June 27, 1939. E. K. BENEDEK 2,164,171

HYDRAULIC PUMP OR MOTOR

Filed Oct. 16, 1935 4 Sheets-Sheet 1

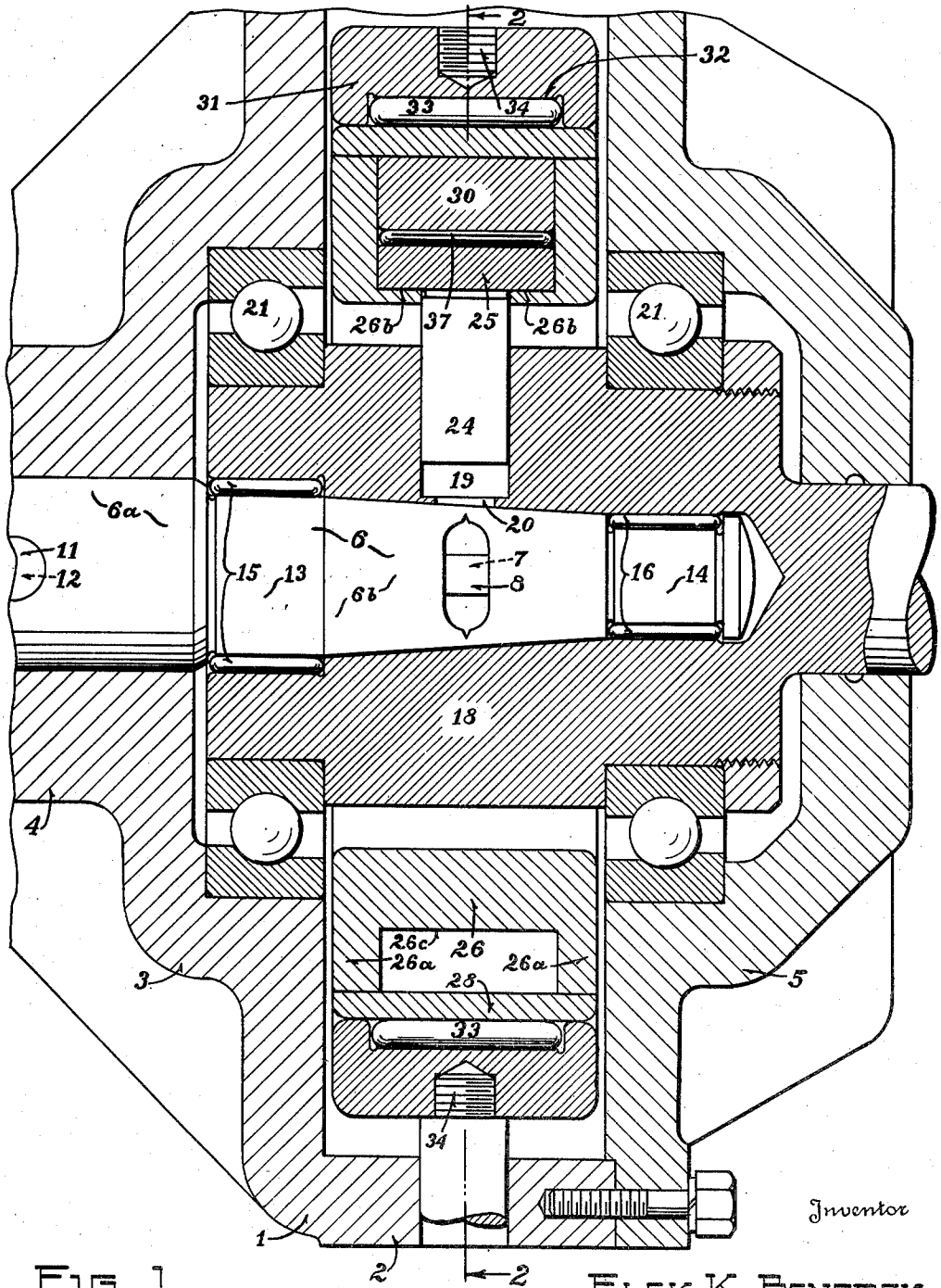


FIG. 1.

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HYDRAULIC PUMP OR MOTOR

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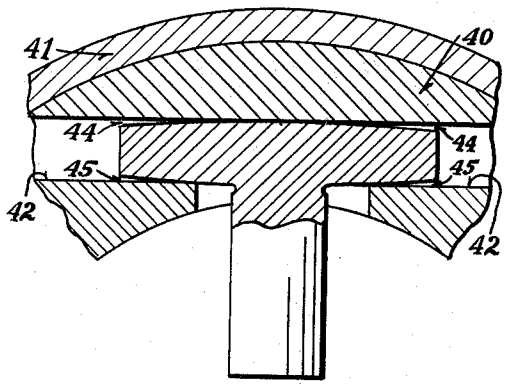


FIG. 3.

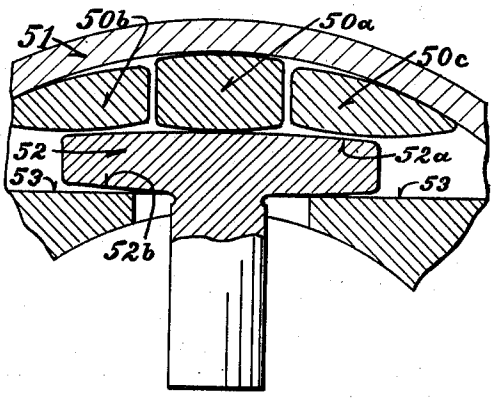


FIG. 4.

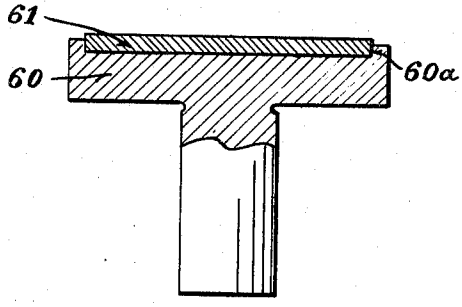


FIG. 5.

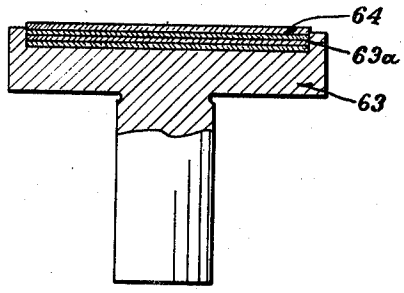


FIG. 6.

