

Dec. 7, 1937.

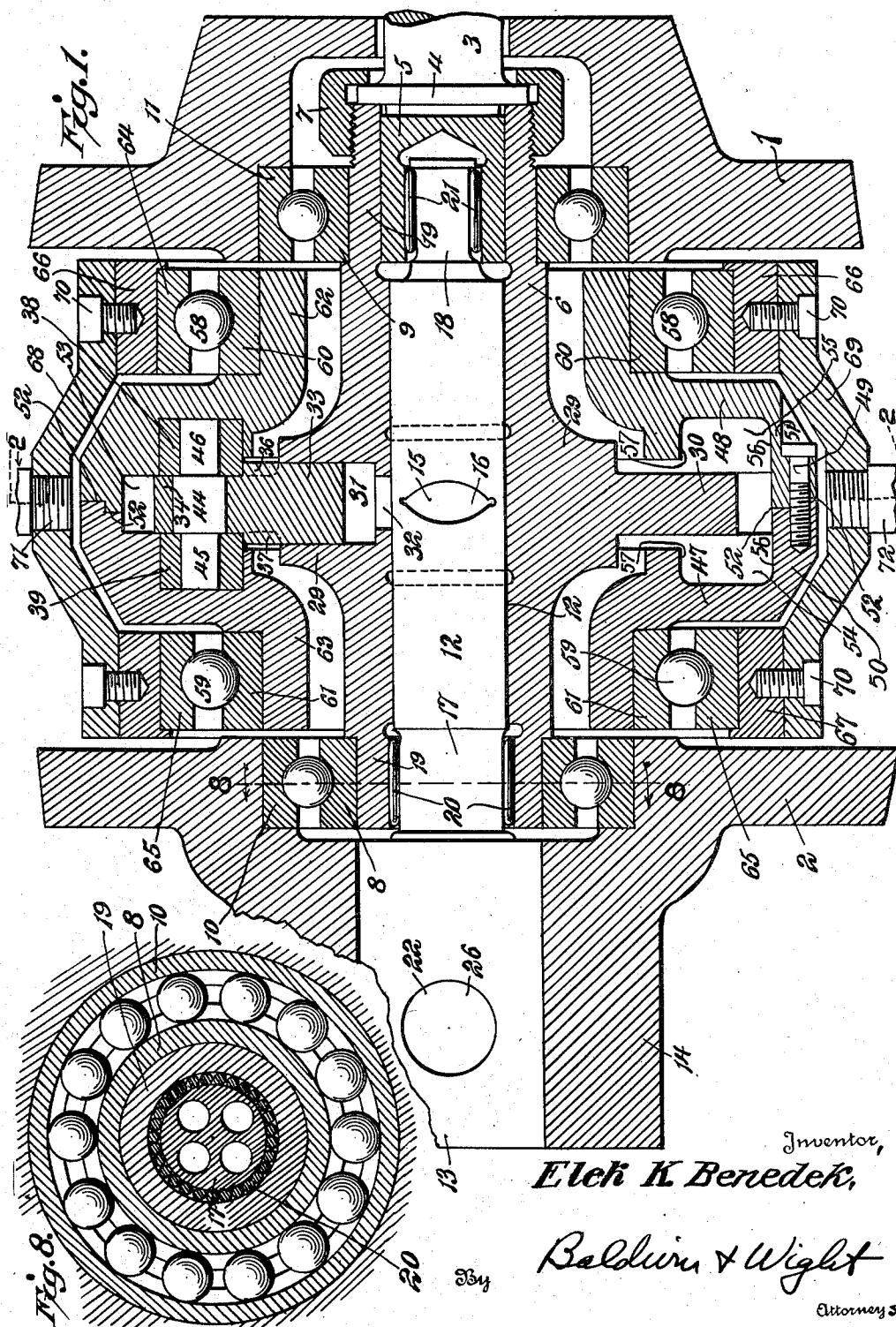
E. K. BENEDEK

2,101,732

HYDRAULIC MACHINE

Original Filed July 24, 1933

3 Sheets-Sheet 1



Dec. 7, 1937.

