Sept. 13, 1938. E. K. BENEDEK 2,129,642

VARIABLE DELIVERY PUMP OR MOTOR

Filed Aug. 3, 1934 3 Sheets-Sheet 1

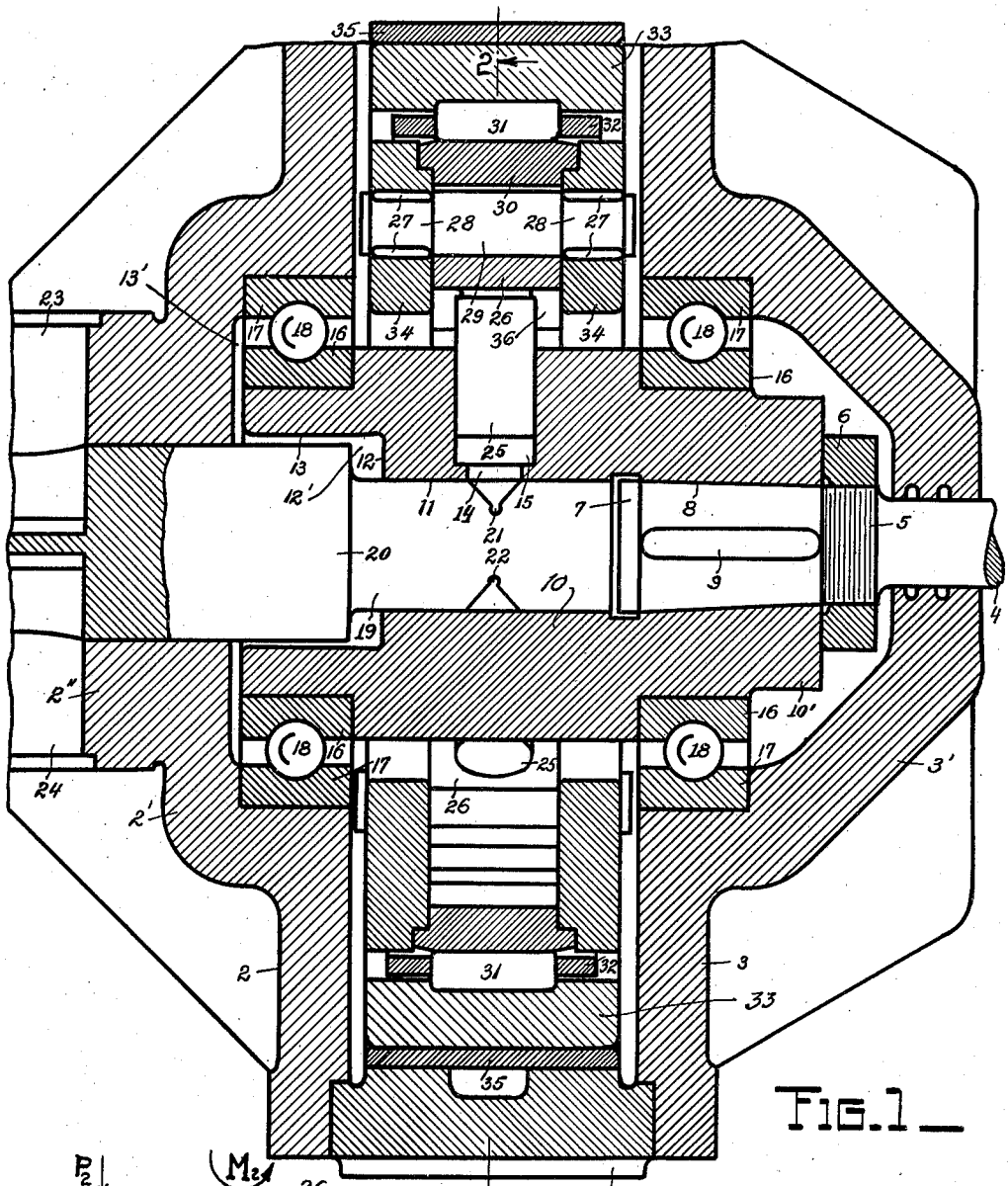


FIG. 1

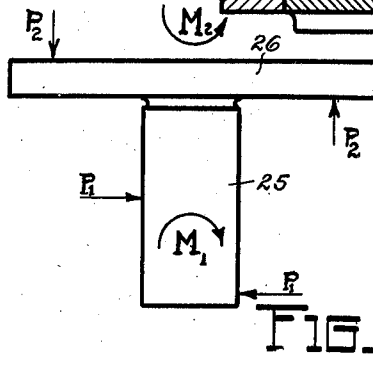


FIG. 3

Inventor

ELEK BENEDEK

By Soule & Leonard

