June 12, 1951 E. K. BENEDEK 2,556,717

PUMP OR MOTOR

Filed Nov. 14, 1944 7 Sheets-Sheet 1

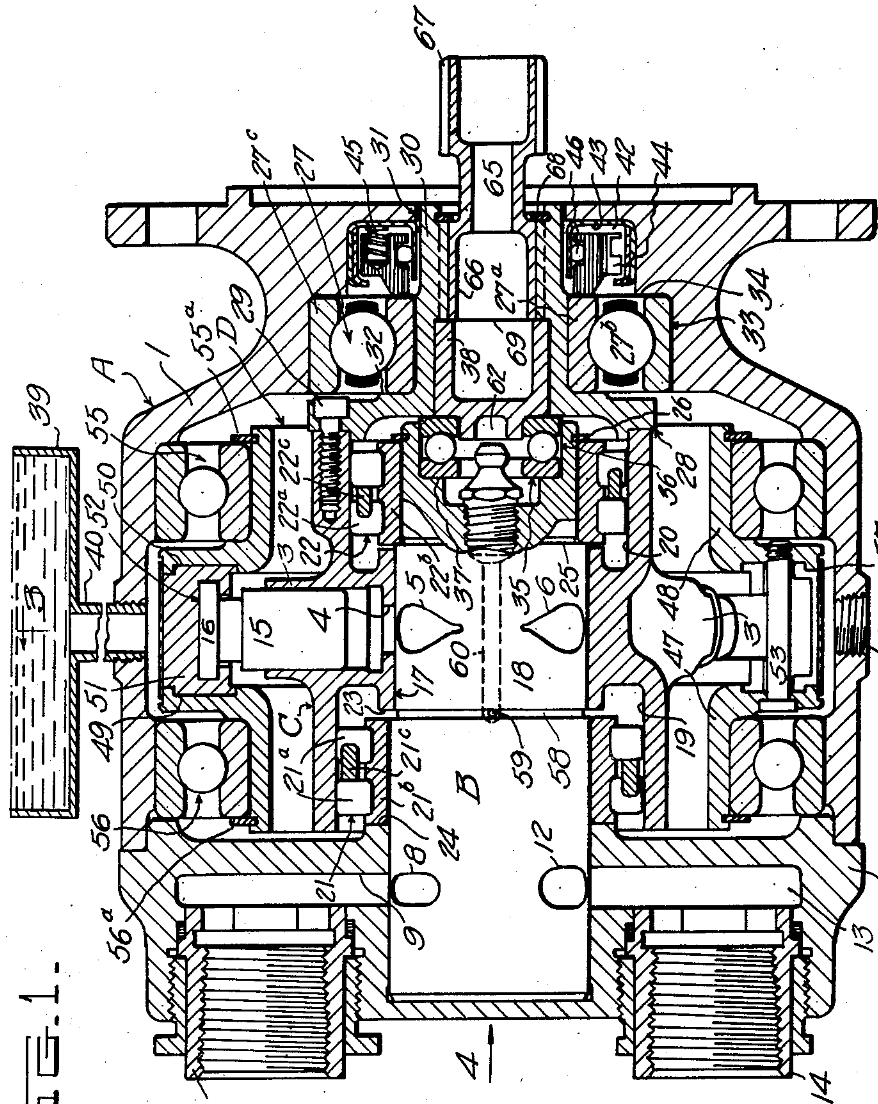

FIG. 1.

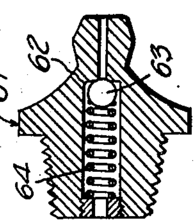

FIG. 5.

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PUMP OR MOTOR

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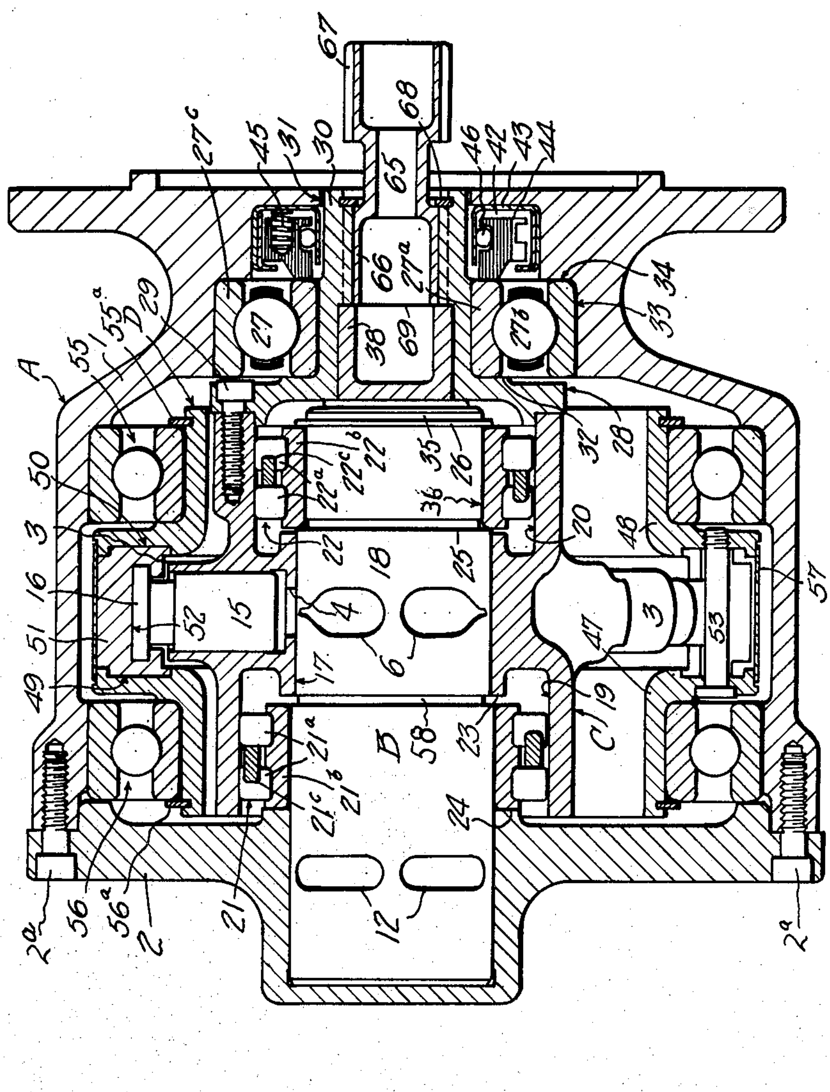

FIG. 2.

