Oct. 14, 1958 L. J. LARSEN ET AL 2,855,858

POSITIVE DISPLACEMENT PUMP

Filed April 12, 1957 2 Sheets-Sheet 1

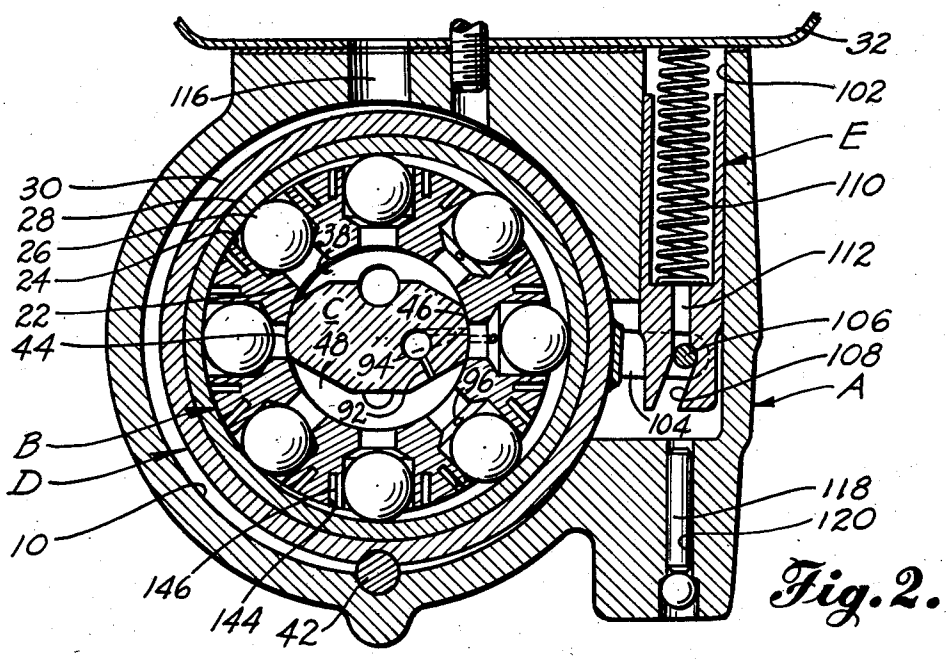


Fig. 2.