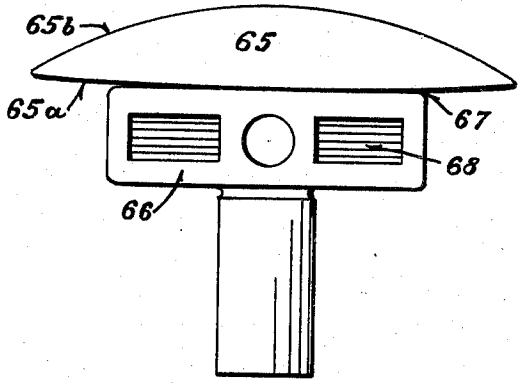


FIG. 7.

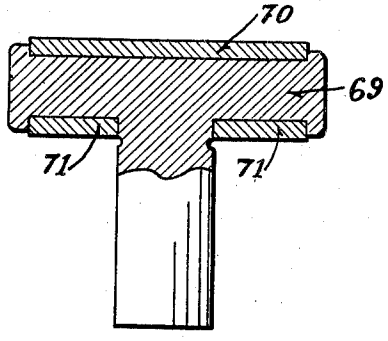


FIG. 9.

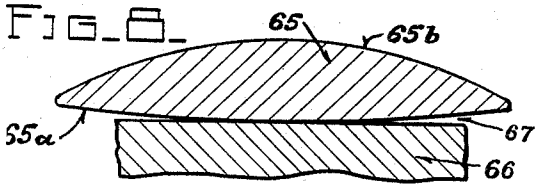


FIG. 8.

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HYDRAULIC PUMP OR MOTOR

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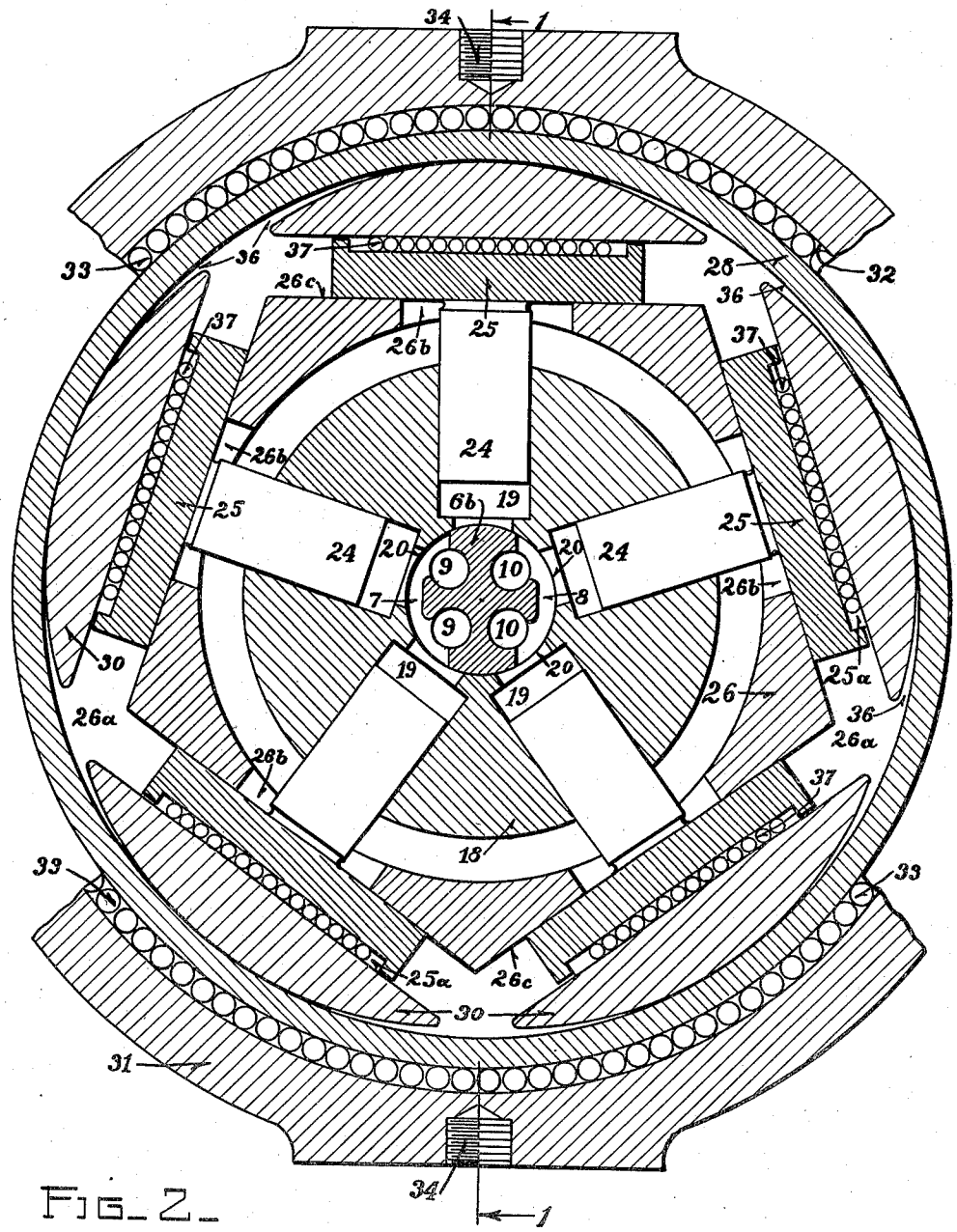


FIG. 2.

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HYDRAULIC PUMP OR MOTOR

Filed Oct. 16, 1935 4 Sheets-Sheet 4

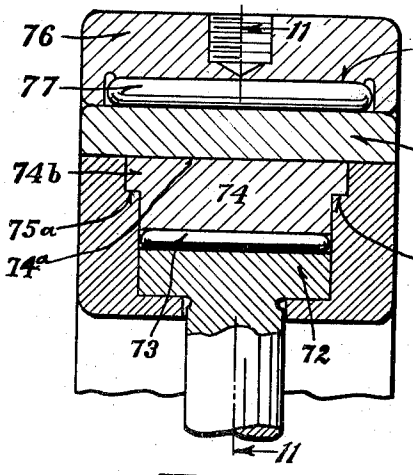


FIG. 10.

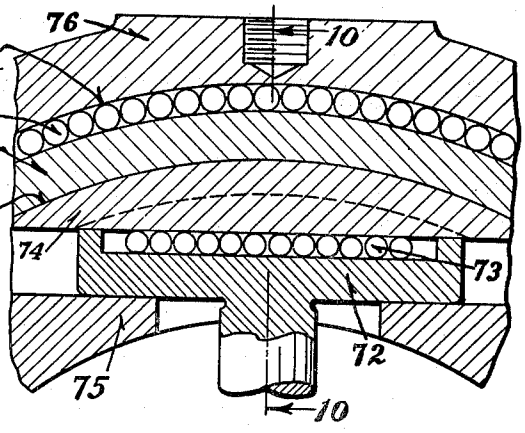


FIG. 11.