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FIG. 3.

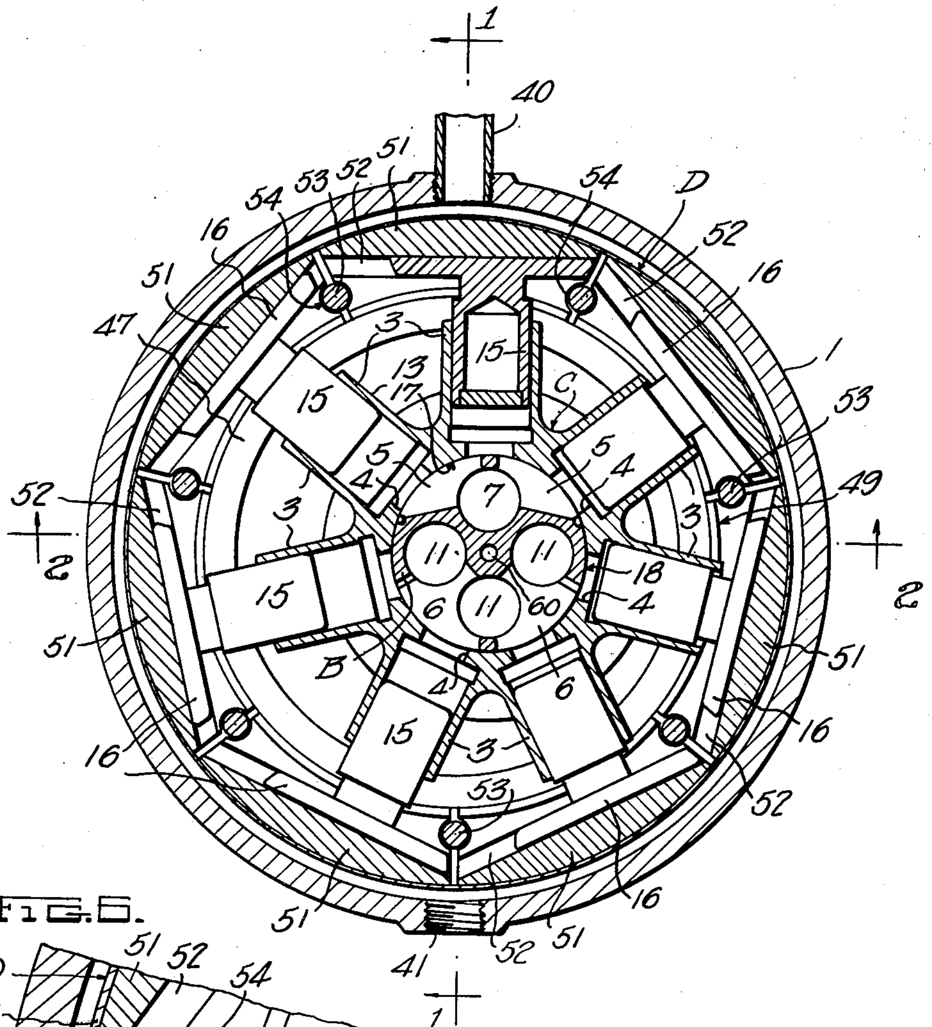

FIG. 6.

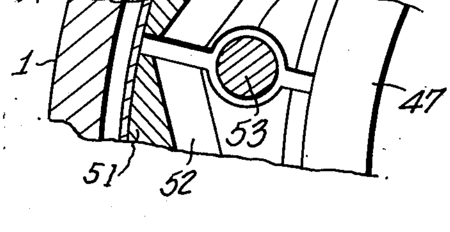

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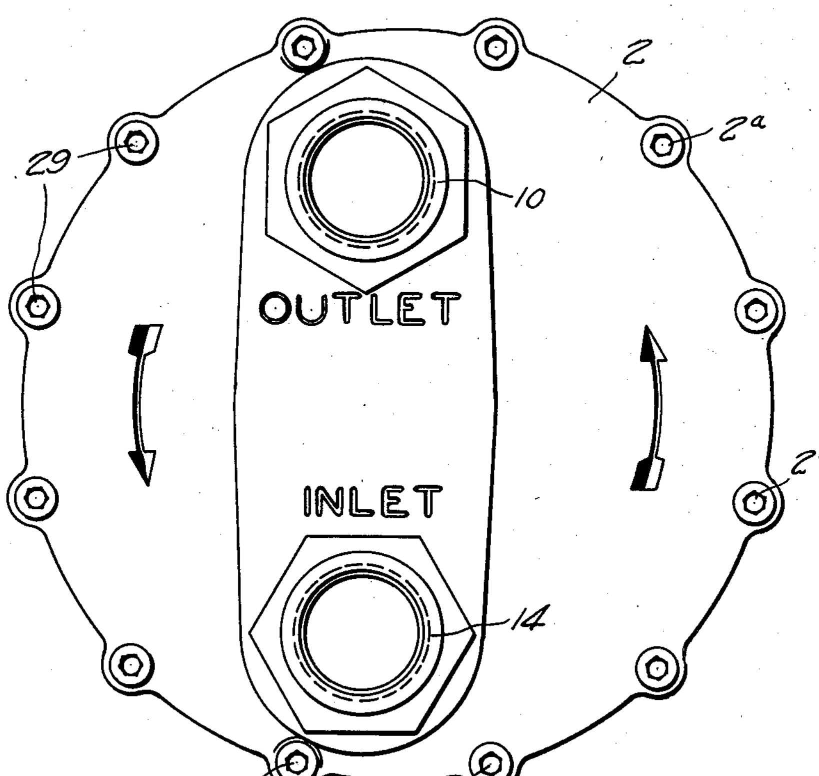
FIG. 4.