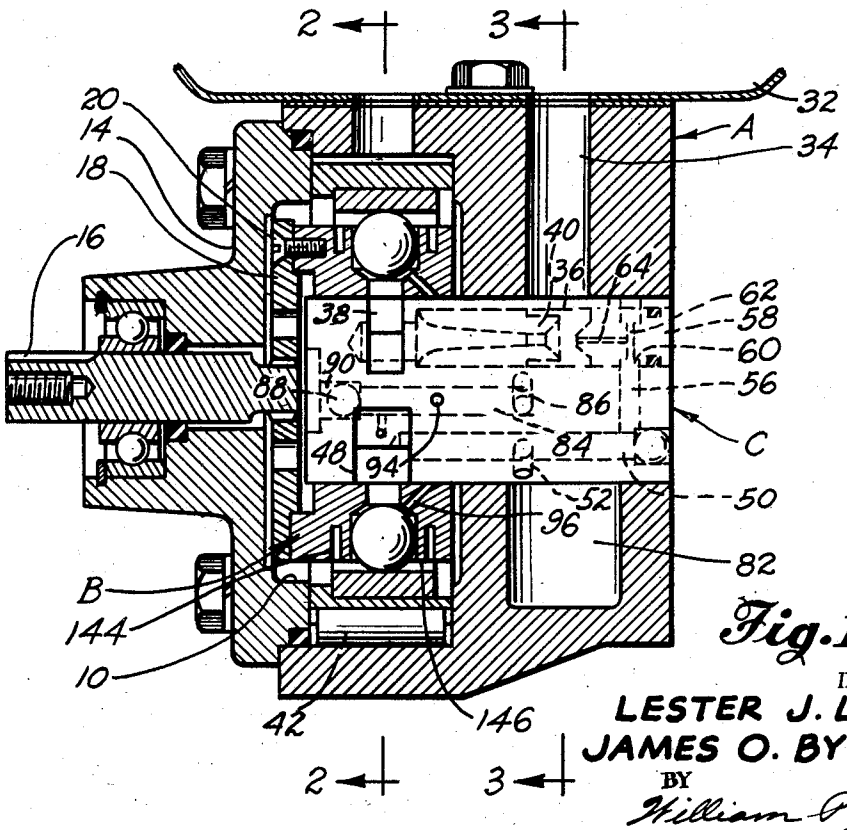


Fig. 1.

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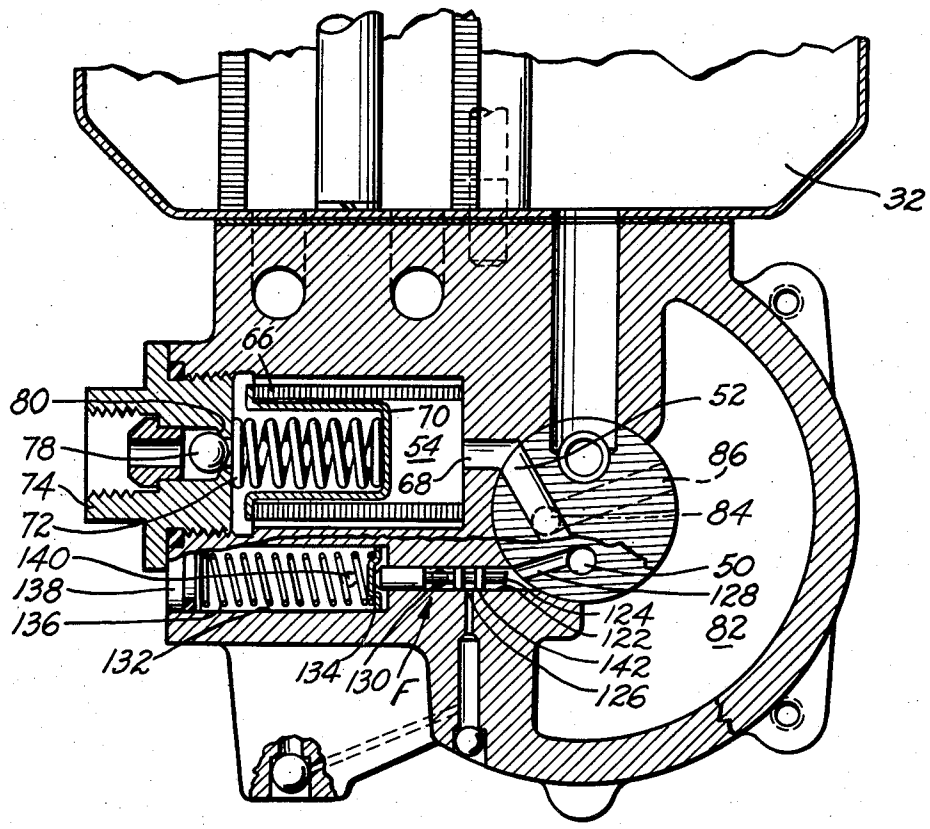


Fig. 3.

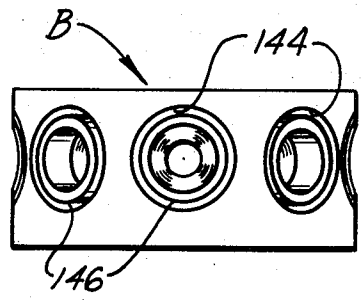


Fig. 4.

