

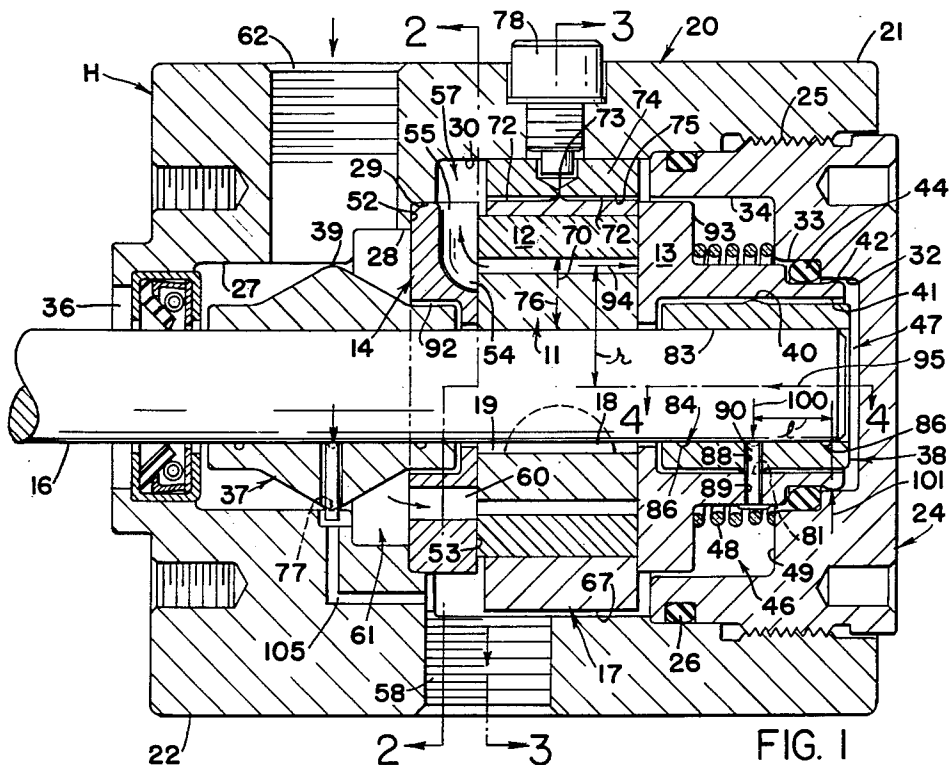
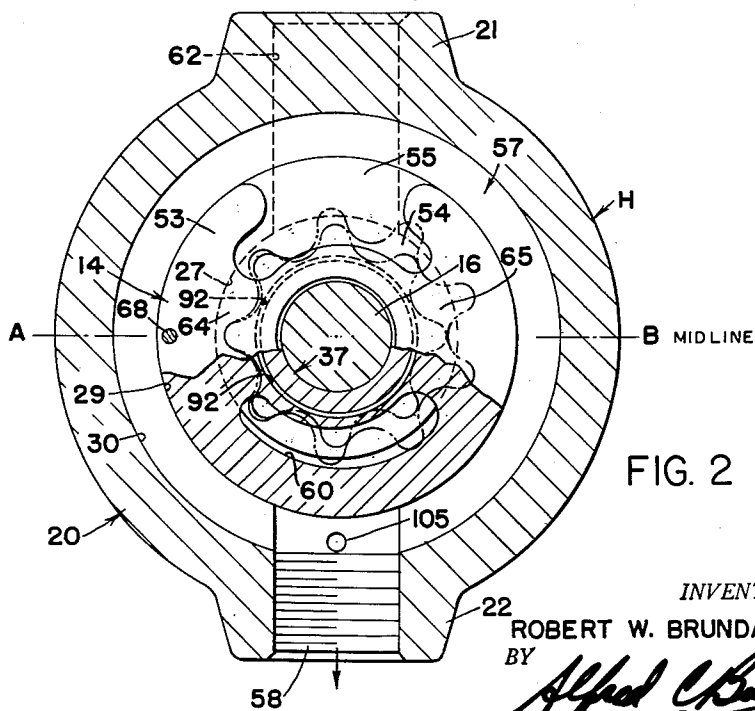
May 15, 1962

R. W. BRUNDAGE
HYDRAULIC PUMP OR MOTOR

3,034,447

Filed May 19, 1959

3 Sheets-Sheet 1



INVENTOR.

