

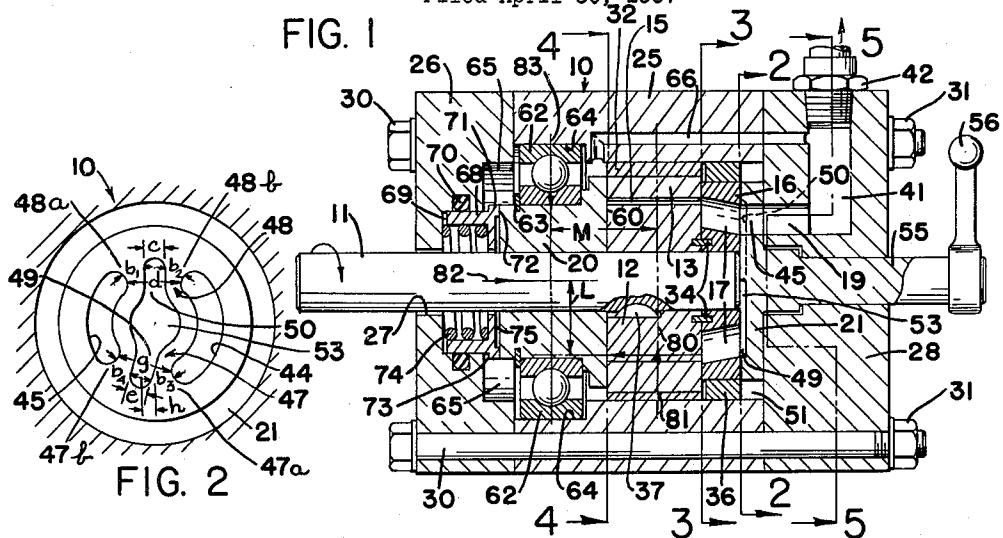
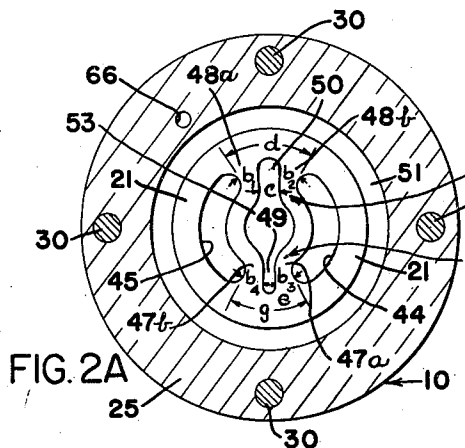
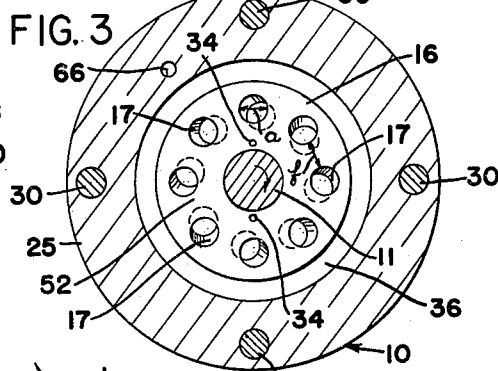
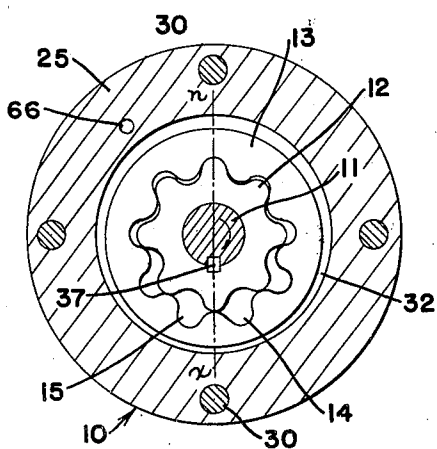
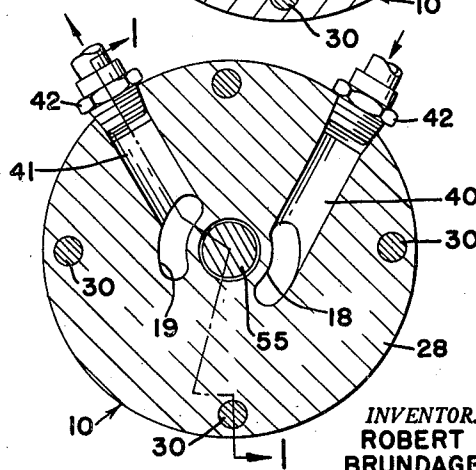
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VARIABLE DELIVERY HYDRAULIC PUMP OR MOTOR

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1

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VARIABLE DELIVERY HYDRAULIC PUMP OR MOTOR

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This invention pertains to the art of hydraulic pumps or motors, and more particularly to a hydraulic pump or motor of the variable volume or speed type.

