United States Patent

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[54] **HYDRAULIC PUMP OR MOTOR**

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[52] **U.S. Cl.** **418/71,** 418/75, 418/79, 418/171

[51] **Int. Cl.**.....**F01c 1/10,** F03c 3/00, F04c 1/06

[58] **Field of Search**.....418/71-75, 79, 418/166, 171, 180

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[57] **ABSTRACT**

A high pressure hydraulic motor (or pump) of the type comprised of an externally toothed gear inside of an internally toothed ring gear in turn rotating on a spaced axis inside of a sleeve bearing surface wherein grooves are provided in the bearing surface generally on the neutral axis which are communicated to high pressure in combination with a low pressure groove on the low pressure side of the ring gear communicated with low pressure to prevent distortion of the ring gear under the unbalanced high hydraulic pressures on the side thereof. Two additional grooves are provided, one on each side of the low pressure groove, which are alternately communicated with low pressure as the gears rotate to shift the direction of the hydraulic forces on the outside of the low pressure side of the ring gear as the maximum volume chamber switches from high to low pressure (or vice versa) and shifts the direction of hydraulic forces on the inside of the high pressure side of the ring gear.

13 Claims, 9 Drawing Figures

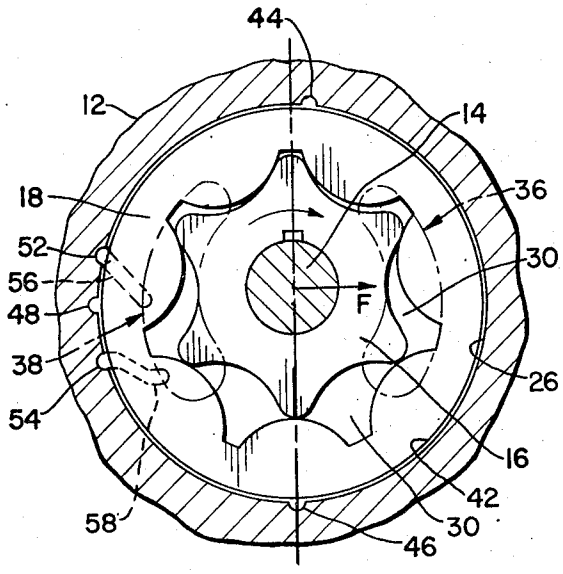
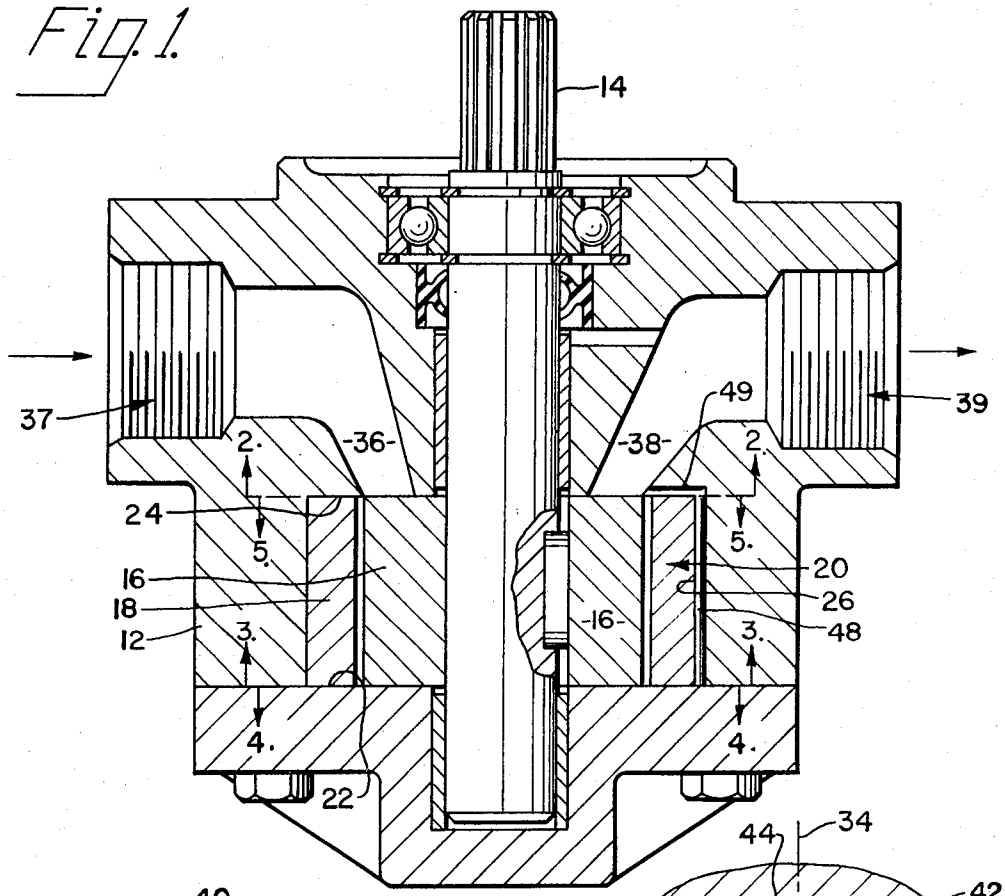


Fig. 1.



