Sept. 13, 1938. E. K. BENEDEK 2,129,643

VARIABLE DELIVERY PUMP

Filed Nov. 4, 1935 3 Sheets-Sheet 1

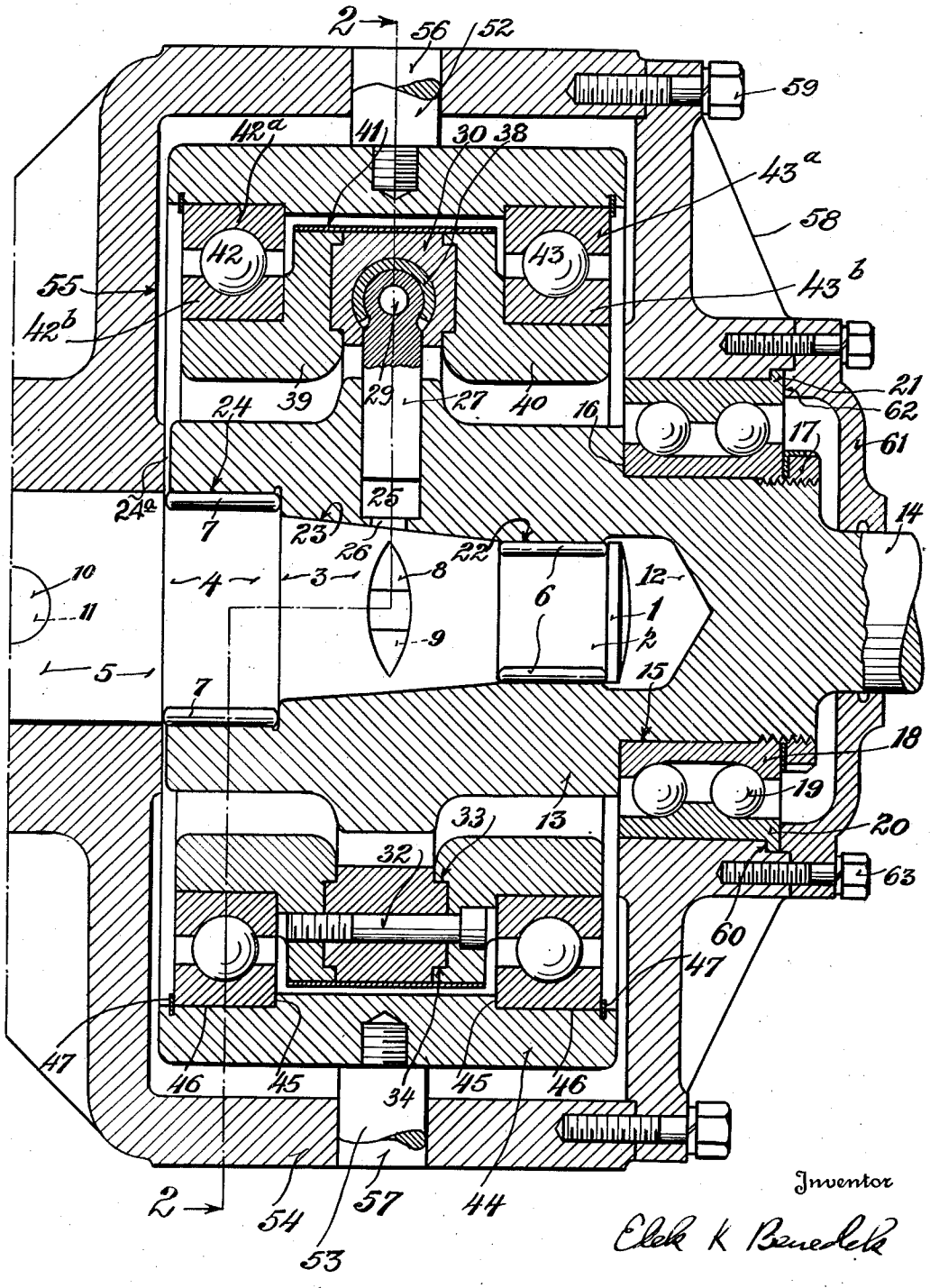



Fig. 1.