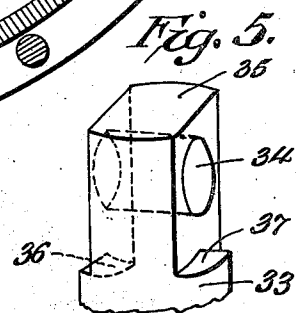
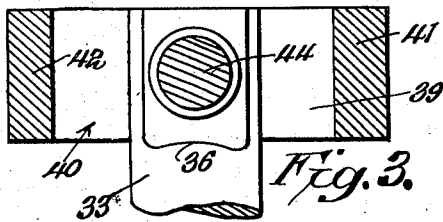
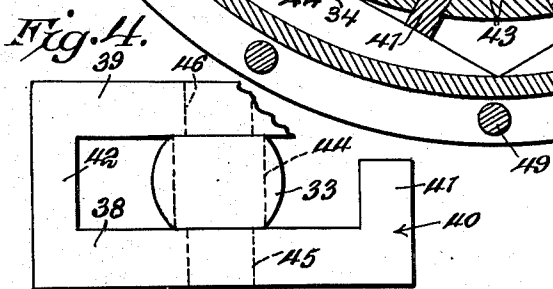
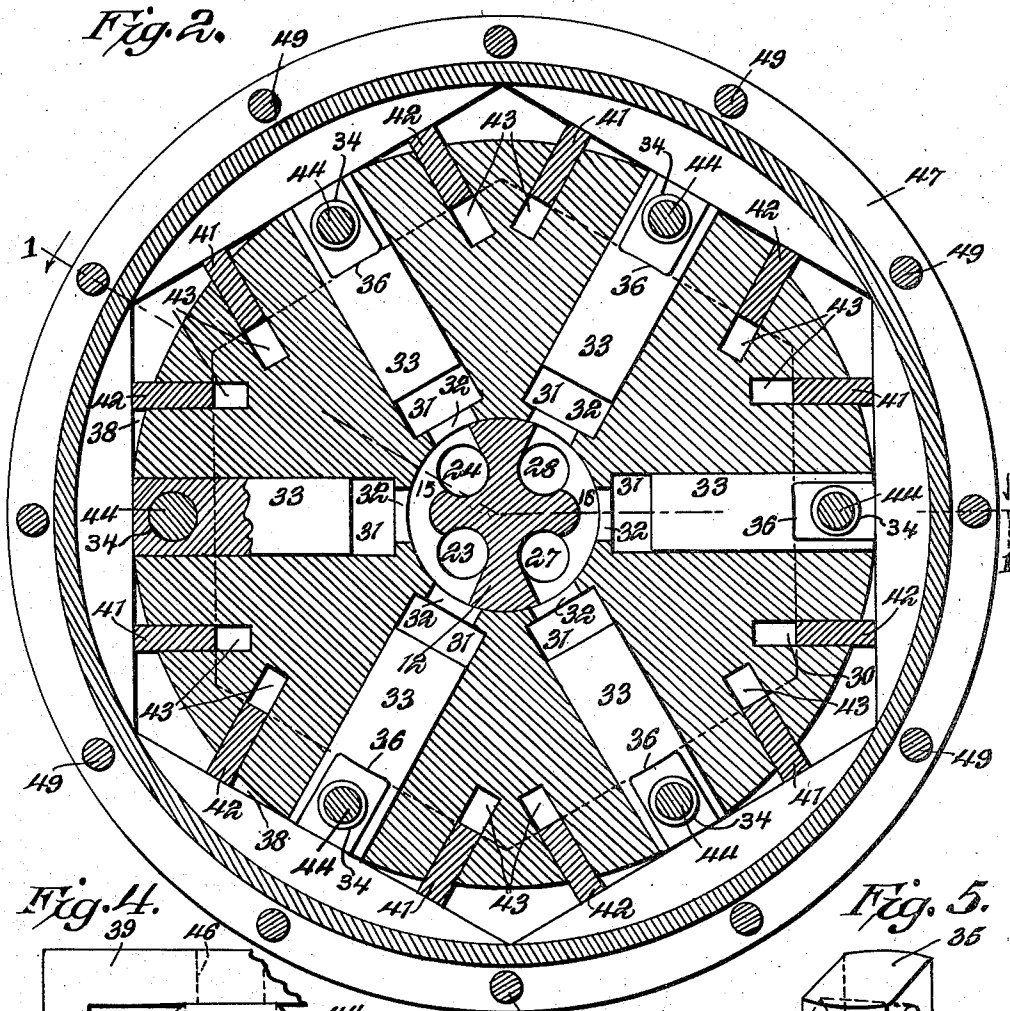
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HYDRAULIC MACHINE