The invention is particularly applicable to a hydraulic pump of the internal gear type and will be described with particular reference thereto although it will be appreciated that the invention is equally applicable to hydraulic motors of the internal gear type or to other pumps or motors having positive-displacement, rotating chambers, such as those of the vane or barrel-cylinder type.

The terms as employed hereinafter will relate to a pump. Obviously if the invention or claims are to be considered in relation to a hydraulic motor, outlet or discharge pressure will become inlet pressure and inlet pressure will become discharge or outlet pressure.

In pumps of the type to which the invention pertains, a plurality of members rotate to define a plurality of rotating increasing and decreasing volume chambers which communicate through passages of a known width with inlet and outlet manifolds separated by stops or lands, past which the passages must move as they alternately communicate with the two manifolds.

In such pumps, the rotational width of the lands have heretofore been made slightly greater than the rotational width of the passages so that when a passage faces a land, high pressure fluid cannot flow from the discharge manifold to the inlet manifold around the land and through the passage.

For full volume output, the lands are normally located at the points of minimum and maximum volume of the chambers, i.e. on the neutral axis of the pump. It is known that the output volume can be varied by rotating the lands away from these points on the passage line of movement. However, when the lands of this width are rotated away from these points, they then face passages from chambers whose volume is changing rapidly. Because of the relative width of the lands and passages, there is an instant in the movement of the passages past the lands wherein fluid can neither flow into nor out of the chambers. Where fluid cannot flow out of the chambers, excessively high pressures result. This is called "trapping" and is highly objectional. Where fluid cannot flow into a chamber, a vacuum or void results. This is called "cavitation" and is generally not objectionable unless the chamber as it leaves the land comes into communication with the high pressure manifold. In such instance, the high pressure fluid rapidly flows into the void created and causes hammering and a great loss in efficiency.

In my co-pending application Serial No. 548,022 filed November 21, 1955, now Patent No. 2,956,506, this problem is dealt with by making each land substantially wider than the passages and then dividing each land into a pair of auxiliary lands separated by intercommunicated trapping ports. The pair of auxiliary lands which are to be rotated to face passages from the decreasing volume chambers are each slightly less in width than the passage. The pair of auxiliary lands which are to be rotated to face passages from the increasing volume chambers are slightly greater in width than the passages. The lands are located diametrically opposite from each other.

When the narrow auxiliary lands face passages from the decreasing volume chambers, trapping does not result. The arrangement is such that when a decreasing volume

2

chamber communicates with a trapping port, fluid can flow therefrom through the intercommunicated trapping port to an increasing volume chamber. The hydraulic fluid acts like a motor in the increasing volume chambers and its energy received from the decreasing volume chambers is to a large extent recovered.

With this arrangement some cavitation results in the increasing volume chambers. However, if the position of the wider pair of the auxiliary stops is moved from the point of minimum chamber volume over a 90 degree arc of the increasing volume chambers, the effect is such that when the increasing volume chamber with a void or vacuum therein comes into communication with a manifold, it is the inlet manifold and the problem of hammering does not result.

With this arrangement, however, when the stops are adjusted to a zero-volume, output position, for example, the chambers formed by the gear teeth at closed mesh contain fluid at high pressure. This high pressure exerts a radial force on the gears which forces the ends of the gear teeth at open mesh into pressure engagement. Excessive wear results.

If the wider of the stops is varied from a point of maximum volume of the chambers over a 90 degree arc of the increasing volume chambers, the chambers formed by the teeth at open mesh contain fluid at high pressure which force the gear teeth together at closed mesh. This is not objectionable. However, with this arrangement cavitation resulted and when the increasing volume chamber came out from under the wide auxiliary stop, the void or cavity was filled from the high pressure manifold and the hammering and loss of efficiency above referred to resulted.

The present invention contemplates a hydraulic pump of the general type referred to which overcomes all the above-referred to difficulties and others, provides a pump wherein there are no problems of trapping or cavitation, and where the open mesh gear teeth form the high pressure chambers.

In accordance with the present invention, there is provided a hydraulic device such as a pump or motor, comprised of a plurality of members relatively movable to define a plurality of revolving increasing and decreasing volume chambers; a passage of circumferential width a rotating with and leading from each chamber and separated by lands of circumferential width f ; high and low pressure manifold facing said passages; a pair of mainlands of circumferential widths d , g , a pair of intercommunicated trapping ports facing said passages, one in each of said mainlands of line of movement widths c , e , and dividing each main-stop into a pair of auxiliary stops of line of movement widths b_1 , b_2 , b_3 , b_4 , each less than the width a . With this arrangement, the inlet and outlet manifolds will each be in communication with one of the trapping ports for an instant as each passage moves past an auxiliary stop. The relative line of movement widths and/or the angular relationship of the trapping ports and lands are all so constructed and arranged that the high and low pressure manifolds are never in communication with a trapping port at the same instant.

