April 23, 1963

C. A. L. RUHL

3,086,477

VARIABLE DISPLACEMENT PUMP

Filed May 9, 1960

2 Sheets-Sheet 1

Fig.1

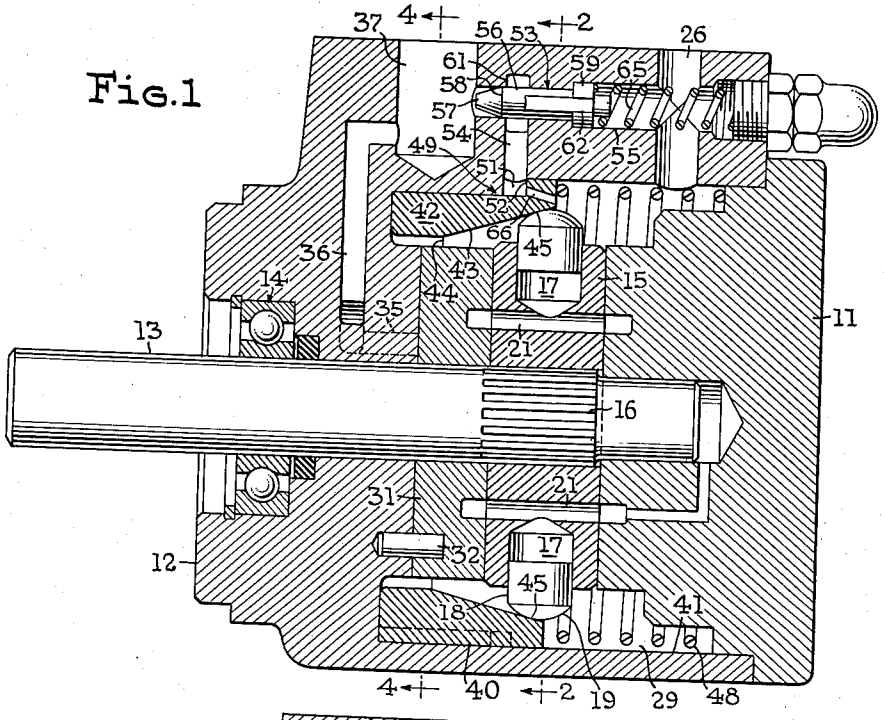
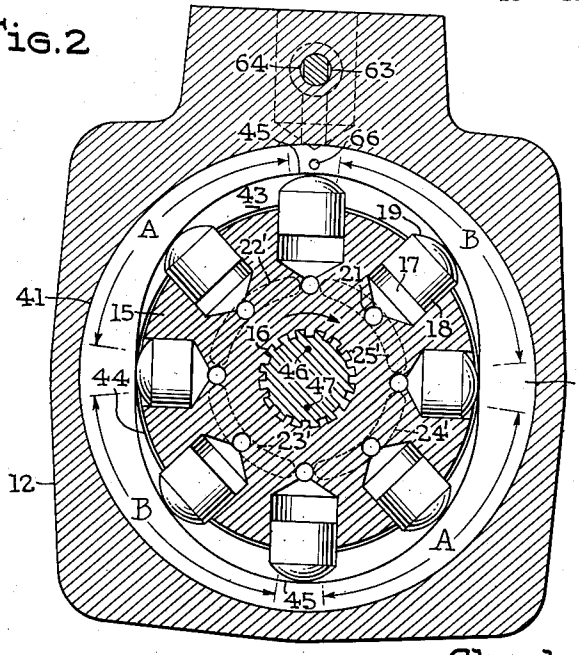


