United States Patent [19]

Ankeny et al.

[11] **3,747,476**

[45] **July 24, 1973**

[54] **BALANCED HYDRAULIC DEVICE**

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[22] Filed: **Apr. 4, 1972**

[21] Appl. No.: **240,974**

Related U.S. Application Data

[62] Division of Ser. No. 24,163, March 31, 1970, Pat. No. 3,682,044.

[52] **U.S. Cl.** **91/489**, 91/506

[51] **Int. Cl.** **F01b 13/04**

[58] **Field of Search** 91/504–506; 917/222, 486–489

[56] **References Cited**

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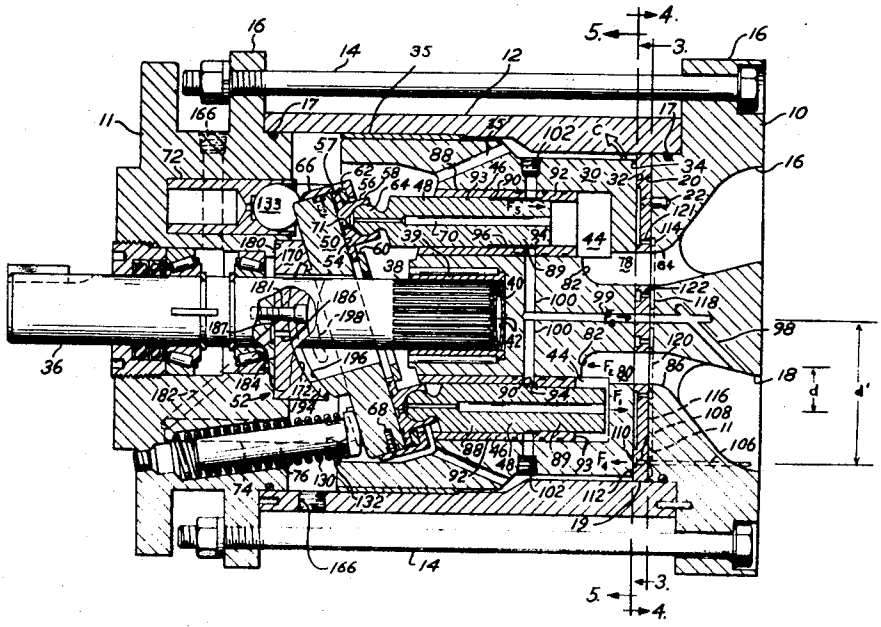
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Attorney—Daniel M. Riess

[57] **ABSTRACT**

In an axial piston hydraulic device, having an angularly tiltable tilt plate for controlling the stroke of the pistons, the cylinders in the rotor communicate with the high and low pressure ports of the device by way of passages which are positioned between the axes of the cylinders and the rotational axis of the rotor to reduce resistances to fluid flow resulting from rotor rotation when the cylinders are on their low pressure stroke. Additional passages are provided which communicate the high pressure port with the rotor and/or the bearing of the tilt plate to exert counterbalancing forces on these components during operation of the device. On and off-stroke control pistons may also be positioned on the side of the tilt plate opposite the pistons to exert counterbalancing forces on the tilt plate.

