Sept. 21, 1954 E. K. BENEDEK 2,689,531

HYDRAULIC MACHINE

Original Filed Sept. 27, 1945 4 Sheets-Sheet 1

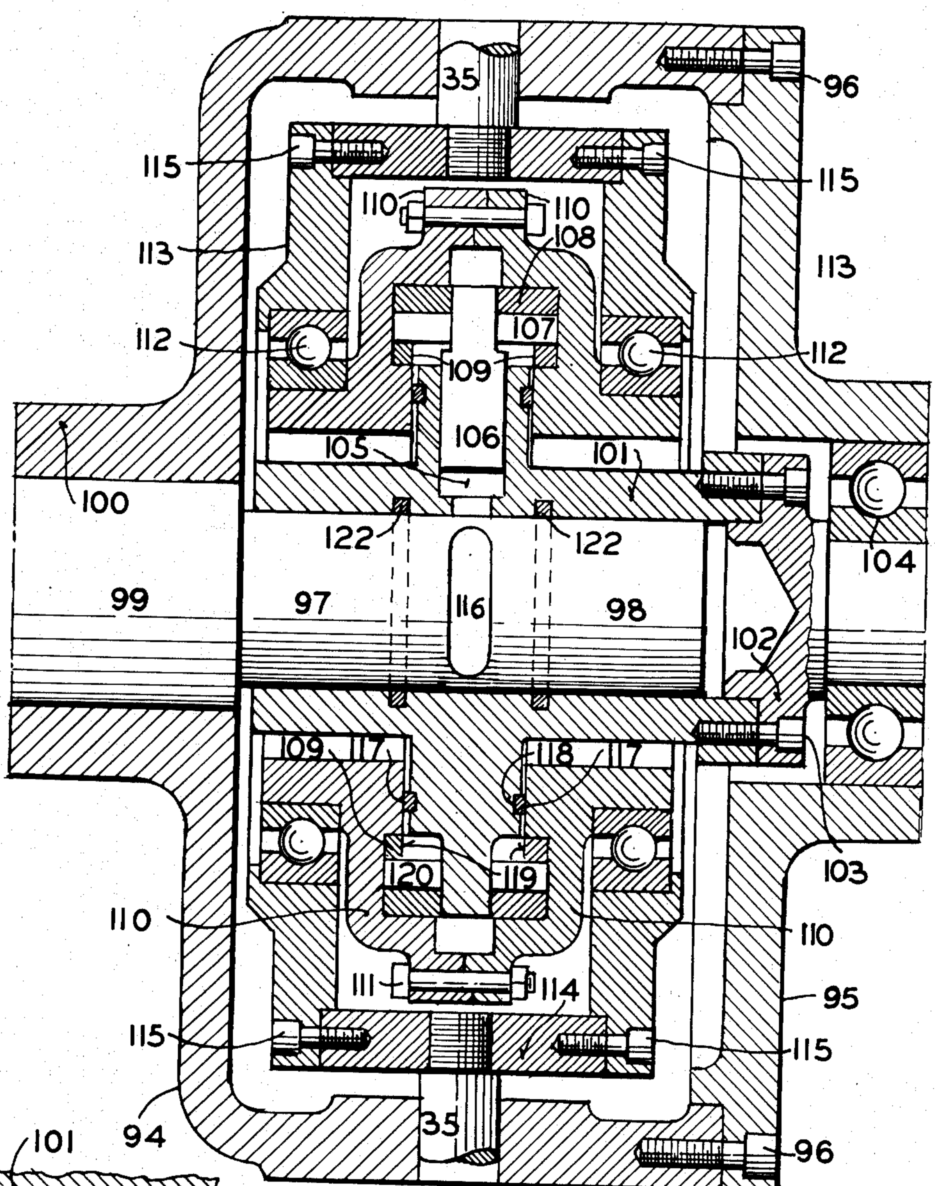

FIG_1_

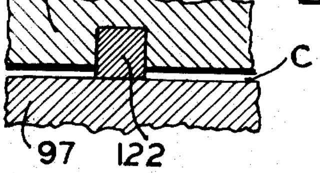

FIG_2_

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HYDRAULIC MACHINE

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FIG. 4.

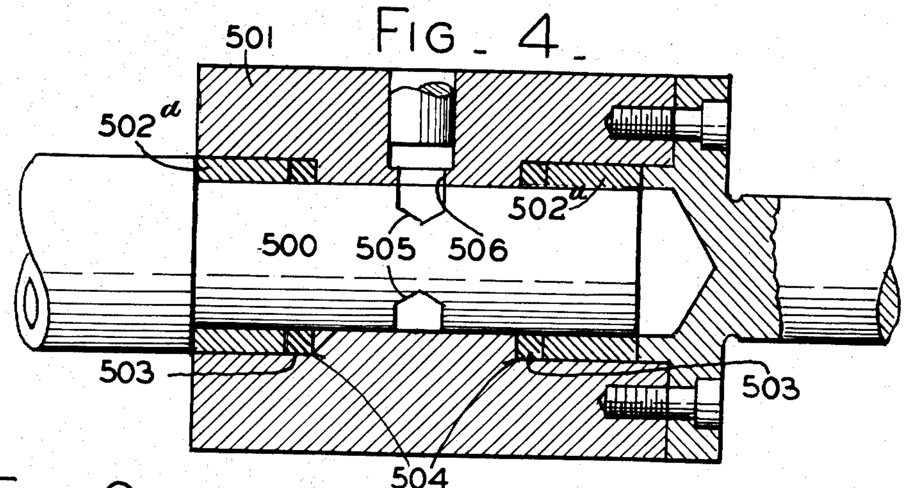
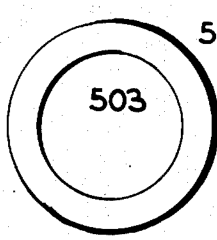


FIG. 6.



