

Nov. 20, 1962

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3,064,583

VARIABLE DISPLACEMENT PUMP

Filed March 29, 1957

3 Sheets-Sheet 1

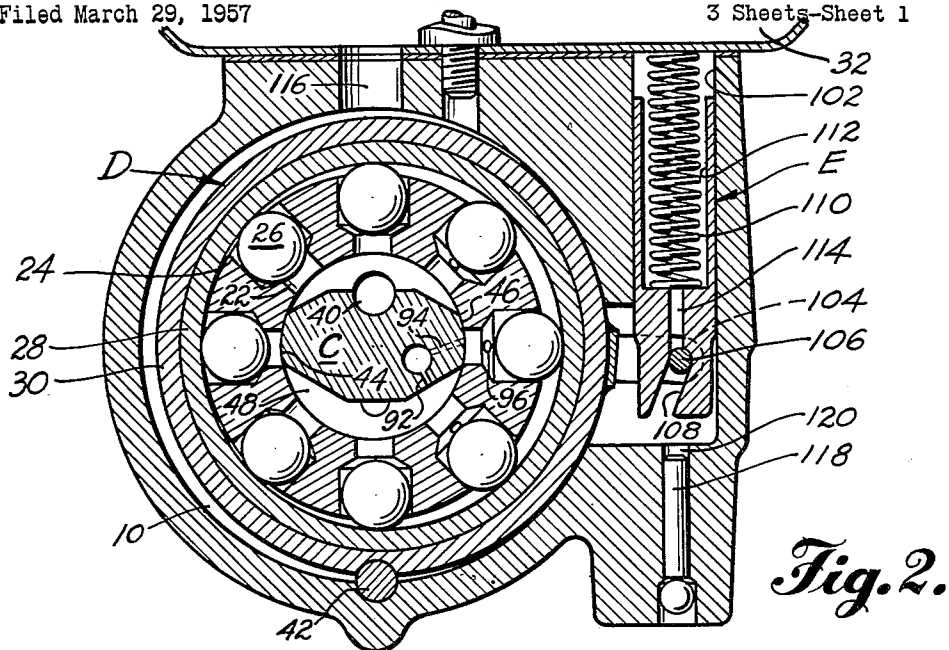


Fig. 2.

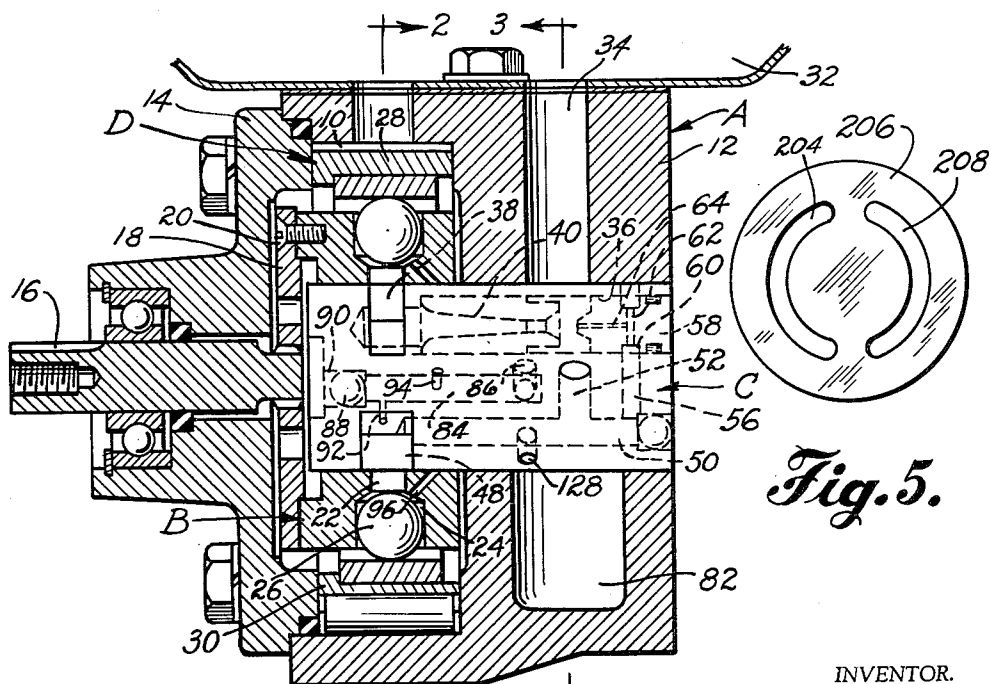


Fig. 5.

Fig. 1.