Attorneys

Sept. 13, 1938. E. K. BENEDEK 2,129,642

VARIABLE DELIVERY PUMP OR MOTOR

Filed Aug. 3, 1934 3 Sheets-Sheet 2

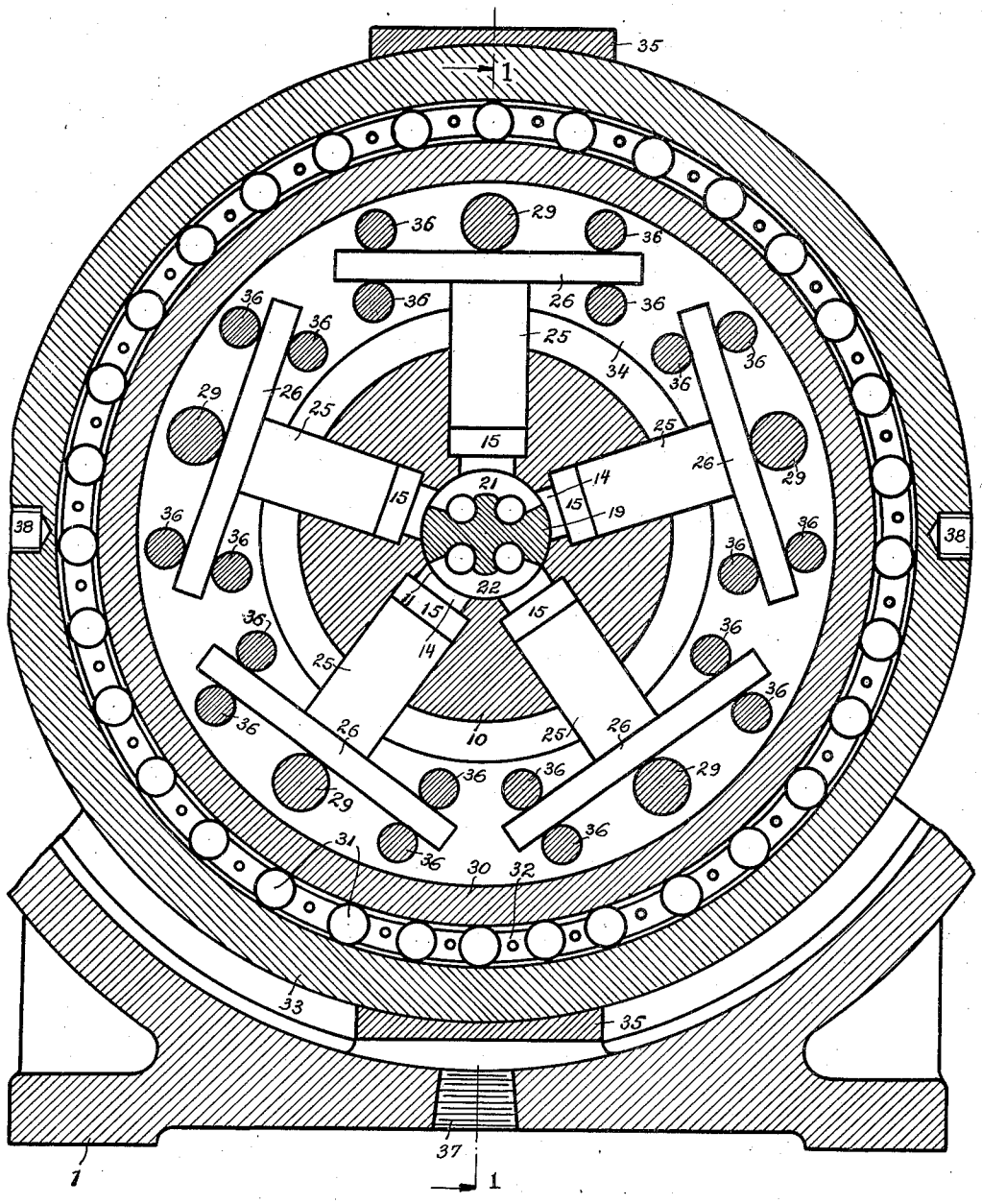


FIG. 2.

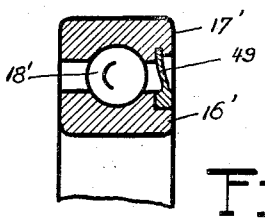


FIG. 9.

Inventor
ELEK BENEDEK.