19 Claims, 11 Drawing Figures



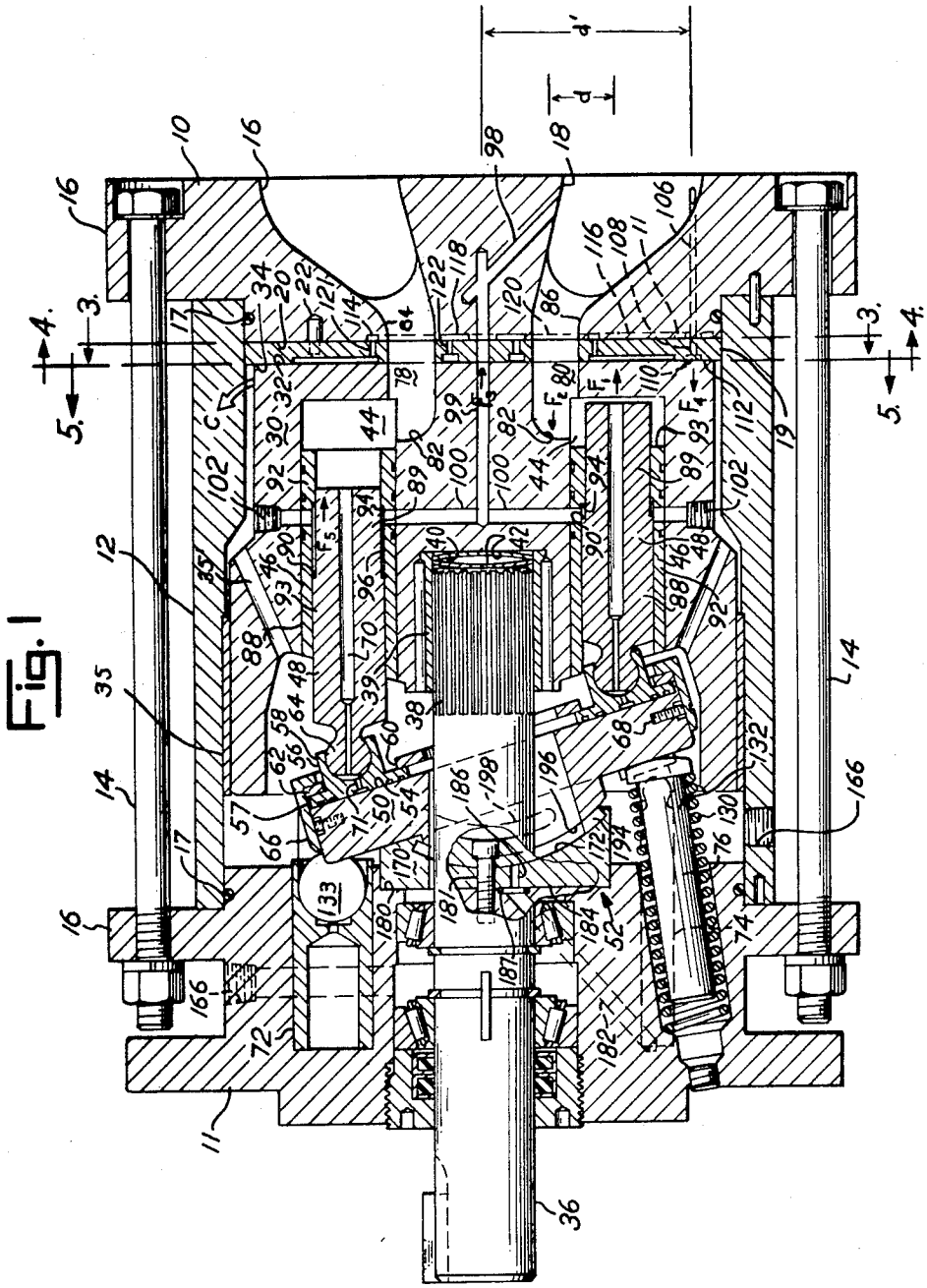


Fig. 1

Fig. 2

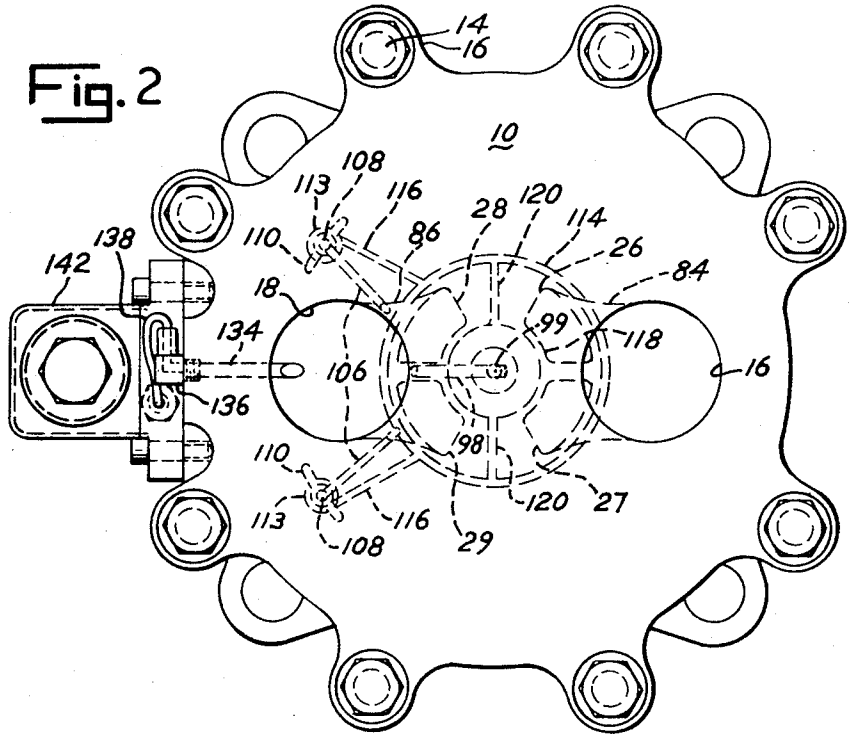

14
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113
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Fig. 3