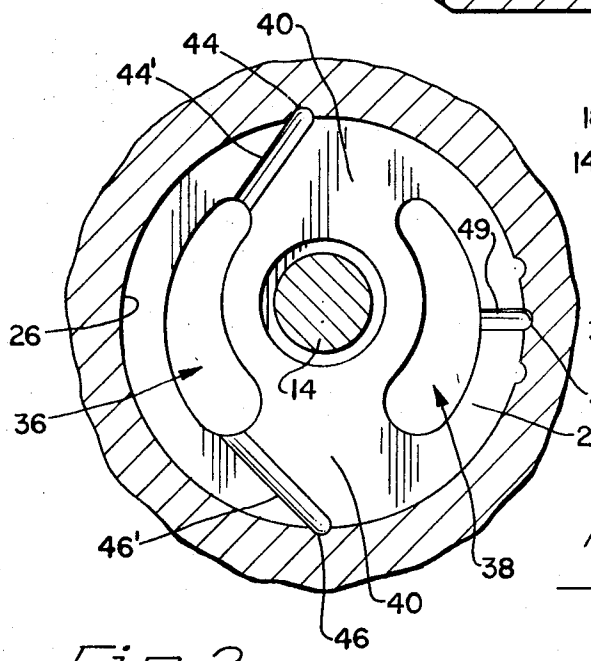


Fig. 2.

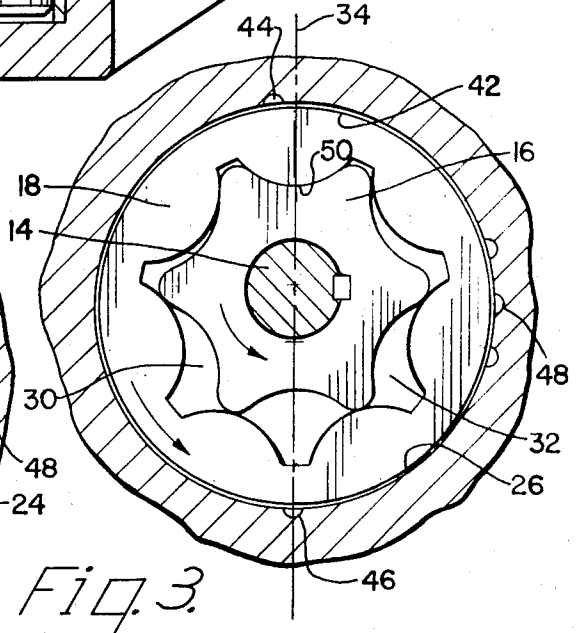
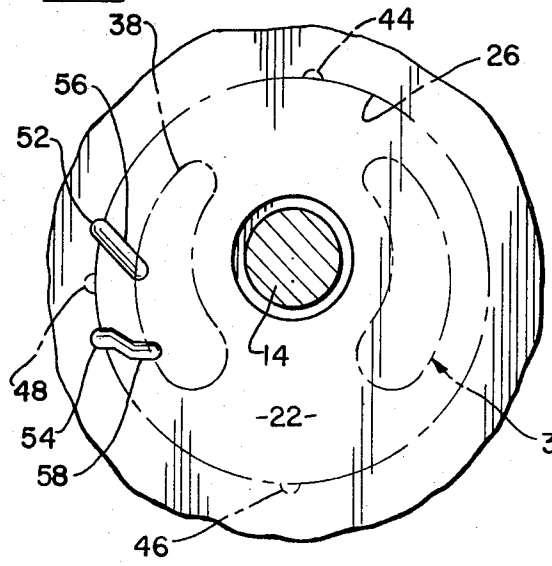


Fig. 3.

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Fig. 4.



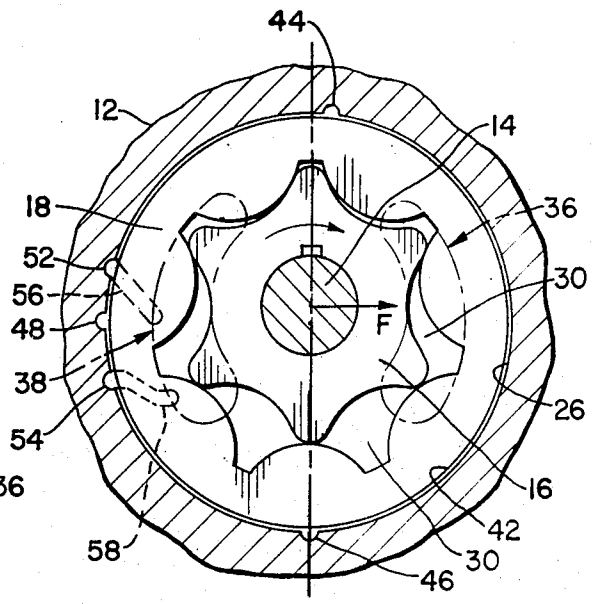


Fig. 5.

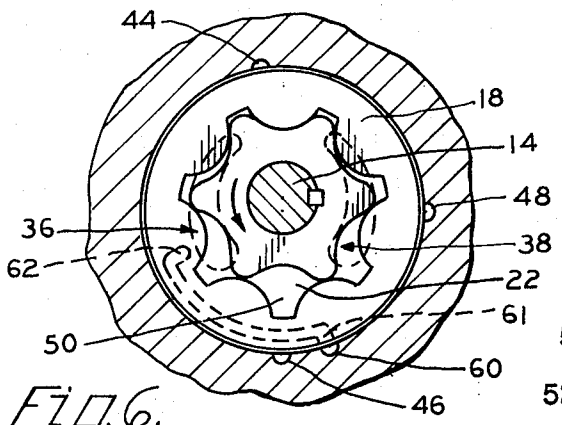


Fig. 6.