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VARIABLE DELIVERY PUMP

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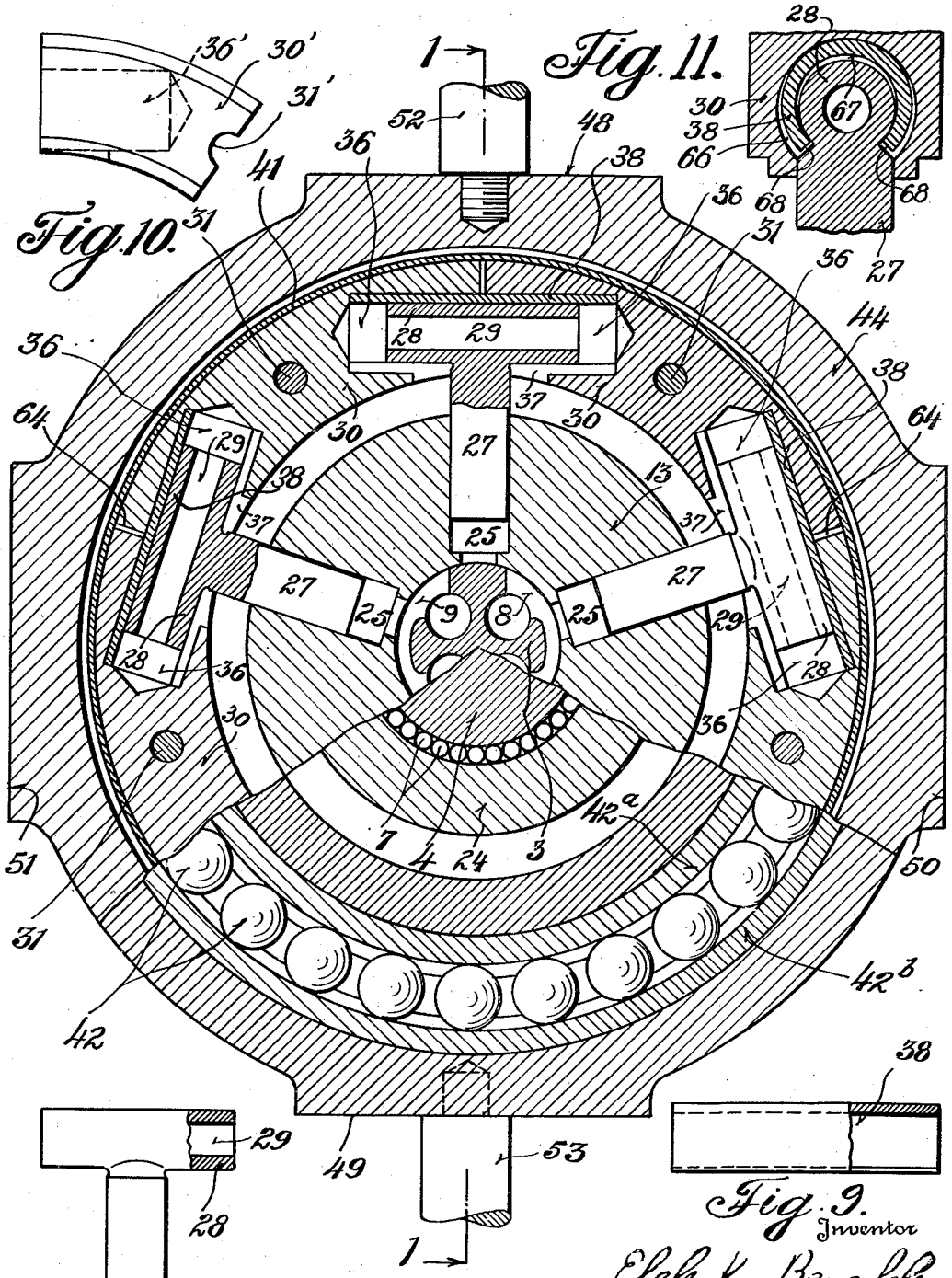



Fig. 11.

Fig. 10.










Fig. 3.




Fig. 9.

Fig. 2.

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VARIABLE DELIVERY PUMP

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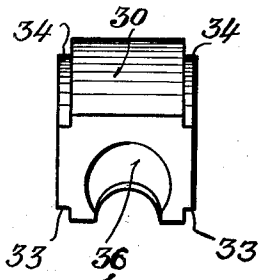



Fig. 5.

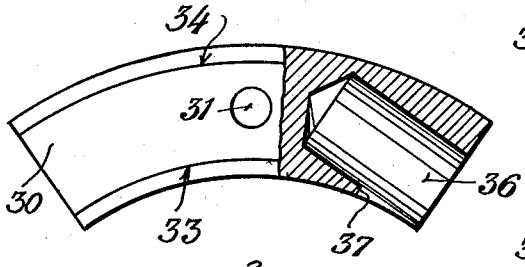



Fig. 4.

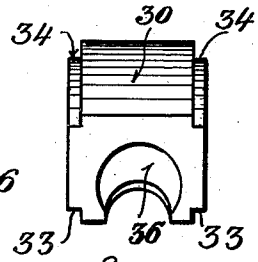



Fig. 6.

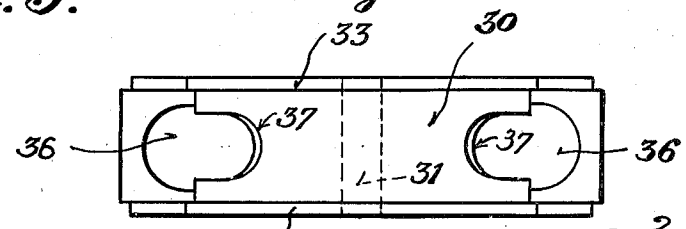



Fig. 7.