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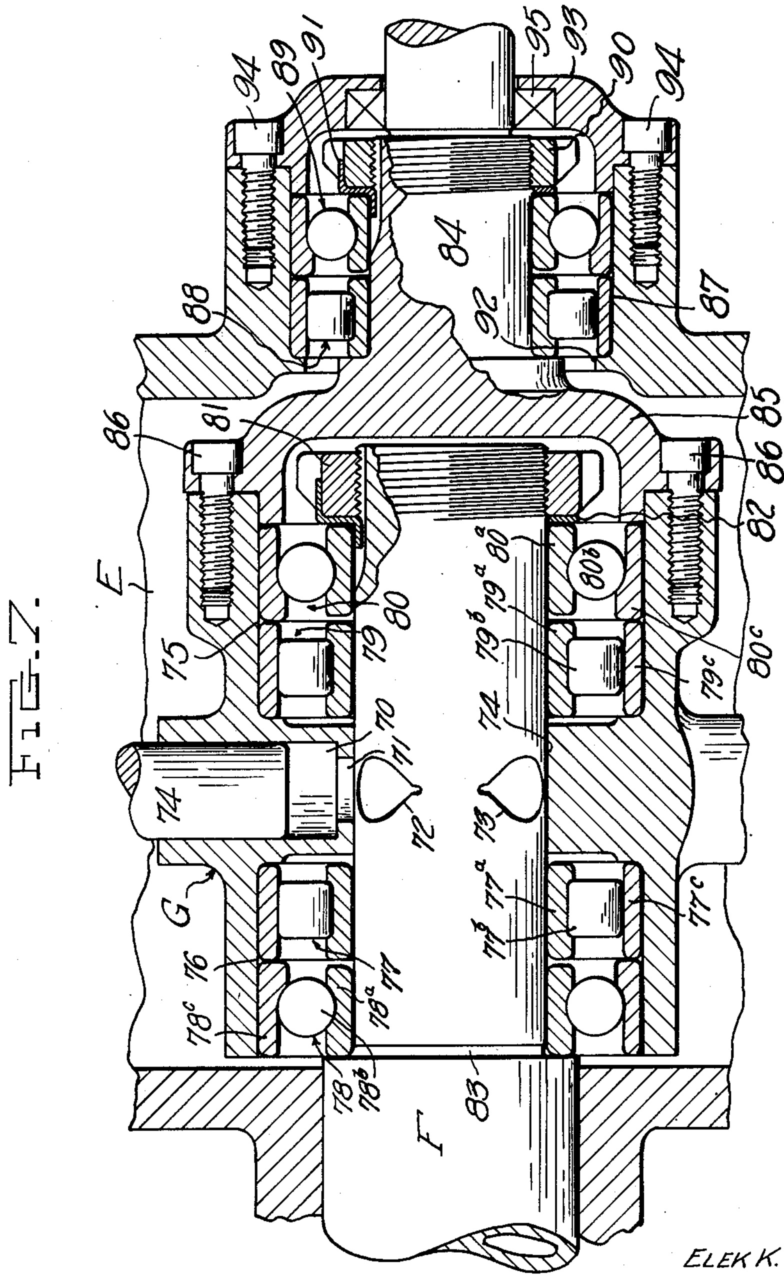

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PUMP OR MOTOR

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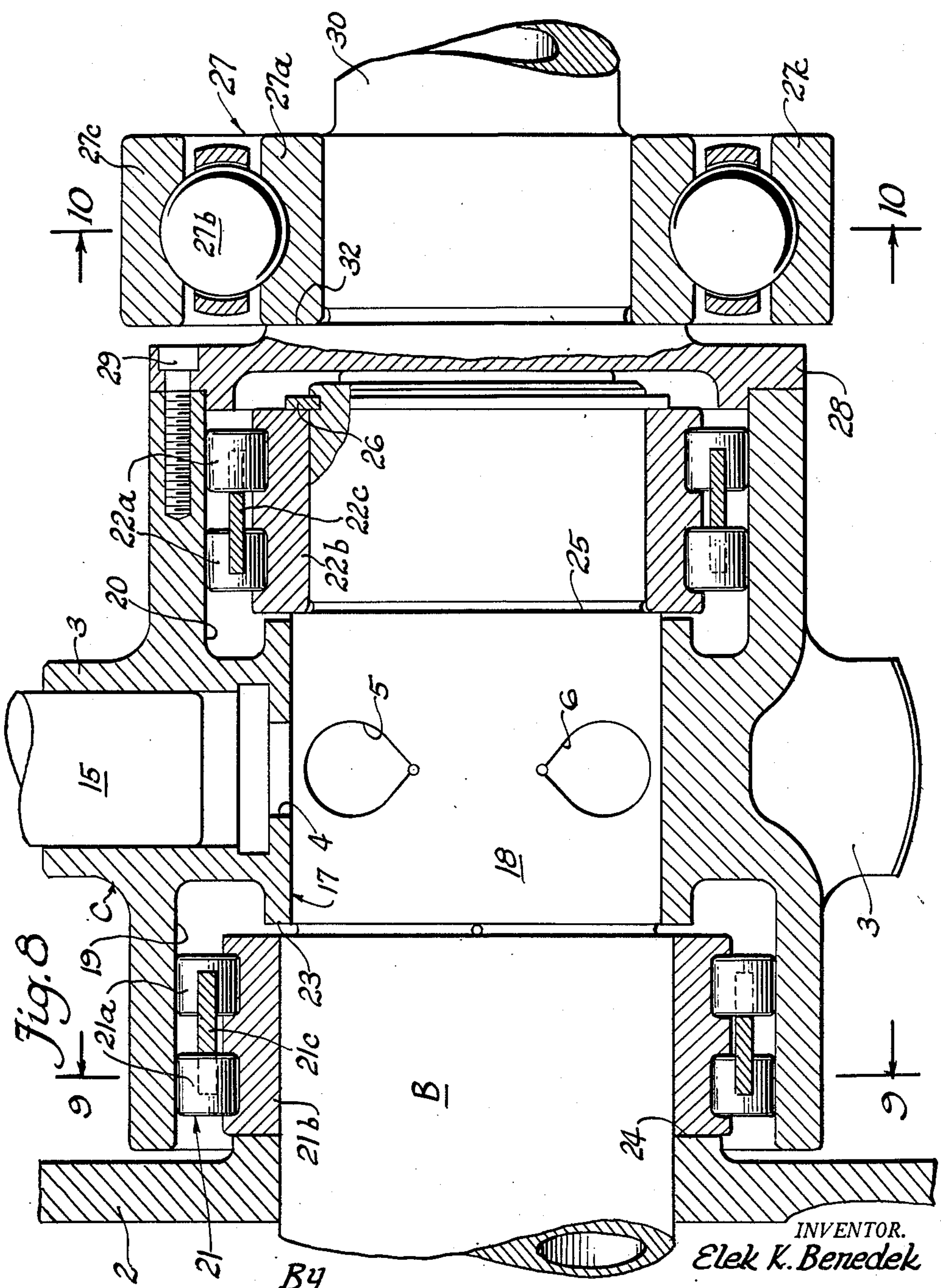