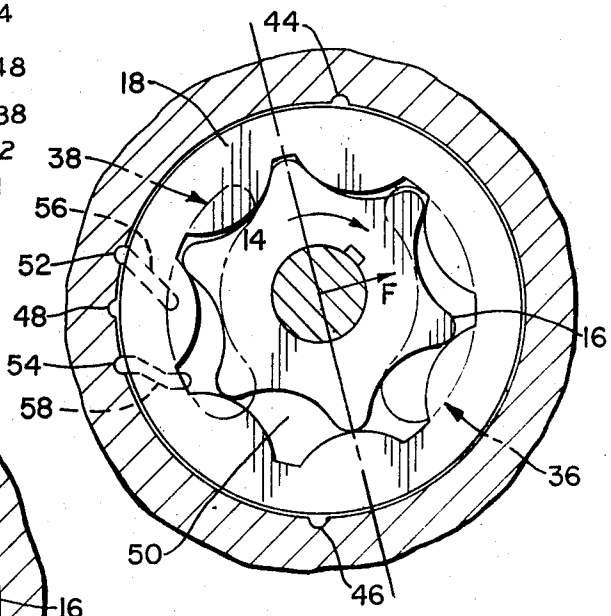


Fig. 5B.

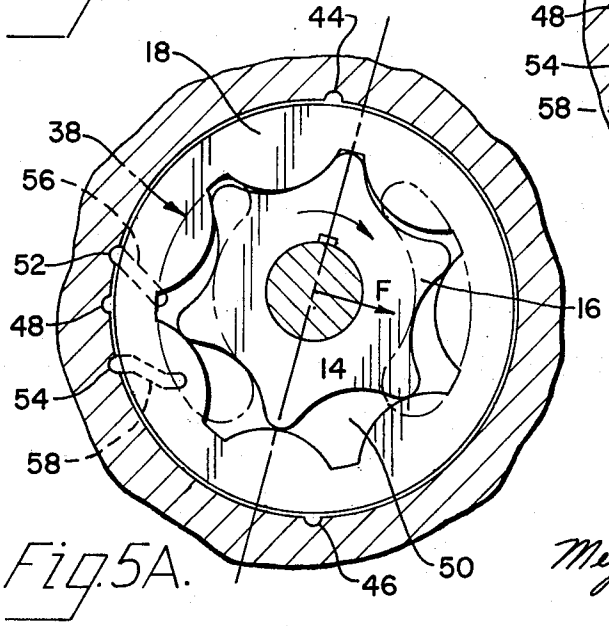


Fig. 5A.

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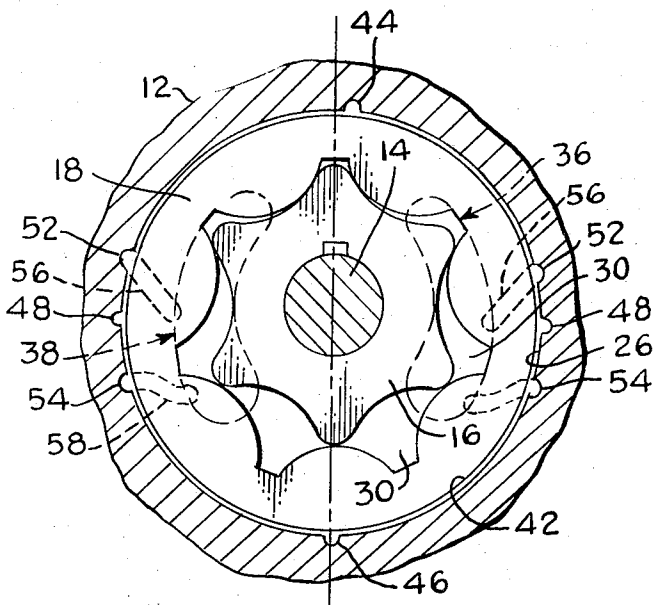


Fig. 7.

HYDRAULIC PUMP OR MOTOR

This invention relates to the filed of hydraulic devices and, more particularly, to an improved arrangement for maintaining the sealing engagement of the gear teeth in internal gear pumps or motors of the type commonly known as gerotors.

Although the invention is particularly applicable to internal gear or gerotor type motors and will be described with particular reference thereto, it will be appreciated that the invention has broader applications and may be employed with other types of hydraulic devices, including pumps, such as external gear type devices.

For the purposes of simplicity of disclosure, the invention will be described primarily in relation to a hydraulic motor 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 high and low pressures respectively. It will be appreciated that, as applied to a hydraulic pump, the relationship of the low and high pressures will be reversed. Moreover, as will be understood, reversal of direction of rotation of the hydraulic device will merely reverse the inlet and outlet ports and manifolds and the high and low pressure chambers.

Hydraulic devices of the type to which this invention is applicable are normally comprised of a housing, a fluid cavity in the housing and a shaft extending into and rotatably supported in the housing in suitable bearings. A plurality of displacement members such as an internally toothed ring gear and an externally toothed pinion gear are disposed in the cavity and rotate with the shaft in a bearing surface eccentric to the axis of the shaft to define a plurality of increasing the decreasing volume fluid chambers. Inlet and outlet ports are formed in the housing and communicate with these chambers. Normally, when functioning as a motor, the chambers increasing in volume communicate with an inlet port and are at relatively high hydraulic pressures while the chambers decreasing in volume communicate with an outlet port and are at relatively low hydraulic pressures.

A line drawn through the points of minimum volume and maximum volume define what may be termed a neutral plane or axis. The gear teeth, particularly at the point of maximum volume or open mesh position, should be in sealing engagement with each other thereby to separate the fluid pressures in the increasing volume chambers on one side of the neutral axis from the fluid pressures in the decreasing volume chambers on the outer side of the neutral axis. To the extent that the teeth are not in sealing engagement and fluid pressure in the high pressure chambers is permitted to leak over into the low pressure chambers on the opposite side of the neutral axis, efficiency of the device is thereby impaired.

