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(54) **HYDRAULIC MOTOR PLATES**

HYDRAULIKMOTORSCHEIBEN

PLAQUES DE MOTEUR HYDRAULIQUE

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Description

Background of the Invention

[0001] Hydraulic pressure devices are efficient at producing high torque from relatively compact devices. Their ability to provide low speed and high torque make them adaptable for numerous applications. White U.S. Patents 4,285,643, 4,357,133, 4,697,997 and 5,173,043 are examples of hydraulic motors.

[0002] The present invention relates to a gerotor hydraulic pressure device according to the preamble of claim 1.

Description of Prior Art

[0003] Rotating valve hydraulic motors are well known in the art. Examples include the McDermott U.S. Patent 3,572,983, the Vengers U.S. Patent 3,749,195, the Thorson U.S. Patent 4,343,600 and the Uppa, etal U.S. Patent 4,762,479. The motors themselves, while serviceable, have complicated housing parts, necessitating numerous machining, drilling and other secondary operations in order to manufacture the unit. Each of these additional manufacturing steps adds to the complexity of the hydraulic motor, increasing the manufacturing, maintenance and other costs attendant to the motors. In addition, complicated seals are needed between adjoining housing parts in order to insure the hydraulic integrity of the motor.

[0004] U.S. Patent No. 3,826,596 discloses a gerotor hydraulic pressure device according to the preamble of claim 1, having a rotor and a stator surrounding the rotor. The device also includes a housing having two clamping parts with the stator disposed between the two clamping parts. Two side plates are provided such that when the two clamping housing parts are drawn together end faces of the housing parts act on the plates and the stator and clamp the components firmly together. Complicated seals are needed between the joining housing parts in order to ensure the hydraulic integrity of the motor.

[0005] The present invention is designed to simplify the construction of hydraulic motors and more particularly hydraulic motors having a rotating valve.

Objects and Summary of the Invention

[0006] It is an object of the present invention to simplify the construction of hydraulic devices;

[0007] It is another object of the present invention to increase the efficiency of hydraulic motors;

[0008] It is a further object of the present invention to strengthen hydraulic motors;

[0009] It is still another object of the present invention to reduce the cost of hydraulic motors;

[0010] It is yet another object of the present invention to increase the adaptability of hydraulic motors;

[0011] The objects are attained by the features of the characterizing part of claim 1.

[0012] Other objects and a more complete understanding of the invention may be had by referring to the drawings in which:

Brief Description of the Drawings

[0013]

FIGURE 1 is a longitudinal cross-sectional view of a hydraulic pressure device incorporating the invention of the application;

FIGURE 2 is a lateral cross-sectional view through the hydraulic pressure generating gerotor structure of figure 1 taken substantially along the lines 2-2 in such figure;

FIGURE 3 is a cross-sectional view of the wear plate of the embodiment of figure 1 taken generally from line 3-3 in that figure;

FIGURE 4 is a cross-sectional view of the wear plate of figure 3 taken generally from line 4-4 in that figure;

FIGURE 5 is a cross-sectional view of the manifold plate of the embodiment of figure 1 taken generally from line 5-5 therein;

FIGURE 6 is a lateral face view of the back side of the manifold plate of figure 5 taken generally along lines 6-6 in figure 7;

FIGURE 7 is a lateral cross-sectional view of the manifold plate of figure 1 taken generally from lines 7-7 in figure 6;

FIGURE 8 is an enlarged representational view of the orientation of the edge of the wear plate to the edge of the housing in figure 1 prior tightening of the assembly bolts;

FIGURE 9 is a view like figure 13 after the tightening of the assembly bolts;

FIGURE 10 is an enlarged view of the top of the manifold in figure 1 highlighting the limited contact thereof;

FIGURE 11 is a view of the stator of figure 10 detailing the orientation of the limited contact of the manifold therewith;

FIGURES 12-16 are selective cross-sectional views of the plates in the rotating valve of the gerotor device of figure 1;

FIGURE 17 is a perspective view of the plates of figures 12-16 prior to assembly into a valve; and,

FIGURES 18-20 are cross-sectional views of a modified White motors incorporating embodiments of the invention:

Detailed Description of the Invention

[0014] This invention relates to an improved pressure device. The invention will be described in its preferred embodiment of a gerotor motor having a rotating valve separate from the gerotor structure.

[0015] The invention would also be amenable to gerotor valved gerotor motors such as the White Model RE, shaft valved devices such as the White Model RS, separate wobblestick toe valved devices such as the White Model HB, TRW M Series, and/or other devices.

[0016] The gerotor pressure device 10 includes a bearing housing 20, a drive shaft 30, a gerotor structure 40, a manifold 50, a valving section 80 and a port plate. 100.

[0017] The bearing housing 20 serves to physically support and locate the drive shaft 30 as well as typically mounting the gerotor pressure device 10 to its intended use such as a cement mixer, mowing deck, winch or other application.

[0018] The particular bearing housing of figure 1 includes a central cavity having two roller bearings 21 rotatively supporting the drive shaft therein. A shaft seal 22 is incorporated between the bearing housing and the drive shaft in order to contain the operative hydraulic fluid within the bearing housing 20. Due to the later described integral drain for the cavity 25 within the bearing housing 20 this shaft seal 22 can be a relatively low pressure seal. The reason for this is that the later described drain reduces the pressure of the fluid within the cavity 25 from full operational pressure, typically 137.90 - 275.80 bar (2,000-4,000 PSI), down to a more manageable number, typically 6.895 - 13.790 bar (100-200 PSI). The use of roller bearings 21 in the pressure device encourages the flow of fluid within the cavity 25 due to the fact that the bearings 21 inherently will move fluid from their small diameter section to their large diameter section. This facilitates in the lubrication and cooling of these critical components. In addition to the above, a series of radial holes 32 in the drive shaft further facilitates the movement of fluid within the cavity 25 across the bearings 21.

[0019] Note that due to the fact that the bore or cavity 25 of the bearing housing 20 has a set of reducing size diameters with no blind grooves or increasing size diameters, it is possible to machine the bearing housing from one side thereof at a single machine with a single setup. This insures the alignment of all of the bores of the bearing housing 20 (i.e. the four steps shown) in addition to reducing the complexity and cost of manufacture of this part.

[0020] A wear plate 27 completes the bearing housing 20. This wear plate is a separate part from the bearing housing 20. As such it can be, and preferably is, made of different materials than the housing proper. Further, the wear plate 27 has a axial length slightly greater than the cavity 28 within which it is inserted. This allows the wear plate 27 to be axially clamped between the later described gerotor structure 40 and the remainder of the bearing housing 20 (while also allowing a solid bedding contact therebetween outside of such wear plate as later described). This construction serves to strengthen the housing as well as reducing the leakage from the pressure cells of the gerotor structure, thus improving the efficiency of the gerotor motor (contrast fig 8 with fig 9). The wear plate 27 in addition serves to lock the bearings 21 in place in respect to the bearing housing 20 eliminating the need for a separate retainer.

[0021] The difference between the axial length of the wear plate 27 and the cavity 28 within which it is inserted is primarily based on the modulus of elasticity (Young's modulus) between the materials of the wear plate and housing in combination with the compressive stress-strain curve for both materials. This will insure that the captured part will be compressed without the device being physically damaged during assembly and/or subsequent use. (See the explanation of material properties in the full text of, and particularly the stress-strain section, of Mark's Handbook for Mechanical Engineers Published by McGraw-Hill of New York, New York, incorporated by reference for explanation of stress-strain curves and Young's modulus.) In addition, the thickness/length/area of any relatively uncompressed sections must be considered; with too elastic a plate 27, bowing may exist. Therefore, the dimensions and function of the remainder of the device should also be considered. (It may also be determined on the basis of experimentation or otherwise if desired.)

[0022] In the particular preferred embodiment disclosed, the bearing housing 20 is made of ductile steel having a Young's modulus of elasticity of 162,716 N/mm^2 (23.6 x 10^6 lbs./square inch) while the wear plate 27 is a powder metal part having a Young's modulus of elasticity of 124,106 N/mm^2 (18 x 10^6 lbs./square inch). By having the wear plate 27 a powder metal part, the natural porosity thereof aids in the lubrication of the rotor 45 of the later described gerotor structure 40, thus increasing the mechanical efficiency of the overall device. The axial length of this wear plate 27 is selected such that with designed assembly torque on the main housing bolts that hold the device 10 together, the stator 41 bearingly engages the bearing housing radially outward of the wear plate 27 (contrast fig 8 bolts not tightened with fig 9: bolts tightened - gap shown out of scale for clarity of presentation). In the embodiment disclosed, this axial length is 0.0254 mm to 0.0762 mm (.001" to .003") greater than the cavity 28 within which the wear plate 27 is located (up to 0.254 mm (.010") could be utilized in this example). This allows the stator 41 of the gerotor structure 40 to fully seat against the bearing housing 20. It also allows for the use of a single simple face seal 54 between the housing 20 and the stator 40 to seal this location - a joint of three parts, reducing manufacturing and maintenance costs. Indeed, the wear plate 27 could be left out without effecting the hydraulic integrity of this seal under pressure.

[0023] The wear plate 27 shown is some 76.2 mm (3") in diameter and 16.51 mm (65") in axial length with a 32.512 mm (1.280") central hole therein. The preferred wear plate 27 is some 0.127 mm to 0.254 mm (.005" to . 010") smaller in diameter and 0.0254 mm to 0.0762 mm (.001" to .003 ") longer in axial length than the cavity 28 in which it resides. (Note that the 16.51 mm (.65") total axial length of the wear plate 27 is such that no compensation to reduce bowing is needed, in contrast with the later described manifold 60.) A further 50.8 mm (2") diameter and 3.683 mm (.145") depth clearance groove on the inside surface thereof allows the wear plate to contact only the outer race of the bearing stack, thus allowing relatively unfettered rotation of the bearings 21. A small 15.748 mm (.062") alignment slot 26 and similar slot 24 in the inner surface of the bearing housing 20 allows a small pin to be inserted to initially lock the two in relative rotary position. This insures that upon clamping assembly the roll recesses 29 and balancing recesses 31 will be properly aligned with the rolls 43 and gerotor cells 47 of the adjoining gerotor structure 40. The roll recesses 29 are 9.525 mm (.375") in diameter and 0.889 mm (.035") in depth, axially aligned with an adjoining stator roll 43 (themselves some 12.7 mm (.5") in diameter on a 58.42 mm (2.30") diameter bolt circle). These recesses reduce the axial pressure on the rolls 43 while still providing for a relative seal between adjoining gerotor cells 47. The balancing recesses 31 are aligned with the gerotor cells 47 (and later described manifold openings), extending 10.16 mm (.4") wide and 6.35 mm (.250") high and some 0.508 mm (.020") deep on a 28.575 mm (1.125") circle. These balancing recesses 31 serve to reduce chattering in the gerotor structure 40.