Fig.2



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Fig. 3

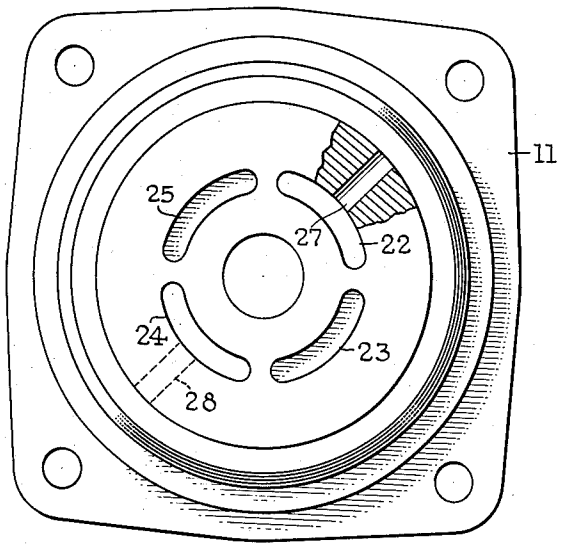
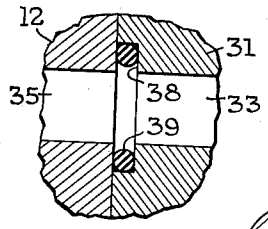


Fig. 5



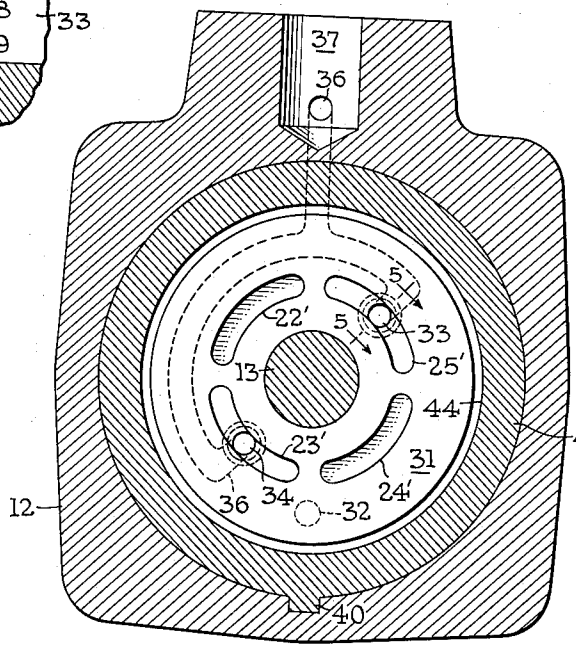


Fig. 4

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3,086,477

VARIABLE DISPLACEMENT PUMP

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1 Claim. (Cl. 103—161)

This invention relates to fluid pressure engines of the reciprocating piston type wherein the pistons are mounted in radial cylinder bores. The term engine is used herein in its generic sense and it will be understood that it includes motors as well as pumps.

The object of this invention is to provide an improved mechanism for varying the displacement of this kind of engine.

According to the invention, there is provided a cylindrical cam member which has an inner peripheral surface that is coaxial with the cylinder barrel and which is in operative engagement with the pistons at their outer ends for moving them on their discharge strokes during relative rotation between the cylinder barrel and the cam member. This inner peripheral surface is a smooth compound curve which connects a transverse circular contour at one axial location with a transverse generally elliptical contour at a spaced axial location; the minor diameter of the generally elliptical contour and the diameter of the circular contour being equal. Relative longitudinal movement between the cylinder barrel and the cam member along their common axis causes the pistons to engage generally elliptical contours of varying major diameters and thus results in a gradual change in the length of the strokes of the pistons; the length of the strokes decreasing as the pistons draw nearer the circular contour.

The preferred embodiment of the invention will now be described in detail with reference to the accompanying drawings, in which:

FIG. 1 is an axial sectional view of a rotary cylinder barrel pump incorporating the invention; the arcuate inlet and discharge ports being rotated into the plane of section for a clearer understanding of the arrangement of the parts.

FIG. 2 is a sectional view taken on line 2—2 of FIG. 1.

FIG. 3 is a view of the inner face of housing section 11 showing the arrangement of the arcuate inlet ports, a portion of the section having been broken away to show one of the passages connecting the inlet manifold with the arcuate ports.