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3 Sheets-Sheet 2

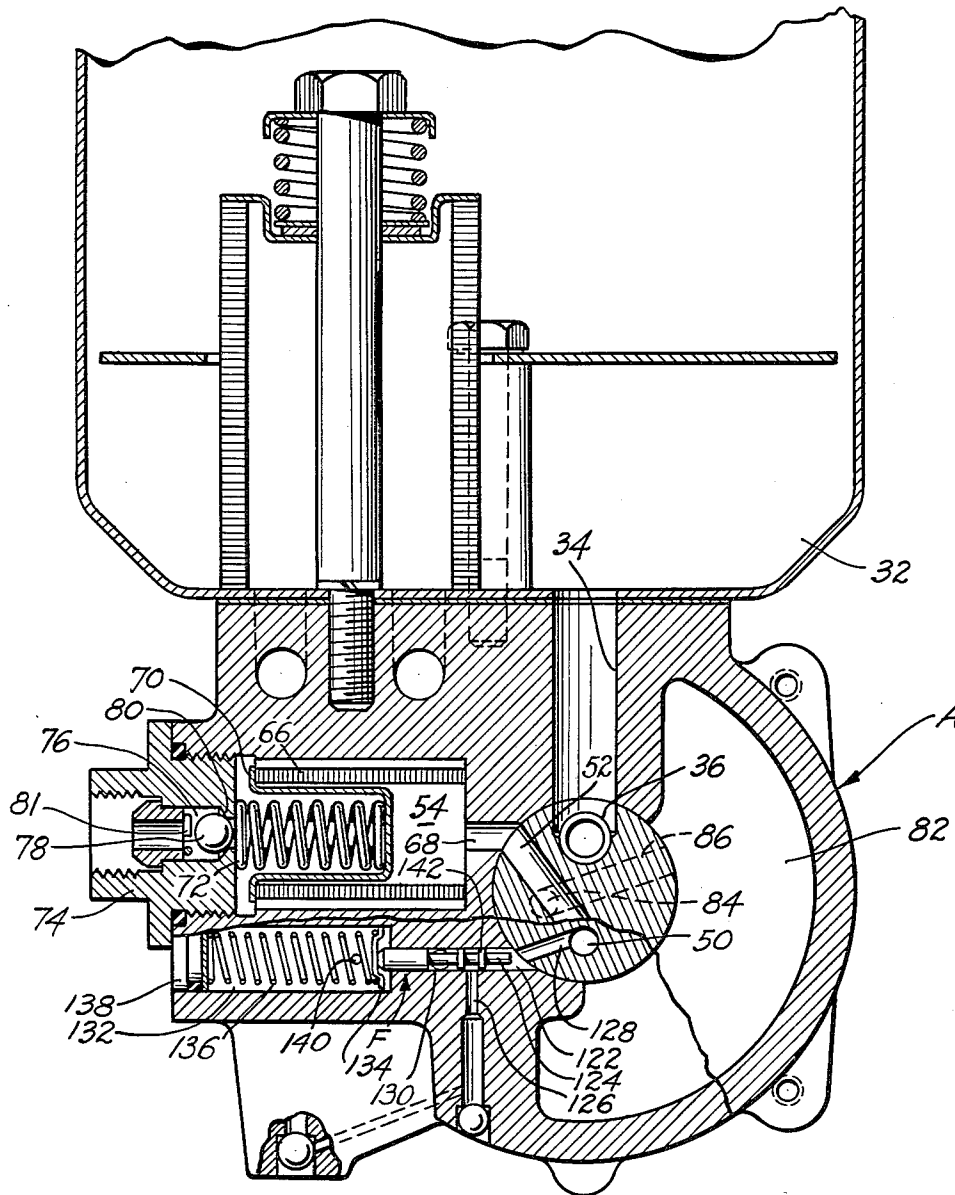


Fig. 3.

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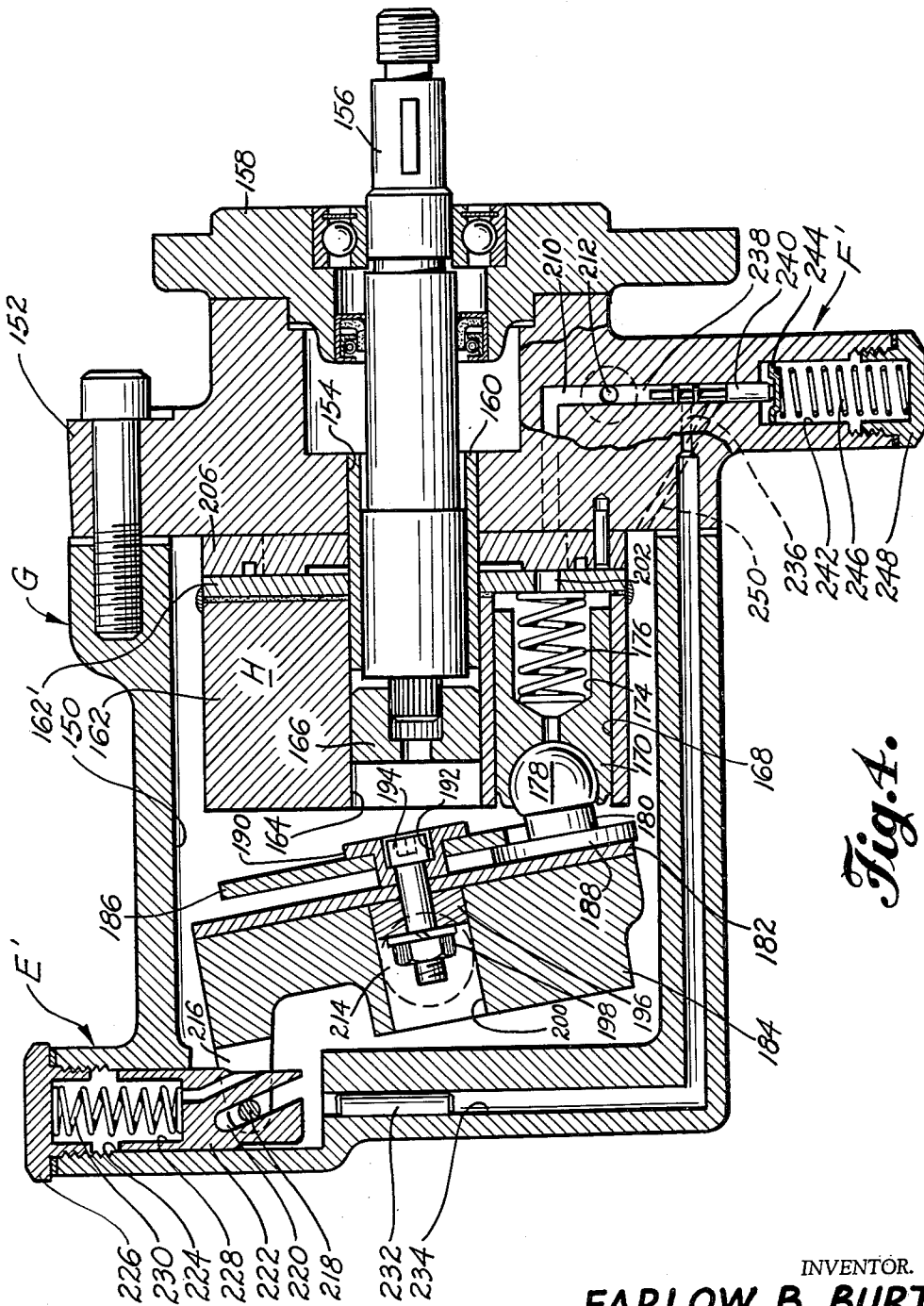
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3 Sheets-Sheet 3



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1

3,064,583

VARIABLE DISPLACEMENT PUMP

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7 Claims. (Cl. 103—161)

The present invention relates to positive displacement pumps having a movable stroke determining member whose position can be changed to vary the pump displacement, and more particularly to means for supporting and positioning said movable member.