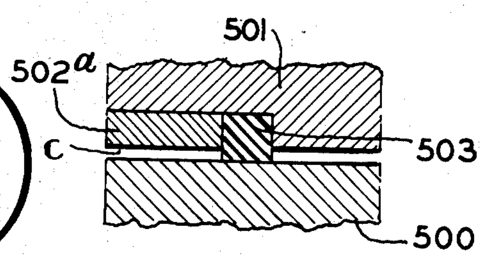


FIG. 5.

FIG. 7.

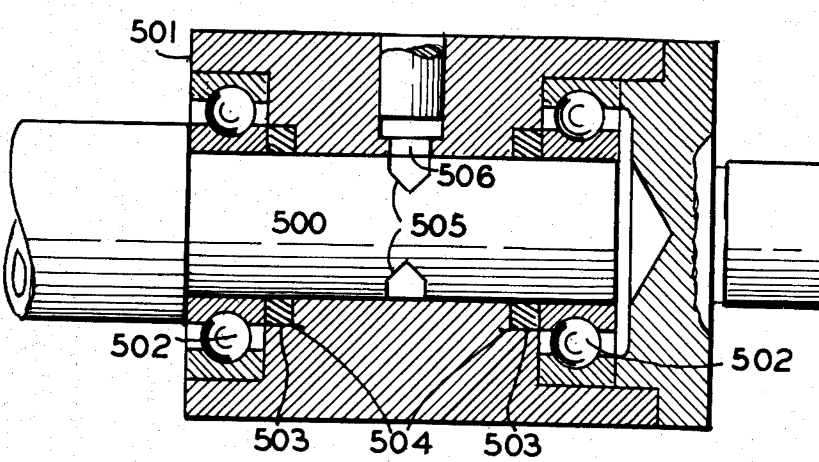


FIG. 3.

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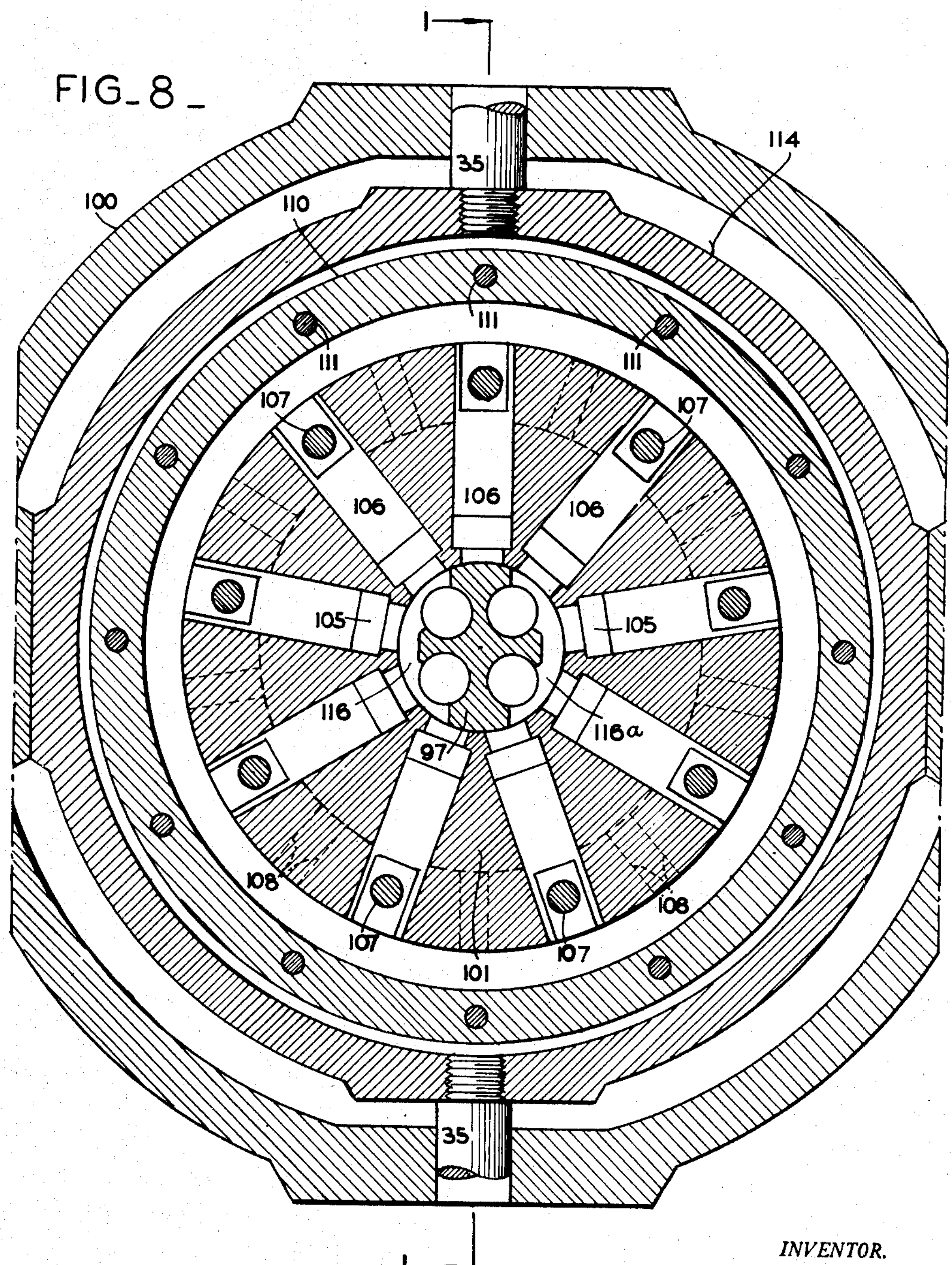
HYDRAULIC MACHINE