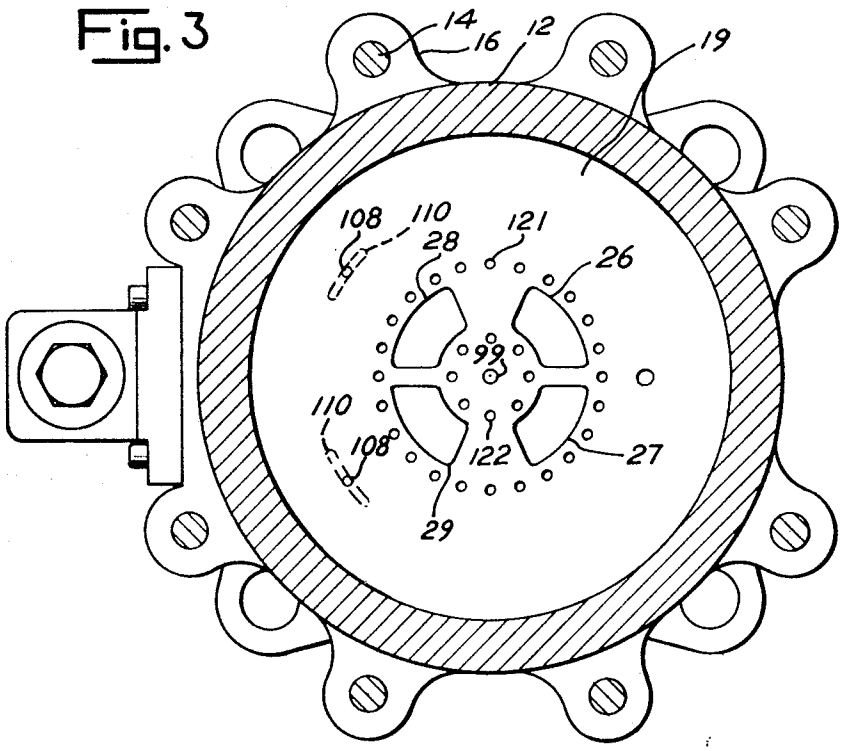

14
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Fig. 4

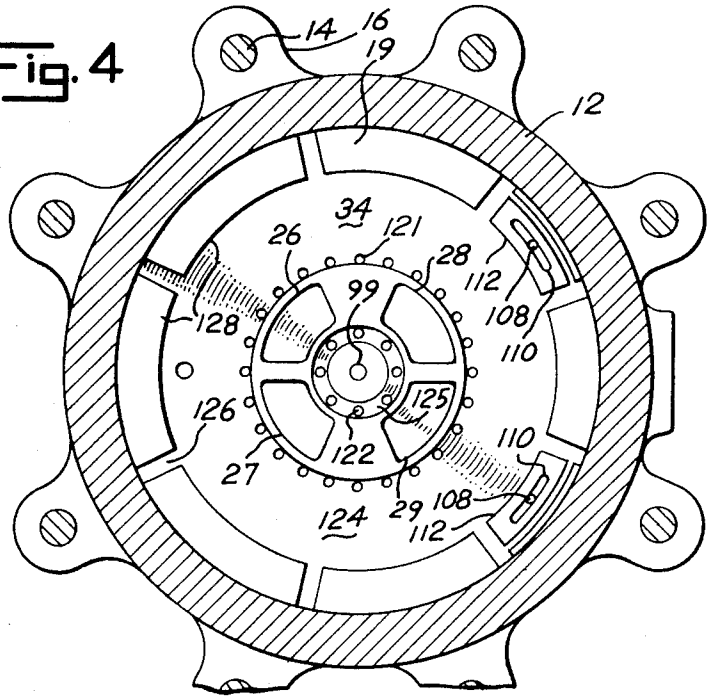


Fig. 5

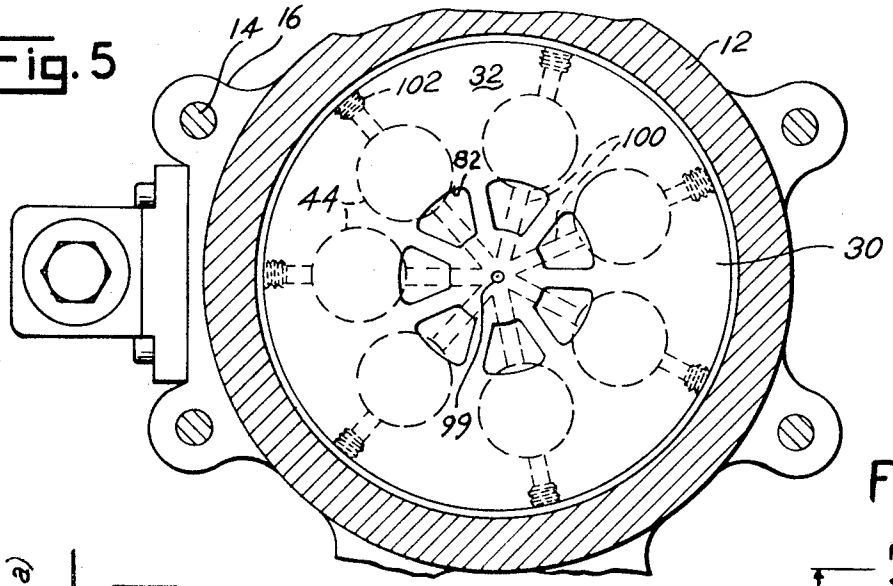


Fig. 6

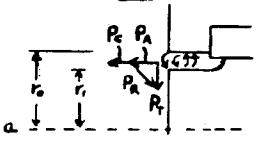


Fig. 6A

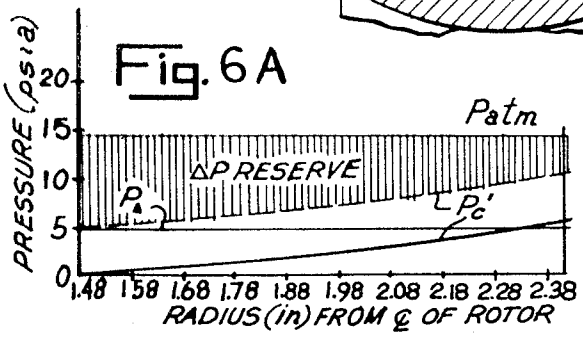
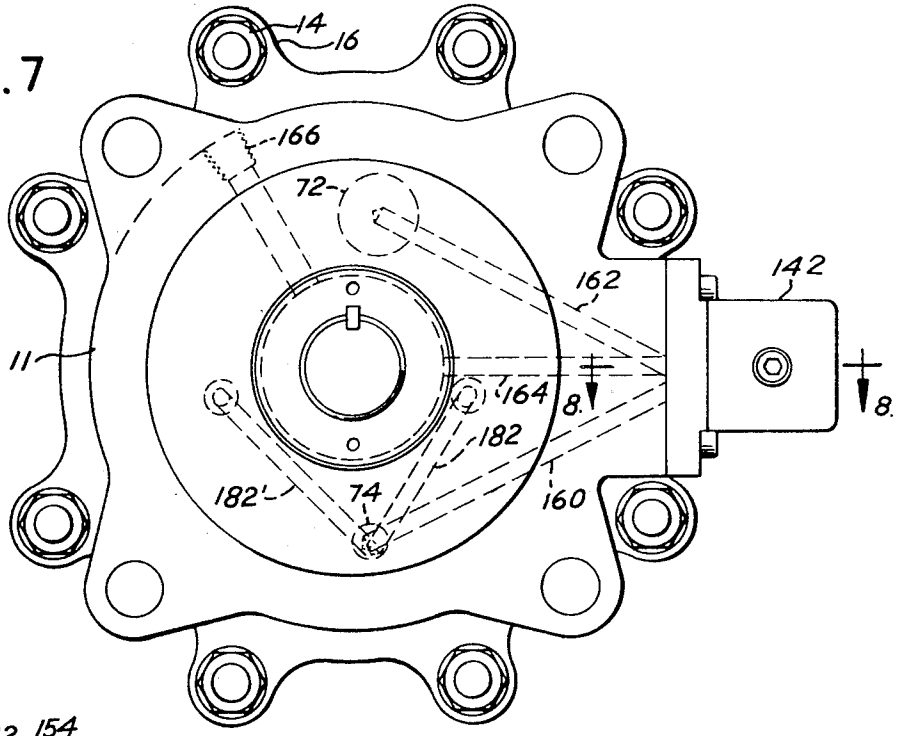


Fig. 7



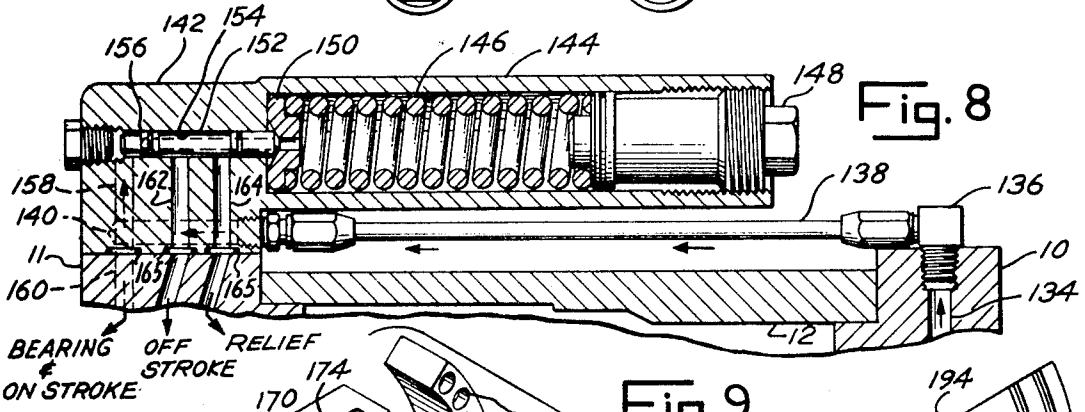


Fig. 8

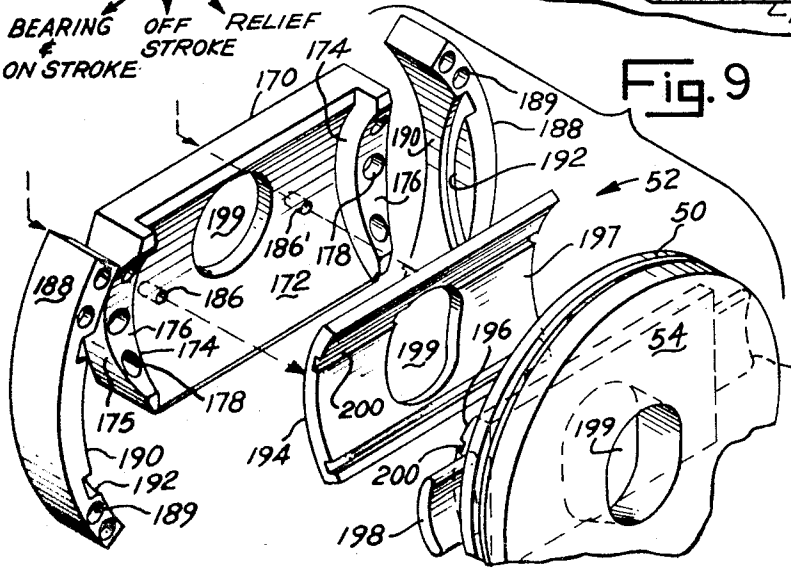


Fig. 9

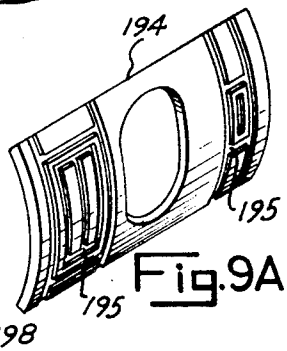


Fig. 9A

BALANCED HYDRAULIC DEVICE

This is a division of application Ser. No. 24,163, filed Mar. 31, 1970 now U.S. Pat. No. 3,682,044.

