Dec. 7, 1937. E. K. BENEDEK 2,101,730

HIGH PRESSURE HYDRAULIC PUMP OR MOTOR

Filed June 20, 1935 3 Sheets-Sheet 1

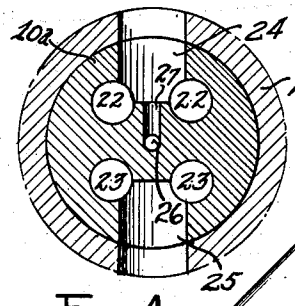


FIG. 4.

FIG. 1.

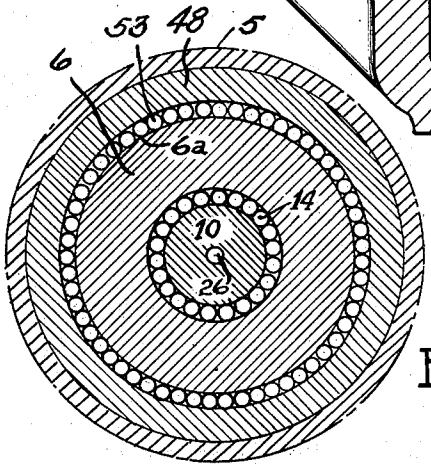


FIG. 3.

Inventor

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Dec. 7, 1937. E. K. BENEDEK 2,101,730

HIGH PRESSURE HYDRAULIC PUMP OR MOTOR

Filed June 20, 1935 3 Sheets-Sheet 2

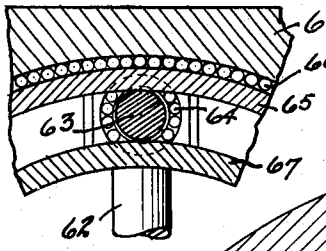


FIG. 6.

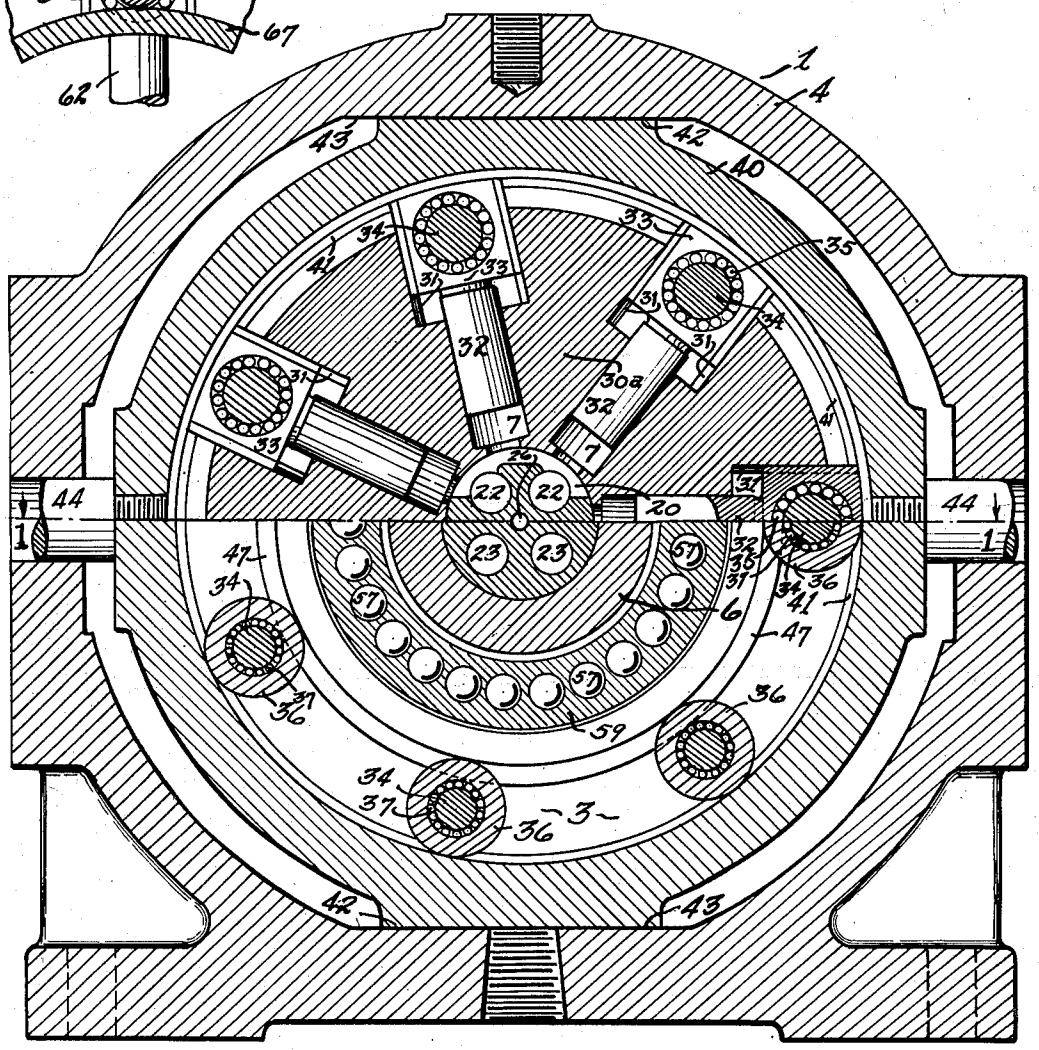


FIG. 2.