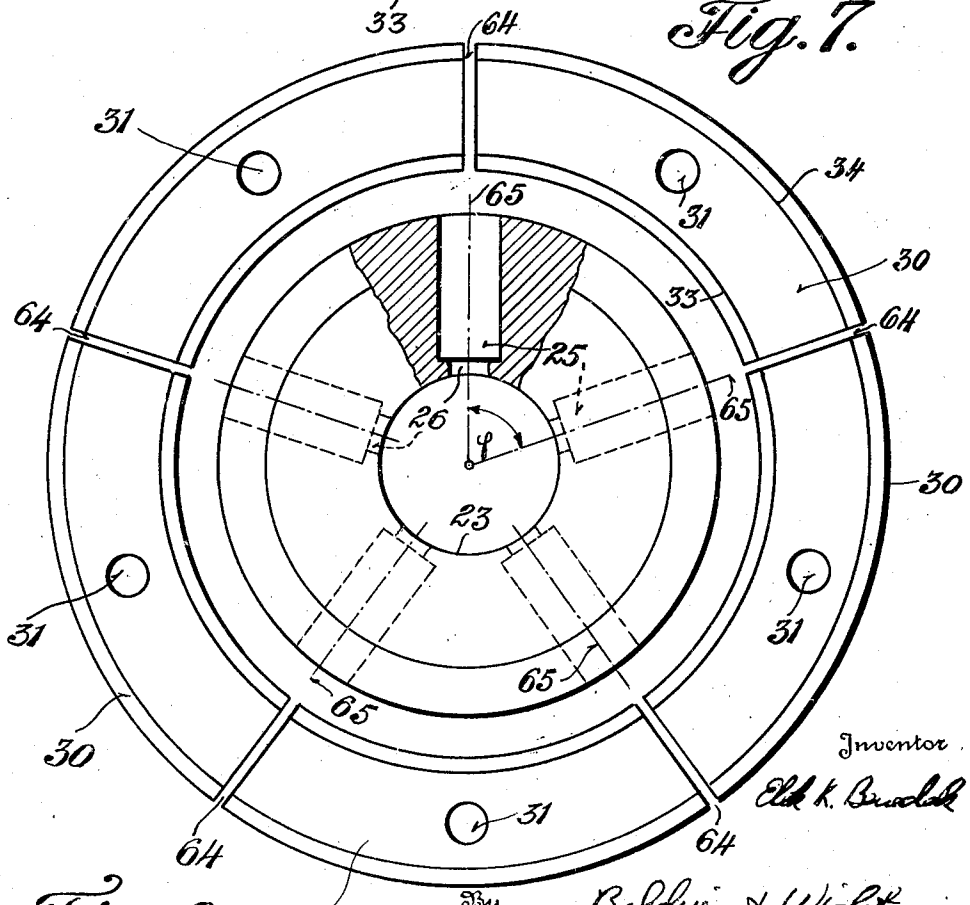



Fig. 8.

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# UNITED STATES PATENT OFFICE

2,129,643

## VARIABLE DELIVERY PUMP

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Application November 4, 1935, Serial No. 48,154

10 Claims. (Cl. 103—161)

This invention relates to hydraulic pumps or motors and more particularly to pumps or motors of the type including a plurality of radial piston and cylinder assemblies.

In conventional pumps of this general type heretofore known in the art, the piston and cylinder assemblies usually have included reciprocable elements such as T-shaped cross heads mounted to slide in ways or guides formed in a surrounding reactance ring adapted to be positioned eccentrically with respect to the axis of rotation of the piston and cylinder assemblies and to exert driving thrust upon the crossheads. When high hydraulic pressures, of the order of 2,000 pounds per square inch and higher, are encountered, the thrust bearing pressures developed between the crossheads and the reactance members are so great that it has been difficult to maintain close fits and to transmit with small frictional loss. High pressure oil film bearings have been found to give the best service under sustained heavy load conditions, but when working under very high pressures, it has been difficult to maintain the accurate alignment of the crossheads, the crosshead guides, and associated parts necessary to provide a high pressure oil film bearing.

A general object of this invention is to provide, in a pump or motor of the class referred to, an improved arrangement of cross heads and associated elements by means of which constructional and operating difficulties heretofore encountered are eliminated.

Another object is to provide a novel T head which will operate in a simplified guide way and which is easy to make and finish to fit the respective guide blocks.

Another object is to provide a bearing sleeve for the novel T head which may be made of high grade and polishable material such as nitralloy and the like, which can be made and polished separate from the guide blocks of the T heads and lapped in with its T head in order to obtain mirror finish for the intervening oil film between the bearing sleeve of the T head.

A further object is to provide a novel disposition of the guide blocks between the secondary rotor halves in such a manner that in the zone of clamping and securing means of the halves, the blocks will be solid and thus a pinching of the T heads in the guide blocks will be eliminated, and thereby the free and efficient reciprocation of the T heads in the guide means will be positively assured.

