

June 30, 1964

T. BUDZICH

3,139,037

HYDRAULIC APPARATUS

Filed Aug. 18, 1959

5 Sheets-Sheet 1

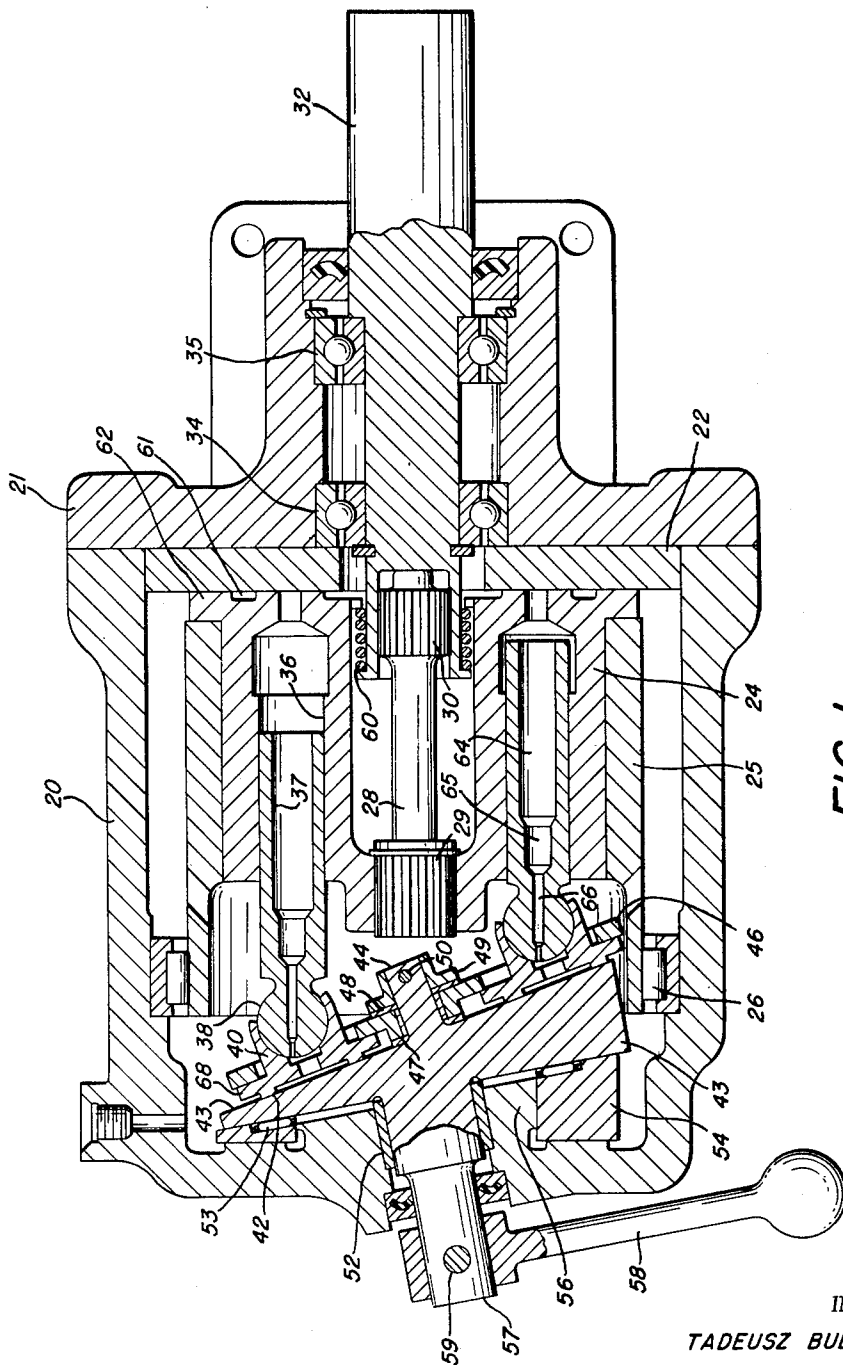


FIG. 1

INVENTOR

TADEUSZ BUDZICH

BY

Richard H. MacCulloch

ATTORNEY

June 30, 1964

T. BUDZICH
HYDRAULIC APPARATUS

3,139,037

Filed Aug. 18, 1959

5 Sheets-Sheet 2

