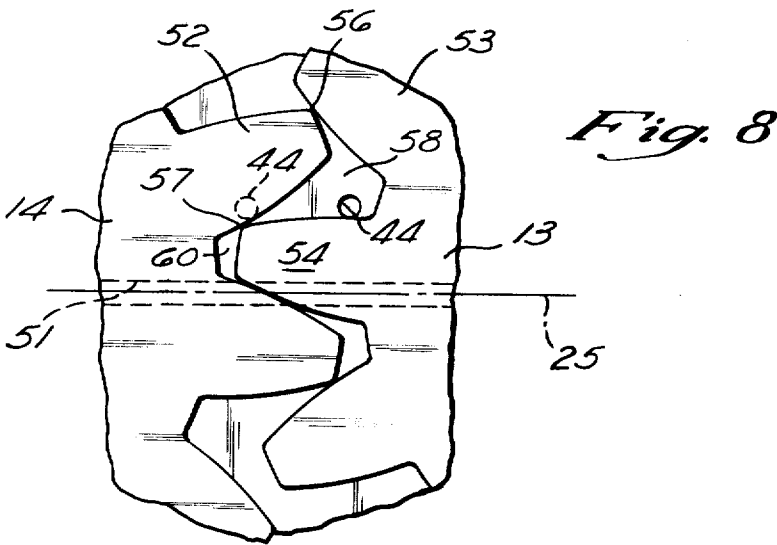


Fig. 9

Fig. 10

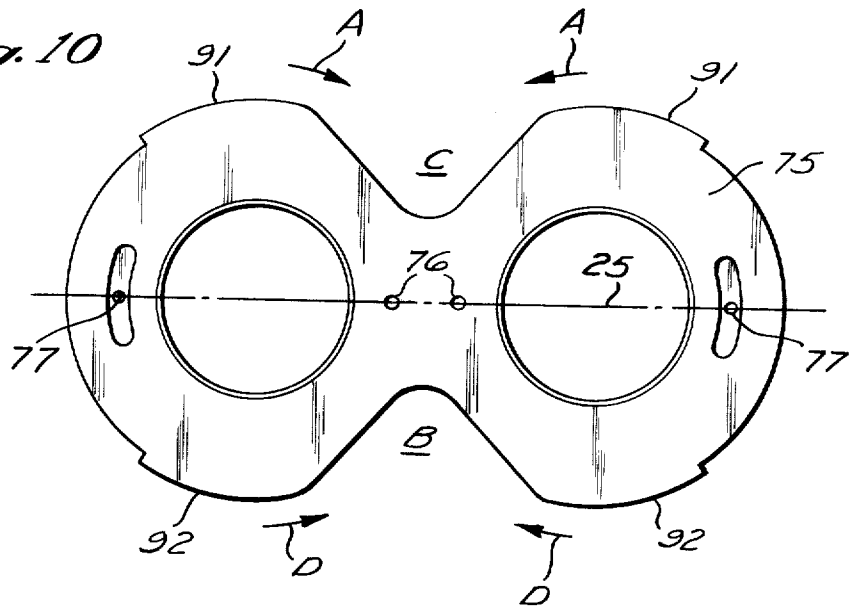
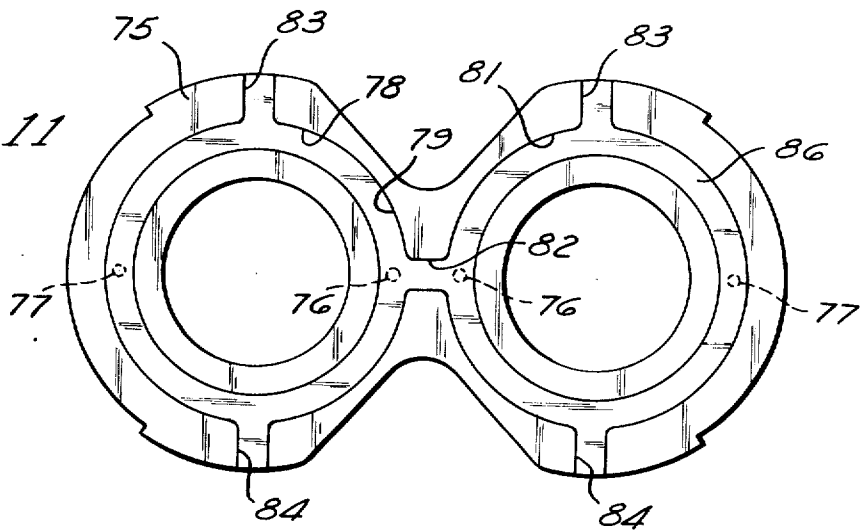


