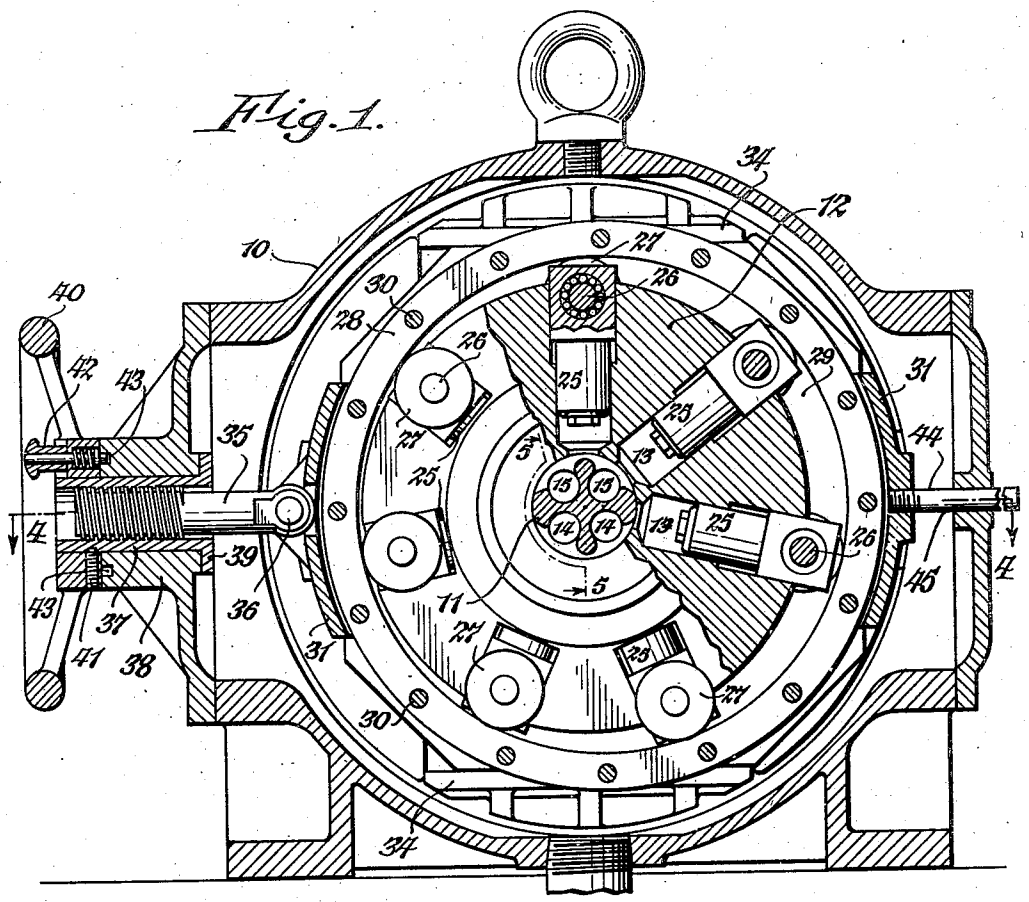
June 25, 1940. T. F. STACY 2,205,913

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Fig. 1.



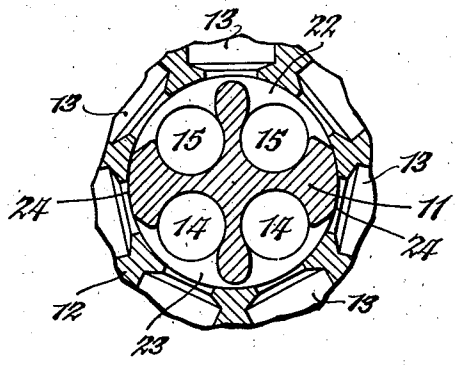


Fig. 2.