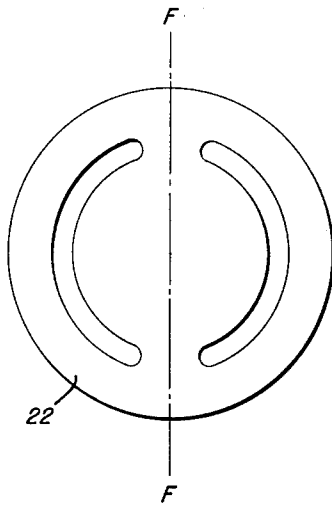


FIG. 14

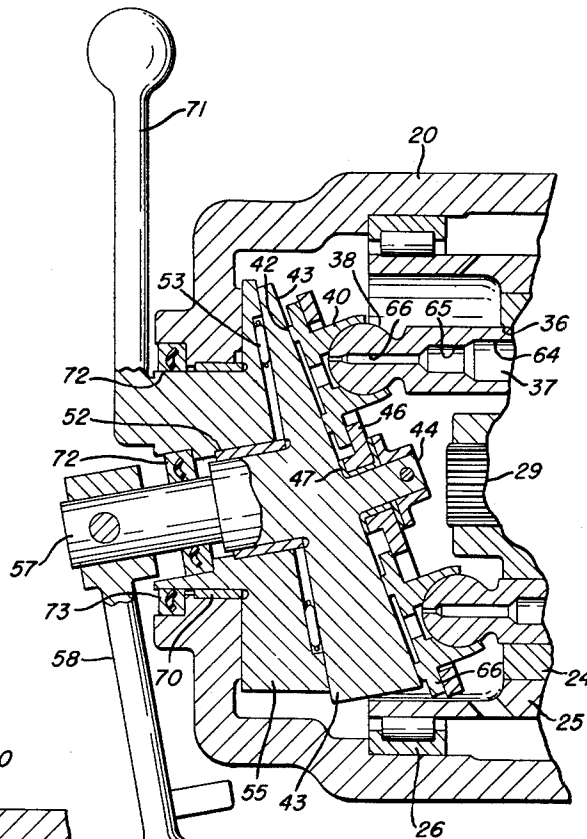


FIG. 3

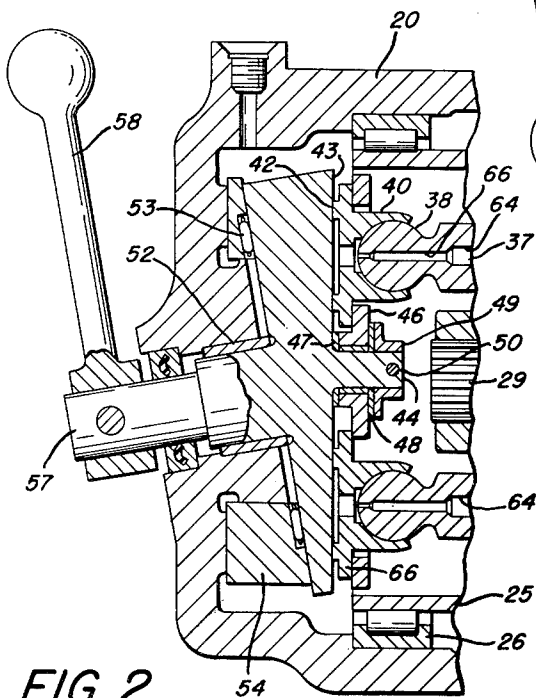


FIG. 2

INVENTOR
TADEUSZ BUDZICH

BY *Richard H. MacCuteh*

ATTORNEY

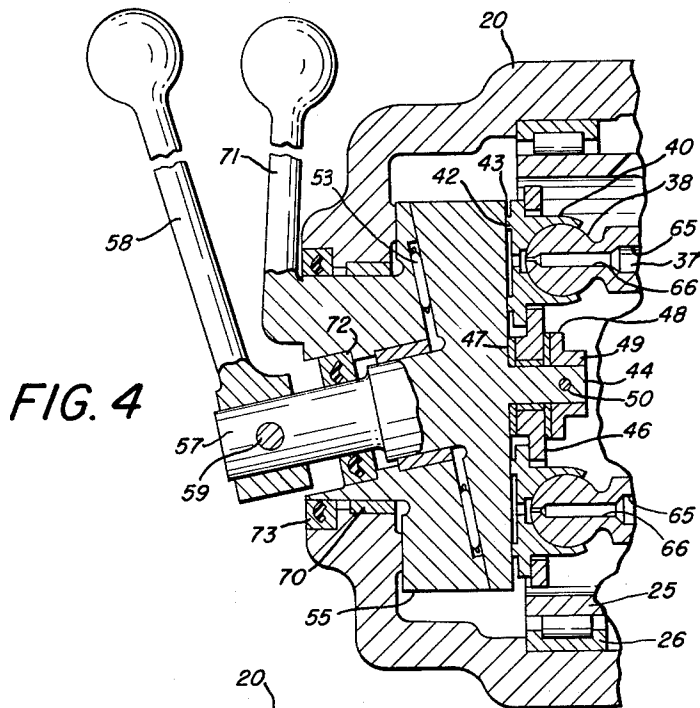
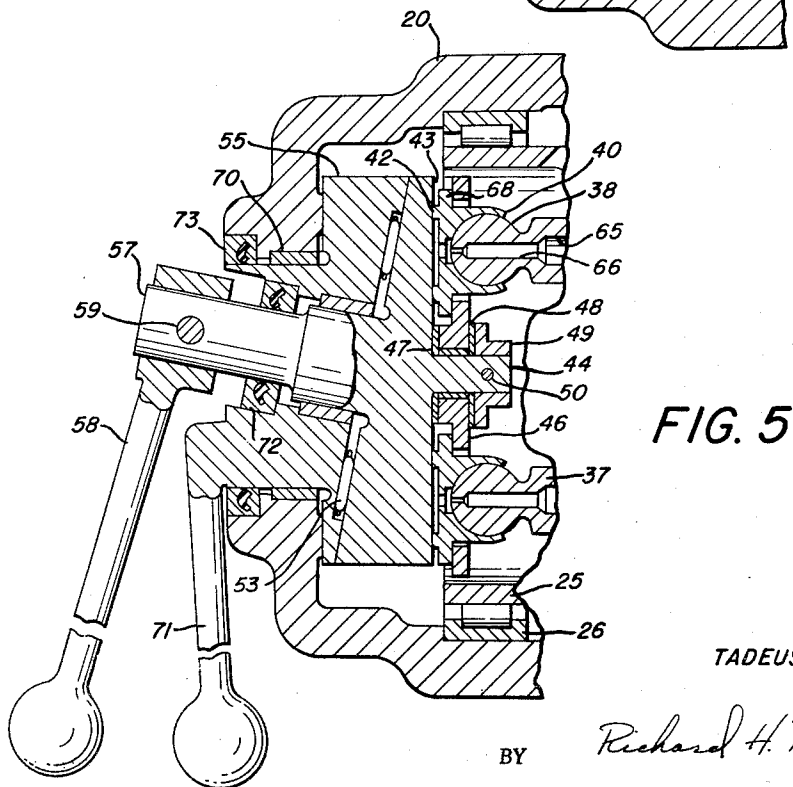
June 30, 1964