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PUMP OR MOTOR

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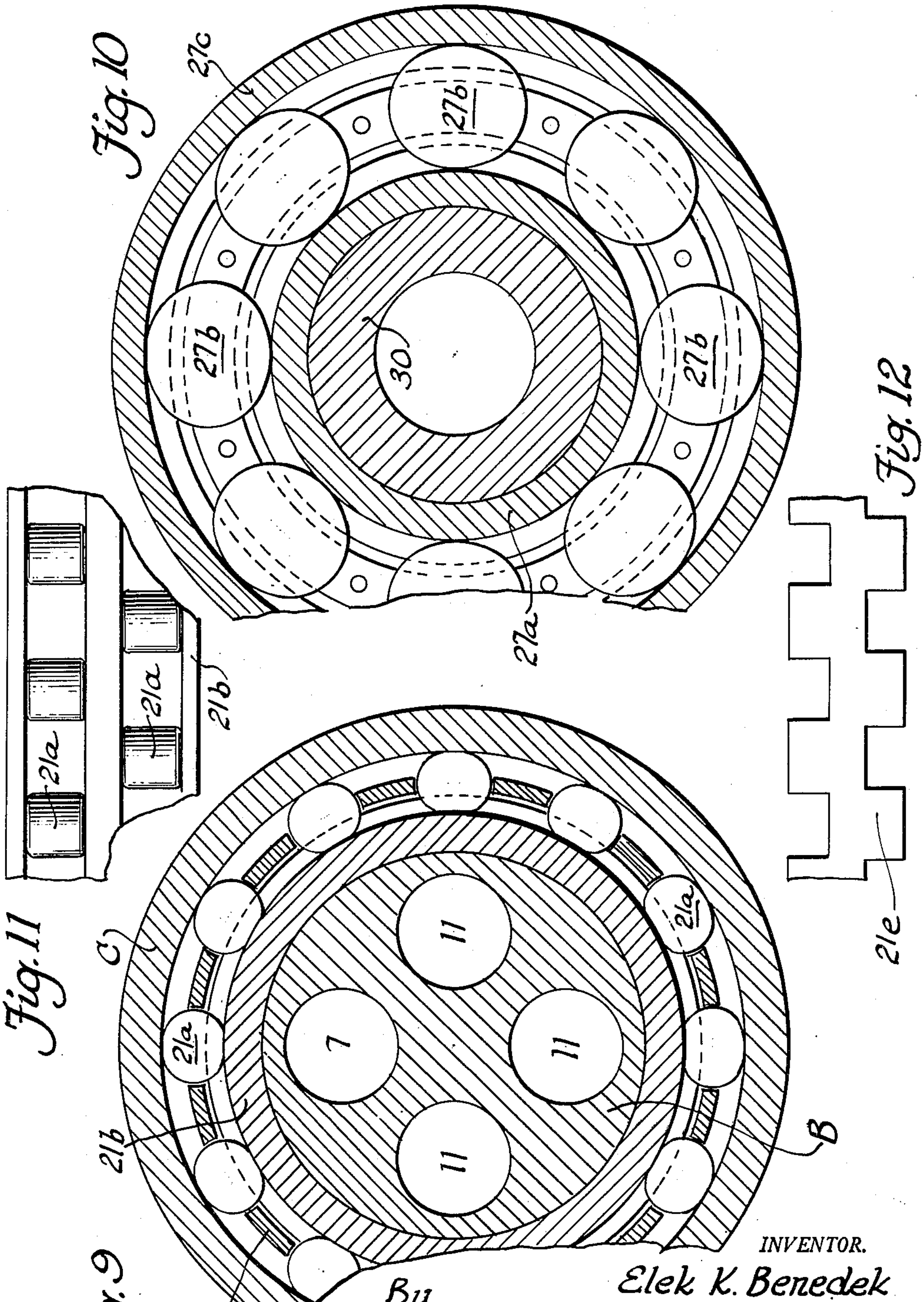

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UNITED STATES PATENT OFFICE

2,556,717

PUMP OR MOTOR

Elek K. Benedek, Chicago, Ill.

Application November 14, 1944, Serial No. 563,353

7 Claims. (Cl. 103—161)

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This invention relates to pumps or motors and more particularly to hydraulic pumps or motors of the kind including a cylinder barrel rotatable about a fixed pintle and in which pistons reciprocable in circumferentially spaced radial cylinders on the barrel are operated by thrust-transmitting means engageable with a reactance member or assembly eccentric to the pintle axis.

An object of the invention is to provide structural improvements in a pump or motor of the class referred to by which the machine is made capable of operating with increased reliability and efficiency and at very high working pressures.

Another object of the invention is to provide a pump or motor having these desirable characteristics and which is of light, compact construction rendering it especially suitable for exacting gruelling service, such as for operating aircraft controls.

Another object of the invention is to provide such a pump or motor apparatus which is constructed to enable the high speed working parts to be lubricated more efficiently than heretofore, so as to make very high operating speeds safe, thereby enabling a small compact unit to carry large loads.

Another object of the invention is to provide an improved preloaded bearing mounting for rotating parts of such machines.

Another object of the invention is to provide an improved drive coupling arrangement for a pump or motor of the kind referred to.

A further object of the invention is to provide an improved reactance assembly. Other objects will become apparent from a reading of the following detailed description, the appended claims, and the accompanying drawings, in which:

Figure 1 is a longitudinal section of a hydraulic pump or motor embodying the invention, the section being taken on the line 1—1 of Figure 3;

Figure 2 is a longitudinal section of the pump or motor shown in Figure 1, the section being taken on the line 2—2 of Figure 3, with rotary parts shown as being angularly displaced from the positions shown in Figure 3;

Figure 3 is a transverse section on the line 3—3 of Figure 1;

Figure 4 is an end elevation looking in the direction of the arrow 4 in Figure 1;

Figure 5 is a detail longitudinal sectional view of a check valve drawn on an enlarged scale;

Figure 6 is a fragmentary detail section of part of a reactance rotor showing particularly

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the relation of a pair of adjacent thrust blocks to a clamping bolt;

Figure 7 is a fragmentary longitudinal section of a modified arrangement for journaling a rotatable cylinder barrel;

Figure 8 is an enlarged copy of Fig. 1 showing only the preloaded bearing mounting of the pintle and barrel, and the preloaded bearing of the impeller shaft;

Figure 9 is a section through line 9—9 of the larger pintle bearings of Fig. 8, showing the deformation of the rolling element slightly exaggerated to illustrate the preload on the rolling elements in the direction of their rotation;

Figure 10 is a transverse section taken along line 10—10 of Fig. 8 showing the deformation of the preloaded antifriction elements slightly exaggerated for clearness of illustration;

Figure 11 shows a fragmentary portion of one of the bearings shown in Fig. 9 with the cage removed, and with the staggered, double row straight roller elements, and

Figure 12 shows a fraction of the staggered cage element of the bearings shown in Fig. 11.