[0024] The wear plate 27 shown, some 0.0762 mm (. 003") greater in axial length than the cavity 28 within which it is located, is under the pressure of four 9.525 mm (.375") diameter grade 8 bolts having national fine threads (24 per 25.4 mm (inch)). When tightened to 67.8 Nm (50 foot lbs.) of torque, each bolt produces approximately 43,593 N (9,800 lbs. of force). This force compresses the wear plate 27 sufficiently to solidly seat the bearing housing 20 to the stator 41. Further, the resistance of the wear plate 27 to being compressed aids in retaining the bolts in place by preloading same.

[0025] The drive shaft 30 is rotatively supported within the bearing housing 20 by the bearings 21. This drive shaft serves to interconnect the later described gerotor structure 40 to the outside of the gerotor pressure device 10. This allows rotary power to be generated (if the device is used as a motor) or fluidic power to be produced (if the device is used as a pump). In addition to the previously described radial hole 32, a hole 33 drilled in the radial surface of the drive shaft 30 and the pumping action of the radial bearings 21 facilitate the movement of fluid throughout the cavity 25 thus to further facilitate the lubrication and cooling of the components contained therein. The drive shaft 30 includes a central axially located hollow which has internal teeth 35 cut therein. The hollow provides room for the wobblestick 36 while the internal teeth 35 drivingly interconnect the drive shaft 30 with such wobblestick 36. Additional teeth 37 on the other end of the wobblestick drivingly interconnect the wobblestick 36 to the rotor of the later described gerotor structure, thus completing the power generating drive connection for the device. A central hole 38 drilled through the longitudinal axis of the wobblestick 36 further facilitates fluid communication through the device.

[0026] The gerotor structure 40 is the main power generation apparatus for the pressure device 10 (fig 2). The particular gerotor structure 40 disclosed includes a stator 41 and a rotor 45 which together define gerotor cells 47. As these cells 47 are subjected to varying pressure differential by the later described valve, the power of the pressure device 10 is generated. This occurs because the axis of rotation 46 of the rotor is displaced from the central axis 42 of the stator (the wobblestick 36 accommodates this displacement). As the rotor 45 moves, the inner section of the lobes 48 of such rotor define an inner limit circle 49. This inner circle 49 defines the innermost extension of the gerotor cells 47. In the invention of the present application, there are fluid passages 50 which extend from this innermost extension 49 to the central area 52 within the pressure device 10. Due to this extension, an amount of fluid can be parasitically drawn off of the cells 47 to pass into the central area 52. This serves simultaneously to lubricate the critical moving components of the pressure device 10 in addition to providing a cooling and lubrication function therefor.

[0027] In the preferred embodiment disclosed in figure 1, these passages 50 are "T" slots formed in the surface of the wear plate 27 adjoining the gerotor structure 40 (see fig 3). With the slots so positioned, there is one slot interconnected to the dead pocket in a top dead center position rotor with a second slot 53 leaking to the central area 52 of the pressure device. In a corresponding bottom dead center position, there is again one leakage path going to the dead pocket and a further slot 54 starting to have leakage to the central area 52. The radial extension 55 at the outer end of the passages 50 allow for an increased amount of leakage over a longer period of time than would be possible with a straight laterally extending passage 50 (i.e. without the radial extension 55). This facilitates the continuity of the flow of the lubrication fluid into the central area 52 of the device. The location of the passages 50 in the wear plate 27 is preferred to a location in the later described manifold due to its axial separation from the later described pressure release mechanism in the rotating valve of the valving section 80. Note that although the passages 50 are shown located in a nonmoving part, the wear plate 27, they could also be located in the rotor 45 as long as the same conditions are met - i.e. there is a leakage path from the gerotor cells 47 into the central area 52 of the device. This would be accomplished by placing a small inwardly extending passages 58 within the rotor 45, preferably at the base of the lobes thereof, sufficiently long enough to extend into the central hole of the wear plate 27 or later described manifold 60 thus to provide for the desired leakage.

[0028] The manifold 60 in the port plate 100 serves to fluidically interconnect the later described valve to the gerotor cells 47 of the gerotor structure 40, thus to generate the power for the pressure device 10. In the particular embodiment disclosed, since the valve is an orbiting valve, phase compensation is not necessary. As such, the through valving passages 62 can extend straight through the manifold 60. With differing valving mechanisms such as the valve in rotor of the White Model RE or the separate wobblestick toe driven orbiting valve of figure 20 or the TRW M Series, phase compensation may be included in the manifold (typically 90o or so). With a shaft valve such as that in the White Model RS, the positions of the wear plate 27 and manifold 60 may be reversed (see fig 18). The particular manifold disclosed includes recesses 64 directly centered on the rolls 43 of the stator 41 (figs 5-7). These serve to reduce the axial pressure on such rolls 43 while maintaining a good seal between adjoining gerotor cells 47 (corresponding recesses 29 and the wear plate 27 provide a similar function at the other end of the rolls 43). In addition, the manifold opening 61 are expanded at their interconnection with the gerotor cells 47 relative to the openings of the through valving passages 62 on the other side of such manifold (contrast fig 5 with fig 6). (Balancing recesses 31 in the wear plate 27 serve to equalize the pressure on the other sides of the rotor 45).

[0029] As with the wear plate 27, the axial length of the manifold 60 is preferably greater than the axial length 65 of the cavity in the port plate 110 within which it is contained as previously set forth: this serves to clamp the gerotor structure 40 with pressure on both sides thereof, thus to reduce leakage and improve the overall mechanical efficiency of the pressure device. Similarly, the manifold 60 is of powder metal construction. A small groove 119 extending about the manifold 60 adjacent to the outer circumference of the valve 80 provides clearance for any incidental burrs thereon.

[0030] By having the manifold 60 a powder metal part, again as with the wear plate 27, the natural porosity thereof aids in the lubrication of the rotor 45 of the later described gerotor structure 40, thus increasing the mechanical efficiency of the overall device. The axial length of this manifold 60 is selected such that with normal torque on the main housing bolts that hold the device 10 together, the stator 41 engages the port plate 110 radially outward of the manifold 60. It is preferred that this axial length be similar to that of the wear plate 27 so as to provide for substantially equal forces on both sides of the gerotor structure 40. In the embodiment disclosed this axial length is again 0.0254 mm to 0.0762 mm (.001 " to .003 ") greater than the cavity within which the manifold 60 is located. This allows the use of a second single simple face seal 73 to seal this location, reducing manufacturing and maintenance costs. Indeed, the manifold 60 could be left out without effecting the integrity of this seal (Note that this seal 73 could be located symmetrically with the wear plate seal. However, as later described, this would disturb the function of the centering of contact area on either side of the manifold (via ring 115) due to the reduction of the area 117 outside of such ring 115. This would necessitate movement/redesign of the ring 115 to provide is function).

[0031] The outer diameter of the manifold is 0.127 mm to 0.254 mm (.005" to .010") smaller than the inner diameter of the port plate 110 surrounding the same and 0.0254 mm to 0.0762 mm (.001" to .003") longer in axial length than the cavity within which it resides. Again the axial length oversize was determined based on the modulus of elasticity in combination with the other factors set forth in respect to the wear plate 27.

[0032] In order to facilitate the cooperation between the captured part and neighboring pieces, it is possible that at least one of the adjoining elements in compressive contact would have a reduced surface area in respect to the total area otherwise available (i.e. the full overlapping of adjoining surfaces). This reduced surface area increases the unit loading, thus to facilitate the compression process. The surface area of contact can therefor be adjusted to such a certain cooperation of parts and/or the location and amount of resulting forces. Adjustments include a) the materials (i.e. relative Young's modulus), b) the materials dimensions including thickness and diameter, c) the value of the compressive forces, d) the total surface area of contact, e) the concentration of the area of contact over the surface area of contact available (for example a single narrow groove vs. knurling the entire surface area to produce the same area of contact, f) the symmetry of the area of contact, g) the location of the area of contact, h) the direction of the compressive force, and, i) the concentration of the compressive force and other variables.

[0033] For example, in the case of the manifold 60 there normally would be a contact area with the port plate 110 of some 1.5 square inches (the outer diameter of the manifold 60 being 73.66 mm (2.9") and the inner diameter of the port plate 110 about the valve 80 being 64.77 mm (2.55"). As there if 43,593 N (9800 pounds) of loading on four bolts (174,372 N (39,200 pounds) total) the pressure on the contact area normally would be 180.18 N/mm^2 (26,133 pounds per square inch). However, as later set forth, in this embodiment the contact area is reduced to a 63.5 mm (2.5") center to center diameter ring 115 some 0.762 mm (.030") in width. This reduces the surface area to 158.06 mm^2 (.245 square inches). With the same loading this produces a revised pressure on the reduces contact area of 1103.16 kg/mm^2 (160,000 pounds per square inch) - a unit loading over six times greater than before. This allows for an adjustment of the amount of unit loading.

[0034] This reduced surface area is preferably provided first in the element having the higher modulus of elasticity. This lowers the chances of structural damage to the reduced area. It also increases the volume of material available in the other element to absorb the increased loading (this in recognition of its lower strength). Note that with certain combinations of materials it is possible for the reduction in surface area to increase the unit loading so far that the combination is self-destructive. Therefor a limit exists that such destruction not occur during the designed service life of the device containing the elements. This life including the extra margin of life normally included in the device.

[0035] The actual reduction in surface area can be provided by grooves, slots, cross-hatching, impressed dimples, knurling or other technique as previously set forth. The choice depends primarily on the materials to be utilized together with other design criteria.

[0036] In respect to materials, in general the closer the modulus of elasticity between materials, the higher the ratio can be between the reduced surface area to the full available area and the higher a given individual point of loading can be (it being assumed that the full available area is itself greater than that necessary to transfer the loading without destruction of either elements; if it is not no reduction is possible). The reason for this is that the materials having similar values are less likely to destroy each other under higher unit loading. In respect to design, the thicker the materials and the closer the transfer of forces is compression of the elements (as opposed to angular, shear, etc.) the greater the unit loading can be. The reason for this is both that materials are typically stronger with straight compression loads and that any angular forces reduce this strength. These angular forces also might compromise the location and/or operation of the involved parts (and/or others).

[0037] As an example of the above, the particular manifold 60 shown is thinner than the wear plate 27 while being the same material. In the particular preferred embodiment this would ordinarily cause the manifold 60 to bow slightly towards the rotor 45, potentially reducing the mechanical efficiency for the device. The reduced contact area between the port plate 110 and the manifold 60 also reduces this bowing by machining the ring 115 at a certain location between the surfaces of contact therebetween. In specific the ring 115 is located such that the contact area between the manifold 60 and stator 41 outside 117 of a circle 116 equal in diameter to the center of the ring 115 is substantially the same as the contact area between the manifold 60 and stator inside 118 such circle (figs 10 and 11). The ring 115 is thus located substantially in the center of the area of contact with the stator on the opposite side of the manifold 60. This balances out the forces on either side of the manifold (i.e. both contacts are located on the same radius from the center of the manifold, albeit with different surface area). This reduces the loading imbalance of the manifold and thus any bowing tendency. Alternately, the manifold 60 could have been made 2.54 mm to 5.08 mm (.10" to . 20") thicker.