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POSITIVE DISPLACEMENT PUMP

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Application April 12, 1957, Serial No. 652,574

10 Claims. (Cl. 103—161)

The present invention relates to fluid handling devices having close fitting cylinder and piston arrangements therein which are subject to changing temperature conditions during their operation; and more particularly to hydraulic ball piston pumps for developing comparatively high discharge pressures.

Ball piston pumps must usually be made with much closer fits between their pistons and cylinders than would be provided in conventional piston pumps used in the same service. This is true because ball piston pumps have only a line seal preventing leakage past their pistons, while conventional cylindrically shaped pistons have a comparatively large surface to effect a seal between its opposite ends. Because of this fact, an extremely close fit must be used in ball piston pumps—which fit will normally be in the order of four ten thousandths of an inch where hydraulic pressures over approximately 1000 p. s. i. are to be developed.

Pumps of the ball piston type have never met with appreciable commercial success in high pressure hydraulic systems inasmuch as none have given a satisfactory service life, particularly when used in circuits having comparatively limited volumes. Considerable scuffing of the cylinder walls usually occurs after a period of time—greatly increasing the leakage across the ball pistons. In order to reduce the amount of scuffing experienced, most prior art pumps with which we are familiar have used extremely hard materials for their rotors and balls. Even these pumps have not given satisfactory service in hydraulic circuits such as are used in farm machinery for operating power steering, power lifts, and the like.

An object of the present invention is the provision of a new and improved radial piston ball pump capable of developing pressures of up to approximately 2500 pounds per square inch in a hydraulic circuit of the type used in farm machinery, and under which conditions the pump will have a service life which is comparable to that of other types of pumps.

A further object of the invention is the provision of a new and improved radial piston ball pump in which the cylinder containing rotor is formed in such a way that heat transferred to the rotor and ball pistons from the liquid being pumped produces an expansion of the cylinder walls comparable to that produced in the ball piston so as to substantially maintain the desired clearance between the ball piston and cylinder wall as the temperature of the hydraulic fluid being pumped increases.

A still further object of the invention is the provision of a new and improved radial piston pump, of the above described type, in which annular recesses are formed around the cylinder chambers to restrict the flow of heat from the side walls of the cylinder to the main portion of the rotor containing the cylinders sufficiently to cause the side walls of the cylinders to expand at a rate comparable with that of the ball pistons during changing temperature conditions of the fluid being pumped.

Another object of the invention is to correct similar

problems in other types of machinery wherever they occur.

The invention resides in certain constructions, and combinations 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 the preferred embodiment described with reference to the accompanying drawings forming a part of this specification, and in which:

Figure 1 is a cross-section view of a radial ball piston pump showing the cylindrical pintle about which the rotor revolves, unsectioned;

Figure 2 is a cross-sectional view 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; and

Figure 4 is a plan view of the rotor shown in Figures 1 and 2.

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 journalled about an axially extending pintle C which projects into the internal chamber 10 from one end wall 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 journalled 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 antifriction 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 notch 38 supplying the rotor and the inlet passage 34. 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 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 of both the body member A and the support member 30 in such manner as to limit all but arcuate movement 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 cam member B 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 land 44. 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.

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 starting or most inwardly position. Just after the time that the ball pistons 26 start to move inwardly, the inner ends of the cylinders 24 move off of the land 46 into communication with a discharge groove 48 in the lower surface of the pintle C. The discharge groove 48 corresponds in length to the inlet groove 38 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. High pressure fluid is also supplied for the pressurizing of the pump 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 orifice section 40. The fluid to be discharged from the pump passes through an annular filter 66 which is 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 therethrough containing a check valve 78 and cooperating valve seat 80 for the prevention of return flow through the pump.