Structures embodying the invention may be adapted to operate either as pumps or motors. In the following detailed description of representative embodiments of the inventive structure reference will be made to its functioning when operating as a pump, but it will be understood that like other structures of the same general class it may operate reversely, that is as a hydraulic motor when supplied with working fluid under pressure.

Throughout the specifications of this invention, the word or term "preloaded bearing or bearings" is intended to comprehend such antifriction ball or roller bearing structure in which in assembly the races, which are the relatively rotating bearing members, as well as the balls or rollers which are the load transmitting rolling elements, are under positive preload. This preload is obtained in assembly by designing the relatively rotating races in such relation with respect to the inner and outer bearing housing members, that by press-fitting the races in their respective housing the space defined therebetween for the anti-friction rolling elements will be less radially than the radial dimensions of the rolling elements. By this simple provision and without any extra member or costs the whole bearing and in it, its rolling elements will be positively preloaded and all radial looseness from the bearing itself, eliminated.

The embodiment shown in Figures 1 to 6 inclusive comprises a pump casing generally designated A and including a cylindrical body portion 1 and a manifold cover plate 2 mounted at the open end of the body 1 and clamped and held in assembled relation to the body 1 by bolts 2ª or other suitable means. The end cover 2 supports a fixed pintle B which extends into the casing and is surrounded by a rotatable cylinder barrel member C provided with a plurality of circumferentially spaced radial cylinders 3. Cylinder ports 4 communicating with the cylinders 3 are adapted to cooperate in a known manner with pintle discharge ports 5 and intake ports 6. The pintle discharge ports 5 communicate with a discharge passage 7 extending longitudinally within the pintle to a radial discharge passage 8 in constant communication with a chamber 9 in the manifold plate 2, the chamber 9 being adapted to communicate with exterior piping fitted to a connection 10. Similarly, the pintle intake ports 6 communicate with intake passages 11 extending longitudinally through the pintle and being connected to radial passages 12 which communicate with a manifold intake chamber 13 adapted to communicate with outside piping through a connection 14.

Pistons 15 mounted for reciprocation in the cylinders 3 respectively are formed or equipped with tangential cross heads or thrust-transmitting connections 16 adapted to have rotary drive and thrust-transmitting connection with a reactance ring member or assembly D, the construction of which will be explained later. For the present it is sufficient to state that the reactance ring assembly D is mounted for rotation about an axis eccentric to the common axis of the pintle and cylinder barrel.

In operation, the cylinder barrel C is rotated about the pintle B whereby the pistons 15 and the cross heads 16 impart rotary drive to the reactance ring assembly D and, due to the eccentricity of the assembly D, the pistons are reciprocated. The registration of the cylinder ports 4 with the pintle ports 5 and 6 is so timed with respect to the in-and-out travel of the pistons that fluid is drawn in to the cylinders during outward strokes of the pistons and is forced from the cylinders when the pistons move in. This mode of operation as generally described thus far is well known to those familiar with pumps or motors of the rotary, radial cylinder, eccentric reactance class.

In accordance with one feature of the present invention, the cylinder barrel C is journaled in a new and improved way by preloaded antifriction bearings which hold the barrel so accurately aligned with respect to the pintle as to permit operation under very high working pressures without undue slip fluid loss and with minimum wearing of the parts under heavy load operating conditions. In the form shown in Figures 1 to 6 and 8 to 12 the cylinder barrel C is formed with a central bore portion 17 in the region of the cylinders 3. The bore 17 fits closely over the central or adjacent pintle surface 18 with small operating clearance insufficient to permit substantial slippage of fluid under pressures as high as 3000 pounds per square inch. On opposite sides of the barrel bore 17 are barrel end bore portions 19 and 20 of somewhat larger diameter than the bore 17 so as to be spaced radially from the pintle portions which they respectively surround. An antifriction bearing assembly 21 including rolling elements 21ª, an inner race 21ᵇ, and a spacer ring or cage 21ᶜ is interposed in the annular space between the barrel bore 19 and the pintle B. Similarly an antifriction bearing assembly 22 comprising rolling elements 22ª, an inner race 22ᵇ, and a spacer ring or cage 22ᶜ is interposed between the barrel bore 20 and the outer end portion of the pintle. The races 21ᵇ and 22ᵇ are recessed to provide depressed annular trackways for the rollers 21ª and 22ª. No separate outer races are provided in the embodiment shown, but it shall be understood that I do not limit my invention to this embodiment alone, but may use any standard modification of these external two row straight roller bearings, such as a complete bearing having both inner and outer races or a so-called inverted form wherein the rollers are carried by an outside race and bear directly and without an inner race on the pintle. As shown best in Fig. 8, the rollers 21ª and 22ª instead engage the bores 19 and 20 directly and the bores 19 and 20, which thus serve as race-ways, are smoothly finished and hardened so as to have all of the properties required for a bearing race.

Anti-friction straight roller bearing assemblies 21 and 22, respectively, when preloaded by radial compression forces of the associated pintle B and barrel flange C, respectively, will be deformed as in Fig. 9 to the extent of the maximum rated pump load in such a manner that irrespective of the instantaneous load or operating pressure of the pump, all rolling anti-friction elements will be loaded to the extent of the maximum rated load of the pump, thus as shown in Fig. 8, they will be compressed radially and assume slightly deformed egg or elliptical form in the direction of their rotation. Due to this precompression of the rollers and associated race members of bearings 21 and 22, the operating clearance between pintle and barrel not only can be kept down to a practical minimum and up to the maximum operating pressure of the pump, but this operating clearance can be stabilized and maintained during the useful life of said preloaded roller bearings 21 and 22. Such greatly reduced and evenly distributed running clearance between pintle and barrel will result in reduced slip and reduced mechanical friction.