[0038] The manifold 60 shown is some 73.66 mm (2.9") in diameter and 14.224 mm (.56") in axial length with a 29.718 mm (1.170") central hole therein. A small 3.175 mm (.125") alignment slot and similar slot in the inner surface of the port plate 110 lock the two in relative rotary position. This insures that upon clamping assembly the roll recesses 64 and manifold openings 61 will be properly aligned with the rolls 43 and gerotor cells 47 of the adjoining gerotor structure 40. The roll recesses 29 are 9.525 mm (.375") in diameter and 0.889 mm (.035 ") in depth, axially aligned with an adjoining stator roll 43 (themselves some 12.7 mm (.5") in diameter on a 58.42 mm (2.30") diameter bolt circle). These recesses reduce the axial pressure on the rolls 43 while still providing for a relative seal between adjoining gerotor cells 47. Again, the manifold is subjected to the approximately 43,593 N (9,800 lbs.) of force per each of the four main assembly bolts to compress such plate. The circle 116 is located at a 33.02 mm (1.30") radius with the ring 115 itself 0.762 mm (.030") wide and 1.016 mm (.040") above the remainder of the port plate 110.

[0039] The valving passages complete the manifold 60.

[0040] On one side of the manifold there are narrow valving passages 62 alternating with a series of balancing recesses 63. These are both substantially equal size and equally spaced 4.2672 mm (.168") wide and 8.8392 mm (.348") long on a 28.65 mm (1.128") radius circle. The balancing recesses 63 extend some 0.889 mm (.035") deep in a groove extending some 1.016 mm (.040") wide about the perimeter of the recess. (The center portion of the recesses extend to the surface of the manifold 60 - full area depth is not necessary to provide the balancing function: the center portion aids in supporting the rotary valve 81.)

[0041] On the other side of the manifold there are broad manifold openings 61 alternating with a series of balancing recesses 64. The manifold openings 61 are substantially 11.938 mm (.470") wide and 8.839 mm (.348") long on a 28.65 mm (1.128") radius circle. (There is a transition between the narrow valving passages 62 and the wider manifold openings by an angular transition section in the middle third of the manifold 60. The difference provides for accurate valving on one side and maximum flow/low pressure drop on the other.) The balancing recesses 64 are again 9.525 mm (.375") in diameter and 0.889 mm (.035") deep centered on the rolls 43 of the gerotor structure on a 58.42 mm (2.30") bolt circle.

[0042] The combination of wear plate 27 and manifold 60, trapped as they are, provide for increased manufacturing efficiency (being of pressed metal and not forged or machined), increased mechanical and pressure efficiency (by reliably and predictably closing and lubricating both sides of the gerotor structure 40) and increased serviceability (replacing the two parts in combination with at least the rotor (and in some cases the stator) produces the equivalent of a new gerotor structure.

[0043] The manifold 60 in the port plate 110 also can serve as a location for an additional/alternate dedicated leakage path. Although not preferred as a location for a leakage path (due to its proximity to the case drain in the valve) it was discovered that the area 71 immediately surrounding the manifold 60 was subjected to high pressure when the outer port 113 pressurized, primarily via leakage past the outer surface of the valve 80. This provided a relatively convenient source or lubrication fluid for a leakage path. In addition a leakage path at this location would lower the relative pressure at this location (and on the seal 73). The inclusion of a hole 72, or series of holes 72, from this area 71 to the center 70 of the manifold 60 provides this. (If the outer port 113 is connected to low pressure, since the later described case drain in the valve would be. Also, the hole 72 is relatively pressure balanced between its inner and outer ends. It would thus not compromise the volumetric efficiency of the device under this alternate connection.) The aggregate cross-sectional size of the hole(s) 72 is preferably selected such that it is larger than the smallest of the leakage path from about the valve 80 to the area 71 on the outside circumference of the manifold 60. This allows the fluid to drain from such area 71 to the center 70 of such manifold 60 without relative restriction. Note, however, that in certain applications it may be appropriate to size the hole 72 such that it does limit flow - for example where such flow would unduly compromise the volumetric efficiency of the device or where a back pressure is desired (typically for a secondary purpose). The particular hole has a diameter of about 3/16 of an inch, providing about .25 gallons of lubrication fluid for a 25 gallon input. This hole 72 may be included in addition to or instead of the previously described first dedicated leakage passage.

[0044] The second fluid leakage passage 72 in the manifold 60 could also form part of a separate case drain for the hydraulic device (for use with or instead of the later set forth valve case drain). This would be attractive for applications wherein a separate drain line isolated from the valve 80 or ports 110, 113 is desired. To provide for this separate case drain a drain port 75 would be located extending from the area 71 to the outside of the device, preferably directly radially outwards so as to simplify its manufacture. The drain port 75 would be threaded or otherwise rendered into a form for an external drainline (not shown). Multiple holes 72 would be preferred on an outer circumferential groove so as to increase the connection dwell time between the port 75 and the center 70 of the manifold 60 (via holes 72). This drain port 75 would simultaneously lower the unit pressure on the area 71 (especially if port 113 is pressurized) while also providing for a case drain for the center 52 of the device 10. This simplifies the device while simultaneously reducing the design limits of the parts therein. Towards this end if the first set of dedicated leakage paths is eliminated it is preferred that longitudinal hole 31 be included in the wobblestick 36 (fig 1). This hole 31 allows movement of fluid down the center of the wobblestick towards the drive connection 35, such movement assisted by the centripetal radial forces on the fluid provided by hole 32 and the previously described pumping action of the front bearing 21. The holes 23 and the back bearing 21 further encourage movement of fluid in the center of the device and across the back drive connection 37. These connections are cooled and lubricated by this fluid flow.

[0045] The valving section 80 selectively valves the gerotor structure to the pressure and return ports. The particular valve 81 disclosed is of brazed multi-plate construction including a selective compilation of five plates (figs 12-17). The particular valve 81 is a eleven plate compilation of two communication plate 82's, four first transfer plates 83, a single second transfer plate 84, a radial transfer plate 85 and three valving plates 86.

[0046] The communication plate 82 contains an inner area 83 which communicates directly to the inside port 111 in the port plate. The communication plate 82 also contains six outer areas 89 which are in communication with the outside port 113. The plate thus serves primarily to interconnect the valve 81 to the pressure and return ports of the gerotor pressure device 10.

[0047] In order to provide for the necessary alternating passages 105, 106 in the valving plate 86, the first and second transfer plates 83, 84 shift the fluid from the inner 88 and outer 89 areas.

[0048] The first transfer plate 83 contains a series of three first intermediate passages 90 which serve to begin to transfer fluid from the inner area 88 outwards. It also includes a series of six second outward passages 91 which communicate with the outer areas 89 in the communication plate to laterally transfer fluid. Since the outside port 113 directly surrounds the valve 81, these passages 91 also serve to interconnect to the outside port 113.

[0049] A second transfer plate 84 completes the movement of the fluid from the inner and outer areas of the communication plate 82. It accomplishes this by a series of three second intermediate passages 93 which serve to complete the radial movement of fluid from the inner area 88 of the communication plate 82. A set of third outer passages 94 interconnect with the second outward passages 91 in the transfer plate 83 and the first outer area 89 in the communication plate 82 to complete the lateral movement of fluid therefrom. Again, since the outside port 113 surrounds the valve, the third outer passage 94 also directly interconnects to the outside port 113.

[0050] The radial transfer plate 85 segments the second intermediate passages 93 so as to provide for the alternating valving passages in the valving plate 86. This is provided by cover sections 96 for the middle of such passages 93. This separates the two passages 97, 98 therein to initiate alternated placement thereof.

[0051] The valving plate 86 contains a series of alternating passages 105, 106 which terminate the inner 88 and outer 89 areas of the commutation plate 82 to complete the passages necessary for the accurate placement of the valving openings in the device. In the valving plate 86 the first 105 of the alternating valving passages is thus interconnected to the inside port 111 while the second 106 of the alternating passages is connected to the outside port 113 by the previously described passages.

[0052] Due to the use of a multiplicity of plates, the opening available for fluid passage is increased over that which would be available if only a single plate of each type was utilized.

[0053] In addition to the above valving function, the valving section 80 also includes a pressure release mechanism for the central area 52 of the gerotor pressure device. This pressure release mechanism includes three through holes 100-102, each containing a ball check valve 107, in combination with two valve seats. The holes 100-102 extend through the communication plate 82 and transfer plate 83, 84. These holes service to allow for the passage of fluid through the valve 81 in addition to providing a physical location for the two balls 107 contained within the holes 100-102. The balls 107 themselves cooperate with two valve seats in plate 84 in order to interconnect the central area 52 to the inside port 111 or outside port 113 having the lowest relative pressure. This provides for a self-contained case drain for the cavity 25 of the hydraulic device, thus allowing the circulation of fluid therein as well as lowering the pressure thereof. Of the two valve ball retaining holes, the outermost is interconnected to the outside port 113 while the inner hold interconnects to the inside port 111. The middle hole sweeps the area covered by the later described balancing piston 120. Due to the fact that the valve seats and middle hole are connected to the central area by passages 103, 104 in the radial transfer plate 85 respectively, the fluid in the cavity 25 is free to flow to the port having the lowest relative pressure. By integrating these pressure release valves with the rotating valve, the overall complexity and cost of the gerotor pressure device. Note it desired multiple leakage paths could be provided - for example to include two more sets of holes 100-102 on 120o and 240o spacing from the first set (with the appropriate arms in the plates slightly widened to provide for same).

[0054] The valve 81 is itself rotated by a valve stick interconnected to the rotor 45 and thus through the wobblestick 36 to the drive shaft. This provides for the accurate timing and rotation of the valve 81.

[0055] A balancing piston 120 on the port plate 110 side of the valve 81 separates the inside port 111 from the outside port 113, thus allowing for the efficient operation of the device. This balancing ring is substantially similar to that shown in the U.S. Patent 3,572,983, Fluid Operated Motor.

[0056] The port plate 110 serves as the physical location for the valving section 80 in addition to providing a location for the pressure and return ports (not shown). It thus completes the structure of the gerotor pressure device 10. Note that in this port plate 110 again the bore about the manifold 60 and the plates 86 of the valve 80 together with the groove for the balancing piston 120 can be machined by a single machine with a single setup, again reducing the cost of this part while insuring the alignment of all bores (i.e. two steps and a groove). The ports 111 and 113 together with their fittings are relatively non-critical, having no low tolerance high placement accuracy requirements. Note also that there are no passages, check valve seats, access plug fittings or other fluidic opening machined in the body of the housing of the device; only the threaded holes for the fittings to the two ports 111, 113 need be especially machined. The device is thus very simple compared to the complicated machined housings of the art cited in the prior art section of this application.

[0057] Although the invention has been described in its preferred form with a certain degree of particularity, it is to be understood that numerous changes can be made without deviating from the invention as hereinafter claimed.