Positive displacement pumps of the type whose displacement can be varied usually employ a rotor having at least one fluid pressure chamber therein, and cooperating camming member which produces a reciprocating movement of the fluid displacement member in each fluid pressure chamber during the rotation of the rotor. The displacement of the pump is determined by the relative positioning of the rotor and camming surface; and is usually varied by the positioning of the camming member in accordance with either the discharge pressure of the pump, or the differential pressure across a flow control orifice. In most pumps of the above type, the reaction forces exerted against the camming member vary in direction and amount depending upon the angular position of the rotor during its rotation to produce a fluctuating movement of the camming member. This fluctuating movement of the camming member in turn produces a corresponding variation in the pump displacement, and troublesome noise and vibration in the pump and the system to which it is connected results.

It is an object of the present invention to provide a new and improved positive displacement pump whose output is varied in accordance with discharge conditions, and which is so constructed and arranged that changing reaction forces exerted upon its displacement controlling member will not be reflected in its positioning of the displacement controlling member.

A further object of the invention is the provision of a new and improved pump of the above described type having control means which is easily adjusted by control forces but which will not be moved by the reaction forces exerted upon it by the camming member.

A still further object of the invention is the provision of a new and improved pump of the above described type whose camming member is movable in a direction which changes the pump displacement, said camming member being controlled by a slide which is reciprocable generally at right angles to the camming members displacement changing motion, and being connected to the slide by means acting on a line of force which never overcomes the co-efficient of friction between the slide and its supporting member to effect a shifting of the slide. The camming member and rotor preferably being so constructed and arranged that the forces exerted upon the camming member decrease substantially to zero at some period during each revolution of the rotor, and at which time substantially no side loading is exerted on said slide so that the slide can be accurately positioned with very little force exerted upon it by its control member.

A more particular object of the invention is the provision of a new and improved radial piston pump comprising a rotor surrounding a pintle suitably ported to provide suction and discharge for its cylinders at opposite sides of the pintle, a camming member surrounding the rotor and pivoted generally on the centerline of the arc through which the cylinders discharge, and whereby arcuate movement about the pivotal support changes the displacement of the pump—said displacement changing motion being restrained by a slide extending at generally right angles to the camming member's arcuate movement,

2

and which slide is connected to the camming member by means acting on a line of force which never overcomes the coefficient of friction between the slide and its supporting member.

The invention resides in certain constructions and combination and arrangements of parts, and further objects and advantages will become apparent to those skilled in the art to which the invention relates from the following description of several preferred embodiments described with reference to the accompanying drawings forming a part of this specification, and in which:

FIGURE 1 is a cross-sectional view of a radial piston pump embodying principles of the present invention;

FIGURE 2 is a cross-section taken on the line 2—2 of FIGURE 1;

FIGURE 3 is a cross-sectional view taken on the line 3—3 of FIGURE 1;

FIGURE 4 is a cross-sectional view (having a portion broken away to better show details beneath the plane of the section) of an axial piston pump embodying principles of the present invention; and

FIGURE 5 is an end view of a porting plate used in the pump shown in FIGURE 4.

The radial piston pump shown in FIGURE 1 generally comprises a body member A having an internal chamber 10 therein in which an annular rotor B is journaled about an axially extending pintle C which projects into the internal chamber 10 from one end wall 12 of the pump. The opposite end wall 14 of the pump is made in the form of a removable cover member suitably bolted in place; and the annular rotor B is adapted to be rotated about the pintle by means of a drive shaft 16 journaled in the cover member 14. The inner end of the drive shaft 16 is splined to a drive plate 18 positioned over the adjacent end of the pintle and the outer edges of which are fastened to the annular rotor B by means of a plurality of machine screws 20, only one of which is shown.

The annular rotor member B is provided with a plurality of radially extending openings 22, therethrough, the radially outer ends of which are accurately counterbored to form cylinders 24 in which individual ball pistons 26 are positioned. The ball pistons 26 are retained within the cylinder by means of an annular camming member D which extends around the outer surfaces of the rotor and on which the balls 26 are adapted to roll. The annular camming member D shown in the drawing utilizes the race 28 of a commercially obtained anti-friction bearing, pressed into a support member 30 which is suitably guided and supported for eccentric movement with respect to the rotor B.

The pump shown in the drawing is adapted to be supplied with oil from a reservoir 32 which is bolted directly to the top surface of the pump. Oil from the reservoir passes through a vertical opening 34 in the body member A to a longitudinally extending drilling 36 in the axially extending pintle C. The top surface of the pintle C directly beneath the annular rotor B is notched out as at 38 to provide inlet communication between the inner end of the cylinders 24 and the inner end of the longitudinally extending drilling 36; and a venturi section 40 is pressed into the longitudinal drilling 36 between the inlet passage 34 and the notch 38 supplying the rotor. The particular embodiment shown in the drawing utilizes a pressurized suction wherein fluid from the inlet passage 34 is forced into the throat of the venturi section 40 to the cylinders 24 by means of a high pressure impinging stream presently to be described.