The bearings 21 and 22 are preloaded radially in order to obtain more stability of the operating clearance between the pintle midsection 18 and the barrel bore 17. The preloading is imposed on the bearings by forcing the sets of rollers and their respectively associated inner races on the pintle to provide a press fit of the races on the pintle. The races are machined to inside diameters slightly smaller than the normal diameters of the cooperating pintle portions so that when the races are pressed upon the pintle the races are expanded or forced radially outwardly, thus forcing the associated rollers 21ª and 22ª radially outwardly against the barrel bores 19 and 20. On the other hand, barrel bores 19 and 20 are so machined that their respective inside diameters will be less than the expanded outside diameters of bearings 21 and 22, with a predetermined amount. The rollers 21ª and 22ª thus are assembled under radial compression from the inside as well as from the outside thereby providing the desired preload condition.

Inasmuch as the inner bearing race 21ᵇ is press fitted on the pintle, it is not necessary to hold it in fixed axial position, but if there should be any tendency for it to shift longitudinally of the pintle, its shifting movement would be limited by a shoulder 23 on the cylinder barrel and a shoulder 24 on the end cover 2. The race 22ᵇ is held against shifting movement by a shoulder 25 on the pintle and a snap ring 26.

The barrel C also is journaled externally at one end by a preloaded antifriction bearing 27 interposed between the casing body 1 and a barrel end part 28 made separate from but having a flange fixed to the barrel C by bolts 29. The part 28 includes a shaft part or sleeve extension 30 which projects into an opening 31 in the casing body 1 coaxial with the pintle B. The preloaded bearing 27 includes an inner race member 27^a, the left side face of which abuts and has fluid tight engagement with a radial shoulder 32 on the part 28. The race 27^a is grooved to receive ball rollers 27^b which also are received by a grooved outer race 27^c mounted in a counter bore 33 formed in the casing body 1. In opposed relation to and axially spaced from the radial shoulder 32 is another shoulder 34 which abuts the right hand outer face of the bearing race 27^c.

After bearing 27 is preloaded, either in the assembly or in the process of manufacture as shown in Fig. 10, under the balanced compressing forces of the races and their respective housing, the originally and geometrically pure spheres now become compressed to egg-shaped or ellipsoidal rolling elements (see Fig. 10) and instead of having point contact with their respective race walls, they will have a contact involving an elliptical area of substantial magnitude. Due to this preload to the approximate amount of the maximum rated load of the pump, during maximum load operation of the drive shaft 30, there will be no additional deformation in the balls 27^b of the preloaded ball bearing 27, consequently the drive shaft 30 will operate with a minimum relative eccentricity with regard to the axis of pintle and the barrel respectively. With such close eccentricity the cylinder barrel with its end flanges 19 and 20, and supporting preloading roller bearing 21 and 22 will operate with a minimum wabbling effect, thus the preload of bearings 21 and 22, and the preload of bearing 27 are closely interdependent, and the success of the one depends greatly on the success of the other. The illustration of Fig. 10 is slightly exaggerated, but actual preload in this specification is defined as one which produces positive precompression of the rolling elements of a preloaded bearing. Since the rolling of the balls is confined to a channel shaped race, the compression of the balls is most striking in the direction of the rotation of the bearing.

A preloaded pintle nose thrust bearing 35 is received in an opening 36 in the end of the pintle and abuts a shoulder 37. A plug 38 press fitted into the inner end of the sleeve extension 30 also abuts or bears against the nose bearing 35.

When the parts are assembled and the cover 2 and body 1 are drawn together axially, the shoulder 34 presses the race 27^c toward the left, causing the balls 27^b and the inner race 27^a also to be forced toward the left. Thus the inner race 27^a is pressed against the radial shoulder 32, forcing the cylinder barrel toward the left and causing the plug 38 to press the nose bearing 35 against the pintle shoulder 37. In this way the bearings 27 and 35 are preloaded and anchor the right hand end of the cylinder barrel against axial movement.

A further feature of the invention resides in the provision of means for flooding the interior of the casing A and the operating parts contained therein with fluid lubricant independent of the working fluid flowing to and from the cylinders. As indicated in Figure 1 a reservoir 39 located exteriorly of the pump communicates constantly with the interior of the casing A by a pipe or conduit 40. The reservoir 39 constitutes an exterior source of fluid pressure which, in the illustrated apparatus, is a static pressure due to location of the reservoir at a higher level than the pump. It will be understood that the lubricating fluid may be maintained under pressure by other suitable means. When it is desired to remove the fluid from the casing, it may be drained through an opening 41 which normally is closed by a plug (not shown).

In order to prevent the escape of fluid through the space between the sleeve 30 and the casing opening 31, the casing body 1 is formed with a seal chamber 42 in which is mounted an expansive sealing or packing device including a cage 43 and an annular sealing element 44 pressed by springs 45 into sliding or rubbing contact with the adjacent face of the bearing race 27^a which is smoothly finished and suitably hardened. The sealing element 44 may be maintained centralized by a ring of packing or the like 46 which is disposed between the cage 43 and the sealing element 44 and serves to prevent the escape of lubricating fluid from the rear of the sealing element 44. Thus it will be observed, that a balanced mechanical seal is provided, which is not only very compact, but is so constructed that a plurality of axial springs function only to press the seal nose against the polished bearing race at a constant pressure and the springs are not required to act against the hydrostatic pressure of the housing.

The rotary reactance assembly D previously referred to only generally comprises two end plate members 47 and 48 spaced from each other axially of the pintle and formed respectively with opposed circularly grooved seats 49 and 50 for receiving a plurality of thrust blocks 51 having tangential slide ways or working faces 52 adapted to accommodate the piston cross heads 16. In order that the individual thrust blocks 51 may each be aligned in exact tangential relation to its associated piston 15, the blocks are sized so as to leave a substantial clearance space between contiguous end faces of adjacent blocks. This clearance permits the blocks to be adjusted circumferentially without touching each other so that each block may be positioned in exact tangential relationship to the associated piston. After the blocks 51 have been adjusted, the end members 47 and 48 are drawn towards each other and are thus clamped against the blocks 51 by means of bolts 53. Each bolt 53 passes between contiguous end faces of two adjacent blocks 51 and in order that the bolts will not interfere with the circumferential adjustment of the blocks, the block ends are formed with cut-outs or recesses 54 through which the bolts extend with substantial clearance. The recesses 54 are sufficiently large to insure that the blocks 51 will always be free of engagement with the bolts 53.