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 provide 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 discharge by means of a bleed passage adapted to produce a small but substantially continuous flow to the accumulator chamber. A longitudinal drilling 84 is provided in the pintle—the inner end of which drilling is intersected by 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. During this time 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 instance 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. The drillings 96 are positioned in such a way as to register with the opening 94 during the time that the cooperating cylinder 24 is valved off from both the suction and discharges 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 drillings 94 and 96 are further positioned such that the groove 96 moves out of register with the opening 94 while the individual chamber is valved off from the suction and discharge grooves 38 and 48 and immediately thereafter the cylinder is communicated 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 without producing a surge of pressure from the pump discharge into each of the cylinder 24 as they are opened to the discharge system. 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 the accumulator 82 from being transmitted to the pump 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, 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 demand for pressure fluid falls off, the camming member D is shifted in a direction decreasing the displacement of the pump until the amount delivered equals the system demand at the predetermined set pressure of the pump.

The control structure shown comprises a slide E 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 line of motion of the slide E. 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 communicating the vertical drilling 112; and the upper end of the slot 108 and the chamber 102 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 reservoir.

Shifting movement of the slide E in the embodiment shown in the drawings is accomplished by using changes in pressure of the fluid discharge from the pump. Inasmuch as the pressure reaction forces produced upon the

camming member during operation of the pump fluctuate to 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 by reason of the angle on which they act, 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 against 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 communciated 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.

Radial ball piston pumps of the general type above described have had very poor service life in hydraulic systems such as are used on farm machinery, where the hydraulic systems contain only a few gallons of fluid which is continually circulated through the system. Large amounts of heat are generated by leakage of the fluid being pumped past the ball pistons, as well as by other throttling processes through out the hydraulic system. The temperature of the fluid being pumped may be raised rapidly by the fluid friction so induced to temperatures above 200° F. even though fluid cooling means are employed. The heat generated by leakage past the ball pistons is particularly troublesome, in that it is transferred, with no possible cooling effect by the body of the pump, directly to the ball pistons; which must transmit their heat to the body of the rotor through a single line contact with the cylinder walls. Radial ball piston pumps have never met with much success in high pressure systems of this type because the balls have produced sufficient scuffing of their cylinder chambers after only a short service life as to be unable thereafter to adequately operate the system. Attempts have been made to build radial ball piston pumps for services of the above type, but in all instances with which applicants are familiar, the approach has been to use extremely hard alloys for the ball pistons and their rotors to reduce scuffing, and even these pumps have not met with appreciable success in systems of the above described type.

Applicants have found that the scuffing of the cylinder walls by the ball pistons is not produced, per se, by a rubbing action therebetween as the balls roll around the cam, but is produced by reason of the fact that the rotor does not heat up as fast as the ball pistons do during the operation of the pump. The design clearance between the ball pistons and their cylinder walls in the pump above described is .0004", and during the operation of the pump, sufficient temperature difference between the balls and rotor is achieved to produce an interference fit between the balls and their cylinder walls. Thereafter, the balls stop rotating in their cylinders and a scuffing action is first produced on the balls by the sliding action of the balls over the hardened race ways. This creates a rough spot on the balls, and if the pump is not ruined by continued operation at this time, it will rapidly deteriorate when the balls do start to rotate again by the scuffing action of the roughed balls on the softer cylinder walls. This was discovered when a ball and rotor having a fit permitting the ball to fall out of its cylinder at room temperatures, was placed in oil having a temperature of approximately 200° F. for approximately 5 minutes. When the rotor containing the ball was removed from the hot oil, the ball could not be moved in its cylinder. Thereafter an annular recess was milled in a rotor of the same design around one of its cylinder chambers; and the above experiment repeated. It was found that the ball piston would fall out of its cylinder regardless of the length of time that it had been immersed in the hot oil. Still other experiments have shown that an appreciable length of time is required for the rotors to reach an equilibrium temperature with the oil. In the case of the rotor used in the above experiments, approximately one hour was required for the rotor to assume a temperature of within approximately twenty degrees of the oil in which it was immersed. In other experiments ball pistons which were scuffed were placed in a rotor and operated for short periods of time. In all cases, scuffed balls caused excessive wear on cylinder walls.