The radial piston pump shown in the drawings is adapted to be driven clockwise as seen in FIGURE 2. The annular camming member D is supported for eccentric movement with respect to the annular rotor B by means of an abutment pin 42 recessed into the lower end

3

of both the body member A and the support member 30 in such manner as to limit all but a rocking motion of the cam member D with respect to the rotor B. Maximum displacement for the pump will be provided when the camming member D is in the position shown in FIGURE 2 of the drawing. With the camming member D in the position shown, the ball pistons 26 will be in their innermost position with respect to their cooperating cylinders 24 when the inner end of the cylinders 24 are moved out of engagement with the land portion 44 of the pintle into communication with the inlet groove 38 on the upper surface of the pintle. The positioning of the camming member D is such that centrifugal force moves the ball pistons 26 radially outwardly in their cooperating cylinders 24 as the balls roll around the race 28 to a position approximately 180° from the start of the inlet stroke. As the ball pistons approach their outward limit of travel, the inner end of the cylinders 24 move out of communication with the inlet groove 38 to a position wherein a land 46 valves off or completely isolates the cylinders 24 from both suction and discharge.

Continued rotation of the rotor during the second half of each revolution causes the ball pistons 26 to roll around the lower half of the race 28 thereby causing the pistons 26 to be moved inwardly to their most inwardly or starting position. Just after the time that the ball pistons 26 start to move inwardly, the inner ends of the cylinders 24 move off the land 46 into communication with a discharge groove 48 in the lower surface of the pintle C. The discharge groove 48 is quite similar to the inlet groove 38—extending over a similar arc of the pintle, but is separated from the inlet groove 38 by the land portions 44 and 46. Fluid forced into the discharge groove 48 by the inward movement of the ball pistons 26 passes through a longitudinal discharge drilling 50 in the pintle C to a transverse drilling 52 leading to a discharge chamber 54 in the body member A. Some discharge fluid is used for the pressurizing of the pump's inlet stream by means of a transverse drilling 56 communicating the longitudinal discharge drilling 50 with a nozzle 58 in the inlet drilling 36. The passage 56 opens into an annular groove 60 in the nozzle 58; and a transverse drilling 62, between opposite sides of the recess 60, communicates with a small longitudinal drilling 64 which directs the high pressure stream into the throat of the venturi section 40. Fluid discharged from the pump passes through an annular filter 66 held into engagement with the bottom end of the discharge chamber 54 surrounding its inlet 68 by means of a retainer 70 and a coil spring 72. The coil spring 72 is in turn held in place by a threaded outlet fitting 74 screwed into the outer end of the discharge chamber 54. The fitting 74 is provided with a centrally located discharge opening 76 therethrough containing a check valve 78 and cooperating valve seat 80 for the prevention of return flow through the pump. A bent wire 81 is inserted between the ball 78 and a tube fitting in the opening 76 to prevent the ball from restricting flow out of the discharge connection.

The pump shown in the drawing is adapted to provide discharge pressures up to approximately 2500 pounds per square inch—at which pressures sufficient compressibility is encountered in the oil circulated through the pump to create a troublesome noise and vibration problem. In order to help alleviate difficulties created by the compressibility of the fluid being pumped, the embodiment shown in the drawing is provided with an accumulator chamber 82 adapted to hold an isolated supply of hydraulic fluid at a pressure slightly below the pump discharge pressure. The accumulator chamber 82 is filled with pressure fluid from the pump's discharge by means of a bleed passage adapted to produce a small and substantially continuous flow to the accumulator chamber. A longitudinal drilling 84 for this purpose is provided in the pintle—the inner end of which drilling is intersected by

4

means of a transverse drilling 86 opening into the reservoir 82, and the outer end of which drilling is closed off by means of a ball 88 pressed into a counterbore 90 in the outer end of the drilling. A small transverse drilling 92 of predetermined size communicates the longitudinal drilling 84 and the discharge groove 48 of the pintle to limit the rate at which pressure flow is supplied the accumulator from the discharge of the pump.

Fluid pressure from the accumulator 82 is used to pressurize each of the cylinders 24 when the ball pistons 26 therein have reached their outermost positions, and during the time that each cylinder is valved off from both the inlet groove 38 and discharge groove 48 by the land 46. Fluid pressure from the accumulator 82 is bled to each cylinder at this instant in a controlled amount by means of a small transverse drilling 94 which communicates the accumulator passage 84 with the surface of the pintle C adjacent one side of the land 46. A plurality of cooperating drillings 96 (one for each of the cylinders 24) are provided in the annular rotor B and are positioned in such a way as to register with the opening 94 during the time that each cylinder 24 is valved off from both the suction and discharge grooves 38 and 48. During the times that the drillings 94 and 96 are in register, fluid pressure from the accumulator 82 is communicated to the cylinders 24. The grooves 94 and 96 are further positioned such that the groove 96 moves out of register with the opening 94 just prior to the time that the cylinders 24 move into communication with the discharge groove 48. By means of this valving process, each of the cylinders 24 are rapidly pressurized to a pressure approximately equal to the pump discharge pressure prior to the time that the individual cylinders are communicated to the pump discharge system and immediately following which the cylinders are opened to pump discharge before leakage has had a chance to appreciably reduce their pressure. It will be seen that this filling of the cylinders with pressure fluid prior to valving to discharge is accomplished from a pressure system substantially isolated from the pump discharge system—the only connection being the small transverse drilling or filling orifice 92 which for all practical purposes prevents fluctuations in accumulator pressure from being transmitted to the pump's discharge system.

The displacement of the pump shown in the drawing is adapted to be controlled in accordance with the demand of the system to which it is connected. The pump will maintain a more or less constant discharge pressure within certain limits; and as the demand of the system for more fluid increases (as sensed by a slight drop in discharge pressure), the camming member D is rotated to increase the pump displacement until the pump again achieves its predetermined set discharge pressure. Conversely as the system's demand for pressure fluid falls off, a slight increase in pressure is sensed by the pump causing its camming member D to be shifted in a direction decreasing the displacement of the pump until the amount delivered equals the system's demand at the predetermined set pressure of the pump.