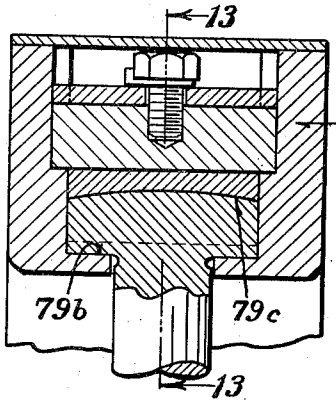


FIG. 12.

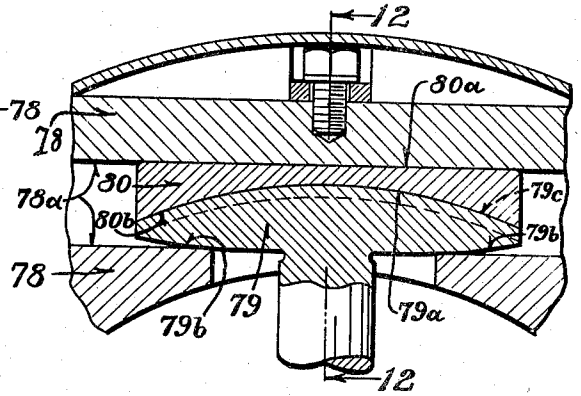


FIG. 13.

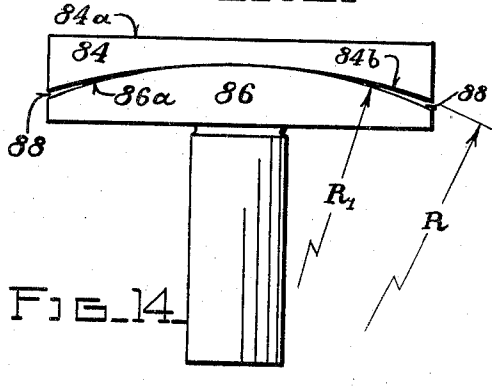


FIG. 14.

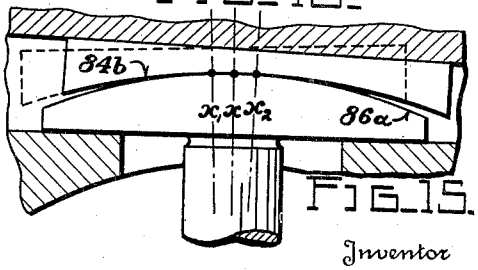


FIG. 15.

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Patented June 27, 1939 2,164,171

UNITED STATES PATENT OFFICE

2,164,171

HYDRAULIC PUMP OR MOTOR

Elek K. Benedek, Bucyrus, Ohio

Application October 16, 1935, Serial No. 45,325

15 Claims. (Cl. 103—161)

This invention relates to pumps and motors of the rotary radial piston type in which T-head pistons are employed for effecting positive driving relation through the medium of the pistons, between the primary and secondary rotors.

The principal object of the present invention is to provide combined hydrodynamic and capillary oil film bearings in cooperation with the piston T-heads and rotary reactance through the medium of which films hydraulic load and torque are transmitted to and from the T-head in the most efficient manner and by which an elastic and uniform distribution and application of eccentric load and torque forces are maintained.

More specifically, an object of the present invention is to provide a piston reactance interponent for cooperation with the T-head and rotary reactance, with which interponent the T-head is cooperated through the medium of wedge-shaped hydrodynamical oil films and the interponent is in oscillating rolling cooperation with the reactance rotor through the medium of a capillary oil film so that its oscillation is damped and controlled by the capillary oil film.

A correlative object is to provide a floating reactance interponent between the T-head of the piston and the rotary reactance which is of such configuration in relation to the piston head and reactance that both hydrodynamic and capillary pressure oil films are maintained therebetween for increasing the efficiency of cooperation between the block, reactance, and T-head.

Another object is to provide a hydraulic pump or motor of this general type which will withstand higher operating pressures and speeds while, at the same time, operating with reduced frictional resistance, wear and noise, and in which the risk of seizure during load transmission is substantially eliminated.

Another object is to provide a more improved and compact pump structure with higher operating efficiency.

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 sectional view of a pump or motor embodying the principles of the present invention, and is taken through the axis of rotation of the cylinder barrel and pintle, as indicated by the line 1—1 in Fig. 2;

Fig. 2 is a vertical cross sectional view taken on the plane indicated by the line 2—2 in Fig. 1;

Figs. 3 and 4 are fragmentary sectional views similar to Fig. 2 illustrating respectively certain modified structures by which the required oil films and coaction are obtained;

Figs. 5 and 6 are elevations respectively of T-head pistons, partly in section, showing specialized piston T-heads made of different materials for reducing friction and maintaining oil films;

Figs. 7 and 8 are a side elevation and a fragmentary sectional view, respectively, of a piston T-head and interponent combination illustrating still another modified structure for creating the combination oil films;

Fig. 9 is an elevation, partly in section, illustrating a babbited piston T-head on each face of the head;

Fig. 10 is a fragmentary longitudinal sectional view of a pump or motor piston and reactance combination illustrating a modified bearing structure for the adjustable reactance assembly;

Fig. 11 is a sectional view taken on a plane indicated by the line 11—11 of Fig. 10;

Fig. 12 is a fragmentary longitudinal sectional view of a reactance and piston combination illustrating a modified T-head for reactance cooperation;

Fig. 13 is a sectional view taken on a plane indicated by the line 13—13 of Fig. 12;

Fig. 14 is an elevation of the piston T-head and interponent combination illustrating a modification of the present invention; and

Fig. 15 is a diagrammatic illustration of the operation of the structure illustrated in Fig. 14.

Referring to the drawings, the present invention is shown embodied in a pump or motor in which the cylinder barrel carries the impeller shaft, its operation in connection with other pumps or motors wherein the rotary reactance is the primary rotor being readily apparent therefrom.