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Dec. 7, 1937. E. K. BENEDEK 2,101,730

HIGH PRESSURE HYDRAULIC PUMP OR MOTOR

Filed June 20, 1935 3 Sheets-Sheet 3

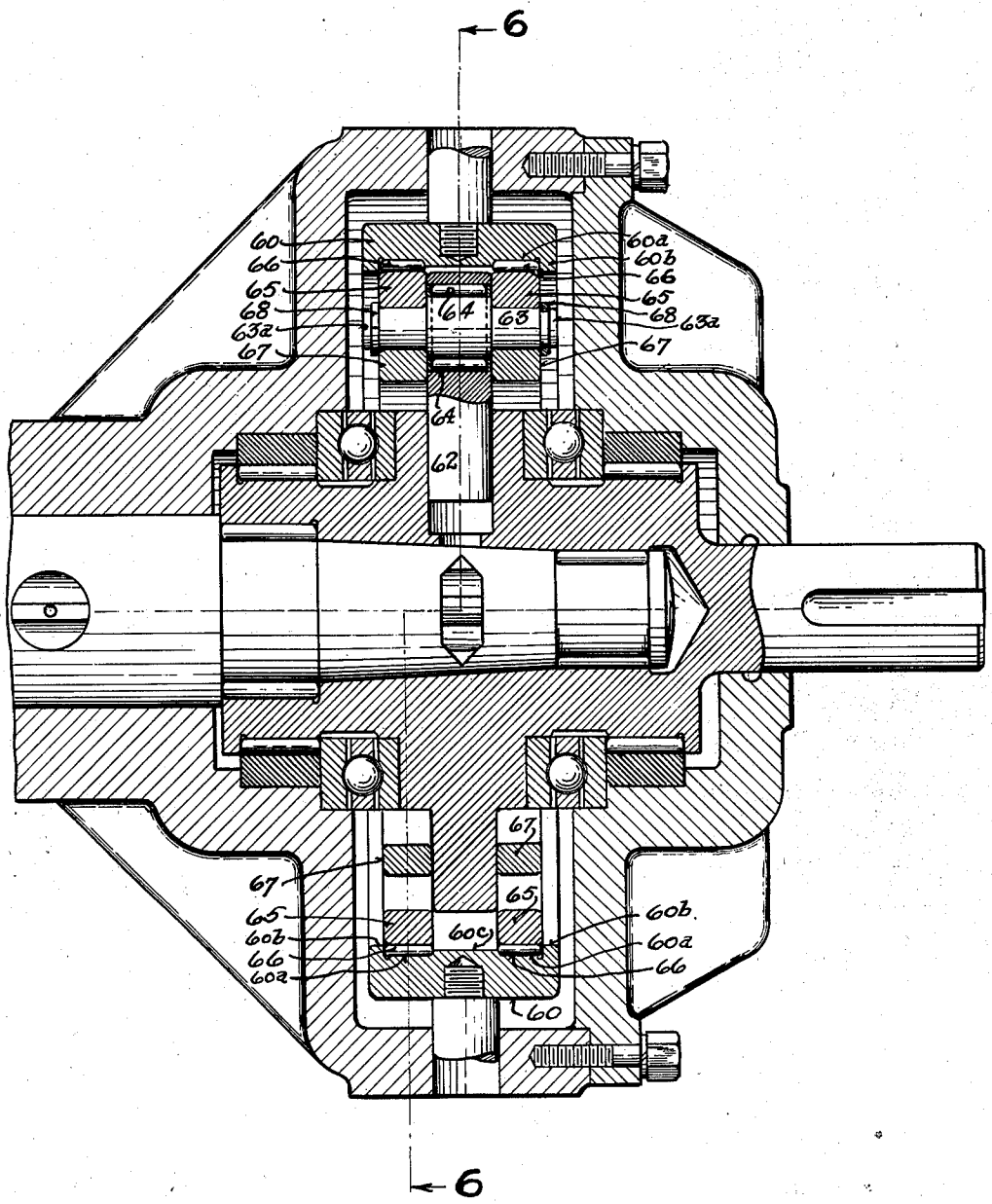


FIG. 5.

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Patented Dec. 7, 1937 2,101,730

UNITED STATES PATENT OFFICE

2,101,730

HIGH PRESSURE HYDRAULIC PUMP OR MOTOR

Elek K. Benedek, Bucyrus, Ohio

Application June 20, 1935, Serial No. 27,558

11 Claims. (Cl. 103—161)

This invention relates to rotary, radial plunger pumps or motors in which high fluid pressures are to be developed or utilized, particularly in connection with a hydrostatically balanced pintle, one of the principal objects being to maintain a high degree of efficiency and durability at high operating fluid pressures.

Another object is to accomplish this result with improvements contributing to the higher efficiency incorporated in the structure in space heretofore unused in apparatus of this character.

Still another object is to provide a balanced drive between the primary rotor or barrel and the reactance or secondary rotor.

Other objects and advantages will become apparent from the following specification wherein reference is made to the drawings in which—

Fig. 1 is a horizontal longitudinal sectional view of a pump or motor embodying the principles of the present invention;

Figs. 2, 3 and 4 are vertical cross sectional views taken on the planes indicated by the lines 2—2, 3—3 and 4—4 respectively in Fig. 1;

Fig. 5 is a fragmentary sectional view taken on a plane corresponding to the plane 2—2 of Fig. 1, but illustrating a modified cooperative structural relationship between the plungers and rotary reactance; and

Fig. 6 is a cross sectional view taken on the plane indicated by the line 6—6 of Fig. 5.

For the purposes of illustration the structure herein shown will be described as a pump, its use as a motor being readily apparent therefrom, and, in the cross sections of the drawings, the needle rollers are shown in elevation, for clearness in illustration.