The reaction forces exerted upon the camming member B by the pistons 26 varies in accordance with the number and the positioning of the pistons, and the manner in which the cylinders are valved to the inlet and discharge pressures. These reaction forces tend to produce a fluctuating movement of the camming member, which if not firmly resisted, would produce a rapid fluctuation in the displacement of the pump to produce troublesome noise and vibration in the pump and the system to which it is connected. According to the principles of the present invention, supporting and regulating means are provided for the camming member which prevents the previously referred to fluctuating forces from being transmitted back against the pump's control system in a manner affecting the positioning of the camming member by the pump's control system. The embodiment about

to be described is constructed such that the fluctuating forces upon the camming member D will at some time during each revolution of the rotor substantially balance out, such that no force is then exerted upon the pump's regulating or control system. Control movement may take place during these intervals of rotor movement to produce a precise positioning of the camming member D.

Although the annular camming member D may be otherwise supported relative to the body member A of the pump, the preferred embodiment will utilize an abutment positioned approximately on the center line of the discharge cycle of the pump such that the line of force of the discharge pressure upon the rotor passes through the abutment. Displacement changing movement of the camming member D takes place in a direction substantially at right angles to this line of force and is accomplished in the preferred embodiment by arcuate movement of the camming member about the abutment 42.

According to the principles of the present invention, the displacement changing movement of the camming member D is opposed by a slide reciprocable in a direction substantially at right angles with the displacement changing movement of the camming member. A suitable connection is provided between the camming member and the slide to control the movement of the camming member. In the preferred embodiment, the connecting mechanism between the camming member and slide will act upon a line of force extending at an angle relative to the sliding motion of the slide which is greater than the coefficient of friction between the slide and its supporting member under well lubricated conditions. The fluctuating forces produced upon the camming member during the rotor pumping action under such an arrangement only serves to force the slide into firmer engagement with the supporting member, and will not move the slide to change the fluid displacement of the pump.

In the preferred embodiment the slide E is formed from a cylindrically shaped member positioned in a vertical drilling 102 in the body member A. A U-shaped bracket 104 is welded to the adjacent end of the camming member D in such a position as to straddle the opposite sides of the slide E. A pin 106 is positioned across the outer ends of the U-shaped bracket; and the pin 106 is received in a milled slot 108 extending at a slight angle relative to the displacement changing movement of the camming member. Reciprocation of the slide E therefore produces movement of the pin 106 at substantially right angles to the movement of the slide E causing the camming member D to pivot about the abutment pin 42. Camming member D is normally biased into its largest pump displacement producing position by means of a coil spring 110 positioned between the bottom of the reservoir 32 and the lower end of the vertical drilling in the slide E. The volume enclosed behind slide E is relieved to the internal chamber 10 of the pump by means of a drilling 114 communicating the vertical drilling 112 and the upper end of the slot 108, and the chamber 110 is in turn communicated with the reservoir 32 by means of the drilling 116 in the upper end of the body member A. An adequate supply of lubricating fluid is therefore assured all moving parts of the pump; and any high pressure leakage is adequately relieved to the system's reservoir.

Shifting movement of the slide E in the embodiment shown in the drawing is accomplished by the utilization of a slight change in pressure of the discharge pressure of the pump. Inasmuch as the fluctuating forces produced upon the camming member during operation of the pump are in the order of from 50 to 100 pounds; the design of the pump is such that these forces pass through a null during each revolution of the rotor and the construction of the slide and cam shifting structure is such that the fluctuating forces do not tend to shift the slide, very little force is required for moving the slide. A hydraulic piston 118 is positioned in a bore 120 in the body member A directly

beneath the lower end of the slide member E. Fluid pressure supplied to the lower end of the hydraulic piston 118 forces it up into engagement with the lower end of the slide E to oppose the coil spring 110 and effect a shifting of the slide E. Inasmuch as the forces required to shift the slide are quite small, a control valve F is utilized to regulate the amount of the pump discharge pressure which is supplied to the hydraulic piston 118. The structure shown comprises a bore 122 having a spool valve 124 therein, the annular flanges or lands of which normally straddle a control port 126 which is communicated to the bottom side of the hydraulic piston 118 by suitable drilled passageways. The inner end of the bore 122 of the control valve is communicated with the discharge drilling 50 in the pump pintle C by a drilling 128, and the outer end of the bore 122 of the control valve is communicated with the pump chamber 10 by means of an opening 130 in the body member A. The outer end of the spool valve 124 projects into a spring chamber 132 where it is abutted by a spring retaining plate 134 which is biased inwardly by a coil spring 136. The outer end of the spring chamber 132 is closed off by a suitable sealing member 138 held in place by the threaded outlet fitting 74; and the spring chamber 132 is also vented to the internal chamber 10 by means of a drilling 140 in the body member.

Operation of the pump should be readily discernible by those skilled in the art from the above description reciting the cooperation between the various pump elements. Suffice it to say that hydraulic fluid from the reservoir 32 passes through the inlet passageway 34 to the venturi section 40 where the impingement of a high pressure stream through the longitudinal drilling 64 into the throat of the venturi section produces a positive pressure in the inlet groove 38 of the pintle C. Rotation of the rotor B successively communicates the cylinders 24 with the inlet groove 38 during the portion of the rotor cycle wherein the ball pistons 26, which are in rolling contact with the camming member D, move radially outwardly in their cylinders. Outward movement of the ball pistons 26 causes a quantity of fluid to be added to each cylinder while the cylinders are communicated to the inlet groove 38; and at approximately the time that the ball pistons 26 have reached their outer limit of travel, the inner openings of the cylinders 24 slide over the land portion 46 of the pintle to isolate the cylinders from communication with both the inlet and outlet systems of the pump. Shortly after the cylinders 24 become valved off from the inlet groove 38, and prior to the time that the cylinders are communicated with discharge groove 48 of the pintle, each cylinder is pressurized with fluid from the accumulator 82 by the rotation of each cylinder's pressurizing groove 96 into communication with the pressurizing groove 94 of the pintle. Each cylinder is thereby rapidly brought up to a pressure approximating that of the pump discharge pressure; and immediately thereafter each chamber is successively valved off from the accumulator, and then communicated with the discharge groove 48 of the pintle. Continued rotation of the rotor with respect to the camming member D causes the ball pistons 26 to move inwardly in their cylinders 24 discharging the fluid into the discharge groove 48, through passageways 50 and 52 in the pintle to the discharge chamber 54. At the same time a small side stream is supplied to the suction pressurizing nozzle 58 through the transverse drilling 56; and a second side or auxiliary stream of high pressure fluid is supplied to the accumulator 82 through the small filling orifice 92 extending between the discharge groove 48 and the accumulator passage 84 of the pintle. It should be stated that the accumulator 82 is sized sufficiently large to control its pressure drop within limits each time a drilling 96 is communicated with the drilling 94; and the passageway 92 is sized sufficiently large to maintain the reservoir 82 at a pressure approximately equal to that of the pump discharge, while at the same time preventing