The operating structure of the pump or motor is enclosed in a casing 1, comprising an annular wall 2 closed at one end by an internal radial end wall 3, the radial wall terminating in a hub portion 4, and the opposite end of the casing being closed by a rigid end cover 5. Mounted within the hub portion 4 of the casing in fixed and non-rotative position is the shank 6*a* of a valve pintle 6 which pintle protrudes from the hub portion 4 into the interior of the casing and is only indirectly supported at the opposite end. The protruding portion of the pintle 6 has a tapered valve portion 6*b* in which are a port 7 and a diametrically opposite and aligned port 8, the ports 7 and 8 being reversible and in valving cooperation with the cylinder ports of the barrel. The internal hydraulic circuit is completed by longi-

tudinal ducts 9 and 10 in the pintle which connect the internal ports 7 and 8 with main ports 11 and 12, respectively, disposed in the shank of the pintle. The ports 11 and 12, in turn, are in communication respectively with the fluid supply lines of the external hydraulic circuit through suitable bores formed in the hub portion 4 of the casing, in a well known manner.

At the ends of the valve portion 6b of the pintle are bearing races 13 and 14 respectively, which are formed directly on the pintle and accommodate elongated sets of rollers 15 and 16. The sets of rollers 15 and 16, in turn, engage aligned race portions of the wall of the barrel valve bore for positively maintaining the pintle and barrel in coaxial relation with each other and accurately spaced relation for providing positive capillary clearance between the pintle valve, portion 6b and a correspondingly tapered and aligned portion of the barrel bore wall. All above mentioned races are formed of extremely hard material, such as nitralloy or equivalent, to assure the performance of the full complement bearing structures 15 and 16 respectively.

Mounted within the casing for rotation about the pintle is a cylinder barrel 18 which has a set of circumferentially spaced radial cylinders 19, each cylinder being provided with a valve port 20 for cooperation successively with the ports 7 and 8 consequent upon rotation of the barrel. The barrel is rotatably supported in the casing, independently of the sets of rollers 15 and 16, on heavy duty combination load and thrust bearings 21, so as to maintain the barrel in fixed position axially and radially with respect to the casing. In each cylinder is carried a radial piston 24 having a T-head 25 adapted for cooperating the piston with a rotary reactance for transmitting load and torque.

The rotary reactance surrounds the barrel in the zone of the pistons and T-heads and comprises an annular member 26 having radial side walls or flanges 26a which lie alongside the heads of the pistons. The member 26 has chordal surfaces 26b positioned between the flanges near their base and providing chordal or tangential bearing surfaces or guideways 26c for engagement with the radially inward face portions of the T-heads 25, respectively. Surrounding the annular member 26 and forming a part of the reactance rotor is a rigid cylindrical load reactance support 28 which, in combination with the annular member 26 and shoulders 26b, defines a series of circumferential or chordal recesses in which the T-heads 25 are accommodated.

In the recesses and interposed one between the support 28 and each piston head are reactance interponents 30, each of which, at one face, engages the load face of the associated T-head 25, and which, at its other face, is in engagement for load reactance with the support 28, the specific cooperation and configuration of the parts being more fully described below. For rotatably and adjustably supporting the rotary reactance, a reactance stator 31 is provided, the stator having an internal recess 32 which is preferably coextensive axially of the stator with the T-heads 25 and interponents 30, projected.

Mounted within the recess 32 are heavy, elongated rollers 33 which rotatably mount the support 28 in coaxial relation with the stator 31. The rollers 33 are spaced a capillary distance from each other for maintaining oil films therebetween and for distributing the load more uniformly, and likewise terminate lengthwise of the recess 32 in closely spaced relation to the ends thereof so as to provide positive axial clearance spaces between the ends of the rollers 33 and the end walls of the recess 32. Thus the rollers are constrained to the desired circumferentially spaced relation with respect to each other by capillary oil films and also to positive rolling relation with respect to their races. Suitable adjusting means 34 are connected to the stator 31 for shifting the stator to different positions of eccentricity with respect to the axis of rotation of the barrel for varying or reversing the stroke of the pump.

Referring next to Figs. 1 and 2, in detail, and to the cooperation among the reactance interponents 30, the reactance rotor, and the T-heads 25 of the pistons, the interior surface of the load supporting member 28 is cylindrical, polished, and hardened. Each interponent 30 is preferably formed with a generally chordal inner face for engagement with the outer face of the associated T-head and an arcuate, generally cylindrical or spherical outer face for coaction with the inner surface of the member 28, both surfaces of the interponents being hardened and polished. The outer cylindrical surface of each interponent 30, however, is of slightly less radius than the radius of the inner cooperating surface of the member 28, as better illustrated in Fig. 2 wherein the difference in curvature is exaggerated for clearness in illustration. As a result of this difference in radius, it is apparent that very small tapered interspaces 36 will be defined between the outer surface of the interponent 30 and the cooperating bearing surface of the load supporting member 28. When the interponent 30 is in normal intermediate position or dead center position, these spaces will be present at each end of the interponent, being thickest radially at the ends of the interponent and tapering gradually to the midportion, and being generally wedge-shaped. Thus each interponent 30 is free to oscillate and pivot by rolling cooperation with the inner surface of the member 28 consequent upon eccentric disposition of the load of the T-head or upon torque transmission, instead of sliding therealong, as in prior structures. During oscillation the pivotal axis migrates to and fro along the support 28.

Assuming that the direction of rotation in Fig. 2 is clockwise, it is apparent that as the piston head leads from its normal central position with respect to the interponent 30, the interponent will roll slightly toward its leading end, decreasing the space 36 at that end and increasing the space 36 at the opposite or trailing end. Conversely, as the piston head oscillates and lags toward the trailing end of the interponent, the leading space 36 is increased and the trailing space 36 is decreased. Since these spaces are of capillary size, however, fluid is maintained therein at all times and forms therein tenacious high pressure capillary films. Each film, though partially displaced due to the slight rolling action at the particular end, is concurrently augmented at the other end, and, upon reversal of the rolling movement of the interponent 30, the displaced oil film is replenished both by dynamic suction and capillary force. At no time, however, is the film completely displaced from the spaces 36. The mechanical rolling oscillation of the interponent is, therefore, resisted and cushioned by an opposing hydrostatic reactance force, uniformly applied. Thus both by actual mechanical pumping action or suction and by capillary action, te-

nacious films are provided which damp and stabilize the rolling of the interponent, eliminate friction, and tend to distribute the load uniformly and produce hydrodynamic balance of the eccentric load of the piston. This film is a hydrostatic film.

In addition to the hydrostatic capillary film, however, dynamic pressure fluid films are formed between the coacting faces of the interponents **30** of the T-head **25**, due to the relative motion of the interponent with respect to the crosshead. This dynamic film is due to three causes: original misalignment between the interponent and crosshead resulting from manufacturing inaccuracies, relative reciprocation thereof, and the eccentric piston load on the interponent.