According to the principles of the present invention, recesses **144** are milled into the outer surface of the rotor B surrounding each of its cylinder chambers **24**. The recesses **144** are spaced from the cylinder chamber a sufficient distance to provide a cylinder wall **146** of sufficient thickness to withstand the hydraulic pressures to be developed and to adequately support its ball piston **26** while it rolls around the camming member D. The depth of the recess **144** should preferably extend to a point inwardly from the point where the ball abuts the cylinder at the ball's innermost point of movement. The recess **144** therefore requires heat added to the side walls of the cylinder to be transferred down the length of the side walls **146** into the base of the rotor. This greatly restricts the heat transfer to the remaining portion of the rotor from what is experienced in an unrecessed rotor. The recesses **144** should be at least co-extensive with the travel of the cylinder contacting portions of the pistons **26**; and the thickness of the cylinder walls **146** will preferably be no thicker than required to adequately contain the pressures involved and to adequately support the pistons. If the rotor B were to be made by means of a casting process, annular bosses extending radially outwardly from the base, or inner portion of the rotor, could be provided to form the cylinder walls **146**. Another suitable construction would be to form the cylinder walls by means of tubing which projects radially outwardly of the inner or base portion of the rotor.

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, are moving 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 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 the 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 following pressurization 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. The accumulator **82** is sized sufficiently large to control its pressure drop each time a drilling **96** is communicated with the drilling **94** to within predetermined limits; 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.

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. When the discharge pressure of the pump exceeds its predetermined set pressure, the coil spring **136** yields sufficiently to permit the spool valves inner land **142** to be moved sufficiently to communicate the pump discharge pressure to 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 attached, 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 forces 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 increasing 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.

Conversely, a drop in pump discharge pressure, decreases the pressure exerted on the hydraulic piston **118**, permitting the slide E to move downwardly, and the displacement of the pump to increase until it equals the demand of the system being supplied with fluid by the pump.

As previously explained, the recesses **144** around each of the rotor cylinders are so proportioned as to restrict the passage of heat from the side walls of each cylinder to the base of the rotor B sufficiently to cause the side walls **146** to expand and contract at a rate similar to that of the ball pistons **26** during the operation of the pump.

While the preferred embodiment of the invention has been described in considerable detail, we do not wish to be limited to the particular constructions shown and described, and it is our 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.

We claim:

1. In a fluid handling device having an inlet and discharge port: a body member having a cylinder chamber therein, a piston in said cylinder chamber forming a variable volume chamber with the side walls of said cylinder chamber and constructed and arranged to reciprocate in said cylinder chamber, and means for causing fluid to flow in and out of said variable volume chamber, the portion of said body member forming the side walls of said cylinder chamber being formed by a generally tubularly shaped projection, which projection is substantially coextensive in length with the length of the side walls swept by said piston, said side walls being of a sufficient thickness to adequately support the pistons but being thin enough to restrict the flow of heat from the side walls of the cylinder chamber to the base of the body member sufficiently to cause the side walls of the cylinder chamber to expand at a rate comparable to that of said pistons as the temperature of the fluid flowing through said variable volume chamber changes.

2. In a fluid handling device having an inlet and discharge port: a body member having a cylinder chamber therein opening into its external surface, a piston in said cylinder chamber forming a variable volume chamber with the side walls of said cylinder chamber and constructed and arranged to reciprocate in said cylinder chamber, and means for causing fluid to flow in and out of said variable volume chamber, said body member having a recess in its external surface surrounding said cylinder chamber, said recess being substantially coextensive in depth with the length of the side walls swept by said piston, and being constructed and arranged such that heat added to the portion of the body member between said recess and the cylinder chamber from the fluid passing through said cylinder chamber will be conducted into the remaining portion of the body member at a rate causing the portion of the side walls of the cylinder chamber swept by said piston to expand and contract at a rate substantially equal to that of the piston in said cylinder chamber.