T. BUDZICH
HYDRAULIC APPARATUS

3,139,037

Filed Aug. 18, 1959

5 Sheets-Sheet 3



INVENTOR
TADEUSZ BUDZICH

BY

Richard H. MacCutchens

ATTORNEY

June 30, 1964

T. BUDZICH
HYDRAULIC APPARATUS

3,139,037

Filed Aug. 18, 1959

5 Sheets-Sheet 4

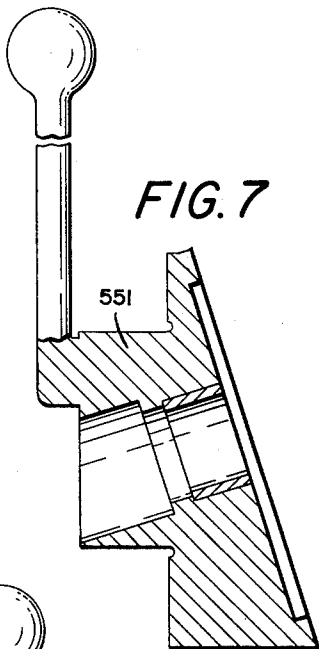


FIG. 7

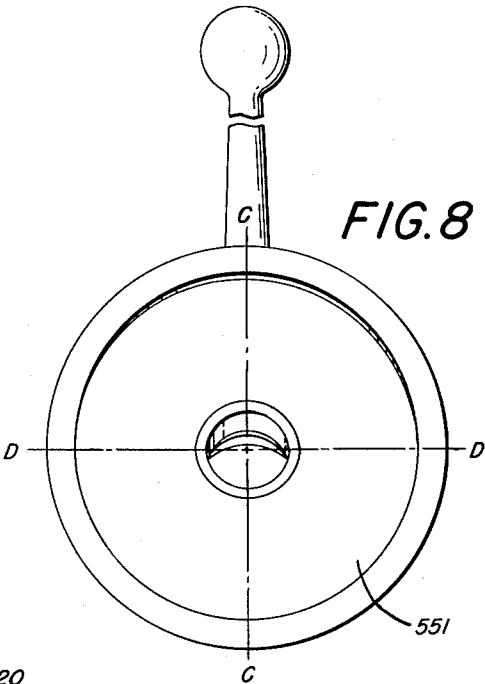


FIG. 8

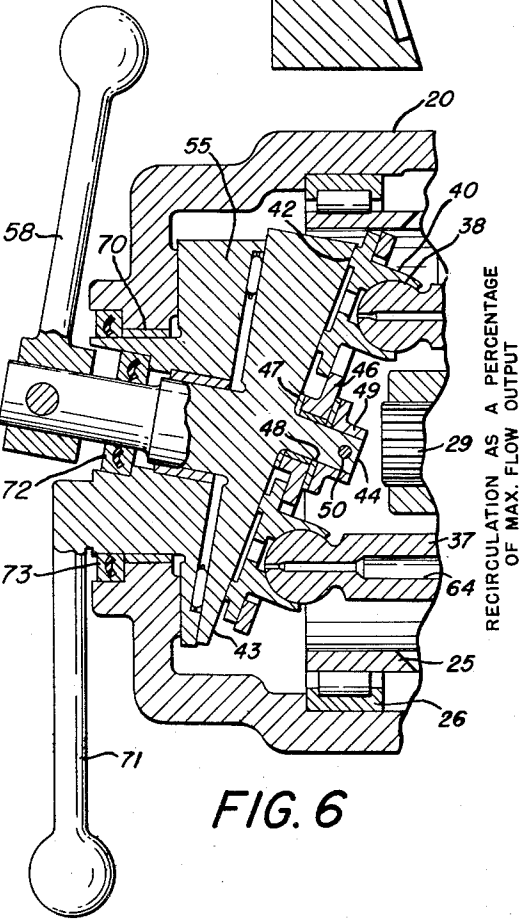


FIG. 6

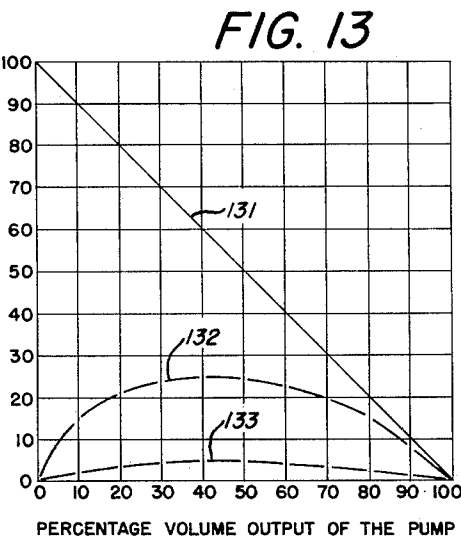


FIG. 13

INVENTOR
TADEUSZ BUDZICH

BY *Richard H. MacCutcherson*
ATTORNEY

June 30, 1964

T. BUDZICH
HYDRAULIC APPARATUS

3,139,037

Filed Aug. 18, 1959

5 Sheets-Sheet 5

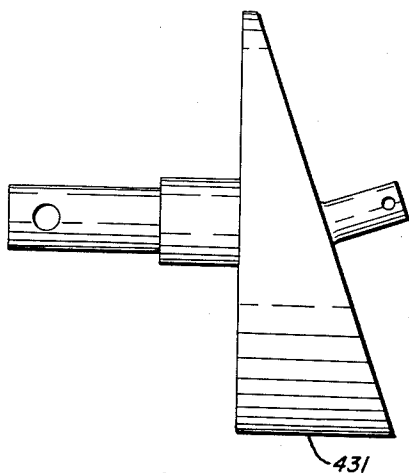


FIG. 9

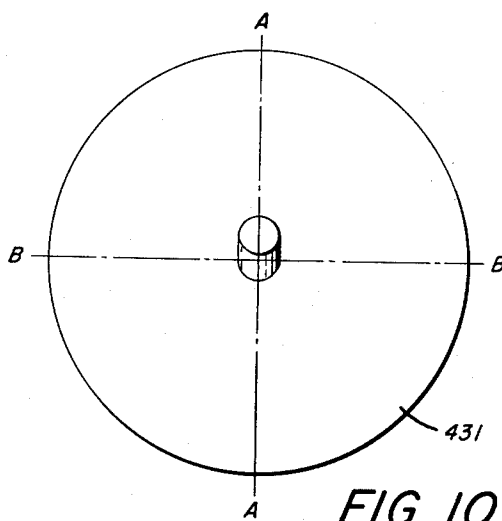


FIG. 10

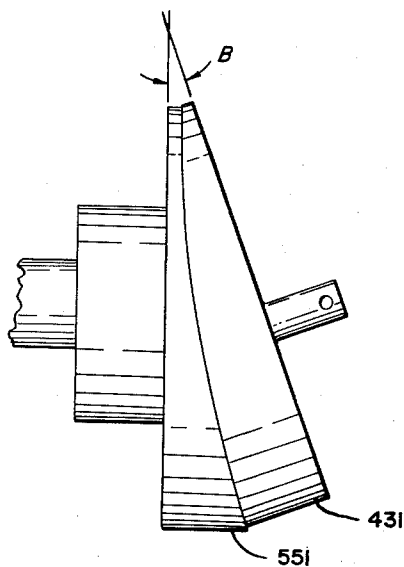


FIG. 11

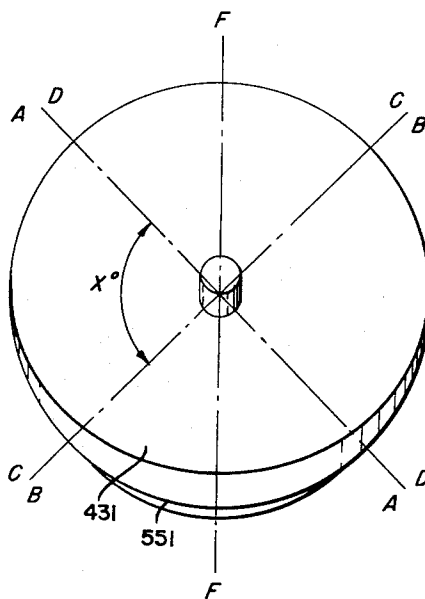


FIG. 12

INVENTOR
TADEUSZ BUDZICH

BY *Richard H. MacCuteh*

ATTORNEY

1

3,139,037

HYDRAULIC APPARATUS

Tadeusz Budzich, 3344 Colwyn Road, Cleveland 20, Ohio

Filed Aug. 18, 1959, Ser. No. 834,516

17 Claims. (Cl. 103-162)

This invention relates generally to hydraulic apparatus, and more particularly to fluid pressure, energy translating pumps and motors of the axial piston type in which a rotatable cylinder barrel carries pistons which are reciprocated by a swash plate device affecting motion of spherical joints associated with the pistons.

In more particular aspects the invention relates to variable output fluid pumps and motors of the rotatable cylinder barrel, axial cantilever piston type where the cylinder barrel is supported directly in the housing or on a shaft retained in the housing in such manner that the support of cylinder barrel is located on the intersection of the longitudinal axis of the cylinder barrel with a plane connecting the centers of the spherical joints associated with the pistons, so as to eliminate all transverse moments and permit free floating of the cylinder barrel and allow it to align itself against a flat stationary valve plate.

Whenever stroke changing mechanism is necessary for such pumps, according to a conventional solution the volume output of the pump is affected by changing the angle of inclination of the cam plate in relation to the axis of the pump, but heretofore the conventional suspension of cam plate has suffered from serious disadvantages associated with its expense, its complexity, and its bulk. The space required inside the housing for a complete sweep of cam plate is large making the housing big and heavy and this also results in complicated control linkages.