Obviously, as the interponent oscillates by rolling, the working face will tend to rock slightly relative to the coacting working face of the T-head and form wedge-shape clearance spaces into which slip fluid is forced. As a result, tenacious wedge-shaped oil films are formed between the T-head and the inner face of the interponent and distribute the load uniformly to the interponent. Thus automatic hydrodynamic equilibrium is maintained between acting and reacting radial forces, the interponent actually having hydraulic load transmitted to it through the medium of hydraulic films, and, in turn, transmitting the load to the reactance through hydraulic films. Reactance forces are correspondingly transmitted. The exact benefits and effects of this operation are illustrated more clearly by comparison of the present interponent with circumferentially sliding interponents, such as described in my copending application, Serial No. 736,550, filed July 23, 1934. The interponent of my copending application is similar to the interponent **30** but has an outer surface congruent to the inner surface of the support **28**. Such an interponent slides bodily back and forth on the interior of the ring **28** to compensate for eccentric disposition of the load during oscillation of the piston head, instead of rocking or rolling as does the interponent **30**. Thus alignment between the head and block results and oil films are formed betwen the interponent and piston head. However, the sliding friction oftentimes is too great to allow free movement of the interponent thus it is delayed in its action, especially at starting and at high speeds, so that instantaneous pinching and seizure often results before the interponent can adjust itself to the proper self aligned position. Also extremely high temperatures are developed which not only thin the oil films and render them less effective but, in fact, soften and anneal the bearing surfaces of the interponent and T-head if they are made of hardened steel. By providing the rolling interponent **30**, such as herein described, instantaneous alignment is effected as the interponent can easily roll into position unopposed by frictional resistance, and pinching, and delayed action of the interponent in moving into alignment is not encountered in this novel structure.

Due to its instantaneous and more perfect alignment, the present interponent distributes the load more effectively. It might be that disadvantages appear to have been introduced in that the rocking action does not appear to provide as large and stable areas of contact between the interponent, the T-head, and the reactance rotor, but the contrary is true. By providing a rocking interponent of this character instead of a firmly seated sliding interponent, and positive clearance and oil films instead of mechanical contact, sufficiently tenacious and flexible support is provided to damp but not unduly restrict the rocking and self-alignment of the crosshead and, at the same time, to distribute the load over the entire bearing surfaces hydraulically and uniformly even under eccentric load disposition, as the interponent operates flexibly at all times between oil films. Thus the load and torque and reactance force transmitted between the piston and the interponent and between the interponent and reactance is uniformly distributed through the medium of the films, an effect which cannot be maintained by direct mechanical engagement of the operating surfaces due to their more sluggish or unbalanced action under friction and to inaccuracies in manufacture. An equivalent of free reciprocation of the crosshead is obtained such as could be obtained only by maintaining the crosshead surfaces and interponent surface in exact theoretical alignment at all times.

Without the intervening oil films, rocking of the interponent would be transmitted to the piston and cause wearing out and subsequent rocking of the piston in its cylinder. With the present interponent, however, some slight rocking tendency of the piston in its cylinder will be perfectly counteracted or balanced by means of the oil film between cylinder wall and piston caused by the pressure in the cylinder. After wearing in of the piston no additional wear or rocking will occur as pressure oil films will form between the piston and its cooperating cylinder. The slight rocking movement transmitted to the piston by the present interponent is thus balanced by this film, due to the cooperation of the above described oil films, and consequently all load and torque transmission will be balanced through the medium of capillary and pressure films, either static or dynamic. The forces between the interponent **30** and support **28** are through the medium of films in the spaces **36**, these forces being transferred to the T-head **25** through films between the outer surface of the T-head and inner face of the interponent, the transfer of load forces and torque from the piston T-head to the barrel being through the medium of the piston and the oil film formed between the outer portion of the piston and outer portion of its associated cylinder. These films necessarily damp and hydrostatically and hydrodynamically balance all forces and automatically maintain hydrodynamic equilibrium.

It should be noted that while the oil films between the interponents **30** and ring **28** are static and will remain indefinitely when once formed, the films between the inner faces of the interponents and the associated T-head are dependent upon the motion between the two and will disappear when the pump is stopped. They will shortly be renewed, however, upon resumption of the operation.

While not necessary, it is desirable to guard against even remotely possible pinching of the T-head between the surfaces **26***c* and inner face of the interponent **30** at the short interval before the oil film is renewed therebetween. For this purpose, elongated cageless rollers **37**, spaced a capillary distance from each other, are interposed between the inner face of the interponent **30** and the outer face of the T-head **25**. A suitable retaining trough **25***a* of such length as to provide a total clearance in the direction of oscillation only sufficient to permit capillary spac-

ing of the individual rollers from each other and relative oscillation of the rollers as an assemblage is formed in the outer face of the head. The rollers **37**, as a set, travel only one half of the relative travel of the head **25** and associated interponent **30** during the relative oscillation thereof. Since the rollers are free capillary needle rollers they will maintain their parallelism and be held together in proper position as an assembly by the capillary oil films formed thereabout and, in turn, will retain the films which act as an elastic cage. Once the oil films are established therefor the rollers will be spaced thereby and at the same time constrained from additional separation even under high speed oscillation. Therefore, the slot **25***a* may be just enough longer than the set of rollers in the direction of oscillation to permit the oscillation of the rollers to the extent necessary during full stroke operation of the piston. It should be noted that the needles **37** are, in fact, rollers and not merely fixed needles which latter would provide only a slide bearing surface. In the conventional type of rollers, concentration of the load on the rollers at one side of the piston axis occurs which, if existing for a short interval without proper lubrication, causes a flattening of the roller. Once the roller has become flattened in this manner, it thereafter ceases to operate as a roller but operates only as a short fixed sliding surface which will soon destroy the cooperating bearing surface of the interponent and T-head. With the present disposition of the rollers, however, absolute uniformity and distribution of the load is effected, the same as though the full faces of the elements were in contact, and furthermore, the films are retained by capillary action so that lubrication is present at the instant of starting.

Due also to the oil films, not only is the hydraulic load equalized and balanced, but the parts are rendered so freely cooperable that the torque is also equalized and evenly distributed so that each piston crosshead carries and transmits an equal portion of the torque, whereas, were the interponents pinched or fixed in position, the loaded pistons or pressure pistons would tend to bind therewith so that all of the torque transmission would be through the medium of the loaded pistons only. In the present design, due to the capillary oil films maintaining all the interponents in almost the same relationship to their respective T-heads, each T-head will transmit its proportionate part of the torque irrespective of the hydraulic pressure under which it is operating at the particular instant.