FIG. 4 is a sectional view taken on line 4—4 of FIG. 1.

FIG. 5 is an enlarged sectional view taken on line 5—5 of FIG. 4 showing one of the O-ring encircled biasing areas on the rear face of the valve plate.

As shown in the drawings, the pump comprises a housing having two separable sections 11 and 12 which are connected by bolts (not shown). Extending into the housing is a drive shaft 13 which is journalled in housing section 11 and in a ball bearing 14 carried by the housing section 12, and which is connected in driving relation with a rotary cylinder barrel 15 by splines 16. The rotary cylfinder barrel 15 is provided with four pairs of diametrically opposed cylinder bores 17, each of which receives a reciprocable piston 18 whose outer end carries a spherical surface 19 which is centered on the longitudinal axis of the piston.

A plurality of bores 21 extend through the cylinder barrel 15 and are arranged so that one intersects the inner end of each cylinder bore 17. The right ends of these bores, as viewed in FIG. 1, are positioned to register successively with four equi-angularly spaced arcuate ports 22, 23, 24 and 25 formed in the inner end face of

housing section 11. These ports are spaced apart a distance equal to the diameter of bore 21. Two of these ports, namely 22 and 24, communicate with the inlet port 26 formed in housing section 12 through radial passages 27 and 28, respectively, and the annular inlet manifold 29, and the other two ports 23 and 25 are blind and function merely as balancing chambers. The left ends of the through bores 21 are positioned to register with four similar arcuate ports 22', 23', 24' and 25' formed in the front face of valve plate 31 which floats freely between housing section 12 and cylinder barrel 15 but which is prevented from rotating by a pin 32. The arcuate ports 23' and 25', which are directly opposite ports 23 and 25, communicate with bores 33 and 34, respectively, that extend through the valve plate 31 and connect with the longitudinal legs 35 (only one shown in FIG. 1) of a discharge manifold 36. This manifold 36 leads into discharge port 37 formed in housing section 12. The rear face of the valve plate 31 is provided with two counterbores 38 (one being shown in FIG. 5) which are coaxial with the bores 33 and 34 and which receive the resilient O-rings 39 which are squeezed between the valve plate 31 and housing section 12. The regions enclosed by the outer margins of these O-rings define biasing areas which are subject to discharge pressure and which develop forces that urge the valve plate 31 into sealing engagement with the cylinder barrel and, in turn, urge the cylinder barrel into sealing engagement with the inner end face of housing section 11. The arcuate ports 22' and 24', which are directly opposite ports 22 and 24, are blind and, as in the case of arcuate ports 23 and 25, serve merely as balancing chambers.

Housing section 12 is provided with a stepped bore 41 which receives a sliding cylindrical cam member 42. The cam member 42 is prevented from rotating by an integral key 40 which slides in a longitudinal slot formed in housing section 12. A portion 43 of the inner peripheral surface of cam member 42 takes the form of a smooth compound curve which connects a transverse circular contour 44 at one axial position with a transverse generally elliptical contour 45 at an axially spaced position; the elements of the surface portion 43 being straight lines. The generally elliptical contour comprises two circular arcs centered at the points 46 and 47 (see FIG. 2) and two straight portions which are tangent to and connect the adjacent ends of the two arcs. The minor diameter of the generally elliptical contour is the same as the diameter of the circular contour 44. The surface portion 43 engages the spherical outer ends 19 of the pistons 18 and serves, during rotation of the cylinder barrel, to move the pistons on their discharge strokes. Since transverse planes intersecting surface portion 43 at different axial positions between contours 44 and 45 cut generally elliptical contours of varying major diameters, the longitudinal position of cam member 42 relative to cylinder barrel 15 determines the lengths of the piston strokes.