Original Filed Sept. 27, 1945

4 Sheets-Sheet 3

FIG_8_



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HYDRAULIC MACHINE

Original Filed Sept. 27, 1945 4 Sheets-Sheet 4

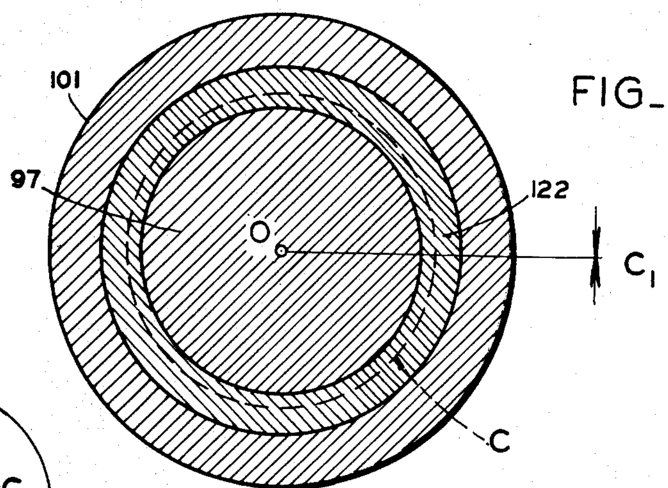

FIG_9_

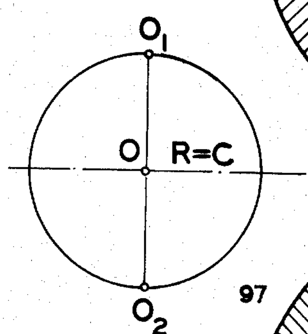

FIG_12_

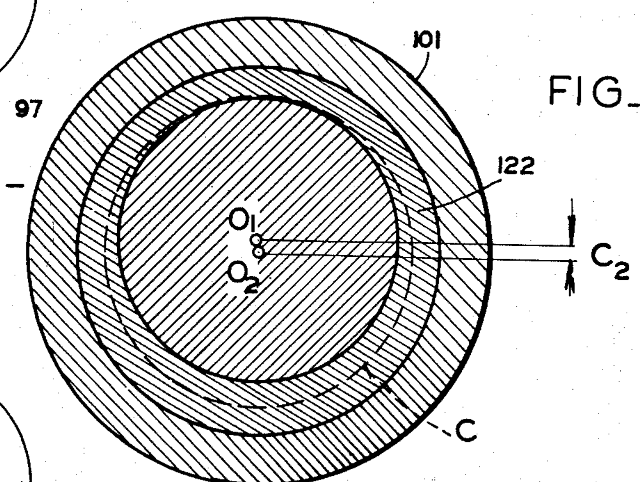

FIG_10_

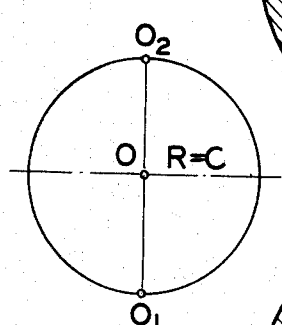

FIG_13_

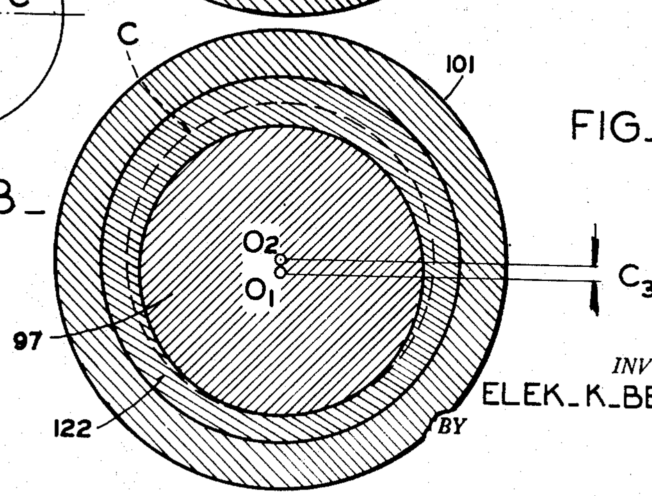

FIG_11_

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Patented Sept. 21, 1954

2,689,531

UNITED STATES PATENT OFFICE

2,689,531

HYDRAULIC MACHINE

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Original application September 27, 1945, Serial No. 618,890, now Patent No. 2,452,541, dated November 2, 1948. Divided and this application October 29, 1948, Serial No. 57,167

4 Claims. (Cl. 103—161)

1

This application is a division of my copending application Serial No. 618,890, filed September 27, 1945, now Patent No. 2,452,541, issued November 2, 1948.

