June 27, 1967 T. BUDZICH 3,327,642

FLUID APPARATUS

Filed Feb. 11, 1965 2 Sheets-Sheet 1

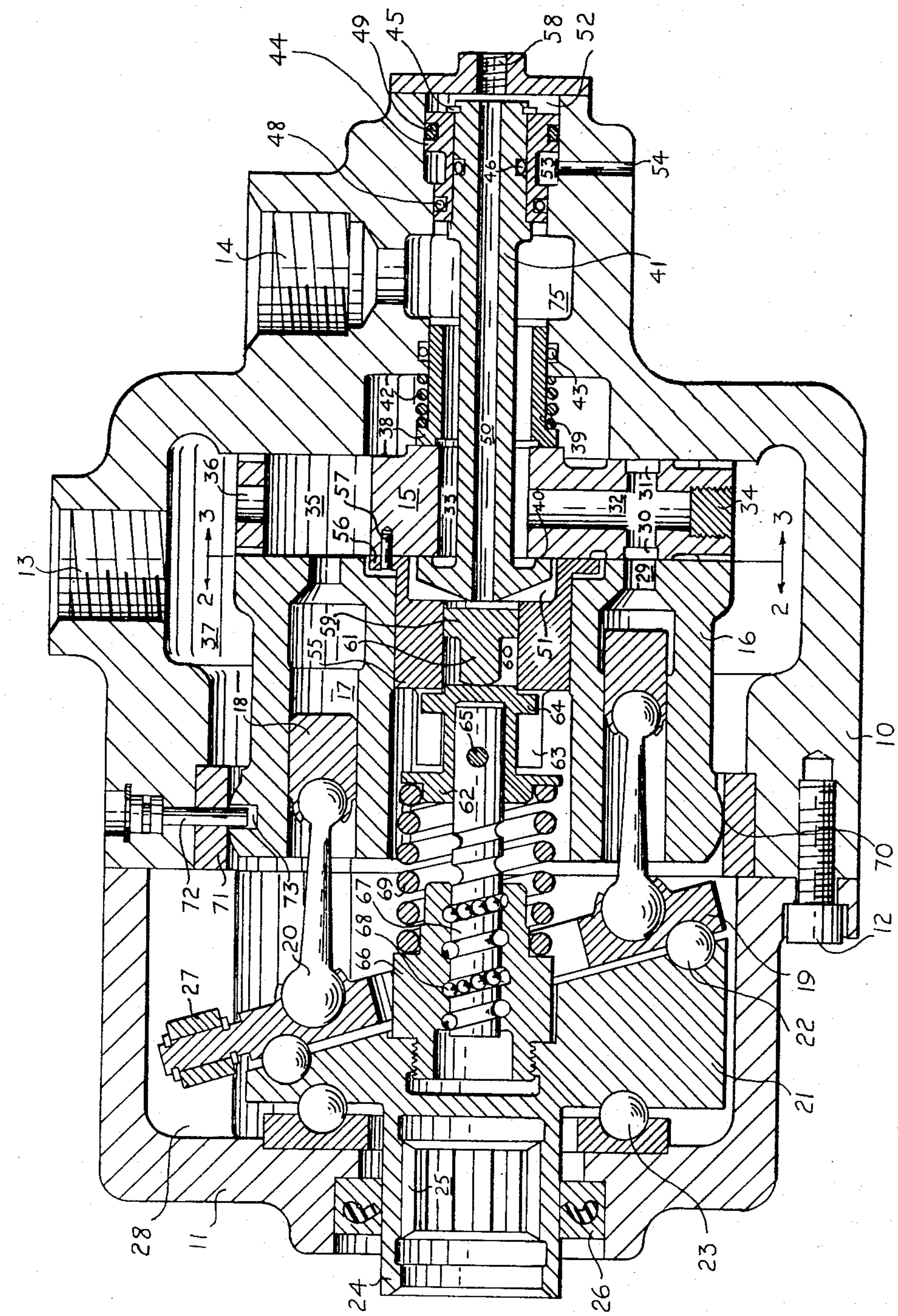

FIG. 1

INVENTOR.