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3 Sheets-Sheet 2



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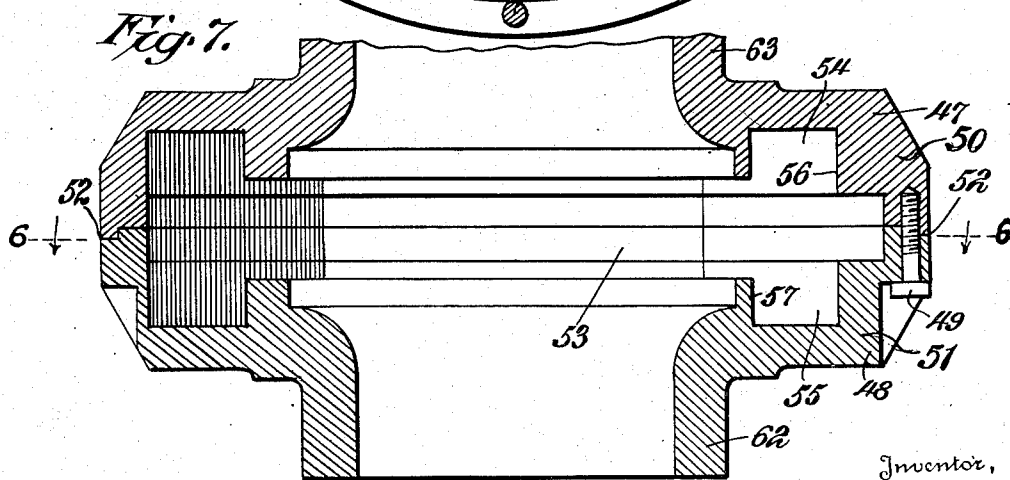
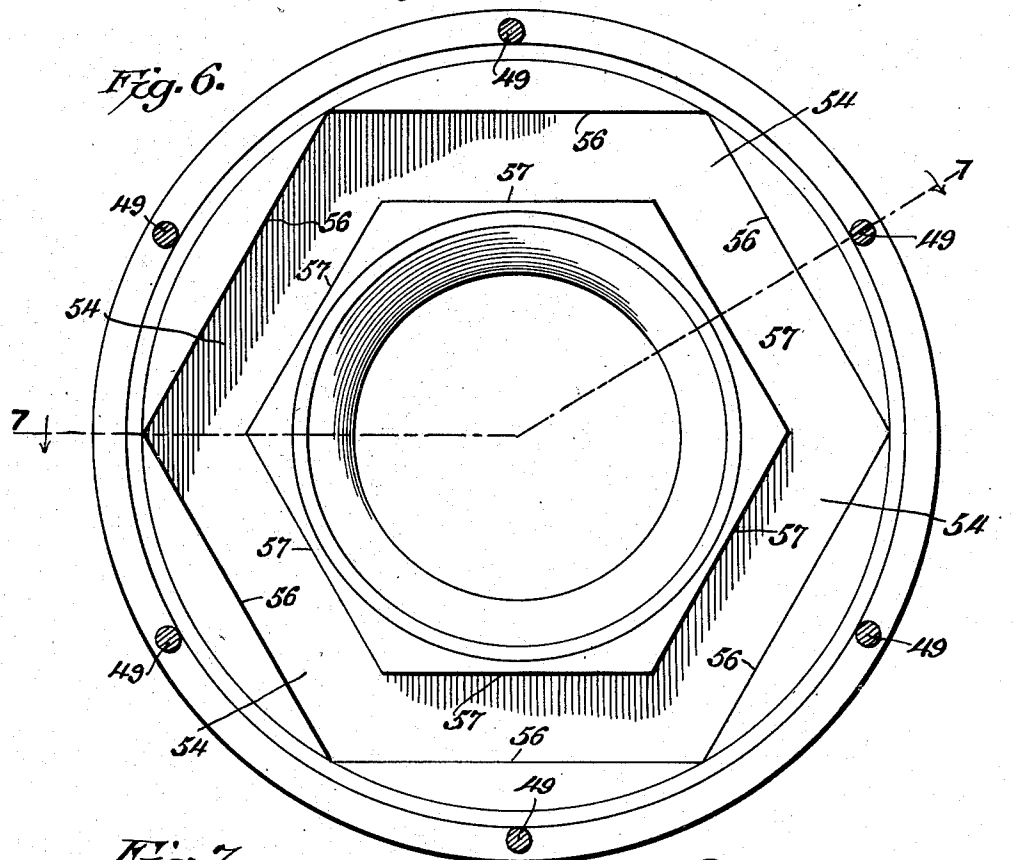
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HYDRAULIC MACHINE

Original Filed July 24, 1933

3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,101,732

HYDRAULIC MACHINE

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Original application July 24, 1933, Serial No. 681,-
991. Divided and this application December
3, 1936, Serial No. 114,072

7 Claims. (Cl. 103—161)

This application is a division of my copending application 681,991, filed July 24, 1933, now Patent No. 2,074,205, issued March 16, 1937.

This invention relates to fluid pressure generators and more particularly to pumps of the kind embodying a plurality of rotatable, radially disposed piston and cylinder assemblies.

The term "fluid pressure generator" is herein used for the purpose of clearness, in order to distinguish the apparatus forming the subject matter of this invention from the conventional class of pumps in which very high pressure rating is not an absolute prerequisite of the usefulness of the device. With the imperative demand of nature's laws of economy, particularly in regard to the usefulness of a modern machine, it is a primary prerequisite to build the machine so that in a certain size a maximum amount of power will be obtainable, or the machine be susceptible of a maximum flow of input energy or power. To do this, fluid pressure generators have to operate first at high speed, and second at high pressure and thus turn out a maximum possible power from the unit. This being an absolute requirement for the usefulness of this type of generator or hydraulic power machine, I find it convenient to use this new term instead of or descriptive of the term "pump", which, as commonly accepted, now designates a hydraulic machine which sucks and delivers fluid, practically speaking, at highly difficult pressures. Therefore, good suction is the main requirement as to the operativeness of such machines. On the contrary, a pressure fluid generator is absolutely useless and inoperative for many present day purposes if it is unable to maintain commercial pressure of the order of 250 atmospheres, or the like. For example, a 10 inch diameter ram of a hydro-power press has to exert a force of approximately 1200 tons, and if a pump is not able to create that tonnage and maintain that tonnage for a determined length of time, it will not be useful and commercial for that press and therefore will not be used as a generator for that press system.