[0058] Examples are set forth in figures 18, 19 and 20 which are variations of the White Model RS (U.S. Patent 4,285,643) (fig 18), the embodiment disclosed in figure 1 herein (fig 19) and HB (U.S. Patent 4,877,383) (fig 20) the contents of which are included by incorporation. In the figures 18 and 20, the plates 100 are dimensioned in respect to their respective cavities so as to bow out slightly towards a moving part on assembly of the device (the rotor in fig 18 and the valve in fig 20). Both plates are approximately 9.525 mm (.375") thick. This bowing out biases the moving part so as to aid in the equalization of fluidic pressure thereon (fig 18) or to aid in the equalization of the mechanical pressure and sealing (fig 20). The manifold in figure 19 is substantially the same as the manifold 60 herein except that it is made of steel. Further in any embodiment the joint between surrounding housing parts could be the confines of the manifold (dotted lines 130 - figs 18-20 - these new joints could replace on supplement those adjoining).

[0059] Other modifications are also possible within the scope of the claims.

Claims

1. A gerotor hydraulic pressure device having a rotor with a surface adjoining a housing and a stator surrounding the rotor,

   said housing having a cavity adjacent to a surface of the rotor, said cavity having a depth,

   said stator closing one side of said cavity, and in a tightened configuration of the device said stator being seated to the housing,

   a plate being in said cavity, and in a tightened configuration of the device said plate being clamped between the stator and the housing,

   <u>**characterized in that**</u>

   said plate is compressible,

in an un-tightened configuration of the device said plate has an axial length greater than said depth of said cavity, and

in said tightened configuration of the device said axial length of said plate is compressed to said depth of said cavity, and the difference between the axial length of the plate in an un-tightened configuration and the depth of the cavity is primarily based on the difference of modulus of elasticity between the materials of the plate and housing in combination with the compressive stress-strain curve of both materials.

2. The gerotor hydraulic pressure device of claim 1 **characterized in that** said plate is a manifold.

3. The gerotor hydraulic pressure device of claim 1 or 2 **characterized in that** said plate is of powder metal.

4. The gerotor hydraulic pressure device according to any one of the preceding claims wherein the housing is held together by bolts and **characterized in that** tightening the bolts fully compresses said plate.

5. The gerotor hydraulic pressure device according to any one of the preceding claims wherein the housing, stator, and plate each have modulus of elasticity and **characterized in that** the modulus of said plate is lower than either the modulus of the housing or stator.

6. The gerotor hydraulic pressure device according to any one of the preceding claims wherein said plate has a compressive stress/strain curve and **characterized in that** said plate is compressed within the limits of such curve.

7. The hydraulic pressure device according to any one of the preceding claims **characterized in that** said cavity has a cavity surface substantially parallel to the surface of the rotor, said plate having a plate surface, said plate surface overlapping said cavity surface, and at least one of said plate surface or said cavity surface being reduced to provide a reduced surface area of contact therebetween.

8. The hydraulic pressure device of claim 7 **characterized in that** said reduced surface area of contact is a ring.

9. The hydraulic pressure device of claim 8 **characterized in that** said ring is substantially centered on the overlap between said plate surface and said cavity surface.

10. The gerotor hydraulic pressure device according to any one of the preceding claims wherein the rotor has a second surface parallel to the surface and **characterized by** the addition of a second cavity, said second cavity being in the housing adjacent to the second surface of the rotor, said second cavity having a depth,

a second plate, said second plate being compressible, said second plate having an axial length, in said un-tightened configuration of said device said axial length of said second plate being greater than said depth of said second cavity, said second plate being in said second cavity, in said tightened configuration said axial length of said second cavity compressed to said depth of said second cavity,

said stator closing one side of said second cavity, and said stator being seated to the housing.

11. The gerotor hydraulic pressure device of claim 10 **characterized in that** said second plate is a manifold.

12. The gerotor hydraulic pressure device of claim 10 or 11 wherein the housing is held together by bolts and **characterized in that** tightening the bolts fully compresses said plate and said second plate.

13. The gerotor hydraulic pressure device according to any one of claims 10 to 12 wherein the housing, stator, said plate and said second plate each have modulus of elasticity and **characterized in that** the modulus of said plate and said second plate is lower than either the modulus of the housing or stator.

14. The gerotor hydraulic pressure device according to any one of claims 10 to 13 wherein said plate and said second plate have a compressive stress/strain curve and **characterized in that** said plate and said second plate are compressed within the limits of such curve.

15. The gerotor hydraulic pressure device according to any one of claims 10 to 14 **characterized in that** said second cavity has a second cavity surface substantially parallel to the second surface of the rotor, said second plate having a second plate surface, said second plate surface overlapping said second cavity surface, and at least one of said second plate surface or said second cavity surface being reduced to provide a reduced surface area of contact therebetween.

16. The gerotor hydraulic pressure device of claim 15 **characterized in that** said reduced surface area of contact is a ring.

17. The gerotor hydraulic pressure device of claim 16 **characterized in that** said ring is substantially centered on the overlap between said second plate surface and said second cavity surface.

18. The gerotor hydraulic pressure device according to any one of the preceding claims **characterized in that** said plate has an outside, an inside cavity, and means in said plate to connect said outside to said inside cavity.

19. The gerotor hydraulic pressure device of claim 18 **characterized by** the addition of a case drain and means in the housing to connect said outside to said case drain.

20. The gerotor hydraulic pressure device of claim 19 **characterized by** the addition of a case drain and means in the housing to connect said inside cavity to said case drain.

21. The gerotor hydraulic pressure device according to any one of the preceding claims **characterized by** an area, said area being located in said cavity outside of said plate, said plate having an inside cavity, a hole, and said hole being in said plate fluidically connecting said area to said inside cavity of said plate.

22. The gerotor hydraulic pressure device of claim 21 **characterized by** the addition of a case drain, said case drain being in said housing substantially outwards of said cavity, and said case drain connected first to said area and then said hole.

23. The gerotor hydraulic pressure device of claim 21 or 22 **characterized by** the addition of a case drain, said case drain being in said housing, and said case drain connected first to said inside cavity in said plate and then said hole.

Patentansprüche

1. Gerotor-Hydraulikdruckvorrichtung, welche einen Rotor mit einer an ein Gehäuse grenzenden Oberfläche und einen den Rotor umgebenden Stator aufweist,
wobei das genannte Gehäuse nahe einer Oberfläche des Rotors einen Hohlraum aufweist, wobei der genannte Hohlraum eine Tiefe aufweist,
wobei der genannte Stator eine Seite des genannten Hohlraums schließt und der genannte Stator in einer festgezogenen Konfiguration der Vorrichtung an dem Gehäuse aufsitzt,
wobei sich eine Platte in dem genannten Hohlraum befindet und die genannte Platte in einer festgezogenen Konfiguration der Vorrichtung zwischen dem Stator und dem Gehäuse eingeklemmt ist, **dadurch gekennzeichnet, dass**
die genannte Platte komprimierbar ist,
die genannte Platte in einer entspannten Konfiguration der Vorrichtung eine axiale Länge aufweist, welche größer ist als die genannte Tiefe des genannten Hohlraums, und
in der genannten festgezogenen Konfiguration der Vorrichtung die genannte axiale Länge der genannten Platte auf die genannte Tiefe des genannten Hohlraums komprimiert ist und die Differenz zwischen der axialen Länge der Platte in einer entspannten Konfiguration und der Tiefe hauptsächlich auf der Elastizitätsmodul-Differenz zwischen den Materialien der Platte und des Gehäuses in Verbindung mit der Kompressions-Spannungs-Dehnungs-Kurve beider Materialien beruht.

2. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 1, **dadurch gekennzeichnet, dass** die genannte Platte ein Verteiler ist.

3. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 1 oder 2, **dadurch gekennzeichnet, dass** die genannte Platte aus Pulvermetall besteht.

4. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, wobei das Gehäuse von Bolzen zusammengehalten wird und **dadurch gekennzeichnet, dass** durch Festziehen der Bolzen die genannte Platte vollständig komprimiert wird.

5. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, wobei das Gehäuse, der Stator und die Platte jeweils ein Elastizitätsmodul aufweisen und **dadurch gekennzeichnet, dass** das Modul der genannten Platte niedriger als das Modul entweder des Gehäuses oder des Stators ist.

6. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, wobei die genannte Platte eine Kompressions-Spannungs-Dehnungs-Kurve aufweist und **dadurch gekennzeichnet, dass** die genannte Platte innerhalb der Grenzen einer derartigen Kurve komprimiert wird.

7. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, **dadurch gekennzeichnet, dass** der genannte Hohlraum eine zu der Oberfläche des Rotors im Wesentlichen parallele Hohlraum-Oberfläche aufweist, wobei die genannte Platte eine Platten-Oberfläche aufweist, wobei die genannte Platten-Oberfläche die genannte Hohlraum-Oberfläche überlappt und wenigstens eine der beiden Oberflächen, nämlich der genannten Platten-Oberfläche oder der genannten Hohlraum-Oberfläche, verkleinert ist, um einen verkleinerten Oberflächen-Kontaktbereich zwischen ihnen zu schaffen.

8. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 7, **dadurch gekennzeichnet, dass** der genannte verkleinerte Oberflächen-Kontaktbereich ein Ring ist.

9. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 8, **dadurch gekennzeichnet, dass** der genannte Ring im Wesentlichen zu der Überlappung zwischen der genannten Platten-Oberfläche und der genannten Hohlraum-Oberfläche zentriert ist.

10. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, wobei der Rotor eine zu der Oberfläche parallele zweite Oberfläche aufweist und **gekennzeichnet durch** die Hinzufügung eines zweiten Hohlraums, wobei sich der genannte zweite Hohlraum in dem Gehäuse nahe der zweiten Oberfläche des Rotors befindet, wobei der genannte zweite Hohlraum eine Tiefe aufweist, eine zweite Platte, wobei die genannte zweite Platte komprimierbar ist, wobei die genannte zweite Platte eine axiale Länge aufweist, wobei bei der genannten entspannten Konfiguration der genannten Vorrichtung die genannte axiale Länge der genannten zweiten Platte größer als die genannte Tiefe des genannten zweiten Hohlraums ist, wobei sich die genannte zweite Platte in dem genannten zweiten Hohlraum befindet, wobei bei der genannten festgezogenen Konfiguration die genannte axiale Länge des genannten zweiten Hohlraums auf die genannte Tiefe des genannten zweiten Hohlraums komprimiert ist, wobei der genannte Stator eine Seite des genannten zweiten Hohlraums schließt und der genannte Stator an dem Gehäuse aufsitzt.

11. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 10, **dadurch gekennzeichnet, dass** die genannte zweite Platte ein Verteiler ist.

12. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 10 oder 11, wobei das Gehäuse von Bolzen zusammengehalten wird und **dadurch gekennzeichnet, dass** durch Festziehen der Bolzen die genannte Platte und die genannte zweite Platte vollständig komprimiert werden.

13. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der Ansprüche 10 bis 12, wobei das Gehäuse, der Stator, die genannte Platte und die genannte zweite Platte jeweils ein Elastizitätsmodul aufweisen und **dadurch gekennzeichnet, dass** das Modul der genannten Platte und der genannten zweiten Platte niedriger als das Modul entweder des Gehäuses oder des Stators ist.

14. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der Ansprüche 10 bis 13, wobei die genannte Platte und die genannte zweite Platte eine Kompressions-Spannungs-Dehnungs-Kurve aufweisen und **dadurch gekennzeichnet, dass** die genannte Platte und die genannte zweite Platte innerhalb der Grenzen einer derartigen Kurve komprimiert werden.

15. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der Ansprüche 10 bis 14, **dadurch gekennzeichnet, dass** der genannte zweite Hohlraum eine zu der zweiten Oberfläche des Rotors im Wesentlichen parallele zweite Hohlraum-Oberfläche aufweist, wobei die genannte zweite Platte eine zweite Platten-Oberfläche aufweist, wobei die genannte zweite Platten-Oberfläche die genannte zweite Hohlraum-Oberfläche überlappt und wenigstens eine der beiden Oberflächen, nämlich der genannten zweiten Platten-Oberfläche oder der genannten zweiten Hohlraum-Oberfläche, verkleinert ist, um einen verkleinerten Oberflächen-Kontaktbereich zwischen ihnen zu schaffen.

16. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 15, **dadurch gekennzeichnet, dass** der genannte verkleinerte Oberflächen-Kontaktbereich ein Ring ist.

17. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 16, **dadurch gekennzeichnet, dass** der genannte Ring im Wesentlichen zu der Überlappung zwischen der genannten zweiten Platten-Oberfläche und der genannten zweiten Hohlraum-Oberfläche zentriert ist.

18. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, **dadurch gekennzeichnet, dass** die genannte Platte einen äußeren und einen inneren Hohlraum aufweist und Mittel in der genannten Platte, um den genannten äußeren mit dem genannten inneren Hohlraum zu verbinden.

19. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 18, **gekennzeichnet durch** die Hinzufügung eines Gehäuse-Abflusses und Mittel in dem Gehäuse, um das genannte Äußere mit dem genannten Gehäuse-Abfluss zu verbinden.

20. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 19, **gekennzeichnet durch** die Hinzufügung eines Gehäuse-Abflusses und Mittel in dem Gehäuse, um den genannten inneren Hohlraum mit dem genannten Gehäuse-Abfluss zu verbinden.

21. Gerotor-Hydraulikdruckvorrichtung nach einem beliebigen der vorangehenden Ansprüche, **gekennzeichnet durch** einen Bereich, wobei der genannte Bereich in dem genannten Hohlraum außerhalb der genannten Platte angeordnet ist, wobei die genannte Platte einen inneren Hohlraum aufweist, ein Loch und wobei das genannte in der genannten Platte befindliche Loch eine Fluidverbindung des genannten Bereichs mit dem genannten inneren Hohlraum der genannten Platte herstellt.

22. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 21, **gekennzeichnet durch** die Hinzufügung eines Gehäuse-Abflusses, wobei sich der genannte Gehäuse-Abfluss in dem genannten Gehäuse im Wesentlichen außerhalb des genannten Hohlraums befindet und der genannte Gehäuse-Abfluss zunächst mit dem genannten Bereich und dann mit dem genannten Loch verbunden ist.

23. Gerotor-Hydraulikdruckvorrichtung nach Anspruch 21 oder 22, **gekennzeichnet durch** die Hinzufügung eines Gehäuse-Abflusses, wobei sich der genannte Gehäuse-Abfluss in dem genannten Gehäuse befindet und der genannte Gehäuse-Abfluss zunächst mit dem genannten inneren Hohlraum in der genannten Platte und dann mit dem genannten Loch verbunden ist.

Revendications

1. Dispositif de pression hydraulique à rotor denté ayant un rotor muni d'une surface contigüe à un boîtier, et un stator entourant le rotor,
ledit boîtier ayant une cavité adjacente à une surface du rotor, ladite cavité ayant une profondeur,
ledit stator fermant un côté de ladite cavité, et dans une configuration serrée du dispositif, ledit stator étant mis en place dans le boîtier,
une plaque se trouvant dans ladite cavité, et dans une configuration serrée du dispositif, ladite plaque étant immobilisée entre le stator et le boîtier,
caractérisé en ce que
ladite plaque est compressible,
dans une configuration non serrée du dispositif, ladite plaque à une longueur axiale supérieure à ladite profondeur de ladite cavité, et
dans ladite configuration serrée du dispositif, ladite longueur axiale de ladite plaque est comprimée à ladite profondeur de ladite cavité, et la différence entre la longueur axiale de la plaque dans une configuration non serrée et la profondeur de la cavité est principalement basée sur la différence de module d'élasticité entre les matériaux de la plaque et du boîtier en combinaison avec la courbe contrainte de compression-déformation des matériaux.

2. Dispositif de pression hydraulique à rotor denté selon la revendication 1, **caractérisé en ce que** ladite plaque est un collecteur.

3. Dispositif de pression hydraulique à rotor denté selon la revendication 1 ou 2, **caractérisé en ce que** ladite plaque est constituée de métal en poudre.

4. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications précédentes, dans lequel le boîtier est maintenu par des boulons, et **caractérisé en ce qu'**un serrage des boulons comprime complètement ladite plaque.

5. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications précédentes, dans lequel le boîtier, le stator et la plaque ont chacun un module d'élasticité, et **caractérisé en ce que** le module de ladite plaque est inférieur au module du boîtier ou du stator.

6. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications précédentes, dans lequel ladite plaque à une courbe contrainte de compression/déformation, et **caractérisé en ce que** ladite plaque est comprimée dans les limites d'une telle courbe.

7. Dispositif de pression hydraulique selon l'une quelconque des revendications précédentes, **caractérisé en ce que** ladite cavité a une surface de cavité sensiblement parallèle à la surface du rotor, ladite plaque ayant une surface de plaque, ladite surface de plaque chevauchant ladite surface de cavité, et au moins l'une de ladite surface de plaque ou de ladite surface de cavité étant réduite pour fournir une aire de surface de contact réduite entre celles-ci.

8. Dispositif à pression hydraulique selon la revendication 7, **caractérisé en ce que** ladite aire de surface de contact réduite est un anneau.

9. Dispositif à pression hydraulique selon la revendication 8, **caractérisé en ce que** ledit anneau est sensiblement centré sur le chevauchement entre ladite surface de plaque et ladite surface de cavité.

10. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications précédentes, dans lequel le rotor a une seconde surface parallèle à la surface, et **caractérisé par** l'ajout d'une seconde cavité, ladite seconde cavité étant dans le boîtier adjacente à la seconde surface du rotor, ladite seconde cavité ayant une profondeur,
une seconde plaque, ladite seconde plaque étant compressible, ladite seconde plaque ayant une longueur axiale, dans ladite configuration non serrée dudit dispositif, ladite longueur axiale de ladite seconde plaque étant supérieure à ladite profondeur de ladite seconde cavité, ladite seconde plaque étant dans ladite seconde cavité, et dans ladite configuration serrée, ladite longueur axiale de ladite seconde cavité étant comprimée à ladite profondeur de ladite seconde cavité,
ledit stator fermant un côté de ladite seconde cavité, et ledit stator étant mis en place sur le boîtier.

11. Dispositif de pression hydraulique à rotor denté selon la revendication 10, **caractérisé en ce que** ladite seconde plaque est un collecteur.

12. Dispositif de pression hydraulique à rotor denté selon la revendication 10 ou 11, dans lequel le boîtier est maintenu par des boulons, et **caractérisé en ce qu'**un serrage des boulons comprime complètement ladite plaque et ladite seconde plaque.

13. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications 10 à 12, dans lequel le boîtier, le stator, ladite plaque et ladite seconde plaque ont chacun un module d'élasticité, et **caractérisé en ce que** le module de ladite plaque et de ladite seconde plaque est inférieur au module du boîtier ou du stator.

14. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications 10 à 13, dans lequel ladite plaque ladite seconde plaque ont une courbe contrainte de compression/efforts, et **caractérisé en ce que** ladite plaque ladite seconde plaque sont comprimées dont les limites d'une telle courbe.

15. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications 10 à 14, **caractérisé en ce que** ladite seconde cavité a une seconde surface de cavité sensiblement parallèle à la seconde surface du rotor, ladite seconde plaque ayant une seconde surface de plaque, ladite seconde surface de plaque chevauchant ladite seconde surface de cavité, et au moins une parmi ladite seconde surface de plaque ou ladite seconde surface de cavité étant réduite pour fournir une aire de surface de contact réduite entre celles-ci.

16. Dispositif de pression hydraulique à rotor denté selon la revendication 15, **caractérisé en ce que** ladite aire de surface de contact réduite est un anneau.

17. Dispositif de pression hydraulique à rotor denté selon la revendication 16, **caractérisé en ce que** ledit anneau est sensiblement centré sur le chevauchement entre ladite seconde surface de plaque et ladite seconde surface de cavité.

18. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications précédentes, **caractérisé en ce que** ladite plaque a un extérieur, une cavité intérieure, et des moyens situés dans ladite plaque pour relier ledit extérieur à ladite cavité intérieure.

19. Dispositif de pression hydraulique à rotor denté selon la revendication 18, **caractérisé par** l'ajout d'un drain de carter et de moyens dans le boîtier pour relier ledit extérieur audit drain de carter.

20. Dispositif de pression hydraulique à rotor denté selon la revendication 19, **caractérisé par** l'ajout d'un drain de carter, et de moyens dans le boîtier pour relier ladite cavité intérieure audit drain de carter.

21. Dispositif de pression hydraulique à rotor denté selon l'une quelconque des revendications précédentes, **caractérisé par** une aire, ladite aire étant positionnée dans ladite cavité à l'extérieur de ladite plaque, ladite plaque ayant une cavité intérieure, un trou, et ledit trou étant dans ladite plaque en re-liant de manière hydraulique ladite aire à ladite cavité intérieure de ladite plaque.

22. Dispositif de pression hydraulique à rotor denté selon la revendication 21, **caractérisé par** l'ajout d'un drain de carter, ledit drain de carter étant dans ledit boîtier sensiblement à l'extérieur de ladite cavité, et ledit drain de carter étant relié premièrement à ladite aire, et ensuite audit trou.

23. Dispositif de pression hydraulique à rotor denté selon la revendication 21 aux 22, **caractérisé par** l'ajout d'un drain de carter, ledit drain de carter étant dans ledit boîtier, et ledit drain de carter étant relié premièrement à ladite cavité intérieure située dans ladite plaque, et ensuite audit trou.