By Soule & Leonard
Attorneys

Sept. 13, 1938. E. K. BENEDEK 2,129,642

VARIABLE DELIVERY PUMP OR MOTOR

Filed Aug. 3, 1934 3 Sheets-Sheet 3

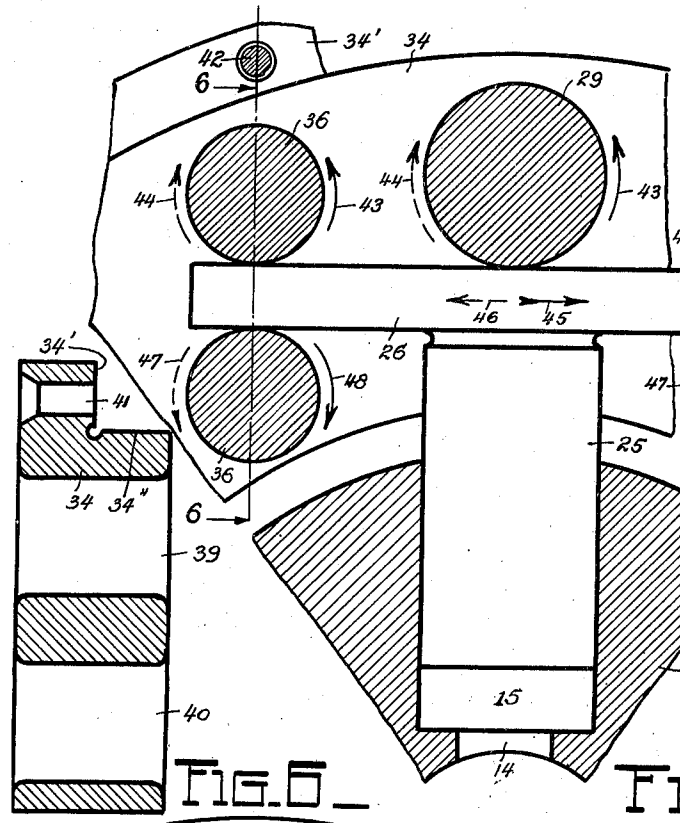


FIG. 4

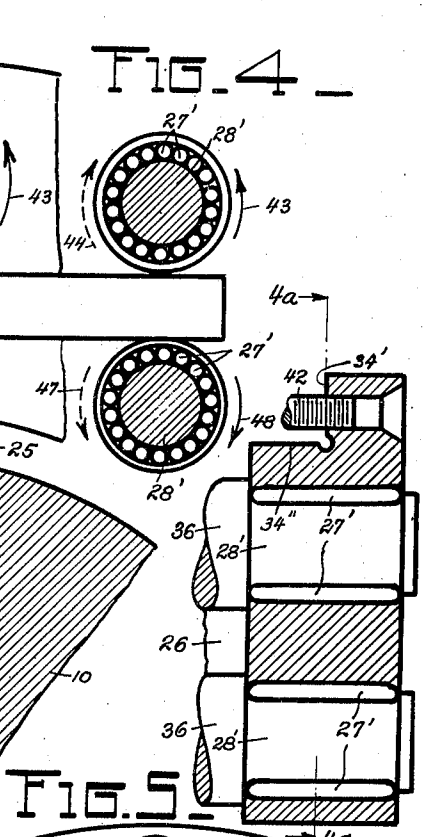


FIG. 5

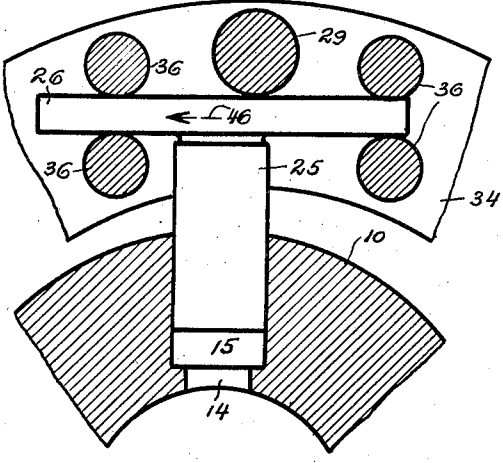


FIG. 6

FIG. 7

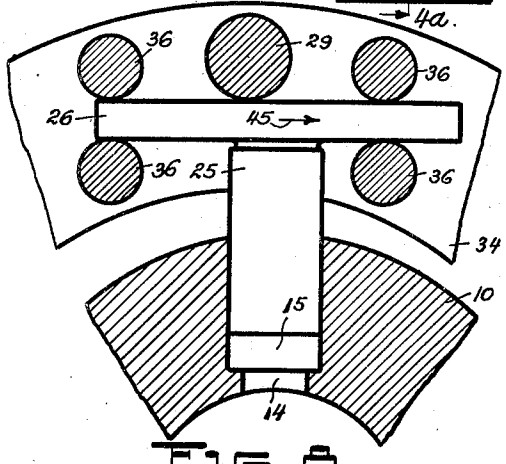


FIG. 8

Inventor
ELEK BENEDEK

By Louis & Leonard
Attorneys

Patented Sept. 13, 1938 2,129,642

UNITED STATES PATENT OFFICE

2,129,642

VARIABLE DELIVERY PUMP OR MOTOR

Elek K. Benedek, Bucyrus, Ohio

Application August 3, 1934, Serial No. 738,184

14 Claims. (Cl. 103—161)

This invention relates to pumps or motors of the variable delivery radial piston type and more particularly to that type of pumps or motors, wherein the radial piston load is carried by a reciprocating T-head, and the T-head itself is supported on anti-friction rollers contained in a common cage and cooperable with the reciprocating crosshead in such manner that said roller assembly is interposed between said reciprocating crosshead and a relatively stationary reaction plate or drum which is substantially parallel with the reciprocating crosshead, whereby said roller assembly is forced to roll between the parallel track surfaces which are in relative reciprocation, thus the roller-cage assembly also must perform a relative reciprocation in regard to both of said track elements. As it is obvious from the kinematics of such combination, that the rollers will travel a distance determined by the distance of reciprocation of the crosshead, which actually results in the reciprocation of the roller-cage assembly relative to the crosshead one-half the distance of reciprocation of the crosshead. The total amount of travel of the crosshead in either direction is, of course, equal to the stroke of the piston at a given stroke setting. In the pumps of this type herein above referred to, there is still sliding friction between the inner face of the reciprocating crosshead and cooperating relatively stationary surface of the polygonal driving head and the driving force is carried only on an oil film, provided by and from a spray of operating fluid, which escapes as slip through and from the outer ends of the pistons. For the sake of hydraulic efficiency the escapage of fluid as slip is minimized as much as possible by close fitting of the pistons and when slip does occur the fluid is, of course, thrown outwardly centrifugally, away from the innermost sliding surfaces, thus greatly reducing lubrication of these surfaces. Thus the partial lubrication of the radially inwardly disposed sliding surfaces of the reciprocating crossheads often becomes insufficient to withstand the twisting effect of the driving torque on the driving crosshead against its guiding surfaces and results in gaulding of such sliding surfaces, thus preventing further reciprocation of the crossheads and wrecking the entire mechanism.