In my co-pending application, Serial No. 681,991, filed July 24, 1933, of which the present application is a division, there is disclosed a fluid pressure generator of such construction as to eliminate many of the difficulties, limitations upon utility, and other shortcomings of prior art pumps. Briefly stated, the apparatus disclosed in the parent application embodies, among other improvements, a novel arrangement for coupling the reciprocable members of radial rotatable piston and cylinder assemblies with a rotary re-

actance member or assembly, such arrangement being claimed in said parent application; and it also discloses, in common with this application, an improved arrangement for mounting and operatively connecting a pintle and a rotor in a casing, which constitutes the subject matter claimed in this divisional application.

In pumps or pressure generators of the rotatable radial piston and cylinder assembly type the enormous piston pressure up to 500 atmospheres, sometimes encountered, creates an equivalent inwardly acting force, which reacts against a central cylindrical valve known as the "pintle" in the American art. This pintle in the pumps of the prior art usually is a long member, supported in the manner of a cantilever beam. This cantilever beam pintle is also supported along the rotating cylinder barrel, being, however, free to flex under the periodic pressure of the resultant piston load, according to the laws of elastic deflections and vibrations. The elastic vibrations and excessive friction created between barrel and pintle combines with the similar action of the piston thrust load transmitting crosshead pin and creates a characteristic radial pump noise under pressure, as well known to those familiar with this art.

To minimize the danger of the hard rubbing mechanical contact between pintle and barrel and aforesaid periodic vibrations of the pintle created by the periodic piston loads, I provide a novel bearing support for the pintle at the end thereof which is opposite to the walled-in or anchored end.

According to the well recognized laws of beam deflection, the unsupported end of a cantilever beam will have a deflection under a given uniformly distributed load about 67 times as great as the maximum deflection of the same beam supported at both ends and the same load.

One object of my invention is to provide an apparatus of the class referred to in which a cantilever pintle is anchored rigidly directly on and immovably with respect to the casing and in which the cantilever pintle nevertheless has a deflection no greater than it would have if supported at both ends directly in the casing.

Another object is to provide an apparatus of the kind referred to in which a novel arrangement of antifriction bearings is provided for maintaining the pintle and the rotor or cylinder barrel in accurate axial alignment.

A further object is to provide a novel organization of pintle, rotor and interposed antifriction rolling bearings so arranged and of such form

as to provide a fluid seal to prevent fluid slip between the rotor and the pintle, as well as functioning as bearings to center the rotor with respect to the pintle.

- 5 It will be understood that the foregoing statement of some objects is not inclusive of all objects, and other objects will become apparent from a reading of the following description, the accompanying drawings, and the appended
10 claims. The provision of the structure defined in each claim constitutes the attainment of at least one object of the invention.

In the drawings:

- 15 Figure 1 is a longitudinal sectional view of apparatus constructed in accordance with the invention, the section being taken on the line 1—1 of Figure 2;

Figure 2 is a section taken on the line 2—2 of Figure 1, with some parts omitted;

- 20 Figure 3 is a detail view drawn on an enlarged scale showing a piston and coupling element assembly partly in elevation and partly in section;

Figure 4 is a top plan view of the assembly shown in Figure 3;

- 25 Figure 5 is a perspective view of a piston head, drawn on an enlarged scale;

Figure 6 is a transverse sectional view of a piston actuating rotary member, the section being taken on the line 6—6 of Figure 7;

- 30 Figure 7 is a longitudinal sectional view taken on the line 7—7 of Figure 6; and

Figure 8 is a fragmentary vertical sectional view taken on the line 8—8 of Figure 1.

- 35 For the sake of simplicity, the illustrations show a hydraulic generator or pump with an even number of pistons, which, however, may be constructed with any other number of pistons, as hereinafter will be specifically described.

- Referring to Figures 1 and 2, the pump selected
40 for the illustration of the invention comprises a liquid-tight casing comprising a body portion (not shown), a shaft-end cover 1, and main covering head 2.

