

[54] **HYDRAULIC PUMP/MOTOR WITH
HYDROSTATICALLY BALANCED ROTORS**

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[22] Filed: **July 2, 1973**

[21] Appl. No.: **376,031**

[52] U.S. Cl. **418/72; 418/73; 418/77;
418/79; 418/81; 418/102; 418/169; 418/178**

[51] Int. Cl. **F01c 1/10; F01c 21/00; F03c 3/00**

[58] Field of Search **418/71-73,
418/75, 77, 79, 81, 102, 107, 168-171, 178**

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Primary Examiner—John J. Vrablik

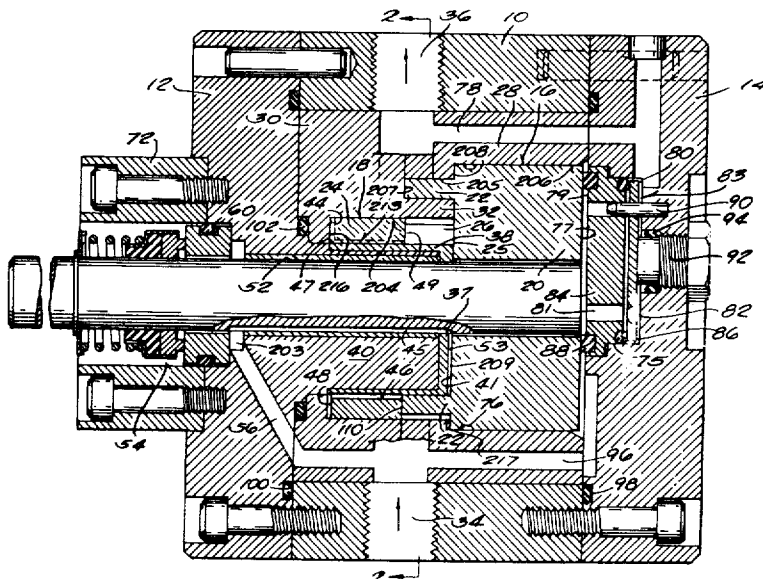
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and Fuller

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ABSTRACT

A power rotor and idler rotor having intermeshing teeth are rotatably mounted within bushings in a housing on offset axes. The idler rotor is smaller in diameter than the power rotor and rotates substantially within the circumference of the power rotor teeth. The teeth of the two rotors mesh in one portion of their circumference and are separated by a crescent baffle in the diametrically opposed portion of their circumference. The power rotor teeth make sealing contact between the outer surface of the crescent baffle and between its bores and the adjacent end surfaces of the working chamber within which the teeth rotate and act to pump fluid from an inlet port in the housing to an outlet port therein or to be driven by fluid under pressure moving from one port to the other. The idler rotor teeth make sealing contact between the inner surface of the crescent baffle and the diametrically opposite part of its bore section and between the idler rotor boss sleeve bearing and the adjacent end surfaces of the working chamber within which the idler rotor teeth rotate. The meshing portion of the idler rotor teeth and power rotor teeth are shaped in the form of matching concave-convex surfaces having different radii of curvature to provide for rolling contact thereinbetween. Hydraulic pressure chambers are formed between the hub of the idler rotor and its bearing to balance the axial, radial and tilting hydraulic pressures thereon so that the idler rotor tends to float friction free in its annular cavity. A duct conveys pressurized fluid from the high pressure end of the power rotor to its other end or to a shoulder to counterbalance the hydraulic working load and to produce a selfcompensating wear zero clearance fit within all end surfaces of working parts.