ROBERT W. BRUNDAGE

BY

Alfred C. Brady

ATTORNEY

May 15, 1962

R. W. BRUNDAGE
HYDRAULIC PUMP OR MOTOR

3,034,447

Filed May 19, 1959

3 Sheets-Sheet 3

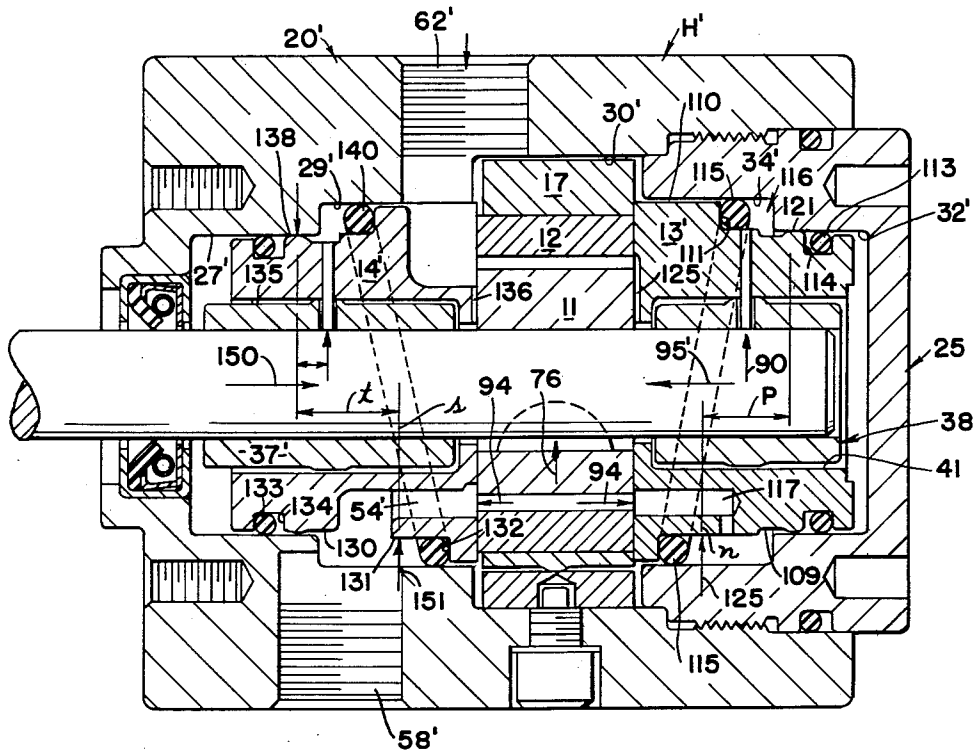


FIG. 5

INVENTOR.

ROBERT W. BRUNDAGE

BY

Alfred C. Barty
ATTORNEY

1

3,034,447

HYDRAULIC PUMP OR MOTOR

Robert W. Brundage, Willoughby Lake, Ohio

Filed May 19, 1959, Ser. No. 814,319

9 Claims. (Cl. 103-126)

This invention pertains to the art of hydraulic pumps or motors, and more particularly to a hydraulic pump or motor of the positive displacement revolving chamber type.

The invention is particularly applicable to what is generally known as internal gear type pumps or motors, and will be described with particular reference thereto, although it will be appreciated that the invention has broader applications and in many instances is equally applicable to vane or rotating cylinder type hydraulic pumps or motors.

Furthermore, the present invention is particularly applicable to hydraulic pumps or motors operable at what may be termed very high hydraulic pressures, that is to say, above 1,000 pounds per square inch and oftentimes approaching or exceeding 4,000 pounds per square inch. At such pressures, construction and expedients usable at the lower pressures are often unsatisfactory and inapplicable to the problems of higher pressures.

For the purpose of simplicity, the invention will be described only in relation to a pump and reference will be made to inlet and outlet ports, inlet and outlet manifolds and increasing and decreasing volume chambers, all of which will be at low and high pressures respectively. The description may be applied to a hydraulic motor by reversing the relationship of the high and low pressures.

Internal gear type hydraulic pumps are normally comprised of an internally-toothed and an externally-toothed gear member rotating on spaced axes with the teeth in sliding, sealing engagement, all within an internal cavity of a housing. A shaft extends from the outside of the housing through this cavity and is supported for rotation within the housing on spaced bearings with the externally-toothed gear mounted on the shaft between these bearings. The internally-toothed gear in turn is mounted for rotation on an axis spaced from that of the shaft by means of an eccentric ring supported within the housing. Sealing members engage each axial face of the gear members so that when the gear members rotate, the members define a plurality of chambers which revolve about the axes and, depending upon their position with relation to the plane containing the spaced axes, progressively increase from a point of minimum volume corresponding to closed gear mesh to a point of maximum volume corresponding to open gear mesh and then decrease in volume to the point of minimum volume. Normally chambers which are decreasing in volume all communicate with a discharge port and are at high hydraulic pressures, while the chambers which are increasing in volume all communicate with an inlet port and are at relatively low pressures.

It will be appreciated that the high hydraulic pressures are non-symmetrically located relative to the axes of rotation and create non-symmetrical forces, both axial and radial, of a very substantial magnitude on the various parts and members of the pump. It is the deleterious effects of these non-symmetrical forces with which the present invention primarily deals.

Thus the high pressure fluid in the decreasing volume chambers exerts a very high force on one outwardly facing surface of the externally toothed gear. This force is on the radial line through the center of all the high pressure chambers and this radial line with the axis of rotation forms what will hereinafter be referred to as the "force plane." This force is transmitted first

2

to the shaft and then to the bearings, all in the force plane.

Because of the magnitude of this radial force, sleeve bearings heretofore could not be employed to rotatably support the shaft, and it has been considered necessary to use the much more expensive and bulky roller or ball bearings.

In accordance with the present invention, however, a sleeve-type bearing is employed and the problems heretofore existent therewith are prevented by communicating controlled amounts of the high pressure hydraulic fluid to the inside of the sleeve bearing in a point or line located diametrically opposite from the high pressure chambers and on the force plane. This hydraulic fluid exerts a hydraulic force on the shaft in opposition to the radial force on the externally-toothed gear and effectively counterbalances it. The loading on the sleeve bearing is thus either eliminated or reduced to a degree where a sleeve bearing is capable of carrying the load.

As a result of the present invention, it was further found that the forces on the externally-toothed gear were transmitted to the shaft at a point generally midway between the bearing supports for the shaft and because of the magnitude of the force, the shaft was deflected or flexed slightly in an amount proportional to the hydraulic pressure. Thus if the sleeve bearings were aligned with the shaft at zero pressure, they were out of line with the shaft when the pump was operating under pressure and excessive bearing wear occurred.