- A drive shaft 3 having an annular shoulder 4
45 and hollow end portion 5 is rigidly secured by a conventional key connection, not shown in the drawings, to one of the supported ends of cylinder barrel or primary driving member 6; which end is pulled tightly against shoulder 4 of drive shaft 3
50 by the lock nut 7. Cylinder barrel 6 is supported at both ends on the inner ring of conventional antifriction bearings 8 and 9 respectively. The outer races 10 and 11 respectively of said bearings
55 are mounted in covers or heads 2 and 1 as shown in Figure 2. The pintle 12 is stationarily held through its enlarged portion 13 in the hub portion 14 of the main head cover 2 by being pressed into said hub portion 14 and in addition, by means of a conventional key connection, not
60 shown in the drawings.

- The central portion of pintle 12 is adapted to distribute the working fluid and sustain the inwardly acting hydraulic pressure. To provide for this and minimize all possible friction, deflection
65 and periodic vibrations of the pintle 12, its ends on both sides of its suction and pressure ports 15 and 16, respectively, have reduced bearing portions as at 17 and 18, respectively. Bearing portion 17 cooperates with the end portion 19 of the
70 barrel 6 in the zone of bearing race 8, as a pilot antifriction bearing by means of interposed needle bearings 20. Reduced portion 18 is similarly disposed in the working zone of antifriction bearing race 9 in such a manner that it projects into the
75 hollow end 5 of the drive shaft 3, which forms

the outer bearing race for the needle bearings 21 of this pilot bearing. Thus, it is evident that while the cylinder barrel 6 is supported at its two ends in antifriction bearings 8 and 9 respectively, the central valve or pintle is also indirectly supported on same bearings through the provision of pilot needle bearings 20 and 21, respectively. Thus, pintle 12 does not rely on the support of its enlarged extension 13, as hitherto has been customary, in the manner of a cantilever beam, i. e., on one end only. Pintle 12 has its reversible ports 15 and 16 formed in the conventional manner with the exception of their contour which provides a continuous curve of a butterfly. Port 15 is in communication with port 22 through axial
15 channels 23 and 24, which further communicates with main line of the outer circuit. Ports 16 and 26 similarly communicate through axial passages 27 and 28, to the outside oil circuit. Cylinder barrel 6 outside of its wider disc portion 29 is provided with a narrow outermost disc portion 30. Each cylinder bore 31 thus extends radially from the outside of the narrow rim 30 to the inner cylinder port 32 in a radial direction, to receive the piston 33. Each piston has an opening or eye 34
25 and an outer end 35 flattened on both sides in a parallel manner (see Figure 5). The piston ends in arcuate shoulders 36 and 37 to engage directly the parallel flanges 38 and 39 of the slide 40. The slide 40 is provided with ends 41 and 42 which
30 are engaged by the slots 43 provided in the narrow disc portion 30 of cylinder barrel 6. Guide slots 43 are parallel with their respective piston bore so that slide 40 through its ends 41 and 42 is thus guided radially and driven tangentially
35 or angularly by the barrel 6. The crosshead pin 44 is engaged by eye or opening 34 of the piston 33 and has lateral extensions, as at 45 and 46 to engage the sides 38 and 39 of the slide 40 and thereby operate the piston during suction stroke
40 only.

The slide 40 as shown in Figures 3 and 4 is rectangular and comprises the slides 38 and 39 interconnected at their ends by the ends 41 and 42, thus forming an open frame. A transverse
45 bore in the sides 38 and 39 is adapted to engage the extensions 45 and 46 of cross pin 44, and thereby the piston 33. Thus, it will be seen that the piston has a free rocking mounting on its crosshead pin 44 or its extensions 45 and 46,
50 thereby ensuring free alignment in its respective bore 31 without strain or stress.

An eccentrically mounted piston actuating assembly comprises connected parts 47 and 48.

It will be seen that when under heavy hydro-
55 static pintle load, with great torque on the slide assembly at full load and full volume, the driving force between cylinder barrel extension 30 and piston actuating eccentric parts 47 and 48 tends to rotate or twist the slides about the cross-
60 head pin 44. In order to accommodate freely for this elastic twist of the slides or slide assembly, the shoulders 36 and 37 of the pistons shown in Figure 5 are formed arcuately, affording rocker
65 bearings. Thus, each piston under its maximum load will be perfectly free from the driving stresses which will be entirely exerted by the slide assembly itself as coupling member between
70 cylinder barrel 30 and piston actuating eccentric members 47 and 48, which form a secondary driven member or reactance rotor. In order to clearly illustrate how the slides 40 are engaged by the eccentric discs 47 and 48, I show in Figures 6 and 7 the eccentric itself. According to these figures the assembly comprises the two
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eccentric discs 47 and 48, which are bolted together by a plurality of cap screws 49, as shown. Each eccentric disc 47 or 48 is provided with a concentrically disposed circular flange portion as at 50 and 51, respectively, affording mountings therefor, as will be set forth later on. The eccentric discs have mating circular faces as at 52 which are slightly staggered to keep their concentricity the same. The eccentric assembly or secondary driver member is provided with circular chamber 53 which will accommodate the flange rim 30 of the cylinder barrel movable therein for the purpose of fluid control. Adjacent and in communication with chamber 53 are a plurality of chordal grooves forming chordal slideways as at 54 and 55 in the eccentric discs, having normally transversely aligned disposition to receive the longitudinal flanges 38, 39, respectively, of the slide structures. Each slideway thus has a load transmitting straight bearing surface 56 and a guide and suction surface 57 to guide the slides during its operation.