12 Claims, 16 Drawing Figures



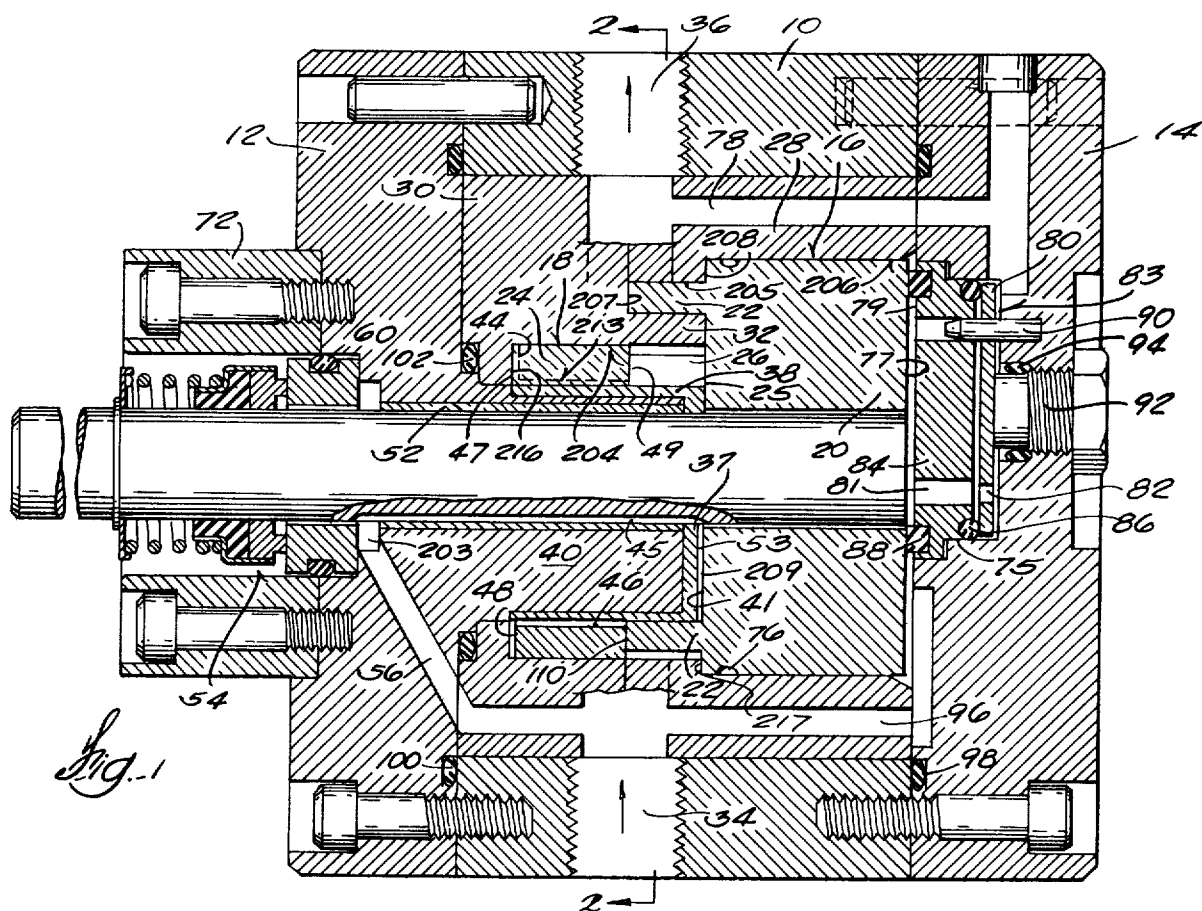


Fig. 1

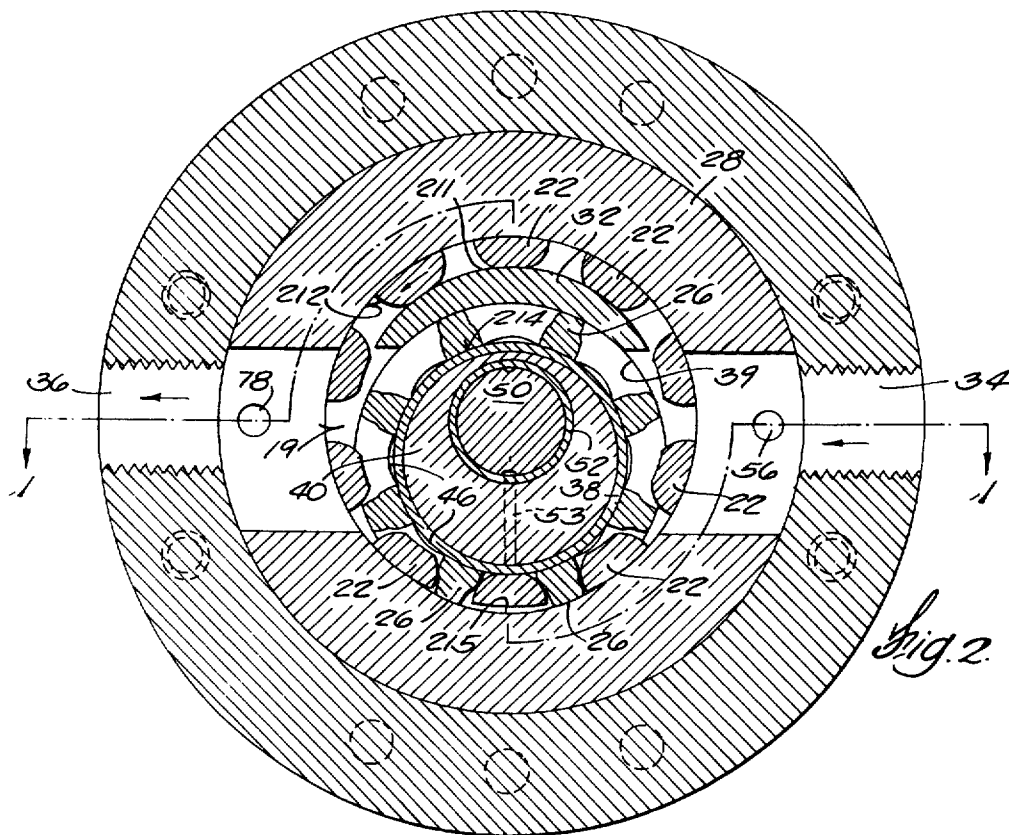
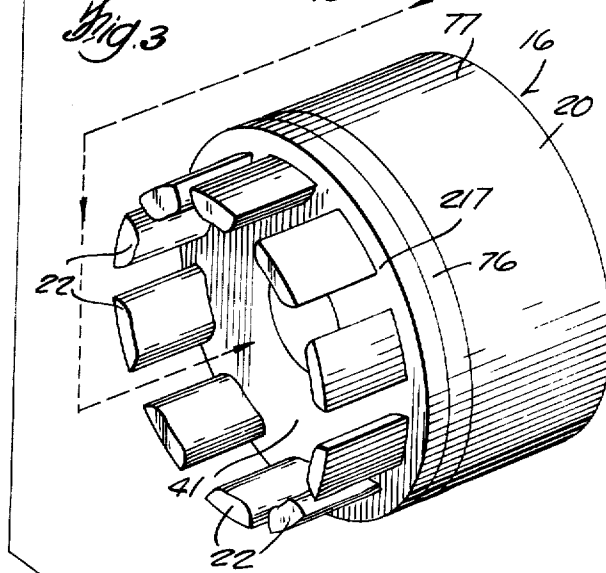
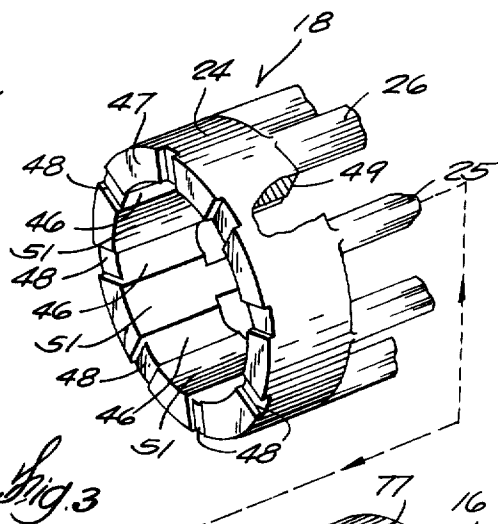
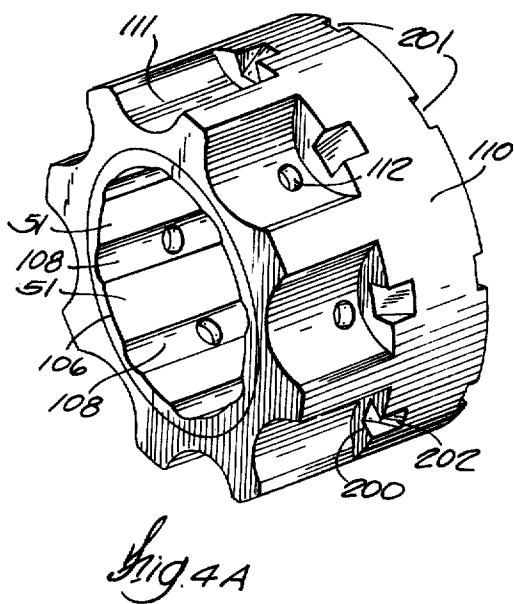
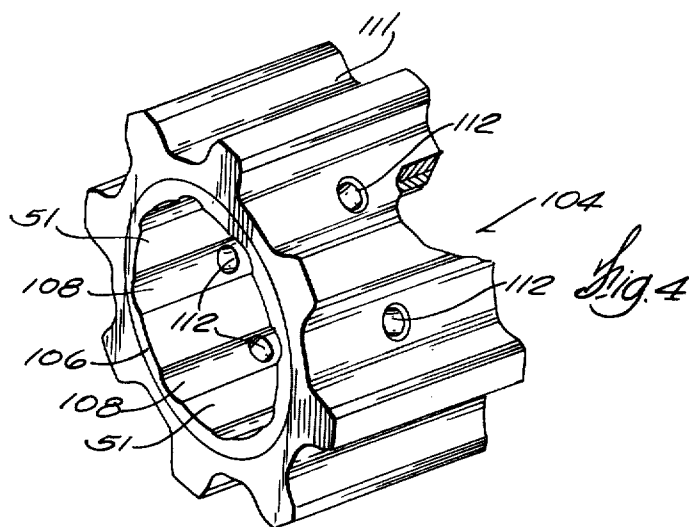