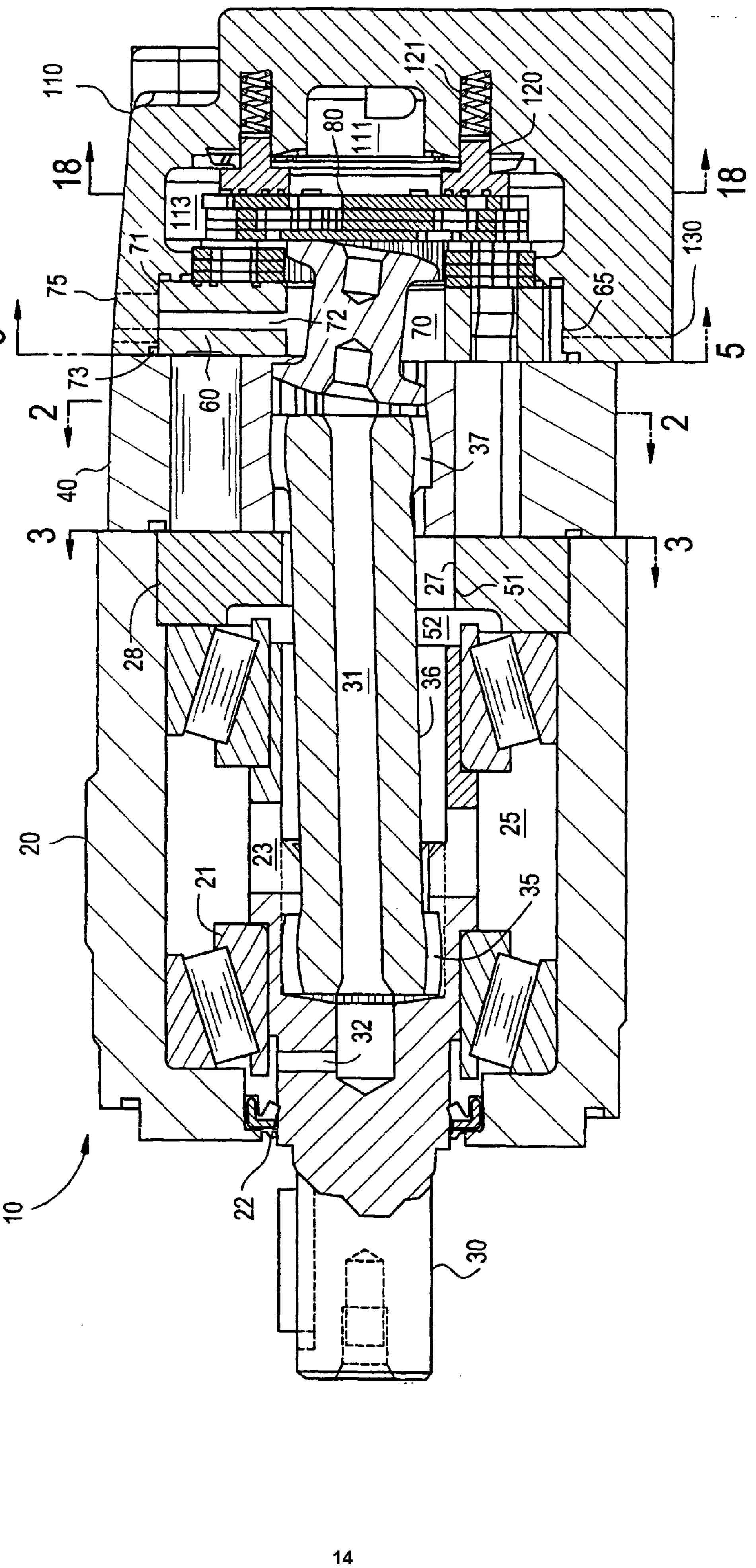
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FIG.1