BACKGROUND AND SUMMARY OF THE INVENTION

This invention relates to a pressure device and, more particularly, to a hydraulically balanced axial piston pump or motor.

Where a hydraulic pressure pumping device is to be supplied with liquid from a source at atmospheric pressure, starvation of the cylinders of the pump which are on the suction stroke frequently occurs as a result of resistances developed within the passages supplying fluid to the cylinders which oppose the filling of the cylinders. These resistances arise as the result of several design features common to such axial piston pumps. One of the design features of such pump which contributes in more than an insignificant amount to the creation of resistance, is the necessary radial spacing of the pumps's cylinders and the passages which supply the cylinders from the rotational axis of the rotor of the pump. This spacing, combined with the relatively high rotational speeds of the rotor, imparts centrifugal as well as tangential forces to the fluid enroute to the cylinders. These forces, as well as axial forces due to orifice effect, tend to resist the flow of fluid into the cylinders. Where the pump is large and where the fluid is to be supplied at atmospheric pressure, these resistances may become sufficiently large so as to preclude the use of the pump without appropriate auxiliary booster devices. In addition, where the pump is operated at or near such starvation level, both an increase in wear and operational noise level result.

In one principal aspect of the invention, the pump constructed in accordance with the principles of the invention, is capable of a substantial reduction of the likelihood of starvation and the attendant undesirable consequences of starvation or near starvation. In the pump of the invention, resistance resulting from centrifugal as well as tangential effect may be substantially reduced. By reducing these resistances, the pump constructed in accordance with the principles of the invention may be larger than those previously employed for pumping fluid from fluid sources at atmospheric pressure and the need for booster devices is substantially reduced. In the pump of the Invention, the passages which communicate the ports with the cylinders of the device are radially offset from the cylinders of the pump toward the rotational axis of the rotor so as to substantially reduce resistance to the flow of fluid into the cylinders during the suction stroke of the pistons.

Also in such axial piston devices, substantial unbalancing forces and mechanical couples are encountered during operation on various components of the device including the tilt plate, piston shoes and rotor. These unbalancing forces not only increase the wear on these components, but also result in substantial increases in the noise level of the pump during operation.

In the fluid pressure device constructed in accordance with the principles of the invention, these several unbalancing forces and couples are counterbalanced to effect a substantial reduction in both wear and operational noise of the device. Counterbalancing is accomplished by one or both the direction of high pressure fluid to various of the components and the positioning of the tilt plate control pistons. This high pressure fluid may be supplied directly from the high pressure port of the device.

In one preferred embodiment of device, fluid at high pressure from the high pressure port of the device is ported, by way of passages, both to the cylinders and to the end of the rotor adjacent the high pressure ports and there exerts forces on the rotor along its rotational axis and adjacent the cylinders of the device which communicate with the high pressure port respectively so as to offset or counterbalance couples which tend to tilt the rotor of the device.

In another preferred embodiment of device, high pressure fluid from the high pressure port of the device is ported to the tilt plate of the device to exert a force on the tilt plate which tends to counterbalance the oppositely directed forces exerted by the working pistons during operation of the device. This fluid force is preferably directed to the bearing of the tilt plate so as to form a fluid cushion upon which the tilt plate is pivoted. This cushion not only results in reduced bearing wear, but also sound insulates the device against the transmission of noise through the device during operation. In addition,excess pressure obtaining in the high pressure port of the device may be relieved through the latter mentioned bearing of the invention.

In addition to directing high pressure fluid so as to counterbalance the forces generated in the device during normal operation, the control pistons, which selectively vary the angular disposition of the tilt plate, are also positioned on the side of the tilt plate opposite the working pistons of the device. Thereby, the forces exerted by these pistons during operation, also tend to counterbalance the forces exerted on the tilt plate by the working pistons of the device, as well as effect a substantial reduction of the overall diameter of the device. These control pistons may also be operated by way of high pressure fluid ported thereto from the high pressure ports of the device

In a fluid pressure device incorporating the principles of the invention sound transmission during operation is substantially reduced by way of fluid insulation.

These and other objects, features an advantages of the present invention will be more clearly understood from a consideration of the following detailed description.

BRIEF DESCRIPTION OF THE DRAWINGS

In the course of this description, reference will frequently be made to the attached drawings in which:

FIG. **1** is a cross sectioned plan view of a hydraulic pump incorporating the principles of the invention, the right end port cover having been rotated **90°** for the purpose of clarity;

FIG. **2** is an end elevation view of the pump as viewed from the right in FIG. **1**;

FIG. **3** is a cross sectioned elevation view of the port plate of the pump taken substantially along line **3—3** of FIG. **1**;

FIG. **4** is a cross sectioned elevation view showing the other side of the port plate and taken substantially along line **4—4** of FIG. **1**;

FIG. **5** is a cross sectioned elevation view of the rotor taken substantially along line **5—5** of FIG. **1**;

FIG. **6** is a schematic showing centrifugal, axial and tangential effects on the fluid entering the device during suction;

FIG. 6A is a plot of pressure v. passage raduis from the center-line of the rotor and showing the effect on the suction head of the pump as the radius of spacing of the passages increases;

FIG. 7 is an end elevation view of the pump as viewed from the left in FIG. 1 and showing the hydrostatic pressure passage arrangement of the control pistons and tilt plate bearing of the pump;

FIG. 8 is a cross sectioned elevation view of the discharge compensator of the pump taken substantially along line 8—8 of FIG. 7 and showing the connection of the hydrostatic pressure passages of FIG. 7;

FIG. 9 is an exploded view of the tilt plate trunnion bearing; and

FIG. 9A is a view of the opposite side of the trunnion bearing insert shown in FIG. 9.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIG. 1, the pump incorporating the principles of the invention is of the axial piston hydraulic type in which a tilt plate is provided for determining and/or adjusting the stroke of the pistons and hence capacity of the pump.

By way of general description, the pump includes a pair of end covers 10 and 11 having a cylindrical housing 12 extending therebetween. The end covers and housing are drawn firmly together by a plurality of tie bolts 14 extending parallel to the axis of the housing between flanged ears 16 on the covers and suitable seals 17 are provided to form a sealed enclosure.

Referring to FIGS. 1 and 2, end cover 10 includes a pair of relatively large circular ports 16 and 18, port 16 being a suction port and the other port 18 a discharge port. Each of these ports is adapted to be coupled to suitable hydraulic lines (not shown) for supplying the pump with hydraulic fluid and for receiving the discharge of the pump.

As shown in FIGS. 1–4, a port plate 19 is stationarily attached to the inner face 20 of the cover 10 by way of one or more pins 22. The port plate 19 has a plurality of ports 26, 27, 28 and 29 for communicating ports 16 and 18 respectively with the respective ones of the cylinders of the pump.