Fig. 2



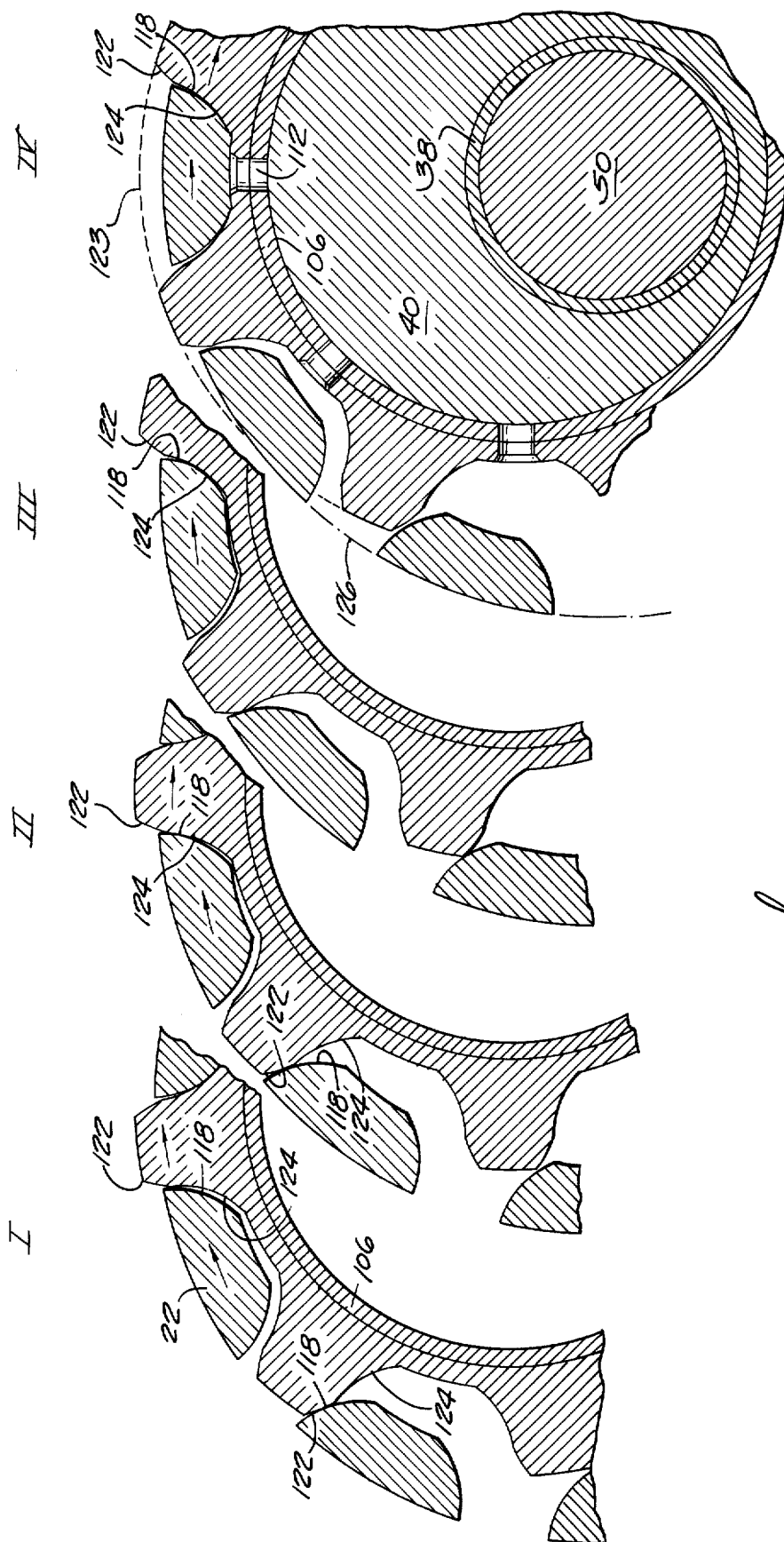
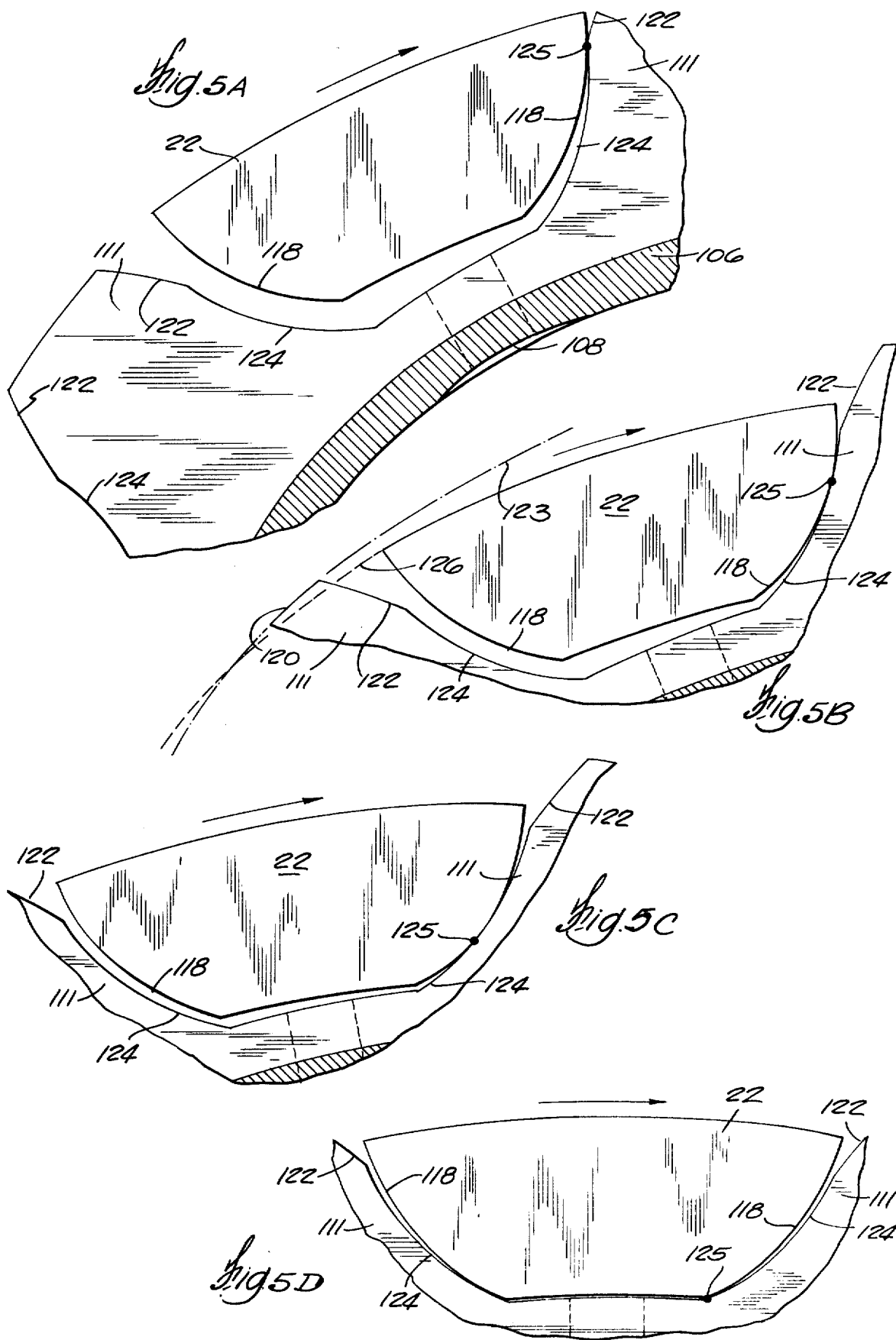
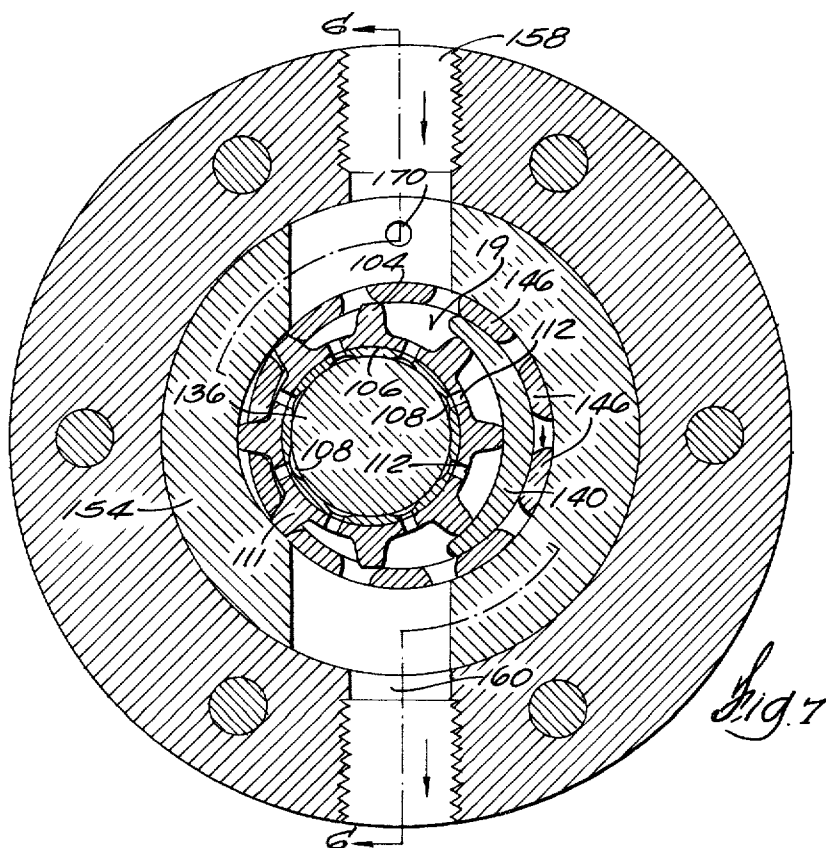
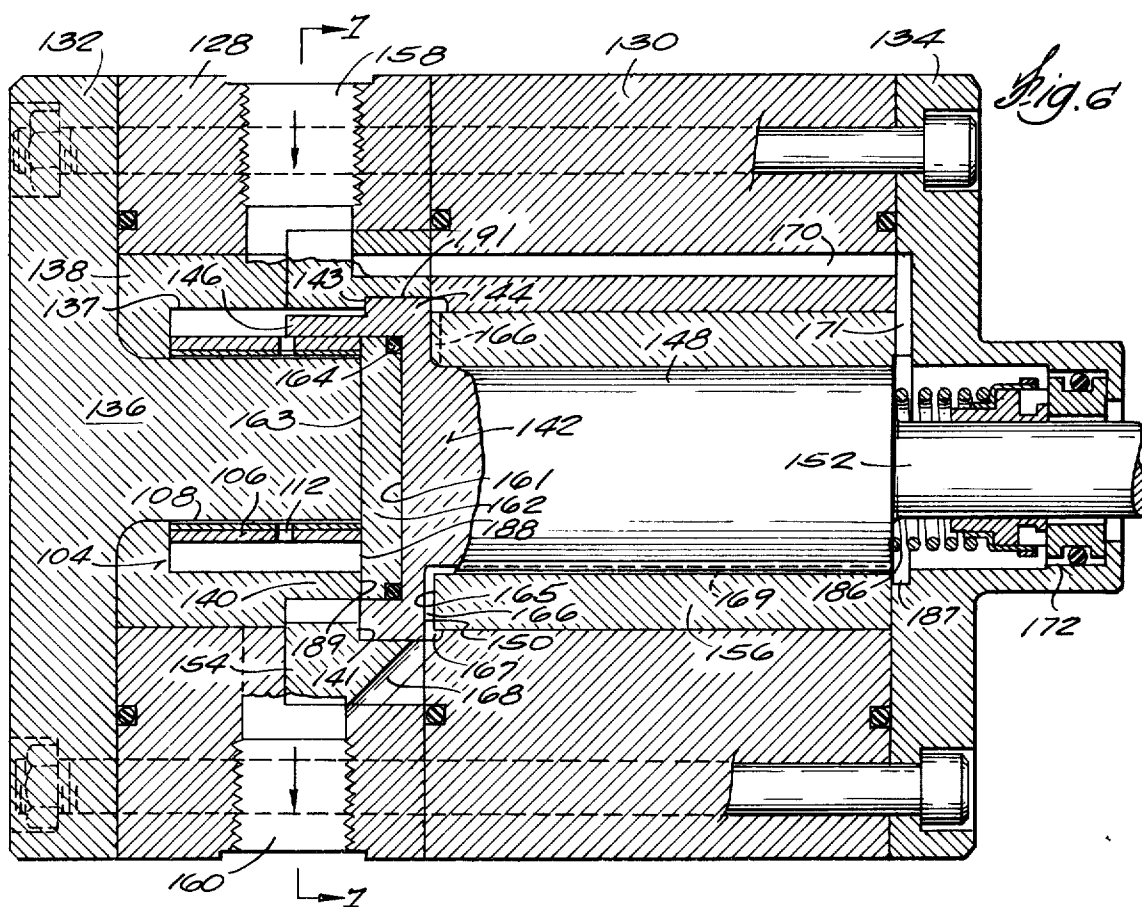
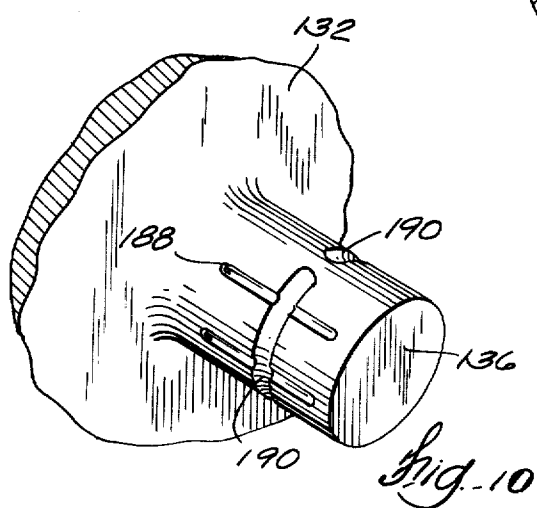
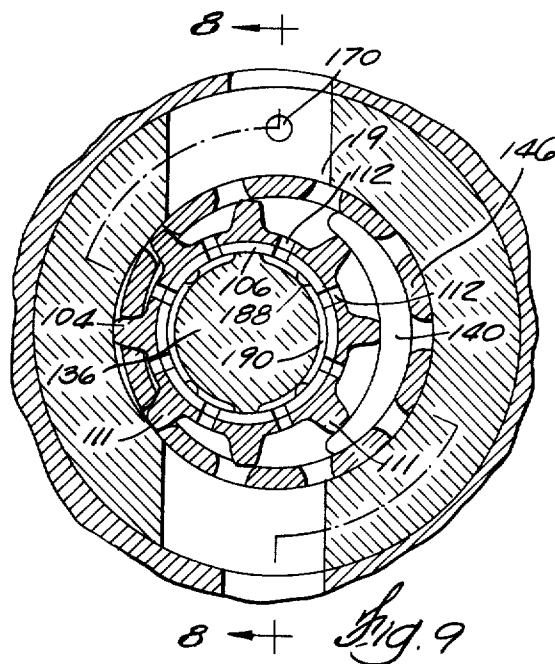
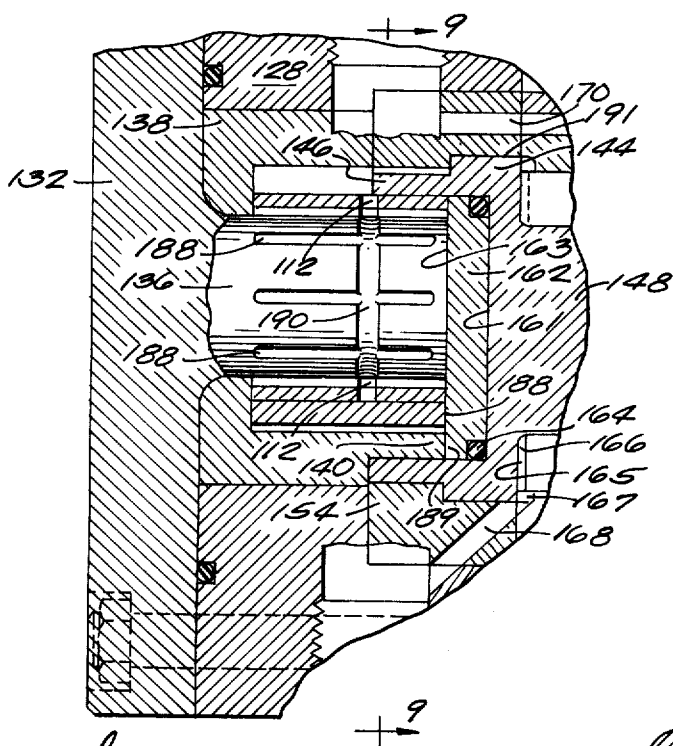