A further object is to eliminate the bearing at one end of the cylinder barrel between the cylinder barrel and the casing and thereby shorten the projecting lengths of the pintle into the casing and thus the entire length of the pump so that a more effective and rigid cooperation between the pintle and the cylinder barrel will be obtained under increased pressure.

A further object is to provide novel mounting means for the driven end of the cylinder barrel, to efficiently transmit driving torque from the impeller shaft to the cylinder barrel and associated pistons respectively, and take up the axial component of the driving force, caused by the coupling means, which connects the impeller to a prime mover, and which component would otherwise tend to lock and close the tapered fit of the barrel and pintle axially.

A further object is to support one end of the barrel on the pintle entirely, and therefore to provide a rigid cylindrical portion on the walled-in pintle and appropriate means to receive full complement anti-friction bearing means which will constrain and support the barrel radially only, thus an axial floating mounting for this end of the barrel is provided. Another object is to over bridge the coacting ends of the guide blocks by appropriate bearing means to provide an unbroken oil film and support for the reciprocating crosshead.

A further object is to reduce the weight of the crossheads and thereby reduce mass forces of the reciprocating crossheads.

A further object is to retain as much oil as possible with the reciprocating crossheads and their respective bearing means, to thereby effect efficient operation at these points.

Another object is to provide a novel secondary rotor or reactance structure, comprising a plurality of guide blocks or block elements, having solid spacing and clamping receiving portions, to clamp the block elements between supporting and mounting flanges of the structure.

Further object and advantages will be apparent from the following descriptions of the accompanying drawings, forming part of the specifications.

In the drawings:—

Figure 1 is an axial longitudinal section of the pump disclosed for illustration as an example, taken thru line 1—1 in Figure 2.

Figure 2 is a transverse sectional view taken on line 2—2 in Figure 1, but with the pump casing omitted.

Figure 3 is a front elevation of the novel T head

piston, in which one end of the hollow head is shown in section.

Figure 4 is a front elevation as well as a partial sectional view of my novel crosshead guide block, showing the guide bore for one end of the piston crosshead of Figure 3.

Figures 5 and 6 are end views of the block, shown in Figure 4.

Figure 7 is a view of the block of Figure 4 looking from the concave side thereof from radial direction.

Figure 8 is an end elevation of the cylinder barrel and surrounding guide blocks in assembly, the barrel being partially shown in transverse sectional view. This figure illustrates the relative disposition of the blocks with respect to the cylinder bores, when the pump is in its neutral position.

Figure 9 shows my novel crosshead bearing sleeve in elevation and partial sectional view respectively.

Figure 10 is a modified form of my invention, and shows the block of Figure 4 a further subdivided form for the sake of simplicity of production and assembly.

Figure 11 is a slightly enlarged sectional view of the T head and its guiding supporting means showing the effect of the flexibility of the bearing sleeve under pressure and suction stroke respectively, with the operating clearances slightly exaggerated for the purpose of clearness of operation.

With reference to the drawings and more particularly to Figures 1 and 2 respectively, the pump disclosed for illustration comprises a heavy stationary cylindrical valve or pintle 1. In order to reduce the eventual bending and deformation of the pintle under the hydraulic pressure, the valving portion 3 of the pintle is substantially tapered toward the driving end of the pump in such a manner that the tapered portion 3 will interconnect a small cylindrical portion 2, and a large cylindrical portion 4 in such a manner that the walled-in end of the pintle as at 5, will afford substantially constant resistance in every of its transverse cross sections along its projecting longitudinal central axis. Thus the large bearing portion 4 is immediately adjoining the walled-in portion whereas the smaller bearing portion 2 is located in a mating cylindrical bore of the barrel 13. This small bearing portion is adjacent to a pressure chamber 12 provided in the barrel 13.

The cylinder barrel is provided with mating tapered bearing surfaces such as 23 for the actually acting hydraulic pintle portion 3, whereas the adjoining cylindrical bores 22 and 24 are provided in radially spaced concentric relationship to the respective pintle bearing surfaces 2 and 4, in order to receive full complement roller bearings 6 and 7 respectively for the purpose of positive mounting and partially supporting one end of the barrel at least thereon. It will be seen later on that while the bearing elements 7 are provided for total support for one end of the barrel the small roller bearings 6 are provided only for centering the pintle, and cooperating for hydrostatic balance purposes.

The cylinder carries a plurality of radial cylinders as at 25. Each cylinder bore 25 has a radial port as at 26 whereby it communicates alternatively with the internal ports 8 and 9 of the pintle respectively. The internal pintle ports 8 and 9 are in further communication with the external pintle ports 10 and 11 in a well known manner.