In my copending application, Serial No. 754,753, filed November 26, 1934, is described a rotary, radial plunger pump employing a pintle which is balanced hydrostatically so as to prevent deflection and binding between the pintle and the bore of the associated cylinder barrel of the pump. In such a balanced pintle structure, it is apparent that the projected end area of the pintle and the end wall of the barrel bore are subjected fully to the specific pressure of the operating fluid with the result that heavy axial thrusts are imposed on both. This fluid pressure tends to force the barrel and pintle axially apart, shifting the pintle axially within its mounting in one direction and forcing the barrel in the opposite direction. At the lower operating fluid pressures, suitable standard anti-friction roller bearings, capable of resisting both radial load and axial thrust can be obtained and used for resisting axial displacement of the barrel, axial displacement of the pintle being resisted by securely mounting it within the pump or motor casing by press fitting. However, such commercial combination radial load and axial thrust roller bearings, having a specified rating for thrust, are available only within comparatively narrow limits of size and capacity. Beyond this range, they are supplied only on special order and even then the manufacturer refuses to rate them or guarantee any particular degree of performance. In fact, for pressures and sizes herein required, theoretical and empirical knowledge is totally lacking in this connection. Furthermore, as to such bearings, even within commercial limits, radial load tends to decrease the efficiency of the bearing as to axial thrust and vice versa. Consequently, such bearings are largely a compromise predicated on a necessary acceptance of a lowered overall efficiency.

If the pintle has a projected end area of less than ten square inches, such bearings often prove satisfactory for pressures up to two to three thousand pounds per square inch. In pumps of the present type, however, in which a projected pintle area is more than ten square inches,—for example, about twenty square inches,—and a specific fluid pressure of from three thousand pounds per square inch is employed, it is apparent that the total effective hydraulic pressure urging the barrel and pintle apart from each other axially amounts to thirty tons. Under such terrific axial pressures, combined with correspondingly increased radial load and large pitch diameter, the usual commercial combination bearings are unsatisfactory and will either fail quickly or will operate at an extremely low efficiency and only for a comparatively short period. Even when the rated load limits of these commercial bearings are approached, though not exceeded, the efficiency of the bearings drops off rapidly.

Again, as will be explained more fully hereinafter, hydrostatic balance of the pintle transfers the unbalanced hydraulic radial load on the barrel and pintle into axial thrust therebetween, with the result that the radial load is reduced to driving torque reactance only while the axial thrust is multiplied so many times that the ratio between the load and thrust does not fall within the required ratio limits of combination bearing design.

Though useful for low pressures and small capacity pumps and motors, the present pump structure is particularly concerned with operating fluid pressures of three thousand pounds per square inch and up, and with large diameter pintles, in which instances more highly efficient

radial load and axial thrust provision must be made for the reasons above mentioned.

Referring, therefore, to the drawings, the pump is mounted in a casing, designated generally as 1, which comprises a hub portion 2 having at one end a radial flange wall 3 terminating at its outer radial limit in an annular, axially extending flange 4. The opposite end of the flange 4 is closed by a removable end plate 5 securely fastened to the free end of the flange 4 for completing the casing.

Mounted within the casing 1 and coaxial therewith is a rotatable cylinder barrel 6 having a plurality of circumferentially spaced, radial cylinders 7 intermediate its ends. The barrel is provided with an axial dead end bore 8 with which the cylinders 7 communicate through suitable respective ports 9 for valving cooperation with a valve pintle 10.

The valve pintle 10 has formed thereon an enlarged shank portion 10*a* which is press fitted into a suitable bore in the hub portion 2 of the casing for fixedly securing the pintle in position coaxial with the barrel and for preventing axial movement thereof. The pintle protrudes from the supporting hub 2 into the dead end bore 8 of the barrel and is provided intermediate the ends of the protruding portion with a tapered valve portion which snugly fits a corresponding tapered portion of the wall of the bore 8. At the ends of the valve portion of the valve pintle are annular surfaces forming inner bearing races 11 and 12 on which operate sets of capillary cageless needle rollers 13 and 14 respectively, so that, with the barrel independently supported rotatably within the casing, as will later be described, the rollers 13 and 14 may constrain the pintle in accurate radial position with respect to the bore 8 and with capillary clearance between the pintle and bore wall. This clearance permits the formation of a capillary oil film along the entire length of the pintle instantaneously for equalizing the hydrostatic load and thereby preventing hydrostatic deflection of the pintle.

The bearing race portion 12 of the pintle is preferably in the form of a groove and, with the corresponding wall portion of the barrel bore, provides the races for the bearings 14 and constrains the bearings 14 to fixed axial position. The barrel bore 8 is counterbored to greater diameter in the barrel portion in radial alignment with the bearings 13, as indicated at 15, so that the bearings 13 are constrained against axial displacement by the end wall 16 of the counterbore and the radial wall 17 between the race portion 11 and larger diameter shank portion 10*a* of the pintle.

The pintle valve portion is disposed between the races 11 and 12 and has a pressure port 20 and a suction port 21 which are connected respectively to longitudinal ducts 22 and 23 in the pintle. The ducts 22 and 23, in turn, communicate with radial ducts 24 and 25, respectively, formed in the shank portion 10*a* of the pintle, and are not connected with each other. The ducts 24 and 25 are continued radially through the hub portion 2 of the casing and connected to the device to be operated and to a sump respectively, not shown, which completes the hydraulic circuit.