A further defect of above referred to roller and cage load transmission lies in the fact that there are no means provided to positively control the roller and cage assembly with respect to the centerline of the piston and its T-shaped crosshead, except a longitudinal slot in the longitudinal axis of said cage, and a stop screw fastened to the reaction plate or roller track of the rollers. It is evident, that due to the enormous acceleration of the cage and roller assembly, and the cocking load of the driving rotating force, the cage tends to become displaced from its theoretical dynamic position with respect to said stop screw and crosshead so that sooner or later a pounding of the cage against said stop takes place which may result in one or more of the rollers leaving the cage, blocking the further reciprocation of the crossheads and wrecking the pump.

The inherent faults of such structures as hereinabove referred to are further aggravated by the fact, that the leading inner ends of the driving crossheads, when cocked or tilted by the driving torque, at the contact points of the crosshead create considerable mechanical friction, cause loss of useful power, and decrease the efficiency of the machine.

In order to alleviate the above encountered deficiencies, I have provided a novel pump structure, in which the hydraulic piston load, the torque forces of the driving couple acting on the crossheads, as well as the piston actuating forces during the suction cycles are all taken up in a more efficient manner and by a novel load transmission structure; which structure, while it provides frictionless support for the cross heads in withstanding all three of the previously enumerated forces, is relatively stationary with respect to the reciprocating crossheads and therefore may be very rigidly supported.

It will be hereinafter noted that in addition to the improved results and simplified structure of my novel mechanism, there are several mechanical features which simplify its manufacture as a precision instrument, in permitting more simple manufacturing expedients, for example, centerless grinding etc.

A further object of the invention is to dispose all bearings in a more compact arrangement, to thereby eliminate elastic deflections from the working structure of the pump and reduce noise and vibrations so that the rotary pump may properly, efficiently and silently execute its hydraulic functions for long periods of time.

Other advantages will be described in connection with the accompanying drawings, forming a part of this specification.

In the drawings:

Fig. 1 is a vertical sectional view of a pump, constructed in accordance with the invention, taken on line 1—1 in Fig. 2;

Fig. 2 is a transverse fragmentary sectional view taken on line 2—2 in Fig. 1;

Fig. 3 is a front elevation of my integral T-head solid piston shown separately from the pump to illustrate various forces on the same;

Fig. 4 is a fragmentary sectional view taken on line 2—2 in Fig. 1 with a part of the cheek plate broken away, to show the bearing mounting of my novel crosshead operating means, part of the view being taken along the line 4a—4a in Fig. 5;

Figs. 5 and 6 are fragmentary sectional views of the cheek plate taken on line 6—6 in Fig. 4, the rollers and their needle bearings being removed from the plate in Fig. 6; and

Figs. 7 and 8 are fragmentary sectional views similar to Fig. 4, illustrating two relative positions of the piston and anti-friction load transmitting means.

Fig. 9 is a fragmentary cross sectional view of a slightly modified form of bearing for use in connection with the cylinder block.

The mechanism disclosed for illustration is enclosed within a casing comprising a central body section 1 and end heads 2 and 3, which are carefully fitted to the end faces of the cylindrical body portion through appropriate shoulders and recesses respectively, to provide for substantially permanent alignment for the central axis of the entire case during the operation of the machine.

The end head 2 is provided with a central hub portion 2'' to receive the pump valve, as well as carry the main fluid pipe connections of the pump, for suction and delivery of operating fluid. The end heads have bell-shaped portions 2' and 3' respectively, which are appropriately recessed for rigidly supporting stationary rings 17 of anti-friction bearings 18 for supporting a rotary cylinder block or barrel 10. The recesses also provide axial abutment surfaces for the respective bearing rings. The inner bearing rings 16 tightly embrace and axially abut suitable shoulders on the cylinder barrel—the spaced bearing arrangement forming a balanced and sturdy support for the barrel as well as holding the barrel in place against axial shifting.

The cylinder barrel 10 is provided with a plurality of radial bores as at 15 in a well known manner for pistons 25, which cylinders are in communication with main pump ports 21 and 22 respectively through radial ports 14, one for each piston cylinder.

An impeller shaft 4 projects through a central bore of the end head 3 into the pump casing, terminating inwardly at 7, the inner end portion fitting a tapered bore 8 of the cylinder barrel 10, and being drivingly rigid therewith through the medium of a feather key 9. A suitable locknut 6 on the threaded portion 5 of the shaft pulls the tapered surfaces together to form a very rigid connection between the cylinder barrel and shaft. The tapered and keyed connection is preferably in the transverse plane of the bearing 18 surrounding the smaller end 10' of the cylinder barrel thereby assuring proper support for the shaft at the zone of application of driving torque to the barrel to the end of assuring quiet rotation of the cylinder block.

The cylinder barrel 10 adjacent its tapered bore 8, is provided with another slightly tapered bore as at 11 to receive the reduced diameter portion 19 of a central valve or pintle 20. The reduced portion of the pintle is tapered in order to limiit the escape of pressure fluid axially, as well as to prevent air suction around the peripheral surfaces of the pintle. In order to further seal the pintle engaging end of the cylinder barrel, between the pintle and bore 11 of the barrel, I provide a circular fluid chamber as at 12', by counterboring the block at 12—13, which chamber will become filled by pressure fluid escaping from the barrel to prevent air suction, during the suction strokes, particularly when the machine is operating as a pump and is set for short piston stroke service, and also to maintain the desired slip pressure. The chamber 12' is constituted by a free uninterrupted clearance space or oil-receiving void between the longer diameter portion or shank of the pintle and the wall of the barrel counterbore 13. As shown in Figure 1, the pintle shank extends into but terminates short of the inner end of the counterbore.