The position of cam member 42 can be regulated in several different ways, but in the preferred embodiment it is controlled automatically by a discharge pressure compensator. In this embodiment, the cam member 42 is biased toward its maximum displacement-establishing position (shown in FIG. 1), wherein its left end is in abutment with housing section 12, by a coil compression spring 48 and is moved in the opposite direction against this bias by a fluid pressure control motor 49 which includes an annular working chamber 51 and an annular piston 52. The pressure in the working chamber 51 is controlled by a valve 53 which is connected with it by a passage 54. The control valve 53 comprises a bore 55 formed in housing section 12 and interconnecting inlet and exhaust ports 26 and 37, and a plunger 56 carrying

a tapered nose 57 that defines a control edge 58 and an enlarged portion 59 that acts as a stop to limit leftward movement. The bore 55 is encircled by an annular chamber 61 which communicates with passage 54. Plunger 56 is formed with two diametrically opposed flats which, together with aligned longitudinal slots 62 in enlarged portion 59 and the bore 55, define flow passages 63 and 64 that are in continuous communication with inlet port 26 through the right end of bore 55. The valve plunger 56 is biased to the position shown in FIG. 1 by a coil compression spring 65 and, in this position, the left ends of the flats extend into chamber 61. The plunger 56 is shifted in the opposite direction by the discharge pressure in port 37 which acts upon the nose 57.

Operation

When the pump is put in operation, the drive shaft is rotated in the direction of the arrow in FIG. 2 and the parts are in the positions shown in FIG. 1. All of the pistons are urged outward in the radial direction into contact with the cam member 42 by centrifugal force and that pair of pistons 18 in the regions A of the elliptical contour will move outward relatively to their cylinder bores. The through bores 21 associated with these pistons will, at this time, be in communication with the arcuate ports 22 and 24 so that fluid entering inlet port 26 may flow to their cylinder bores through manifold 29, radial passages 27 and 28, and the arcuate ports 22 and 24. Those bores 21 associated with the pair of pistons in the regions B of the elliptical contour register with arcuate ports 23′ and 25′, and the inward movement imparted to these pistons by the cam member 42 will cause them to discharge fluid through arcuate ports 23′ and 25′, bores 33 and 34, legs 35, manifold 36, and discharge port 37. The remaining two pairs of pistons 18 are at their outermost or innermost positions and thus they will neither draw in or expel fluid. An inspection of FIG. 2 will show that each piston 18 completes two pumping cycles, each comprising an inlet and discharge stroke, during each revolution of the cylinder barrel 15. It also will be apparent that the use of pairs of diametrically opposed pistons results in a balance of the radial components of the thrust forces transmitted between pistons 18 and cam member 42.

As long as the discharge pressure in port 37 is below that value required to produce a force on nose 57 which is greater than the bias of spring 65, valve plunger 56 will remain in the FIG. 1 position and the working chamber 51 of control motor 49 will be vented through passages 63 and 64 and through the restricted passage 66 formed in the cam member 42. Because of this, spring 48 will keep the cam member 42 in the FIG. 1 position and the length of the piston strokes will be a maximum.

When the demand for high pressure fluid decreases to such an extent that the discharge pressure force acting on nose 57 becomes greater than the force of spring 65, valve plunger 56 shifts to the right and closes the vent passages 63 and 64. Simultaneously control edge 58 overtravels annular chamber 61 and opens a path from discharge port 37 to working chamber 51 through passage 54. Due to the fact that nose 57 is tapered, rightward movement of plunger 56 permits a progressively greater quantity of fluid to flow to working chamber 51 and, since the fluid in the working chamber can escape only through the restricted passage 66, this results in a rising pressure in the working chamber. When the force developed on annular piston 52 by the pressure in working chamber 51 becomes greater than the bias of spring 48, cam member 42 moves to the right. This shift of the cam member 42 has the effect of presenting to the pistons 18 a generally elliptical cam surface of smaller major diameter and this results in a decrease in the lengths of the piston strokes and in the displacement of the pump. As discharge pressure continues to rise, the pressure in working chamber 51 increases and cam member 42 moves further to the right. When discharge pressure reaches the desired maximum, cam member 42 will be in a position in which circular contour 44 is in engagement with the pistons. At this time, rotation of cylinder barrel 15 is ineffective to reciprocate the pistons 18 and the displacement of the pump is zero.