As more fully explained in my copending application, it is desirable that a pressure film be provided on the pintle and that this film be introduced as a circumferentially continuous film from the free end of the pintle. In some instances this is effected by virtue of the slip from the port 20, this slip filling the dead end of the bore 8 and thereafter assisting in maintaining a circumferentially continuous film. However, upon starting or upon a sudden shift to a high pressure, such a film may not be obtained instantaneously. Operation for an instant without such film may and usually does cause binding, heating, and seizure of the pintle in the barrel bore. Consequently, a pressure fluid chamber into which pressure fluid is continuously supplied from the working fluid of the pump and which is so related to the pintle as to supply pressure fluid thereonto at the free end is provided. The preferred structure for this purpose is illustrated and comprises a small axial duct 26 in the pintle extending entirely to and opening through the free end of the pintle into the dead end of the bore 8. In referring to the bore 8 as a "dead end" bore it is meant that no bleed vent for draining the fluid from the end of the bore adjacent the free end of the pintle is provided but instead, this end of the bore is sealed due to its original formation to terminate short of the end of the barrel or, if originally extended entirely therethrough, by subsequent plugging and sealing. Even though overload relief valves are provided for safety therein, yet the bore remains in operative effective "dead end". The pressure fluid chamber may be provided in such instance by axial clearance between the free end of the pintle and the end wall of the dead end bore 8. The duct 26 communicates through a small radial duct 27, as better illustrated in Fig. 4, with the main pressure duct 24 of the pintle so that the fluid is supplied into the fluid chamber at the operating pressure of the pump or motor. Such a chamber insures introduction of a circumferentially continuous film at the free end of the pintle. This fluid is forced over the bearings 14 continuously and unbrokenly around the valve portion of the pintle 10 and to the bearings 13 so that a complete and unbroken capillary film is maintained both on the valve portion and the bearings. Since the bearings 13 and 14 reduce the effective overhang of the pintle and the unsupported length of the pintle relative to the barrel, and since the valve portion of the pintle is comparatively short axially and is held by the bearings with axial clearance relative to the bore 8, the film is formed instantaneously with the result that the hydrostatic pressure acts radially inwardly around the entire circumference of the pintle. As a result, the pintle is hydrostatically balanced as also is the barrel. Here it should be noted that one of the greatest forces creating radial load on the barrel load supporting bearings results from the unbalanced hydraulic load reactance, created due to exposure of only half the circumference of the pintle and barrel bore wall to the pressure fluid. At high fluid pressures, this load equals the specific pressure multiplied by the projected pintle area of exposed pintle length times average pintle diameter at the exposed portion. The axial thrust is comparatively small, being only the pressure effective due to the slight pintle taper. When the pintle is hydrostatically balanced, however, this radial load is transferred into axial thrust leaving the radial load only the reactance from torque transmission. This transfer effects several beneficial results. Some noteworthy results are that the radial load occasioned by torque reactance is not augmented by the hydraulic load and the axial thrust does not tend to cause misalignment

or deflection of the pintle or barrel but instead creates a beneficial effect on both. Again, the axial thrust created in this manner necessarily acts only on the projected end or cross sectional area of the pintle, which area is for less than the circumferentially exposed area, and consequently the total thrust pressure is greatly reduced. However, at exceedingly high pressures and with a large projected end area of the pintle thus freely subjected to the full specific fluid pressure of the apparatus, it is apparent that a very severe axial thrust is imposed on the pintle and on the end wall of the barrel bore which tends to move the pintle toward and further into the hub portion 2 and the barrel in the opposite direction. Combination radial load and axial thrust bearings, however, are designed only for limited low ratios of load to thrust and, with present knowledge, cannot be made to operate efficiently beyond such ratio. The present hydrostatic balance, however, renders the axial thrust so greatly predominant that combination bearings are useless for high pressure structures.

Accordingly, the barrel is provided, midway between the bearings 13 and 14 and in the zone of the cylinders 7, with a radial flange 30 having an axially thickened portion 30*a* near its base and juncture with the barrel 6. The cylinders 7 extend radially to the periphery of the thickened flange portion 30*a*. In the flange 30, beginning at the outer ends of the cylinders, are radial guideways 31, each guideway extending radially and being aligned with a corresponding cylinder 7.

Mounted within the cylinders are plungers 32, each plunger having an enlarged head 33 formed thereon which is reciprocable radially in the associated guideway 31. Each piston head 33 is provided with a transverse bore which extends parallel to the axis of rotation of the barrel and in which is mounted a thrust pin 34. Each pin 34 has an enlarged central portion lying within the bore of the associated head 33, and reduced diameter end portions at each side of the head. The crosspin is supported in the bore of the head on suitable capillary cageless rollers 35 so as to be freely rotatable therein. Mounted on the protruding ends of each pin 34 are reactance engaging rollers 36, these rollers being supported on anti-friction capillary cageless needle rollers 37 for rotation independently of the pin. Each plunger head 33 has flat, lateral walls lying in the planes of the radial walls of the flange 30 respectively, and the abutting ends of the rollers are flat so as to lie thereagainst with only operating clearance. Circumferential grooves 38 are formed in the reduced end portions of the pin near the outer ends for receiving snap springs 39 which, when seated in the grooves 38, cooperate with the lateral faces of the plunger head for retaining the rollers 36 and the corresponding needle rollers 37 in the proper position on the pins. The rollers 36 are in rolling engagement with a reactance stator 40 which has annular radially extending shoulders 41 near its ends in position to engage the outer ends of the rollers 36 with operating clearance for constraining the rollers to proper position axially relative to the stator. The stator 40, in turn, has diametrically opposite slide surfaces 42 on its outer surface which slide surfaces engage complementary slide surfaces 43 formed in the flange portion 4 of the casing. These slide surfaces cooperate to support the stator with its axis parallel to the axis of the barrel while permitting adjustment of the stator to different degrees of eccentricity relative to the barrel, whereby the plunger stroke may be adjusted.

Suitable control rods 44 are connected to the stator and extend parallel to the surfaces 42 and through suitable bores in the casing so that the stator may be adjusted thereby from the outside of the casing.