To the end of maintaining such pressure in the chamber 12' and prevent sealing fluid loss centrifugally, I preferably use, instead of the bearings 18 above noted, a special bearing 18' at this end of the cylinder barrel 10, a detail of which is shown in Fig. 9. This bearing is substantially sealed by a ring plate 49 at one transverse end face, the plate 49 being suitably secured to one race member (16' e. g.) and continuously extending into a peripheral groove in the other member (17' e. g.). The ring plate may be snapped into place to assemble it into the bearing. Thus a substantial volume of operating fluid under pressure can be maintained from the slippage of the pump to prevent sucking air on the suction strokes of the pistons. This is particularly important after long continued service has worn either the pintle or barrel, and has increased the clearance therebetween. The excess slip of pressure fluid leaves the pump casing through a suitable drain orifice 37 in the casing section 1.

Reversible mains of the pump 23 and 24 respectively in the hub portion 2'' provide for inlet and outlet pipe connections.

The pistons 25 have T-heads 26 rectangular in cross section, integrally secured to respective pistons proper and adapted to operate the pistons during rotation of the piston and cylinder assembly in the following manner.

The stroke controlling mechanism comprises a pair of cheek plates 34, one at each side of the reciprocating crossheads supported in the casing section 1 by an anti-friction bearing comprising outer ring 33, inner ring 30, and a plurality of rollers 31, caged in an appropriate retainer 32, The cheek plates 34 are secured to the inner ring 30 of the antifriction bearing as illustrated in Figs. 5 and 6. It is evident from these figures, in connection with Fig. 1, that the outer circumferences of the plates 34 are recessed to obtain circular axial and radial shoulders 34'' and 34', respectively, as better illustrated in Figs. 5 and 6 and that the ring 30 is complementarily formed so that the plates may be assembled thereon as by press fitting onto said surfaces 34'' and against the shoulders 34'. The plates 34 may be retained in their respective shoulder seats by screws 42, extending through appropriate holes 41 of the cheek plates into threaded openings of the inner race 30 aligned therewith. To facilitate pulling the cheek plates off the press shoulders of the ring 30 when desired, a few tapped holes may be provided in the outer flanges of the plates so that screws engaging the threads may abut the ring surfaces which abut the shoulder faces 34' of the plates. Thus it will be seen that a very rigid mounting is secured for the cheek plates 34, the rigid cheek plate and bearing ring assembly being well adapted to be supported by a more or less standard type of anti-friction bearing dis-

posed in the central plane of the machine which coincides with the axes of the pistons. The bearing shown is adapted for high speed and excessive pressures. The cheek plates closely embrace the piston crossheads to prevent axial displacement of the cheek plate and bearing ring assembly. Thus the cheek plates provide radial flanges which, with the ring **30**, define a circumferentially closed trough for retaining lubricant. The rollers of the bearing operate in a suitable annular raceway in the ring **33** so as not to become axially displaced.

Pursuant the requirements of a highly special structure of the class described, I further provide as a reactance support for the piston crosshead, a plurality of rollers mounted at their ends in the parallel walls of the cheek plates **34**, the rollers having their axes parallel to the rotational axis of the pump.

A main or load roller **29** is provided in normal radial alignment with the reciprocating piston, as shown in Fig. 4, having somewhat reduced end portions as at **28**, said end portions being mounted in heavy duty bearing means **27** in axially aligned bores **28'** of the cheek plates **34**. For the purpose of illustration, a plurality of small (needle) rollers are used without mechanical cages, and the small rollers are located against axial displacement in undercut recesses provided in the roller ends **28**. It is understood, however, that any high grade bushing may be used instead of the needle rollers **27**, to thereby reduce the price of the unit, and increase its commercial utility, particularly in cases where the centrifugal pressure oil bath, confined by the cheek plates **34** and by the inner race **30**, is available for ample lubrication of the bearings **27**, and where exchange and replacement of the rollers **29** may be easily effected. The rollers are subject to mass production by centerless grinding which cuts their cost to a point where, due to interchangeability, the needle rollers are not prohibitively expensive for this class of mechanism. The axial position of each of the rollers **29** is secured by having its center portion slightly larger in diameter, than the supporting bores in the cheek plates.

Referring again to Fig. 4, a pair of torque rollers **36**—**36** are provided at each end of the reciprocating crosshead **26**, the axes of each pair of the rollers being disposed in planes parallel to the inner and outer crosshead surfaces respectively and extending parallel to the rotational axis of the barrel **10** so as to engage the crosshead for rolling to and fro therealong tangentially of the path of rotation of the T-heads. The reduced end portions of the torque rollers **36** are constructed as previously described—see Fig. 5—slightly recessed as at **28'** to thereby accommodate the needle bearings **27'**. For convenience, the large roller **29** will be considered the pressure roller or main roller, due to the fact that its main function is, particularly, to take up the hydraulic pressure of the piston. As will be observed in Fig. 4, the main load will be inherently taken up by the main roller due to the unavoidable slight deflection of the crosshead. Referring to Fig. 3, it will be seen that each piston and T-head is subjected not only to radial pressure but also to driving torque. Assuming the rotor is driven clockwise, each piston will be subjected to the driving torque pressure P_1 which tends to rotate the piston about a transverse axis, as indicated by the arrow M_1, thus cocking the head forwardly. This moment must be resisted by an equal and opposite moment M_2 if equilibrium and driving of the reactance rotor are to be maintained. The moment M_2 is provided by the two reactionary forces P_2, one such force being exerted by one outer torque roller **36**, and the other force P_2 by an inner torque roller **36** at the opposite side of the piston axis from the first torque roller. This relation to the torque rollers is effected by the forward cocking of the piston T-head and resultant warping into positive contact with one inner and one outer roller. If the roller **29** were omitted, the radial outward forces or hydraulic load would have to be resisted by both of the outer rollers **36**, due to the fact that the T-head would bow outwardly, tending to counteract the engagement with the inner roller **36** produced by the cocking of the head. As a result of such omission, the effective positive driving connection between the crosshead ends and the rollers **36** could not be provided but reliance would have to be placed on rolling frictional resistance. The application of hydraulic load pressure reactance by the roller **29** to either side of the piston axis may assist in withstanding the moment M_1 or may aggravate the condition slightly, depending upon which side of the piston axis it is disposed at the particular incident, but it restrains outward bowing of the piston T-head so that the T-head can warp and cooperate properly with one outer and one inner roller **36**, as described, for providing anti-friction positive torque transmission between the barrel **10** and the reactance rotor. Naturally the functions of the rollers **29** and **36** are not limited entirely to radial pressure and torque respectively for the pistons are only aligned with the rollers **29** a small portion of the time and the deflection of the crossheads is very slight, but a preliminary analysis of the operation of my novel combination suggests a performance as hereinabove outlined.