In another prior art type of construction the volume output of the pump is controlled by rotating a constant angle cam plate, relative to a valve plate, to render the pumping action of pistons ineffective during predetermined periods of rotation of cylinder barrel. While this has the minor advantage that the volume changing mechanism is small and inexpensive, it is very disadvantageous from the standpoint that the stroke of the pumping mechanism is the same irrespective of pump volume output, the same volume of fluid is circulated in the pump irrespective of pump discharge which gives a marked reduction in efficiency especially at low volume output, the locus of centers of spherical piston ends is caused to move away from the point of cylinder barrel support when changing the volume output of the pump which upsets the equilibrium of the cylinder barrel, and the pistons are caused at high velocity to pass the sealing lands of the valve plate producing noise and shock loading which compares very unfavorably with the swinging cam plate solution where the pistons pass the sealing lands of the valve plate only at zero reciprocal velocity.

It is an object of the present invention to provide simple and inexpensive means for overcoming the above mentioned difficulties.

Another object of the invention is to provide an improved solution of variable volume pump mechanism which retains the advantages of both types above mentioned while avoiding the majority of their inherent disadvantages.

Another object of the invention is to provide an improved stroke changing mechanism adapted to vary the output of a pump by changing the stroke of the piston assemblies without using the conventional type of swinging cam plate.

Another object of the invention is to provide a non-

2

swinging type of stroke changing mechanism in which at all cam angles the locus of centers of spherical piston ends remains stationary.

Another object is to provide an improved stroke changing mechanism in which rotation of a fixed angle cam will change the length of stroke of pistons.

Another object is to provide a stroke changing mechanism in which by rotation of a fixed angle cam, supporting the cam plate, the pumping mechanism can be brought into overcenter position to reverse the polarity of the valve plate ports.

Another object is to provide a stroke changing mechanism in which volume output of the pump can be varied and overcenter operation of the pump can be achieved while using less than 180° angle of rotation of the cam plate.