This invention relates to rotary hydraulic machines and more particularly to hydraulic pumps or motors of the kind which include a rotary barrel and a centrally located valve pintle about which the barrel rotates. The barrel is formed with a plurality of circumferentially deployed radial cylinders in which reciprocable pistons are mounted. Hydraulic machines of this general class are well known in the art and have been used extensively under exacting conditions of fluid pressure and rotative speed. Although numerous improvements and refinements have been contributed to the art in recent years, some difficulties remain to be eliminated or reduced in effect. One of the difficulties is that of maintaining proper lubrication of the pintle and barrel, and of parts which transmit radial thrust to the reciprocable pistons. Another undesirable characteristic of machines of this class heretofore provided has been their tendency to suck air into the cylinders resulting in noisy operation, and vibration frequently leading to break downs. In nearly all rotary hydraulic machines of the class referred to the driving torque is transmitted from the cylinder barrel through the radial pistons themselves. At any stage of operation one half of the pintle surface and one half of the pistons are operating on suction strokes, and in prior art constructions the suction pistons must be operated without the protection of pressure film lubricant.

An object of the invention is to overcome these difficulties by providing an hydraulic machine of the class refered to in which a fluid tight chamber filled with working or lubricating fluid lubricates the coacting pintle and cylinder barrel surfaces which seal the working fluid of the cylinders by viscosity. With such a construction the corotating parts are sealed in a film of lubricant which assures complete and efficient lubrication at all times. Furthermore the fluid in the chamber surrounds or covers the coacting surfaces of the pintle and cylinder barrel and prevents the sucking in of air between the surfaces, and provides lubricant under pressure between the pintle and its respectively associated barrel when the pump is performing a suction cycle. Thus the working fluid which also acts as a lubricant is maintained air-free and always available for pressure lubrication.

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In accordance with the invention the foregoing stated object is achieved by providing a lubricant and working fluid containing space between two relatively rotating rotors, which clearance space is sealed at both ends of the barrel so that fluid will be maintained in said clearance space at all times of pump operations, during suction as well as pressure periods.

Another object of the invention is to provide a construction of the character stated in which the fluid tight clearance is formed or provided by the cylinder barrel element, the pintle element, and sealing means so arranged between the barrel element and the pintle element as substantially to retain fluid between the two elements but so as to permit relative movement between the elements.

Another object of the invention is to provide a construction of the kind described above and in which means are provided for delivering fluid to the sealed clearance space.

Another object of the invention is to provide a construction of the kind referred to in which the body of fluid sealed in the clearance space minimizes the dry running and resultant excessive wearing of the suction side of the pintle and the coacting wall of the cylinder barrel bore, which have been characteristic of rotary hydraulic machines heretofore known.

Another object of the invention is to provide an hydraulic machine having a fluid tight chamber in the dead-end bore of the barrel, the retained fluid of which chamber will supply supercharging fluid between the clearance space of the barrel and pintle, which fluid tends to repel incoming air at the left hand end of the pintle and barrel.

A further object of the invention is to provide an hydraulic machine including a pintle element and a surrounding cylinder barrel element and novel and improved means for sealing the clearance between the pintle and the cylinder barrel bore on opposite sides of registering pintle and cylinder barrel parts.

A further object of this invention is to provide improved sealing means between pintle and barrel bore at both sides of registering pintle and cylinder barrel ports, which means include capillary clearance space whereby a small uniform capillary clearance is provided between closely finished barrel bore and lapped and ground pintle. In the capillary space the sealing occurs by the viscosity of working fluid which in this instance comprises high grade mineral oil, such as Gargoyle D. T. E. light, medium, heavy or extra-heavy.

Since each grade of oil is suitable for a certain size capacity pump, it will be used with a certain clearance. Lighter oils are suitable for smaller pintles and smaller pumps, and heavier oils are suitable for larger pintles and larger pumps. The capillary clearance has a linear magnitude of one-half of one thousandth of an inch per one inch pintle diameter. Thus a one inch pintle requires one-half of a thousandth of an inch total diametral clearance, which amounts to one quarter of one thousandth on the side. Under the term running clearance this specification means the amount of the side clearance around the entire periphery of pintle and barrel bore respectively. Diametral clearance meaning the total running clearance at diametrally opposite points of the pintle and barrel bore, which is twice as great as the radial or side clearance. But I do not limit my invention to any specific amount, as long as the clearance space is small enough to seal the pressure and permit free rotation for the barrel.

One main object of this application is to combine the sealing effect of a capillary seal with the sealing effect of positive resilient sealing means in such a manner that the capillary seal, at high operating pressures such as 3000 p. s. i., will take care of the pressure seal by permitting limited predetermined pressure slip between the capillary parallel circular surfaces of the pintle and the barrel. While the resilient sealing means are not able to seal the high pressure, they are capable of sealing the air against the suction vacuum pressure, and thus preventing the suction of air. The novel function achieved by inserting the resilient sealing rings between the pintle and barrel under compression, and in such a manner that the sealing means will take up any and all clearance between pintle and barrel, is that it will slide in pressure engagement against one of the coacting pintle and barrel members. This pressure contact will not prevent axial flow of slip from the pintle and barrel parts axially out of the capillary clearance, but it is strong enough to prevent the reverse flow of air into the pump.