In order to actuate the plungers on the suction strokes, free floating rings 47 are provided, these rings engaging the extreme ends of the pins 34 outwardly beyond the rollers 36. Since the pins are anti-frictionally mounted for rolling in the bores of the head 33, the end portions thereof roll relative to the rings 47. This mounting of the plunger and formation of the barrel is more fully described in my copending application, Serial No. 25,449, filed June 7, 1935. Obviously, in such a structure, considerable space exists between the outer radial limits of the barrel adjacent each side of the flange 30 and the inner limits of the rollers 36 at minimum stroke. This space has heretofore been occupied only by material of the barrel providing strength thereof. In the present invention it is utilized for accommodating the axial thrust bearing assembly.

First, for withstanding radial load, the barrel is provided, near its ends, with axial shoulders 6*a* and at the outermost ends with radial annular shoulders 6*b*. These surfaces are preferably treated and hardened to withstand wear. Mounted in the hub portion 2 of the casing and the end plate 5 are hardened annular race rings 48 of tool steel, these rings being seated on suitable axial annular shoulders 49 and 50 respectively, which join, at their outer ends, with radial annular shoulders 51 and 52 respectively. The inner or bearing faces of the rings 48 are parallel to each other and to the axis of the barrel. Thus the rings are supported against radial load and axial thrust outwardly from the zone of the cylinders. Sets of capillary cageless needle rollers 53 are disposed between the rings 48 and the corresponding race portions 6*a* of the barrel for rotatably supporting the barrel and providing adequate bearing surface to withstand the pressures. However, the needle rollers 53 are not constrained to fixed axial position except in the inward direction nor are the rings 48 except by the races 59, later to be described.

In order to accomplish this result and at the same time withstand axial thrust on the barrel, the barrel is provided, adjacent the enlarged portion 30*a* of the flange, with axial annular shoulders 54. Correspondingly, the enlarged portion 30*a* of the flange is provided with a flat annular radially extending face portion 55.

Firmly seated on the two surfaces 54 and 55, one at each side of the flange portion 30*a*, are inner races 56 of sets of axial thrust bearings 57. Each race 56 is provided with an annular groove on its outer face for receiving the balls 57 and constraining the balls against radial displacement. The balls are separated from each other by a suitable spacing ring 58. Correspondingly grooved outer races 59 are provided as the complementary races of the balls 57. The races, however, are so grooved that, when cooperating with the balls, the race 59 extends radially inwardly beyond the inner radial limit of the race 48. Thus the race 59 constrains the race 48 and the needle rollers 53 to fixed axial position and, in turn, is supported against outward axial thrusts thereby. The races 59, however, engage the inner end of the associated race 48 and the

corresponding radial wall of the flange portion 3 and plate 5 of the casing with slight operating clearance so that the races 59 are free to float radially unconstrained except by the balls 57 themselves, and are relieved from all radial thrusts.

Therefore, within the space, which has heretofore been occupied by additional material of the barrel, means are provided for resisting the axial thrust on the barrel imparted by the hydraulic pressure or otherwise and in addition these means effect the same reinforcement against bending stresses and the like. Since both the radial load bearings 53 and the axial thrust bearings 57 withstand only the load and thrust respectively for which provided, and need not withstand any other thrusts, highly efficient and smooth operation is assured and the thrust capacity is unlimited except by the ability of the bearing elements and the races to withstand a direct and a uni-directional load or thrust. The races of the radial load and axial thrust bearings are so related that no increase in the working span of the barrel, between the radial bearings results but, on the contrary, the barrel is reinforced by the races 48, and the interposed assembly of the thrust bearings 57, and their respective bearing races 56 and 59. The cylinder flanges of the barrel are also reinforced against bending forces by the same bearing structure. It will be seen that a downward deflection of the barrel would be resisted in axial thrust reactance on the shoulders 51 and 52 of the end covers of the casing. Again, the diameter, or size, of the remaining parts is not increased due to the thrust bearings. In fact, with the arrangement described, a more compact structure is obtained even with the increased hydrostatic pressures and efficiency than can be obtained by standard commercial thrust bearings.

At the higher pressures, however, it becomes desirable, and oftimes necessary to utilize a balanced load reactance. Referring to Figs. 5 and 6, such a reactance is illustrated and cooperates with the high pressure cylinder and plunger assemblies in a manner afford'ng lubrication from the slip spray therefrom necessarily occuring at high pressures. The structure of Figs. 5 and 6 is similar to the structure illustrated in Fig. 1 insofar as the barrel mounting and radial load and axial thrusts thereon are withstood. The pintle is also hydrostatically balanced in the manner described in connection with Fig. 1, as a result of which the barrel is subjected to axial thrusts equal to the operating fluid pressure in pounds per square inch multiplied by the projected end area of the pintle or the projected cross sectional area of the bore of the barrel. In this structure, the stator 60, corresponding to the stator 40 above described, is provided with axially spaced circumferential bearing surfaces 60*a* arranged one at each side of the zone of the plungers 62. Instead of providing the rollers 36 on the ends of the thrust pins, each thrust pin 63 is anti-frictionally mounted on capillary cageless needle rollers 64 in the associated plunger head bore so that the pin may roll freely therein. Aligned radially with the bearing surfaces 60*a* are outer reactance rings 65. Sets of capillary cageless needle rollers 66 are interposed between the rings 65 and the bearing surfaces 60*a* of the stator so that the rings 65 are supported for free rotation within the stator, the needle rollers 66 being constrained to fixed axial position by suitable shoulders 60*b* on the stator at their outer ends and retaining shoulder 60*c* at their inner ends. Inner reactance rings 67, aligned radially respectively with the rings 65 are provided and are in rolling engagement with the end portions of the pins 63. The ends of the pins 63 protrude axially beyond the outer ends of the cooperating rings, as indicated at 63*a*, and in these outermost protruding end portions are formed annular grooves which receive snap springs 68. It will be noted that the springs 68 retain the rings 65 and 67 in proper axial position and, since these rings abut the ends of the enlarged central portion of the pins 63 near the outer marginal limits of the large central portion, and likewise lie in face to face relation with the plunger heads and barrel flange, all the parts are held together and in proper operating position by virtue of the springs 68.