Thus, in Figures 6 and 7 we have an axially detachable rigid eccentric or piston actuating member. It is understood that the load transmitting shoulders inside of the eccentric disc members 47 and 48 may be made separably from the disc members and inlaid and fastened inside of said members in a proper manner, or not fastened at all, but kept floatingly in proper relation by the slides as spacers. Eccentric members 48 and 47 are mounted on antifriction bearings 58 and 59 in a well known manner, by having the inner rings 60 and 61 mounted directly on ring flanges 62 and 63 respectively, and the outer rings 64 and 65 arranged in bearing retainer rings 66 and 67. Retainer rings 66 and 67 are provided with diametrically opposite parallel bearing sliding surfaces as a resultant crosshead, and are supported in mating parallel bearing surfaces of end covers 1 and 2 respectively as known in the art. Therefore, they are not shown separately in the drawings.

Bearing retainers 66 and 67 are shifted by yokes 68 and 69 respectively, they being connected in one rigid assembly by cap screws 70.

Control rods 71 and 72 connect the control means to the yoke assembly and control the stroke during operation according to the nature of the job for which the pump will be used.

The pump will operate in a well known manner. When the cylinder barrel and piston crossheads are in concentric relation, the pistons will stand still in their cylinders and no pumping action will take place. However, when the piston actuating members 47 and 48 are adjusted by the control rods 71 and 72 to one side or the other of the center of the cylinder barrel or pintle, the pump will deliver through passages 22 and 26, according to the relative position of the primary and secondary rotors.

The assemblage of parts which has been hereinbefore set forth involves the provision, generally speaking, of the primary driving member or piston barrel 6 associated with the secondary driven member consisting of the eccentric or piston actuating unit including the parts 47 and 48, the said primary and secondary members or units equipped with the usual pistons 33 carried by the primary driving member 6, and the secondary driving member or unit having means for the actuation of the pistons in the manner set forth.

In accordance with the invention, the rolling needle bearings 20 and 21 interposed between

the pintle and the cylinder barrel are of elongated shape and the ratio of their length to their individual diameters, and their closely contiguous positioning is such as to provide between adjacent rolling elements a plurality of capillary tube spaces. Generally, the desired capillary characteristics will be provided if the total length or longitudinal span of the needle rollers is approximately equal to the diameter of the pintle. In the apparatus shown as an illustrative embodiment, such characteristics are obtained by using single sets of needle rollers in unit lengths approximately equal to the pintle diameter, but it will be understood that the arrangement may be modified somewhat, depending on the size of related parts, particularly the pintle. In any case, the total bearing extent of needle elements at one end of the pintle should be approximately equal to the pintle diameter. The spaces between contiguous needle rollers will attract and hold bodies of oil, which, in combination with the rolling elements or needles and the associated pintle and barrel surfaces, act as a fluid seal to prevent flow of fluid or slip between the pintle and barrel. Thus, the needle bearings attract fluid by capillary action, so as to maintain a film of oil between the pintle and the barrel, yet they prevent flow completely along the pintle, i. e., past the bearings. These needle bearings thus have an extremely important function. They have about ten times the load capacity of any other bearing that could be supplied and distribute the load on the pintle and barrel. They maintain a very close spacing of the barrel and pintle so that the space can be truly capillary and receive slip fluid instantly. They have a large bearing surface themselves and not only attract, but retain oil by attracting and establishing a capillary oil film initially around the pintle. These needle rollers should be hardened and ground and preferably, the total or aggregate circumferential spacing between the rollers of each assembly should be not more than the diameter of one roller. I have found that this arrangement results in eliminating substantially all eccentricity of the pintle and barrel, with resultant increase in load capacity.

In some prior art structures, ordinary ball or rolling bearings are interposed between the ends of the pintle and the barrel. As higher pressures are utilized in such structures, not only is the pintle subjected to greater hydrostatic unbalance, but also the clearance itself must be reduced to prevent excessive slip.

In order both to reduce the clearance and yet antifrictionally center the pintle in the barrel, a much greater bearing capacity of the antifriction mounting is required. At the same time, at higher pressures, greater heats are developed in the hydraulic fluid and in the pintle and rotor and if any mechanical contact occurred, this heating effect would be greatly augmented. In order to solve all of these problems concurrently, the specialized capillary needle rollers illustrated are provided between the pintle and barrel.