Thus, to prevent the low pressure manifold from coming into communication with the high pressure manifold around any one land, the following dimensional relationships are maintained: (1) b_1 , b_2 , b_3 , b_4 are all less than a , and d and g are greater than a ; (2) either c is less than f , or d is greater than $2a+f$, and either e is less than f , or g is greater than $2a+f$. These dimensional relationships must hold for both an even or odd number of passages.

If an even number of passages are used, then to prevent intercommunication around diagonally opposite

auxiliary lands through the inter-communicated trapping ports, the major lands are diametrically opposed and e is less than c by an amount at least equal to the amount that b is less than a . Alternatively e is the same width as c , and one of the main lands is offset from the diametrical line through the other main land by an amount h at least equal to the amount that b is less than a . Alternatively, a combination of a difference in width of the trapping ports, namely the width e and c , together with an offset of the main lands from the diametrical line to the other land may be employed.

With the arrangement of lands as above defined, trapping and cavitation are at all times prevented. It is thus possible to rotate the lands relative to the points of minimum and maximum volume of the chambers to obtain a variation in the output volume of the pump. While this rotation may be in either direction, in accordance with a limited aspect of the invention, the lands are rotated in a direction such that the low pressures in the case of an internal gear pump will exist over the portion of the gears where the teeth are in closed mesh.

Obviously the lands can be fixed and the volume varied by other means such as by adjusting the eccentricity of the members.

The principal object of the invention is the provision of a new and improved variable volume, hydraulic pump or motor, of the type wherein the chambers rotate, and the inlet and outlet manifolds are rotated relative to the chambers to vary the output volume, wherein the problems of leakage, trapping, and cavitation are eliminated, or minimized.

Another object of the invention is a provision of a new and improved hydraulic pump of the general type described which has a high volumetric and mechanical efficiency which is economical to construct and will be safe and dependable in service.

Another object of the invention is the provision of a hydraulic pump or motor having new and improved means for minimizing leakage and preventing trapping or cavitation.

Another object of the invention is the provision of a hydraulic pump or motor of the type generally described wherein the chambers rotate and the lands separating the inlet and outlet manifolds are provided with inter-communicated trapping ports, the dimension all being so relatively controlled that leakage is minimized and trapping and cavitation are prevented.

The invention may take physical form in certain parts and arrangement of parts, the preferred embodiment of which will be described in detail in this specification and illustrated in the accompanying drawing which is a part hereof and wherein:

FIGURE 1 is a side cross-sectional view of a variable volume hydraulic pump illustrating an embodiment of the present invention taken approximately on line 1—1 of FIGURE 5.

FIGURE 2 is a cross-sectional view of FIGURE 1 taken approximately in the line 2—2 thereof and showing in detail the manifold plate;

FIGURE 2A is a modification of FIGURE 2 showing a different arrangement.

FIGURE 3 is a cross-sectional view of FIGURE 1 taken approximately in the line 3—3 thereof and showing the arrangement of a ported disc containing passages communicating with the pump chambers;

FIGURE 4 is a cross-sectional view of FIGURE 1 taken approximately in the line 4—4 thereof and showing the pump elements defining the pump chambers; and,

FIGURE 5 is a cross-sectional view of FIGURE 1 taken approximately in the line 5—5 thereof and showing the inlet and outlet manifolds.

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 purpose of

limiting same, the various figures show a housing 10; a drive shaft 11; inner and outer members 12, 13 respectively, defining therebetween increasing and decreasing volume chambers 14, 15 respectively; a ported disc 16 rotatable with the inner member and having a plurality of passages 17, one for each chamber 14, 15 for the purpose of communicating the chambers with inlet and outlet manifolds 18, 19; a sealing disc 20 mounted on the shaft 11, and a manifold housing 21, the adjustment of which will vary the output volume of the pump as will appear hereinafter.

The housing 10 may take a number of different forms but in the embodiment shown, is comprised of a central cylindrical portion 25, a left-end plate 26 having an opening 27 through which the shaft 11 extends, and a right-end plate 28 in which the inlet and outlet manifolds 18, 19 are formed. The members 25, 26 and 28 all have abutting surfaces in sealing engagement to define closed interior cavity 65 in which the various members of the pump are positioned. Bolts 30 and nuts 31 hold the members in assembled relationship.