It will be seen that due to the provision of the lips 60*b* at the ends of the stator 60 and due to the radial alignment of the inner edge of the lips and the inner race diameter of the rollers 66, that a complete circumferential trough or lubricating chamber will be provided for the rollers 66. This chamber is such that high pressure fluid and spray escaping between the plungers and cylinders will be caught therein and retained under high pressure due to centrifugal force. As a result the rollers 66 will provide a combination bearing having three distinct and beneficial characteristics. By using capillary cageless needle rollers 66 in the trough or chamber thus formed, the needle roller assembly as a whole will rotate at reduced speed relative to the rotation of the rings 65. But this rotation will maintain high centrifugal force with resultant high centrifugal fluid pressure in the trough, inasmuch as the fluid cannot escape except by overflowing the trough. Furthermore, the capillary attraction of the fluid will maintain a continuous pressure film between the rollers 66 due to their capillary size, as described in my copending application, Serial No. 641,186, filed November 4, 1932. Further, due to the relatively great accuracy and uniformity of the needles, they will tend to contact their pathways in metal to metal contact as do anti-friction bearings with the exception that the conventional radial clearance allowed in anti-friction bearings will be very greatly reduced and only sufficient to maintain a high pressure oil film by capillarity and centrifugal force. This combination bearing, therefore, operates as a high pressure oil bearing, with the additional advantage of greatly reduced relative rotation between the supported parts because of the reduced rotation speed of the needle rollers as an assembly. The rollers operate as anti-friction bearings with the additional advantage that the needles roll on high pressure oil films instead of in metal to metal contact with the supported parts. Thirdly, since the needle rollers are constrained to proper circumferential spacing only by the high pressure oil film and each is entirely surrounded thereby, the load thereon is distributed over a large area both by the film and the large number of rollers so that the bearing surface is comparable to the area that is provided by the conventional high pressure oil film bearing. The circumferential trough provides an additional advantage in that, with it, it is not necessary that true capillary needle rollers be used but instead larger and shorter, or larger and longer, rollers may be used because they are operating in a centrifugal high pressure oil bath and can be spaced a capillary distance apart and held in

such condition by the high pressure bath. Thus the cages can be disposed of. Once the rollers are thus disposed in the bath, the spacing capillary films will remain between them for retaining the fluid when the pump or motor is stopped, and spacing films will be afforded by centrifugal force when the motor is in operation and greater forces tending to dislodge the films are present. The inner shoulder 60c preferably has an inner diameter about equal to or less than the pitched diameter of the rollers so that the rollers are constrained to proper axial position. Thus the entire trough between the shoulders 60b of the stator forms a centrifugal oil pressure trough sufficient for effective lubrication and spacing of the rollers. At the same time this bearing structure does not require any more space than the structure illustrated in Fig. 1, and less space than conventional anti-friction bearings.

In order to balance the load, however, the rings 67, instead of being free radially, are in sufficiently tight engagement under pressure with the end portions of the pins 63 so as to be in rolling engagement therewith at all times. The rings 67 are of sufficient radial dimension and elastic capacity to transmit the unbalanced radial load applied on the rings 65 without permanent distortion. Since the end portions of the pins 63 are in tight engagement both with the rings 65 and the rings 67, the heavy radial load on the outer rings 65, which causes elastic deformation thereof, especially at the loaded portions engaged by the pin ends, is transmitted to the pin ends at the unloaded side of the pump or motor due to the slight elastic deformation of rings 65. This transmitted load, in turn, reacts against the rings 67 at the unloaded side of the barrel and ring 65. The ring 67, in turn, reacts against the pins originally imposing the load at the loaded side of the barrel. These forces and reactances are transmitted by rolling frictional pressure so that no appreciable losses result therefrom. Thus any tendency of elastic deformation of the ring 65 is immediately transmitted to the inner ring at the unloaded portion by virtue of the pins and by the rings 67 and loaded pins is retransmitted back to the ring 65 so that the rings mutually react against each other through the medium of the pins due to slight elastic deformation. It will be noted that all of the pins act on the opposite surfaces of the rings 65 and 67 respectively and thus mutually support and reinforce each other, forming an extremely rigid structure as the secondary rotor in which the load is thus balanced equally and uniformly about the circumference thereof. In this manner the rings 67 act as a load equalizing and transmitting means for balancing the load on the rotary reactance. This action, by distributing the reactance forces more uniformly, relieves the rollers 66 from concentrated stresses. As above mentioned, since at high pressures an adequate spray of fluid is always present for the rollers 66, it is possible to use rollers which are not true capillary rollers. In such instance any size of roller may be used but should be positioned with a total circumferential clearance of from one to three roller elements, according to the total number of rollers required in the bearing assembly. This clearance when prorated evenly among the roller elements provides only interspaces for capillary or high pressure oil films therebetween which, during operation, are filled with fluid and will then retain sufficient fluid by capillary attraction and centrifugal force to lubricate the parts under all operating conditions. Due both to the balanced load exerted by the ring 65 and the oil films, the reactance forces are more evenly distributed over the bearing assembly, and at the same time the roller elements 66 are constrained to circumferentially evenly spaced relation with respect to each other and may roll individually about their own axes at all times without mechanical friction between adjacent rollers.