3. In a fluid handling device having an inlet and discharge port: a body member having a cylinder chamber therein opening into its external surface, a piston in said cylinder chamber forming a variable volume chamber with the side walls of said cylinder chamber and constructed and arranged to reciprocate in said cylinder chamber, and means for causing fluid to flow in and out of said variable volume chamber, said body member having an annular recess machined into its external surface surrounding said cylinder chamber, said recess being substantially coextensive in depth with the length of the side walls swept by said piston, and being constructed and arranged such that heat added to the portion of the body member between said recess and the cylinder chamber from the fluid passing through said cylinder chamber will be conducted into the remaining portion of the body member at a rate causing the portion of the side walls of the cylinder chamber swept by said piston to expand and contract at a rate substantially equal to that of the piston in said cylinder chamber.

4. In a positive displacement pump having inlet and discharge connections; a body member having a cylinder chamber therein, a piston in said cylinder chamber forming a variable volume chamber with the side walls of said cylinder, means constructed and arranged to re-

ciprocate said piston in said cylinder chamber to alternately increase and decrease the volume of said variable chamber, and valve means constructed and arranged to communicate said variable volume chamber with said inlet connection when its volume is increasing and to communicate said variable volume chamber with said discharge connection when its volume is decreasing, said body member having a recess therein surrounding said cylinder chamber, said recess being substantially coextensive in length with the length of the side walls of said cylinder swept by said piston, and being constructed and arranged such that heat added to the portion of the body member between said recess and said cylinder chamber by the heat of the fluid passing through said variable volume chamber will be conducted into the remaining portion of the body member at a rate causing the portion of the side walls of the cylinder swept by said piston to expand and contract at a rate substantially equal to the piston in said cylinder.

5. In a radial piston pump: a housing having an internal pump chamber therein, a pintle in said pump chamber, a generally annularly shaped rotor positioned around said pintle, said rotor having at least one cylinder chamber extending radially therethrough between its surface adjacent said pintle and its radially outer surface, a camming member eccentrically positionable around said rotor, a piston in each cylinder chamber adapted to abut said camming member in a manner causing each piston to move in and out with respect to its cylinder chamber, said pintle having suitable inlet passages for communication with each cylinder chamber as its piston moves radially outwardly and suitable discharge passages for communication with each cylinder chamber as its piston moves radially inwardly, and said rotor having a recess in its radially outer surface surrounding each cylinder chamber, each recess being substantially coextensive in depth with the length of cylinder wall abutted by its piston during its in and out motion, and being constructed and arranged such that heat added to the portion of the rotor between said recess and said cylinder chamber from the fluid during the pumping operation will be conducted into the remaining portion of the body member at a rate causing the side walls abutted by each piston to expand and contract at a rate substantially equal to that of the piston in said cylinder.

6. In a radial ball piston pump: a housing having an internal pump chamber therein, a pintle in said pump chamber, a generally annularly shaped rotor positioned around said pintle, said rotor having at least one cylinder chamber extending radially therethrough between its surface adjacent said pintle and its radially outer surface, a camming member eccentrically positionable around said rotor, a ball piston in each cylinder chamber adapted to roll around said camming member in a manner causing each piston to move in and out with respect to its cylinder chamber, said pintle having suitable inlet passages for communication with each cylinder chamber as its piston moves radially outwardly and suitable discharge passages for communication with each cylinder chamber as its piston moves radially inwardly, and said rotor having an annular recess machined into its radially outer surface surrounding each cylinder chamber, each recess being substantially coextensive in depth with the length of cylinder wall abutted by its piston during its in and out motion, and being constructed and arranged such that heat added to the portion of the rotor between said recess and said cylinder chamber from the fluid during the pumping operation will be conducted into the remaining portion of the body member at a rate causing the side walls abutted by each piston to expand and contract at a rate substantially equal to that of the piston in said cylinder.