TADEUSZ BUDZICH

BY

McNENNY, FARRINGTON, PEARNE & GORDON

R. H. Dickinson Jr.

ATTORNEYS

June 27, 1967 T. BUDZICH 3,327,642

FLUID APPARATUS

Filed Feb. 11, 1965 2 Sheets-Sheet 2

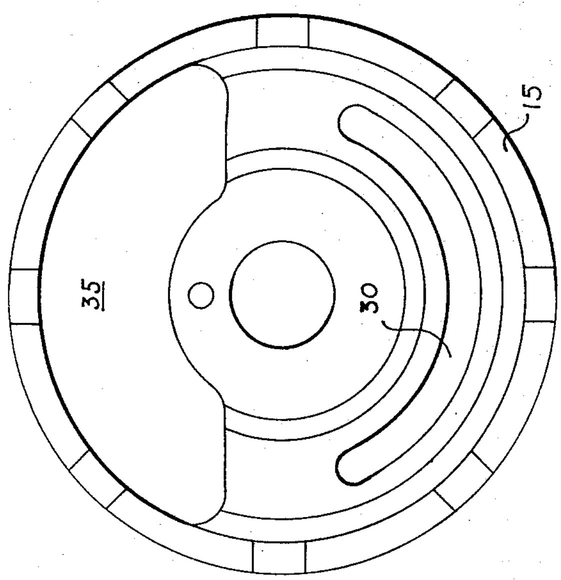

FIG. 3

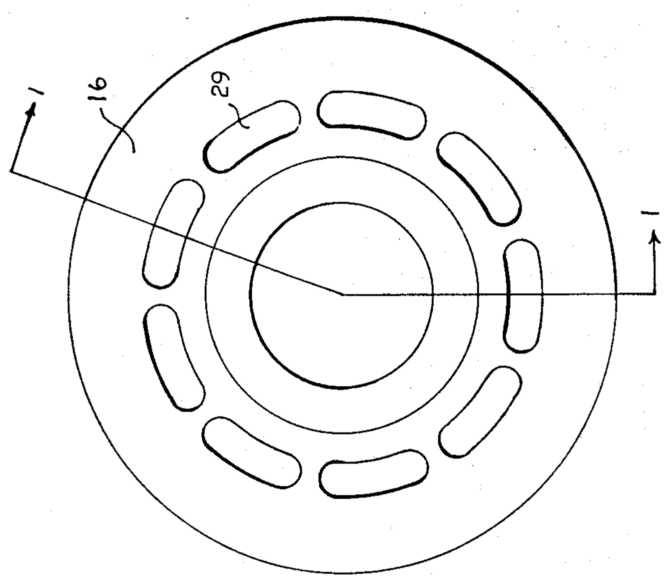

FIG. 2

INVENTOR.
TADEUSZ BUDZICH
BY
McNENNY, FARRINGTON, PEARNE & GORDON
R. H. Dickinson Jr.
ATTORNEYS

United States Patent Office 3,327,642
Patented June 27, 1967

3,327,642
FLUID APPARATUS

Tadeusz Budzich, 80 Murwood Drive,
Moreland Hills, Ohio 44022

Filed Feb. 11, 1965, Ser. No. 431,896
3 Claims. (Cl. 103—173)

The present invention relates generally to fluid pressure energy translating devices of the piston pump or motor type, and mort particularly to fluid piston pumps and motors employing a stationary cylinder barrel and a rotary cam plate driving a rotary valve plate which sequentially conducts fluid to and from multiple cylinders in the stationary cylinder barrel.

A principal object of this invention is to provide a novel variable displacement axial piston pump or motor in which the effective displacement of the pump or motor is altered by changing the angular relationship between the rotary valve plate and the rotary cam plate.

Another object of this invention is to provide in a variable displacement axial piston pump or motor a novel mechanism permitting the change in angular relationship between the cam plate and the valve plate while maintaining the valve in a state of hydrostatic balance.

Another object of this invention is to provide a novel variable displacement axial piston pump or motor of the type set forth in the preceding objects in which the angular relationship between the rotary valve plate and the rotary cam plate may be varied by linear movement of a control member along the axis of rotation of these members.

Another object of this invention is to provide a novel variable displacement axial piston pump or motor as set forth in the preceding object in which the movement of the control member may be controlled by the use of an expansible chamber fluid motor.

Still another object of this invention is to provide a novel variable displacement pump or motor of the type set forth in the preceding objects in which the angular relationship between the rotary valve plate and the rotary cam plates is effected by means of a ball screw mechanism interconnecting the valve plate and cam plate so that by linear actuation of the screw, the resulting rotation of the nut will produce the change in angular relationship.