In accordance with the present invention, this problem is solved by mounting the sleeve bearings in the housing for a pivoted movement in the radial force plane. The sleeve bearing is thus automatically able to change its position in the housing and remain in alignment with the shaft even though the shaft is slightly flexed under the effects of the hydraulic forces.

The high pressure hydraulic fluid in the high pressure chambers also exerts a radially-offset axial force on each of the sealing members, also in the force plane. This force tends to turn the sealing members in a direction such that the sealing pressures between the sealing members and the gear faces adjacent the high pressure chambers, where the sealing pressures are really needed, was substantially less than the sealing pressures adjacent the low pressure chambers where the sealing pressures are not needed.

In accordance with the present invention, this problem is solved by pivotally mounting at least one of the sealing members in the housing about a point in the force plane and providing a radial force in the force plane on the sealing member so located relative to such point as to pivot the sealing member about the point in a direction to increase the pressure on the gear face adjacent the high pressure chambers such that a more uniform sealing pressure of the sealing member across the entire gear face results.

This radial force tending to so pivot the sealing member in the indicated plane may be derived from the unbalanced radial force of the hydraulic fluid in the high pressure chambers transmitted to the shaft and thence to the sealing member to the housing, or may be created by exposing non-symmetrical areas of the sealing member to the high pressure fluids, or both.

A still further problem with internal gear type hydraulic pumps has been to maintain the gear teeth at open mesh in sealing relationship as the various parts of the pump wear in use to thus prevent leakage between the teeth at this point.

In accordance with the invention the eccentric bearing ring (in which the internally-toothed gear rotates) is radially movable within the housing cavity and means are provided for urging or biasing the teeth of the internally-

toothed gear at open mesh toward the teeth of the externally-toothed gear. The result is that as the parts of the pump wear, the clearances normally created thereby will be automatically taken up within the range of movement of eccentric ring in the housing.

A still further problem with hydraulic pumps has been the bulk and complexity of the housing necessitated by the extremely high forces of the hydraulic pressures. The housing provides the physical support for the various members of the pump to hold them in proper sealing relationship. It has been conventional in all high pressure hydraulic pumps to provide the various parts of the housing with flat abutting surfaces and to hold these parts in assembled relationship by means of a plurality of bolts, studs, dowel pins or the like. A threaded engagement of the various parts of the housing has been completely unsatisfactory because of the clearance apparently necessary in the threads. Thus the high forces involved caused a slight cocking or shifting of one of the parts of the housing relative to the other resulting in a mis-alignment and a separation of the pump sealing surfaces.

In accordance with the present invention, the housing is formed in two parts which are threadably engaged one with the other and the members of the pump, e.g. the bearing and sealing member relying on at least one of the housing parts for physical support are pivotally supported thereby such that when the said one housing part shifts relative to the other, such shifting is not transmitted to the said members of the pump.

The principal object of the present invention is the provision of a new and improved hydraulic pump or motor which is relatively simple in construction, which can be manufactured for a minimum cost, and which has a high volumetric and mechanical efficiency.

Another object of the invention is the provision of a new and improved high pressure hydraulic pump or motor which may successfully employ relatively inexpensive sleeve type bearings.

Another object of the invention is the provision of a pump of the general type described wherein the sealing members will have a uniform sealing pressure against the end faces of the gears.

Still another object is the provision of a pump or motor of the type described wherein the parts of the housing may be screw-threaded into assembled relationship.

Another object of the invention is the provision of a new and improved hydraulic pump or motor of the type described, wherein one or both of the parts of the housing may be formed by relatively inexpensive impact extrusion processes.

Yet another object of the invention is the provision of a new and improved internal gear type hydraulic pump or motor wherein the wear of the parts normally permitting leakage at open mesh is automatically taken up.

The invention may take physical form in certain parts and arrangement of parts, the preferred embodiments of which will be described in detail in this specification and illustrated in the accompanying drawings which are a part hereof and wherein;

FIGURE 1 is a side cross sectional view of a hydraulic pump or motor illustrating a preferred embodiment of the invention.

FIGURES 2 and 3 are cross sectional views of FIGURE 1 taken approximately on the lines 2—2 and 3—3 thereof, respectively.

FIGURE 4 is a cross section view of a bearing member; and

FIGURE 5 is a view similar to FIGURE 1 but illustrating an alternative embodiment of the invention.

Referring now to the drawings wherein the showings are for the purposes of illustrating a preferred embodiment of the invention only, and not for the purposes of limiting same, the figures show what may alternatively be used as either a hydraulic pump or a hydraulic motor,

but which for the purpose of simplicity, will be described simply as a hydraulic pump.

The pump of FIGURE 1 includes a housing H, having an interior pumping cavity in which are mounted a plurality of pumping members defining a plurality of closed chambers which progressively increase and decrease in volume as the members move relative to each other. While such members may take a number of conventional forms such as rotating cylinders with axially reciprocating pistons, rotating vanes, in the embodiment shown they comprise generally an externally toothed gear member 11, an internally toothed gear member 12, a sealing member 13 and a manifold member 14 engaging the right-hand and left-hand axial faces of the gears 11 and 12 respectively.

The gear member 11 is slidable on and keyed to a rotatable shaft 16 by means of a key 18 and a key way 19 on the member 11. The internally toothed gear member 12 is supported for rotation about an axis, spaced from the axis of the shaft 16 in a bearing member 17 which, as will appear, is loosely mounted within the housing cavity. The gear member 12 has one or more teeth than that of the gear member 11 and these teeth are in sliding, sealing engagement so that as the gear members 11 and 12 rotate, they along with the sealing and manifold members 13 and 14 define a plurality of closed chambers revolving on a closed path of movement and which progressively increase in volume from a point A of a minimum volume to a point B of maximum volume and then decrease to the point of minimum volume A. The points A and B are on what may be termed the neutral plane through the two axes of rotation and it will be further noted that the gear teeth at the point A are in what may be termed "closed mesh" and at the point B at "open mesh."