The prior art consistently neglected this important issue of the efficient operation of a high pressure pump, and did not provide air seals against the suction of air during suction periods of the pump. According to the spirit of this invention, a combination seal between pintle and barrel is so provided that one seal provides sufficient resistance for sealing the high pressure fluid during pressure cycles, to limit pressure losses to about five to ten percent of the rated G. P. M. of the pump, while the other seal provides suction seal against the incoming air between the free ends of pintle and barrel respectively. It has been discovered that the exclusion of air from the pump and connected hydraulic system is not highly desirable but it is imperative. Elastic air under 3000 p. s. i., included and mixed with operating fluid is very destructive, causes extreme vibrations in the system, leaks through the pipe joints where the oil itself is self sealing, but the air opening a way for escape blows the oil out with itself. Furthermore, a broken or cracked pipe would be very dangerous at this pressure, when the contaminated air reaches a certain point of accumulation in the working fluid.

Other objects will become apparent from a reading of the following description, the appended claims, and the accompanying drawings, in which:

Figures 1 to 7, inclusive, show one specific form of hydraulic machine embodying the invention.

Figure 1 is a central, horizontal, longitudinal section taken on the center line 1—1 of the pintle and the control rods of the stationary reactance, in Fig. 8.

Figure 2 is an enlarged fragmentary section of the suction seal as it is assembled between pintle and barrel.

Figure 3 is a fragmentary longitudinal section of a simple pintle, cylinder barrel, and anti-friction bearing arrangement equipped with novel sealing means in accordance with one feature of the invention.

Figure 4 is a view similar to Figure 3 but showing the sealing means in connection with bushing bearings instead of the anti-friction bearings shown in Figure 3.

Figure 5 is an enlarged fragmentary fractional view of my novel sealing means, taken from Figure 4.

Figures 6 and 7 are side and sectional views of a resilient sealing ring provided as one embodiment of my novel resilient sealing means.

Figure 8 is an end view in cross-section further illustrating the hydraulic machine shown in Figure 1.

Figure 9 is an enlarged sectional view taken through the pintle, barrel and resilient seal assembly in Fig. 1, showing the pintle, barrel and the resilient seal in concentric operating positions.

Fig. 10 shows an enlarged sectional view taken through the pintle, barrel and resilient seal assembly in Fig. 1, showing the three elements in eccentric operating positions.

Figure 11 is an enlarged sectional view taken through the pintle, barrel and resilient seal assembly in Fig. 1, showing the three elements in eccentric relative positions to one another.

Fig. 12 is a diagram showing the circle of eccentricity on which the center of the cylinder barrel and the pintle respectively are moving during the operation of the two. This diagram corresponds to the showing of Fig. 10.

Fig. 13 is a diagram showing the circle of eccentricity on which the center of the cylinder barrel and the pintle are moving during the operation of the two. This diagram illustrates the showing Fig. 11.

The machine shown in Figures 1 and 2 include a casing 94 provided with an end plate 95 held in place by screws 96. A pintle 97 is formed with a valve portion 98 and with a mounting portion 99 fixedly secured in a casing boss 100. A cylinder barrel 101 is mounted for rotation about the pintle portion 98 and is connected to a shaft cap 102 by means of screws 103. The shaft or cap is journaled in ball bearings 104 carried by the casing end plate 95.

The cylinder barrel 101 is formed with a single set of circumferentially deployed radial cylinders 105 equipped with reciprocable pistons 106. Pins 107 respectively fitted to the outer ends of the pistons engage reactance tracks 108 and 109 carried by a rotatable inner reactance assembly formed of two halves 110 secured together with bolts 111. The inner reactance element is mounted for rotation by ball bearings 112 carried by an outer reactance structure comprising end plates 113 secured to a central ring 114 by means of screws 115.

Referring now more particularly to Figures 1 to 7, to the novel features of the form shown in this division, namely to the pintle 99, cylinder barrel 101 and interposed resilient sealing rings 122—122, and the capillary sealing means C, as in Figure 2, it will be seen that the invention comprises two main features, and their combination. Namely, first the effect of a uniformly distributed and capillary clearance space C, which means seals by the viscosity of the oil (the working fluid) and the new phenomena of varying the sealing effect of the capillary seal by varying the concentricity of the pintle 97 and the barrel 101. The second feature of this invention lies in the application of resilient sealing means, such as 122—122 made out of compressible seal material, such as various compounds of artificial rubber, cork, "neoprene," etc. Also important non-metallic seal is the graphite and its various mixtures with bearing metals such as lead, copper, tin, etc., whereby the non-metallic seal ring may slide upon the pintle in direct contact with the pintle as shown more specifically in the enlarged view of Figure 2.