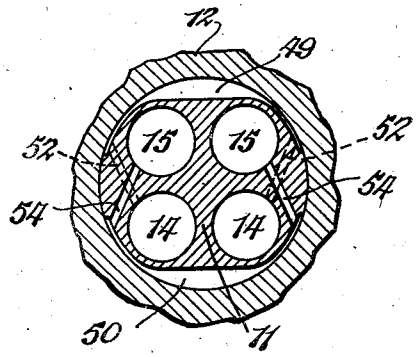


Fig. 3.

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Fig. 4.

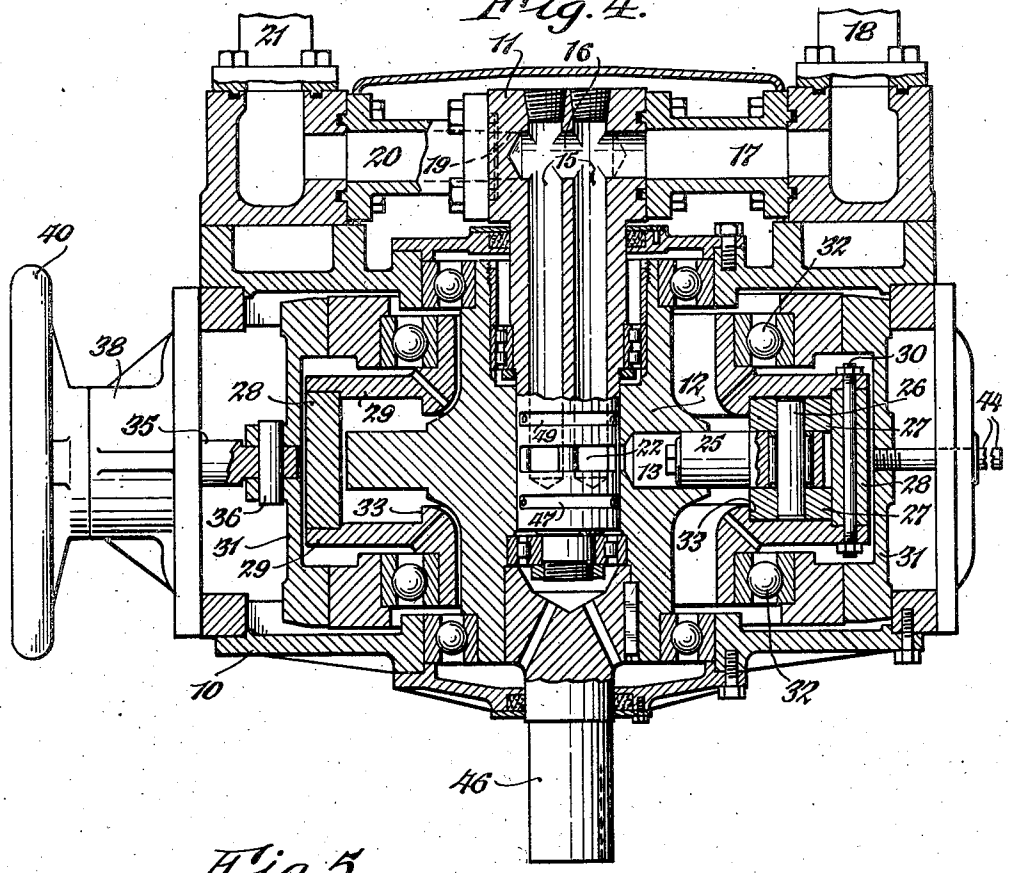


Fig. 5.