the pressure surges experienced within the accumulator 82 from reaching the discharge system of the pump.

It has previously been explained that the positioning of the abutment pin 42 is such that the line of force of the pressure forces upon the rotor pass substantially through the center of the pin 42. Continuously changing forces produced upon the camming member as the individual cylinders are valved to suction and discharge, produce fluctuating components tending to alternately rock the camming member to opposite sides of the abutment pin 42. Fluctuating movement of the camming member D is restrained by the slide structure E which is rigidly supported against movement in this direction, but which is positionable at substantially right angles to the fluctuating forces applied to the camming member. The slot 108 which receives the pin 106 attached to the camming member D is preferably formed at a shallow enough angle with respect to the direction of fluctuating movement of the camming member such that the component of these forces in the direction of movement of the slide will not overcome the coefficient of friction between the slide and its receiving groove 102. The fluctuating forces on the camming member D therefore produce a locking action of the slide E in its receiving groove which prevents these forces from being transmitted back into the control system adapted to position the slide E.

The slide E is positioned by means of the fluid pressure piston 118 which receives its actuating pressure from the discharge of the pump through a control valve F. As pressure is admitted to the inner end of the spool valve 124, its force on the spool valve is exerted against the coil spring 136. Upon yielding of the coil spring 136 (which will occur at a predetermined set pressure) the spool valve's inner land 142 will be moved sufficiently to communicate the pump discharge pressure in the inner end of the bore 122 to the control passage leading to the lower end of the hydraulic piston 118. Should the adjustment of the camming member D be such as to provide a greater amount of pressure fluid than is being used by the system to which the pump is connected, the back pressure exerted by the system will be reflected in an increased discharge pressure of the pump—resulting in the opening of the control passage to the pump discharge pressure as previously explained. This pressure against the lower end of the hydraulic piston 118 forces it upwardly to engagement with the lower end of the slide E to exert an additional amount of force upon the slide in opposition to the coil spring 110. As previously indicated, the fluctuating forces on the camming member D alternately force the pin 106 in opposite directions during each revolution of the rotor to produce nulls during which no rocking forces are being exerted upon the slide. The increased force exerted upon the hydraulic piston 118 as a result in the rise in pressure in the pump discharge is therefore free to move the slide E during these null conditions without being opposed by the fluctuating forces produced upon the camming member during the pump operation. Upward movement of the slide E produces a rocking action of the camming member D about the abutment pin 42 in a direction decreasing the pump displacement, and will continue to do so until the amount of fluid being delivered by the pump just balances the consumption of the system to which it is connected at a pressure corresponding to the predetermined set pressure of the pump as controlled by the biasing action of the coil spring 136.

As second embodiment of the invention employing a slide and control structure quite similar to that just described for the first embodiment is shown in FIGURE 4 of the drawings. The embodiment shown in FIGURE 4 is what is known as an axial piston pump. The pump shown in the drawings generally comprises a body member G having an internal chamber 150 therein which is closed off by a removable end plate 152 suitably bolted into position. An axially extending opening 154 is pro-

vided in the end plate 152 for the reception of a drive shaft 156. The outer end of the drive shaft 156 is journalled by anti-friction means in a bearing plate 158, suitably bolted to the end plate 152, and the inner end of the drive shaft 156 is journalled in a sleeve 160 pressed into the inner end of the axially extending opening 154 and projecting into the pump internal chamber 150. A rotor member H comprising a generally cylindrical shaped body section 162 having an end closure plate 162' brazed thereto is journalled about the inner end of the sleeve 160. The sleeve 160 is received into an axially extending opening 164 in the body and closure members and a splined bushing 166 is pressed into the opening to provide a driving connection with the inner end of the shaft 156.

The rotor member H is provided with a plurality of axially extending cylinders 168 uniformly spaced about its axis of rotation, and only one of which is shown in the drawings—each having a cooperating piston 170 therein. The inner end of each piston 170 is bored out, as at 174, to receive a coil spring 176 which is positioned between the end closure plate 162' and the bottom of the bore to bias the pistons outwardly with respect to their cylinders 168. The outer end of each piston 170 is spherically shaped to receive the ball end 178 of a slipper shoe 180, the other end of which bears against a bearing plate 182 suitably retained on the inner face of a tiltable swash plate 184. The slipper shoes 180 will normally be biased against the bearing plate 182 by pressure forces during the pumping operation of the unit; and are additionally held adjacent the bearing plate 182 by an annular spider plate 186 which is suitably recessed around its periphery to extend over a portion of each shoe's flange 188. The spider plate 186 is retained adjacent the bearing plate 182 by means of a spacer member 190. A suitable counterbored opening 192 is provided in the spacer member 190 to receive the headed end of a bolt 194, the other end of which extends through a bushing 196 is locked in place by a nut 198. The bushing 196 is positioned in an opening 200 extending through the swash plate 184 behind the bearing plate 182, and is held in position by the inside face of the bearing plate 182.