The drive shaft 11 has mounted thereon in side by side relationship, a sealing disc 20, the inner member 12 and the ported disc 16, all of which rotate with the shaft.

The inner and outer members 12, 13 may take a number of different forms but in the embodiment shown are respectively, externally and internally toothed gear members with the inner member having one, or more, less teeth than the outer member. The outer member is rotatably supported on an axis spaced from the axis of the shaft 11 by an eccentric ring 32 mounted on the inside of the housing 10. The eccentricity of the ring, the shape and dimensions of the teeth are all so proportioned that one portion of each tooth is always in sliding, sealing contact with a portion of the tooth on the opposite member to define chambers sealed from each other circumferentially. Thus, as the shaft 11 rotates, the inner and outer members 12, 13 also rotate to cause the chambers 14, 15 to increase and decrease in volume. With this arrangement, the chambers will have a point of minimum volume indicated generally by the point "n" and a point of maximum volume indicated by the point "x" located on what may be termed a neutral axis. Assuming a clockwise direction of rotation, as viewed in FIGURE 4, the chambers just past the point "n" will be increasing in volume and are indicated by the reference character 14 while the chambers which have just moved past the point "x" are decreasing in volume and are indicated by the reference character 15. Obviously, if the direction of rotation were reversed, this situation would also be reversed.

It is to be noted that the chambers 14, 15 extend axially and that the members 12, 13 have a substantial, but equal, axial length relative to their diameter. The axial length of the eccentric ring 32 is slightly less than that of these members.

The ported disc 16 provides passages 17 each of controlled line of movement width a for communicating each chamber 14, 15 with the inlet and outlet manifolds 18, 19. Lands 52 of a width f separate the passages 17. As shown, the ported disc 16 is keyed to rotate with the inner member 12 by means of pins 34 and one passage 17 for each chamber 14, 15. Preferably, the axis of each passage 17 corresponds with the root of the teeth of the inner member 12. The line of movement width a of each passage 17 is preferably equal to one half the pitch of the teeth on the inner member 12.

The ported disc 16 is supported for rotation in a ring 36 also mounted on the inside of the housing 10 and has one surface in sealing contact with the adjacent surface of the members 12, 13 and the other in sealing contact with a corresponding surface on the manifold housing 21.

As the inner and outer members 12, 13 rotate and the

5

chambers alternately increase and decrease in volume, hydraulic fluid is respectively sucked into and discharged from such chambers through the passages 17.

To assist the flow of fluid into the increasing volume chambers, the passages 17 are sloped outwardly in a direction towards the chambers whereby centrifugal forces on the hydraulic fluid assist its movement through the passages into the increasing volume chambers.

Furthermore, the passages are sloped in the direction of rotation away from the chamber whereby as the passages 17 move by the inlet manifold, a scooping action results and the inertia of the hydraulic fluid is employed to assist its being forced into the increasing volume chambers.

The inlet and outlet manifolds 18, 19 are formed in the end plate 28, and communicate with the exterior of the end plate 28 through passages 40, 41, each terminating in a suitable means such as the fitting 42 for connecting hydraulic fluid pipes to the pump.

The inlet and outlet manifolds are communicated with the passages 17 as they rotate through arcuate ports 45, 44 respectively in the manifold housing 21. These ports 44, 45 are spaced from the axis of rotation of the shaft 11 the same distance as the right-hand end of the passages 17 so that as the passages 17 rotate, they will communicate alternately with the ports 44, 45. The adjacent ends of the ports 44, 45 and thus the inlet and outlet manifolds 18, 19 are sealingly spaced from each other by main lands 47, 48 of line of movement widths g , d . The land 47 is, in effect, divided into two auxiliary lands 47a and 47b of widths b_3 , b_4 by a trapping port 49 of a width e . In a like manner, the stop 48 is divided into two auxiliary lands 48a, 48b of widths b_1 , b_2 separated by a trapping port 50 of a width c communicated with the trapping port 49 through a passage 51 formed between the outer edge of the plate 21 and the housing 10.

The line of movement width of the auxiliary lands 47a, 47b, 48a, 48b and the trapping ports 49, 50 as well as the positioning of the lands 47, 48 relative to each other, form an important part of the present invention.

In all cases, the line of movement widths b_1 , b_2 , b_3 , b_4 (referred to jointly hereinafter as b) of each auxiliary land are not greater than and preferably slightly less than line of movement width a of the passages 17. The width of each auxiliary land need not be exactly equal to the other so long as the relative dimensions to be indicated are maintained.