Certain inherent advantages of the axial piston pump and motor employing a stationary cylinder barrel and revolving valve plate have been recognized for some time. These advantages include the absence of the disturbing centrifugal couple trying to upseat the cylinder barrel from the valve plate and reduction in throttling losses in the passages of the timing mechanism.

Heretofore, a particular disadvantage of the stationary cylinder barrel has been that it is very difficult to produce a satisfactory solution of the stroke changing mechanism. The present invention provides a simple, cheap, accurate and reliable method of changing the angular relationship between the cam plate and the valve plate, in a gradual fashion, thus obtaining perfect control over displacement of the unit. This mechanism, working at minimum friction level, may be used in a control of a pressure compensated pump, where the volume output must change proportionally to discharge pressure. In addition, this mechanism besides providing a simple and reliable way of changing the angular relationship between the cam plate and valve plate permits the change in direction of rotation with unit acting as a motor.

As it is well known in the art with the principal axis of the cam plate coinciding with the line dividing the valve plate into low pressure and high pressure zones, the pump or motor is in its full displacement, handling the maximum volume of hydraulic fluid. An angular rotation between the principal cam axis and the valve plate

will gradually decrease the displacement of the unit, until the volume output becomes zero once 90° relationship is reached. Continued rotation of the axis past 90° will gradually start increasing the displacement of the unit from zero to maximum reversing the direction of rotation of the unit.

Further objects and advantages of the invention will become apparent from a study of the following specification when taken in conjunction with accompanying drawings in which:

FIGURE 1 is a sectional elevation of a motor according to the preferred embodiment of this invention.

FIGURE 2 is the end view of the cylinder barrel taken along the line 2—2 and showing the detail of the cylinder barrel porting.

FIGURE 3 is an outside view of the valve plate taken along the line 3—3 and showing the port details.

The embodiment shown in the drawings will be described as a motor, although it can work equally well as a pump. As shown in FIGURE 1 a motor housing **10** and a motor cover **11** are suitably bolted together with bolts **12**. The motor housing **10** is equipped with a low pressure port **13** and a high pressure port **14** which are connected by suitable means to drain and to a source of high pressure hydraulic fluid. The flat inner end bore of the motor body **10** provides a working surface for a rotary valve plate **15**. The rotary valve plate **15** works between this flat face and the end face of a cylinder barrel **16**. The cylinder barrel **16** is equipped with a number of cylinder bores **17** in which pistons **18** are arranged for reciprocation. The reciprocating motion is transmitted to pistons **18** from a reaction plate **19**, by articulated connecting rods **20**, suitably retained in the reaction plate **19** and pistons **18**. A cam plate **21**, working in operational contact with the reaction plate **19**, through an antifriction bearing **22**, is located in the motor cover **11** by another antifriction bearing **23**. A cylindrical extension **24**, of the cam plate **21**, is equipped with splines **25** and sealed by a conventional type seal **26**. A radially extending guide bearing **27**, free to slide in a longitudinal slot **28**, permits the wobbling motion of the reaction plate **19**, while maintaining the angular relationship between the articulated connecting rods **20** and pistons **18**. The cylinder bores **17** of the cylinder barrel **16** are equipped with kidney shaped ports **29** (see FIGURE 2) which work in operational connection with high pressure inlet kidney slots **30**, provided in the valve plate **15** (see FIGURE 3). The rotary valve plate **15** is equipped with identical kidney shaped inlet slots **30** and **31**, which are connected through a drilled radial passage **32**, with space **33**, located in the center of the valve plate **15**. The drilled passage **32**, at the outer end is suitably closed by a plug **34**. The cylinder bores **17**, of the cylinder barrel **16**, on the low pressure stroke are in direct communication with open slot **35** which through passages **36** is connected to the chamber **37** inside the motor housing **10**. The chamber **37** is in direct communication with the low pressure or drain port **13**. The rotary valve plate **15** on the side away from cylinder barrel **16** works in operational contact with end **38** of a transfer sleeve **39**. On the other side the valve plate **15** works in operational contact with sealing face **40** of a hollow balancing member **41**. A spring **42** is located between the motor housing **10** and the transfer sleeve **39**. The transfer sleeve **39** is suitably sealed by an O-ring **43**. The hollow balancing member **41** is connected to a balancing piston **44** by a snap ring **45** and sealed by O-ring **46**. The balancing piston **44** is equipped with two sealing O-rings **48** and **49**. A longitudinal passage **50** in member **41** connects inner chamber **51** with outer chamber **52**. Space **53** is enclosed between the stepped diameters of balancing piston **44** and is vented to atmosphere by suitable passage **54**. A control cylinder member **55** is non-rotatably connected through flange **56** to the rotary valve plate **15** by dowel pins **57** and suitable bolts (not shown). The circular extension of the control cylinder member **55** encloses inner chamber **51** connected through passage **50** with outer chamber **52** and control port **53**. The outer surface of the control cylinder member **55** works in operational contact with the inside bore of cylinder barrel **16** and locates it with respect to rotary valve plate **15**. A control piston **59** works in bore **60** of the control cylinder **55** and with its cylindrical extension **61** rests against flat face of a control spring guide **62**. Slots **63**, machined in the end of the control cylinder **55**, slidably receive projecting ears **64** on the control spring guide **62**, which in turn is fastened by a pin **65** to the control ball screw **67**. The control ball screw **67** engages, through a series of balls **68**, control ball nut **66**, permanently retained in the cam plate **21**. A compression spring **69** is located between the control ball nut **66** and control spring guide **62**. The cylinder barrel **16** with its spherical surface **70** engages the ring **71** and is secured against rotation by pin **72** engaging an oval slot **73**. The oval slot **73** permits the cylinder barrel **16** to align itself to the rotary valve plate **15** while preventing it from rotating.