Fig. 5







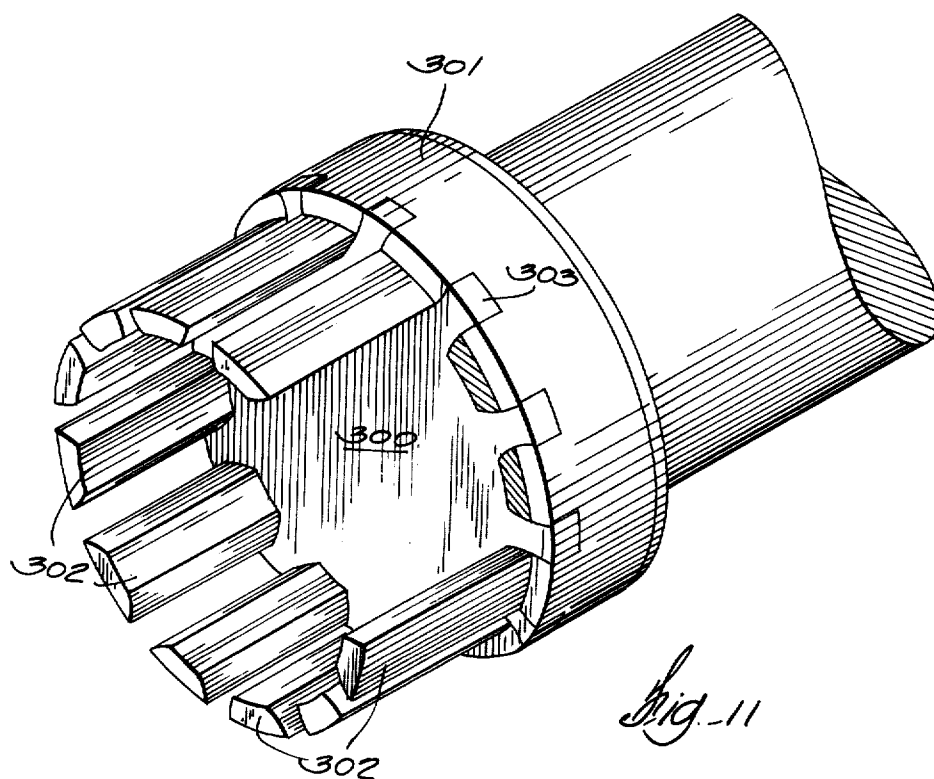


Fig. 11

HYDRAULIC PUMP/MOTOR WITH HYDROSTATICALLY BALANCED ROTORS

BACKGROUND OF THE INVENTION

This invention relates to improvements in internal circumferential piston pump/motors such as disclosed in my U.S. Pat. No. 3,364,868.

An internal circumferential piston pump/motor is a rotary device which is similar in structure to a gear pump but which differs from a gear pump in that the idler rotor teeth and power rotor teeth do not seal with each other but rather seal between the surface of a crescent baffle and the bushing within which the power rotor teeth rotate. The power rotor pumps fluid, but the idler rotor simply acts as a seal, and is without torque. Internal circumferential piston pump/motors have positive displacement, are pulseless, have low slip, will produce high pressure, will pump water and lighter liquids without lubrication, can also pump heavier chemicals, are usable as a pump or a motor, and are compact in structure.

Such internal circumferential piston pump/motor units have application to a wide variety of pumping or motor applications, especially for the high pressure pumping of non-lubricating liquids. Prior art designs of such units, however, are subjected to unbalancing hydraulic static loads, such as radial, axial and tilting loads on the power rotor and idler rotor which tended to press the rotors against their bearing surfaces and thus acted to speed wear on these surfaces, increases losses and reduce efficiency.

SUMMARY OF THE INVENTION

An important objective of the present invention is to provide means for hydraulically balancing the hydrostatic unbalanced pressure on the power rotor and idler rotor.

Hydraulic balancing of the idler rotor is achieved by providing hydraulic pressure chamber means between the idler rotor and its bearing surfaces. This chamber means is ducted to the zone of high pressure adjacent the interdigitated teeth of the power and idler rotors, thus to pressurize the chamber and counter-balance the unbalanced loads aforesaid. In some embodiments of the invention such hydraulic pressure chamber means comprise axially extending grooves formed between the hub of the idler rotor and its bearing to balance radial hydraulic pressures thereon and radial grooves formed between the end wall of the idler rotor and its adjacent bearing surface to balance axial hydraulic pressures thereon, so that the idler rotor tends to float friction free in its working chamber.

In like fashion the power rotor is hydraulically balanced. Pressurized fluid is ducted from the high pressure end of the power rotor to its other end or to an annular shoulder intermediate its ends to balance hydraulic forces thereon. In addition to the pressure of the hydraulic fluid, the power rotor is desirably subjected to the pressure of a mechanical preload element. Thus the power rotor is both hydraulically and mechanically biased against its sealing surfaces, even during low pressure operation.

In preferred embodiments of this invention, a reversible fluid seal which can be oriented either for low pressure or high pressure is applied to the power rotor shaft to prevent leakage of fluid therealong.

Other objects, features and advantages of the invention will appear from the following description.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial sectional view taken along the line 1—1 of FIG. 2 of one illustrative embodiment of the invention.

FIG. 2 is a radial sectional view taken on the line 2—2 of FIG. 1.

FIG. 3 is an exploded perspective view of the power rotor and crown toothed idler rotor shown in FIGS. 1 and 2.

FIG. 4 is a perspective view of a spur toothed idler rotor which can be used in place of the crown toothed idler rotor shown in FIGS. 1 through 3.

FIG. 4A is a perspective view of a modified idler rotor, incorporating features of both a crown gear-shaped and a spur gear-shaped idler rotor.

FIG. 5 is a composite view of a reduced scale showing sequential stages in the intermeshing of the crown gear teeth with the idler gear teeth.

FIGS. 5A through 5D are sequential fragmentary views partly in section and partly in elevation through intermeshing idler and power rotor teeth. These views show the convex-concave curves of different radii which provide rolling contact between the power rotor teeth and idler rotor teeth, and the relative portions of the respective tooth surfaces as the teeth move through meshing engagement.

FIG. 6 is an axial sectional view taken on the line 6—6 of FIG. 7 through another embodiment of the invention.

FIG. 7 is a radial sectional view taken on the line 7—7 of FIG. 6.

FIG. 8 is a fragmentary axial sectional view through a third embodiment of the invention taken on the line 8—8 of FIG. 9.

FIG. 9 is a fragmentary radial sectional view taken on the line 9—9 of FIG. 8.

FIG. 10 is a fragmentary perspective view showing the boss upon which the idler rotor of FIGS. 8 and 9 rotate.

FIG. 11 is a perspective view of a modified embodiment of power rotor with carbon thrust bearing covering substantially the entire end wall of the rotor, certain crown teeth of the rotor being omitted to expose said thrust bearing.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Although the disclosure hereof is detailed and exact to enable those skilled in the art to practice the invention, the physical embodiments herein disclosed merely exemplify the invention which may be embodied in other specific structure. The scope of the invention is defined in the claims appended hereto.