Many factors may cause the teeth not to seal properly but a primary factor is the substantial pressure differentials inherently present in devices of this type. Thus, in normal operation, high hydraulic pressures act over a substantial arc on one portion of the inner surface of the ring gear while, simultaneously, a substantial arc on another portion of the ring gear will be subjected to low hydraulic pressures. The net affect of these two widely different pressures acting on diametrically opposite arcs on the ring gear at the same time is a

tendency to cause the ring gear to assume an oval shape with the long axis generally coincident with the neutral axis. Separation of the teeth at open mesh results. As there is a high pressure differential at this point internal leakage results.

My U.S. Pat. No. 3,043,446 issued May 15, 1962 and No. 3,427,983 issued Feb. 18, 1969 provided partial solutions to this problem by using pressure balancing on the outer surface of the ring gear. These patents introduced high hydraulic pressure continuously to approximately an 180° arc of the clearance space between the outer peripheral surface of the ring gear and the bearing surface in which the gear is rotatably suppored on the high pressure side of the neutral axis. A return groove in the bearing surface on the low pressure side of the neutral axis is continuously communicated to low pressure. There is thus continuous fluid flow in the clearance from high to low pressure and thus a pressure gradient between the high pressure and low pressure arcs. There is a uniform high pressure on the high pressure side of the device. These pressures exert hydraulic forces on the outer surface of the ring gear to reduce or prevent the tendency of the gear to assume an oval shape.

Although the foregoing arrangement has been generally satisfactory, it has not provided a complete solution to the problem. One reason for this appears to be that the direction of the resultant of all the forces of the hydraulic pressures in the high pressure chambers continually shifts as the fluid chamber defined by the gear teeth at the open mesh position continually and abruptly switches between a high pressure condition and a low pressure condition as the chamber at open mesh moves from communication with the high pressure port to communication with the low pressure port. Thus, at one position, the open mesh chamber will be at high pressure while, after a few degrees of rotation of the gears, the chamber will abruptly switch from high pressure to low pressure. As the hydraulic pressures of this "switch" chamber change, the direction of the resultant of the high pressure forces acting on the inside of the ring gear shifts. When this occurs, the return groove particularly in U.S. Pat. No. 3,427,983 is no longer in the ideal position to create the proper opposing force. The result is that hydraulic forces on the external surface of the ring gear which may have assured a good sealing relationship between the meshing teeth when the switch chamber was at high pressure are less than satisfactory to maintain the same condition with the switch chamber is at low pressure. The net effect is that the operating characteristics of the device depends, at any given instant, on the rotated relationship of the ring gear and the pinion relative to the port. For example, it has been found that under stalled conditions the volume of leakage in a device of this type may be increased six fold simply by rotating the device a few degrees.

One prior way of preventing this leakage has been to force the ring gear into an oval shape with the short axis on the neutral axis. Such an arrangement tends to overseal at times with a resultant tooth wear and loss of torque through internal friction. At low speeds, and when the device is a hydraulic motor supplied by a constant volume pump, the varying leakage rate can be a significant factor in the output speed. Thus, when the

gears are in the rotated position where the leakage rate is low, the output speed is high. However, when the gears rotate to a position in which the leakage rate is high, the speed is reduced. Such a pulsating speed in many cases is intolerable.

The prior art has recognized this problem of variable leakage in hydraulic devices operating at low speed; however, a satisfactory solution to this problem has not heretofore been found. Thus, the prior art has attempted to mitigate the effect of the problem by using complex and expensive servo valving and feedback circuits. These circuits compensate for variations in leakage by varying the supply of hydraulic fluid. In this manner, the leakage rate remains variable but the speed is maintained at a relatively constant level. Circuits of this type are somewhat complex and it is not unusual to have the cost of such circuits exceed the cost of the hydraulic motor itself.

The present invention contemplates a hydraulic device which remedies the foregoing problems and assures a good sealing relationship between the meshing teeth irrespective to the rotated position of the pinion and ring gears and the pressure condition in the fluid chamber at the open mesh position.

In accordance with the present invention there is provided, in a pressure balanced hydraulic device of the general type described, at least one switched bearing clearance groove which is periodically connected to and disconnected from port pressure as the ring gear rotates and so located as to compensate for the shift of the line of action of the high pressure forces as the pressure in the chamber at open mesh switches from high to low as the gears rotate.

In a pump this groove can be selectively communicated to either a pressure or suction port. In a reversible motor the groove is selectively communicated to a return or discharge port. The location of the groove (or grooves) will depend on whether it is periodically connected to high or low pressure and on the existence and location of other bearing clearance grooves which are continuously connected to either high or low pressure.

In another aspect of the invention, a pair of switched circumferentially spaced grooves are provided in the bearing clearance on the return side of the ring gear. Each of the switched grooves communicates with separate manifold grooves in the end wall of the housing extending radially inwardly to approximately the root circle of the ring gear. The ends of these grooves are so spaced circumferentially, i.e. other than the pitch of the ring gear teeth or multiples thereof, that rotation of the ring gear acts as a valve or switch to alternately interconnect one or the other of the manifold grooves and this its associated switched bearing clearance groove with the low pressure of the device. Preferably a third groove intermediate the pair of grooves communicating continuously to low pressure may be provided.

In accordance with the principles of this invention, as the chamber at open mesh switches from high pressure to low pressure, so also first one switched bearing clearance groove will be in communication with low pressure and then the other switched bearing clearance groove will be placed in communication with low pressure. The result of this switching is to shift the position of the low pressure return groove in the bearing clearance to compensate for the shift of the line of action of the hydraulic forces acting on the ring gear in the high pressure chambers of the device.

In accordance with another aspect of the preferred form of the invention, it is contemplated that switched manifold and bearing clearance grooves may be placed on both the high pressure and low pressure sides of the device so that, for a reversible motor, the switching action may be achieved irrespective of the direction of rotation of the motor.