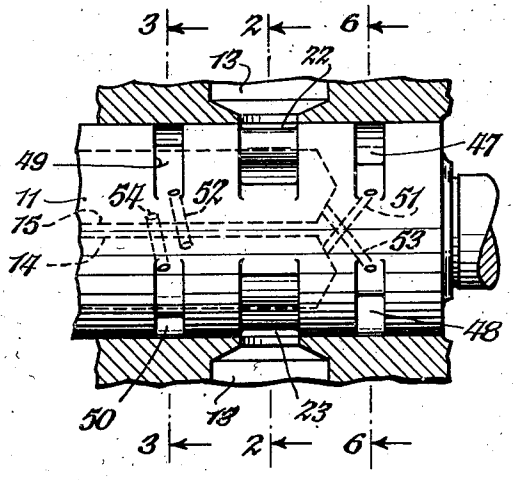
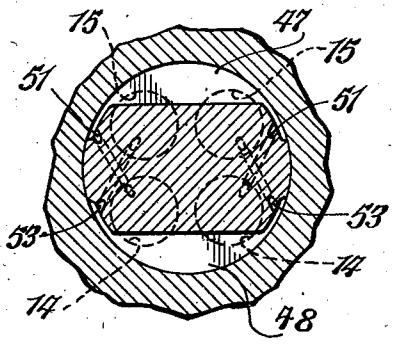


Fig. 6.



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Application February 4, 1938, Serial No. 188,617

8 Claims. (Cl. 103—161)

This invention relates to pumps and more particularly to those of the rotary piston type.

One object of the invention is to improve the smoothness and quietness of operation, and decrease the wear of operation, in pumps of this type.

Another object of the invention is to provide an improved pump of the rotary piston type with which the deflection of the timing shaft or center pintle of the pump may be materially reduced, with which the pressures between the shaft and the cylinder block may be substantially balanced, with which the wear between the timing shaft or bearing pintle and the cylinder block may be reduced, with which the load on the bearings supporting the cylinder block and shaft may be reduced, with which quieter operation will be obtained, with which the cost of manufacture will not be materially increased, and which will be generally superior in durability and performance and relatively simple and inexpensive.

Another object of the invention is to provide an improved method of reducing deflection of the timing shaft or bearing pintle of a rotary piston type of pump in use, of decreasing wear between the timing shaft and relatively rotating cylinder block, of reducing the load on the bearings, and of producing quieter operation.

Another object of the invention is to provide an improved pump of the rotary piston, variable delivery, reversible type, with which the wear between the timing shaft and the relatively rotating cylinder block will be reduced, with which the load on the bearings will be reduced, with which the deflection of the timing shaft during use may be reduced, and with which quieter operation may be obtained, whether the pump is operating in one direction or the other to deliver a fluid in either direction.

Other objects and advantages will be apparent from the following description of one embodiment of the invention, and the novel features will be particularly pointed out hereinafter in connection with the appended claims.

In the accompanying drawings:

Fig. 1 is a transverse sectional elevation through a reversible, variable delivery, rotary piston type of pump constructed in accordance with this invention;

Fig. 2 is a transverse sectional elevation through the same at the timing shaft, and illustrating on a larger scale certain details of the construction, the section being taken approximately along the line 2—2 of Fig. 5;

Fig. 3 is another transverse sectional elevation through the pump at the timing shaft, the section being taken approximately along the line 3—3 of Fig. 5;

Fig. 4 is a sectional plan of the pump, the section being taken approximately along the line 4—4 of Fig. 1;

Fig. 5 is a longitudinal sectional elevation through the pump at the timing shaft, and adjacent the intake and discharge ports of the same, to show the relation of the improvement thereto, the section being taken approximately along the line 5—5 of Fig. 1; and

Fig. 6 is a transverse sectional elevation through the pump at the timing shaft, the section being taken approximately along the line 6—6 of Fig. 5.

In the illustrated embodiment of the invention, the pump selected for the purpose of illustration is of the variable delivery, reversible, radial piston type which, in its general features, is a common article of commerce and available in the open market, and since such pumps are well known in the art only those features thereof directly concerned with the present improvement will be particularly described.