Other objects and advantages will become apparent and the invention may be better understood from consideration of the following description taken in connection with the accompanying drawings, in which:

FIG. 1 is a vertical cross sectional view through an axial piston pump (or motor) according to the invention and with a single fixed angle rotary cam plate shown in maximum stroke position;

FIG. 2 is a part section of the pump in the same plane as FIG. 1 but showing the single fixed angle rotary cam plate inclined at zero angle;

FIG. 3 is a part section view showing a modification in which an axial piston pump has two fixed angle rotary cam plates here shown in maximum stroke position;

FIG. 4 is a part section of a pump as in FIG. 3 but with the pump shown in zero stroke position;

FIG. 5 is a part section of the pump of FIG. 3 but showing the two fixed angle rotary cam plates inclined at zero angle but rotated to operate in the overcenter position;

FIG. 6 is a part section of the pump as in FIGS. 3-5 but with the fixed angle rotary cam plates shown in the maximum stroke overcenter position;

FIG. 7 is a section view through a modified fixed angle overcenter cam for 90° operation of the control as hereafter described;

FIG. 8 is an end view of the cam shown in FIG. 7;

FIG. 9 is a section view through a modified fixed angle cam plate for 90° operation of the control;

FIG. 10 is an end view of the plate shown in FIG. 8;

FIG. 11 is a view through the assembly of both fixed angle overcenter cam (of FIG. 8) and cam plate (of FIG. 9) useful for 90° operation of the control and shown in FIG. 11 with their principal axes rotated 90°;

FIG. 12 is an end view of the assembly shown in FIG. 11 and showing the relationship of principal axes of the two cams for 90° operation of the control;

FIG. 13 is a graph showing recirculation on the base of percentage volume output; and

FIG. 14 is an end view of the valve plate 22 of FIG. 1 which is according to prior art and which is shown merely for the purpose of explanation.

Description

Referring to the drawings, the pump or motor comprises a housing having separable sections including a pump body 20 and a pump cover 21 suitably connected by bolts (not shown). A valve body 22 having ports (not shown) has its outer periphery in contact with pump body 20 and a flat face engaging a corresponding flat face of pump cover 21. The valve body 22 has a face in operational contact with a cylinder barrel 24 at one end of the cylinder barrel. The cylinder barrel 24 is supported in a cylinder block sleeve 25 which at the opposite end is held in a roller bearing 26 suitably located

in the pump body 20 while a coupling 28 with splined ends 29 and 30 acts as a universal joint between cylinder barrel 24 and a drive shaft 32. The drive shaft 32 is mounted for rotation in two anti-friction bearings 34 and 35 supported within the pump cover 21.

Cylinder block 24 is provided with cylinder bores 36 and pistons 37 work in operational contact with these cylinder bores. Each piston 37 terminates in a spherical end 38. A plurality of shoes 40 each closes over one of the spherical ends 38 of the pistons 37 while having opposite faces forming balancing lands 42 which rest against the flat face of a cam plate 43. On the same side the cam plate 43 is provided with a cylindrical protruding part 44, and a nutating plate 46 is arranged radially around the cylindrical protruding part and is axially retained by bearings 47, 48 and a thrust sleeve 49 secured by a pin 50. This arrangement keeps the piston sphere engaging shoes 40 in contact with the cam plate 43 which is radially located by a bearing 52 and rests against a suitable thrust bearing 53. Bearing 52 is mounted with respect to the end of the housing 20, and bearing 53 is located in a fixed cam 54 retained radially on a boss 56 of housing 20 and suitably prevented from rotation. A control shaft 57 is associated with cam plate 43 and supports a control handle 58 secured by a pin 59.

In understanding operation of the embodiments shown in FIGS. 1 and 2, it should be understood that the engine may function either as a pump or as a motor although its operation will be described as though it were acting as a pump. Accordingly, the drive shaft 32 would be connected to a suitable prime mover (not shown). The inlet and outlet ports of the pump (likewise not shown but assumed emanating from the valve body 22 in conventional manner) would be connected to a hydraulic system. The drive shaft 32 revolving in the bearings 34, 35 will induce rotation in the splined universal coupling 28 and cylinder barrel 24. In known manner the cylinder barrel, while revolving, will sequentially register with inlet and discharge ports of the pump and the cam plate 43 will cause the pistons to reciprocate in timed relation with this sequential registration and thereby will produce a pumping action. Proper sealing between the mating faces of cylinder barrel 24 and valve body 22 is affected by hydraulic pressure acting within the cylinder bores which may be augmented by the preload force in a spring 60 interposed between a portion of the shaft 32 and a portion of the cylinder barrel 24 as shown. The free floating cylinder barrel 24 is free to slide in a longitudinal direction and align itself to the valve body and therefore those forces acting on it will produce the effect of rapidly closing any gap existing between the mating members. Although these expedients are well known it can be further explained that the spring 60 anchored against a seat provided in the drive shaft 32 reacts against the cylinder barrel 24 to keep it in contact with the valve body 22 during starting and very low discharge pressures, while the cylinder barrel 24 contains a plurality of circumferentially spaced leakage grooves 61 defining dynamic pads 62 so that the flat face of cylinder barrel 24 mating with valve body 22 is hydrostatically balanced in such manner that the actual contact pressure existing between the two parts is caused by only a small fraction of the hydraulic reaction forces. Also in known manner progressively smaller longitudinal passages 64, 65, 66 extend through each piston and connect with passages provided in the piston shoes 40 to effect hydrostatic balance and distribution of hydraulic forces acting on the flat face of the piston shoe, so that during the discharge stroke there exists a small resultant force keeping the piston assembly against the flat face of the cam plate, while during the suction stroke the piston assembly is maintained in contact with the cam plate by the nutating plate 46 which has suitable openings for engaging an annular shoulder 68 provided on each piston shoe 40.

The nutating plate revolves with the piston assemblies while pivoted on the cylindrical portion 44 provided on the cam plate 43. During the suction stroke the inertia forces needed to accelerate the mass of the piston assemblies, to keep them in contact with the cam plate 43, are supplied by the nutating plate, the thrust washers 47 and 48, and the flange 49 retaining it and providing a fulcrum.