The reactance assembly D is journaled in the casing A by antifriction bearing assemblies 55 and 56 interposed between the casing body 1 and the end members 48 and 47 respectively. The bearings 55 and 56, which are held in place by snap rings 55^a and 56^a, mount the reactance member D for rotation about an axis eccentric to the pintle axis. A shroud ring or band 57 surrounds the blocks 51 and engages the peripheral portions of the end members 47 and 48.

The bearing 35 is not flooded directly with free lubricant from the exterior source 39. In order to assure constant and effective lubrication of the bearing 35, the pintle is formed with an annular groove 58 which is always flooded with lubricant and from which lubricant flows into a radial pintle passage 59 and an axial pintle passage 60 which discharges through a relief valve 61 into a lubricant chamber 62. The valve 61 may be of conventional construction including a valve seat 62 and ball valve 63 urged against the seat by a spring 64.

In operation, fluid under pressure in the small clearance between the pintle surface 18 and the cylinder barrel bore 17 works its way into the chamber 62, maintaining the latter flooded so that when the plug 38 is rotated, the body of lubricant in the chamber 62 will be whirled around and forced against the rollers or balls of the bearing 35 to maintain the latter in a body of lubricant under pressure created by centrifugal force. Any excess pressure built up in the chamber 62 is relieved by opening of the valve 61. When the pump stops operating, the check valve 61 closes so as to maintain the supply of lubricant in the chamber 62 in readiness for lubricating the bearing 35 centrifugally immediately upon starting of the pump.

For coupling the pump to a driving motor or engine a flexible coupling member comprising a reduced stem section 65 and splined ends 66 and 67 is loosely held within the shaft part or sleeve extension 30, which is splined to receive one splined end 66 of the coupling member. A snap ring 68 retains the coupling member in place with its left hand end abutting a shoulder 69 provided by the adjacent end of the plug 38. The other splined end 67 of the coupling member is adapted to be extended into a splined sleeve or the like driven by the motor or engine.

By forming the sleeve 30 of the separate member 28 hollow throughout, broaching to form the splines or teeth is facilitated. The splined sleeve is then closed at its inner end by the press-fitted plug 38 which provides the abutment shoulder 69 for the coupling.

In the embodiment shown in Figure 7, the preloading of the cylinder barrel journals is effected partially by press fitting bearings between the cylinder barrel and the pintle and casing respectively and partially by adjustment of the ball bearing assemblies effected after the bearings have been put in place. The illustrative embodiment of this form of the invention includes a casing element E, pintle F, and cylinder barrel G formed with cylinders 70 adapted to communicate by ports 71 with pintle ports 72 and 73. Reactance mechanism similar to that previously described or of other suitable form may be provided for reciprocating pistons 74 in the cylinders 70.

The cylinder barrel element G is formed with end bores 75 and 76 which are spaced radially from the pintle element F to provide for the insertion of antifriction bearings 77 and 78 in the bore 76 and antifriction bearings 79 and 80 in the bore 75. The bearing assembly 77 comprises an inner race 77^a grooved to receive rollers 77^b which roll also on an outer race 77^c. The assembly 77 is press fitted between the barrel bore 76 and the pintle so as to place the rollers 77^b and the associated races under radial compression. The bearing 79 is similar to the bearing 77 and includes an inner race 79^a grooved to receive rollers 79^b which operate within an outer race 79^c. The bearing assembly 79 is press fitted between the pintle and the barrel bore 75.

The bearing assembly 78 is of the ball type and it is preloaded by press fitting both of its races 78^a and 78^c with a positive interference fit between the pintle F and barrel flange G. This can be achieved by making the outside diameter of the pintle greater than the bore of the race 78^a, and the inside diameter of flange G smaller than the outside diameter of race 78^c. This bearing is adapted to be preloaded by forcing one of its races axially with respect to its other race. It is shown as including an inner race 78^a, a set of balls 78^b, and an outer race 78^c. Similarly the bearing assembly 80 includes an inner race 80^a, balls 80^b, and an outer race 80^c.

Press fitting of the bearings 77 and 79 between the pintle and the associated barrel bores 76 and 75 respectively tends slightly to increase the diameter of the barrel bore walls. In order that the bearings 78 and 80 may be additionally and equally preloaded to compensate for the expansion of the barrel bores a nut 81 threaded on the free end of the pintle F is drawn against a washer 82 to press on the inner race 80^a of the bearing assembly 80. This axial pressure is transmitted through the complete bearing assembly 80, the outer bearing race 79^c, the barrel G, the outer bearing race 77^c, and the complete bearing assembly 78, the inner race 78^a of which abuts shoulder 83 of the pintle. Although this does not tend to load the bearing assemblies 77 and 79 additionally, the axial pressure imposed on the assemblies 78 and 80 does produce in them a radial preload which can be made to equal or to bear a predetermined relation to the press-fit preloading of the bearings 77 and 79.

A shaft extension 84 has a flange 85 connected to the barrel G by screws 86. The extension 84 projects through a bore 87 in the casing E and is journaled in the casing by antifriction bearing assemblies 88 and 89 interposed between the pintle and the casing bore 87. The bearing assembly 88 is similar to the assemblies 77 and 79 previously described and is preloaded by being press fitted between the pintle and the casing. The bearing assembly 89 is similar to the bearings 78 and 80 and is preloaded also by being press fitted between the drive shaft 84 and housing flange bore 87 with positive interference fit with its respective races 89^a and 89^c between the shaft 84 and the flange bore 87 respectively. This can be achieved by making the outside diameter of shaft 84 greater than the inside bore of inner race 89^a, and by making flange bore 87 less than the outside diameter of race 89^c. The preload of this bearing can be adjusted, and additionally increased by turning a nut 90 threaded on the shaft extension 84 to draw it up against a washer 91 so as to press the bearing assembly 89 to the left, which in turn presses the outer race of the bearing 88 against a casing shoulder 92. A cap 93 secured to the casing E by screws 94 is equipped with shaft packing 95.