The advantages of the needle rollers are as follows: First, the use of needle rollers permits a tremendous reduction in radial dimension because of the small diameter of the needles. Secondly, due to the fact that ordinary commercial bearings necessarily have relatively greater radial clearance, whereas needle rollers need only have the very slightest operating clearance and commercial bearings must allow clearances and tolerances for both the rollers and the races, 75

whereas the needles require only clearance between them and the pintle and barrel, an additional decrease in radial dimension is permitted. Thirdly, in the prior art structures, only a relatively few rollers can be provided in a race of given diameter, whereas with the same diameter from ten to twenty times as many individual needle antifriction roller elements can be provided, depending upon the diameter of the latter, which at all events is small. This latter permits not only a saving in radial dimension, but increases the actual bearing surface between the pintle and the barrel from ten to twenty times, so that much greater loads and pressures can be withstood without distortion of the elements themselves and without distortion of the cooperating parts between which the elements are interposed, and without concentrated stresses and elastic deformation, which latter increases frictional heat.

In addition to the advantages of greater bearing capacity due to the large surface of contact, there is also provided a much greater path for the metal-to-metal conduction of heat from the space between the pintle and barrel; that is, from the slip fluid, so that this heat which would otherwise be entrapped may be conducted through the bearings themselves and more rapidly dissipated. Thus, by using the needle rollers, the radial clearance between the pintle and barrel may be reduced. This, in turn, causes a great reduction in slip and the reduction in slip, in turn, causes a great reduction in heat as well as an increase in efficiency.

Again, the actual wear or instantaneous deformation of the needle rollers themselves and of the cooperating parts in engagement therewith is eliminated, thus effecting further reductions in the generation of heat, greater accuracy in the clearance, with a co-existing reduction in the chances for mechanical friction. Thus, by the mere change from commercial rollers to needle roller bearings, much higher efficiency due to reduced slip, much cooler operation with resultant greater uniformity in operation is obtained. As a result of permitting the closer clearances and the other features mentioned, greater pressures are obtainable and an increase in the pressure of the fluid is accomplished by an increase in the mechanical efficiency of the structure.

In the larger commercial bearings, regardless of whether caged or not, the entire load must be carried on a very comparatively few spaced points of contact. Under the heavy loads and pressures involved in pressure fluid generators and motors, the concentration of load due to the widely spaced points of contact results in extreme elastic deformation, both of the antifriction elements and the cooperating parts, not only developing heat, but permitting misalignment of other parts of the apparatus, particularly the barrel and pintle where the clearance should be uniform and as small as possible without danger of seizure. In connection with the pintle and barrel clearance, it should be noted that with absolute concentricity of the barrel and pintle with a resultant uniform clearance circumferentially of the pintle, the slip may have a fixed value based on the pressure times X , where X is the clearance. If, however, the pintle and barrel become eccentric so that the greater portion of the clearance is disposed to one side only, the resultant slip may be increased in accordance with the formula C times $(X)^2$, that is, in accordance with a higher power of X .

In such structures, also, there is a periodic vibration as each piston discharges or receives its charge. Thus, with five pistons at 1200 R. P. M., there is a periodic vibration of a frequency of 600 per minute for each discharge of the piston and 600 for each recharge. This, in effect, creates a frequency of 20 vibrations per second. These vibrations are reflected in the space between the pintle and barrel with the result that they must be withstood directly by the needle rollers. Consequently, the bearings 20 and 21 must not only rotate at high speed, but concurrently must be capable of resisting these periodic shock forces of the hydrostatic pressure.

In addition to these more apparent advantages of the needle roller mountings of the pintle and barrel, attention is directed to the following operating effects: the needle rollers at the ends of the pintle resist shock loads of the pressure cycles, thereby more accurately maintaining the clearance space between the pintle and barrel. By maintaining this clearance space between the pintle and barrel, they permit the limited capillary spacing of the pintle and barrel so that a capillary fluid film forms and provides hydrostatic balance on the opposite sides of the pintle. Further, they provide a pressure seal in two directions between the pintle and barrel during the pressure cycle. This results from the fact that they maintain a capillary oil film which is highly tenacious. Further, they effect a suction seal in both directions during the suction cycles, thus eliminating the entrance of air between the barrel and pintle and subsequently therefrom into the fluid circuit.

Due to their capillarity, they maintain lubrication between the pintle and barrel not only at the bearings themselves, but elsewhere along the entire valve portion during idle or zero stroke periods of the pump or motor, as well as during operation. Due to their capillary action, they provide a fluid cushion for purposes of starting when the pump is standing idle and effect pre-lubrication to prevent instantaneous seizure of the pintle and barrel upon starting and scoring of the pintle by contact with the barrel or particles of foreign matter accumulated between the pintle and barrel.

This latter results from maintaining lubricant between the barrel and pintle at all times. Due to the fact that an oil seal can be obtained by the bearings without completely blocking the end of the barrel bore, foreign particles can be discharged gradually over a period of time from the barrel bore. Due to the effective oil seal provided by the bearings, not only is the operating oil film between the valve portion of the pintle and barrel bore maintained during idle periods and prevented from draining away but also the gradual drainage of fluid from an individual cylinder during stop periods is prevented, so that air does not enter and replace the fluid. In case of ordinary bearings, such fluid would quickly drain from the cylinders and escape so that upon starting the operation, the motor would be subject to air entrapment and would require a considerable period of running until the air was entirely discharged from the system.