Fig. 11



PRESSURE BALANCING SYSTEM FOR GEAR PUMPS OR MOTORS

BACKGROUND OF THE INVENTION

This invention relates generally to hydraulic gear pumps, motors, or the like and more particularly to such hydraulic devices having novel and improved means for pressure loading the sealing bushings or end plates.

PRIOR ART

Hydraulic gear motors or pumps are often provided with pressure-loaded bushings or end plates which engage and provide a dynamic seal with the ends of the rotating gears. In such devices, pressure from selected portions in the device is supplied to one or more loading zones on the side of the plate opposite the gears so that the loading of the plate is a function of the particular pressure condition existing in the operating device. An attempt is made in such systems to provide a load on the end plates which produces the desired seal with the associated gears without excessive force which can, of course, result in excessive frictional losses and wear.

In some instances, the pressure-loaded plates engage the adjacent ends of the gears to the commencement of the gear rotation with sufficient force to cause high breakaway torques. In other instances, the loading of the pressure plate is sufficiently high to produce excessive friction losses that result in decreased operating efficiencies.

SUMMARY OF THE INVENTION

A hydraulic pump or motor in accordance with the present invention incorporates a pressure loading system for the end plates which eliminates high breakaway torques in motor units and which provides optimum sealing with the ends of the gears during operation of the device.

In the embodiments of this invention illustrated, the pressure plates are provided with pressure seal means which separate the pressure side of the pressure plate into two separate zones and which provides a loading force which is a function of the entrapped pressure produced by the inter-meshing of the gears. The seal means are connected by ports to the meshing zone of the gears where entrapped liquid is pressurized during gear rotation. The seal means are also connected by a flow restricting orifice to the high pressure zone of the device.

The system, even when applied to a motor, functions to prevent any substantial loading of the pressure plate until the gears start to rotate. After gear rotation is established, the entrapped liquid under pressure is communicated to the seal to provide the desired loading of the pressure plates. As the speed of the motor increases, the entrapped pressure tends to increase and the restriction orifice connecting the seal with the high pressure portion of the device functions to prevent excessive pressures from being developed.

In the illustrated embodiments of this invention, the pressure plates are supplied with pressure from the zone in which the gears intermesh and entrap fluid. The entrapped fluid is supplied to a floating U-shaped seal located in a 3-shaped groove in the pressure side of the pressure plate. Since the entrapped fluid pressure is a function of the speed of gear rotation, the resulting pressure induced force on the pressure plate is essen-

tially zero until gear rotation commences. This insures that the device does not have a high breakaway friction. However, as the speed of the device increases, the pressure-induced force increases to insure optimum sealing against the ends of the gears.

The seal is also connected through an orifice to the high pressure zone of the device. This prevents excessive pressures from occurring in the seal and thereby prevents the occurrence of excessive pressure loading of the pressure plates against the ends of the gears.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a section taken along the axis of the two gears with the gears themselves and their supporting shafts illustrated in full view;

FIG. 2 is an end view taken generally along 2—2 of FIG. 1;

FIG. 3 illustrates the pressure side of one of the end plates with the floating seal removed for purposes of illustration;

FIG. 4 illustrates the pressure plate of FIG. 3 from the opposite side thereof;

FIG. 5 is a section taken along 5—5 of FIG. 3;

FIG. 6 is a section taken along 6—6 of FIG. 4;

FIG. 7 is a fragmentary perspective view illustrating the end of the floating seal;

FIG. 8 is an enlarged fragmentary view illustrating the meshing portions of the gears;

FIG. 9 illustrates a pressure plate modified for use in a motor device;

FIG. 10 illustrates a pressure plate modified for use in a bi-rotational motor; and

FIG. 11 illustrates the seal side of the pressure plate of FIG. 10.