FIGS. 1, 2 and 3 show one illustrative embodiment of the invention. This particular embodiment of the invention is contained within a three-piece housing having a main housing sleeve 10, an idler housing end plate or cap 12 and a balancing disk end plate or cap 14. The three pieces of the housing are held together by conventional means, such as bolts. The main housing sleeve 10 has a generally cylindrical bore formed therein which contains carbon bushings 28, 30 within which a power rotor 16 and an idler rotor 18 rotate. The power rotor 16 is shaped like a crown gear and has

a large cylindrical hub portion 20 and a plurality of arcuately arranged crown teeth 22 projecting axially therefrom. In order to reduce the radial hydraulic load of the idler rotor and to make feasible application of the counter-balancing chambers, the idler rotor 18 is shaped into a thin wall spur gear toothed or crown gear shaped sleeve. In this particular embodiment, the idler rotor 18 is also shaped like a crown gear and has a cylindrical hub portion 24 and a plurality of arcuately arranged crown teeth 26 projecting axially therefrom and into interdigitated relation to the crown teeth of the power rotor. All rotating parts rotate within carbon or other suitable low friction material bearing surfaces. The power rotor 16 rotates within bores 206, 205 in a first carbon bushing 28 which is fixed within main housing 10, and partly rotates within radial face 211 of crescent baffle 32 which projects axially from bushing 30. The power rotor 16 is keyed to and rotates with its shaft 50 within carbon bushing 52 which is fixed in its bore in the boss 40 which extends axially from end plate 12. The idler rotor 18 rotates within a bore 204 in a second carbon bushing 30 which is also fixed within main housing 10, and it rotates on idler boss carbon sleeve 38. The carbon bushings provide low friction bearing for this pump which is designed especially for pumping non-lubricating liquids.

The idler rotor 18 is smaller in diameter than the power rotor 16 and rotates substantially within the circumference of the power rotor teeth 22, but on an axis offset or eccentric to the axis of rotation of the power rotor 16. Power rotor 16 rotates on the axis of shaft 50 and idler rotor 18 rotates on the axis of boss 40. As best shown in FIG. 2, the teeth 22 of power rotor 16 mesh with the teeth 26 of idler rotor 18 where their paths intersect and are radially separated where their paths diverge by a crescent-shaped baffle 32 which projects from carbon bushing 30. Although the teeth 22 and 26 mesh with each other, they are not required to make sealing contact with each other as is the case in gear pumps. Power rotor teeth 22 pump fluid on their teeth of rotation outside crescent baffle 32. Teeth 26 of idler rotor 18 seal across the return path (where the teeth mesh) and across the path of tooth rotation inside the crescent rotor. Power rotor 16 is keyed to and is driven by shaft 50 and the motion of rotor 16 is transmitted to idler rotor 18 through their intermeshing teeth.

The bores within which power rotor 16 and idler rotor 18 rotate together comprise a rotor chamber or working chamber 19 (see FIG. 2) which communicates with a hydraulic fluid inlet port 34 and an outlet port 36 in the main housing sleeve 10 and in the carbon bushings 28 and 30. Liquid enters the working chamber 19 through inlet port 34 and is forced through the working chamber 19 and out of the outlet port 36 by the teeth 22. As best shown in FIG. 2, the power rotor teeth 22 make sealing contact between the outer surface of crescent baffle 32 and the bore in bushing 28 and the adjacent end faces within which it rotates. In like manner idler rotor teeth 26 make sealing contact with the inner surface of crescent baffle 32. As the teeth 22, 26 rotate, they entrap fluid in the spaces bounded by the sides of the teeth, the crescent baffle 32, and its diametrically opposite surface 215. These entrapped pockets of fluid are transferred from the neighborhood of inlet port 34 to the neighborhood of outlet port 36 by the rotation of the power and idler rotor teeth 22, 26. Power rotor 16 exerts torque on the

liquid and idler rotor 18 acts as a seal without torque load. The delivery of these entrapped pockets of fluid to the outlet port 36 rises the pressure at the outlet port 36 and forces the fluid out. Leakage of fluid past the power rotor back to the inlet port 34 is prevented by the sealing contact of the idler rotor teeth 26 between the inner surface of crescent baffle 32 and internal surface of the carbon bushing 28 at the opposite side of the working chamber 19. These internal surfaces are shaped to sealingly engage the tips of idler rotor teeth 26. The sealing surfaces are specifically as follows:

Power rotor teeth ID. against crescent baffle OD. surface 211; power rotor teeth OD. against bore surface 212; hub face 41 and hub shoulder 217 against adjacent surfaces 209 and 208; crescent baffle end against adjacent surface 41. In the case of the idler rotor, sealings are between the next-described contact faces: idler rotor ID. 51 against sleeve surface 214, idler rotor OD. against bore surface 204 which extends to the baffle internal face 39, and to the diametrically opposite section surface 215; idler hub and idler gear end against adjacent surfaces 216 and 41. The tips of the idler rotor teeth 26 project slightly beyond the tips of power rotor teeth 22 at the side of working chamber 19 opposite crescent baffle 32.

The idler rotor 18 is rotated torque-free by mechanical engagement between the power rotor teeth 22 and the idler rotor teeth 26. The meshing surfaces of the teeth 22 and 26 are, however, specially shaped to provide a rolling contact thereinbetween as will be explained hereinafter, thus to reduce wear to a minimum.

The idler rotor 18 rotates on a cylindrical carbon bushing sleeve 38 which is supported by fixed cylindrical boss 40 which projects inwardly from the central portion of idler housing end plate 12 on an axis offset from the axis of shaft 50. Carbon sleeve 38 desirably has an end cap 37 which serves as an axial thrust bearing and which fits over the end of boss 40. The innermost annular ends 25 of idler rotor teeth 26, and bearing end cap 37, are adjacent to and seal on the power rotor internal end face 41. The opposite annular end 47 of idler rotor hub 24 rotates adjacent to and seals against a shoulder 44 formed on carbon bushing 30. Accordingly, the axial ends of crescent baffle 32, idler rotor teeth 26 and bushing end cap 37 are coplanar and seal against the end face 41 of the power rotor. Power rotor 16 is axially movable on its bearing support. The axial hydraulic counter-balancing force exerted on the power rotor, and the mechanical preload on the power rotor, as hereinafter explained, cause slight axial movement of power rotor 16 and urge these parts into sealing contact for zero clearance therebetween.

The unbalanced hydraulic load which creates the problem with which this invention deals, as they relate to the idler rotor, can be visualized from FIGS. 1 and 2. The pressure at outlet 36 greatly exceeds the pressure at inlet 34. Accordingly, there is a net unbalance of hydraulic force tending to force the hub 24 of idler rotor 18 to the right in FIG. 2, against its sleeve 38, bearing surface 214 and against its bearing surface 39 of the bore bearing surface 204 in which it rotates. There is a much smaller net unbalance of forces exerted against the crown teeth 26, as they present a much smaller effective pressure area to the unbalanced pressure. This net unbalance of forces radially presses the idler rotor 18 against bearing surfaces 38 and 39.

In the case of the crown gear-shaped idler rotor 18, and as can be visualized from FIG. 1, the pressure in the pumping zone between the teeth 22, 26 acts on the intertooth spaces at the annular end ring 49 of idler rotor 18, to force the hub 24 of idler rotor 18 to the left in FIG. 1 and against shoulder or race 44. Thus there is an axial force exerted by the pumped liquid in rotor chamber 19 on the idler rotor 18.

To the extent that the axial force aforesaid is greater near the outlet port 36 than it is near the inlet 34, because the pressure of the pumped liquid increases as it is forced through the rotor chamber 19, there is tilting force imposed on the idler rotor which tends to tilt it on its bushing 38 and on the internal surface 39 of the bore in which it rotates.

Accordingly, the heretofore net unbalanced forces to which the idler rotor 18 and its bearing are subject, impose radial, axial and tilting loads thereon.

The hydraulic balancing of the idler rotor can be achieved in various ways. In the embodiment of FIGS. 1, 2 and 3, radial, axial and tilting pressures on the idler rotor are compensated for or counter-balanced by means of radial and axial grooves in hub 24 and the hydraulic film spaces fed thereby and which together form hydraulic pressure chamber means between the rotor and its bearing surfaces. As best shown in FIG. 3, a plurality of axial grooves 46 are formed in the inner periphery of the idler rotor hub section 24 and a plurality of intercommunicating radial grooves 48 are formed in the outermost annular end 47 of hub section 24. End 47 is axially remote from teeth 26. The portions of end ring 47 between grooves 48 carries a hydraulic film pressurized by the fluid in grooves 48. Portions of the inner bore 51 of hub 24 between grooves 46 also carry a hydraulic film pressurized by the fluid in grooves 46. Each axial groove 46 communicates with a corresponding radial groove 48. The axial grooves 46 conduct hydraulic pressure from the intertooth spaces at inner end 49 of idler rotor 18 to the outer end 47, and the radial grooves 48, and film spaces therebetween act as hydraulic pressure chambers between idler rotor end ring 47 and bearing shoulder 44, thus to tend to counterbalance the axial pressure on the end ring 49 and on end faces 25 of teeth 26. Referring to FIG. 2, it is noted that there will be a pressure gradient from right to left in working chamber 19 as pressure increases from the inlet 34 to the outlet 36 of the pump. However, this pressure gradient will reproduce itself at both ends of the idler rotor, thus to counter-balance tilting loads as well as axial loads. The axial grooves 46 and film spaces therebetween act as hydraulic pressure chambers between the idler rotor hub 24 and its bearing 38, thus to tend to counter-balance the hydraulic radial pressure on bearing 38, 39, 204.