A rotor 30 is next positioned in the housing such that its end face 32 rotatably bears against the left face 34 of the port plate as shown in FIG. 1. The rotor is spaced from the inner wall of the cylindrical housing 12 and is rotatable relative thereto. A hydrodynamic journal bearing 35 is positioned between the rotor and cylindrical housing to receive the lateral load on the rotor as a result of the piston's conjunction with the tilt plate. Passages 35' extend angularly through the rotor to provide a pumping action between the sides of the rotor. This pumping action provides a source of cool fluid from the casing to the bearing.

The rotor is driven by way of a drive shaft 36 which extends through end cover 11 where it is adapted to be coupled with a suitable drive source (not shown). The drive shaft 36 also extends through the tilt plate and its bearing and the inner end of the shaft is formed with a plurality of longitudinally extending external splines 38 which are adapted to engage corresponding internal splines 39 in a recessed portion 40 of the rotor, wherely the drive shaft rotatably drives the rotor. One or more dished spring discs 42 are providPd between the end of the shaft 36 and the rotor to allow for some degree of axial play between the rotor and the drive shaft during operation, yet urge the rotor into rotating contact with face 34 of the port plate 19.

The rotor 30 also includes a plurality of cylinders 44 which are spaced parallel from each other and also radially from the rotational axis of the rotor. Each of the cylinders 44 includes a cylinder sleeve 46 which is press fitted or the like in the cylinder and which receives a piston 48 therein. One end of the piston 48 extends outward beyond the sleeve and toward the tilt plate.

The tilt plate assembly comprises an annular plate 50 which is tiltable upon a trunnion bearing assembly, generally 52, so as to vary the stroke of the pistons 48. The face 54 of the tilt plate toward the rotor provides a flat cam surface upon which a plurality of piston shoes 56 are positioned, the bases 57 of each of the piston shoes being slideable relative to the face 54 of the tilt plate. Each of the piston shoes 56 also includes a socket portion 58 which extends outward from the base 57 through apertures 60 in a piston shoe holddown plate 62. The sockets 58 receive a complimentary ball 64 formed integrally on the end of the respective pistons 48 and provide for pivotal movement of the pistons relative to their shoes. The holddown plate 62 is clamped in overlying relationship to the bases 57 of the piston shoes by suitable brackets 66 attached about the periphery of the tilt plate as by bolts 68. The tilt plate 50, holddown plate 62, and piston shoes 56 are of well known design and will not be described further in detail. In addition, according to preferred conventional construction, a longitudinal oil passage 70 extends through the length of the pistons 48 to provide for lubrication of the ball sockets and the sliding surfaces of the bases 57 of the piston shoes 56 by way of passage 71 and also for hydrostatic balance of the ball and bases of the pistons and shoes.

In order to adjustably vary the pump capacity, one or more control pistons are also provided for varying the tilt of the tilt plate 50. As shown in FIG. 1, piston 72 is an off-stroke piston and piston 74 is an on-stroke piston. In addition to pistons 72 and 74, a spring loaded piston 76 may also be provided which exerts a continuous force on the tilt plate to place the tilt plate on full stroke upon starting of the pump when sufficient hydraulic pressure is not yet available to operate control pistons 72 and 74.

In a preferred embodiment, the suction and discharge passages of the pump which communicate the ports 16 and 18 with the cylinders, are radially offset from the rotor axis by a distance substantially less than the radial spacing of the axes of the respective cylinders from the rotor axis. These passages 78 and 80 are formed by passages 82 in the rotor which communicate with cylinders 44 at their innermost side. In turn, passages 82 communicate in sequence with ports 27, 26, 28 and 29 in port plate 19 as the rotor rotates. These ports are also spaced closer to the rotor axis than the cylinder axes. Ports 26 and 27 in port plate 19 both communicate by way of a varied cross section passage 84 with suction port 16 and ports 28 and 29 communicate with port 18 also by way of a varied cross section passage 86. Passages 84 and 86 where they open to the port plate ports, are of substantially the same cross section as the latter ports.

The offset of passages 82 toward the rotor centerline or axis makes it possible to utilize larger pumps where the pump is to be supplied with a source of fluid which

is at atmospheric pressure and obviates the need for a booster of priming pump to prevent starvation of the pump during the suction stroke. Where the pump size is to be increased with an attendant increase in the radial spacing of the respective axes of the cylinders and the passages **82** from the rotation axis, both the tangential and centrifugal effect due to rotor rotation is such that resistance to filling of the cylinders on the suction stroke increases.

Referring to FIG. **6**, the pressure which must be present to fill the cylinders without starvation occuring must be sufficient to overcome the resistance to filling due to three separate factors, the axial effect, the tangential effect and the centrifugal effect.

The axial effect creates a pressure resistance due to the pressure drop encountered in passage **82** at its ported interface adjacent the port plate **19.** This effect, denoted P_A, varies inversely to the square of the smallest cross sectional area of the passages **82.** This axial resistance may be represented by the formula

$$P_A \alpha Q/A^2$$

where P_A is the axial resistance pressure, Q is the flow rate and A is the minimum cross sectional area of the passages **82.**

The tangential effect also creates a resistance to filling by way of the tendency of the inflowing fluid to miss the rotor passage **82** as it speeds by. This effect may be likened somewhat to the problem encountered in boarding a merry-go-round as it speeds by. The tangential effect is denoted by P_T in FIG. **6** and varies by the formula

$$P_T \alpha (RPM^2) \times (r_o^2)$$

where RPM is the rotational speed of the rotor, and r_o is the maximum radial spacing of the passages **82** from the rotational axis a of the rotor.

The pressure in passage **84** must be at least of a magnitude which is sufficient to generate a resultant force or pressure P_R which will overcome both these effects or the pump will be starved for fluid. In addition to the resistances P_A and P_T to filling the cylinders, a resistance also results from a centrifugal effect due to the rotational speed of the passages **82.** This resistance may be generally represented by the formula

$$P_C \alpha RPM^2(r_o^2 - r_i^2)$$

where P_C is the centrifugal effect, RPM and r_o represent the variables previously described and r_i represents the minimum spacing between the passages and axis a. This centrifugal effect results in a tendency of the fluid to commence rotation and produce eddy currents just as it enters the passage **82** in the rotor. These eddy currents, as shown in FIG. **6**, seriously impede the flow of fluid into the cylinder and are frequently accompanied by cavitation which results in wear of the passage walls. Therefore Δ P must also be large enough to overcome P_C as well as P_A and P_T as previously described.