The third and really the resultant phenomena caused by the combination of the above two sealing means is the effect of direct sealing of the capillary clearance C by the non-metallic or resilient seals 122—122. The seals 122—122 will prevent the suction port 116 from taking in air through the capillary clearance C, which obviously occurs when a sustained long range suction period is taking place in the pump. During a long suction period, when there is no new slip coming forth, and the oil film contained in the circular concentric clearance C between pintle and barrel gradually will be sucked into the suction port 116ª and thus destroyed, air will enter into the suction port 116ª, contaminate the working fluid and cause destructive vibration and noise in the pump and in the interconnected hydraulic system. Additionally, the intruding suction air dries up and destroys all vestige of the oil film between pintle and barrel, at least upon the suction side area of the working portions 97 and 98 of the pintle.

When the oil film is sucked out under the barrel, in the construction of Figure 1, the barrel will drop down and will touch the upper side of the pintle in a metal to metal rubbing contact. Equally detrimental is the phenomena which occurs after the pintle and barrel come into metal to metal contact on the one side of the pintle. This is the phenomena of the offset eccentric position of the pintle and barrel and the distortion of the previously uniform circumferential capillary clearance C between pintle and barrel. The metallic contact between pintle and barrel causes the rapid wear and eventual galling, seizure and welding of the two corotating elements, while the eccentric position of the previously uniform clearance C will increase the working slip to two to three times of its previous normal value, as hereinabove was also mentioned.

In the particular construction of Figure 1 the cylinder barrel 101, the pintle mid portion around the pintle port area 116, and the sealing rings 122—122 at both sides of the port 116 form a fluid tight chamber to retain lubricating pressure fluid for this area. The source of fluid is the pressure in port 116, which tries to escape right and left under the operating pressure. It is obvious as previously described that to prevent excessive leakage between pintle and barrel, the capillary clearance C and seal length between the port 116 and the seal rings 122—122 are provided to effect the so-called high pressure seal. The so-called "suction seal," to prevent the suction of air and to keep the above named clearance space C with fluid at whatever pressure for the sake of capillary lubrication, is the main duty of the resilient seal 122—122. As long as the pump is in operation the pump pressure port 116 will deliver discharge fluid into the closed chamber C. It is true that the suction port 116ª, opposite to the pressure port 116 is trying to suck in right away whatever film fluid it can from this closed clearance chamber C, so that the pressure discharge in port 116 tends to be short circuited and flow into suction port 116ª. However, only the excess or high pressure slip can escape the resilient seals 122—122, and not the low pressure or idling pressure, which will be retained for air seal and lubrication.

In operation, when the cylinder barrel is rotated and the reactance structures are set eccentrically to the pintle axis, the pistons will be reciprocated so as to draw in and expel fluid through pintle ports 116ª and 116. Preferably the slip of fluid along the pintle is minimized by the provision of sealing rings 122 which assist in preventing the sucking in of air at the inner ends of the cylinders. The rings 122 form the ends of a pressure film chamber between the pintle and cylinder barrel bore and thereby confine a pressurized film of lubricant circumferentially around the entire intervening portion of the pintle instead of allowing the lubricant to leak along the pintle to the ends of the cylinder barrel. Since fluid under pressure leaks along the pintle to act on the end of the shaft cap 102, axial pressure thrust on the cap and on the barrel will be taken up by bearing 104.

The provision of sealing rings between the pintle and the wall of the barrel bore is advantageous generally in hydraulic machines including a cylinder barrel element and a pintle element, one of which elements is rotatable relatively to the other. Figure 3 shows a simple sub-assembly including a pintle 500, a cylinder barrel 501, and anti-friction bearings 502 mounting the elements 500 and 501 for rotation, one with respect to the other. In order to seal the small working clearance C between the pintle and barrel bore wall, non-metallic sealing rings 503 are mounted in seats 504 in the barrel 501 so as to have intimate running contact with the pintle on opposite sides of the usual registering pintle and barrel ports 505—506. The non-metallic, e. g. carbon or graphite rings 503 can be fitted so closely to the pintle as to prevent the loss of lubricant from the clearance between the pintle and barrel, and also to prevent the sucking of air through this clearance and into the cylinder or cylinders. The barrel-pintle sealing rings 503 are particularly advantageous when used in connection with modern axially short pintle and barrel combinations.

The construction illustrated in Figure 4 is similar to that shown in Figure 3 differing only in the use of plain bushing bearings 502ª in place of the anti-friction bearings 502 shown in Figure 3. The non-metallic sealing rings 503 may be used advantageously in connection with either of these or with other types of bearings.

In effect, the prevention of loss of the pressure slip by the opposite sealing means 122—122 not only provides an efficient and economic means for lubrication of the pintle, but it prevents destructive wear, suction of air and subsequent noise and vibration not only in the pump, but in the entire system. The fact is that the slip when retained in this manner constitutes a pressure oil cushion between pintle and barrel, which pressure is sufficient to keep the pintle and barrel in substantially concentric relation as shown in Figures 2 and 5 respectively. Otherwise it is necessary to mount the barrel or the pintle on additional pilot bearings, such as 502—502 in Figure 2 to maintain concentricity between pintle and barrel. Thus in the broad aspect of the invention, the pressurized body of oil in clearance C actually constitutes forced feed lubrication, and hydraulic means to center the pintle with respect to the barrel bore at all operating pressures. Namely, when the seals 122—122 as in Figure 1, 503—503 as in Figure 3, and Figure 4 are inserted between pintle and barrel, pressurized fluid body is trapped and preserved in the telescoping capillary clearance C therebetween in such a manner that during a subsequent pressure cycle, when the pressure acts in one port of the pintle and creates a heavy hydrostatic wedging action between the coacting surfaces, the trapped fluid will resist the wedging action and permit the new pressure to propagate in the entire body of entrapped lubricant and thus balance the entire pressurized body of oil and with it the pintle with respect to the barrel. On the other hand, if before an immediate pressure impulse or shock, the suction side of the pintle is empty, the wedging action of the pressure will pull the pintle and the barrel into metallic contact, and no balancing can take place, because there is no fluid on the suction side of the pintle and barrel which would serve as fluid medium to either take up and resist the wedging action by pressure, or by transmitting pressure through propagation of pressure impulse, from the pressure side to the suction side of the relatively rotating assembly of the pintle 500 and barrel 501.