Because of the narrowness of the auxiliary lands, high pressure fluid can flow around the lands when they are opposite any one passage into the trapping ports or from the trapping ports to the inlet or low pressure manifold. If adjacent auxiliary lands, that is, lands of any one main land happen to be simultaneously opposite a passage, it will be appreciated that it might be possible for hydraulic fluid to flow from the high pressure manifold to the trapping port and then to the low pressure manifold. To prevent this, the dimensions b and c are carefully controlled in relation to the dimensions a and f . Thus, in accordance with the invention, d and g are each greater than a and either c is less than f or d is greater than $2a+f$ and either e is less than f or g is greater than $2a+f$. With these dimensions, at no time will two passages be simultaneously directly opposite to adjacent auxiliary lands. This relationship must be maintained whether there are an even or an odd number of passages 17 leading from the chambers. It will be appreciated that the number of passages correspond to the number of chambers. With internal gear type pumps and with axial discharge through a ported disc 16 as shown, the number of passages will normally conform to the number of teeth on the gear 12. The same is true if a radial passage inwardly through the gear 12 is employed. However, if radial passages with the chambers outwardly through the gear 13 are employed, the number of passages

6

will be greater depending upon the excess of teeth in the gear 13 over that of the gear 12.

In the event there are an even number of passages 17, there is then a possibility that high pressure fluid can leak around one auxiliary land through the trapping ports and thence around a diametrically opposite auxiliary land to the low pressure manifold. The present invention contemplates two separate arrangements for preventing this. Thus, in FIGURE 2, the width e is made equal to the width c and the land 47 is displaced from the diametrical line through the land 48 by an amount such that the dimension h shown in FIGURE 2 is at least equal and preferably slightly greater than the difference between the average b i.e. $(b_1+b_2+b_3+b_4)/4$, and a .

In FIGURE 2A the lands 47 and 48 are diametrically opposite one from the other and the dimension e is made less than the dimension c by an amount at least equal and preferably slightly greater than the difference between b and a .

In either event, leakage from the high pressure manifold to the low pressure manifold around the stops is completely prevented. Also because each decreasing volume chamber never at any time has its discharge passage completely closed, is never subjected to trapping pressures. In a like manner because no increasing volume chamber ever has its inlet passage closed, no cavitation will result.

The lands are all preferably so relatively dimensioned that as one passage is cut off from the high pressure chamber and this discharging fluid is discharged into the trapping ports which are at that instance closed, the trapping ports will then come into communication with a low pressure chamber so that the fluid being discharged into the trapping port will flow into an increasing volume chamber and its energy will be recovered.

It is to be noted that with reference to FIGURE 2 that one of the manifolds will have a less circumferential extent than the other manifold so that the chambers will then receive and discharge slightly different amounts of fluid. In actual practice however, the offset angle of the lands is so small and the ratio of inlet and discharge into the land at full volume is so small that only a slight decrease in pressure in the trapping ports from its normally expected one half pressure results. This means that slightly more fluid has leaked into the trapping chambers from the high pressure manifold than leaks from the trapping ports into the low pressure manifolds.

It will be noted that the trapping ports 49, 50 are intercommunicated by a passage 53 in the left hand face of the manifold plate 21 and the edges of this passage 53 generally parallel the adjacent or inner surfaces of the inlet and outlet ports 45, 44. The result is that the sealing surfaces on the left hand side of the manifold plate 21 are all symmetrical one to the other so that they will have uniform pressure gradient there across resulting in increased sealing efficiency and decreased leakage.

In the embodiment shown, the manifold housing 21 is positioned so that the trapping port 50 is located opposite the point of maximum volume of the chambers. The pump will thus have its maximum volume output for any given speed.

So that the output volume of the pump may be readily varied, the manifold housing 21 is rotatable in housing 10 and for this purpose has a shaft 55 extending outwardly through the right hand side of the housing and a handle 56 enables the shaft 55 and control plate 21 to be readily rotated to any position.

In accordance with the invention, the land 48 located opposite the point n is preferably rotated opposite to the direction of rotation so that the high pressure exists in the chambers at the open mesh of the gears. The radial force resulting from these pressures forces the gears into pressure engagement at closed mesh where there is a substantial area of relative engagement. If the lands were rotated in the opposite direction the high pressure would

exist between the gear teeth at closed mesh and would force the teeth into pressure engagement at open mesh. As will appear from FIGURE 4 the engagement at this point is rather limited and substantial wear would result.

In the embodiment shown, the manifold housing 21 is positioned so that the trapping ports 49, 50 are located opposite the points of minimum and maximum volume, "n," "x," respectively. The pump will thus have its maximum volume output for any given speed.