Referring now to FIG. **6A**, a plot is shown of pressure (psia) v. the average radial distance (in.) between the passages **82** and the centerline or rotational axis a of the rotor. Where the supply of hydraulic fluid is at atmospheric pressure, e.g., 14.7 psia, that pressure is the maximum pressure which is available to supply the pump and is represented by P_{atm}. P_{atm} will be diminished by P_A which is substantially constant for a given flow rate Q and passage area A. This P_A may be for example on the order of **5** psia, and thus, where the radial spacing of the ports is 1.48 in. from the rotor centerline, the Δ P RESERVE still available to supply the cylinders over resistance P_A is 9.7 psia. As the distance of the passage **82** to the rotor axis is further increased, the Δ P RESERVE will still be further diminished by the increasing centrifugal effect P_C and tangential effect P_T, these combined effects being represented by the curve P_C' in FIG. **6A**. Thus, as the distance between the passages **82** and rotor axis a is increased, it will be seen that the Δ P RESERVE as shown by the hatched area, will progressively decrease due to P_C' which is shown superimposed on P_A in FIG. **6A**. In fact, if the distance is increased enough, P_C' plus P_A will intersect P_{atm} and no reserve will remain, starving the pump.

By axially offsetting the passages **82** toward the rotor axis from the respective axes of the cylinders with which they communicate, this loss of Δ P RESERVE due to P_C' is substantially reduced for a given rotor speed since r_o and r_i are decreased. The passages **82,** rather than communicating axially with the cylinders **44,** now communicate with the side of the cylinders nearest the rotor axis and the major portion of the passages lie even closer to the axis. This not only decreases P_C and P_T, but also tends to assist flow of the fluid present in passages **82** into the cylinders by a centrifuge effect. Thus, a substantial increase in the Δ P RESERVE is realized and it will be seen that the size of the pump may be increased substantially so long as passages **82** remain relatively close to the rotor axis.

Referring again to FIG. **1,** it will be recognized that during operation of the pumP, when a given piston is to be moved outward in its cylinder on the suction stroke, the piston and its shoe will tend to be drawn away from the tilt plate **50** due to viscous drag and inertia. For example in FIG. **1,** as the upper piston is being drawn outward, a force will be exerted on its shoe toward the right.

In order to compensate for this force and to avoid the need to lighten the pistons by boring and/or light weight inserts in the pistons as frequently found in convention pistons, the piston and its shoe are urged into firm contact at all times against the surface **54** of the tilt plate by hydrostatic pressure. Each of the pistons **48** is stepped in two diameters, a larger diameter **88** and a smaller diameter **89,** the stepped annular surface at the transition between the two piston diameters providing a piston surface **90.** Corresponding portions of the cylinder sleeves **46,** or of the cylinder walls where sleeves are not provided, are also stepped to provide a smaller diameter portion **93** and a larger diameter portion **92** with a stepped surface **94** at the transition between the two portions. Surface **94** is positioned in the cylinder such that when its piston is fully extended into the cylinder, the surface **90** of the piston will still be slightly spaced from the surface **94** of the cylinder. Thus a small annular chamber **96** of varying volume is provided between each of the pistons **48** and their cylinder sleeves **46.**

Referring now to FIGS. **1–5,** pressure passage **98** is bored in the end cover **10** and communicates with the discharge passage **86** at one end and with an axially extending passage **99** at the other end. Passage **99** extends axially through the center of the port plate **19** and partially into the rotor **30.** A plurality of radially extending passages **100** each communicate passage **99** with one of the stepped piston chambers **96.** The short remain-

ing portions of passages **100** between the chambers and the housing which result from the manufacturing operation, are plugged by threaded plugs **102.**

Thus each of the piston chambers **96** will be maintained at discharge pressure by way of passages **98, 99** and **100.** This pressure will act upon the annular piston surfaces **90** of the pistons to urge the pistons, particularly during the suction stroke, where the pistons tend to lift from the tilt plate, to the left to firmly seat their shoes against the tilt plate **50.**

It will be seen that the chambers **96** are effectively interconnected with each other by way of passages **100.** Due to this interconnection, little, if any, flow will occur through passages **98** and **99** since fluid is simply transferred from the chamber of a given piston which is on a discharge stroke to one on a suction stroke by way of passages **100.** Passages **98** and **99** do, however, insure that the pressure in the chambers is maintained at the elevated discharge pressure.

Where the passages **82** are offset toward the rotor axis as previously described, a force couple will be generated on the rotor which tends to tilt the rotor about an axis perpendicular to its rotational axis, or for example, in a counterclockwise direction as viewed in FIG. **1** and as indicated by the arrow C. This couple is caused by the fact the pistons which are on the discharge stroke will exert a force F_1 on the ends of the cylinders and the high pressure discharge fluid will exert a force F_2 on the rotor in a ection opposite the F_1. If the passages **82** extened axially from the cylinders, F_1 and F_2 would balance each other. However, since the passages are offset from the cylinder axes, F_1 and F_2 will be spaced from each other by the offset distance d creating couple C which tends to tilt the rotor. The tendency of the rotor to tilt results in wear to its end face **32** and face **34** of the port plate **19** which are in sliding contact with each other and may result in wear at bearings **35.** The couple C is substantially aggravated where the pump is of large size. This tilting couple is cancelled by way of the stepped piston construction and balance pad construction which will hereafter be described.

Discussing first the stepped piston construction, it will be seen that as pressure fluid is admitted to chambers **96,** the fluid will exert a force on shoulders **94** of the sleeves **46.** These forces will combine to produce a resultant force F_3 which acts along the rotor axis as shown in FIG. **1.**

In addition to the force F_3 exerted by the stepped pistons, a balancing arrangement, as shown in FIGS. **1–4,** is also preferably provided. The balancinG arrangement includes one or more bore passages **106** in end cover **10** which communicate with the discharge passage **86** at one end and with a drilled passage **108** which extends through the thickness of the port plate **19** in that arc of the port plate which is on the discharge side of the pump. The passages **108** in the port plate communicate with elongated slots **110** in lands **112** on the side of the port plate facing thP rotor **30.** End face **32** of the rotor slidably bears against lands **112** and fluid, at discharge pressure, is thereby distributed in the slots **110** to act against the rotor face and exert a force F_4 on the rotor. Forces F_3 and F_4 are separated by distance *d'* to produce a couple which opposes and cancels couple C.

To provide lubrication for the stationary face **34** of the port plate **19** which is in sliding contact with the end face **32** of the rotor **30,** suitable fluid chambers are provided as shown in FIGS. **1–4.** A plurality of grooves may be formed in the inner face **20** of the cover **10,** and may include, for example, a circular groove **113** which surrounds the opening **108** in the port plate **19.** Groove **113** may communicate with another circular groove **114** which surrounds ports **26, 27, 28** and **29** by way of drain grooves **116** in the end cover **10.** Thus, a small amount of the high pressure discharge fluid is leaked from passage **106** to the circular groove **113,** groove **116** and circular groove **114.** In addition, the circular groove **114** may also communicate with an inner circular groove **118** by way of radially extending grooves **120.** The grooves **114** and **120** communicate by way of a pluarlity of drilled passages **121** and **122,** with larger draining recesses **124** and **125** respectively in the face **34** of the port plate which faces the rotating end face **32** of the rotor as shown in FIG. **4.** These recesses also communicate by way of grooves **126** between the bearing lands **128** of the port pltae to the housing casing.