The other end of the barrel 13 is provided with impeller means as at 14 which impeller means projects outwardly from the pump casing for the purpose of being connected to a suitable prime mover, such as an electro motor, or any kind of internal combustion engine. Ordinarily the prime mover and the impeller are interconnected by a commercial coupling and due to this coupling drive connection, ordinarily an axial component of the driving force is caused to react against the impeller 14. In order to counteract this axial thrust of the driving force component, I provided appropriate thrust bearing means for the barrel which will positively secure it against axial movement with respect particularly to pintle portion 3. Would any small amount of axial movement be possible between tapered pintle 3 and cooperating tapered bore 23 of the barrel, it is evident that the useful clearance therebetween would be taken up or be enlarged by a movement inwardly or outwardly axially of the barrel with respect to the pintle. In order to clear such movement I form the driving end of the barrel, between impeller and the portion which carries the cylinders with a radial thrust shoulder as at 16 and an axially spaced thrust nut 17, on this intermediate portion of the barrel. A large axial shoulder as at 15 is provided between the thrust means 16 and 17 respectively in order to receive the inner race of double acting thrust bearing means 19. The outer race 20 of this special bearing means is formed with a central circular shoulder, whereas the inner race 18 is provided with a unitary central groove. As shown on the drawings in Fig. 1, an axial thrust in either direction of the axis of rotation, will be taken up between one thrust shoulder of the outer race 20 and an opposed shoulder of the inner race 18 and vice versa. The outer race 20 is disposed appropriately in support cover 58 in such a manner that the inner abutment of the race is free and in line with the inner face of supporting cover 58, whereas the other end is formed with a small radial flange or shoulder as at 21 which shoulder is engaged in one direction by associated shoulder 60 of the support 58, and in the other direction it is secured against recess shoulder 60 by wall 62 of an end cap 61. The inner race meanwhile is rigidly secured between thrust shoulder 16 and thrust nut 17 respectively in such a manner that the radial thrust surfaces will prevent any creeping or rotation thereof with respect to the mounting seat 15 of the barrel 13.

For the opposite end of the barrel a full complement roller bearing is provided, between an enlarged portion of the pintle 4 and between appropriate bearing race 24 of the barrel. The antifriction elements 7 are provided with such intermediate circumferential clearance which is necessary to separate the individual elements by an oil film in order to eliminate contact friction during operation of the bearing. This end of the barrel is provided with small axial clearance 24a with respect to the inner wall 55 of the body 54 of the pump. This axial clearance space will allow slight axial adjustment of the barrel, when it is adjusted for the hydraulic clearance space between pintle portion 3 and barrel surface 23. This small axial clearance furthermore will provide for heat expansion axially of the barrel since the other end is positively fixed and locked with respect to the support 58. Therefore it will be seen that bearing means 7 between pintle and barrel constitutes an axially floating mounting.

Each cylinder bore or individual cylinder is

provided with a radial piston 27. This piston is shown in neutral position with respect to the cylinder 25 in Fig. 1 and Fig. 2. Each piston is provided with a head portion as at 28 which is reciprocably engaged by a liner or bearing sleeve 38. This bearing sleeve is more fully shown in Fig. 9. Its main feature comprises a cylindrical sleeve which is open toward the piston in such a manner that it can be easily slipped over the crosshead from the direction of the axis of the crosshead. The crosshead is made cylindrical so that the polished cylindrical head will fit telescopically with small operating clearance the inner lapped and polished surface of the liner or bearing sleeve 38. It will be seen that this liner or bearing sleeve will provide a continuous unbroken oil film for the thrust projection of the crosshead surface during the operation, and allows an easy assembly between the piston and the sleeve.

The liner or sleeve 38 with the crosshead 28 will be assembled in two adjacent bores 36 of crosshead guide blocks 30. The guide blocks are provided with appropriate end bores 36 in order to receive bearing sleeve 38. The facing ends of the guide blocks 30 are disposed substantially in the axis of the piston and cylinder assembly with a small circumferential clearance as at 64 for the purpose of adjustment and proper alignment of the crosshead and sleeve assembly in the bores 36. I thus have provided a simple construction whereby the guide bores 36 of the blocks 30 and the crossheads will be aligned accurately, the interruption of the guide bores 36 by the radial spaces 64 being bridged by the bearing sleeves 38 in such manner that a continuous and unbroken high pressure oil film may be maintained between crossheads and their respective bearings.

Bearing thrust sleeve 38 is open to receive the T head 28 of the piston from a direction coincident with the axis of the reciprocation of the head in the sleeve. This will allow one to slip the sleeve on the crosshead or T head by accommodating the neck of the piston in the opening in the sleeve provided by the cut-away portion of the sleeve as shown in Fig. 1 and Fig. 9.

It will be seen that in spite of the cut-out in the sleeve 38, the sleeve will cooperate with end bores 35 and 36 of the blocks 30, to form therewith a self contained fluid supply for the lubrication of the crosshead 28 and sleeve 38, because of the sleeve being snugly fitted in the bores, it will close fluid tightly the chambers, comprising sections 35 and 36, and hole 29 of the crosshead against the centrifugal force and thus the slip fluid escaping the cylinders will be accumulated in said chambers and retained therein, without the necessity of an additional liquid tight cover as at 41. Thus the primary function of cover 41 is only to retain an additional supply of oil in the crevises 64, as well as in the elongated cut-outs 37 in order to increase the centrifugal force in chamber 29 and 36 respectively, by the medium of additional mass of oil subjected to the centrifugal pressure of the rotation.