As a result of the angular inclination of the cam plate 43, a radial force, acting through the center of each piston spherical end 38, is transferred to the cylinder barrel 24. The sum of these radial forces, from all the pistons subjected to pressure, is transmitted from the cylinder barrel to the cylinder block sleeve 25 to the bearing 26 and the pump body 20. For maximum efficiency of the pump the cylinder barrel 24 must be capable of aligning itself to the valve plate 22, irrespective of the change in angularity of the pump cover 21 due to deflection of the housing. To insure this freedom of alignment there is a clearance drive at 28, a clearance bearing at 26 and a geometrical arrangement of spherical piston ends and centers as hereafter described. Thus, in more detail, the torque is supplied to the cylinder barrel 24 from the shaft 32 by a splined universal coupling 28, with clearance in both spline joints so selected that the resulting universal action permits freedom for alignment of the cylinder barrel. The flanged end of drive shaft 32, working in operational contact with spring 60, angularly locates cylinder barrel 24 in such a way that a predetermined radial clearance, necessary for maintenance of squareness of timing surface is provided. A sufficient clearance is provided in roller bearing 26 to permit requisite angular freedom of cylinder barrel 24. Meanwhile, a combined transverse couple transmitted to the cylinder barrel 24 and caused by the side loading of the piston assemblies is eliminated by positioning the plane passing through the centers of the rollers of bearing 26 at a point at which a plane through the centers of the spherical piston ends 38 bisects the center line of the pump. This geometrical relationship is preserved at all inclinations of the cam plate (and thus regardless of volume output of the pump) as is important in maintaining of efficient operation of the pump.

Heretofore the free floating feature of a cylinder block at all cam angles has been achieved by swinging a cam plate around an axis perpendicular to the pump axis and heretofore the axis of motion of the swinging cam plate has passed through the center of suspension of the cylinder block at a point in a plane connecting the spherical ends of pistons. This prior arrangement is not only expensive and bulky but has other undesirable design limitations. But with the arrangement of the present invention the stroke changing mechanism is substantially simplified while the advantages of the swinging cam plate are still maintained and the stroke of the piston may be effectively varied while at the same time a strict position relationship is maintained between the piston assemblies and the point of support of the rotating cylinder barrel.

According to FIGS. 1 and 2 the cam plate 43 is retained by the radial bearing 52 and works against the thrust bearing 53. These two bearings supporting the cam plate 43 are positioned at an angle to the center line of the pump through the medium of a stationary cam plate 54 and the annular flange 56 provided in the pump body 20. The effective angle of inclination of the cam plate assembly is then the sum of angular inclination of cam plate 43 and stationary cam plate 54 with the principal axis of both inclined planes coinciding. Principal axis of an inclined cam face (e.g., a front face) can be defined as the axis along such face and passing through the center of rotation and inclined at the maximum angle (e.g. to the "back" face of the cam, or if a composite assembly is involved, the back face of the assembly perpendicular to the center line of the pump).

As shown in FIG. 1 the cam plate assembly is inclined

5

at its maximum possible angle with the principal axes of both inclined planes coinciding and with the pump working at its maximum stroke. The rotation of handle 58 will effect the angular relationship between the principal axis of both inclined planes, effectively changing the resultant angle of inclination of the cam plate assembly until with 180° of rotation the zero angle and zero stroke position is reached as shown in FIG. 2. The nutating plate assembly and therefore the piston assemblies constrained by it are made to follow this angular change. The geometry of the rotating cam plate is so arranged that center line of rotation of cam plate intersects the longitudinal axis of the pump. In this manner the point of intersection between a plane connecting the centers of the spherical piston ends and the longitudinal axis of the pump may remain stationary during rotation of the cam plate 43. Thus change in piston stroke (and pump output) will not effect the free floating cylinder block feature.

In FIG. 3 like parts are like numbered as in FIG. 1. The fixed angle cam plate 43 carries the same nutating plate assembly as before and is radially located by bearing 52 and axially located by bearing 53 but in this case the stationary cam (54 of FIG. 1) and the inclined face housing boss (56 of FIG. 1) are replaced by an adjustable fixed angle cam 55 which operates as an overcenter cam. This revolving cam 55 in turn is located radially by a bearing 70 and rests against the pump body 20. A control handle 71 is connected to the overcenter cam 55 while the control handle 58 is connected to the shaft 57 of cam 43. The double cam assembly is suitably sealed by shaft seals 72 and 73.

In FIG. 3 the stroke changing mechanism has provided the same adjustment as in FIG. 1 but is capable of operating in overcenter position. With the overcenter control handle 71 in the vertical position as shown in FIG. 3 the operation will be the same as with the arrangement of FIG. 1. 180° rotation of the cam plate 43 (handle 58) will bring the pump into its zero stroke unloaded position (see FIG. 4). Since no fluid is pumped the discharge pressure drops, unloading the bearings and relieving the contact pressure between cam assemblies and pump body. The same zero stroke effect is achieved if only the other handle (71) is rotated 180° (from the position shown in FIG. 3) so that the handles and cams take the position shown in FIG. 5 but in this case the timing mechanism of the pump is in effect reversed because rotation of handle 58 and associate cam 43 (from the position of FIG. 4) will reverse the polarity of the ports (as soon as there is any movement away from zero stroke position) putting the pump into the so-called overcenter position.

Rotation of the cam assemblies may be affected by the respective control handles when the pump is unloaded, or, with adequate bearings between the parts, under load.

One overcenter position is shown in FIG. 6 and those in the art will understand that if the device was previously (according to FIG. 3) acting as a pump the former suction port will become a discharge port and vice versa, reversing the direction of flow through the pump (while increments of the 180° rotation of the control handle 58 will vary the output of the pump from zero to maximum), or if the device was previously (according to FIG. 3) acting as a motor reversal of both handles to bring the cams into the overcenter position of FIG. 6 will cause it to reverse its direction of rotation.

Thus FIGS. 1 and 2 demonstrate a single cam solution, and FIGS. 3-5 a double cam solution both predicated on 180° cam rotation control, but the 180° angle of rotation (to change output from zero to maximum) was selected only for ease of graphical representation. Actually the angle of rotation (to change output from zero to maximum) can be substantially reduced up to or even below 90°. Such a reduction provides advantages from the standpoint of both performance and ease of operation.

6

FIGS. 7, 8 and 9 show the construction of overcenter and rotating cam plates for 90° control operation. FIGS. 11 and 12 show the assembly of these two components with the principal axis of the cam 431 marked AA and the principal axis of cam 551 marked CC. The line FF denotes the vertical axis of the pump dividing the timing mechanism into suction and discharge sectors.