Thus in the concept of the invention, there is an automatic pressure balancing method which comprises the provision of a fluid tight chamber such as is provided by end seals 503—503, pintle 500 and barrel 501, a uniform clearance space C between 500 and 501, and fluid pressure means to maintain the fluid tight chamber with working fluid at all phases of the pump operation.

Figure 5 also shows that where a sleeve bearing is used in combination with pressure seals 503, the preferred clearance between the pintle and the sleeve is the same as the preferred clearance between pintle 500 and barrel 501.

The seal rings 503 are cast or molded and finished to such sizes as to give the necessary sealing action around its entire periphery by compression or close machined fit, like in the case of graphite rings. The sealing pressure of these rings shall be around a couple hundred pounds per square inch, so that actual pressure will be maintained in the sealed chamber C. The compression on the packing 503 is greater than the pressure to be sealed in chamber C.

The operation of my novel resilient seal rings, shown in Fig. 1 may best be described and understood in connection with Fig. 9, Fig. 10 and Fig. 11 respectively. This invention supplements the heretofore conventional close-ground metallic seal C between pintle 97 and barrel 101 by resilient, non-metallic or metalloid seal rings 122—122 at opposite sides of the pintle ports 116a—116 respectively. By this combination of the viscous fluid seal, which is the metallic or inorganic seal with the non-metallic or metalloid seal, many heretofore unknown advantages are obtained. One basic advantage is the one shown in the Figures 9, 10 and 11, whereby the relative eccentric rotation of pintle 97 and barrel 101 upon the eccentricity circles R=C in Fig. 12 and in Fig. 13 respectively, the metallic clearance C provided for the relative rotation of pintle and barrel does not remain concentric to the pintle and barrel as shown in Fig. 9, but due to the diametrally opposite location of the suction and pressure ports 116a and 116 of the pintle 97, the pintle and barrel always tend to rotate in an eccentric manner, as shown in Fig. 10 and Fig. 11 respectively.

Fig. 12 and Fig. 13 show this eccentric relative rotation on the enlarged eccentricity circles having for radii R the amount of C, the evently distributed pintle and barrel clearance. Thus, in Fig. 9 the ideal relative position, for which this invention is aiming, by the provision of seals 122—122 is shown. In this figure, the resilient seal 122 and the seal clearance C are also concentric with the pintle 97 and barrel 101 and all are disposed about one center of rotation marked O in Fig. 9. In Fig. 10, however, two rotational centers manifest, such as O_1 and O_2, O_1 being located on the top, O_2 on the bottom of the circle of eccentricity as in Fig. 12. In Fig. 11, on the other hand, O_2 appears on the top, O_1 on the bottom of the circle of relative eccentricity, as a consequence of a pressure and flow reversal of the variable delivery reversible pumps. While the location of the center of relative rotation are reflected by Fig. 12 and Fig. 13, the facts pertaining to the variation of the bodily shape of the resilient seal 122 are shown in Fig. 9, Fig. 10 and Fig. 11. The change in the shape of the resilient seal is just capable to seal the eccentric clearance spaces C_1, C_2, C_3 respectively, in such a manner that the high pressure viscous seal will be additionally increased by the sealing power of resilient member 122 and at the same time the entrance of air during suction cycles will be entirely eliminated. As the viscosity of air is much less than the viscosity of oils, it is obvious that the large openings between pintle and barrel as shown by the dotted lines of C_1, C_2 and C_3 in Fig. 9, Fig. 10 and Fig. 11 would provide prohibitive entrance for low viscous fluids such as air, into the pump during suction periods. While the important structural merits of the invention are shown in Fig. 1, it is obvious that the non-metallic or metalloid seals 122—122 are so located in the barrel 101 or in the pintle 97, as the case may be, that the high pressure at sudden development cannot blow out the seals 122—122, because they are well recessed, housed and protected in one of the said members only, while they are disposed in axially floating position with respect to the other of the said members.

The distance of the seals 122—122 with respect to the pressure and suction ports 116—116a may vary as much as the pintle length, yet due to the linear drop of pressure from the ports out to the right or to the left, it will be seen that in any disposed axial positions the seals 122—122 will achieve substantially the same sealing, lubricating, balancing and centering effects for the pintle 97 and the barrel 101. When the rings 122—122 are made out of synthetic or natural rubber, under the high pressure of the pressure port 116 they will be tightly pressed in their respective grooves and will seal also the viscous passages C as shown in Fig. 2. Fig. 1 also shows similar effects of rings 117—117 for the benefit of the pistons 106 and cylinders 105 and associated thrust transmitting means 107 and 108 respectively. These latter advantages are claimed and more specifically described in my U. S. Patent No. 2,452,541, issued Nov. 2, 1948.