Housing

The housing H in the embodiment shown is formed in two parts, namely, a main part 20, generally in the shape of a cup, and a closure part 24 over the open end of the cup 20, both parts being so arranged as to have radially opposed surfaces relative to each other. Opposed axially facing surfaces between the axial ends of the radially opposed surfaces such as threads 25 retain the part 24 in assembled relationship with the cup 20. An O-ring 26 mounted in a groove in the outer surface of the closure part 24. This construction may be distinguished from conventional pump housings wherein the parts have axially facing surfaces in abutment and bolts extending through these surfaces holding them in assembled relationship. The closure part 24 on the cavity side of the threads 25 provides a seal to prevent leakage of hydraulic fluids longitudinally past the threads 25. As will appear, the threaded fit of the closure part 24 and the main part 20 is not particularly critical in the construction of the pump.

The main part 20 has a pair of external diametrically opposed longitudinally extending ribs 21, 22 and on the inside has a plurality of inwardly facing cylindrical surfaces 27, 28, 29 (reading from left to right), each of which is larger than the predecessor for receiving and coacting with the various parts or members of the pump itself as will appear.

In a like manner, the closure part 24 has a plurality of cylindrical surfaces 32, 33, 34 (reading from right to left), each of which is larger than the predecessor.

In the preferred embodiment, both of the parts 20, 24 are preferably made from aluminum and by virtue of the symmetry thereof, can be formed from impact extruded aluminum. Furthermore, because of the design of the pump as will appear, the diameters and finishes of the cylindrical surfaces 27, 28, 29, 30, 32, 33, 34 can have rather rough tolerances, and a rough finish, as may be characteristic of impact extrusion tools as well

as the slight taper necessary to effect withdrawal of the extrusion die.

It is to be further noted that the threads 25 must have a slight clearance for ready assembly. Such threads under the high pressure forces which will be developed on the inside of the housing H permit the closure part 24 to cock slightly with reference to the main part 20. However, because of the mounting of the pump members relative to the closure part 24, this cocking is not a problem.

Shaft and Bearings

In the embodiment of the invention shown, the shaft 16 extends through an opening 36 in the left hand end or base of the main part 20 and its left end in the housing H is rotatably supported in a bearing member 37 while its right hand end in the housing H is rotatably supported in a bearing member 38 with the gears in between.

The bearing member 37 is loosely mounted in the portion of the cavity defined by the cylindrical surface 27 in a manner so as to pivot relative to the housing H. Thus, in the embodiment of the invention shown, the bearing member 37 has a cylindrically extending bead 39 around its outer surface having a rounded or circular contour in cross-section, the outer diameter of which bead is just slightly less than the diameter of the surface 27 so that the bead engages the surface 27 at the point of tangency only.

In a like manner, the bearing member 38 is mounted for pivoted movement in the sealing member 13 and for this purpose has a circumferentially extending bead 40 with a circular contour in cross-section on its outer surface which is loosely engaged in a cylindrical opening defined by a surface 41 in the right hand end of the sealing member 13.

Sealing and Manifold Member

The sealing member 13 in turn is mounted for a pivoted movement in the closure part 24 and for this purpose has a circumferentially extending bead or flange 42 with a circular contour on its right hand end which has a diameter slightly less than the diameter of the surface 32 and is loosely fitted into the cavity formed by such surface. An O-ring 44 just to the left of the bead 42 fits around the right end of the sealing member 13 and is in sealing engagement with the cylindrical surface 33. This O-ring 44 seals the cavity 46 which is at high fluid pressure from the cavity 47 which, as will appear, is at low hydraulic pressure.

A coil spring 48 bears between the right hand end of the sealing member 13 and the shoulder 49 of the closure part 24, between the cylindrical surfaces 33, 34 and biases the sealing member 13 into sealing engagement with the right hand end of gear 11, 12.

The manifold member 14 is generally in the shape of a disc and is positioned, with an interference fit in the cavity defined by the cylindrical surface 29 and rests against the shoulder 52 formed between the cylindrical surfaces 28, 29. The manifold member 14 has a sealing surface 53 in sealing engagement with the left hand side of the gears 11 and 12, and this surface has an arcuate extending port 54 formed therein generally in the path of movement of the pumping chambers. A passage 55 communicates the port 54 with the outer periphery of the manifold member 14 and thus with the cavity 57 formed between the outer periphery of the manifold member 14 and the cylindrical surface 30. The housing 20 has an opening 58 extending from the outer surface of the rib 22 inwardly to communicate with this cavity 57 and forms the outlet for the pump shown. It will be noted that the passage 55 is diametrically opposite from the opening 58 so that the outlet fluid from the pump flows circumferentially through the cavity 57 and thence out through the outlet 58. This path of fluid flow contributes to the cooling of the gear members 11, 12 and manifold member 14.

The manifold member 14 also has an arcuate extending port 60 diametrically opposite from the port 54 and identically shaped thereto, which port 60 is in the path of movement of the pumping chambers and extends axially through the manifold member 14 to communicate the pumping chambers with the cavity 61 defined by the cylindrical surface 28. An opening 62 through the rib 21 communicates with this cavity 61 and forms the inlet for the pump. It will be noted that the inlet opening 62 is diametrically opposite from the port 60 and inwardly flowing hydraulic fluid thus flows over the bearing 37 and gives a cooling effect thereto.