DETAILED DESCRIPTION OF THE DRAWINGS

FIGS. 1 and 2 illustrate the assembled pump having a main housing 10 which is closed at one end by an end cap 11 and at its other end by an end cap 12. These three elements cooperate to provide the housing assembly in which the gears are journaled and rotate. A pair of meshing gears 13 and 14 are provided with shaft extensions 16 and 17, respectively, which are journaled in roller bearings 18 and 19, respectively. One end of each shaft extension 16 and 17 extends beyond the end cap 12 and provides means for connecting the pump to a suitable power drive. A pressure plate 21 is positioned against the left ends of the gears 13 and 14 as viewed in FIG. 1, and a similar pressure plate 22 engages the right ends of the gears. Seals 20 prevent leakage of fluid out along the projecting shaft portions. In addition, a molded seal 25 is mounted in each end face of the main housing to prevent leakage between the main housing and the two end caps 11 and 12.

Referring to FIG. 2, the end cap 11 is provided with an inlet port 23 which is connected to a source of hydraulic fluid and an outlet or high pressure port 24 connected to the system supplied by the pump.

The pump is driven so that the gear 13 rotates in a counterclockwise direction as viewed in FIG. 2 to carry the hydraulic fluid from the low pressure port 23 past a sealed zone at 26 to the high pressure zone of the pump. In the illustrated pump, the high pressure zone extends completely around the periphery of the gear chamber from the outlet port 24 to each seal zone 26. To insure that the pressurization of the fluid takes place at the seal zone 26, the pressure plate is relieved from

the points 27 along the remaining periphery of the device.

A floating pressure seal 28 is positioned in a 3-shaped groove 29 on the pressure side of the pressure plate to separate the high pressure zone 31 from the low pressure zone 32. The floating seal 28 is U-shaped in section as described in greater detail below and functions to produce the necessary pressure-induced engagement between the pressure plate and the adjacent ends of the gears.

The shape of the entire groove 29 is best illustrated in FIG. 3. the groove extends from one peripheral portion 33 along a radial section 34 and then along an arcuate portion 36 to a location 37 substantially adjacent to a center plane 25 containing the axis of rotation of the two gears 13 and 14. The portion of the groove 38 extends parallel to the plane 25 to a second arcuate section 39. The groove is provided with a second radial section 41 joining the end of the arcuate section 39 with the periphery at 42. The seal 28 is molded to fit along the entire length of the groove 29 from one peripheral portion 33 to the other peripheral portion 42. Preferably, the seal is provided with a U-shaped section best illustrated in FIGS. 5 and 7. The ends of the seal at the two peripheral portions 33 and 42 are closed by an end wall 43 illustrated in FIG. 7.

A pair of ports 44 extend from the sealing face 46 of the pressure plates to the groove at locations adjacent the ends of the straight section 38 of the groove 29. Similarly, a smaller port or orifice 47 is provided to communicate with each of the arcuate sections 36 and 39. The sealing face is provided with an arcuate relief 49 at the pressure face side of each of the orifices 47. A shallow groove 51 is formed in the sealing face 46 of the end plate which extends along a plane 25.

Referring to FIG. 8, as the teeth of the gear 13 move into the meshing zone with the teeth of the gear 14, engagement of the tooth 52 of the gear 14 occurs with two adjacent teeth 53 and 54 of the gear 13. This engagement occurs at 56 and 57. The teeth then cooperate to form an entrapment zone 58 which decreases in volume as the gears continue to rotate. Consequently, the entrapped fluid is pressurized as the entrapment zone moves past the associated port 44 causing the pressurized entrapped fluid to flow through the ports 44 into the passage defined by the floating seal 38. This causes the floating seal 28 to be pressurized and to produce pressure-induced force on each pressure plate urging the pressure plates toward the adjacent ends of the gears to provide a sealing contact therewith. The entrapped zone 60 is illustrated after it moves past the other port 44 and before it reaches the groove 51. After the entrapment zones pass beyond the ports 44, they moved into communication with the groove 51 which functions to relieve the pressure of the entrapped fluid and provide flow to lubricate the bearings.