Further in accordance with the invention the continuous pressure balancing grooves may be so positioned as to over compensate for the inner forces on the ring gear in one arcuate position, i.e. press the teeth at open mesh into engagement at one point in their arc of rotation, in combination with an additional switched bearing clearance groove selectively communicated with a port so as to relieve the over compensation when the ring gear would normally be in its over compensated position.

Alternatively the continuous grooves can be so arranged as to under compensate in one arcuate position in combination with a switched groove selectively communicated with a port to press the ring gear toward the inner gear when the ring gear would normally be in its under compensated position.

It is the principal object of this invention to provide a hydraulic device which can operate both as a high performance high speed motor and an efficient low speed motor.

It is a further object of this invention to provide a hydraulic device in which the sealing relationship between the meshing teeth is maintained irrespective of the line of action of the hydraulic forces in the device and the rotated position of the pinion and ring gear.

It is a more specific object of the invention to provide a hydraulic device in which the point of return pressure in the bearing clearance is shifted automatically as the line of action of the hydraulic forces of the device shifts.

It is a further object of this invention to provide a hydraulic device which drastically reduces the variable leakage and variable output torque when the device is running close to stalling speed.

Other objects, features and advantages of the invention will become more apparent upon a complete reading of the following description which, when taken with the attached drawings, discloses but a preferred embodiment of the invention.

Referring now to the drawings wherein like reference numerals indicate like parts in the various views:

FIG. 1 is a cross-sectional view of an internal gear hydraulic motor illustrating a preferred embodiment of the invention.

FIG. 2 is a sectional view taken along lines 2—2 of FIG. 1.

FIG. 3 is a sectional view taken along lines 3—3 of FIG. 1.

FIG. 4 is a sectional view taken along lines 4—4 of FIG. 1 with the manifold ports being shown in phantom outline.

FIGS. 5, 5A and 5B are sectional views of FIG. 1 taken along the lines 5—5 thereof and illustrating the varying line of action of the hydraulic forces as the gears rotate to three different positions and the manner

in which the switch grooves are selectively connected and disconnected from low pressure.

FIG. 6 is a view similar to FIG. 5 but to somewhat smaller scale and showing an alternative embodiment of the invention wherein a switch groove is selectively connected to high pressure.

FIG. 7 is a view similar to FIG. 5 and shows an alternative embodiment wherein switch grooves are provided on both sides of the neutral axis.

Referring now more in detail to the drawings wherein the showings are for the purpose of illustrating a preferred embodiment of the invention only and not for the purposes of limiting same, FIG. 1 illustrates a hydraulic device such as an internal gear type motor comprised generally of a housing 12, a driven shaft 14, an externally toothed pinion gear 16 keyed to the shaft 14 and an internally toothed ring gear 18 surrounding the externally toothed gear and rotatable therewith.

The housing 12 defines an internal cavity or fluid chamber 20 defined by opposed end walls 22, 24 and a cylindrical outer surface 26. The pinion and ring gears 16, 18 are disposed within the fluid cavity 20 with the ring gear 18 being supported for rotation by the cylindrical bearing surface 26.

The pinion gear 16 is keyed for rotation to the shaft 14 while the internally toothed ring gear is mounted for rotation on an axis spaced from the axis of rotation of the gear 16 and the shaft 14. As is conventional in devices of this type, the ring gear 18 has one or more teeth than the gear 16 with the teeth of both gears being in sliding sealing engagement to form a plurality of fluid chambers 30, 32 which progressively increase and decrease in volume as the gears rotate. As shown in FIG. 3, these gears have a neutral axis 34 which is defined by a line drawn through the maximum and minimum volume points of the chambers 30, 32. The point of minimum volume comprises the closed mesh position of the gears while the point of maximum volume comprises the open mesh condition of the gears.

As illustrated, the opposed wall surfaces 22, 24 are each in sliding sealing engagement with the axial end faces of the gears 16, 18 thereby to close the axial ends of the fluid chambers. The wall 24 has arcuate extending inlet and outlet manifold ports 36, 38 respectively formed therein and spaced from the axis of rotation so as to communicate with the increasing and decreasing volume chambers. The inlet manifold port 36 has a supply passage 37 through which a source of pressurized fluid may be introduced and the outlet manifold port 38 has a passage 39 through which the fluid is discharged. The spaces 40 on the wall 24 between the arcuate ends of the manifolds 36, 38 function as lands to prevent communication from one manifold port to the other. These lands have an arcuate width at least equal to the maximum arcuate width of the gear chambers 30, 32.

The cylindrical surface 26 functions as a sleeve type bearing for the ring gear 18 and, as is normal in devices of this type, the cylindrical surface 26 has a diameter slightly greater than the diameter of the external surface of the gear 18 to provide a bearing clearance space 42 into which hydraulic fluid is introduced.

As shown in FIG. 3, a pair of high pressure grooves 44, 46 are formed in the cylindrical bearing surface 26 connected continuously to the high pressure port by grooves 44', 46' in the wall 24. The high pressure groove 46 is located along the neutral axis 34 of the device while the high pressure groove 46 is located along the neutral axis 34 of the device while the high pressure groove 44 is spaced circumferentially approximately 175° from the high pressure groove 46 and on the high pressure side of the neutral axis 34. A low pressure or return groove 48 is formed in the bearing surface 26 with this groove being positioned 90° from the neutral axis 34 and continuously communicated to the low pressure manifold 38 by groove 49 in wall 24.