The pump includes a housing 10 from one end of which projects the timing shaft hub or center pintle 11, the shaft or pintle projecting across the interior chamber of the pump housing 10, and rotatably supporting thereon a cylinder block 12 which is concentric to the shaft. The cylinder block 12 is provided with a plurality of piston chambers 13 arranged radially of the timing shaft or pintle 11, and opening at their inner ends into the bearing surface of the cylinder block on the shaft 11. These cylinders 13 are arranged in circumferential alignment about the timing shaft, and the latter is provided with a plurality of passages 14 and 15 which extend in directions endwise of the timing shaft, into the zone thereof having the bearing surface with the cylinder block 12. For convenience in manufacture, each passage 14 and 15 (Figs. 1 and 4) is formed of two separate ducts, and these ducts may communicate with one another where they enter and leave the timing shaft.

The ducts forming the passage 15 are connected together, at the anchored end of the timing shaft, by a transverse passage 16 leading outwardly through a duct or conduit 17, the outer end of this conduit or duct 17 being connected to a service pipe 18. The ducts forming the passage 14 are connected at the same end of the timing shaft to one another by a transversely extending duct or passage 19, which is

connected through a passage 20 to a service pipe 21. The pipes 18 and 21 serve as inlet and outlet pipes alternately, depending upon the direction in which the pump delivers. Adjacent the end portion of the timing shaft 11 which is within the relatively rotatable cylinder block 12, the ducts forming the passage 15 communicate with one another through an external or outwardly opening port 22. This port is formed in the portion of the shaft in direct bearing contact with the cylinder block 12 and extends in a direction circumferentially of the timing shaft, through somewhat less than a half circumference, as shown in Figs. 2 and 4, particularly.

The ducts forming the passage 14 are connected to one another at the portion within the cylinder block 12 by means of a common port or opening 23 (Figs. 2 and 5) which is diametrically opposite from the port 22, and the ends of port 23 are spaced somewhat from the ends of the groove 22. The grooves 22 and 23 are, however, approximately circumferentially aligned, and their ends are separated from one another by uncut zone 24 of the periphery of the timing shaft. One of the ports, such as 22, is thus at all times in communication with the service pipe 18, and the other port 23 is always in communication with the service pipe 21. The cylinders 13 in the cylinder block open at their inner ends through the bearing surface between the block and the timing shaft, in circumferential alignment with the ports 22 and 23, so that as the cylinder block rotates about the timing shaft, the cylinders 13 will open in succession into the ports 22 and 23.

The openings from the cylinders 13 into the bearing surface on the timing shaft 11, are shorter, circumferentially of the shaft, than the zones 24 on the timing shaft, so that none of the cylinders 13 will, at any time, be in communication at the same time with more than one of the ports 22 or 23 and form a direct passage between the ports 22 and 23. Within each cylinder 13 is a reciprocatable piston 25. Each cylinder 13 opens outwardly to the outer periphery of the cylinder block 12, and each piston 25 at its outer end rotatably supports a pin 26 which extends through the outer end of the piston and at its ends carries rollers 27 fixed thereon, there being one roller on each side face of each piston. The two rollers, however, on each piston are connected to one another through the pin 26, and thus will be rotatably supported by the common bearing in the outer end of the piston 25.

These rollers 27 of all the pistons roll along the inner periphery of a floating rotor ring 28, which is mounted between plates 29 in any suitable manner such as by bolts 30. The ring 28 and the plates 29 thus form a rotor frame which is rotatably supported in a yoke 31 of the pump in any suitable manner, such as by ball bearings 32. The plates 29 of the rotor frame have shoulders 33 on their inner faces facing the inner circumference of the ring 28, and form parallel, outwardly facing tracks on which the rollers 27 also run, so that the rollers 27 will be confined in the annular track between the ring 28 and the shoulders 33. The yoke 31 is mounted for sliding movement between guides 34. A rod 35 is pivoted by pin 36 to one side of the yoke 31, and the free end of this rod 35 is threaded through a sleeve 37, which is rotatably confined in a bearing 38 by a flange 39 at one end of the sleeve engaging against one end of the bearing 37, and by a hand wheel 40 which is fixed, by means of set screw 41, upon the end of the sleeve 37 and abutting against the end of the bearing 38.