In regard to the motion of the rollers, it is evident that upon movement of the crosshead in the direction of the arrow **45**, this is accompanied by the simultaneous rotation of all the rollers according to the full line arrows **43** and **48** respectively, and, upon movement in the direction of the arrow **46**, a reverse rotation of the rollers will take place as shown by arrows **44** and **47** respectively.

One of the outstanding advantages of my small roller bearing construction (**27** and **27'**) is that the rollers gradually move in conformity with the arrows **43** and **44**, consequently, any wear caused by the outwardly acting piston load will tend to equalize itself by transference of load and wear from the smaller rollers to the large roller. It will be seen that due to the free and unrestricted operation of the rollers about their longitudinal axes they tend to uniformly distribute wear, both on the crosshead engaging surfaces and on the needle bearings and raceways thereof.

As above described, the entire stroke control mechanism or secondary rotor is mounted in a completely antifriction bearing, and, I may provide for controlling the piston stroke and reversing the pumping operation by slidably supporting the outer bearing ring **33**. To facilitate accurate alignment of the control connections **38** (Fig. 2) with the transverse planes of the pistons and the valves **21**—**22**, I provide detachable bearing blocks **35** which support said bearing ring and slide on appropriate bearing pads of the body **1**.

When the stroke control assembly and the cylinder barrel are in the concentric positions shown in Figs. 1 and 2, the entire pump mecha-

nism rotates as one body. No reciprocation or relative motion takes place between the working parts of the pump, thus no fluid will be pumped in or out of the mains 23 and 24. When, however, the cylinder barrel and stroke control mechanism is shifted into eccentric relation, as shown in Figs. 7 and 8, the pistons reciprocate and will deliver a continuous stream of fluid in a direction determined by the eccentric setting of the axis of the cheek plate and roller assembly at one side or the other of the cylinder barrel axis.

It will be seen that I have substantially eliminated sliding friction from the crosshead operating mechanism with consequent saving of power and wear on the parts, thereby greatly increasing the efficiency of this type of mechanism and making it possible to safely greatly increase the running speed and output of the machine.

I claim:

1. In a rotary radial piston pump or motor of the character described, including a rotatable barrel, a series of circumferentially spaced radial cylinders therein, a valve pintle coaxial with the barrel and in valving cooperation with the cylinders, and pistons having T-heads and reciprocable in the cylinders respectively, a reactance rotor eccentric to the pintle and surrounding the pistons and having radially extending flanges disposed alongside the T-heads, load rollers respective to the pistons, each roller being anti-frictionally mounted at its ends in the said flanges with its axis parallel to the pintle axis and spanning the space between the flanges, and each roller being in rolling contact intermediate its ends with the outer surface of one of the T-heads normally at the axis of the associated piston for oscillation therealong in both directions from the axis of the piston associated therewith, and inner and outer torque transmitting rollers for each piston correspondingly anti-frictionally mounted in the flanges for rolling contact with the inner and outer surfaces respectively of the associated T-head, one inner roller and one outer roller at each end portion of the T-head, whereby free tangential oscillation of the heads relative to the reactance rotor is permitted during concurrent load and torque transmission thereto by the T-heads.

2. In a rotary radial piston pump or motor of the character described, including a rotatable barrel, a series of circumferentially spaced radial cylinders therein, a valve pintle coaxial with the barrel and in valving cooperation with the cylinders, and pistons having T-heads and reciprocable radially in the cylinders respectively, a reactance rotor eccentric to the pintle and surrounding the pistons and having radially extending flanges disposed alongside the T-heads, load rollers respective to the pistons, each roller being rotatably mounted anti-frictionally at its ends in the said flanges with its axis parallel to the pintle axis and spanning the space between the flanges, and each being in rolling contact with the outer surface of one of the T-heads adjacent the axis of the piston associated therewith, and inner and outer torque transmitting rollers for each piston correspondingly anti-frictionally mounted at their ends in the flanges and being in rolling contact with the inner and outer surfaces respectively of the associated T-head, one inner roller and one outer roller at each end portion of the head, and anti-friction roller bearings surrounding the ends of said rollers and anti-frictionally supporting the same in said flanges, whereby positive torque transmission is effected while maintaining free tangential oscillation of the heads relative to the reactance rotor.