The rotation of the rotor member H about the sleeve 160 causes the slipper shoes to slide around the periphery of the bearing plate 182 to produce a reciprocation of the pistons 170 in their cylinders 168. The pump shown in the drawing is adapted to be rotated in a clockwise direction as viewed from the shaft end of the pump; and as such the pistons 170 will move outwardly with respect to their cylinders 168, as the piston moves from the position shown in the drawings to a position adjacent the upper end of FIGURE 4. During this outward movement of the pistons 170, fluid is introduced into the cylinders 168 through an inlet opening 202 in the rotor closure plate 162' and an arcuately shaped inlet port 204 in a porting plate 206 positioned between the outer end of the rotor member H and the removable end plate 152 of the pump. The arcuately shaped inlet port 204 communicates with suitable drilled passageways in the body portion of the pump—which inlet passageways are not shown in the drawings inasmuch as they are positioned in the portion of the pump lying above the plane of the paper.

Clockwise rotation of the cylinders 168, during the second half of each rotor revolution, carries the pistons 170 from a position adjacent the upper end of the swash plate 184 to the lower position shown in the drawings. The pistons 170 during this half of the cycle are biased inwardly by the sliding action of the shoes 180 on the bearing plate 182. Inward movement of the pistons 170 forces fluid out of the cylinders 168 through the openings 202 in the rotor closure plate 162' to an arcuately shaped discharge port 208 in the other half of the porting plate 206. Pressure fluid from the pressure port 208

passes through a discharge passageway 210 in the front cover plate 152 to a discharge connection 212.

The swash plate 184 is preferably pivoted about a point positioned on the center line of the pump such that the forces exerted upon the swash plate will produce a fluctuating movement tending to alternately rotate the swash plate in opposite directions to produce null periods when substantially no tilting force is exerted thereon. The structure E' used in this embodiment for controlling the camming member of the previous embodiment. The swash plate 184 is journaled about opposite pin sections fixed in the side walls of the body member, and is provided with a bifurcated portion 216 carrying a pin 218 which is received in a groove 220 milled into the slide 222. The slide 222 is generally cylindrically shaped and is received in a bore 224 extending at right angles to the axis of the pump, and the upper end of which bore is closed off by means of a threaded closure member 226. The upper end of the slide 222 is bored out in similar fashion to that of the previous embodiment, and a coil spring 230 is positioned between the closure member 226 and the bottom of the bore 228 to bias the slide downwardly towards the center line of the pump. The groove 220 is inclined angularly with respect to the center line of the pump such that reciprocatory movement of the slide 222 will tilt or rock the swash plate 184 about its pin sections 214 from an inclined position producing a maximum stroke of the pistons 170 to a generally parallel relationship with respect to the rotor 162 wherein substantially no stroke of the pistons 170 is produced.

The positioning of the slide 222 in this embodiment is accomplished by structure similar to that of the previous embodiment. Inasmuch as very little force is required to move the slide 222, and this pump also is adapted to produce extremely high discharge pressures, a control valve F' similar to that of the previous embodiment is provided to regulate the amount of pressure delivered to the slide opposing piston 232. The slide opposing piston 232 is positioned in a bore 234 beneath the slide 222 in the body member G, and is supplied with pressure from the control port 236 of the control valve F'. The control port 236 communicates with an extension 238 of the discharge passage 210 in which a spool valve 240 similar to that of the previous embodiment is positioned. The pump discharge pressure is therefore applied to the inner end of the spool valve 240; and the outer end of the spool valve 240 projects into a spring chamber 242 where it is abutted by a spring abutment plate 244 biased inwardly by the coil spring 246. The other end of the coil spring 246 is held in place by a threaded closure member 248 which is screwed into the outer end of the spring chamber 242. An exhaust passage 250 communicating with the pump internal chamber 150 is provided for that portion of the valve bore 238 which is positioned on the opposite side of the control port 236 from the portion containing the pump discharge pressure.

The control valve F' and the slide structure E' operate in a similar fashion to the corresponding portions of the preceding embodiment and will not be described in detail. Suffice it to say that an increase in pump discharge pressure above the predetermined set pressure as determined by the spring 246, causes a shifting of the spool valve 240 to admit additional pressure to the bottom side of the piston 232 to produce an upward shifting of the slide 222 during those portions of the rotor cycle wherein substantially no rocking forces are exerted upon the swash plate 184. Upward movement of the slide 222 will of course cause the swash plate 184 to gradually assume a position more nearly parallel with respect to the rotor 162, until the displacement of the pump substantially corresponds to the consumption of the system at the predetermined set pressure of the pump. A decrease in pump discharge pressure permits the spring 246 to bias the slide valve 240 inwardly until the con-

trol port 236 is communicated with the passage 250, whereupon the pressure in control port 236 is decreased sufficiently to permit the slide 222 to be moved downwardly into a position increasing the pump displacement sufficiently to balance the consumption of the system to which the pump is connected.

Although the invention has been described in considerable detail, I do not wish to be limited to the particular constructions shown and described; and it is my intention to cover hereby all adaptations, modifications and arrangements thereof which come within the practice of those skilled in the art to which the invention relates.

I claim:

1. In a positive displacement hydromechanical device: a body member, first and second members in said body member at least one of which first and second members rotates relative to the other, at least one positive displacement means operatively connected between said first and second members so as to produce chambers whose volume varies during relative rotation between said first and second members, one of said first and second members also being movable in a first direction relative to the other of said members in a manner changing the displacement of said chambers, a slide in said body member held against movement in said first direction, said slide being reciprocable in a direction generally perpendicular to said first direction, and a mechanical connection between said slide and said one of said first and second members which transfers tension and compression forces therebetween in a direction which differs from a normal to the direction of movement of said slide by an angle whose tangent is less than the coefficient of friction between said slide and said body member, whereby said one of said members is held by said slide against fluctuating movement produced during operation of the device, and whereby the displacement of said device can be varied by the positioning of said slide.