A spring operated pin 42 in the hub of the hand wheel engages in any of the recesses 43 in the abutting end face of the bearing 38, so as to secure the hand wheel releasably in different adjusted positions. By turning the hand wheel 40, the rod 35 may be threaded through the sleeve 37 in one direction or the other, depending upon the direction of rotation of the hand wheel, and this shifts the yoke 31 in one direction or the other so as to produce varying eccentricity of the ring 28 with respect to the timing shaft. A rod 44 projects from the yoke 31 through a passage 45 in the frame or housing of the pump so as to guide the other end of the yoke in this back and forth movement.

In the operation of pumps of this type, the cylinder block, for example, is connected to a driving shaft 46 and is rotated by any suitable source of power. As the cylinder block 12 rotates about the timing shaft or center pintle, the openings of the cylinder 13 into the bearing surface engaging the timing shaft will successively cover and uncover the ports 22 and 23. By turning the handle 40 the yoke 31 may be shifted laterally, and since the yoke rotatably mounts the ring 28 and the side plates 29 forming the rotor frame, this frame will be shifted horizontally to the left or right in Fig. 1 so as to carry the ring 28 out of concentricity to the timing shaft and to one side or the other thereof selectively. Because of this eccentricity of the ring 28 and the shoulders 33 on the side plates 29, the rollers 27 will be cammed into the cylinders and then out again in succession as the cylinder block rotates, and thus they act as pump cylinders.

Assuming that the yoke of the pump is in the position shown in Fig. 1, for example, and that the direction of rotation of the cylinder block is clockwise in Fig. 1, the pistons will be pulled outwardly progressively as they move from the left to the right in Fig. 1, and will have their maximum outward movement when they reach a nearly horizontal position at the right side of the timing shaft (Fig. 1). During most of this movement while the pistons are being drawn outwardly in succession, the inner ends of the cylinders 13 in which those pistons are moving will be in communication with the upper port 22. As the cylinders in succession closely approach an approximate horizontal position at the right in Fig. 1, they will move out of communication with the upper port 22 and as soon as they pass downwardly below the horizontal position they will be connected to the lower port 23 and continue in communication with that lower port 23 until they closely approach the horizontal position at the left in Fig. 1.

During this last movement below the timing shaft, however, while the cylinders 13 are in communication with the port 23, the eccentric cam ring 28 will force the pistons of those cylinders inwardly and expel any liquid or fluid which was drawn into those cylinders while travelling in communication with the port 23. Thus, if the cylinder block is rotated clockwise in Fig. 1, the cylinders when travelling through the upper arc of movement while in communication with port 22 will create a suction in the port 22, and through the upper passage 15 the outwardly moving pistons of those cylinders will draw a fluid into those cylinders, through the pipe 18, and then as the same cylinders move through the lower arc of movement while they are in communication with

the port 23, the pistons of those cylinders will be forced inwardly by the ring 28 and will expel the fluid from those cylinders through the passage 14 and the service pipe 21.

If the direction of rotation of the cylinder block is counter-clockwise in Fig. 1, the intake portion of the cycle will be while the cylinders are moving through the lower arc of movement while they are in communication with the port 23, in which case the fluid is drawn in through service pipe 21, and then the cylinders will expel this fluid through the service pipe 18 as the cylinders move in the upper arc of movement while in communication with the port 22. By shifting the yoke horizontally, the extent to which the pistons will be drawn outwardly may be varied and the degree of reciprocation of the pistons may be thus regulated, so as to obtain various rates of delivery selectively. If the yoke is shifted so that the ring 28 is fully concentric with the timing shaft, then there will be no reciprocating movements of the pistons, and therefore no delivery of the pump. If the yoke is shifted past concentric position to eccentricity on the other side of the concentric position, then the direction of delivery will be reversed while the pump continues to rotate in the same direction. This is the well known operation of the reversible, variable delivery type of radial piston pumps.