FIG.2

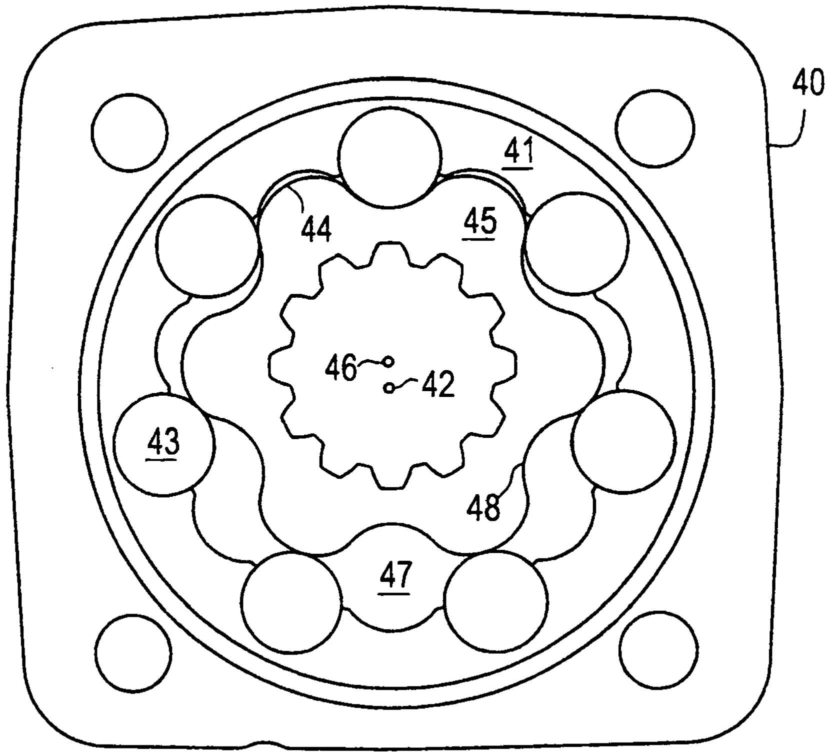


FIG.3

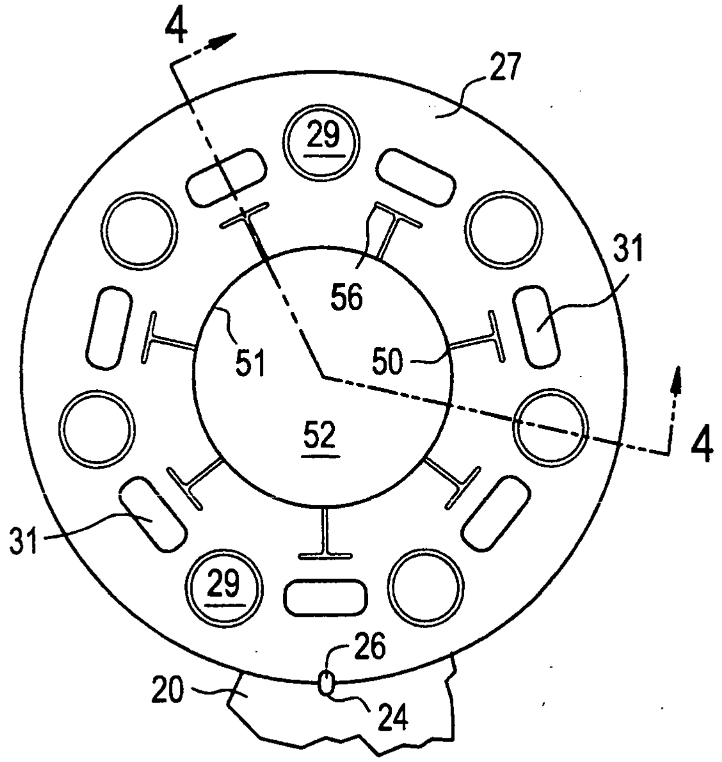


FIG.4

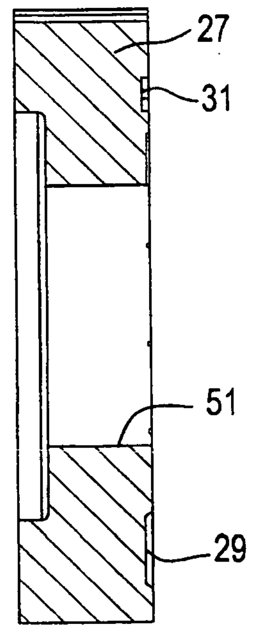


FIG.5

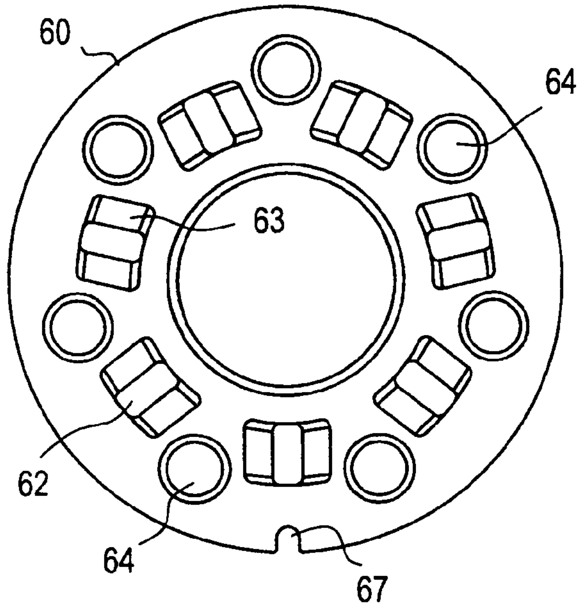


FIG.6

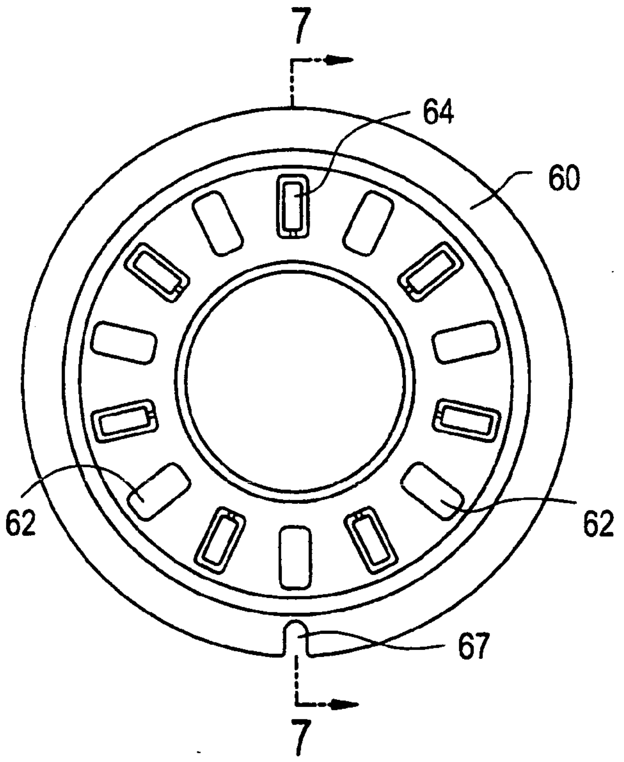


FIG.7

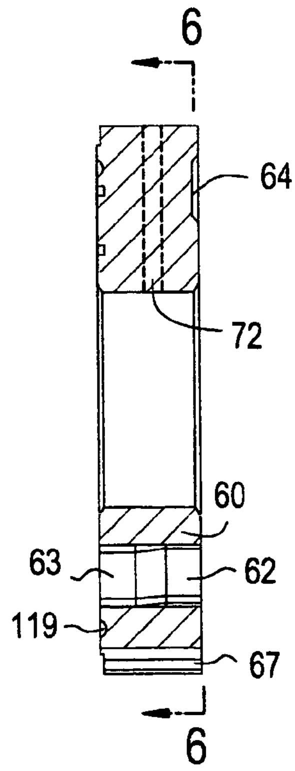


FIG.8

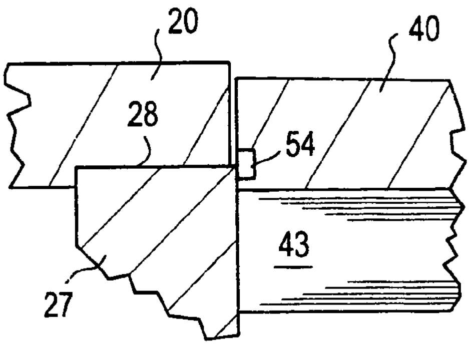


FIG.9

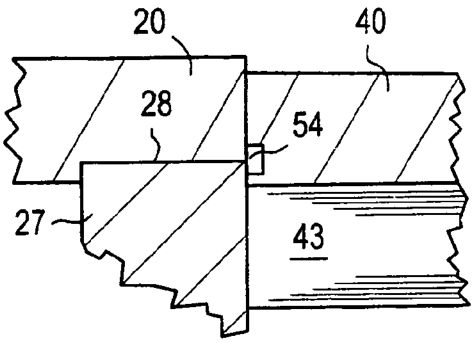


FIG.10

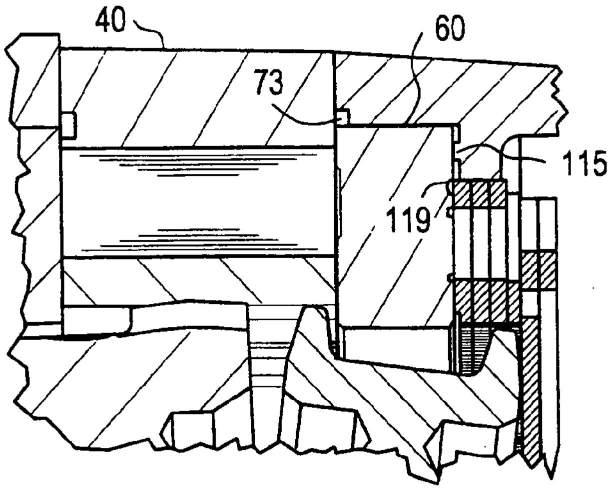


FIG.11

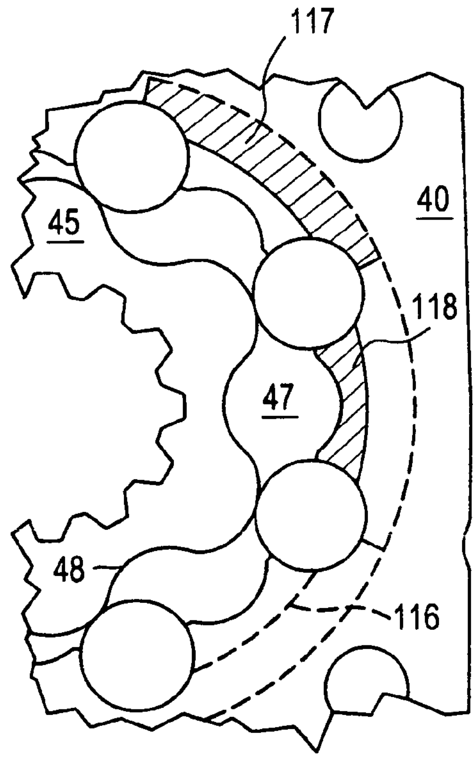


FIG.12

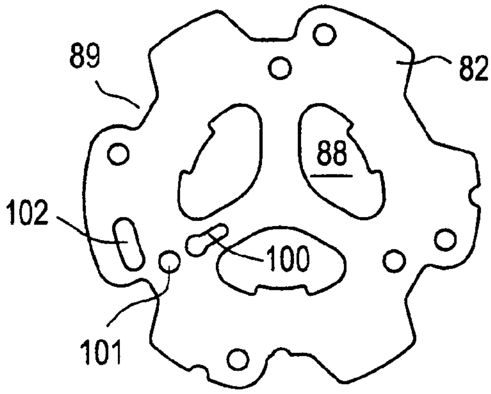


FIG.13

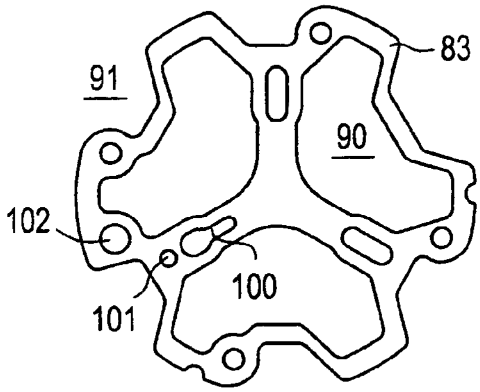


FIG.14

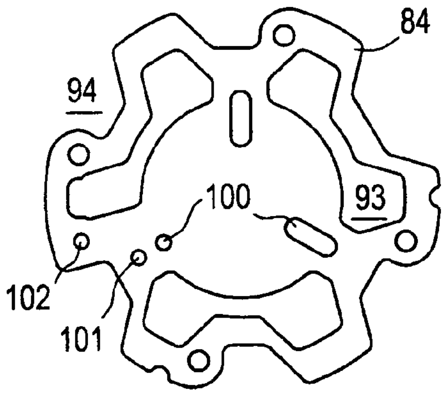


FIG.15

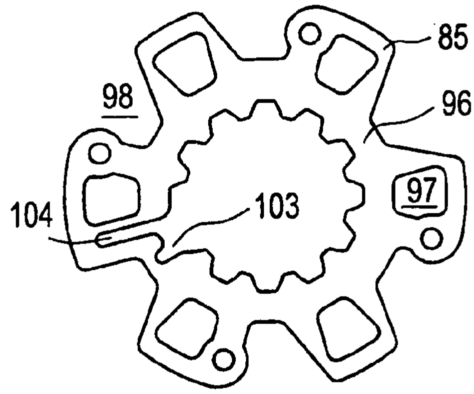
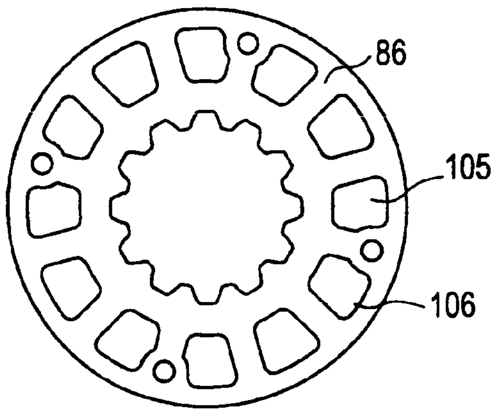