The reduction in angle of rotation of the control is achieved by using for the fixed angles of component cams more than half of the required angle of inclination of the component cam assembly. Such steep angle cams (431, 551) as illustrated in FIGS. 7-12 are then positioned in the pump body with their principal axes displaced by an angle at which the angular inclination of the resulting plane gives the maximum required stroke. Thus as illustrated in FIGS. 11 and 12, the cam 551 from FIG. 7 with its principal axis CC (FIG. 8) and the cam 43 from FIG. 9 having a principal axis AA (FIG. 10) are located with these axes oppositely inclined to the vertical axis of the pump as illustrated in FIG. 12. As illustrated in many well known patents and in FIG. 14 the vertical axis divides the valve plate into suction and discharge sectors. With the arrangement of FIG. 12 the resultant angle of inclination along the line FF, which angle is illustrated at "B" in FIG. 11, is less than the sum of the angles of inclination of the principal axes of the component cams. Thus the arc of rotation of the control is reduced to a rotation equal to the angle of 180° minus the angle equal to the angular displacement of the principal axis of cam plate 431 with respect to that of overcenter cam 551 (i.e., angle "X" of FIG. 12) for maximum pumping stroke. Since the magnitude of this angular displacement depends on an angle of inclination of the fixed cams, the bigger this angle of inclination over half of maximum required angularity of the combined cam assembly, the smaller the angular travel of the control. As already mentioned, the reduction in angle of rotation of the control offers substantial advantages.

With the above arrangement the principal axis of the resultant plane coincides with the vertical axis of the pump (FF) and provides the position of maximum pumping stroke. Movement of control handle 58 equal to X° will bring the principal axis AA of the rotating cam plate 431 over the principal axis CC of the overcenter cam 551, bringing the stroking mechanism into zero angle position. In this position, with the pump unloaded, the overcenter cam 551 can be rotated by angle X while still maintaining the unloaded condition of the pump (by rotating the other cam) and bringing the control into the overcenter sector, where axis CC takes the original position of axis AA. From this position further rotation of the cam plate (i.e., of AA) will reverse polarity of the ports and reverse the direction of the flow through the pump. With full rotation of 90° the position of maximum overcenter pump output will be reached.

As already mentioned the prior art provides a rotating cam with its axis of rotation coinciding with the longitudinal axis of the pump. This arrangement, though changing effective volume output, retains full stroke of the pump. Therefore a constant volume of oil is circulated through the pump with the associated losses. The advantages of the present invention are clearly demonstrated in FIG. 13 in which recirculation, as a percentage of maximum volume output, is plotted on the base of percentage volume output of the pump. As previously noted, one conventional solution gives 100% of recirculation at zero percentage output of the pump as is represented by line 131 of the graph. The curve 132 shows the recirculation for apparatus according to the present invention and having 180° movement of the control as shown in FIGS. 1-6. The advantages (at smaller percentages of volume output of the pump) are evident. But such 180° apparatus (as shown by curve 132) was adopted only because of the ease of graphical

representation. The recirculation losses with 90° rotation of control for stroke changing mechanism are demonstrated by curve 133. It must be noted that the recirculation losses with this arrangement are negligible, being less than 5%.

As already indicated in connection with FIG. 1, for maximum stroke the principal axis of fixed cam 54, and of revolving cam plate 43, and of the resultant plane, fall in a plane through the vertical axis of the pump. The net displacement of each piston, measured by resultant axis inclination from vertical between top and bottom, will give the actual displacement of the pump. The rotation of the principal axis of the cam plate 43 will induce a rotation in the principal axis of the resultant plane, its angular displacement from the vertical axis of the pump being equal to half of the angle travelled by cam plate 43 (assuming the two cams have equal inclination). Therefore the effective pumping stroke of the mechanism will no longer be the maximum stroke. With 180° rotation of the principal axis of the cam plate 43 the principal axis of the resultant plane will travel 90°, and both of those axes will be inclined at zero angle. The geometry of this stroke changing mechanism can be readily changed so that the difference between the maximum stroke of the mechanism and the effective stroke is small, as shown by FIG. 13, or zero as described hereafter. The difference between the maximum stroke and effective pumping stroke is proportional to excess of oil circulation induced in the pumping mechanism when some volume of the fluid circulated through the pump is rejected (e.g., into the suction port) without any pumping action. Referring to FIG. 13 the advantages of the 90° control arc in reducing such parasite circulation are self evident.

But as above described the principal axis of the resultant plane moves away from the vertical axis dividing the valve plate into the suction and discharge sectors, causing some unnecessary recirculation. This recirculation can be stopped, when changing the piston stroke, by moving both component cams simultaneously one in a clockwise and the other in a counterclockwise direction. In this way the principal axis of the resultant plane will remain stationary, and the effective pumping stroke may at all times remain the maximum stroke of the pump. To maintain this condition the rotation of both cams must be synchronized, the angular displacement from vertical axis of one cam will then correspond to the same angular displacement of the other cam from this axis but in the opposite direction. Control means of conventional design can be utilized to synchronize both cams. With such an arrangement to change the volume output of the pump from maximum to zero only half of the arc of rotation of each cam is required, the relative rotation between both cams and the angular displacement of their principal axes remaining the same as in the case of single cam plate operation though each cam went only half as far. With both cams synchronized and rotating, once the position of zero stroke is reached further rotation of the cams will reverse the direction of fluid flow through the pump, the volume output gradually increasing with rotation from zero to maximum in the overcenter position. During this operation, when using synchronized cams, the principal axis of the resultant plane will always coincide with the vertical axis dividing the valve plate into suction and discharge sectors, and the circulating losses are zero.

While I have illustrated and described a particular embodiment, various modifications may obviously be made without departing from the true spirit and scope of the invention which I intend to have defined only by the appended claims taken with all reasonable equivalents.