It will be noted that since part of the pistons will be discharging fluid while the others are drawing fluid, one of the ports 22 or 23 will be under continuous discharge pressure, and the other port will be under continuous suction pressure, and consequently this difference in pressure on opposite sides of the timing shaft will tend to cause a substantial pressure on one side of the bearing surface between the cylinder block and the timing shaft than on the other side. To lessen this inequality of pressure on this bearing surface, the shaft is provided in the peripheral surface with outwardly opening, balancing grooves 47, 48, 49 and 50. One pair of the grooves 47 and 48 is disposed endwise of the shaft to one side of the ports 22 and 23, and the other pair 49 and 50 is disposed endwise of the shaft to the other side of these ports. The grooves 47 and 48 forming one pair, are on diametrically opposite sides of the timing shaft from one another, are circumferentially aligned with one another, and are also preferably coextensive in circumferential length and alignment with the circumferential lengths of the ports 22 and 23.

Similarly, the grooves 48 and 50 are aligned endwise with the port 23, and are coextensive in circumferential length with the ends of that port. These grooves 47 to 50, inclusive, in the periphery of the timing shaft are normally closed at their outer faces by the bearing surface of the cylinder block on the timing shaft. The groove 47 is connected near its ends by ducts 51 to the passage 14 in the timing shaft, so that the space or chamber in the groove 47 will be at all times under the same pressure as exists in the passage 14. The groove 49 is also connected near its ends by ducts 52 to the passage 14, and it also will be subjected to the same fluid pressure as exists in the passage 14. Similarly, the grooves 48 and 50 are connected by ducts 53 and 54 respectively to the passage 15 in the timing shaft. Thus the grooves 48 and 50 will be subjected to the same pressures as exist in the passage 15.

Assuming that the direction of rotation of the pump and the position of the yoke are such that the pump is discharging under pressure through the port 22, this pressure on the upper surface of the timing shaft tends to increase the pressure between the bottom of the timing shaft and the rotating cylinder block, but since the grooves 48 and 50 are subjected to the same pressure as the port 22, there will be an upward pressure in the grooves 48 and 50 opposing the downward pressure in the port 22. By making the face opening areas of the grooves 48 and 50 together equal to the area of the face of port 23, and the areas of the grooves 47 and 49 approximately equal to the area of the port 22, the fluid pressure on the timing shaft at the port 22 will be balanced by the opposing pressures on that shaft in the grooves 48 and 50. Since the pressure in the port 23 is low, there will be a corresponding low pressure in the grooves 47 and 49. Thus, there will be no unequal pressures between the timing shaft and the cylinder block by reason of the unequal pressures in the ports 22 and 23, and this remains true, even if the direction of rotation of the cylinder block or the direction of delivery is changed. It will be noted from Figs. 4 and 5, for example, that when there is a substantial fluid pressure in the discharge port 22, this pressure will react in opposite directions against the side walls of that port 22, that is, the side walls which extend transversely of the timing shaft, and such pressure, with a relatively low pressure on the corresponding side walls of the opposite port 22, causes a bending moment which tends to deflect the shaft at opposite sides of the ports 22 and 23. This deflection causes unequal wear on the bearing surfaces between the shaft and the cylinder block, and disturbs the balance betweeen the shaft and block, and therefore causes noisy operation. Grooves 47, 48, 49 and 50 are therefore made of substantial depths, so that there will be substantial areas on the side walls of these grooves, and since the grooves 48 and 50, by reason of their connection to passage 15, will be under the same fluid pressure as the side walls of the port 22, there will be approximately the same pressure per unit area on the side walls of grooves 48 and 50, as upon the side walls of the port 22.