Each end of the supporting blocks 30 are cut away as at 37 to clear the reciprocation of the neck of the piston with respect to the blocks. Since the maximum amplitude of reciprocation right and left at each side of the axis of the piston amounts to the maximum eccentricity of the pump—therefore the cut-out 37 is proportional to the eccentricity, measured from the coacting ends of the blocks. As shown in Fig. 1, when adjacent ends of two adjacent blocks meet in assembly, the cut-outs 37 of two ends will form an elongated opening around the piston to facilitate relative tangential oscillation of the crosshead 28 and piston respectively in the adjacent blocks.

It will be understood that my invention is not limited to the use of the specific form of sleeve shown at 38. Sleeves employed in accordance with my invention should be formed at their central portions with clearance spaces accommodating the pistons and permitting relative movement of the pistons and sleeves tangentially with respect to the pistons, but otherwise the sleeves 38 are shown merely as representing a practical and preferred form.

In the form shown in Fig. 2 of the invention, the pistons will be assembled from a radial direction together with their respective blocks and sleeves into their cylinders. At the start of the assembly the circumferential distance between the coacting ends of the blocks will be sufficient to introduce the ends of the sleeves 38 into the bores 36 whereupon the blocks will be pushed in radially and simultaneously to form a smaller and smaller circle until the circumferential gaps or crevises will be not less than 64 respectively. In this particular position, the shoulders 33 and 34 of blocks 30 just will fit the normally aligned grooves of supporting flanges 39 and 40. The blocks will be carefully lined up with respect to the sleeves 38 in such a manner that by the medium of gaps 64 each sleeve and crosshead will be free when moving eccentrically barrel 13 with respect to the assembled blocks 30 and supporting means 39 and 40. After the sleeves 38 and the supporting blocks 30 have been aligned properly, the plates 39 and 40 are drawn up tight against the blocks 30 by means of the bolts 31.

It will be seen that due to the form of guide means 35 and 36, of the bores around the coupling means the blocks will be solid and a very forceful clamping will be possible without affecting at all the shape of the bores 35 and 36. See lower portion of Fig. 1. A forceful clamping needless to say, will result in a more rigid substructure, necessary for heavy load and noiseless operation of the pump.

The modified block structure shown in Fig. 10 will simplify the assembly above described in that, in this instance each piston can be assembled with its blocks independently into the cylinder. Namely each piston 27, after being assembled with its sleeve 38, and two respective blocks 30' will form a self contained sub-assembly unit ready for final assembly. Also in this instance the advantage of freeing the guide holes 36' of deformation by the clamping assembly process, is preserved.

It will also be seen that by the virtue of the subdivision of the block 30 as at 30', one single operation for the essential finish of the block namely to bore and lap hole 36' only will be necessary. Thus all blocks will remain the same, but simplified as to size and operation.

By cutting the blocks 30 in two to form the half blocks 30', a clearance equal to the thickness of material removed in making the cut will be provided between the half blocks 30'. This clearance further facilitates assembly and alignment of the cross heads and their associated elements. Assembly is facilitated despite the fact that when completely assembled, the portions 31' of the half

blocks 30' are contacted by the bolts 31, because the half blocks are positioned between the members 39 and 40 before the bolts 31 are put in place. The provision of clearance between the contiguous faces of the adjacent half blocks permits the half blocks to be shifted slightly and after all the half blocks are arranged between the members 39 and 40, the respective half blocks can be moved to just the right positions for admitting the bolts 31. If no clearance between the half blocks 30' were provided, it would be more difficult to move the half blocks so as to bring the portions 31' into the positions for receiving the bolts 31.

One novel feature of the invention is shown in Fig. 11 and will be described in connection with that figure.

The bore 35 or 36 of member 30 is considered cylindrical and its shape to be permanent, since block 30 is substantially rigid, and any clamping action which would cause distortion is curbed as hereinabove described.