Accordingly, the axial, radial and tilting loads and pressure between the idler rotor hub and its bearing surfaces are substantially relieved and the idler rotor tends to float on its bearing.

Referring again to FIG. 1, the means for balancing the hydraulic load on power rotor 16 includes an annular balancing groove 76 in the power rotor hub 20 and associated hydraulic film space which frees the power rotor 16 from radial loads other than that of its driving torque.

Axial hydraulic pressure in rotor chamber 19 on the end faces 110 of power tooth teeth 22 and the annular end faces 41, 217 of the power rotor 16 is equalized or

balanced by counter-balancing means which includes a duct 78 through carbon bushing 28 and balancing disk end plate 14 for transmitting hydraulic pressure from the high pressure zone of working chamber 19 to outer annular end 77 of power rotor 16. Duct 78 communicates at one end with high pressure outlet port 36 and its other end with a mechanical preload disk 80 in chamber 83. The fluid passes through chamber 83, an opening 82 in preload disk 80 and duct 81 in balancing disk 84, thus to pressurize both sides of balancing disk 84. Disk 84 is sealed by O-ring 86 within a bore 75 in balancing disk end plate 14. The inner face of balancing disk 84 carries a carbon ring 88 which bears against the adjacent axial end 77 of power rotor 16. Both the balancing disk 84 and the preload disk 80 are keyed together by a pin 90 to balancing disk end plate 14, thus to prevent these parts from rotating.

In the operation of this embodiment, the high pressure fluid from outlet port 36 is conveyed through duct 78 to the outer end of balancing disk 84. The chamber 79 between disk 84 and end face 77 of power rotor 16 is also pressurized. The area of chamber 79 within carbon ring 88 is slightly less than the chamber 83. Accordingly, most of the load will be relieved from carbon ring 88, which will be subject only to the difference of pressure of the fluid in chambers 79 and 83.

The high pressure fluid in chamber 83 presses against the outer end of balancing disk 84 which functions as a piston and thereby causes the carbon ring 88 and the pressurized fluid in chamber 77 to press against the adjacent axial end 77 of power rotor 16, thus to counter-balance working chamber axial pressures thereon.

Balancing disk 84 and associated parts are offset from the axis of power rotor 16 toward high pressure outlet 36 to compensate for the tilting force developed due to the fact that the end face of power rotor 16 near outlet 36 is exposed to high pressure fluids while its other side near inlet 34 is exposed to low pressure fluid. Thus the center of balancing disk 84 is offset toward the high pressure side of power rotor 16 far enough to balance out the above-noted tilting force. The outer surface area of balancing disk 84 upon which the fluid pressure acts is dimensioned in accordance with well-known hydrostatic principles to provide substantial equilization of the axial force developed by the pumped fluid on the axial end of power rotor 14 within working chamber 19.

The mechanical preload disk 80 bears against the outer face of balancing disk 84 through O-ring 86 and is pressed thereagainst by a preload screw 92 which can be adjusted to vary the amount of preload pressure. O-ring 86 is resilient and stores energy to add a spring effect to the preload functions. O-ring 86 spring effect could be replaced by metallic spring to add resilience and stored energy to the preload. The pressurized fluid under the end of adjustable preload screw 92 is prevented from leaking out around the screw by an O-ring 94. In practice, this preload system provides an independent mechanically controlled axial balancing safety force for starting or low pressure pumping operation of the pump when there might not be enough hydraulic pressure to chamber 83 to so bias the power rotor. In the disclosed embodiment, the preload screw 92 is adjusted to apply sufficient pressure to the end 77 of power rotor hub 20 to bias the power rotor to the left in FIG. 1 and seal the ends of the power rotor to their bearing surfaces during low pressure pumping condi-

tions. Because the counter-balancing axial hydraulic force is greater than the opposed working hydraulic force, the power rotor axial counter-balancing force not only counter-balance the unbalanced axial working load on the power rotor, but by the constantly applied differential hydraulic force zero clearances are maintained between all the working rotary and stationary end sealing surfaces of the pump.

The fluid which leaks from the high pressure working chamber through hub 16 and around carbon ring 88 is ducted back to inlet port 34 through a duct 96 which is formed in carbon bushing 28 and in balancing disk end plate 14. The joint between balancing disk end plate 14 and main housing 10 is sealed by an O-ring 98. A similar O-ring 100 seals the joint between main housing 10 and idler housing end plate 12. A smaller O-ring 102 seals the joint between idler housing end plate 12 and carbon bushing 30.

The power rotor 16 is attached to and is driven by a shaft 50 which extends through a bore in boss 40, the latter being lined by a relatively long carbon bearing sleeve 52 which provides a low friction supporting surface for the shaft 50. The relatively long carbon bearing sleeve 52 is an important feature of the invention because it significantly reduces wear on the shaft 50.

There is a high pressure lubrication for the shaft 50 and its bearing 52 which is channeled from the high pressure groove 53 through shaft or bushing groove 45 to the lower pressure sealing chamber 203, which is communicating through duct 56 with the inlet port 34.

A simple commercial shaft seal assembly 54 is attached around the shaft 50 on the end of idler housing end plate 12 to prevent fluid leaking to the outside.

Although the idler rotor 18 of the embodiment of FIGS. 1-3, inclusive, has a crown gear configuration, it could have a spur gear configuration, if desired. FIG. 4 shows such a spur gear-shaped idler rotor 104 which can be used in place of the crown gear-shaped idler rotor 18 in FIG. 3. The hub of the idler rotor 104 is lined with a carbon bushing sleeve 106 which has hydraulic pressure chamber means comprising a series of axially extending grooves 108 and the hydraulic film spaces therebetween which serve the same purpose as the corresponding hydraulic pressure chamber means in the crown gear-shaped idler rotor 18. Such pressure chamber means can be formed directly on the hub of the idler rotor 104, where the bushing is part of the boss on which the rotor turns.

The spur gear-shaped idler rotor does not need radially extending grooves in its end ring, like grooves 48 of the crown gear-shaped idler rotor 18 of FIG. 3. This is because the spur gear-shaped idler rotor 104 extends completely across the working chamber 19 and does not have a radial surface (such as surface 49 of the crown gear-shaped idler rotor 18) subject to axial hydraulic pressure. Hence there is no need in the spur gear-shaped idler rotor to counterbalance such non-existent axial force.

FIG. 4A, however, shows a modified idler rotor which has both the spur gear type idler rotor advantages and the crown gear type idler rotor advantages. This type of idler rotor incorporates the spur gear fixed in carbon bearing advantages with the full length applied balancing grooves and the advantage of crown hub section, additional internal and external bearing areas and its toothed strength reinforcement hub section characteristic. Basically, motor has a spur gear

shape, but the valleys between the teeth 111 are filled in at 110 beyond the ends of the crown teeth on the power motor which intermesh therewith. Accordingly, in FIG. 4A there are surfaces 200 and 202 which are exposed to axial hydraulic loads. These are counter-balanced in the same manner as the crown gear idler rotor by providing radially extending grooves 201 on the outer end face of the rotor. These communicate with axial grooves 108 through ducts 112 and function as do grooves 48 in the embodiment of FIG. 3.

The pressure chamber means 108 can also be formed on the outer surface of the boss, as in FIG. 10. In the embodiment of FIGS. 4 and 4A, the bearing sleeve 106 is regarded as part of the rotor hub and the grooves 108 are formed in the hub. A series of radial holes 112 disposed in the valleys between spur teeth 111, one for each of the grooves 108, duct high pressure hydraulic balancing fluid to the axial grooves 108 and through this axial groove 108 to the radial balancing groove 201 and the film spaces therebetween. Such holes are not required in the crown gear configuration inasmuch as the high pressure fluid can enter the axial grooves 46 in the crown gear through the open spaces or gaps between the crown gear teeth 26.

Moreover, the power and idler rotors can have tooth thickness for a maximum tooth strength and extent of tooth tip sealing surfaces. The teeth have a unique rolling drive characteristic. This rolling tooth driving system consists of two kinds of driving contact mechanism, namely, primary and secondary driving contact surfaces or curves.

The primary driving contact mechanism is established by the convex surface of the power rotor teeth and by the lower concave surface of the idler rotor teeth. The secondary or backup driving mechanism is established by the convex surface of the power rotor teeth and by the upper convex surface of the idler rotor teeth.

The rotor sizes can have any magnitude, but the tooth dimensions, shapes, curves magnitude and their angular positions, which represent a part of novelties of the driving system, have close relationship to each other and they have to be proportional.

The theory of the tooth primary and secondary driving relationship is best shown in FIG. 5 which illustrates the teeth successive driving positions and their relationship to each other.

The rolling and the backup tooth contact is illustrated in FIG. 5 and in FIGS. 5A through 5D which illustrate successive positions of meshing tooth surfaces. Each power rotor tooth 22 has driving convex contact surface 118. Each idler rotor tooth 111 has a short convex side portions 122 which provide secondary drive contact surfaces and long concave drive contact portions 124 which provide primary drive contact surfaces with the power rotor tooth convex curve 118. The concave surfaces 124 of idler teeth 111 are of slightly larger radius than the corresponding convex surfaces 118 of power rotor teeth so as to provide rolling contact therebetween as the teeth move into and out of mesh, as illustrated in FIGS. 5, 5A through 5B.