I claim:

1. An energy translating fluid pressure device comprising, a housing, a cylinder barrel, means rotatively supporting said cylinder barrel in said housing, said cylinder barrel having a plurality of cylinder bores, a piston disposed in each of said bores and reciprocal therein, valving

means including ports disposed in said housing and positioned to sequentially register with said cylinder bores, said device having a principal axis about which the cylinder barrel rotates, support means disposed in said housing, said support means including a surface inclined with respect to a plane normal to the principal axis and intersecting said axis, a primary cam plate having a first face parallel to and supported by said surface of the support means, said primary cam plate having a second face operatively engaging said pistons, said second face being inclined with respect to said first face, means to rotate said primary cam plate about an axis inclined with respect to the pump axis from a first position wherein there is a minimum cam angle and the length of the piston stroke is a minimum to a second position wherein the cam angle is a maximum and the length of the piston stroke is a maximum, the location on said primary cam plate defining the point of maximum piston displacement being moveable angularly with respect to the valve ports with the rotational movement of the cam plate from its first position to its second position, said cam plate having at least one additional operative position intermediate its first and second positions, whereby rotation of the primary cam plate will change both the length of the piston stroke and the angular position of the high and low points of the cam plate with respect to the ports.

2. An energy translating device as in claim 1 further characterized by the second support means being fixed with respect to the housing.

3. The combination of claim 1 characterized by the surface of the support means having its principal axis falling on a plane which contains an axis dividing the valving means into intake and discharge ports.

4. The device of claim 1 further characterized by said second support means being itself a support cam plate, and means to rotate said second cam plate with respect to said housing.

5. The device of claim 1 further characterized by said axis of rotation of the primary cam plate intersecting the principal axis.

6. The combination of claim 1 further characterized by said pistons including part spherical ends, and piston shoes closing over said spherical ends, and the means rotatively supporting the cylinder barrel including a bearing the center of which coincides with the intersection of the principal axis with a plane passing through the centers of the part spherical ends of the pistons.

7. The device of claim 6 further characterized by a nutating plate arranged to axially restrain said piston shoes while permitting some radial movement thereof with respect to said principal cam plate.

8. The device of claim 7 further characterized by thrust means associated with said primary cam plate and said nutating plate adapted to maintain the shoes against said cam plate.

9. The combination of claim 1 further characterized by said cam plate having an infinite number of operative positions intermediate its first and second positions.

10. An energy translating fluid pressure device, comprising, a housing, a cylinder barrel rotatively disposed in said housing, said cylinder barrel having a plurality of cylinder bores, a piston disposed in each of said bores and reciprocal therein, valving means including ports disposed in said housing and positioned to sequentially register with said cylinder bores, said device having a principal axis about which the cylinder barrel rotates, a support cam disposed in said housing, said support cam having a surface inclined with respect to a plane normal to the principal axis and intersecting said axis, a primary cam plate, said primary cam plate having a first face parallel to and supported by said surface of the support cam, said primary cam plate having a second face operatively engaging said pistons, said second face being inclined with respect to said first face, said primary cam plate and said support cam each being rotatable with respect to

each other, the axis of rotation of said primary cam plate being inclined with respect to the principal axis, whereby rotation of the primary cam plate and support cam equally in opposite directions will change the angle of the second face of the primary cam plate with respect to the principal axis while maintaining the high point thereof unchanged angularly with respect to the ports thereby minimizing recirculation losses.

11. In a fluid pressure device as in claim 10 a configuration and arrangement of cams, control means therefor and housing such that the first mentioned cam may be rotated for varying the resultant angle of inclination of the cam assembly and thus the piston stroke from zero to maximum while the second mentioned cam may be rotated for reversing the direction of fluid pressure through the device.

12. In a fluid pressure device as in claim 10, first and second rotatable cams as provided in said claim further characterized by the angle of inclination of each of these component cams being greater than half the maximum required angle of the resultant plane.

13. In a fluid pressure device as in claim 10, first and second cams as provided in said claim further characterized by the first cam having its principal axis rotated to be disposed at an angle to the principal axis of the second cam when providing maximum required resultant plane angle of inclination for providing maximum stroke.

14. In a fluid pressure device as in claim 10, the valve structure comprising a plate having kidney shaped suction and discharge ports with a vertical axis bisecting the lands therebetween, and the first and second rotatable cams providing a resultant plane for affecting the stroke of the pistons with the principal axis of said resultant plane coinciding with a longitudinal plane through the vertical axis dividing the valve plate into suction and discharge sectors when said resultant plane is inclined at maximum angle equivalent to maximum stroke of the device.

15. In a fluid pressure device as in claim 10, first and second rotatable cams as provided in said claim and dual control means one for each cam with the control means for the first mentioned cam characterized by rotation through a control angle less than 180° to change the piston displacement from maximum to zero.

16. In a fluid pressure device as in claim 10, first and second rotatable cams as provided in said claim and dual control means one for each cam with the control means for the second mentioned cam characterized by rotation through a control angle less than 180° to reverse the direction of fluid pressure action.

17. In a fluid pressure device as in claim 10, first and second rotatable cams as provided in said claim, and control means for simultaneously moving the two cams in opposite directions but each the same angular distance while preserving the principal axis of the resultant plane in a longitudinal plane through the vertical axis of the device to obviate recirculation losses while varying piston displacement from maximum to zero.

References Cited in the file of this patent

UNITED STATES PATENTS

1,819,715	Le Bret	Aug. 18, 1931
1,908,612	Johnson	May 9, 1933
2,407,013	Infield	Sept. 3, 1946
2,513,758	Talbot	July 4, 1950
2,565,582	White	Aug. 28, 1951
2,737,899	Bonnette et al.	Mar. 13, 1956
2,753,802	Omohundro	July 10, 1956
2,847,938	Gondek	Aug. 19, 1958
2,926,032	Cook	Feb. 23, 1960
3,010,339	Brock	Nov. 28, 1961

FOREIGN PATENTS

977,328	France	Nov. 8, 1950
---------	--------	--------------