Fig. 8 shows more particularly the pressure and exhaust ports 116 and 116a, and that one-half of the total number of pistons are under pressure while the other half are under negative pressure, that is vacuum. It is this vacuum, which is more dangerous for the life of the piston and cylinder assemblies than the pressure, for which this invention provides positive remedies in the form of better lubrication, by the elimination of friction and air between the suction pistons and the suction cylinders and between the suction half of the pintle and barrel respectively.

The importance of the advantage obtained by providing the fluid containing sealed chamber can be appreciated fully only when the importance of lubrication is considered. For instance, in purely mechanical machines complete lubrication is provided, but in a conventional pump, one-half of the pintle surface is running without lubrication. With conventional or previously known structures the coefficients of friction of the pintle and barrel on the suction side may be about .3 to .5, whereas with pressure lubrication afforded by the present invention the coefficient will be reduced to less than .01. The above named improvement is necessitated by the novel pump structure which reduced a large capacity pump to relatively small component parts. This could not have been achieved without the provision of more effective lubrication than heretofore was provided by the prior art.

Although several specific embodiments of this invention are herein shown and described, it will be understood that numerous details of the construction shown may be altered or omitted without departing from the spirit of this invention as defined by the following claims.

I claim:

1. In an hydraulic machine of the character described, a relatively fixed pintle, a cylinder barrel rotatably mounted on said pintle and having a central disc portion with radially extending lateral side walls limiting said disc portion, pistons reciprocable respectively in said cylinders and projecting beyond the outer ends thereof; a reactance structure having axially separable side walls lying in face-to-face abutting relation with the adjacent lateral side walls of the disc portion of said barrel and surrounding and eccentric to the axis of the pintle; thrust transmitting means interposed between said reactance structure and the outer ends of the pistons; means providing fluid seal between said lateral side faces of said cylinder barrel and between said axially separable side members of said reactance structure, said sealing means said cylinder and said reactance structure providing a substantially fluid tight chamber pressure fluid lubricating bath and accommodating said outer ends of the pistons, associated cylinders and said thrust transmitting means respectively, for providing fluid seal and forced feed lubrication for said pistons and cylinders and said thrust transmitting means respectively, said means including a seal and groove-way combination, said seal grooveway being provided in one of the said barrel disc walls and axially separable side members of said reactance structure and operable against the other of said members.

2. In an hydraulic machine of the character described, a relatively fixed pintle, a cylinder barrel rotatably mounted on said pintle and having a central disc portion with radially extending lateral side walls limiting said disc portion, pistons reciprocable respectively in said cylinders and projecting beyond the outer ends thereof; a reactance structure having axially separable side walls lying in face-to-face abutting relation with the adjacent lateral side walls of the disc portion of said barrel and surrounding and eccentric to the axis of the pintle; thrust transmitting means interposed between said reactance structure and the outer ends of the pistons; hydraulic means providing fluid seal between said lateral side faces of said cylinder barrel and between said axially separable side members of said reactance structure, said sealing means said cylinder and said reactance structure forming a substantially fluid tight chamber for accommodating said outer ends of the pistons and associated cylinders, said hydraulic means including metalloid sealing rings, said rings being carried by one and engaged by the other of said barrel and reactance structure respectively.

3. In an hydraulic machine of the character described, a relatively fixed pintle, a cylinder barrel rotatably mounted on said pintle and having a central disc portion with radially extending lateral side walls limiting said disc portion, pistons reciprocable respectively in said cylinders and projecting beyond the outer ends thereof; a reactance structure having axially separable side walls lying in face-to-face abutting relation with the adjacent lateral side walls of the disc portion of said barrel and surrounding and eccentric to the axis of the pintle; thrust transmitting means interposed between said reactance structure and the outer ends of the pistons; means providing fluid seal between said lateral side faces of said cylinder barrel and between said axially separable side members of said reactance structure, said sealing means said cylinder and said reactance structure providing a substantially fluid tight chamber for accommodating said outer ends of the pistons and associated cylinders, said means including resilient sealing rings; said sealing rings being arranged in paired assemblies between each side of said cylinder barrel, and the coacting faces of said reactance structure respectively.

4. In an hydraulic machine of the character described, a relatively fixed pintle, a cylinder barrel rotatably mounted on said pintle and having a central disc portion with radially extending lateral side walls limiting said disc portion, pistons reciprocable respectively in said cylinders and projecting beyond the outer ends thereof; a reactance structure having axially separable side walls lying in face-to-face abutting relation with the adjacent lateral side walls of the disc portion of said barrel and surrounding and eccentric to the axis of the pintle; thrust transmitting means interposed between said reactance structure and the outer ends of the pistons; means providing fluid seal between said lateral side faces of said cylinder barrel and between said axially separable side members of said reactance structure, said sealing means said cylinder and said reactance structure providing a substantially fluid tight chamber for accommodating said outer ends of the pistons and associated cylinders, said means including resilient sealing rings; said sealing rings being arranged in paired assemblies between each side of said cylinder barrel and the coacting faces of the associated reactance structure, and being under compressive forces in all directions of the space to seal a portion of the slip fluid in said fluid tight chamber, and provide an oil bath for said

pistons and cylinders and coacting thrust transmitting means respectively.

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