Referring next to Fig. 3 a slightly modified crosshead and reactance combination having certain advantages is illustrated. In this structure, an interponent **40** is provided which has a chordal face and a cylindrical face, the cylindrical face being of the same radius as the rotary reactance support or ring **41**. The guide surfaces **42** of the reactance are also chordal and plane surfaces. The T-head, however, has slightly cylindrical outer and inner surfaces so as to form positive wedge-shaped spaces **44** and **45** respectively between one surface of the interponent **40** and the supporting ring **41**, and between the other surface of the interponent and guide surfaces **42**, for creating the oil films and obtaining cooperation similar to that obtained with the rolling oscillation of the interponent **30**.

In Fig. 4 is illustrated an interponent comprised of a plurality of segments **50***a*, **50***b* and **50***c*, each of which has a slightly different curvature on its outer face than the cooperating surface of the reactance ring **51**. Each also has on its inner face a slight curved surface, the inner surface of each cooperating with the slightly cylindrical outer face **52***a* of a crosshead **52**. The inner surfaces **52***b* of the crosshead are likewise slightly curved and cooperate with the plane guide surfaces **53** of the reactance. By this structure, a plurality of wedge-shaped films are obtained and each segment can adjust itself instantly to the T-head independently of the other segments.

In Fig. 5 is illustrated a piston having a T-head in connection with the interponent herein described and by means of which frictional resistance between the piston and interponent is reduced. This piston also has an advantage in that the bearing surface of the T-head is replaceable. Referring particularly to the drawings, piston T-head **60** is provided on its working face with a groove which is preferably coextensive transversely of the head therewith. The working face of the head is babbitted or provided with wear-resisting inserts, as indicated at **61**, to provide the desired bearing surface, thus having the advantages of a babbitted sliding bearing as well as ready replacement in case of wear.

In Fig. 6 is illustrated a T-head **63** having at its outer face a recess **63***a* in which are received a plurality of light gauge metal plates **64**, the plates **64** preferably being hardened and polished. The plates **64** define a plurality of capillary interspaces between their contacting flat surfaces which will draw in lubricating fluid by capillary action. This fluid will be partially squeezed out during each pressure period of the pistons but during the succeeding suction period will be released from the pressure applied by the reactance and again drawn in lubricant. This action damps periodic impulses and harmonic vibrations as well as assuring better lubrication and reducing noise and vibration.

In Figs. 7 and 8 is illustrated a piston in which the working face **65***a* of the interponent **65** is slightly curvilinear and cooperates with a flat working surface on the piston head **66** to form capillary wedge-shaped spaces **67**. In this structure, which is especially useful for heavy duty and large pistons, the piston head **66** is provided with transverse rectangular openings in which are received a plurality of metal plates **68** for holding fluid therein by capillary force so as to provide additional damping of harmonic vibrations and shocks. The outer surface **65***b* of the interponent **65** is curved but may be of the same or of a slightly smaller radius than the associated reactance rotor so as to positively assure wedge-shaped spaces for formation of oil films between the interponent and rotor as well as between the interponent and head **66**. The tapered or wedge-shaped spaces assure maintenance of the films at all times.

Fig. 9 illustrates a T-head, for use in connection with the above interponents, which has both the inner and outer surfaces babbitted, the T-head being designated generally as **69**, the outer babbitting **70** and the inner babbitting **71**. In such a structure, wear of the bearing surface does not necessitate replacement of the entire piston, but requires only rebabbitting.

Referring next to Figs. 10 and 11, these figures illustrate a T-head and interponent combination and the manner of cooperating the reactance

rotor with the reactance stator so as to provide substantially continuous circumferential and direct bearing support for the reactance rotor. This combination is particularly useful in connection with multi-stage pumps or motors wherein a number of reactance rotors must be placed very closely end to end. In this structure the piston is provided with a T-head 72 which is recessed on its working face to receive capillary full complement needle rollers 73 or elongated rollers having a total clearance in the direction of operation in the T-head such that capillary interspaces can be provided between the adjacent rollers and the set of rollers can oscillate one half of the relative distance of travel during oscillation of the T-head and interponent. The interponent 74 has a substantially chordal inner face for cooperation with the outer face and rollers 73 of the T-head. The outer face 74a of the interponent is cylindrical and may be the same or slightly less in radius than the inner coacting surface of the reactance rotor 75. The interponent 74 is likewise provided with shoulders 74b which are cylindrical on the outer surface and form a continuation of the surface 74a and which, on their inner surface 74c, may be of slightly greater radius than the cooperating shoulders 75a formed in the reactance rotor for cooperation with the interponent. Thus a rolling action and reaction for transmission of load or suction force as well as for torque is provided. The radial load transmission is thus directed substantially radially of the reactance rotor 75. To transmit this load directly to the stator 76, the stator is provided with a circumferential recess 76a extending axially of the stator so as to be substantially co-extensive with the projected width of the interponent 74. Within the recess 76a are a plurality of elongated needle rollers 77 which may be capillary rollers or rollers of greater than capillary dimensions, but positioned with respect to each other circumferentially of the stator a capillary distance. The recess 76a is substantially the same in depth or very slightly less than the diameter of the rollers 77 so that lubricant entering the recess may constrain the rollers to a capillary distance from each other and may form around the ends of the rollers for maintaining them in a position parallel to the reactance stator. The reactance rotor 75 is received axially in the stator, its outer circumferential surface, which is hardened and polished, being snugly received in the set of rollers 77 and constrained thereby to an accurate co-axial relation with respect to the stator at all times.

Referring next to Figs. 12 and 13 there is illustrated a modified piston T-head and interponent combination for use in connection with a rotary reactance having chordal guide-ways for the head. In this structure the reactance rotor 78 is provided with a number of chordal guide-ways 78a in which are received the piston T-heads 79 and cooperating interponents 80, the walls of the guide-way 78a being parallel to each other. The interponent 80 has a flat working face 80a in cooperation with the working wall of the chordal guide-way 78a and curved surface 80b concave toward the piston T-head 79. The working face 79a of the T-head 79 is curved and convex toward the interponent so as to cooperate with the curved surface 80b thereof.

Referring particularly to Fig. 12, the surface 79a is curved in the direction of rotation about an axis parallel to or coincident with the axis of the reactance rotor, and may be of the same or slightly less radius of curvature than the surface 80b of the interponent. The surface 79a of the T-head is likewise slightly curved transversely, as indicated at 79c in Fig. 12 about an axis transverse to the axis of rotation. Due to the fact that the working surface of the T-head is curved both transversely and longitudinally, the entire working surface of the T-head forms generally a spheroidal or elipsoidal face. As a result, the load applied by the piston T-head and the reactance force ordinarily would be mechanically applied in a small area adjacent the axis of the piston. However, the same surface cooperates with the interponent in such a manner as to define wedge-shaped oil films extending from the central portion of the outer surface of the T-head toward all boundaries thereof. Due to the oil films thus formed the load and reactance forces are hydro-dynamically and hydro-statically distributed uniformly, thus providing the advantages heretofore described and preventing undue concentration of load and forces.