If the trapping ports 50, 49 are moved so as to be generally equidistant from the points "n," "x," the pump will have no output volume. Fluid from half of the decreasing volume chambers will flow through each manifold to half the increasing volume chambers and as the rate of increase is the same as the rate of decrease, there will be no net output of the pump. However, substantially all of the pressure energy in the fluid from the decreasing volume chambers will be delivered to the increasing volume chambers as though the increasing volume chambers were a motor, so there is little loss of efficiency resulting from the bypassing of the output fluids from some of the decreasing volume chambers to some of the increasing volume chambers.

The end of the chambers 14, 15 opposite from the ported disc 16 are closed by means of the sealing disc 20 which is in the form of a sleeve mounted on the shaft 11 and having a sealing surface 60 in sealing engagement with the left-hand end of the inner and outer members 12, 13. The sealing disc 20 is supported for axial and rotational movement in the housing 10. In the embodiment shown, it is mounted in a roller bearing 62 held against axial movements relative to the sealing disc 20 by means of a C-ring 63 mounted in a slot in the outer surface of the sealing disc 20. The roller bearing 62, is, in turn, slidable in a rabbet 64 in the inner surface of the central portion 25 of an axial length of the bearing 62.

The sealing disc 20 thus rotates with the inner member 12 but is axially movable relative thereto. While such action may be obtained in a number of different ways, in the embodiment of the invention shown, the sealing disc 20 is preferably rigidly attached to the shaft 11 in any suitable manner but preferably with an interference fit.

The inner member 12 is slidably mounted on the shaft 11, however, and keyed against rotation relative thereto by the key and keyway 37. The ported disc 16 is slidable axially along the shaft 11 along with the inner member 12.

The hydraulic pressures in the chambers in communication with the outlet manifold exert an axial force on the sealing surface 60 tending to separate the sealing disc 20 from the left-hand end of the inner and outer members 12, 13. Likewise, such pressures will tend to force a separation between the surfaces of the ported disc 16 and the control plate 21. In either case, leakage of the pump will result. To prevent this happening, means are provided for urging the sealing disc 20 towards the inner and outer members 12, 13 with a force sufficient to resist the hydraulic forces tending to separate the disc and the two members. As the output pressure of the pump may vary and thus the force tending to separate the disc and the two members may likewise vary, it is preferred that the force tending to move the disc 20 toward the members 12, 13 be made proportional to the pressure in the outlet manifold. Thus, in the embodiment of the invention shown, the cavity 65 to the left of the sealing disc 20 is communicated with the outlet manifold 19 by means of a passage 66 formed in both the central portion 25 and the end plate 28. As the amount of fluid flowing through this passage is almost negligible, the size of the passage may be made relatively small.

The outlet manifold pressure in the chambers 14, 15 will only exert a force on effectively one-half of the cross-sectional area of the sealing surface 60 bearing against the ends of the members 12, 13. To exactly counteract this force, preferably approximately one-half of the cross-sectional area of the sealing disc 20 on the side remote

from the sealing surface 60 is exposed to the hydraulic pressure in the cavity 65.

To prevent the hydraulic pressure in the cavity 65 from contacting the entire cross-sectional area of the sealing disc 20, an auxiliary sealing disc is provided comprised of a sleeve 68 slidably supported in a counterbore 69 in the surface of the end member 26 facing the cavity 65. An O-ring 70 fits around the sleeve 68 and provides a sealing action to prevent the fluid in the cavity 65 from leaking around the outer surface of the sleeve 68. The sleeve 68 has an axially and radially extending flange 71 on its right-hand end of the sealing disc 20. Inwardly of the flange 71, the space 75 is sealed from the cavity 65 pressures. The flange 71 has a surface 73 opposite from the surface 72 of an area equal to one-half the area of the surface 72. Hydraulic pressure in the cavity 65 exerts an axial force to the right on the surface 73 exactly equal and opposite to the hydraulic pressures to the left on the surface 72. In both cases practically friction free operation results.

A coil spring 74 positioned within the sleeve 68 biases the entire assembly to the right. The force of the spring 74 need only be nominal, however.

The areas of the sealing disc or surface 73 exposed to the outlet manifold pressures may be increased beyond that stated, if desired, while still being within the invention or made less if the pressure of the spring 74 is increased sufficiently.

With the arrangement shown, and dimensions, the axial hydraulic forces are balanced and the only pressures between the sealing surfaces is due to the spring 74. A minimum or no leakage within the pump results.

One of the problems in a pump of the general type described, however, is that the force of the hydraulic fluids on the various parts of the pump are unequally distributed about the axis and axially along the axis. Thus, in all cases, the outlet manifold will be in communication with approximately one-half of the chambers. The hydraulic pressures in such chambers will exert a force to the left on the sealing surface 60 over one-half of the areas of inner and outer members 12, 13 facing the sealing surface 60 which forces may be generally indicated by the vector 80. This force is resisted by the force indicated by the vector 82 along the axis of the shaft 11. These two forces provide a bending moment proportional to the forces and the spacing indicated by the dimension L between the two vectors.