The sealing surface 53 of the manifold member 14, as shown and with the exception of the passage 55, completely surrounds the ports 54, 60 to form between the arcuate ends thereof, lands 64, 65, each of which have a line of movement width slightly greater (by about 10%) than the line of movement width of the opening 55 from the pumping chambers to the ports 54.

If, however, a ported plate is employed as is described in my co-pending application, Serial No. 656,117, filed April 30, 1957, now Patent No. 3,007,418, issued November 7, 1961, then the line of movement width of the opening from the pumping chambers will be substantially reduced and the line of movement width of the lands 64, 65 may be substantially reduced.

Furthermore, in the embodiment of the invention shown, the ports 54, 60 each have the same line of movement width and the lands 64, 65 are symmetrically disposed about the neutral plane through the axis of rotation. The manifold member is held from rotating by an interference fit with the housing surface 29.

Eccentric Ring Member

The eccentric ring member 17 rotatably supports the gear member 12 and the center or axis of its inner surface coincides with the axis of rotation of the gear 12. The outer cylindrical surface 67 of the ring member 12 is eccentric to the inner surface and thus to the axis of rotation of the gear 12 and as an important part of the present invention has an outer diameter less than the diameter of the surface 30. The ring member is thus relatively free to move radially in the housing 20, but is held against rotation by a pin 68 extending from the manifold member 14 into a slot 69 of the member 17. The eccentricity of the ring member 20 is thus located relative to the lands 64, 65 so that the diametrical line through the middle of the lands corresponds to the neutral axis A—B (hereinafter called mid line).

It is to be noted that the hydraulic pressures exert forces over a wide circumferential width of the chambers but for the purpose of this invention these forces may be all summed up by a single large radial force which swings or oscillates about the perpendicular to the land mid line as chambers come newly into communication with the discharge manifold or go out of communication therewith. This force is indicated generally in FIGURE 3 by the vector 70 and for the purposes of describing the invention, its line of action is assumed as being on the radial line perpendicular to the mid line through the lands 64, 65 on the high pressure chamber side of such line. This line of action with the axis of rotation defines a "force plane."

In the embodiment of the invention shown, the lands 64, 65 are symmetrical with the neutral axis AB and thus the vector 70 is perpendicular to this neutral axis AB. This force urges or biases the eccentric ring radially toward the housing.

In accordance with the invention, the ring member 17 engages the housing 20 at a single point located on this line of action. The point or line may be located in any one of a number of different ways, but in the embodiment shown, the eccentric ring has a flat 72 formed on its outer surface with the exception of a single point 73 close to the middle and this point bears against an insert

74 mounted on the surface 30 and having a surface 75 against which the point 73 bears. A screw 78 extends through the rib 21 and engages the insert 74 to locate it and hold it in position. The insert 74 is preferably of a hardened material such as steel.

The construction of the insert 74 is one of convenience in manufacturing and can take a number of different forms without varying from the invention. The point 73 may also be an axially extending rib.

Further in accordance with the invention, means are provided for continuously biasing or urging the teeth at open mesh together. In the embodiment shown such means include a leaf spring 71 mounted in a circumferentially extending groove or slot 79 in the outer surface 67 of the member 17 symmetrical about the neutral axis A—B. The ends of the spring bear against the housing surface 30 and the center against the member 17.

All gears have a slight variation in the height of the gear teeth. In the construction shown, the gear teeth at open mesh take the position of the highest teeth, friction between the gears and the other pump members holding the gears in this position for one revolution when the highest teeth again touch and locate the gears for the next revolution.

The force vector 70 as shown in FIGURE 3 and FIGURE 1 is in an upward direction. The same hydraulic pressure exerts an equal and opposite force indicated by the vector 76 on the axial mid-plane of the gear 11, which force in FIGURE 1 is in a downward direction. This force is transferred from the gear 11 to the shaft 16 and thence to the two bearings 37, 38 in inverse proportion to the axial centerline distance of the bearings 37, 38 from the axial mid-plane of the gear 11. The force transmitted to the bearing member 37 is then transmitted to the cylindrical surface 27 through the spherical surface on the flange 39 all on the "force plane."

In a like manner, the bearing member 38 engages the surface 41 of the sealing member 13 at a point 81 also located on the force plane.

With this arrangement, it will be appreciated that the bearing members 37, 38 each pivot about the support points 77, 81 respectively so that as the shaft 16 is deflected as a result of the force vector 76, the bearing members 37, 38 may deflect therewith resulting in a uniform pressure loading of the shaft 16 along the bearing surfaces of the bearing members 37, 38.

It will be appreciated that in a high pressure pump, the force vector 76 is very large and the pressures of the shaft 16 on the bearing surfaces of the bearing members 37, 38 is very substantial and in many instances greater than a conventional sleeve type bearing is capable of handling.

In accordance with the invention, the high pressure fluid of the pump is communicated to the space between the shaft 16 and the bearing surfaces of the bearings 37, 38 in a controlled manner so that the hydraulic fluid will exert an upward hydraulic force on the shaft 16 in an amount such that the loading of the shaft 16 on the bearings 37, 38 will be within that which the bearings are capable of withstanding. Preferably the amount is equal to the radial force of the shaft 16 on the respective bearing. In the embodiment of the invention shown, the bearing surface 83 of the bearing 38 has a diamond-shaped groove 84 formed therein, symmetrical about the "force plane" and so dimensioned that the circumferential corners 85 are spaced an arcuate distance less than 180° and so that the axial corners 86 are spaced from the axial ends of the surface 83. A circumferential groove 87 interconnects the corners 85.