There is no pressurized entrapped fluid before the two gears 13 and 14 commence to rotate, so the seal 28 is not pressurized before operation of the pump is commenced. As the gears 13 and 14 commence to rotate, they commence to entrap and pressurize fluid to cause pressurization of the floating seal 28. This causes a force on each of the pressure plates causing a sealing engagement between the seal face 46 of each pressure plate and the associated ends of the gears 13 and 14. As the speed of the pump increases, the pressure of the entrapped fluid communicated to the seal also in-

creases. However, since the entrapped fluid is initially at the pressure of the outlet of the pump at the moment it becomes entrapped, the entrapped fluid pressure always exceeds the outlet pressure of the pump.

In addition, the illustrated embodiment provides automatic compensation for variations in output pressure of the pump. The rate of flow of the fluid from the seal through the two orifices 47 is a function of the differential pressure across such orifices. Therefore, as the pump pressure increases causing the differential pressure to decrease, the rate of flow through the orifices decreases and causes the pressure in the seal to increase until equilibrium is again reached. Therefore, the pressure induced force varies with the output pressure of the pump and greater forces are available when greater forces are required by increased pressure. Conversely, when the pump pressure decreases, the seal pressure decreases reducing the pressure induced force on the pressure plate at a time when reduced forces are required for proper operation of the device.

In one instance, a pump was constructed and tested where the ports had a diameter of 0.060 inches, and the ports or orifices 47 were formed with a diameter of 0.030 inches. In such devices where the diameter of the orifices 47 was one-half of the diameter of the ports 44, the pressure within the seal tended to approach the entrapment pressure. However, excessive pressures do not occur, because the two orifices continue to bleed fluid from the seal to the high pressure zone 31. In such device, mechanical efficiencies of about 95% were obtained throughout the range of 1000 RPM to 3000 RPM at rated pressures from 2000 PSI to 3000 PSI. These results demonstrate that a very good seal is obtained between each pressure plate and the associated gear end, but that the force produced by the pressure in the seal 28 did not result in excessive friction which would, of course, materially reduce the mechanical efficiency.

If the device of the general type illustrated in FIGS. 1 and 2 is to be operated as a motor, the supply pressure is connected to the port 24 and the exhaust line or outlet line is connected to the port 23. In such instance, the gear 13 rotates in a clockwise direction, while the gear 14 rotates in a counterclockwise direction.

In the motor embodiment, the entrapment of fluid occurs below the center line 25. Therefore, a pressure plate as illustrated in FIG. 9 is used. In FIG. 9, similar reference numerals are used for similar parts, but a prime (') is added to signify that reference is made to the embodiment of FIG. 9. The groove of both of the pressure plates is modified, as shown on pressure plate 22', so that the section 38' is located below the center line 25', and the two ports 44' are, therefore, positioned below the center line. In instances in which the invention is applied to a motor, the improved results of low breakaway friction are obtained even though the seal is pressurized from the supply pressure through the orifices 47' prior to gear rotation. Any tendency for the pressure to build up in the seal is prevented by leakage through the ports 44'. After gear rotation starts, the entrapped fluid pressure builds up to pressurize the seal. In such a system in which the invention is applied to a motor, the sealing pressure again increases as a function of speed and maintains proper sealing without excessive friction.

FIGS. 10 and 11 illustrate a third form of pressure seal incorporating the present invention which is par-

ticularly adapted for use in a bi-directional or bi-rotational motor. In this embodiment, the ports 76 are located on the center line 25, and the orifices 77 are also located on the center line 25. In this embodiment, the pressure plate 75 and the seal are symmetrical with respect to the center line 25, and the grooves 51 of the prior embodiments are eliminated. Referring to FIG. 11, the pressure side of the pressure plate 75 is formed with an 8-shaped groove 78 having two circular sections 79 and 81 joined by a central portion 82. Extending in opposite directions from each of the circular portions 79 and 81 are grooved projections 83 and 84. A mating seal 86 having a U-shaped section similar to the earlier embodiments is positioned in the groove 78 and is pressurized to a pressure determined by the entrapped pressure entering the seal through the ports 76 and the rate of flow out of the seal through the orifices 77.