When the demand for high pressure fluid increases, discharge pressure decreases and valve plunger 56 moves to the left thereby reducing the flow to and consequently the pressure in working chamber 51. Spring 48 now shifts cam member 42 to the left and thus causes it to increase progressively the lengths of the piston strokes. When the valve plunger 56 again vents working chamber 51, cam member 42 returns to the FIG. 1 position.

During operation of the pump, the point of contact between the surface 43 and the spherical end 19 of each piston 18 is always displaced to the left, as viewed in FIG. 1, from the longitudinal axis of the piston. This arrangement has two desirable effects. First, it produces unidirectional rotation of the pistons 18 in their cylinder bores and thus minimizes wear on ends 19. Second, it causes the axial components of the thrust forces transmitted between the pistons 18 and the cam member 42 to act to the right in FIG. 1 and urge cylinder barrel 15 into sealing engagement with the inner end face of housing section 11.

As stated previously, the drawings and description relate only to a preferred embodiment of the invention. Since many changes can be made in the structure of this embodiment without departing from the inventive concept, the following claim should provide the sole measure of the scope of the invention.

What is claimed is:

A fluid pressure engine comprising a housing containing high and low pressure ports; a drive shaft journaled in the housing; a rotary cylinder barrel connected in driving relation with the shaft but being free to move longitudinally of it, the cylinder barrel containing a plurality of pairs of diametrically opposed radial cylinder bores; longitudinal flow passages formed in the cylinder barrel, one passage intersecting each cylinder bore and each passage opening through the side faces of the cylinder barrel; a reciprocable piston in each cylinder bore, each piston having a spherical outer end; a stepped cylindrical bore formed in the housing coaxial with and encircling the cylinder barrel, the bore having a step which divides it into a small diameter portion and a large diameter portion; a cylindrical cam ring guided for sliding movement in the cylindrical bore and formed with a shoulder on its outer periphery that defines small and large diameter portions which fit the small and large diameter portions, respectively, of the cylindrical bore, the cam ring having an inner peripheral surface which engages the outer ends of the pistons and which is a smooth compound curve that connects a transverse circular contour at one axial positon with a transverse generally elliptical contour at a spaced axial position, the diameter of the circular contour being equal to the minor diameter of the generally elliptical contour; means for transmitting fluid under pressure to the shoulder on the cam ring to thereby develop a pressure force that shifts the cam ring longitudinally in the cylindrical bore; a coil compression spring coaxial with the cylinder barrel and reacting between the housing and the cam ring for opposing movement of the cam ring under the action of said pressure force; means for preventing rotation of the cam ring relatively to the housing; a stationary valve face carried by the housing and lying in a plane that is normal to the axis of rotation, the valve face containing a pair of diametrically opposed low pressure ports and a pair of diametrically opposed balance ports arranged to register sequentially with the longitudinal flow passages as the cylinder barrel rotates, the valve face being so located that the cylinder barrel is urged toward it by the axial components of the reaction forces developed between the pistons and the cam ring; a nonrotary floating

valve plate, having a valving face located in a plane that is normal to the axis of rotation, positioned at the side of the cylinder barrel opposite the stationary valve face, the valve plate contining a diametrically opposed pair of high pressure ports aligned with the balance ports in the stationary valve face and a diametrically opposed pair of balance ports aligned with the low pressure ports in the stationary valve face, the ports in the floating valve plate being arranged to register sequentially with the longitudinal flow passages as the cylinder barrel rotates; means responsive to the pressure in the high pressure ports for urging the valve plate toward the cylinder barrel; and flow passages connecting the housing high and

low pressure ports with the corresponding ports in the stationary valve face and in the valve plate.

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