FIG.16



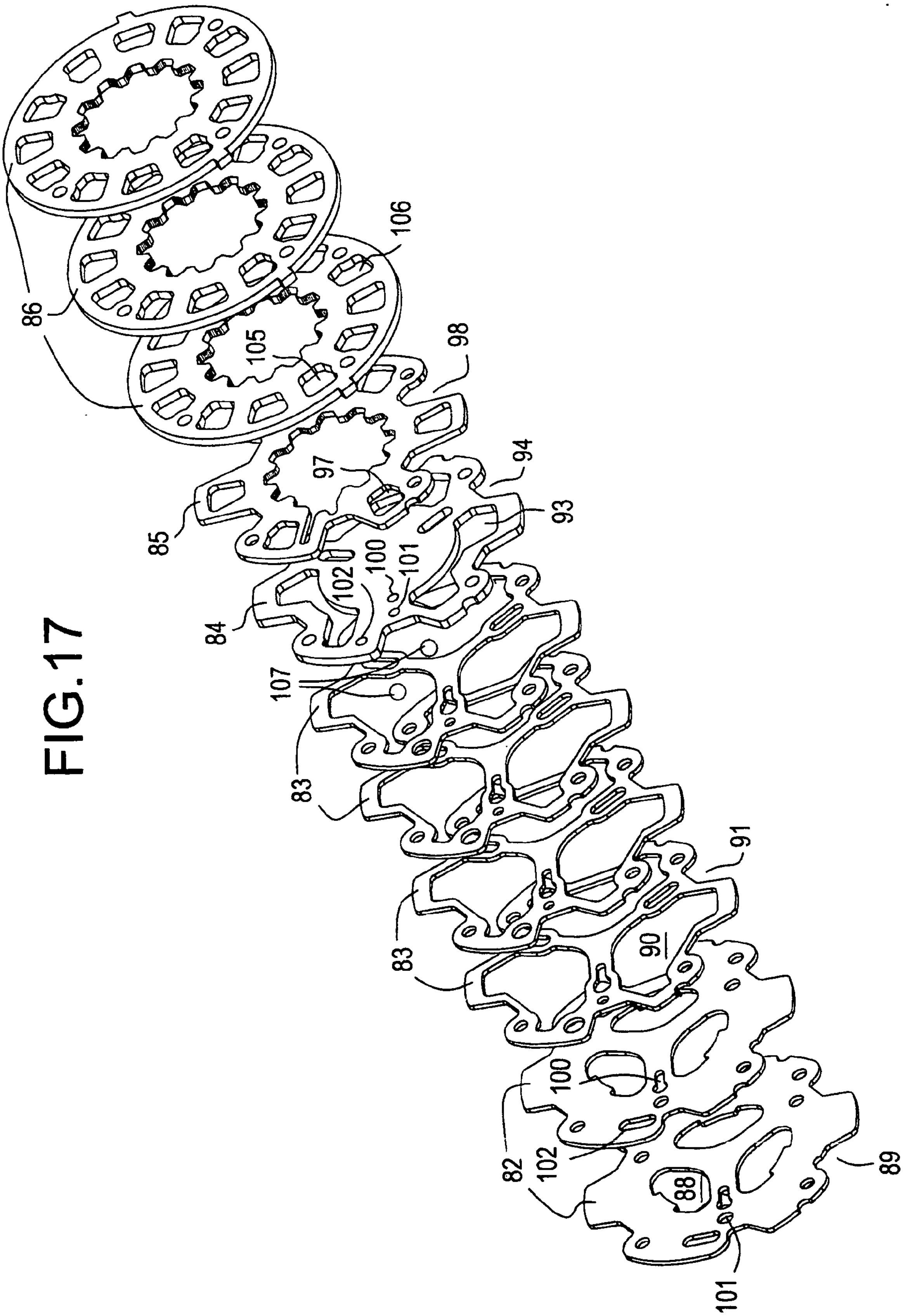

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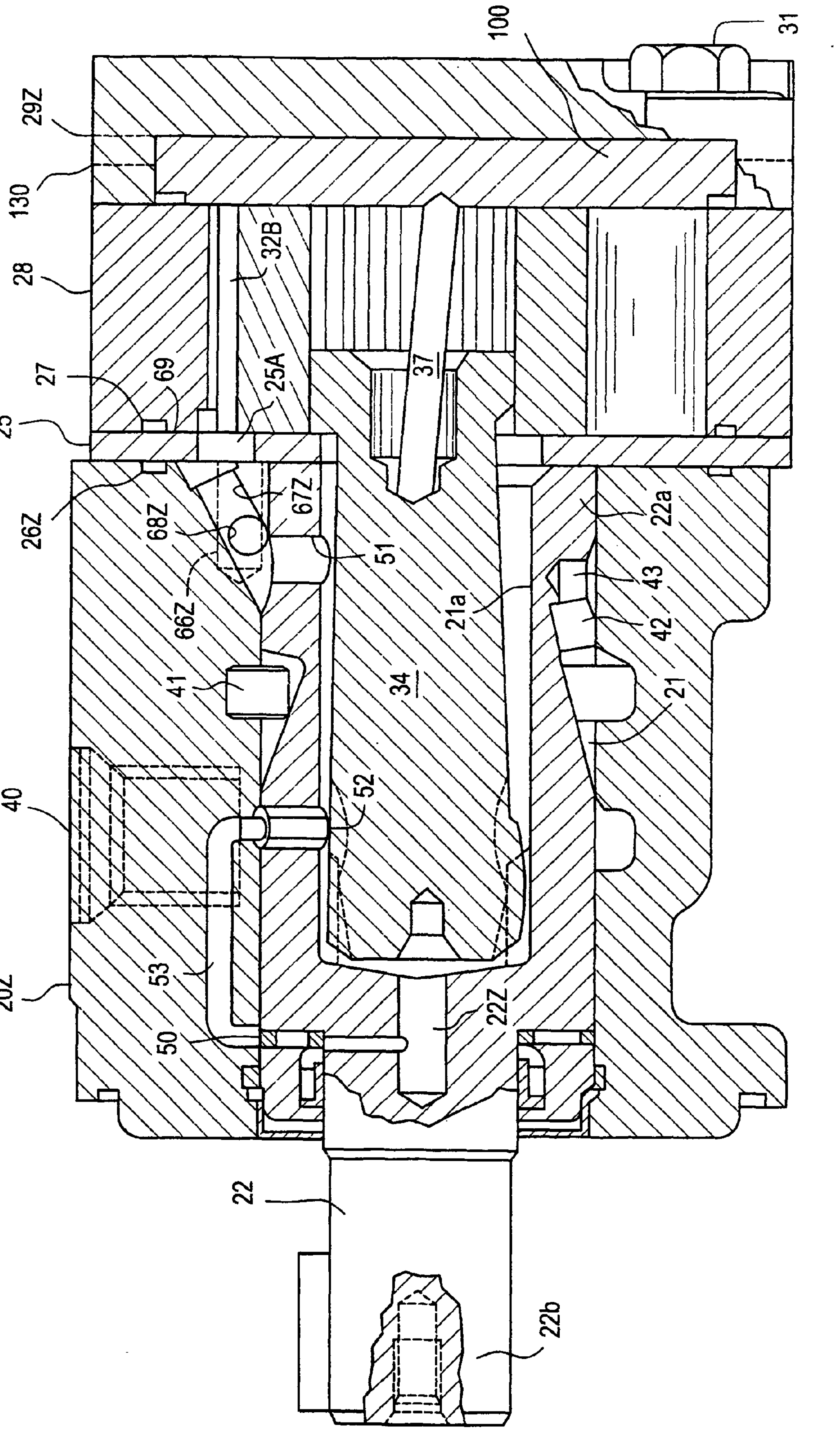
FIG.17

FIG.18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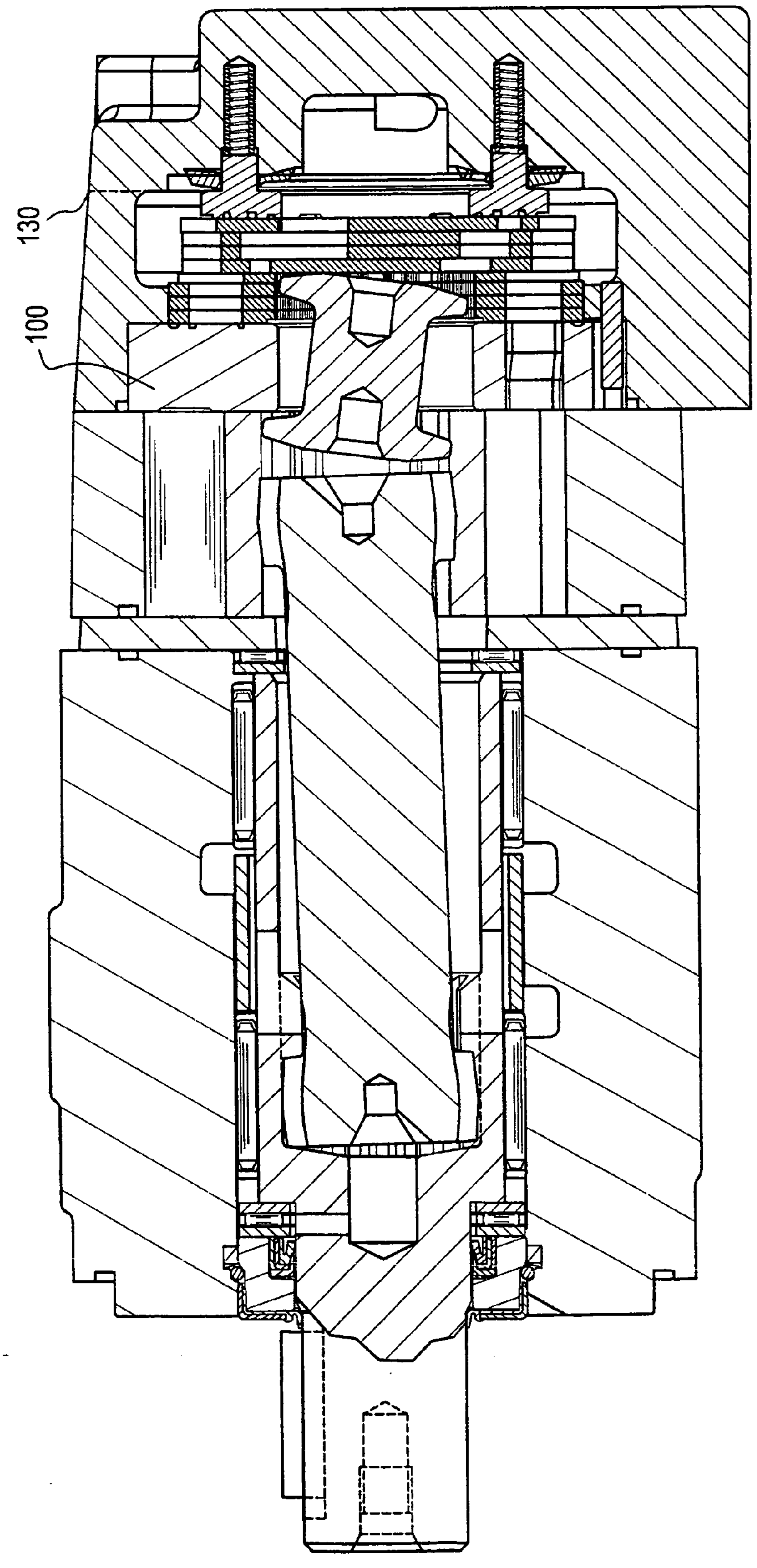
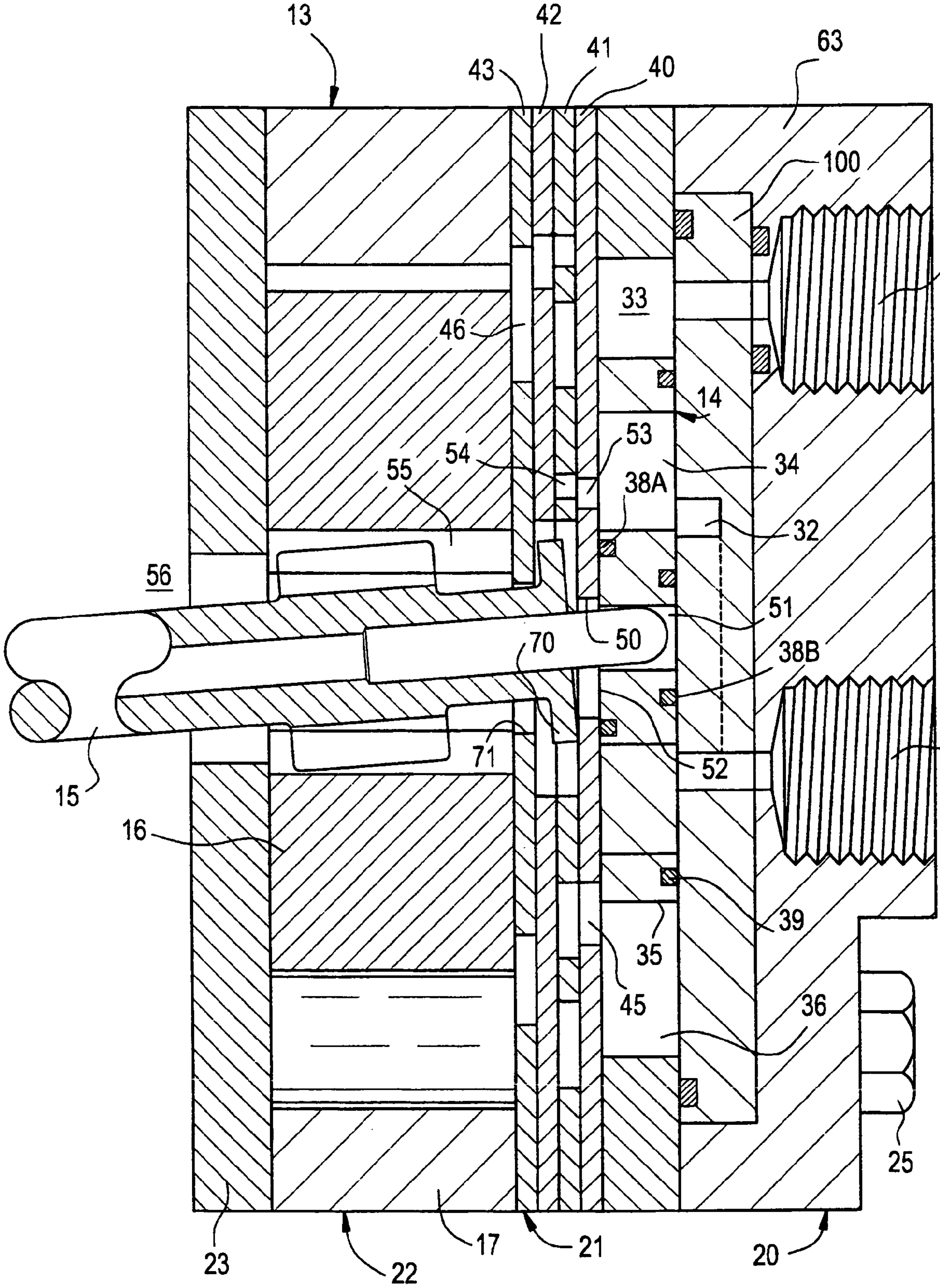


FIG.19

FIG.20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REFERENCES CITED IN THE DESCRIPTION

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