These pressures on the side walls of the grooves 48 and 50 create bending moments opposing or counter to the bending moments exerted on the walls of the port 22, and tend to flex the ends of the timing shaft in the opposite direction from the direction of flexing caused by the bending moments on the side walls of the port 22. By making the grooves 48 and 50 as deep as possible, so as to obtain in these grooves as close an approach as possible to the side wall area of the port 22, there will be a substantial balancing of the tendency to deflect the shaft, and consequently there will be a minimum deflection of the shaft. Hence, one will obtain more uniform and normal bearing pressure between the timing shaft and cylinder block. This decreases the wear between the timing shaft and the cylinder block and the load on the bearings supporting the cylinder block, and quieter operation due to a condition more nearly approaching concentricity between the cylinder block and the timing shaft by reason of the lessened actual deflection in the timing shaft.

In the particular type of pump illustrated, the passages 14 and 15 extend along the timing shaft from the same end, and therefore these passages are preferably stopped short at approximately the ports 22 and 23, as shown in Figs. 4 and 5 particularly, so that the grooves 47 and

48 may be made as deep as possible or necessary in order to counteract the bending moments in the ports 22 and 23. The grooves 49 and 50 are made as deep as possible, as shown in Fig. 3, so as to obtain counter-bending moments as great as possible in those grooves also. The depth of the grooves 49 and 50, however, is limited owing to the fact that the grooves must not be deepened until they enter the passages 14 and 15. If the direction of delivery is changed and the port 23 is under discharge pressure and the port 22 is under suction pressure, then the grooves 47 and 49 will exert the counter-bending moments which reduce the actual deflection of the timing shaft. Thus the pump may be operated in either direction, so that one may use either of the service pipes 18 and 21 as the intake pipe and the other as the delivery pipe, and the pressure between the cylinder block and timing shaft will nevertheless be balanced and actual deflection reduced to a minimum.

The operation of the pump should be clear from the foregoing description. The importance of these counter-bending moments will be apparent by an inspection of the cross section of the timing shaft, as shown in Fig. 2, where, because of the relatively large space occupied by the passages 14 and 15, there is a minimum of structure to resist the deflecting, bending moments in the port that is under discharge pressure.

It will be understood that various changes in the details and arrangements of parts, which have been herein described and illustrated in order to explain the nature of the invention, may be made by those skilled in the art within the principle and scope of the invention, as expressed in the appended claims.

I claim as my invention:

1. In a pump of the rotary piston type in which a timing shaft and a cylinder block are disposed in concentric relation to one another and relatively rotatable one upon the other, with the shaft having spaced inlet and discharge ports, the improvement in the shaft thereof which comprises an auxiliary groove in the side of the shaft opposite to and spaced longitudinally from the discharge port, and of such maximum practical depth and circumferential length that when a fluid under a pressure at least approximately equal to that in the discharge port is admitted to such groove, it will create an internal, expanding pressure on the side walls of the groove and thereby create a substantial bending moment counter to that developed in the shaft at the discharge port, and a duct leading to said auxiliary groove for supplying thereto a fluid under pressure sufficient to create the desired counter-bending moment.

2. In a pump of the rotary piston type in which a timing shaft and a cylinder block are disposed in concentric relation to one another and relatively rotatable one upon the other, with the shaft having spaced inlet and discharge ports, the improvement in the shaft thereof which comprises an auxiliary groove in the side of the shaft opposite to and spaced longitudinally from the discharge port, of such maximum practical depth and circumferential length that when a fluid under a pressure at least approximately equal to that in the discharge port is admitted to such groove, it will create an internal, expanding pressure on the side walls of the groove and thereby create a substantial bending moment counter to that developed in the shaft at the discharge port, and a passage connecting said groove to said discharge port so as to subject said groove to the fluid pressure in said discharge port.

3. A hub for a pump of the type described, having high and low pressure ports opening through the surface of the hub at circumferentially spaced points, recesses spaced in opposite directions longitudinally of the hub from the opening of the low pressure port but adjacent that port and approximately diametrically opposite from said high pressure port, and passages communicating fluid pressure from the high pressure port to said recesses, said recesses having maximum practical depth and length to provide side walls therefor which approach as closely as practical to those in said high pressure port, whereby the fluid pressure in said recesses on the side walls thereof will create in the hub counter-bending moments opposing those bending moments created by said fluid pressure on said hub in said high pressure port.