Hydraulic fluid at the high pressure of the pump is supplied to these grooves. In the embodiment of the invention shown, a radial passage 88 located on the "force plane" and midway between the axial ends of the bearing member 38 is provided, which passage is aligned with

a corresponding passage 89 through the sealing member 13. It will be noted that the passage 88 is symmetrical relative to the bead 40 and has a diameter less than the width of the bead 40, so that, in effect, a portion of the bead 40 which engages the surface 41 is a circle surrounding the passage 88.

The cavity 46 communicates with the outlet 58 through the clearance between the surfaces 67 and 30. Thus, hydraulic fluid at high pressure flows through the passage 89 and the passage 88 to the groove 87. In some instances, it has been found that too much hydraulic fluid flowed to the groove 87 and to limit this, a metering pin 90 is disposed in the openings 88, 89 for the purpose of limiting the hydraulic flow and also for the purpose of continuously maintaining the alignment of the openings 88, 89.

By virtue of the arrangement shown, the area enclosed by the diamond shaped groove 84 is generally at the high hydraulic pressure and this pressure exerts an upward force in the shaft 16. By varying the size of the diamond, this force may be varied.

The fluid from the diamond groove 84 flows axially along the shaft 16 and is discharged into the cavity 47. This cavity is at inlet pressure by virtue of the clearances between the bead 40 and the surface 80, the key-way 19 and the space or clearance 92 between the manifold member 14 and right hand end of the bearing member 37.

The bearing surface of the bearing member 37 is constructed in a manner similar to the bearing surface of the bearing 38 and will not be described further herein. Suffice it to say that a passage 105 from the cavity 57 communicates the high pressure to the grooves.

It will be appreciated that the hydraulic pressures in the pumping chambers also exert an axial force indicated generally by the vector 94 on the sealing member 13. Such axial force is in the force plane and tends to move the sealing member 13 to the right. This force 94 is opposed primarily by the force of the hydraulic fluid in the cavity 46 against the right hand axially facing surface 93 of the sealing member 13, which force is axially symmetrical relative to the member 13 and is indicated by the force vector 95. The force vectors 94, 95 are equal and opposite, but are radially off-set one from the other by a distance r with the result that the sealing member has a turning moment or force couple which tends to turn the sealing member 13 away from the gears 11, 12 at the high pressure chambers.

To prevent this, the invention contemplates providing a radial force on the sealing member so located relative to the point of support of the sealing member in the housing that an opposite turning moment on the member is created.

In the preferred embodiment, the radial force is provided from the force 76. Thus, the portion of the force 76 which is transmitted to the bearing member 38, is in turn transferred through the bead 40 to the sealing member 13 as indicated by the vector 100. The force 100 is opposed by an equal and opposite force 101 at the point of engagement of the bead 42 with the surface 32. It is to be noted that this point of engagement is on the axial side of the force transferral from the bearing 38 to the sealing member 13 remote from the gear 11, 12. By properly proportioning the axial length 1 between the forces 100 and 101, at the time of design of the pump, the turning moment on the sealing member 13 may be proportioned to be equal and opposite to the turning moment of the force vector 94.

It will be appreciated that the force 94 also tends to separate the manifold member 14 from the gears 11, 12 but as the gears are axially movable on the shaft, the forces on the sealing member 13 are transmitted through the gears 11, 12 so that with the construction above described, both member 13 and 14 have a pressure sealing engagement with the axial ends of the gears which is uniform over the entire axially facing surface thereof.

FIGURE 5 shows an alternative arrangement for holding the sealing member and manifold member in uniform pressure sealing engagement with the axial ends of the gears. Here like parts will be designated with like numbers and similar parts will be designated with a like number with a prime (') mark at it.

In the embodiment of FIGURE 5 the principal difference is that both the sealing member 13' and the manifold member 14' are loosely mounted in the housing for pivoted movement about a point in the force plane and the force for creating the turning moment on these members in opposition to the force 94 is created hydraulically by exposing unsymmetrical areas of the members to the high pressure fluids with the centers of the areas so located relative to the pivot point that the required turning moment or force is created. While this embodiment of the invention shows both the sealing member and the manifold member as being so pivoted, it will be appreciated that the same end result can be obtained by fixedly mounting one of the members and increasing the unsymmetry of the other.

In FIGURE 5 the high pressure chambers as distinguished from FIGURE 1 are in the lower half of the figure and the housing part 26' has inner cylindrical surfaces 27', 29', 30' of progressively increasing diameters reading from left to right. The closure part 25 has cylindrical surfaces 32', 34' of progressively increasing diameters reading from right to left.

The sealing member 13' has cylindrical surfaces 109, 110 of progressively increasing diameters from right to left separated by a shoulder 111 facing axially away from the gears 11, 12. The surfaces 109, 110 are of a diameter less than and loosely fit within the cavities defined by the surfaces 32', 34'. An O-ring 113 in a groove 114 provides a seal between surfaces 109, 32'.

The cavity thus formed to the right of O-ring 113 communicates with the low pressure chambers through the space between the bearing member 38 and the surface 80 and thence to a radial passage 120 in the sealing surface of the member 13' extending outwardly to the line of movement of the low pressure chambers.

The surface 109 has a cylindrical bead 121 just to the left of O-ring 113 although it could be to the right of O-ring 113 of a diameter slightly less than surface 32' which engages surface 32' at one point of tangency only on the force plane. The sealing member 13' is otherwise free to move radially in the housing and pivot about this point.

A second O-ring 115 between surface 109 and surface 34' and bearing against the shoulder 111 defines a cavity 116 which is communicated with the high pressure chambers through a passage 117 opening through the sealing member sealing surface in the line of movement of such chambers.