The device so far described operates in the following manner. Referring to FIG. 5 and assuming the device operates as a motor, high hydraulic pressure is introduced through the manifold port 36 to the fluid chambers 30 on the high pressure side of the neutral axis 34. This high hydraulic pressure causes a counterclockwise rotation of the pinion gear 16 and the ring gear 18. The neutral axis 34, as shown in FIG. 5, passes through the points of closed mesh and open mesh of the device. As shown in FIG. 5, the high pressure in the chambers 30 creates forces which may be integrated into a single large radially outward load of force F which acts along a line of action approximately perpendicular to the neutral axis 34. At the same time, high hydraulic pressure is also introduced into the bearing clearance 42 through the bearing grooves 44, 46 with the resulting pressure gradients in the bearing clearance creating forces which tend to oppose the internal forces and maintain the ring gear in meshing engagement with the pinion gear thereby minimizing leakage from the high pressure chambers 30 to the low pressure chambers 32.

As shown in FIG. 5, the bearing clearance on the high pressure side of the neutral axis 34 will be at substantially the high pressure of the device while the bearing clearance on the low pressure side of the neutral axis will have a pressure gradient which decreases to the low pressure of the device immediately adjacent the low pressure groove 48.

The problem presented by devices of this type and which is solved by the present invention is best illustrated in FIGS. 5A and 5B. Referring first to FIG. 5A, the pinion gear and ring gear have rotated from the position shown in FIG. 5. In that position, there is a fluid chamber 50 defined by the meshing teeth which is at high pressure but which, as the gears continue to rotate, will suddenly switch to low pressure. Thus, in FIG. 5B the gears have rotated from the position shown in FIG. 5A and the fluid chamber 50 has come into communication with the low pressure outlet manifold 38 thereby communicating the fluid chamber 50 to the low pressure of the device. This chamber 50 thus constitutes a "switch" chamber which switches between high pressure and low pressure depending on the particular rotated position of the gears 16, 18. It is important to note that the position of the force F also varies as the pressure in switch chamber 50 varies. This, as shown in FIG. 5A, the gears have rotated to a position where the chamber 50 is still at high pressure but is almost ready to "switch" to low pressure. In that position, the force F of the device has rotated to a position where it lies substantially 90° from the high pressure groove 44. In this condition the pressure gradients in

the clearance **42** are less effective to maintain the meshing engagement of the gears than they were when the gears where in the position shown in FIG. **5**. Moreover, as the gears rotate from the position shown in FIG. **5A** to that shown in FIG. **5B** and the chamber **50** is placed in communication with the outlet manifold **38**, the force F suddenly shifts to the position shown in FIG. **5B**. Thus, as the device operates, the line of action of force F continuously alternates between the various positions shown in FIGS. **5**, **5A** and **5B**. In such circumstances, it is readily apparent that an arrangement which may be suitable for maintaining sealing engagement of the gears under one set of conditions will not be satisfactory when the gears have rotated to a different position and different force conditions exist. Thus, the teeth may have a good sealing relationship when the gears are in the rotated position shown in FIG. **5** and cross port leakage from the chambers **30** to the chamber **32** may be minimized. However, as the gears rotate to the position shown in FIG. **5A**, the difference force conditions present in that condition of the gears may result in a slight separation of the teeth and leakage may begin to develop. As soon as leakage between the teeth occurs, the efficiency of the device is reduced and, when operating as a motor, the output torque of the device will begin to diminish. The same problem will occur in the rotated position of the gears shown in FIG. **5B**. The magnitude of this problem of variable leakage is illustrated by one test that was conducted and in which it was found that, with a 1.7 cubic inch per revolution motor, cross port leakage varies from 59 to 370 cubic inches per minute depending on the relative position of the gears. This test was conducted by introducing pressure of 2,000 psi at the inlet port and restraining the shaft from rotation while recording the fluid coming out of the return port. In terms of torque variation, this same test indicated that the stall torque varies from 380 to 512 inch-pounds as compared to a theoretical value of 544 inch-pounds at 2,000 psi.

In accordance with the principles of this invention, the problem of variable cross port leakage and variable torque output has been resolved through the use of a pair of switched bearing clearance grooves on the low pressure side of the ring gear communicating with a pair of manifold grooves which are alternately intercommunicated with the low pressure fluid chambers as the pressure condition in the switch chamber changes. Referring more specifically to FIG. **4**, an additional pair of switched return grooves **52**, **54** are formed in the bearing surface **26**. The return grooves are circumferentially spaced on either side of the main return groove **48** by an amount which is approximately equal to the arc through which the neutral axis **34** shifts from the position shown in FIG. **5**. The groove **52** communicates with a manifold groove **56** formed in the end wall **22** while the groove **54** communicates with a similar manifold groove **58** formed in the wall **22**. The grooves **56**, **58** extend to a position slightly radially inward of the root circle of the ring gear **18** and with the inner ends so located and spaced circumferentially as to be valved at the proper time into and out of communication with the fluid chamber **32** as the ring gear rotates and the switch chamber **50** goes from high to low pressure and vice versa. The radially inner ends of the grooves **56**, **58** must be spaced circumferentially a distance different than the pitch diameter of the ring gear **18** or multiples thereof. Thus, as shown in FIG. **5**, the manifold grooves **56**, **58** at one point in the arc revolution of the ring gear **18** are covered by the adjacent axial end face of the gear **18** and only the center groove **48** is in communication with low pressure.