When a pressure plate as illustrated in FIG. 10 is installed in a motor in which the gears rotate in a direction of the arrows A, high pressure is supplied in the zone B and the zone C is connected to the exhaust of the motor. The pressure on the lower side of the center line 25, as illustrated in FIG. 10, causes the gears and the pressure plate 75 to shift upward, as viewed in the drawings, within the motor housing. Therefore, the high pressure in the gear pockets is maintained until they pass the peripheral section 91. Consequently, the orifice 77 is open to the high pressure zone within the motor. In this embodiment, in which the ports 76 are located on the center line 75, the pressure of the entrapped fluid is not as great as it is in the previous two embodiments since the rate of decrease in volume of the cavities enclosing the entrapped liquid is not as great at the moment the ports are open to such cavities as in the previous two embodiments. Consequently, in a given installation, the area of the seal 86 should be somewhat greater to provide the necessary sealing force in this embodiment.

It is recognized that the ports 76 remain in communication with the entrapped liquid as the cavity passes the center line and commences to increase in volume. However, in practice, this does not produce excessive loss of pressure in the seal because of the dynamic conditions existing in an operating motor. For example, the effect of compression of the liquid as it is entrapped and pressurized tends to maintain the pressure as the entrapped cavities pass the center line, and the inertia of the liquid flowing into the seal tends to maintain the entrainment pressure in the seal.

When the opposite direction of rotation is desired, as indicated by the arrows D, the high pressure supplied to the zone C and the zone B is connected to exhaust. This causes a shift of the gears and the pressure plate downward as viewed in FIG. 10 so that the pressure drop of the liquid in the gear pockets occurs at the peripheral portions 92. Consequently, the orifices 77 are open to the high pressure zone of the motor regardless of the direction of rotation of the motor.

Although preferred embodiments of this invention are illustrated, it should be understood that various modifications and rearrangement of parts may be resorted to without departing from the scope of the invention disclosed and claimed herein.

What is claimed is:

1. A hydraulic pump, motor or the like comprising a housing assembly providing a gear chamber with high

and low pressure ports communicating therewith at spaced locations, a pair of meshing gears in said chamber, a pressure plate adjacent each end of said gears, each pressure plate providing a seal face sealing with the adjacent end of said gears and a pressure force opposite said seal face, said gears when rotating entrapping and pressurizing liquid in a zone where the gears intermesh, pressure seal means in each pressure face operating when pressurized to produce a force urging the associated pressure plates toward sealing engagement with said gears, said seal means dividing said pressure faces into low pressure zones in communication with said low pressure port and high pressure zones in communication with said high pressure port, first passage means connected to supply entrapped liquid to pressurize said seal means, and second passage means smaller than said first passage means connecting one of said pressure zones and seal means for flow in both directions therebetween.

2. A hydraulic pump, motor or the like comprising a housing assembly providing a gear chamber with high and low pressure ports communicating therewith at spaced locations, a pair of meshing gears in said chamber, a pressure plate adjacent to at least one end of said gears, said pressure plate providing a seal face sealing with the adjacent end of said gears and a pressure face opposite said seal face, said gears when rotating entrapping and pressurizing liquid in a zone where the gears intermesh to a pressure higher than the pressure in either of said pressure ports, pressure seal means in said pressure face operating when pressurized to produce a force urging said pressure plate toward sealing engagement with said gears, said seal means dividing said pressure face into a low pressure zone in communication with said low pressure port and a high pressure zone in communication with said high pressure port, first passage means connected to supply entrapped liquid to pressurize said seal means, and second passage means smaller than said first passage means connecting one of said pressure zones and seal means for flow in both directions therebetween, whereby the pressure in said seal means exceeds the pressure in said one of said pressure zones during operation.