The axially spaced and aligned three groups of preloaded combination bearing assemblies cooperate in a unique manner. The mounting of the roller and ball bearings in the barrel flanges offer specific advantages, irrespective of the order in which the roller and ball bearings are combined. If the roller bearings are mounted first and the ball next to it, the excess enlargement of the flange bore due to the preload of the roller bearing can be taken up readily by the additional preload or adjustment of the ball bearing, thus both kind of bearings, the ball and the roller, can be preloaded substantially to the same amount of their respective capacities. The roller bearing will be the principal agent of radial load, while the ball bearing though it will help in radial load, will be the principal agent of axial positioning and thrust. If on the other hand, the order of the mounting is reversed, during preloading, in the same barrel bore, the ball bearing will be less preloaded than the following roller bearing, and automatically the proportion of preload will be in the ratio of the respective capacities of the two different types of bearings, which is a great perfection in the bearing combination and an advantage at the same time.

The same is true for bearings 88 and 89. Depending on the assignment of the pump, either the roller or the ball bearing will be assembled first. If the thrust force is toward the left, the present order is the proper one. If the dominating thrust force points to the right, the ball bearing should come first, and contact with its thrust shoulder the fixed outer race of the roller bearing. The mounting can be obtained by the complete reversal of the two bearings as one unit.

The apparatus disclosed by way of example embodies the invention in the preferred form, but it will be apparent that changes may be made in the particular construction and relative arrangement of the illustrated parts without departing from the invention as defined in the claims.

I claim:

1. In a pump or motor, a casing; a cylinder barrel provided with a plurality of circumferentially spaced radial cylinders; means mounting said cylinder barrel for rotation including an antifriction bearing interposed between said casing and an end part of said cylinder barrel and including an inner race member and associated rolling elements, said end part supporting said inner race and having a radial shoulder against which the inner face of said race abuts; means supplying lubricating fluid under pressure to the interior of said casing to flood said bearing; a seal chamber in said casing adjacent the outer face of said race; and a balanced mechanical seal in said chamber including an annular element pressing axially into sealing engagement with said race outer face and also including packing means preventing leakage between said annual element and said casing.

2. In a pump or motor, a casing; a cylinder barrel provided with a plurality of circumferentially spaced radial cylinders; means mounting said cylinder barrel for rotation including an antifriction bearing interposed between said casing and an end part of said cylinder barrel and including an inner race member and associated rolling elements, said race member being mounted in fluid tight engagement with said end part; means supplying lubricating fluid under pressure to the interior of said casing to flood said bearing; a seal chamber in said casing adjacent the outer face of said race; and a balanced mechanical seal in said chamber including an annular element pressing axially into sealing engagement with said race outer face and also including packing means preventing leakage between said annual element and said casing.

3. In a pump or motor, a casing; a pintle mounted in said casing; a cylinder barrel journalled for rotation on said pintle and being provided with a plurality of circumferentially spaced radial cylinders; an opening in one end of said casing opposite the adjacent end of said cylinder barrel; a shaft part fixed with respect to and extending from said end of said cylinder barrel and into said opening and having a radial shoulder adjacent said barrel end; a counter bore in said casing surrounding said shaft part and terminating in an opposed shoulder facing towards said barrel end and being spaced outwardly from said radial shoulder; an antifriction bearing comprising rolling elements and inner and outer races, the inner race fitting over said shaft part and abutting said radial shoulder, and the outer race fitting in said counter bore and abutting said opposed shoulder; a seal chamber in said casing outwardly beyond said antifriction bearing; and a balanced mechanical seal in said chamber including an annular element pressing axially into sealing engagement with the outer face of said inner race.

4. In a pump or motor, a casing; a pintle mounted in the casing; a cylinder barrel rotatable about said pintle and having a bore surrounding said pintle with working clearance between the pintle and the bore wall; ports in said pintle; cooperating ports in said cylinder barrel; a closed chamber between said cylinder barrel and one end portion of said pintle; bearings in said chamber and operatively interposed between said pintle and said cylinder barrel; a lubricant passage extending through a part of said pintle and communicating at its opposite ends respectively with said chamber and with a source of lubricant; a relief valve in said passage-way for retaining fluid lubricant in said chamber but being operable by excess fluid pressure in said chamber for permitting lubricant to flow from said chamber into said passage-way.

5. In a pump or motor, a casing; a pintle mounted in the casing; a cylinder barrel rotatable about said pintle and having a bore surrounding said pintle with working clearance between the pintle and the bore wall; ports in said pintle; cooperating ports in said cylinder barrel; a closed chamber between said cylinder barrel and one end portion of said pintle; bearings in said chamber and operatively interposed between said pintle and said cylinder barrel; a lubricant passage extending through a part of said pintle and communicating at its opposite ends respectively with said chamber and with the interior of said casing; means for connecting the interior of said casing to a source of fluid lubricant under pressure; a relief valve in said passage-way for retaining fluid lubricant in said chamber but being operable by fluid pressure in said chamber for permitting lubricant to flow from said chamber into said passage-way to thereby control the pressure of said closed chamber and of said working clearance between said pintle and said bore wall of said barrel.

6. In a pump or motor, a casing; a pintle mounted in said casing; a cylinder barrel rotatable about said pintle and being provided with a plurality of circumferentially spaced radial cylinders; a separate member attached to said barrel at one end thereof and being formed with a sleeve extending axially from said cylinder barrel, the outer end of said sleeve being internally splined to receive a splined coupling; a plug fitted tightly in the inner end of said sleeve and providing an abutment for the inner end of the coupling; pistons respectively reciprocable in said cylinders; reactance and thrust means for reciprocating said pistons in response to rotation of said barrel and cylinders; and means for valving flow of fluid to and from said cylinders.

7. In a pump or motor, a casing, a pintle mounted in said casing; a cylinder barrel ro-

tatable about said pintle and being provided with a plurality of circumferentially spaced radial cylinders; a separate member attached to said barrel at one end thereof and being formed with a sleeve extending axially from said cylinder barrel, the outer end of said sleeve being internally splined to receive a splined coupling; a plug fitted tightly in the inner end of said sleeve and providing an abutment for the inner end of said sleeve; an antifriction thrust bearing interposed between said plug and the adjacent end of said pintle; pistons respectively reciprocable in said cylinders; reactance and thrust means for reciprocating said pistons in response to rotation of said barrel and cylinders; and means for valving flow of fluid to and from said cylinders.

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