Although a spur gear configuration is shown for the idler rotor in FIGS. 5A through 5B, the same dimensions can be used for crown gear teeth.

FIG. 5 illustrates both the primary rolling contact drive between surfaces 118 of power rotor tooth 22 and the surface 124 of the idler rotor tooth 111 and the sec-

ondary or backup partly rolling, partly sliding contact drive between surfaces 118 of power rotor tooth 22 and the surface 122 of the idler rotor tooth 22. The secondary drive aforesaid partly relieves overload from the primary drive by distributing contact toward more tooth surfaces and relieves and redistributes the shock of tooth engagement and reduces nose, thereby smoothing operation.

The primary drive mechanism is best illustrated in FIGS. 5A through 5D. Initial line contact 125 between convex surface 118 and concave surface 124 is made near the top ends of said surfaces. As the teeth 22 and 111 turn clockwise in the drawing, the line contact 125 moves down the surfaces 118 and 124 as these surfaces roll together, until the line contact 125 reaches the bottom of these surfaces as shown in FIG. 5D. Thus the matching convex-concave surfaces 118, 124 with different radii of curvature provide a single rolling contact line as the teeth mesh and demesh, thereby eliminating sliding surface contact in the primary drive contact between the teeth and the wear that such sliding surface contact produces.

The foregoing rolling action and the relation between surfaces 118 and 124 and secondary or backup drive between surfaces 118 and 122 is of particular significance in rotor geometries because the tooth tip path 123 of the idler rotor overlaps the tooth tip path 126 of the power rotor. The teeth have to be wide enough for strength and tip sealing purposes. As illustrated in FIG. 5, this geometry requires the teeth 22 of power rotor to cross through the tooth tip path 123 of the idler rotor as the teeth mesh and demesh.

Although devices embodying the invention have been described as pumps, they can also be used as hydraulic motors by introducing liquid under pressure into the outlet port 36 and exhausting the liquid out inlet port 34. Rotors 16, 18 would then turn clockwise in FIG. 2, instead of counterclockwise, and shaft 50 would deliver power. Thus it will be clear that the terms "inlet port" and "outlet port" are used herein in a relative sense and may be interchanged if the function of the device is changed.

FIGS. 6 and 7 show another embodiment of the invention which has the same pumping principle, pumping section and internal arrangement and shape of rotors as described in the previous embodiment, but is designed by a different construction approach. Important features of this embodiment can be summarized as follows:

1. Axial counter-balancing by power rotor shoulder without balancing disk.
2. Shoulder chambers of power rotor for high pressure force lubrication and for axial and radial counter-balancing purposes.
3. Integrally jointed power rotor hub, shaft and bearing arrangement.
4. Integrally fixed carbon thrust bearing of power rotor.
5. Extra bearing and sealing area power rotor shoulder.
6. Reinforced seal spring for mechanical axial preload of power rotor.

Referring to FIGS. 6 and 7, this embodiment of the invention is housed in a four-part housing for practicality, comprising a rotor housing 128, a main bearing housing 130, an idler end cover 132, and a shaft seal end cover 134. The four pieces of the housing are held together by conventional means such as bolts or the

like. The rotors 104, 142 are supported in relative thin walled carbon bushings for economical and for improved thermal expansion reasons. The power rotor 142 rotates in the carbon bushing 154 and with its hub shaft 148 in carbon bushing 156 which are fixed into rotor housing 128 and into main bearing housing 130 respectively. The idler rotor 104 rotates in the carbon bushing 138, which extends through its baffle section to the power rotor teeth for sealing reasons, and it is fixed into idler end cover 132. The idler end cover 132 has a central boss 136 which, for exemplification, rotatably supports a spur gear type idler rotor 104 such as illustrated in FIG. 4. As described above, idler rotor 104 is lined with a carbon bushing sleeve 106 which has axially extending balancing grooves or chambers 108 and radial duct openings 112. These grooves and openings function as hereinbefore described to hydraulically counterbalance radial pressure between the rotor and its bearing surfaces.

Idler rotor 104 rotates within a bore 137 in a carbon bushing 138, which fits within main bearing housing 128 around boss 136. A crescent-shaped baffle 140 projects from a peripheral portion of carbon bushing 138 and separates the idler rotor teeth 111 from the power rotor teeth 146 as best shown in FIG. 7. In this embodiment, the power rotor 142 has a crown gear configuration as in the previous embodiment, but with a relatively short cylindrical hub 144 with an internal hub shoulder 143, a plurality of arcuately arranged teeth 146 projecting from the internal axial end of hub 144 and within shoulder 143 formed therein, and a relatively long cylindrical hub extension 148 which functions as a drive shaft and which projects from the other axial end of hub 144. The power rotor has an external shoulder 150 which is placed within the hub 144 and the smaller diameter of the extended hub 148. The external shoulder 150 is used as a piston surface to balance the axial hydraulic pressure on the power rotor 142, as will be explained hereinafter.

A shaft 152 is integrally joined to the outer end of hub extension 148 and projects therefrom. Making the power rotor 142 and shaft 152 from a single piece of material the over-all compact power rotor design with its extended length power rotor hub construction makes possible a rigid alignment, low PV. factors, close over-all sealing conditions and high pumping performances.

The power motor 142 rotates within bores in two generally cylindrical carbon bushings: carbon bushing 154, which covers the tooth and hub areas of the power rotor, and carbon bushing 156, which covers the hub extension 148 of the power rotor. The carbon bushings 154 and 156 are fixed within bores in their associated housing members.

As shown in FIG. 7, the teeth 146 of power rotor 142 mesh in one portion of their circumference with the teeth 111 of idler rotor 140 and are separated therefrom in the diametrically opposed portion of their circumference by the crescent baffle 140. The power rotor teeth 146 and idler rotor teeth 111 are shaped with matching concave-convex meshing surfaces of different radii, as illustrated in FIGS. 5A through 5D, to provide rolling contact thereinbetween as described in connection with the first embodiment. The idler rotor teeth 111 making sealing contact between the inner surface of crescent baffle 140 and the bore within carbon bushing 154. The bores within carbon bushings

154 and the adjacent carbon bushings 138 comprise a rotor chamber or working chamber 19 (see FIG. 7). An inlet port 158 and an outlet port 160 communicate with the working chamber 19 for conveying fluid into and out of the chamber. The ports 158 and 160 are formed in the main bearing housing 128 and in carbon bushings 138 and 154. Fluid is pumped in through inlet port 158 and out through outlet port 160 by the rotation of the power rotor 142 as described previously in connection with the first embodiment.

Hub 144 is provided with a pocket 161 to receive an axial thrust bearing disk 162 which is fixed and sealed in the pocket 161 by an O-ring seal 164. Face 163 of thrust bearing disk 162 is coplanar with hub shoulder 143 and with the idler rotor end face 188 and baffle end face 189 and the end face of boss 136 and the portions of the hub face of the power rotor between the teeth. Shoulder 143 bears axially on complementary shoulder 141 formed on bearing 154 and further face 163 of bearing disk 162 bears axially on the end of boss 136. Accordingly, except for the tips of teeth 146, the axial thrust of the axially movable power rotor 142 is taken on such coplanar surfaces.

As in the previously described embodiments, rotor chamber 19 contains hydraulic fluid under pressure. This pressure tends to force power rotor 142 axially to the right in FIG. 6. In order to balance the axial and radial hydraulic pressure on the power rotor 142, a pressure chamber consisting of radial grooves 166 which communicate with circumferential chamber 167 and the hydraulic film spaces therebetween is formed between the power rotor shoulder on annular piston face 150 and the annular ring face 165 of fixed bushing 156. A fixed position duct 168 is formed in carbon bushing 154 between high pressure outlet port 160 and pressure chamber 167, 166, thus to apply pressurized fluid to the piston face 150. The area of annular shoulder or piston face 150 is selected to provide an axial force that is substantially equal and opposite to the axial force exerted on power rotor teeth 146 and power rotor hub 144 by the hydraulic pressure of the fluid within the working chamber 19. It should be noted that this arrangement for balancing axial pressure is significantly simpler than that used in connection with the first embodiment.

Accordingly, the annular area of chamber 166 provides hydraulic counter-balancing force exerted at the axial balancing area of the shoulder 150 of the power rotor.

Moreover, circumferential chamber 167 with chamber 166 provide high pressure fluid to the hub bearing surface 191 to counteract radial unbalancing forces. In addition, chambers 166, 167 provides cooling fluid and high pressure force lubrication for the hub shaft bearing through bearing clearance or through grooves 169. Leakage is discharged to the low pressure areas 187 adjacent shaft seal 172 and back to the inlet port 158 through ducts 171, 170.

The shaft 152 is sealed by a simple mechanical shaft face seal 172 which is the same as shaft seal 54 described previously, but which is placed in the opposite direction so as to seal off leakage for either low or high pressure flooded intake conditions.

The reinforced seal spring 186 is designed to act as a mechanical preload on the power rotor 142 and bias it to the left in FIG. 6, even during low pressure operating conditions, in a manner similar to the operation of

the preload disk 80 of FIG. 1. Accordingly, zero clearance is maintained at all end surfaces on the power rotor and its opposed bearing surfaces, including exposed parts of the idler rotor, as hereinbefore explained.