2. In a radial piston hydromechanical device: a body member having an internal chamber therein having a longitudinal axis with first and second generally mutually perpendicularly extending imaginary planes passing through said axis, an axially positioned rotor having at least one generally radially extending fluid pressure chamber therein, a cam member positioned about said rotor member, a piston in said fluid pressure chamber extending radially outwardly thereof for engagement with said cam member, inlet and outlet porting on opposite sides of said first imaginary plane for successive communication with said fluid pressure chamber during rotation of said rotor, pivot means positioned on one side of said first imaginary plane and pivoting said cam to said body member to permit arcuate movement of said cam about said pivot means in the general direction of said first plane, a slide in said body member generally paralleling said second imaginary plane and restraining movement of said slide in the direction paralleling said first plane, and slot and pin means connecting said slide and said cam member, the sides of said slot being generally in abutment with opposite peripheral edges of said pin and extending at an angle relative to said second imaginary plane whose tangent is less than the coefficient of friction between said slide and body member, whereby said cam is held by said slide against fluctuating movement produced during operation of the device, and whereby the displacement of said device can be varied by the shifting of said slide.

3. In a radial piston hydromechanical device: a body member having an internal chamber therein having a longitudinal axis with first and second generally mutually perpendicularly extending imaginary planes passing through said axis, a pintle in said chamber centered on said axis, a rotor positioned about said pintle and having at least one generally radially extending fluid pressure chamber therein, a cam member positioned about said rotor member, a piston in said fluid pressure chamber

extending generally radially outwardly thereof for engagement with said cam member, said pintle having inlet and outlet porting on opposite sides of said first imaginary plane for successive communication with said fluid pressure chamber during rotation of said rotor, abutment means positioned on one side of said first imaginary plane connecting said cam to said body member and permitting movement of said cam in the direction of said first imaginary plane, a slide in said body member held against motion except in a direction generally paralleling said second imaginary plane and positioned to one side thereof, said slide and body member having a coefficient of sliding friction therebetween opposing movement of said slide, a mechanical connection transferring tension and compressive forces from said cam to said slide at an angle relative to a normal to said second imaginary plane the tangent of which angle does not exceed said coefficient of friction, and means for positioning said slide.

4. In a radial piston pump: a body member having an internal chamber therein having a longitudinal axis with first and second generally mutually perpendicularly extending imaginary planes passing through said axis, a pintle in said chamber centered on said axis, a rotor positioned about said pintle and having at least one generally radially extending fluid pressure chambers therein, a cam member positioned about said rotor member, a piston in said fluid pressure chamber extending radially outwardly thereof for engagement with said cam member, said pintle having inlet and outlet porting on opposite sides of said first imaginary plane for successive communication with said fluid pressure chamber during rotation of said rotor, pivotal abutment means positioned on the discharge stroke side of said first imaginary plane connecting said cam to said body member and permitting arcuate movement of said cam in the direction of said first imaginary plane, a cylindrical bore in said body member generally perpendicular to said first imaginary plane and positioned to one side of said second imaginary plane, a slide in said bore, slot and pin means connecting said slide and said cam member, the sides of said slot being generally in abutment with opposite peripheral edges of said pin and extending at an angle relative to said second imaginary plane, the tangent of which angle is less than the coefficient of friction between said slide and body member, whereby said cam is held by said slide against fluctuating movement produced during operation of the pump and whereby the displacement of said pump can be varied by the shifting of said slide, means biasing said slide to its position producing maximum stroke of said piston, and means using pressure discharge of said pump to bias said slide in the opposite direction to decrease the stroke of said piston.

5. In a rotary axial piston pump: a body member having a chamber therein, a rotor in said chamber journaled for rotation about an axis extending through said chamber, said rotor having an end surface extending generally transversely to said axis, at least one generally axially extending chamber in said rotor with an opening to said end surface, a swash plate adjacent said end surface of said rotor and journaled for pivotal movement with respect thereto, a fluid displacement member in said chamber and operatively connected to said swash plate to be stroked thereby, a slide in said body member held against movement in an axial direction but slidable in a direction generally transverse to said axis, and means connecting said slide and said swash plate in a manner transferring compression and tension forces therebetween at an angle relative to a normal to the reciprocating movement of said slide, the tangent of which angle is less than the

coefficient of friction between said slide and body member.

6. In a rotary axial piston pump: a body member having a chamber therein, a rotor in said chamber journaled for rotation about an axis extending through said chamber, said rotor having an end surface extending generally transversely to said axis, at least one generally axially extending cylinder chamber in said rotor with an opening to said end surface, a swash plate adjacent said end surface of said rotor and journaled for pivotal movement with respect thereto, a fluid displacement member in said cylinder chamber and operatively connected to said swash plate to be stroked thereby, a slide in said body member held against movement in an axial direction but slidable in a direction generally transverse to said axis, and slot and pin means connecting said slide and said swash plate, the sides of said slot being generally in abutment with opposite peripheral edges of said pin and extending at an angle relative to a normal to said axis, the tangent of which angle is less than the coefficient of friction between said slide and body members, whereby said swash plate is held by said slide against fluctuating movement produced during operation of the pump, and whereby the displacement of said pump is varied by the positioning of said slide.

7. In a rotary axial piston pump: a body member having a chamber therein, a rotor in said chamber journaled for rotation about an axis extending through said chamber, said rotor having an end surface extending generally transversely to said axis, at least one generally axially extending cylinder chamber in said rotor with an opening to said end surface, a swash plate adjacent said end surface of said rotor and journaled for pivotal movement with respect thereto about an axis generally passing through said first mentioned axis, a fluid displacement member in said cylinder chamber and operatively connected to said swash plate to be stroked thereby, a slide in said body member held against movement in an axial direction but slidable in a direction generally transverse to said first mentioned axis, and slot and pin means connecting said slide and said swash plate, the sides of said slot being generally in abutment with opposite peripheral edges of said pin and extending at an angle relative to a perpendicular plane to said first mentioned axis, said angle having a tangent which is less than the coefficient of friction between said slide and body member, means biasing said slide in a direction which produces maximum stroke of said fluid displacement member, and means using pressure discharge of said pump to bias said slide in the opposite direction reducing the stroke of said fluid displacement member.

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