7. In a fluid handling device having an inlet and discharge port: a body member having a cylinder chamber therein, a piston in said cylinder chamber forming

a variable volume chamber with the side walls of said cylinder chamber and constructed and arranged to reciprocate in said cylinder chamber, and means for causing fluid to flow in and out of said variable volume chamber, the portion of said body member forming the side walls of said cylinder chamber being formed by a generally tubularly shaped projection, which projection is substantially co-extensive in length with the length of the side walls swept by said piston, and being constructed and arranged such that heat added to the portion of the body member formed by said generally tubularly shaped projection from the fluid passing through said cylinder chamber will be conducted into the remaining portion of the body member at a rate causing the portion of the side walls of the cylinder chamber swept by said piston to expand and contract at a rate substantially equal to that of the piston in said cylinder chamber.

8. In a ball piston rotary pump having an inlet and discharge port: a rotor having a cylinder chamber therein, a piston in said cylinder chamber forming a variable volume chamber with the side walls of said cylinder chamber and constructed and arranged to reciprocate in said cylinder chamber, and means for causing fluid to flow in and out of said variable volume chamber, the portion of said rotor member forming the side walls of said cylinder chamber being formed by a generally tubularly shaped projection, which projection is substantially co-extensive in length with the length of the side walls swept by said piston, and being constructed and arranged such that heat added to the portion of the rotor member formed by said generally tubularly shaped projection from the fluid passing through said cylinder chamber will be conducted into the remaining portion of the rotor member at a rate causing the portion of the side walls of the cylinder chamber swept by said piston to expand and contract at a rate substantially equal to that of the piston in said cylinder chamber.

9. In a radial piston pump: a housing having an internal pump chamber therein, a pintle in said pump chamber, a rotor positioned about said pintle, said rotor having at least one cylinder chamber extending radially therethrough between its surface adjacent said pintle and its radially outer surface, a camming member eccentrically positionable around said rotor, a piston in each cylinder chamber adapted to abut said camming member in a manner causing each piston to move in and out with respect to its cylinder chamber, said pintle having suitable inlet passages for communication with each cylinder chamber as its piston moves radially outwardly and suitable discharge passages for communication with each cylinder chamber as its piston moves radially inwardly, the side walls of said cylinder chambers which are swept by the pistons during their in and out movement being formed by a generally annularly shaped projection of metal proceeding outwardly from the base of the rotor, said side walls being of a sufficient thickness to adequately support the pistons but being thin enough to restrict the flow of heat from the side walls of the cylinder to the base of the rotor sufficiently to cause the side walls of the cylinder to expand at a rate comparable to the pistons as the temperature of the fluid being pumped changes.

10. In a radial ball piston pump: a housing having an internal pump chamber therein, a pintle in said pump chamber, a rotor positioned about said pintle, said rotor having at least one cylinder chamber extending radially therethrough between its surface adjacent said pintle and its radially outer surface, a camming member eccentrically positionable around said rotor, a ball piston in each cylinder chamber adapted to roll around said camming member in a manner causing each piston to move in and out with respect to its cylinder chamber, said pintle having suitable inlet passages for communication with each cylinder chamber as its piston moves radially outwardly and suitable discharge passages for communication with

each cylinder chamber as its piston moves radially inwardly, the side walls of said cylinder chambers which are swept by the ball pistons during their in and out movement being formed by a generally annularly shaped projection of metal proceeding outwardly from the base of the rotor, said side walls being of a sufficient thickness to adequately support the ball pistons but being thin enough to restrict the flow of heat from the side walls of the cylinder to the base of the rotor sufficiently to cause the side walls of the cylinder to expand at a rate comparable to the ball pistons as the temperature of the fluid being pumped changes.

References Cited in the file of this patent

UNITED STATES PATENTS

982,632	Prather	Jan. 24, 1911
1,325,434	Todd	Dec. 16, 1919
2,123,391	Whitfield	Apr. 15, 1935
2,646,755	Joy	July 28, 1953