However, the flexible sleeve 38 with the above described cut-out will not be permanent in shape during the pressure and suction cycles of the pump. As will be noted from Fig. 11 the ends of the normal section of the sleeve at the points of the cut-out will touch closely the cylindrical surface of the crosshead 28 as shown in Fig. 11 in such a manner that due to the radial approach of the sleeve to the bore 36 under the pressure of the piston, a slight clearance as at 67 will be formed with the sleeve and crosshead, or assuming that the force conditions are such that the cut-out edge portions of the sleeve 38 will not spring inwardly but rather spread apart from each other in which instant the ends of the normal cross section of the sleeve will close a chamber similar to 67 between the circular bore of the guide block 30 and between the sleeve 38 itself. Thus according to the nature of the flexibility of the sleeve the longitudinal margins or edges of the sleeve at the place of the cut-out as at 68 will define a closed clearance space either with respect to the cylindrical crosshead or with respect to the cylindrical bore 36 of the guide block 30. Assuming the case that the flexible sleeve springs inwardly and will close a clearance space as at 67 with the crosshead by touching the crosshead surfaces in line with a neck portion of the piston as at 68, a clearance space 66 will be formed between the ends of the sleeve and the bore 36 of the rigid block 30. A condition as illustrated in Fig. 11 therefore illustrates the pressure cycle, when under the hydraulic piston load, sleeve 38 will depress and become eccentric with respect to the main bore 36 with which it forms a clearance and subsequent continuous wedge shaped oil film as at 66. During the subsequent suction cycle of the piston however, sleeve 38 will be pulled in eccentric relationship with respect to bore 36 in the opposite direction as during the pressure cycle with the consequence that the clearance conditions will be changed and will be symmetrical to the present clearance conditions with respect to a normal plane to the axis of the piston and laid through the axis of the crosshead. Thus it is obvious that due to the flexibility of sleeve 38 a plurality of possible combinations for the formation of high pressure wedge shaped oil film are provided. The springing tendency of the sleeve member 38 depends on how much load is applied on the piston and how rigid the working cross section of the sleeve longitudinally is. Due to the original stresses imposed upon the sleeve by cutting away a section of its normal cross section, unbalanced forces will be set up in the remainder of the cross section which tend to close the open ends of the normal cross section accommodating the cut-out for the neck of the piston. In addition to the unbalanced internal stresses due to the cut-out, hydraulic forces acting on the outside surface of the sleeve tend also to compress the sleeve and thus close the ends at 68—68 respectively, because under pressure the hydraulic forces will act on a larger surface on the outside of the sleeve than on the inside thereof.

The pump will operate in a well known manner. As far as the cylinder barrel and piston actuating reactance rotor will be disposed about the same axis of rotation by means of control rods 52 and 53 the entire rotating assembly of the pump will rotate bodily, and without any reciprocation of the pistons 27 or the crossheads 28 thus no pumping action will take place. However, as soon as the axis of rotation of the reactance rotor will be established as different from the axis of the barrel 13, the pistons will radially reciprocate in their cylinders 25 whereas crossheads 28 will reciprocate tangentially in their respective thrust sleeves 38. Since, however, the eccentricity of the synchronized rotors can be changed from zero to a predetermined maximum it is evident that pump delivery can be changed and controlled from zero to a maximum value.

The piston actuating reactance assembly comprising the supporting means 39 and 40 and the intermediate supporting blocks 30 are mounted as a sub-assembly on bearing means 42 and 43 respectively in such a manner that the bearing means are disposed symmetrically to the plane of the axis of the pistons. In this manner when control rods 52 or 53 will shift the stator housing 44, the axis of the reactance rotor will shift substantially parallel to itself. In order to assure the bearings 42 and 43 in fixed positive axial relationship with respect to each other and respective to the stator 44 I provided intermediate shoulders 45 on the stator 44 against which the stationary races of the bearing will be held by appropriate spring washers 47. Thus the bearings will be confined between shoulders 45 and washers 47 respectively. The translatory movement of the stator 44 will be assured further on by the disposition of the shifter rods 52 and 53 which are disposed diametrically opposite in the casing 54 and guided in appropriate diametrically opposite guide holes 56 and 57 of the casing. Thus there is no possible chance during adjustment of the reactance assembly to cause other than a parallel shift thereof.

Various changes may be made in the embodiment of the invention without departing from the spirit thereof or narrowing its scope which will be defined in the appended claims.

I claim:—

1. In a hydraulic pump or motor of the type including a plurality of valved radial piston and cylinder assemblies each comprising a relatively non-reciprocable element and a relatively reciprocable element, the combination with said assemblies, of means for effecting reciprocation of said relatively reciprocable elements comprising cross heads, one connected to and disposed tangentially with respect to each relatively reciprocable element; a plurality of circumferentially arranged thrust blocks surrounding said piston and cylinder assemblies and being formed with openings within the blocks and extending perpendicu-

larly to the piston axes in the plane of rotation of the thrust blocks respectively, each cross head being received and guided in two-directional thrust-transmitting relationship by the openings in two circumferentially end-to-end adjacent thrust blocks; and means mounting said thrust blocks eccentrically with respect to the relatively non-reciprocable elements of said piston and cylinder assemblies.

2. In a hydraulic pump or motor of the type including a plurality of valved radial piston and cylinder assemblies each comprising a relatively non-reciprocable element and a relatively reciprocable element, the combination with said assemblies, of means for effecting reciprocation of said relatively reciprocable elements comprising cross heads, one connected to and disposed tangentially with respect to each relatively reciprocable element; a plurality of thrust blocks surrounding said piston and cylinder assemblies in circumferential end-to-end relation, a plurality of liners extending respectively into and being supported by each two circumferentially end-to-end adjacent thrust blocks, each liner receiving and guiding a cross head, and means mounting said thrust blocks eccentrically with respect to the relatively non-reciprocable elements of said piston and cylinder assemblies.