In that condition, grooves **52**, **54** are ineffective and the pressure gradients present in the bearing clearance space **42** are determined by the pressure grooves **44**, **46** and return groove **48**. However, as the gears rotate to the position shown in FIG. **5A**, a different condition exists. Thus, as the line of action of force F rotates to the position shown in FIG. **5A**, the groove **56** becomes effective to communicate the groove **52** and thus flow of fluid from the bearing clearance with the low pressure of the device is blocked by the ring gear and is at the pressure gradient determined by the flow through the bearing clearance. Thus, as shown in that FIGURE the ring gear **18** has rotated to a position wherein the inner end of the groove **56** is communicated with one of the fluid chambers on the low pressure side of the device thereby connecting the low pressure of the device with the groove **52**. The groove **58** remains covered by one of the teeth on the ring gear **18** thereby maintaining the groove **54** ineffective. The net effect of the communication of the groove **52** to the low pressure of the device is to shift the point of return of pressure in the bearing clearance **42** to a point in which is spaced 180° from the line of action of the force F. In other words, the manifolding of groove **52** maintains essentially the same force condition on the gears as was present in the arrangement of FIG. **5** notwithstanding the shifting of the force F to the position shown in FIG. **5A**.

Referring now to FIG. **5B**, essentially the opposite condition now exists. The switch chamber **50** has been dropped to low pressure and the force F has shifted to the position illustrated. In this position of the gears, the groove **56** is once again covered by the ring gear **18** thereby isolating the groove **52** from the fluid chambers of the device. However, the ring gear has now uncovered the groove **58** thereby communicating the groove **54** with the low pressure of the device. It is important to note that with the groove **54** connected to the low pressure of the device once again the bearing clearance **42** is connected to low pressure at a point which is generally 180° from the line of action of force F.

In summary, it is to be noted that in each of the three conditions illustrated in FIGS. **5**, **5A** and **5B**, the return or low pressure of the device is communicated to the bearing clearance at substantially 180° from the line of action of the force F. With this arrangement it is possible to seal the teeth of the gears without requiring unnecessary forces and hence, cut both the friction and the leakage in the device. For example, it has been found that in the test above-described by adding the two manifold grooves, the cross port leakage under precisely the same conditions was reduced to a range of 44 to 47 cubic inches per minute. In other words, with the manifolding groove added, the leakage of the device varied only 3 cubic inches per minute depending on the relative rotated position of the gears. In terms of stalling torque with the manifold grooves, the torque varied from 505 to 525 inch-pounds. Thus, from these

figures it is apparent that the use of the two manifold grooves variable leakage and variable output torque are virtually eliminated even when the motor runs at low speeds and near stalling conditions. As a result, it is entirely practical to produce a high performance, high speed, high pressure motor which is equally capable of operating with good performance at low speeds.

FIG. **6** shows an alternative embodiment of the invention. In this figure like parts are designated with like reference numerals to those of FIGS. **4** and **5**. In this embodiment a switch groove **60** is provided located on the side of the neutral axis remote from the high pressure chambers and adjacent open mesh. This switch groove is communicated with the high pressure by means of a manifold groove **61** which extends in the surface **22** across the neutral axis on a radius greater than the root diameter of ring gear **18** to an end **62** located at a point between the root diameter and the inner diameter of the teeth of the ring gear **18** with the end **62** being so located circumferentially that as the switch chamber **50** switches from communication with the high pressure manifold port **36** to the low pressure manifold port **38**, the end **62** of the manifold groove **61** will come into communication with any one of the high pressure chambers remote from the switch chamber **50**. The groove **60** is located about one half the width of the switch chamber **50** from the neutral axis.

It is thus apparent from the foregoing that the primary concept is characterized in a hydraulic pump or motor in which there is a switch chamber in which the pressure condition switches from high pressure to low pressure and vice versa and in which a stationary groove is manifolded or switched into and out of communication with the fluid chambers of the device as the switch chamber is alternately connected to high and low pressure. Variations from this primary concept will suggest themselves to those having ordinary skill in the art. For example, instead of placing the grooves only on one side of the device it would be equally workable to have switched grooves on both sides of the neutral axis as shown in FIG. **7**. In this way, the hydraulic device would be reversible so that when operating in the manner illustrated in FIGS. **5**, **5A** and **5B** the grooves on the right-hand side of the neutral axis would be effective in the manner described above while the grooves on the left side of the neutral axis would remain ineffective since the entire bearing clearance **42** on that side of the device is already at the high pressure of the device. However, upon reverse rotation of the gears, exactly the opposite condition would obtain and the grooves on the left side of the neutral axis, as viewed in FIG. **5**, would be operative for the desired purpose while those on the right side of the neutral axis would become ineffective.

Still a further modification would be to employ switched grooves in conjunction only with the high pressure grooves **44**, **46**. This would work satisfactorily in a device in which it was intended to operate only in a given direction. However, in a reversible device problems would be incurred since in one direction of rotation return fluid would be introduced to the bearing space at points close to the neutral axis **34** thereby making it more difficult to maintain the sealing engagement of the gear teeth.

Further, it is possible to arrange the location of the continuous pressure grooves so that the tendency for the ring gear to assume an oval shape with the long axis parallel to the neutral axis is over compensated, that is the ring gear tends to periodically assume an oval shape as it rotates with the long axis at an angle, e.g. 90° to the neutral axis and then locate the switched groove (or grooves) closer to the open mesh position than the preferred embodiment and connected periodically to low pressure so as to relieve the over-compensation. In a like manner, the continuous pressure grooves can be so located as to periodically under compensate for distortion and then so locate the switched grooves closer to the closed mesh position than the preferred embodiment so as to compensate for the under compensation.

The essence of the present invention is believed to be the periodic connecting and disconnecting of a line or point on the bearing clearance with either high or low pressure so as to compensate for the varying line of action of the resultant of the forces of the high pressures on the inside of the ring gear as the pressure in the switch chamber changes from high to low or vice versa.