4. In a radial piston type pump, in which the hub has high and low pressure ports opening through the surface of the hub at circumferentially spaced points, with radial pressure balancing recesses on opposite sides of said opening of said low pressure port, longitudinally of the hub, and both connected to said high pressure port, that improvement which comprises said recesses on the approximately diametrically opposite side of the hub from said high pressure port of such substantial depths and lengths that those side walls of such recesses which are transverse to the hub will be of such area that when substantial fluid pressures are created in said recesses, substantial bending moments will be created in said hub opposing, and as closely as possible equaling, the bending moments in the hub that are created by the fluid pressure in said high pressure port.

5. In a radial piston type pump, in which the hub has high and low pressure ports opening through the surface of the hub at circumferentially spaced points, the improved method of reducing any deviation of the hub from a straight condition by fluid pressures in said high pressure port, which comprises creating in said hub on opposite sides of, longitudinally of the hub, but adjacent the opening of, said low pressure port, and approximately diametrically opposite from said high pressure port, substantial internal expanding forces tending to create bending moments opposing those created in said hub by fluid pressure in said high pressure port, of such amount that they substantially offset as nearly as practical, any bending of the hub adjacent the high pressure port by the fluid pressure in that port.

6. In a pump of the radial piston type in which a timing shaft and a cylinder block telescopically fit and have relative rotation, and in which the shaft has inlet and discharge ducts terminating in spaced ports in the periphery of the shaft and also has balancing grooves in its periphery within the bearing surfaces between the shaft and the block, and arranged in pairs on opposite sides of the shaft with the grooves of each pair spaced apart on opposite sides of each port lengthwise of the shaft the grooves of each pair on each side of the shaft being connected to the duct leading to the port on the opposite side of the shaft, that improvement in the shaft which includes said balancing grooves of maximum practical depth and circumferential length to provide side walls therefor which approach

in area as closely as practical to the corresponding side areas of the port at the opposite side of the shaft, whereby the fluid pressure in said balancing grooves will not only balance the radial pressures between said block and shaft, but will create counter-bending moments in the portions of the shaft at the opposite side thereof from each port for opposing those bending moments created in the shaft by the fluid pressure in said port at the opposite side of the shaft.

7. In a pump of the radial piston type in which a timing shaft and cylinder block are arranged concentrically and are relatively rotatable, and in which the shaft has a pair of passages extending in a direction endwise thereof and opening outwardly through the periphery of the shaft to form inlet and discharge ports within the bearing surface between the shaft and block, into register with which block openings are brought successively during such relative rotation, the improved method of opposing and compensating for radial deflection of the shaft by the discharge pressure in one port and suction pressure in the other port, which comprises setting up in said shaft approximately at the side opposite the discharge port and on both sides of a plane through that port, but adjacent the inlet port, internal expanding forces operating in a direction endwise of said shaft and of a strength approximately as closely as practical to the expanding forces acting also in a direction endwise of the shaft in said discharge port, whereby the actual deflection in the shaft by the pressure in the discharge port and the suction in the inlet port will be substantially reduced.

8. In a pump of the rotary piston type in which a timing shaft extends through a cylinder block and the block and shaft are relatively rotatable one upon the other, with the shaft having spaced inlet and discharge ports, the improvement in the shaft thereof which comprises a pair of auxiliary spaced grooves in the side of the shaft opposite from the discharge port and arranged on opposite sides of a plane through said discharge ports transverse to said shaft, each groove being of such maximum practical depth and circumferential length that a substantial bending moment counter but as closely equal as possible to that developed in the shaft at the discharge port will be created when an internal expanding pressure approximately equal to that in said discharge port is applied to said grooves, and passages connecting said grooves to said discharge port, so as to subject said grooves to the fluid pressure in said discharge port.

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