Having thus described my invention, I claim:

I claim:

1. In a rotary radial plunger pump or motor including a casing, a rotatable barrel therein, plunger and cylinder assemblies carried by the barrel, axially floating reactance means for the assemblies, said barrel having a dead end valve bore, a stationary valve pintle fitting said bore for valving cooperation with the assemblies and terminating in spaced relation to the dead end of the bore, said pintle having high and low pressure valve ports and having a duct communicating with the pressure port and discharging into said dead end of the barrel bore, and said pintle being hydrostatically balanced by operating fluid at line pressure maintained between the end of the pintle and dead end of said barrel bore, whereby the pintle and barrel are urged axially apart by fluid at operating pressure, radial thrust anti-friction bearing means rotatably supporting the barrel in the casing and constraining the barrel to fixed radial position only, and anti-friction axial thrust bearings mounted in the casing in cooperation with the barrel for constraining the barrel to fixed axial position against the fluid pressure exerted thereon, said axial thrust bearings comprising an inner radially extending race in fixed position on the barrel, an outer radially extending race arranged to abut the casing under outward axial thrust forces and free to float radially relative to the casing, and free rolling elements interposed between the race for transmitting axially directed forces therebetween, and means operatively connecting the elements to the races for supporting the outer race in proper alignment with the inner race.

2. In a rotary radial plunger pump or motor, a casing, a rotatable barrel therein having a hub portion and a radial flange portion, said flange portion having a radial side wall facing toward the said hub portion and extending radially inwardly of the barrel substantially to the projected surface of the hub portion, said casing having a circumferential shoulder aligned radially with the hub portion and a radial shoulder at the outer end of the circumferential shoulder, a radial load bearing race fitting on said casing shoulders, anti-friction rolling elements interposed between the race and hub and constraining the barrel to radial position only, anti-friction axial thrust bearing means for said barrel comprising an inner race abutting said radial side wall of the flange, an outer race aligned axially therewith and abutting the inner end of said radial thrust bearing race and extending radially inwardly of the barrel beyond the inner bearing surface of said radial thrust race and thereby constraining the said rolling elements from inward axial displacement, rolling elements interposed between said axial thrust races, a plurality of cylinder and plunger assemblies carried by the barrel in the flange portion, reactance means therefor, and valve means in valving cooperation with said assemblies.

3. In a high pressure rotary radial plunger

pump or motor, a casing, a plurality of rotatable plunger and cylinder assemblies therein, a valve pintle in valving cooperation therewith, thrust pins carried by the plungers respectively and mounted for rotation therein about an axis parallel to the axis of rotation of the plungers, and extending laterally from the associate plungers at both lateral limits of the plungers, an annular stator surrounding the extending portions of the pins and having its axis parallel to and offset from the axis of rotation of the plungers, a pair of axially spaced rotatable reactance rings coaxial with the stator, roller elements interposed between each reactance ring and the stator for rotatably supporting the rings therein, said rings being disconnected and rotatable independently of each other and lying closely alongside the outer portions of the plungers, said extending portions of each pin being in rolling engagement with the rings respectively at all times, and the outer end portions of the pins extending axially outwardly beyond the associated rings, means on the extending end portions of the pins engaging the rings and constraining the rings to substantially fixed position axially with respect to the plungers, and axial thrust means constraining the plunger and cylinder assemblies to fixed position axially in the casing.

4. In a high pressure rotary radial plunger pump or motor, a casing, a plurality of rotatable plunger and cylinder assemblies therein, a valve pintle in valving cooperation therewith, thrust pins carried by the plungers respectively and mounted for rotation therein about an axis parallel to the axis of rotation of the plungers, and extending laterally from the associated plungers at both lateral limits of the plungers, an annular stator surrounding the extending portions of the pins and having its axis parallel to and offset from the axis of rotation of the plungers, annular grooves in the inner wall of the stator, a pair of axially spaced rotatable reactance rings coaxial with the stator, roller elements occupying said grooves and operatively interposed between each reactance ring and the stator for rotatably supporting the rings therein, said rings being disconnected and rotatable independently of each other and lying closely alongside the outer portions of the plungers, said extending portions of each pin being in rolling engagement with the rings respectively at all times, the outer end portions of the pins extending axially outwardly beyond the associated rings, grooves in the extending end portions of the pins, snap springs in said grooves and engaging the rings at the outer ends of the rings and constraining the rings to substantially fixed position axially with respect to the plungers, and axial thrust means constraining the plunger and cylinder assemblies to fixed position axially in the casing.

5. In a high pressure rotary radial plunger pump or motor, a casing, a plurality of rotatable plunger and cylinder assemblies therein, a valve pintle in valving cooperation therewith, thrust pins carried by the plungers respectively and antifrictionally mounted for rotation therein about an axis parallel to the axis of rotation of the plungers, and extending laterally from the associated plungers at both lateral limits of the plungers, an annular stator surrounding the extending portions of the pins and having its axis parallel to and offset from the axis of rotation of the plungers, a pair of axially spaced rotatable reactance rings coaxial with the stator, roller elements interposed between each reactance ring and the stator for rotatably supporting the rings therein, said rings being disconnected and rotatable independently of each other and lying closely alongside the outer portions of the plungers, said extending portions of each pin being in rolling engagement with the rings respectively at all times, inner reactance rings aligned radially with the outer reactance rings respectively, said inner rings being constrained to rolling engagement with the extending pin portions respectively at all times and lying closely alongside the outer portions of the plungers, said inner rings being of sufficient rigidity radially to transmit thrusts without elastic deformation from each pin to the other pins, whereby the load reactance on the outtr rings is balanced, and axial thrust means constraining the plunger and cylinder assemblies to fixed axial position within the casing and means on said thrust pins engaging the reactance rings and constraining the rings to proper axial position relative to the barrel and plungers.