Having thus described my invention, I claim:

1. A housing for a hydraulic device, said housing having a cavity having spaced parallel end surfaces and a cylindrical bearing surface between said end surfaces, an opening through said housing into said cavity having an axis spaced from the axis of said cylindrical surface, the plane through the two axes defining a neutral axis, said cavity having an inlet and an outlet manifold one on each side of said axis one of said manifolds adapted to be at high hydraulic pressure and the other at low hydraulic pressure, said cylindrical bearing surface having two grooves, each groove adjacent one of the ends of said neutral axis and means communicating said grooves with the high pressure manifold, said cylindrical bearing surface on the side of said neutral axis adapted to be at low pressure having a pair of switched grooves and a third groove interposed between said switched grooves, means communicating said third groove with the low hydraulic pressure manifold, a pair of grooves in said spaced end surfaces each communicating with one of said switched grooves and extending radially inwardly with the ends thereof circumferentially spaced.

2. In a hydraulic device including a housing, a cavity in the housing defined by parallel spaced end surfaces and a cylindrical bearing surface, inlet and outlet ports in the housing communicating with the cavity, one of said ports adapted to be at high hydraulic pressure and the other said ports adapted to be at low hydraulic pressure, a rotatable shaft extending into the cavity and having an axis spaced from the axis of said bearing surface, a member secured to such shaft for rotation therewith and having alternate teeth and root portions therebetween on the outer periphery, a cylindrical ring member rotatably supported in said bearing surface and having on the inner surface thereof a plurality of alternate teeth and root portions, the teeth and root portions of said members cooperating to define increasing and decreasing volume chambers therebetween as the members rotate, said bearing surface and said cylindrical member cooperating to define a bearing clearance therebetween, said hydraulic device having a neutral

axis defined by a line drawn through the points of maximum and minimum volume of said chambers, at least some of said chambers on one side of said axis being at high hydraulic pressure and at least some of said chambers on other side of said neutral axis being at low hydraulic pressure, the chamber at maximum volume being a switch chamber which is alternately communicated with the inlet and outlet ports of the device and thus to high and low hydraulic pressures, the improvement which comprises means permanently communicating high hydraulic pressure to the bearing clearance on the high pressure side of the neutral axis, at least one fixed point on said bearing surface adapted to be communicated with a low pressure port and means alternately communicating said point with said low pressure port and cutting it off from said port as said cylindrical member rotates.

3. The improvement of claim 2 wherein said point is a switched groove in said bearing surface.

4. The improvement of claim 2 wherein said alternately communicating means is a groove in at least one of said parallel cavity surfaces extending radially inwardly a distance just beyond the root diameter of said cylindrical member so as to be communicated with and cut off from said chambers as said member rotates.

5. The improvement of claim 2 wherein said high hydraulic pressure means includes two grooves in said cylindrical bearing surface each adjacent an end of said neutral axis and means permanently communicating said grooves with said high pressure manifold.

6. The improvement of claim 2 wherein there are two such points in said bearing surface and means alternately communicating said points with and cutting said points off from the low pressure port as said cylindrical member rotates.

7. The improvement of claim 6 wherein a third point in said bearing surface is positioned between said pair of points and means permanently communicate said third point to said low pressure port.

8. The improvement of claim 6 wherein said alternately communicating means comprise a pair of grooves in at least one of said parallel surfaces extending radially inwardly a distance just beyond the root diameter of said ring gear such as to alternately communicate with and be sealed off from the low pressure chambers as said cylindrical member rotates, the inner ends of said grooves being spaced circumferentially a distance such that only one groove is in communication with a chamber at a time.

9. The improvement of claim 6 wherein a pair of grooves in at least one of the cavity parallel surfaces extend radially inwardly from said fixed points a distance just beyond the root diameter of said cylindrical member such as to alternately communicate with and be sealed off from said chambers as said cylindrical member rotates, the inner ends of said grooves being spaced circumferentially such that only one groove is in communication with a chamber at a time.

10. The improvement of claim 6 wherein said spaced points on said bearing surface are axially extending grooves in said cylindrical bearing surface.

11. The improvement of claim 6 wherein a pair of fixed points are provided on said bearing surface on both sides of said neutral axis adapted to be periodically communicated with the chambers on their respective side of said neutral axis and means alternately communicating each point with said chambers as said members rotate.

12. In a hydraulic device including a housing, a cavity in the housing defined by parallel spaced end surfaces and a cylindrical bearing surface, inlet and outlet ports in the housing communicating with the cavity, one of said ports adapted to be at high hydraulic pressure and the other said ports adapted to be at low hydraulic pressure, a rotatable shaft extending into the cavity and having an axis spaced from the axis of said bearing surface, a member secured to such shaft for rotation therewith and having alternate teeth and root portions therebetween on the outer periphery, a cylindrical ring member rotatably supported in said bearing surface and having on the inner surface thereof a plurality of alternate teeth and root portions, the teeth and root portions of said members cooperating to define increasing and decreasing volume chambers therebetween as the members rotate, said bearing surface and said cylindrical member cooperating to define a bearing clearance therebetween, said hydraulic device having a neutral axis defined by a line drawn through the points of maximum and minimum volume of said chambers, at least some of said chambers on one side of said axis being at high hydraulic pressure and at least some of said chambers on other side of said neutral axis being at low hydraulic pressure, the chamber at maximum volume being a switch chamber which is alternately communicated with the inlet and outlet ports of the device and thus to high and low hydraulic pressures, the improvement which comprises means permanently communicating high hydraulic pressure to the bearing clearance on the high pressure side of said neutral axis, a point in said bearing surface communicating with said bearing clearance said point located on the side of the neutral axis remote from high pressure and adjacent a switching chamber at open mesh, means alternately communicating said point with a second chamber at high pressure and cutting it off from said second chamber as said cylindrical member rotates.

13. The improvement of claim 12 wherein said communicating means comprises an arcuate groove in one of said spaced end surfaces extending from said point radially inwardly a distance just beyond the root diameter of said cylindrical member so as to be communicated with and cut off from said chambers as said member rotates.

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