The surface 79a is preferably the same radius as the surface 80b so as to mate therewith or it may be very slightly less in radius so that wedge-shaped oil films may be positively formed between the surfaces as heretofore described. In addition, the surface of the T-head 79 which cooperates with the other wall of the guide-way 78a is slightly cylindrical or curved along the outer portions 79b thereof about an axis parallel to the barrel axis, so as to present to the guide-way slightly convex surface areas. Thus in operation the wedge-shaped oil films may readily be formed between the surface portion 79b of the head and the coacting wall of the guide-way 78a. The coaction resulting from the use of an interponent which is concave toward a convex surface of a T-head is better illustrated in Figs. 14 and 15, in which chordal guide-ways are provided in the reactance rotor and positive creation of oil films such as above described are obtained. In such instance, the interponent 84 has flat or plane face 84a and face 84b which is cylindrical and concave, its axis being coincident with or parallel to the axis of the associated reactance rotor. The piston head 86 has a convex cylindrical reactance surface 86a which is presented to the interponent, the radius of curvature of the face 86a being slightly less than the radius of curvature of the cylindrical surface 84b of the interponent 84. As a result of this structure, wedge-shaped capillary spaces 88 are created between the interponent 84 and the T-head 86. Certain advantages result from this specific structure, in that the capillary spaces 88 may be of more nearly the same dimension radially of the pump throughout their extent than in those instances in which the curvature of the interponent and head, or interponent and reactance rotor, are such as to present a convex surface to a flat or oppositely convex surface. In addition, the desirable effect of the angle of incidence of the surfaces being very small is present, and vibration and pounding are reduced to a minimum. Due to the self adjustment of the interponent 84 on the T-head 86 by rolling thereon, capillary films are provided between the interponent and all coacting surfaces. This action likewise causes the interponent 84 to rock relative to the reactance support with which to be associated, as indicated by the dotted line in Fig. 15, so that wedge-shaped oil films likewise will be formed between the guide-ways of the reactance rotor and the outer surface 84a of the interponent. Obviously the interponent 84a, in roll-

ing, will occupy a normal intermediate position wherein its surface 84b and the outer surface 86a of the piston head are coaxial, as indicated by the line X in Fig. 15. Due to the tangential component of eccentricity as the reactance rotor and cylinder rotor rotate about eccentric axes, the piston head will roll along the surface 84b of the interponent so that the theoretical point of contact and load transmission migrates between limits first at a position and along a line indicated at X—1 and then at a position and along a line indicated by X—2 in Fig. 15. This is necessarily accompanied by a rocking of the interponent relative to the reactance rotor so that oil films are maintained between the interponent and cooperating guide surfaces of the reactance rotor.

In every instance the load and torque transmission forces are uniformly distributed by the oil films and vibrations and impacts are damped. Due to the rolling coaction, as distinguished from sliding coaction, instantaneous adjustment of each interponent is provided.

Having thus described my invention, I claim:

1. In a hydraulic machine of the class described including rotatable radial pistons and rigid crossheads therefor, said crossheads respectively extending fore and aft of the piston axis in the direction of rotation of the pistons, a rotary support having an arcuate bearing surface, a two part guide means for each crosshead carried by the support, one part of the guide means being fixedly secured in the support and the other being elongated fore and aft of the direction of rotation of the pistons and being tangentially reciprocable relative to the crosshead and having an outer arcuate bearing surface in rolling rocking engagement with the said bearing surface of the support, the two cooperating to automatically maintain a wedge shaped oil film between the coacting surfaces of the crosshead and one of said parts, and to reciprocate the pistons through the medium of the crossheads.

2. In a hydraulic machine of the radial piston type, a primary rotor, and a relatively eccentric secondary rotor means, a radial set of pistons carried by the primary rotor and having rigid T-shaped crossheads, an annular recess in the secondary rotor means having an outer arcuate reactance surface, a crosshead guide means associated with each crosshead and disposed in the annular recess of said secondary rotor means, one part of said guide means having an arcuate surface of less radius than said arcuate reactance surface and being in rolling engagement therewith for circumferential adjustment in the recess, and the other part comprising chordal faces carried by the secondary rotor means and in fixed position in said recess relative to the secondary rotor.

3. In a hydraulic machine of the radial piston type, a primary rotor, and a relatively eccentric secondary rotor means, a radial set of pistons carried by the primary rotor and having rigid T-shaped crossheads, an annular recess in the secondary rotor means having a cylindrical reactance surface, a crosshead guide means associated with each crosshead and disposed in the annular recess of said secondary rotor means, one part of said guide means being in rolling engagement with said cylindrical surface for circumferential adjustment in the recess of the said part and the other part of said guide means comprising chordal shoulders carried by the secondary rotor means and in fixed position in said recess relative to the secondary rotor.

4. In a hydraulic machine of the class described, a cylinder barrel having a plurality of radial cylinders, valve means for the cylinders, radial pistons in the cylinders respectively, and a rotary reactance eccentric to the barrel, rigid T-heads on the pistons respectively, said reactance having a circumferential recess accommodating said T-heads and having a load reactance surface exposed in the recess and respective to the pistons, guide-ways in said reactance rotor respective to the T-heads, each guide-way having chordal bearing surfaces in sliding engagement with the one working surface of the associated T-head and having radial flanges in embracing relation with the side walls of the associated T-head, tangentially elongated rockable reactance guide block interponents carried by the reactance rotor in the recess and respective to the T-heads, and each interponent having arcuate working surfaces, one of said interponent surfaces being in rolling engagement with one of the working surfaces of the associated T-head, and the other interponent surface being in rolling engagement with one of said reactance surfaces of the rotary reactance.

5. In a hydraulic machine of the class described, a cylinder barrel having a radial cylinder, valve means for the cylinder, a radial piston in the cylinder, and a reactance rotor member eccentric to the barrel, a rigid T-head member on the piston, said reactance rotor member having a circumferential recess accommodating the T-head member, chordal guide means in the recess in fixed position relative to the reactance rotor and in guiding cooperation with one working face and the side faces of the T-head member for supporting and guiding the T-head member, a reactance guide interponent in said recess, supported for limited bodily rolling movement circumferentially of the reactance rotor member, said interponent having one reactance surface in engagement with a complementary surface on one of said members and an opposite arcuate surface in rolling engagement with a complementary arcuate reactance surface on the other member, and said arcuate surface of the interponent being slightly less in radius than the arcuate reactance surface engaged thereby for pivotally supporting the interponent.