6. In a high pressure rotary radial plunger pump or motor, a casing, a plurality of rotatable plunger and cylinder assemblies therein, a valve pintle in valving cooperation therewith, thrust pins carried by the plungers respectively and mounted for rotation therein about an axis parallel to the axis of rotation of the plungers, and extending laterally from the associated plungers at both lateral limits of the plungers, an annular stator surrounding the extending portions of the pins and having its axis parallel to and offset from the axis of rotation of the plungers, a pair of axially spaced rotatable reactance rings coaxial with the stator, roller elements interposed between each reactance ring and the stator for rotatably supporting the rings therein, said rings being disconnected and rotatable independently of each other and lying closely alongside the outer portions of the plungers, said extending portions of each pin being in rolling engagement with the rings respectively at all times, inner reactance rings aligned radially with the outer reactance rings respectively, said inner rings being constrained to rolling engagement with the extending pin portions respectively at all times and lying closely alongside the outer portions of the plungers, said corresponding inner and outer rings being of sufficient rigidity radially to transmit thrusts by elastic deformation only, from one to the other through the pins, whereby the load reactance on the outer rings is balanced and evenly distributed in the entire periphery of the rings, said extending pin portions extending outwardly beyond the both associated inner rings and outer rings, and abutment means on the outer ends of the pins engaging the outer ends of the inner and outer rings and thereby constraining the rings to substantially fixed axial position with respect to the cylinder and plunger assemblies, and means constraining the plunger and cylinder assemblies to fixed axial position within the casing.

7. In a rotary radial plunger pump or motor, a casing, a rotatable barrel having a plurality of radial cylinders, plungers in the cylinders, and having a dead end axial valve bore for the cylinders, a valve pintle mounted at one end in the casing and having its free end fitting the barrel bore, said pintle terminating at its free end in axially spaced relation to the dead end of the bore, said pintle having a valve portion and valve ports therein in valving cooperation with the cylinders and high and low pressure ducts communicating with the ports respectively and uncommunicated with each other, and having a duct in

communication with said high pressure duct and opening through the free end of the pintle into the dead end barrel bore, spaced sets of anti-friction roller bearings interposed between the pintle and barrel bore wall at the ends of the valving portion for retaining the pintle in fixed radial position in said bore with capillary clearance between the bore wall and valving portion of the pintle, anti-friction radial roller bearings operatively interposed between portions of the casing and barrel in radial alignment with said pintle roller bearings, said barrel having a radial flange portion between the said radial bearings and in the zone of the cylinders, axial thrust rolling bearing elements operatively interposed between the said flange portion and the casing at opposite sides of the flange portion, said elements being positioned inwardly toward the plane of the axis of the cylinders from said radial bearings, and reactance means for the plungers.

8. In a rotary radial plunger pump or motor, a casing, a barrel rotatably mounted therein and having a plurality of circumferentially spaced radial cylinders, plungers in the cylinders, a rotary reactance for the plungers, roller thrust means anti-frictionally mounted on the plungers and in rolling engagement with the rotary reactance for actuating the plungers, valve means having a high pressure port cooperating with the cylinders during revolution thereof along one part of their path of revolution and having a low pressure port cooperating with the cylinders during their revolution along another part of their revolution, whereby an unbalanced hydraulic load condition on the rotary reactance is created, and load transmitting and equalizing means operatively interposed between the roller thrust means and in rolling engagement therewith and constraining all of the roller thrust means to tight engagement with both the rotary reactance and load transmitting and equalizing means whereby said hydraulic load on the rotary reactance is balanced and means on said roller thrust means engaging the rotary reactance and the load transmitting and equalizing means for constraining the same to proper axial position relative to the barrel.

9. In a radial plunger pump or motor, a casing, a rotatable barrel having a plurality of radial cylinders, plungers in the cylinders respectively, said barrel having a dead end axial valve bore, a valve pintle mounted at one end in the casing and fitting said valve bore, said pintle terminating in axially spaced relation to the dead end of the valve bore and having a valve portion with high and low pressure valve ports therein in valving cooperation with the cylinders, and having a duct in communication with said high pressure port and discharging into the dead end of the valve bore, spaced sets of anti-friction radial load bearings operatively interposed between the casing and opposite end portions of the barrel, said barrel having a radial flange portion positioned between said anti-friction bearings and lying in the zone of the cylinders, axial thrust anti-friction bearing means operatively interposed between the flange portion of the barrel and the casing at opposite sides of the flange respectively, said axial thrust elements being positioned inwardly toward the plane of the axes of the cylinders from said radial load bearings, and reactance means for the plungers.

10. In a radial plunger pump or motor, a casing, a rotatable barrel having a central flange portion, reactance means, a plurality of radial cylinders and plunger assemblies interposed between the barrel and reactance means in the zone of the flange portion, said barrel having an axial valve bore, a stationary valve pintle mounted in the casing and fitting said valve bore and in valving cooperation with the assemblies, spaced sets of anti-friction radial load bearings operatively interposed between the casing and portions of the barrel at each side of the flange portion for supporting the barrel for rotation in the casing and having outer races respectively, sets of anti-friction thrust bearings operatively interposed between the faces of the flange portion and the casing and in surrounding relation to the barrel, and having outer races respectively, and being positioned inwardly toward the plane of the cylinders from said radial load bearings, and the outer races of the radial load bearings buttressing the outer races of the axial thrust bearings against axial outward displacement.

11. In a rotary radial plunger pump or motor, a casing, a rotatable barrel mounted therein, a rotary reactance surrounding the barrel in eccentric relation thereto and comprising a stator, axially spaced coaxial reactance rings coaxial with the stator, capillary cageless roller bearings operatively interposed between the rings and the stator for rotatably supporting the rings in the stator and for sealing the radial space between the rings and stator by capillary attraction and defining with the rings and stator a relatively deep slip fluid trough while permitting lubrication of said roller bearings by fluid in the trough, radial piston and cylinder assemblies interposed between the barrel and reactance, said assemblies having portions extending into said trough, means carried by said extending portions and cooperating with said rings for actuating the assemblies, and valve means for said assemblies.

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