In a like manner, the hydraulic forces exert a radial force on the surface of the inner member 12 which may be generally summed up by the vector 81. This force is resisted by the force indicated by the vector 83 through the center line of the bearing 62. These two vector forces 81, 83 have a bending moment on the shaft 11 proportional to the distance M between the lines of action of the two vectors.

The forces 80, 82 and the distance L are fixed distances dependent upon the designed capacities of the pump. On the other hand, the distance M may be varied by positioning the center line of the roller bearing 62 closer to or further from, the line of action of the vector 81. The distance M is so proportioned that the bending moment of the forces 81, 83 will be equal and opposite to the bending moment of the forces 80, 82. With such an arrangement, bending of the various parts making up the pump can be held to a minimum and the sealing surfaces will thus always remain in sealing engagement and leakage can be held to a minimum while at the same time having a minimum of friction within the pump.

It will be appreciated that if the bending did take place that certain of the surfaces would be moved into high pressure engagement which means high friction while other of the surfaces would be moved apart which would permit leakage to occur.

It is to be noted that the sealing disc 20 has a substantial radial and axial dimension and by virtue of its

interference fit on the shaft 11 contributes substantially to the rigidity of the shaft 11.

It is to be noted that both the sealing disc 20 and the ported discs 16 rotate with the inner member 12. Thus, excellent sealing can be obtained here and there is no problem of friction between the members. Further, the sealing disc 20 and the ported disc 16 have a speed or rotation relative to the outer member 13, equal only to the difference in the number of teeth between the two gears and the pitch of the teeth. Thus, there is little frictional loss here.

It is obvious that the improvements and the invention herein described may be employed in hydraulic motors of either the rotating internal gear type, vane type or rotating cylinder type. For the purposes of simplicity in the claims hereinafter, reference will be made only to a pump with the understanding that whenever a pump is mentioned a motor may be indicated. Outlet manifolds and pressures on a pump become inlet manifolds and pressures on a motor.

The invention has been described with particular reference to a preferred embodiment. Obviously, modifications and alterations differing radically in appearance from the preferred embodiment described will occur to others upon a 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 appended claims.

Having thus described my invention, I claim:

1. A hydraulic device comprised of a plurality of members movable relatively to each other and defining a plurality of chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement and then gradually decreasing in volume until they reach said fixed point of minimum volume, means defining an arcuate inlet and an arcuate outlet manifold including first and second main lands one at each arcuate end of said manifold and sealingly separating said manifolds one from the other, one of said manifolds being at high fluid pressure, an opening for each chamber revolving therewith, each of said openings moving past said lands to alternately communicate its associated chamber with either said inlet or said outlet manifold, the peripheries of said openings being spaced on said line of movement a distance f and said openings having a line of movement width a , said first and second lands each having a line of movement width d and g respectively, each land having a trapping port which communicates with each opening as each opening moves past the respective land, said ports having a line of movement width c and e , respectively, said ports dividing their respective lands into a pair of auxiliary lands of widths b_1 , b_2 , and b_3 , b_4 respectively, and passage means intercommunicating said ports, the improvement which comprises said line of movement dimensions being proportioned as follows: b_1 , b_2 , b_3 , b_4 are each less than a , and d and g are each greater than a , c is less than f and g is greater than $2a+f$.

2. The improvement of claim 1 wherein said lands are diametrically opposed and e is less than c by an amount at least equal to $\frac{1}{4}$ the amount that $b_1+b_2+b_3+b_4$ is less than $4a$.

3. The improvement of claim 1 wherein e and c are equal and one of said lands is offset from the diametrical line through the other land by an amount at least equal to $\frac{1}{4}$ the amount that $b_1+b_2+b_3+b_4$ is less than $4a$.

4. The improvement of claim 1 wherein e is less than c by an amount less than $\frac{1}{4}$ the amount that $b_1+b_2+b_3+b_4$ is less than $4a$, and one of said lands is offset from the diametrical line through the other land by an amount at least equal to $\frac{1}{4}$ the amount that $b_1+b_2+b_3+b_4$ is less than $4a$ minus the amount that e is less than c .