FIGS. 8, 9 and 10 show a modification of the invention in which the hydraulic pressure chamber means between idler rotor 104 and boss 136 are formed in the boss 136 instead of being formed on the rotor 104. Referring to FIG. 9, in this modification the carbon sleeve 106 which lines the bore of the spur gear-shaped idler rotor 104 and forms a part thereof does not have axial grooves 108 such as shown in FIG. 7. Instead, the inner periphery of carbon sleeve 106 is smooth and the pressure chamber means for idler rotor 104 is formed by a plurality of axial grooves 188, circumferential grooves 190 formed in the boss 136 upon which the carbon sleeve 106 of idler rotor 104 turns, and the film spaces therebetween. Pressurized fluid is ducted into the axial grooves 188 through circumferentially extending grooves 190 which are formed in boss 136 under the radial openings 112 in the idler rotor 104. As the idler rotor 104 rotates, the radial openings 112 also rotate, but they remain above the circumferential grooves 190 and thus duct pressurized fluid from working chamber 19 into circumferential grooves 190, which communicate directly into the axial grooves 188. The pressurized fluid which is ducted into grooves 188, 190 and the hydraulic film spaces therebetween presses against the inner periphery of the carbon sleeve 106 of idler rotor 104 and thus acts to relieve the radial hydraulic load on the rotor 104.

FIG. 11 shows a modified design of the carbon thrust bearing of the power rotor. In this embodiment the carbon thrust bearing 300 is shaped with radially projecting fingers 303 which fit between the crown teeth 302, thus to cover substantially the entire end face of the power rotor hub 301 including the spaces between the projecting crown teeth 302. In this embodiment the idler rotor teeth end faces can have continuous full bearing contact with the carbon bearing 300, regardless of the movement of the idler rotor teeth into and out of mesh with the power rotor teeth.

Although this invention has been described in connection with several illustrative embodiments thereof, it should be understood that the invention is not limited to the disclosed embodiments since changes can be made in the disclosed structure without departing from the basic principles of this invention. For example, although the improved power rotor disclosed in the second embodiment is integral with its shaft, the shaft could be separate if desired and could extend in the opposite direction if desired, as in FIG. 1. Different dimensions and shapes could be employed for the power rotor shoulder depending on the requirements of particular applications. These and other modifications of the disclosed structure will be apparent to those skilled in the art, and this invention includes all modifications that fall within the scope of the following claims.

I claim:

1. In a pump/motor unit having a housing with a rotor chamber, a liquid inlet at one side of the chamber and a liquid outlet at the other side of the chamber, said inlet and outlet providing for passage of liquid through said chamber, power and idler rotors in said chambers, said rotors having interdigitable teeth, bearings upon which said rotors turn, the improvement for hydraulically

cally balancing hydrostatic pressure on the idler rotor and comprising pressure chamber means between the idler rotor and its bearing, said pressure chamber means comprising narrow grooves, the bearing surfaces between said grooves being wider than the grooves and defining hydraulic film spaces fed by said grooves and duct means from the rotor chamber to said grooves to conduct pressurized liquid to said pressure chamber means and counter-balance the hydrostatic pressure of said liquid on the idler rotor, said idler rotor having a cylindrical hub, said pressure chamber means comprising a plurality of such narrow grooves disposed radially in one axial end of said hub and running out at the inner surface of the hub and a like plurality of such narrow grooves extending axially along the inside of said hub and running out at the end of the hub and intercommunicating with said radial grooves.

2. The improvement defined in claim 1 wherein said idler rotor is crown gear shaped with a hub and axially extending crown teeth, said pressure chamber means being formed between said hub and its bearing.

3. The improvement defined in claim 1 wherein said idler rotor is spur gear shaped with a hub and radially extending spur teeth, said pressure chamber means being formed between the hub and its bearing, said duct means comprising radial openings through the hub and between the teeth.

4. The improvement as defined in claim 1 in which said idler rotor has an over-all thin wall sleeve body configuration.

5. The improvement defined in claim 1 in which said idler rotor has a hub with axially extending teeth and a sleeve beneath said teeth from which said teeth extend radially.

6. In a pump/motor unit having a housing with a rotor chamber, a liquid inlet at one side of the chamber and a liquid outlet at the other side of the chamber, said inlet and outlet providing for passage of liquid through said chamber, power and idler rotors in said chamber, said rotors having interdigitable teeth, bearings upon which said rotors turn, the improvement for hydraulically balancing hydrostatic pressure on the idler rotor and comprising pressure chamber means between the idler rotor and its bearing, said pressure chamber means comprising narrow grooves, the bearing surfaces between said grooves being wider than the grooves and defining hydraulic film spaces fed by said grooves and duct means from the rotor chamber to said grooves to conduct pressurized liquid to said pressure chamber means and counter-balance the hydrostatic pressure of said liquid on the idler rotor, the narrow grooves of said pressure chamber means comprising a plurality of narrow axial grooves in said bearing and a narrow circumferential groove in said bearing interconnecting said axial grooves.

7. In a hydraulic pump/motor unit having a housing with a rotor chamber, a liquid inlet port at one side of the chamber and a liquid outlet port at the other side of the chamber, said inlet and outlet ports providing for passage of liquid through said chamber, power and idler rotors in said chamber, said rotors having interdigitable teeth, bearings on which said rotors turn, and means for applying hydraulic pressure to one axial end of said power rotor to counter-balance axial pressures thereon caused by the pressure of the liquid on the power rotor, the improvement comprising preload means for applying a mechanical preload pressure to

said one axial end of said power rotor to provide an axial pressure thereon in the absence of said hydraulic pressure, said preload means for applying said hydraulic pressure to one axial end of said power rotor comprising a balancing disk seated within said housing and positioned adjacent to said one axial end of said power rotor, a low friction bearing surface on said balancing disk for bearing against said one axial end of said power rotor to convey axial pressure thereto, and duct means for applying hydraulic pressure from the rotor chamber to the side of said balancing disk away from said power rotor, and wherein said means for applying a mechanical preload pressure comprises a preload disk seated within said housing and positioned to bear against the balancing disk, an O-ring between said disks and in radial contact with the seat in the housing for the balancing disk, thus to seal said hydraulic pressure, said O-ring being of resilient material and a preload screw threaded within said housing and positioned to bear against said preload disk to press the O-ring against said balancing disk, thereby providing adjustable mechanical pressure on said balancing disk in the absence of said hydraulic pressure.

8. In a hydraulic pump/motor unit having a housing with a rotor chamber, a liquid inlet at one side of the chamber and a liquid outlet at the other side of the chamber, said inlet and outlet providing for passage of liquid through said chamber, power and idler rotors in said chamber, said rotors having a plurality of teeth which mesh simultaneously, the valleys between teeth on one rotor being only slightly larger circumferentially than the teeth on the other rotor, the improvement for reducing friction between the rotor teeth wherein the meshing portion of said teeth are shaped as matching concave-convex surfaces, the radius of said concave surface being only slightly larger than the radius of said convex surface, said concave-convex surfaces providing rolling contact in the meshing portion of said teeth, thereby reducing the friction between the teeth.

9. The improvement defined in claim 8 wherein the side meshing portions of the power rotor teeth are convex and the side meshing portions of the idler rotor teeth are concave.

10. In a pump/motor unit having a housing with a rotor chamber, a liquid inlet at one side of the chamber and a liquid outlet at the other side of the chamber, said inlet and outlet providing for passage of liquid through said chamber, power and idler rotors in said chamber, said rotors having interdigitable teeth, bearings upon which said rotors turn, the improvement for hydraulically balancing hydrostatic pressure on the idler rotor and comprising pressure chamber means between the idler rotor and its bearing, said pressure chamber means comprising narrow grooves, the bearing surfaces between said grooves being wider than the grooves and defining hydraulic film spaces fed by said grooves and duct means from the rotor chamber to said grooves to conduct pressurized liquid to said pressure chamber means and counter-balance the hydrostatic pressure of said liquid on the idler rotor, and the improvement for reducing friction between the power and idler rotor teeth wherein a plurality of such teeth mesh simultaneously, the valleys between teeth of one rotor being only slightly longer circumferentially than the teeth on the other rotor and the meshing portion of said teeth are shaped as matching concave-convex surfaces, the radius of said concave surface being only slightly

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larger than the radius of said convex surface to provide rolling contact in the meshing portion of said teeth, thereby reducing the friction between teeth.

11. The improvement defined in claim 10 wherein the side meshing portion of the power rotor teeth are convex and the side meshing portions of the idler rotor teeth are concave.

12. A pump/motor unit having an idler rotor and a power rotor with teeth which mesh and demesh as the rotors turn, a plurality of the teeth of the power rotor simultaneously imparting driving thrust to like plurality of the teeth of the idler rotor, said power rotor teeth having side faces which bear on the idler rotor teeth for transmitting force therebetween, said idler rotor teeth having a compound shape including a curved surface near the roots of the idler rotor teeth and which engage the side surface of the power rotor teeth when the teeth are substantially in mesh, thus to provide primary driving engagement between the rotors, said compound

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shape further comprising a curved surface near the tips of the idler rotor teeth and which engage said side faces of the power rotor teeth before the teeth are substantially in mesh, thus to provide secondary driving engagement therebetween, the path of rotation of the tips of the power rotor teeth intersecting and crossing the path of rotation of the tips of the idler rotor teeth as the power rotor teeth move from the secondary driving surfaces of the idler rotor teeth to the primary driving surfaces thereof in the course of moving into mesh, the side faces of the power rotor teeth being convex and the curved surfaces near the roots of the idler rotor teeth being concave, the curvature of said curved surfaces of the idler rotor teeth being only slightly greater than the convex surfaces of the power rotor teeth, one of said tooth surfaces rolling on the other tooth surface as the teeth move into mesh.

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