6. In a hydraulic machine of the class described, a cylinder barrel having a radial cylinder, valve means for the cylinder, a radial piston in the cylinder, and a reactance rotor member eccentric to the barrel, a rigid T-head member on the piston and having an outer reactance surface, said reactance rotor member having a circumferential recess accommodating the T-head member and having a load reactance surface in the recess, one of said surfaces being arcuate, chordal guide means in the recess in fixed position relative to the reactance rotor and in cooperation with the radially inward face of the T-head member for supporting and guiding the T-head member, a reactance guide interponent in said recess and movable bodily to and fro in the direction of rotation within certain limits, said interponent having working surfaces in engagement with the surfaces of said members respectively, one of said interponent surfaces being arcuate, the said arcuate one of said reactance surfaces being in engagement with the arcuate surface of the interponent and of sufficiently slightly different radius of curvature to provide rolling engagement between the arcuate surfaces and wedge-shaped spaces therebetween.

7. In a hydraulic machine of the class described, including a radial piston, a T-head therefor, a rotary support having a load reactance surface, a two part guide for each T-head carried by said support and having one part in fixed position relative to the support and in engagement with one working surface of the T-head for supporting and guiding the same, a movable tangentially extending part elongated in the direction of rotation of the support and having a surface in engagement with the other surface of the T-head and having an opposite surface in engagement with said reactance surface of the support, one surface of the movable part being arcuate and in rolling engagement with the surface engaged thereby.

8. In a hydraulic machine of the character described, a rotatable barrel having a radial cylinder, a radial piston carried thereby, valve means for the cylinder, each piston having a rigid T-head member, a rotary reactance member eccentric to and surrounding the piston and having a chordal slideway in fixed relation to the reactance member and in engagement with one working face and the side faces of the T-head for supporting and guiding the same, a tangentially extending reactance interponent elongated in the direction of rotation of said reactance member and being operatively interposed between the T-head member and the rotary reactance member, said interponent having opposite cylindrical reactance surfaces in rolling engagement with said members respectively, the axes of said cylindrical surfaces being parallel to or coincident with the axis of the reactance member.

9. In a hydraulic machine of the character described, a rotatable barrel having a radial cylinder, a radial piston carried thereby, valve means for the cylinder, each piston having a rigid T-head member, a rotary reactance member eccentric to and surrounding the piston and having a chordal slideway in fixed relation to the reactance member and in engagement with the one working face and the side faces of the T-head for supporting and guiding the same, a reactance interponent operatively interposed between the T-head member and the rotary reactance member and movable relative to the reactance member, said interponent and one of said members having mutually engaging arcuate reactance surfaces of slightly different radius of curvature, said axes of said surfaces being parallel to the axis of rotation of the reactance member and said interponent and the other of said members having cooperating reactance surfaces.

10. In a hydraulic machine of the radial piston type, a primary rotor, and a relatively eccentric secondary rotor means, a radial set of pistons carried by the primary rotor and having rigid T-shaped crossheads, an annular recess in the secondary rotor means having a cylindrical reactance surface, a crosshead guide means associated with each crosshead and disposed in the annular recess of said secondary rotor means, one part of said guide means comprising chordal shoulders carried by the secondary rotor means and in fixed position in said recess relative to the secondary rotor, and the other part of the guide means comprising an interponent having a working surface engaging one working face of the T-head, and a cylindrical surface in engagement with the said cylindrical reactance surface, said cylindrical surfaces not quite meeting and having axes parallel to the axis of rotation of the rotor means, and the radius of curvature of the interponent surface being less than that of the cylindrical surface of the rotor means.

11. In a hydraulic machine of the class described, a cylinder barrel having a radial cylinder, valve means for the cylinder, a radial piston in the cylinder, and a reactance rotor eccentric to the barrel, a rigid T-head on the piston, said reactance rotor having a circumferential recess accommodating the T-head, chordal guide means in the recess in fixed position relative to the reactance rotor and in guiding cooperation with one working face and the side faces of the T-head for supporting and guiding the T-head, reactance guide interponents in said recess, supported for limited bodily movement circumferentially of and relative to the reactance rotor, said interponent having one reactance surface in engagement with a complementary surface of said reactance rotor, and having another and arcuate surface, said T-head having an arcuate working face in engagement with the arcuate surface of the interponent and of different radius of curvature than the interponent for providing slight positive clearance spaces between the arcuate surfaces.

12. In a hydraulic machine of the class described, including a radial piston, a T-head therefor, a rotary support having a load reactance surface, a two part guide interponent for each T-head carried by said support and having one part in fixed position relative to the support and in sliding engagement with the one working face of the T-head for supporting and guiding the same, a circumferentially movable part having a face in engagement with the outer working face of the T-head and an opposite face in engagement with said reactance surface of the support, and the working faces of the T-head converging toward a plane between said faces and normal to the piston axis in each direction along the path of rotation from the mid-portion of the T-head.

13. In a hydraulic machine of the class described, including a radial piston, a T-head therefor, a rotary support having a load reactance surface, a two part guide interponent for each T-head carried by said support and having one part in fixed position relative to the support and in sliding engagement with the radially inward face of the T-head for supporting and guiding the same, a circumferentially movable part having one surface in engagement with a working face of the T-head and an opposite face in engagement with said reactance surface of the support, the working faces of the T-head converging in each direction along the path of rotation from the mid-portion of the T-head, and in each direction transversely of the path of rotation from the mid-portion of the T-head.

14. In a hydraulic machine of the class described, including a radial piston, a T-head therefor, a rotary support having a load reactance surface, a two part guide block for each T-head carried by said support and having one part in fixed position relative to the support and in engagement with the one working surface of the T-head for supporting and guiding the same, a rockable non-rotatable circumferentially movable block part having a face in engagement with the other working face of the T-head and an opposite face in engagement with said reactance surface of the support, each of the faces of the movable part sloping from the mid-portion thereof in both directions of the path of rotation and toward the opposite face.

15. In a hydraulic machine of the class de-

scribed, including radial pistons, T-heads on the pistons respectively, a reactance rotor having a circumferential recess accommodating the T-heads, a plurality of non-rotatable load reactance interponent blocks for each T-head pivotally supported in the recess and coacting with the associated T-head, each of said interponents being pivotally supported therein for rocking movement about pivotal axes parallel to the axis of the reactance rotor, and means carried by the reactance rotor for operating the pistons on their suction strokes.

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