5. A hydraulic device comprised of a plurality of members movable relatively to each other and defining a plurality of pumping chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement and then gradually decreasing in volume until they reach said fixed point of minimum volume, means defining an arcuate inlet and an arcuate outlet manifold including first and second main lands one at each arcuate end of said manifold and sealingly separating said manifolds one from the other, one of said manifolds being at high fluid pressure, an opening for each chamber revolving therewith, each of said openings moving past said lands to alternately communicate its associated chamber with either said inlet or said outlet manifold, there being an even number of said openings, the peripheries of said openings being equally spaced by a predetermined line of movement width, said openings having a predetermined line of movement width, said lands being diametrically opposite one from the other and each having a trapping port which communicates with each opening as the opening moves past said lands, passage means intercommunicating said ports, said ports dividing each land into a pair of auxiliary lands of a line of movement width each less than the line of movement width of said openings, the improvement which comprises said main lands and ports having line of movement widths in relation to the line of movement widths of said openings and the spacing therebetween that leakage will not occur around a main land from a high to a low pressure manifold, and the diametrically opposite lands being so arranged relative to each other that leakage will not occur from a high pressure manifold around an auxiliary land to a port and around a diametrically opposite auxiliary land to the low pressure manifold.

6. The combination of claim 5 wherein said auxiliary lands have a line of movement width less than said passages.

7. The combination of claim 5 wherein said auxiliary lands have a width less than the width of a passage and the total width of each main land is greater than the width of two passages and the land therebetween.

8. A hydraulic device comprised of a plurality of members movable relatively to each other and defining a plurality of pumping chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement and then gradually decreasing in volume until they reach said fixed point of minimum volume, means defining an arcuate inlet and an arcuate outlet manifold including first and second lands one at each arcuate end of said manifold and sealingly separating said manifolds one from the other, one of said manifolds being at high fluid pressure, an opening for each chamber revolving therewith, each of said openings moving past said lands to alternately communicate its associated chamber with either said inlet or said outlet manifold, each land having a port which communicates with each of said openings as they move past said lands, said ports effectively dividing each land into a pair of auxiliary lands, passage means intercommunicating said ports; the improvement which comprises: each auxiliary land having a line of movement width less than the line of movement width of said openings and each main land having a line of movement width greater than twice the line of movement width of each opening.

9. The improvement of claim 8 wherein said trapping ports are diametrically opposite.

10. The improvement of claim 8 wherein said trapping ports are of unequal line of movement widths.

11. The improvement of claim 8 wherein at least one

11

of said main lands have a line of movement width greater than the circumferential width of two adjacent rotating passages including the land therebetween.

12. The improvement of claim 8 wherein at least one of said trapping ports have a line of movement width less than the circumferential width of the land between two adjacent rotating ports.

13. The improvement of claim 8 wherein all of said auxiliary lands are of the same line of movement width.

14. A hydraulic device comprised of a plurality of members movable relatively to each other and defining a plurality of chambers revolving in a fixed closed path of movement, said chambers gradually increasing in volume after they pass a fixed point of minimum volume on said path of movement until they reach a fixed point of maximum volume on said path of movement and then gradually decreasing in volume until they reach said fixed point of minimum volume, means defining an arcuate inlet and an arcuate outlet manifold including first and second main lands one at each arcuate end of said manifold and sealingly separating said manifolds one from the other, one of said manifolds being at high fluid pressure, an opening for each chamber revolving therewith, each of said openings moving past said lands to alternately communicate its associated chamber with either said inlet or said outlet manifold, the peripheries of said openings being spaced on said line of movement a distance f and said openings having a line of movement width " a ," said first and second lands each having a line of movement width d and g respectively, each land having a trapping port which communicates with each opening as each opening moves past the respective land, said ports having a line of movement width c and e respectively, said ports dividing their respective lands into a pair of auxiliary lands of width $b1$, $b2$ and $b3$, $b4$, respectively, and passage means intercommunicating said ports, the improvement which comprises said line of movement dimensions being proportioned as follows: $b1$, $b2$, $b3$, $b4$ are each less than a , and d and g are each greater than a , and d is greater than $2a+f$ and b is less than f .

15. The improvement of claim 14 wherein said lands are diametrically opposed and e is less than c by an amount

12

at least equal to $\frac{1}{4}$ the amount that $b1+b2+b3+b4$ is less than $4a$.

16. The improvement of claim 14 wherein e and c are equal and one of said lands is offset from the diametrical line through the other land by an amount at least equal to $\frac{1}{4}$ the amount that $b1+b2+b3+b4$ is less than $4a$.

17. The improvement of claim 14 wherein e is less than c by an amount less than $\frac{1}{4}$ the amount that

$$b1+b2+b3+b4$$

is less than $4a$, and one of said lands is offset from the diametrical line through the other land by an amount at least equal to $\frac{1}{4}$ the amount that $b1+b2+b3+b4$ is less than $4a$ minus the amount that e is less than c .

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