The left hand side of the O-ring 115 also communicates with the low pressure chambers through the space between the eccentric ring 17 and the surface 30' and inlet opening 62'.

In accordance with the invention, the plane of shoulder 111 and thus O-ring 115 is oblique to the axis of shaft 16 and inclined symmetrically toward the high pressure chambers and away from the low pressure chambers so that the area of the portion of the surface 109 between the O-rings and on the high pressure side of the axis is greater than the area on the low pressure side, the differential area being a function of the angle of inclination and having a center n on the high pressure side midway between the axially transverse planes through the axial limits of the O-ring 115 and on the force plane.

The high pressures in the cavity defined by the two O-rings exert both an axial force 95' on the shoulder 111 equal and opposite to the force 94 and a radial force 125 on the center end proportional to the differential area. As the center n and the line of action of the force 125 is spaced a distance P from the line of bead 121, a turning moment on the sealing member 13' is created

tending to rotate the member toward the high pressure chambers and thus oppose the turning moment of the force 94.

It will be appreciated that by inclining the plane of the O-ring 115 in an opposite direction and by placing the point of tangency contact of the sealing member 13' with the housing between the O-ring and the gears 11, 12, a similar turning moment can be created.

In a similar manner the manifold member 14' has cylindrical surfaces 130, 131 of increasing diameters reading from left to right separated by a shoulder 132 facing axially away from the gears 11, 12. The surfaces 130, 131 are of a diameter less than and loosely fit within the cavities defined by the surfaces 27', 29'. An O-ring 133 in a groove 134 provides a seal between surfaces 130, 27'.

The cavity to the left of O-ring 133 communicates with the low pressure chambers through the space between the bearing member 37' and the inner surface 135 of an axially extending bore in the manifold member 14' and thence through a radial passage 136 communicating with the inlet opening 62'.

The surface 130 has a cylindrical bead 138 just to the left of O-ring 135 of a diameter slightly less than the surface 27' which engages surface 27' at one point of tangency only on the force plane. The manifold member 14' is otherwise free to move radially in the housing and pivot about this point.

A second O-ring 140 between surface 131 and surface 27' and bearing against the shoulder 132 defines with the O-ring 135 a cavity into which the outlet manifold 54 discharges and which is communicated with the outlet opening 58'.

The right hand side of the O-ring 140 is in communication with the inlet opening 62'.

In accordance with the invention, the plane of shoulder 132 and thus O-ring 140 is oblique to the axis of shaft 16 and inclined symmetrically toward the high pressure chambers and away from the low pressure chambers so that the area of the portion of the surface 130 between the O-rings 135, 140 on the high pressure side of the axis is greater than the area on the low pressure side of the axis. The differential area is the function of the angle of inclination and has a center s on the high pressure side midway between the axially transverse planes through the axial limits of the O-ring 140.

The high pressures exert both an axial force 150 on the shoulder 132 equal and opposite to the force 74 and a radial force 151 on the center s proportional to the differential area and as this center is spaced a distance t from the plane of bead 138 a turning moment on the manifold member 14' is created tending to rotate the member toward the high pressure chambers and thus oppose the turning moment of the force 74.

It will be appreciated that the turning moment on either of the members 13' or 14' may be varied or adjusted by two expedients, namely, changing the inclination of the planes of the O-rings or changing the distance from the center of the differential pressure areas from the point of contact with the housing or both.

It will further be appreciated that either the manifold member or sealing member or both may be mounted as above described.

In connection with either embodiment of the invention, it is to be noted that the closure part 24 with reference to the main part 20, because of the necessary clearance between the threaded engagement, may cock slightly under the force of the high pressures on the inside of the pump. In the present invention this is not detrimental for the reason that the sealing member 13 is pivoted relative to the closure part 24 and when the part 24 cocks, this twisting or turning is not transmitted to the sealing disc 13.

It is to be further noted that with reference to the embodiment of FIGURE 1, the internally toothed gear 12 rotates within the eccentric bearing ring 17 and is

lubricated by the fluid being pumped at the high discharge pressure. Inasmuch as the viscosity of the fluid increases with pressure and inasmuch as a high viscosity lubricant is required to withstand the heavy radial loads between the gear and the ring, this is a decided advantage. This is described and claimed in my co-pending application, Serial No. 814,320, filed May 19, 1959.

It will thus be seen that embodiments of the invention have been described in such detail as will enable those skilled in the art to utilize the principles of the invention in the design of high pressure, high efficiency, positive displacement hydraulic pumps and/or motors.

The invention has been described in connection with preferred embodiments. Obviously, modifications and alterations will occur to others upon reading and understanding of this specification and it is my intention to include all such modifications and alterations insofar as they come within the scope of the appendant claims.

Having thus described my invention, I claim:

1. In a positive displacement hydraulic device comprised in combination: a housing having an inwardly facing surface defining a pumping cavity at least portions of which surface are generally cylindrical; a shaft extending into said housing and rotatable on the axis of said cylindrical portion; an externally toothed gear supported on said shaft for rotation therewith; an internally toothed gear having teeth in sliding, sealing engagement with said externally toothed gear and rotatable about an axis spaced from said shaft axis by a predetermined gear eccentricity determined by said gear teeth; a bearing ring having a radially inwardly facing cylindrical surface rotatably supporting said internally toothed gear and a radially outwardly facing outer surface; said gear teeth moving from open to closed mesh as the gears rotate and defining a plurality of revolving increasing and decreasing volume chambers; at least one of said chambers being at high discharge pressure whereby a resultant radially outward force is exerted on said internally toothed gear, the improvement which comprises: said bearing ring outer surface having a clearance from said housing surface less than said gear eccentricity whereby said ring is radially movable in said housing cavity; and means biasing said bearing ring radially inwardly at the open mesh point of said chambers, whereby said gear teeth are biased together at the open mesh point thereof.