3. In a pump or motor of the character described, a rotatable barrel having a series of circumferentially spaced radial cylinders, valve means for the cylinders, radially reciprocable pistons received in the cylinders and having head portions at their outer ends extending fore and aft of their path of rotation, a reactance rotor surrounding said pistons and eccentric to the barrel and having a pair of radial flanges axially separable from each other and spaced apart to lie along opposite sides of the head portions, anti-friction rollers having their axes extending parallel to the axis of said rotor, each of said rollers having an enlarged central portion disposed between the flanges and in rolling contact with a portion of one of the piston head portions with which associated, and reduced diameter circumferentially recessed end portions defining radial shoulders at the ends of said central portion, aligned bores in said radial flanges for receiving the recessed end portions of the rollers axially and being of larger diameter than said recessed portions for accommodating cageless roller bearings in surrounding relation to said recessed portions and being of smaller diameter than the central portion, and means to secure said flanges in fixed axially spaced position with the recessed end portions of the rollers so mounted in the bores and the said shoulders between the central portion and the end portions constraining the rollers against axial dislocation and the recessed portions and flanges confining said needle roller bearings against axial displacement.

4. In a rotary radial piston pump or motor of the character described, including a rotatable barrel, a series of circumferentially spaced radial cylinders therein, a valve pintle coaxial with the barrel and in valving cooperation with the cylinders, and pistons having T-heads and being reciprocable in the cylinders respectively, a reactance rotor eccentric to the pintle and surrounding the pistons, inner and outer torque transmitting rollers for each T-head, said rollers being carried by the reactance rotor and secured thereby in fixed position circumferentially of the reactance rotor and having their axes parallel to the rotor axis, said rollers being in rolling contact with inner and outer surfaces respectively of the associated T-head, one inner roller and one outer roller at each end portion of the associated T-head, the axes of the inner and outer rollers at the same end of a T-head lying substantially in a plane parallel to the plane defined by the associated piston axis and the axis of revolution of the barrel, and load transmitting means carried by the reactance rotor and engaging the outer surfaces of the T-heads respectively normally at the axis of the associated piston.

5. In a radial piston pump or motor of the character described including a rotatable barrel, a radial cylinder therein, a valve pintle for the barrel in valving cooperation with the cylinder, a piston member reciprocable radially in the cylinder, and a reactance rotor member eccentric to the pintle and surrounding the barrel, means for driving said piston member on one stroke including a pair of axially separable radial flange portions on said rotor member spaced apart axially for receiving snugly therebetween the outer end portion of the piston member, a roller having its axis parallel to the pintle axis and hav-

ing an enlarged central portion disposed between and substantially spanning the space between the flanges and in driving relation to the piston member for driving the same, and reduced diameter end portions engaged with the flanges and defining with the central portion annular shoulders engaging the flanges for constraining the roller from axial dislocation, said roller being in rolling engagement with said piston member and having its axis fixed with respect to the reactance rotor member and being anti-frictionally mounted for rotation about its axis in said reactance rotor member.

6. In a rotary radial piston pump or motor of the character described, including a barrel having radial cylinders, a valve pintle in valving cooperation with the cylinders, circumferentially spaced radial pistons carried thereby, a reactance rotor eccentric to the barrel for reciprocating the pistons consequent upon concurrent rotation of the barrel and rotor, heads on the pistons respectively, each head extending fore and aft of its path of rotation in each direction from the axis of the piston with which it is associated, means associated with the heads respectively for transmitting torque from the pistons to the reactance rotor through the medium of said heads, each of said means comprising a pair of outer torque rollers journalled in the reactance rotor and spaced one in advance and one in the rear of the longitudinal mid-portion of the associated head, said rollers being journalled to rotate about axes fixed with respect to the reactance rotor, a correspondingly arranged and mounted pair of inner torque rollers, the outer rollers being cooperable with the outwardly facing surface of the head and the inner rollers being cooperable with inwardly facing portions of the head, and load transmitting means carried by the rotor and engaging the heads respectively at the said mid-portions of the said heads.

7. In a rotary radial piston pump or motor of the character described, including a barrel having radial cylinders, valve means for the cylinders, circumferentially spaced radial pistons carried thereby, and a reactance rotor eccentric to the barrel for reciprocating the pistons consequent upon concurrent rotation of the barrel and rotor, T-heads respective to the pistons, each head extending fore and aft of its path of travel in each direction from the axis of the piston with which it is associated, sets of torque rollers anti-frictionally mounted in the reactance rotor and having their axes parallel to the rotor axis, said rollers being arranged one set to each head, one roller of each set being in rolling torque transmitting engagement with an outwardly disposed face portion of the associated head at one end thereof, and the other roller being in rolling torque transmitting engagement with an inwardly disposed face portion of the associated head at the opposite end thereof concurrently, and a load transmitting roller for each head anti-frictionally mounted in the reactance rotor for rotation about an axis fixed with respect to the reactance rotor and being in rolling engagement therewith adjacent the piston axis.

8. In a pump or motor of the character described, an inner rotor, an outer rotor, piston and cylinder assemblies carried by one rotor and reciprocable consequent upon concurrent rotation of the rotors, valve means for the cylinders, each assembly including a piston member and a cylinder member, one of said members of each assembly having a head, axially separable radial flanges on the other one of said rotors spaced apart axially and accommodating the heads therebetween for tangential oscillation relative thereto, a plurality of anti-friction rollers having their axes extending parallel to the axis of rotation of the rotors, each of said rollers having a relatively large diameter central portion disposed between the flanges and in rolling contact with one of said heads, and reduced diameter end portions defining radial shoulders at the ends of the central portion, aligned bores in said flanges receiving the reduced diameter end portions of the rollers axially and being of larger diameter than said end portions for accommodating elongated cageless roller bearings in surrounding relation to said reduced end portions, and being of smaller diameter than the central portion, elongated cageless roller bearings for the ends of the rollers, means to secure said flanges in fixed axially spaced position with the reduced end portions of the rollers so mounted in the bores and said shoulders constraining the roller bearings from inward axial displacement, and means for constraining the roller bearings from outward axial displacement.