Referring again to FIG. **1,** the control pistons **72, 74** and **76** are positioned in end cover **11** to act against the side of the tilt plate **50** opposite the pistons **48.** This placement of the control pistons effects a substantial reduction in the diameter of the overall pump and tilt plate and also acts to counterbalance the forces exerted on the tilt plate by pistons **48.** This is particularly important where the pump is already of large size.

In the preferred pump, a hydraulically operated off-stroke control piston **72** is provided which acts against the top side of the tilt plate **50** as shown in FIG. **1.** The off-stroke piston exerts a pressure on the tilt plate preferably by way of a fluid loaded ball **133** which rollably contacts the tilt plate. The ball enables the piston to be substantially shortened. In addition, hydraulically operated on-stroke control piston **74** is also provided which acts against the bottom side of the tilt plate tending to increase the tilt of the plate and the stoke of the pistons. A starting on-stroke piston **76** is also preferably provided which acts against the bottom of the tilt plate. This piston **76** is mechanicallly urged against the tilt plate by the force exerted by spring **130** on a plunger member **132** whereas pistons **72** and **74** are preferably hydraulically urged against the plate. The purpose of the starting piston **76** is to tilt the plate to full stroke upon the starting of the pump when sufficient hydraulic pressure has not yet been generated by the pump to actuate control pistons **72** and **74.**

Referring now to FIGS. **2** and **8,** a passage **134** is provided in the port cover **10** which communicates at one end with the discharge passage **18,** as shown in FIG. **2,** and at the other end is adapted to receive an elbow fitting **136** which is threaded into the passage. A high pressure conduit **138** extends between the elbow fitting **136** and a bored passage **140** in a discharge compensator block **142** which is stationarily bolted upon end cover **11.**

The compensator block **142** includes a cylindrical extension **144** having a spring **146** positioned therein which acts between threaded adjustment plug **148** fitted in one end of the cylinder and a movable disc **150** adjacent the other end of the cylinder. A spool valve **152** bears against the disc **150** and is positioned in a longitudinally bored passage **154** in the block. The spool valve **152** includes an enlarged head **156** which is movable from the left hand position shown in FIG. **8** to any one of several positions to the right, by the force exerted by the pressure fluid in passage **154** which is

communicated to head 156 of the spool valve by way of passage 158 which continuously communicates with passage 140.

It will be appreciated that the degree of movement of the spool valve 152 for a given pressure obtaining in passage 154 will depend upon the force opposing such movement exerted by the spring 146 and this force is adjustable by varying the setting of the adjustment plug 148.

Referring to FIGS. 7 and 8, another passage 160 is also bored in the end cover 11 and communicates continuously between passage 140 and on-stroke piston 74 as well as the bearing 52 of the tilt plate as will be explained in more detail later. In addition, passages 162 and 164 are also bored in the compensator block 142 and end cover 11 and communicate the spool valve passage 154 respectively with the off-stroke piston 72 and the casing, the latter passage 164 acting as a relief passage to relieve the off-stroke piston 72 to the pump casing if the pressure in the discharge port 18 and passage 140 drops. Suitable sealing rings 165 may be provided as necessary at the junction between block 142 and cover 11 to prevent leakage of the high pressure fluid.

Prior to describing the operation of the compensator, the trunnion bearing construction 52 will first be described since the bearing acts to a certain extent in conjunction with the control pistons.

Referring now to FIGS. 1, 9 and 9A, the tilt plate bearing 52 includes a trunnion block 170 having a concave surface 172 facing the tilt plate and an arcuate pair of end walls 174 formed at each end of the concave surface. Each of the end walls 174 is convex on its outer surface 175 and concave on the edge 176 which faces toward the tilt plate. A plurality of holes 178 extend through the thickness of the end walls 174 and the trunnion block is stationarily mounted against a step 180 in end cover 11 by screws 181 extending through the holes 178 as shown in FIG. 1.

As shown particularly in FIG. 7, a pair of bored passages 182 and 182' extend through the end cover 11 and one end of each of the passages communicates continuously with passage 160 just before the on-stroke piston 74. The other end of passages 182 and 182' terminates adjacent the back face 184 of the trunnion block 170 as shown in FIG. 1. Each of these passages 182 and 182', in turn, communicates with small bored passages 186 and 186' respectively which extend through the thickness of the trunnion block and open to the concave surface 172 of the trunnion block. Passages 182 and 182' may be slightly countersunk adjacent the back face 184 of the trunnion block so as to receive a suitable sealing ring 187 which is pressed between the back face of the trunnion block and the end cover 11 when the trunnion block has been positioned in the latter on step 180.

A pair of arcuate end pieces 188 are also positioned at each side of the trunnion block 170. Several holes 189 extend through the width of the end pieces to accommodate screws (not shown) for stationarily attaching the end pieces also to the end cover 11. The concave inner surface 190 of each of the end pieces is of a curvature to fit snuggly against surfaces 175 of the trunnion block and are formed with an arcuate groove 192 for receiving the trunnions of the tilt plate.

An arcuate insert 194, having a length substantially equal to the distance between the end walls 174 of the trunnion block, is adapted to be received against surface 172. As shown in FIG. 9A, the side of the insert facing the trunnion block 170, is suitably grooved at 195 and the grooves register with passages 186 and 186'.

The back surface of the tilt plate 50 also includes a convex surface 196 which is adapted to mate with he concave face 197 of the insert 194. A pair of arcuate trunnions 198 extend from each end of the convex surface 196 and are adapted to be movably inserted in the arcuate grooves 192 in the end pieces 188 to provide for tilting of the tilt plate. The trunnion block 170, insert 194 and tilt plate 50 are suitably apertured at 199 to allow for passage therethrough of the drive shaft 36 as shown in FIG. 1.

In the assembled tilt plate bearing, the convex surface 196 of the tilt plate and the concave face 197 of the insert 194 are positioned together and an elongated key (not shown) is inserted from the side into the mating slots 200 in each surface to prevent movement of the insert and tilt plate relative to each other about the radius of curvature of their surfaces 196 and 197. The insert and tilt plate are then positioned in the trunnion block 170 so as to fit between the end walls 174 of the trunnion block. The end pieces 188 are positioned in place with the trunnions 198 extending into their arcuate grooves 192.

During operation of the pump, grooved face 195 of the insert 194 which is adjacent the concave surface 172 of the trunnion block, will be pressurized at all times by fluid at discharge pressure by way of passages 160, 182, 182' and 186, 186'. Thus, a prtssurized hydraulic fluid cushion is provided between these surfaces and a force is exerted on the insert and tilt plate to counterbalance the force exerted in the opposite direction on the tilt plate by the pistons 48. This fluid cushion not only relieves stresses on the tilt plate and bearing assembly, but also acts to insulate the pump against the transmission of sound and thereby effects a substantial reduction in pump noise level during operation.

The control pistons 72, 74 and 76,as well as the trunnion bearing 52, are sized such that their combined forces which act against the left side of the tilt plate are approximately equal to the forces exerted upon the other side of the tilt plate by the pistons 48 during operation of the pump. Thus, these latter forces which occur in all axial pistons pumps are effectively counterbalanced.