2. The improvement of claim 1 wherein said means comprise a spring member disposed between the bearing ring and said housing surface adjacent opening mesh of said gears.

3. The improvement of claim 1 wherein said bearing ring outer surface and said housing surface have a point engagement generally on the radial line through the center of the high pressure chambers.

4. The improvement of claim 1 wherein said bearing ring outer surface has a flat on the high pressure chamber side of the axis symmetrically disposed relative to the high pressure chambers; said housing cylindrical surface has a similarly disposed flat opposite said bearing ring flat and means provide a point engagement between said flats.

5. The improvement of claim 1 wherein said clearance is from .002 to .010 inch.

6. In a positive displacement hydraulic pump comprised of, in combination: a housing having an inwardly facing surface defining a pumping cavity at least portions of which surface are cylindrical; a shaft extending into said housing and rotatable on the axis of said cylindrical portion; an externally toothed gear supported on said shaft for rotation therewith; an internally toothed gear having teeth in sliding, sealing engagement with said externally toothed gear and rotatable about an axis spaced from said shaft axis by a predetermined gear eccentricity determined by said gear teeth; means rotatably

supporting said internally toothed gear; said gear teeth moving from open to closed mesh as the gears rotate and defining a plurality of revolving increasing and decreasing volume chambers; a sealing member in sealing engagement with one axial end of said gears; a manifold member in sealing engagement with the other axial end of said gears, the decreasing volume chambers being at high discharge pressure whereby a resultant radially inward force is exerted on said externally toothed gear and on said shaft on the same side of the shaft as the high pressure chambers and a radially offset axial force is exerted on said members on the same side of said shaft as said high pressure chambers; the improvement which comprises at least one of said members being loosely mounted in said housing and engaging said housing at a single radially facing contact point spaced from the axial end of said gears with which said one member is in sealing engagement a predetermined distance and means transferring at least a portion of said radially inward force on said shaft to said one member at a point between said single point of contact and the axial end of said gears with which said one member is in engagement whereby to create a turning moment on said one member in opposition to the turn-moment of the offset axial force thereon.

7. The improvement of claim 6 wherein said one member has a radially outwardly facing surface exposed to the high hydraulic pressure, said surface being circumferentially unsymmetrical such that the area of the portion of the surface on the high pressure side of the shaft is greater than the area on the low pressure side of the shaft, the size of the differential area thereof and the center of such area being located between said point of contact and said axial end of said gears whereby to create a radial force on said one member, said radial forces on said one member exerting a turning moment about said contact point to oppose the turning moment of said radially offset axial force.

8. In a positive displacement hydraulic device comprising in combination: a housing having an inner generally cylindrical surface defining a cavity, a shaft extending into said cavity and rotatable about an axis, a plurality of members rotatable with said shaft and defining a plurality of chambers which move on a fixed line of movement, sealing means in sealing engagement with each axial end of said members, one of said means having arcuate ports therein communicating with the openings from said chambers as they revolve and diametrically opposed lands spacing the ends of said ports, the line of movement width of said lands being slightly greater than the line of movement width of the openings from said chambers to said ports, the chambers on one radial side of the diametrical line through the lands being at high fluid pressure whereby a resultant radially inward force is exerted on said shaft on the same side of the shaft as the high pressure chambers; bearing members, one on each side of said pumping members rotatably supporting said shaft in said housing; the improvement which comprises said bearing members being loosely mounted in said housing for pivoted movement relative to said housing about a point on the side of said shaft diametrically opposite from said high pressure chambers.

9. In a positive displacement hydraulic device comprising in combination: a housing having an inner generally cylindrical surface defining a cavity, a shaft extending into said cavity and rotatable about an axis, a plurality of members rotatable with said shaft and defining a plurality of chambers which move on a closed fixed line of movement, sealing means in sealing engagement with each axial end of said members, one of said means having arcuate ports therein communicating with said chambers as they revolve and a pair of diametrically opposed lands spacing the arcuate ends of said ports, the line of movement width of said lands being slightly

greater than the line of movement width of the openings from said chambers to said ports, the chambers on one radial side of the diametrical line through the lands being at high fluid pressure whereby a resultant radially inward force is exerted on said shaft on the same side of the shaft as the high pressure chambers, and bearing members, one at each axial end of said pumping members rotatably supporting said shaft in said housing; the improvement which comprises said bearing members being loosely mounted in said housing and at least on the side of said shaft diametrically opposite from said high pressure chambers having a circumferentially extending surface having a circular contour in cross section and means supporting said bearings comprised of a circular surface having a diameter at least slightly greater than the diameter of said circumferentially extending portion, said bearing members being free to pivot about the point of contact of said surfaces.

5

10

15

1,442,828
1,940,410
1,970,146
2,070,413
2,792,788
2,809,595
2,956,512

782,701
1,006,722
1,055,365
1,121,395

References Cited in the file of this patent

UNITED STATES PATENTS

Rotermund Jan. 23, 1923
Fitch et al. Dec. 19, 1933
Hill Aug. 14, 1934
Sandberg Feb. 9, 1937
Eames May 21, 1957
Adams et al. Oct. 15, 1957
Brundage Oct. 18, 1960

FOREIGN PATENTS

Great Britain Sept. 11, 1957
Germany Apr. 18, 1957
(KL 59e 3/01)
Germany Apr. 16, 1959
(KL 59e 3/01)
France Apr. 30, 1956