3. A hydraulic pump, motor or the like as set forth in claim 1 wherein said second passage means connects said high pressure zone to said seal means.

4. A hydraulic pump, motor or the like as set forth in claim 1 wherein said pressure plate is formed with a 3-shaped groove open through said pressure face, and said seal means is an elongated U-shaped resilient seal positioned in said groove.

5. A hydraulic pump as set forth in claim 4 wherein said first passage means includes a first pair of ports having a first predetermined diameter and said second passage means includes a second pair of ports having a diameter substantially equal to one-half of said predetermined diameter and said second pair of ports are open to said high pressure zones.

6. A hydraulic pump, motor or the like as set forth in claim 1 wherein said first passage means is open to entrapped liquid during a first portion of the entrainment, and said pressure plate is provided with relief means to release said entrapped liquid after it is no longer in communication with said first passage means.

7. A hydraulic pump, motor or the like as set forth in claim 6 wherein said relief means is a groove in said seal face along a central plane through the axes of said

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gears, and said first passage means are open through said seal face on the side of said central plane along which said intermeshing gears approach said central plane.

8. A hydraulic pump or the like as set forth in claim 7 wherein said gears operate to pump liquid from said low pressure port to said high pressure port, and said first passage means includes two passages open through said seal face on the high pressure side of said central plane.

9. A hydraulic pump or the like as set forth in claim 8 wherein said second passage means connects said high pressure zone and said seal means.

10. A hydraulic pump or the like as set forth in claim 7 wherein said gears are driven by liquid entering said high pressure port and exhausting from said low pressure port, and said first passage means includes two passages open through said seal face on the low pres-

sure side of said central plane.

11. A hydraulic motor as set forth in claim 10 wherein said second passage means connects said high pressure zone and said seal means.

12. A hydraulic pump, motor or the like as set forth in claim 1 wherein said first passage means are ports located along a central plane containing the axes of rotation of said gears.

13. A hydraulic pump, motor or the like as set forth in claim 12 wherein said seal means is generally 8-shaped and is symmetrical with respect to said central plane.

14. A hydraulic pump, motor or the like as set forth in claim 13 wherein said seal means is provided with projections extending from the 8-shaped portions in opposite directions therefrom.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

PATENT NO. : 3,904,333
DATED : September 9, 1975
INVENTOR(S) : Ulrich W. Stoeckelmann

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Under the caption "PRIOR ART", Column 1, Line 25, after "gears" insert -- prior -- .

Under the caption "DETAILED DESCRIPTION OF THE DRAWINGS", Column 3, Line 54, change "moved" to -- move -- .

Under the caption "DETAILED DESCRIPTION OF THE DRAWINGS", Column 4, Line 33, change "There" to -- These -- .

Under the caption "What is claimed is:", Column 6, Claim 3, Line 45, change "1" to -- 2 -- .

Under the caption "What is claimed is:", Column 6, Claim 4, Line 48, change "1" to -- 2 -- .

Under the caption "What is claimed is:", Column 6, Claim 6, Line 60, change "1" to -- 2 -- .

Column 7, Claim 10, Line 14, change "pump" to -- motor -- .

Column 8, Claim 12, Line 6, change "1" to -- 2 -- .

Signed and Sealed this

sixth Day of January 1976

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents and Trademarks

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,904,333 Dated September 9, 1975

Inventor(s) Ulrich W. Stoeckelmann

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Under the caption "PRIOR ART", and before the caption "SUMMARY OF THE INVENTION", following the second paragraph under "PRIOR ART", the third paragraph was entirely omitted and should be inserted after the second paragraph under the caption "PRIOR ART", as follows: Insert -- The following patents disclose various hydraulic pumps or motors having pressure-loaded end plates therein: 2,319,374; 2,884,864; 2,885,965; 2,981,200; 3,003,426; 3,029,739; 3,043,230; 3,137,238; 3,145,661; 3,174,408. --.

Signed and Sealed this
thirteenth **Day of** *April* 1976

[SEAL]

Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents and Trademarks