The operation of the above described system is as follows. The adjustment plug 148 is screwed into the cylinder 144 by a predetermined amount and thereby sets a predetermined force on the spring 146 depending upon the discharge pressure desired at port 18 of the pump. The pump is then turned on. Upon starting, the hydraulic pressure needed to operate the on-stroke and off-stroke pistons 74 and 72 is not yet available. However, the mechanical spring urged starting piston 76 will tilt the tilt plate to its full tilt-full stroke position. As pressure builds up in the discharge port 18, some of the fluid at discharge pressure will pass through passage 134, fitting 136, conduit 138, and passage 140 and will flow through the continuously open passage 160 to the on-stroke piston 74 and tilt plate trunnion bearing. Thereby the on-stroke piston 74 will now take over primary control of the tilt plate from the starting piston 76.

Fluid at discharge pressure will also be ported from passage **140**, through passage **158** to passage **154** and will exert a force on the left side of the enlarged head **156** of spool valve **152**. As the pressure builds up, the fluid urges the spool valve to the right, as viewed in FIG. **8** against the force of spring **146**. When the discharge pressure reaches the desired magnitude, the spool valve head **156** will move past the left edge of passage **162** to allow pressurized fluid to flow to the off-stroke piston **72**, actuating the piston and removing some of the tilt from the tilt plate. Thereby, the stroke of the pistons **48** will be slightly decreased to decrease the flow in port **18**. The spool valve will thus move back and forth to maintain the desired discharge pressure of the pump as set by the adjustment plug **148**.

If for some reason the discharge pressure in port **18** exceeds a maximum limit, the off-stroke piston **72** is sized so as to exert a force on the top of the tilt plate which is sufficient to cause the trunnion insert **194** to separate from its block **170** by an amount sufficient to relieve some of the pressure to the pump casing from passages **182** through the bearing. Referring to FIG. **1**, suitable fittings **166** are preferably provided for receiving a conduit which communicates with the supply or other reservoir of hydraulic fluid for passage of excess fluid from the housing.

Conversely, if the pressure in port **18** drops, the spool valve **152** will move to the left to the position shown in FIG. **8**. In this position, passages **162** and **164** are communicated with each other and the fluid in the off-stroke piston **72** is relieved to the pump casing by way of passages **162, 154** and **164**, respectively to cause further tilt of the tilt plate and increase the flow through port **18**.

It will be noted that the rotor of the above described pump, may be completely surrounded by a fluid film so as to sound insulate this rotating member to substantially reduce the operating noise level of the pump. Such fluid film is dynamically formed at the bearing **35**. A film is also formed between surfaces **32** and **34** and between the trunnion block **170** and insert **194**.

It will be understood that, although the various features of the fluid pressure device have been described with reference to a hydraulic pump, the features of the invention will be equally applicable to hydraulic motors. It will also be understood that the embodiment of the invention which has been described is merely illustrative of a few of the applications of the principles of the invention. Numerous modifications may be made by those skilled in the art without departing from the true spirit and scope of the invention.

What is claimed is:

1. In a fluid pressure device having high and low pressure ports,

a rotor in which pistons are reciprocal therein,

a tilt plate mounted for pivoting on bearing means whereby the stroke of the pistons may be varied,

means communicating said bearing means with a sorce of fluid under pressure for exerting a force on said tilt plate adjacent said bearing means in a direction opposite to the force exerted by said pistons on said tilt plate, and

meanS for relieving fluid from said high pressure port through said bearing means when the pressure of said last mentioned fluid exceeds a predetermined amount.

2. In the device of claim **1** wherein said communicating means communicates said bearing means with the high pressure port of the device.

3. In the device of claim **1** wherein said bearing means is positioned on the side of said tilt plate opposite said pistons.

4. In the device of claim **1** wherein said bearing means comprises a first block means stationarily mounted in the device and having an arcuate surface facing said tilt plate, a member positioned on and movable with said tilt plate and having an arcuate surface complimenting the arcuate surface of said block means, and passage means communicating the high pressure port with the arcuate surface of said block means, whereby fluid at high pressur is introduced between said arcuate surfaces and exerts a force on the arcuate surface of said member toward said tilt plate.

5. In the device of claim **4**, wherein said arcuate surfaces are positioned on the side of said tilt plate opposite said pistons and said force is exerted on said opposite side toward said pistons.

6. In the device of claim **4**, wherein said bearing means includes arcuate insert means between said arcuate surfaces, said insert means having a grooved surface facing the arcuate surface of said block means, and said passage means extends through said block means to introduce said high pressure fluid between the last mentioned surfaces.

7. In the device of claim **4**, wherein said passage means continuously introduce said high pressure fluid to said bearing means while said device is in operation.

8. In the device of claim **1** wherein said means for relieving the fluid in said high pressure port includes control piston means for tilting said tilt plate and exerting a force on said tilt plate in the same direction as the force exerted on said tilt plate by said bearing means, the force exerted by said control piston means separating said bearing means to relieve said fluid in said high pressure port.

9. In a fluid pressure device having high and low pressure fluid ports,

a rotor mounted for rotation,

a plurality of cylinders in the rotor alternately communicating with said ports,

a plurality of pistons reciprocally movable in said cylinders,

a tilt plate one side of which is drivingly associated with an end of said pistons, said tilt plate being mounted for rotation about a pivot axis adjacent one end of the rotor whereby the stroke of the pistons may be varied,

control means for selectively exerting a force on the tilt plate to pivot the tilt plate about is pivot axis, said control means exerting a first force on said tilt plate in a direction opposite the force exerted on said tilt plate by said pisotns,

bearing means pivotally mounting said tilt plate for said rotation, and

means communicating said bearing means with said high pressure port for exerting a second force on said tilt plate in a direction opposite to the force exerted by said pistons on said tilt plate, said first and second forces together counterbalancing the forces exerted by said pistons.

10. In the device of claim **9** wherein said control means relieves excessive pressures in said high pressure port through said bearing means.

11. In the device of claim 9 wherein said control means is hydraulically operated by the fluid in the high pressure port of said device.

12. In the device of claim 9 wherein said control means comprises first and second hydraulic pistons, said first piston being positioned to exert a force on said tilt plate which increases its tilt and said second piston being positioned to exert a force on said tilt plate which decreses the tilt.

13. In the device of claim 12 wherein said control means include means which is operable independently of the pressure of the hydraulic fluid of said device to exert a force on the tilt plate to fully tilt said plate.

14. In the device of claim 13 wherein said last mentioned means comprises a spring urged piston.

15. In the device of claim 12 including means to connect said pistons with the high pressure port of said device.

16. In the device of claim 12 including compensator means for selectively varying said first and second pistons, said compensator means continuously communicating a source of pressure fluid with said first piston and selectively communicating a source of pressure fluid with said second piston.

17. In the device of claim 16 wherein said sources of pressure fluid are the high pressure port of the device.

18. In the device of claim 16 wherein said compensator means includes a spool valve for selectively communicating said source with said second piston.

19. In the device of claim 16 wherein said compensator means includes relief means for relieving said second piston when the pressure in said high pressure port decreases.

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