3. In a hydraulic pump or motor of the type including a plurality of valved radial piston and cylinder assemblies each comprising a relatively non-reciprocable element and a relatively reciprocable element, the combination with said assemblies, of means for effecting reciprocation of said relatively reciprocable elements comprising cross heads, one connected to and disposed tangentially with respect to each relatively reciprocable element; a plurality of thrust blocks surrounding said piston and cylinder assemblies in circumferential end-to-end relation, a plurality of slotted bearing sleeves extending respectively into and being supported by each two circumferentially end-to-end adjacent thrust blocks, each sleeve receiving and guiding one of said cross heads, the connections between said reciprocable members and the associated cross heads extending through the slots in the respectively associated sleeves, and means mounting said thrust blocks eccentrically with respect to the relatively non-reciprocable elements of said piston and cylinder assemblies.

4. In a hydraulic pump or motor of the type including a plurality of valved radial piston and cylinder assemblies each comprising a relatively non-reciprocable element and a relatively reciprocable element, the combination with said assemblies, of means for effecting reciprocation of said relatively reciprocable elements comprising cross heads, one connected to and disposed tangentially with respect to each relatively reciprocable element; a plurality of thrust blocks surrounding said piston and cylinder assemblies in circumferential end-to-end relation, a plurality of slotted bearing sleeves flexible in cross section and extending respectively into and being supported by each two circumferentially end-to-end adjacent thrust blocks, each sleeve receiving and guiding one of said cross heads, the connections between said reciprocable members and the associated cross heads extending through the slots in the respectively associated sleeves, and means mounting said thrust blocks eccentrically with respect to the relatively non-reciprocable elements of said piston and cylinder assemblies.

5. In a hydraulic pump or motor of the type including a plurality of valved radial piston and cylinder assemblies each comprising a relatively non-reciprocable element and a relatively reciprocable element, the combination with said assemblies, of means for effecting reciprocation of said relatively reciprocable elements comprising cylindrical cross heads, one connected to and disposed tangentially with respect to each relatively reciprocable element; a plurality of thrust blocks surrounding said piston and cylinder assemblies in circumferential end-to-end relation, circumferentially adjacent ends of said thrust blocks being formed with aligned bores, a plurality of slotted, cylindrical bearing sleeves flexible in cross section and extending respectively into and being supported by each two circumferentially end-to-end adjacent thrust blocks, each sleeve receiving and guiding one of said cross heads, there being sufficient clearance between said sleeves and said bores and cross heads to permit flexing of the sleeves for drawing lubricant into the clearance spaces, the connection between said reciprocable members and the associated cross heads extending through the slots in the respectively associated bearing sleeves, and means mounting said thrust blocks eccentrically with respect to the relatively non-reciprocable elements of said piston and cylinder assemblies.

6. In a hydraulic pump or motor of the type including a plurality of valved radial piston and cylinder assemblies each comprising a relatively non-reciprocable element and a relatively reciprocable element, the combination with said assemblies, of means for effecting reciprocation of said relatively reciprocable elements comprising cross heads, one connected to and disposed tangentially with respect to each relatively reciprocable element; a thrust member; means mounting said thrust member eccentrically with respect to the relatively non-reciprocable elements of said assemblies, said thrust member being provided with a plurality of bearing recesses associated with the cross heads respectively; and connections between said thrust member and each of said cross heads comprising an expansible and contractible sleeve interposed between the cross head and the walls of the associated bearing recess and embracing a portion of the cross head, said sleeve being slotted throughout its length and being relatively more flexible than said thrust member and said cross head, and being adapted when thrust-loaded successively in opposite directions to expand and contract successively under the action of successive thrust loadings and to thus effect an oil pumping action.

7. The combination defined in claim 1 and in which each two circumferentially adjacent thrust blocks are separated from each other by a clearance space substantially in line with the axis of reciprocation of the relatively reciprocable element of the associated piston and cylinder assembly.

8. The combination set forth in claim 1 and in which each of the thrust blocks is separated from the adjacent thrust blocks at each of its ends by clearance spaces, the clearance space at one end of each thrust block being substantially in line with the axis of reciprocation of the relatively reciprocable element of the associated piston and cylinder assembly.

9. The combination set forth in claim 1 and in which the means for mounting the thrust blocks comprises spaced supporting flange elements disposed respectively on opposite sides of thrust blocks, and means for securing said sup-

porting flange elements and said thrust blocks together.

10. The combination set forth in claim 1 and in which the thrust blocks are formed with shoulders and in which the means for mounting the thrust blocks comprises spaced supporting flange elements disposed respectively on opposite sides of the thrust blocks, said flange elements being formed with grooves receiving the thrust block shoulders, means securing said thrust blocks and flanges together, and means mounting said supporting flanges for rotary movement.

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