9. In a pump or motor of the character described, an inner rotor, an outer rotor, piston and cylinder assemblies carried by one rotor and reciprocable consequent upon concurrent rotation of the rotors, valve means for the cylinders, each assembly including a piston member and a cylinder member, one of said members of each assembly having a head, axially separable radial flanges on the other one of said rotors spaced apart axially and accommodating the heads therebetween for tangential oscillation relative thereto, a plurality of anti-friction rollers in operative engagement with said heads respectively and having their axes extending parallel to the axis of rotation of the rotors, aligned bores in said flanges receiving the end portions of the rollers axially and being of larger diameter than said end portions for accommodating elongated cageless roller bearings in surrounding relation to said end portions, elongated cageless roller bearings for the ends of the rollers, means to secure said flanges in fixed axially spaced position with the end portions of the rollers so mounted in the bores, and means for constraining the roller bearings from axial displacement.

10. In a rotary radial piston pump or motor of the character described, corotatable eccentric rotors, radial cylinders carried by one rotor, valve means therefor, pistons reciprocable in the cylinders respectively, each piston having a head with a load transmitting surface extending fore and aft in the direction of rotation, a load transmitting roller for each piston anti-frictionally mounted at its ends in the other rotor for rotation about an axis fixed with respect to the said other rotor and being in rolling load transmitting engagement with the load transmitting surface of the associated head substantially at the axis of the piston, and means in addition to said load transmitting roller operatively connecting the head and the said other rotor for effecting synchronous driving relation between the rotors.

11. In a rotary radial piston pump or motor of the character described, co-rotatable eccentric rotors, radial cylinders carried by one rotor, valve means therefor, pistons reciprocable in the cylinders respectively, each piston having a head with a load transmitting surface extending fore and aft in the direction of rotation, rollers respective to the pistons and respectively anti-frictionally

mounted in the other rotor for rotation about an axis fixed with respect to the said other rotor, said rollers being in rolling engagement each with an associated piston head and normally at the axis of the associated piston.

12. In a rotary radial piston pump or motor of the character described, a rotatable barrel; a series of circumferentially spaced radial cylinders therein; a valve pintle coaxial with the barrel and in valving cooperation with the cylinders; pistons reciprocable in the cylinders respectively; heads on said pistons respectively having radially inner and outer thrust-transmitting surfaces extending tangentially fore and aft of the respectively associated piston axes in the direction of rotation of said barrel; a reactance rotor eccentric to the pintle and surrounding the pistons; inner and outer torque-transmitting rollers for each piston head, said rollers being carried by the reactance rotor and journalled therein to rotate about axes fixed circumferentially with relation to the reactance rotor and having their axes parallel to the rotor axis, said rollers being in rolling contact with the inner and outer thrust-transmitting surfaces of the associated piston head, one inner roller and one outer roller being at each end of the associated piston head; and load-transmitting means carried by the reactance rotor and engaging the outer surfaces of the piston heads respectively normally at the axis of the associated piston.

13. In a rotary radial piston pump or motor of the character described, a rotatable barrel; a series of circumferentially spaced radial cylinders therein; a valve pintle coaxial with the barrel and in valving cooperation with the cylinders; pistons reciprocable in the cylinders respectively; heads on said pistons respectively having radially inner and outer thrust-transmitting surfaces extending tangentially fore and aft of the respectively associated piston axes in the direction of rotation of said barrel; a reactance rotor eccentric to the pintle and surrounding the pistons; inner and outer torque-transmitting rollers for each piston head, said rollers being carried by the reactance rotor and journalled therein to rotate about axes fixed circumferentially with relation to the reactance rotor and having their axes parallel to the rotor axis, said rollers being in rolling contact with the inner and outer thrust-transmitting surfaces of the associated piston head, one inner roller and one outer roller being at each end of the associated piston head, the axes of the inner and outer rollers at the same end of a piston head lying substantially in a plane parallel to the plane containing the associated piston axis and the axis of revolution of the barrel; and load-transmitting means carried by the reactance rotor and engaging the outer surfaces of the piston heads respectively normally at the axis of the associated piston.

14. In a rotary radial piston pump or motor of the character described, including a barrel having radial cylinders, a valve pintle in valving cooperation with the cylinders, circumferentially spaced radial pistons carried thereby, a reactance rotor member eccentric to the barrel for reciprocating the pistons consequent upon concurrent rotation of the barrel and rotor member, head members on the pistons respectively, each head member extending fore and aft of its path of rotation in each direction from the axis of rotation of the piston with which it is associated, means associated with the head members respectively for transmitting torque from the pistons to the reactance rotor member through the medium of said head members, each of said means comprising a pair of outer torque rollers journalled in one of said members and spaced one in advance and one in the rear of the longitudinal midportion of the associated head member for rotation about axes fixed with respect to the member in which the rollers are journalled, a correspondingly arranged and mounted pair of inner torque rollers, the outer rollers being cooperable with the outwardly facing surface of the head member and the inner rollers being cooperable with the inwardly facing portions of the head member, and load-transmitting means carried by the rotor and engaging the head members respectively at the midportions thereof.

ELEK K. BENEDEK.