The seal provided by the bearings is effective during operation both against excessive slip or air which might otherwise be occasioned by machining inaccuracies or wear, thus greatly extending the useful life of the pump or motor. In view of these differences, it is readily apparent that the interposition of the capillary needle rollers

ers between the pintle and barrel perform, in this new surrounding, a function separate and apart from their function as bearings, which function results from a new cooperative relation with the barrel and pintle, as described above. For example, the effective seal against the escape of oil or the intake of air in the clearance space between the pintle and barrel is a result effected only in connection with the particular structure with which the bearings are associated. In fact, they make possible the elimination of the usual packing gland used for sealing between the barrel and pintle for retaining fluid, which glands are usually provided in the ordinary types of bearings.

It will be understood that various changes may be made in the construction and relative arrangement of the parts without departing from the invention as defined in the claims.

I claim:

1. In a radial piston pump or motor, a rotor, piston and cylinder assemblies carried thereby, means to actuate the assemblies upon rotation of the rotor, said rotor having an axial bore, a valve pintle received in said bore and having ports for cooperation with the assemblies, and elongated cageless capillary needle rollers interposed between the barrel and pintle, and said rollers being freely rotatable with respect to each other and anti-frictionally constraining the pintle and barrel to coaxial relation.

2. In a radial piston pump or motor, a rotor, piston and cylinder assemblies carried thereby, means to actuate the assemblies upon rotation of the rotor, said rotor having an axial bore, a valve pintle received in said bore and having a valve portion with ports intermediate its ends for cooperation with the assemblies, and elongated cageless capillary needle rollers interposed between the barrel and pintle at each end of the valve portion, and said rollers being freely rotatable with respect to each other and anti-frictionally constraining the pintle and barrel to coaxial relation.

3. In a radial piston pump or motor, a rotor having an axial bore; a piston and cylinder assembly carried by said rotor; means to actuate said assembly upon rotation of said rotor; a valve pintle received in said bore and having a port for cooperation with said assembly; and means for maintaining said rotor centered with respect to said pintle and for obstructing the slip of fluid between the pintle and the wall of the rotor bore beyond one end of the pintle comprising a set of closely contiguous elongated anti-friction bearing rollers interposed between said pintle and the wall of said bore adjacent one end of said pintle, the ratio of the length of said rollers to their individual diameter and the close spacing of said rollers being such as to provide between the rollers a plurality of capillary tube spaces adapted to maintain therein bodies of oil by cap-

illary attraction to thereby provide a combined anti-friction bearing and fluid seal.

4. In a radial piston pump or motor, a rotor having an axial bore; a piston and cylinder assembly carried by said rotor; means to actuate said assembly upon rotation of said rotor; a valve pintle received in said bore and having a port intermediate its ends for cooperation with said assembly; and means for maintaining said rotor centered with respect to said pintle and for obstructing the slip of fluid between the pintle and the wall of the rotor bore beyond the ends of the pintle comprising at each end of the pintle and respectively on opposite sides of said port a set of closely contiguous elongated anti-friction bearing rollers interposed between said pintle and the wall of said bore the ratio of the length of said rollers to their individual diameter and the close spacing of said rollers being such as to provide between the rollers a plurality of capillary tube spaces adapted to maintain therein bodies of oil by capillary attraction to thereby provide a combined anti-friction bearing and fluid seal.

5. A construction as set forth in claim 3 in which at least one of the following elements therein: namely, the rotor and the pintle, is formed with a recess for receiving said bearing rollers, the depth of the recess being no greater than the individual diameters of the bearing rollers.

6. A construction as set forth in claim 3 in which the rotor and the pintle are formed with radially aligned recesses for receiving said bearing rollers, the combined depths of said recesses being no greater than the individual diameters of the bearing rollers.

7. In a radial piston pump or motor, a rotor having an axial bore; a piston and cylinder assembly carried by said rotor; means to actuate said assembly upon rotation of said rotor; a valve pintle received in said bore with sufficient clearance to provide an intervening space for accommodating a body of oil, said pintle having a port for cooperating with said assembly; and means for centering said rotor with respect to said pintle and for maintaining a body of oil in said intervening space while preventing flow or slip of oil therethrough comprising a set of closely contiguous elongated antifriction bearing rollers interposed between and operatively engaging the pintle and the wall of said bore, the ratio of the length of said rollers to their individual diameters and the close spacing of said rollers being such as to provide between the rollers a plurality of capillary tube spaces whose capillary attraction draws oil from the adjacent portion of the space between the pintle and the rotor but resists flow or slip of oil completely past said bearing rollers.

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