The housing **10** with its ring **71** engages the spherical surface **70** of the cylinder barrel **16**. The cylinder barrel **16**, equipped with kidney shaped ports **29**, works in operational contact with rotary valve plate **15**. As rotary valve plate **15** revolves between the flat faces of cylinder barrel **16** and housing **10**, the kidney shaped inlet slots **30**, located on both faces of the valve plate **15**, sequentially register with the kidney shaped slots **29**, conducting the high pressure fluid, supplied to and from the cylinder bores **17** through the ports **29**, space **33**, space **75** and port **14**. Slot **35**, provided in the rotary valve plate **15**, sequentially registers with cylinder bores **17** through kidney shaped ports **29**, conducting the low pressure oil through a series of passages **36** to chamber **37** and port **13**.

The cylinder barrel **16** is prevented from rotating by pin **72** and while pivoting around the spherical surface **70** is free to align itself, under action of unbalanced hydrostatic forces, against the flat face of the rotary valve plate **15**. Since both flat surfaces of the rotary valve plate **15** are identical, the unbalanced force coming from the cylinder barrel **16** will effectively seal both rotary surfaces of the valve plate, permitting operation of the pump or motor at a minimum leakage level. The pin **72** works in oval slot **73** and permits not only a measure of universal freedom of the cylinder barrel **16** but also a measure of axial freedom, permitting the valve plate to work with minimum end clearances. The pistons **18** are mounted for reciprocation in cylinder bores **17** of the cylinder barrel **16** and are connected by articulated connecting rods **20** to the reaction plate **19**, while being universally mounted at their spherical ends. The reaction plate **19**, located on the antifriction bearing **22**, interposed between the reaction plate **19** and the cam plate **21**, is prevented from rotation by the bearing **27**, working in slot **28**, while being permitted to transmit the wobbling and therefore reciprocating motion through the articulated connecting rods **20** to pistons **18**. The cam plate **21** is interposed between the reaction plate **19** and the end cover **11**. The antifriction bearing **23** is positioned between the cam plate **21** and cover **11**. The rotary motion is transmitted to the cam plate **21** by the cylindrical extension **24**, sealed by the seal **26** and equipped with spline **25**, suitably connected to coupling of driving or driven source.

Between the cam plate **21** and the valve plate **15** a mechanism is interposed with can change the angular relationship of the cam plate **21** and the valve plate **15**, while transmitting the rotary motion from one to the other. This mechanism is basically composed of the control ball nut **66** which is secured to the cam plate **21** on its axis of rotation. The control ball nut **66**, through a series of balls **68**, operationally engages the control ball screw **67** which is secured to the control spring guide **62** which in turn, through the projecting ears **64**, engages slot **63** of the control cylinder **55**, permanently secured by dowel pin **57** and bolts, not shown, to the valve plate **15**. The control spring **62** is interposed between the ball nut **66** and the spring retainer **62** and provides the reaction which permits the transmission of rotary motion between the cam plate **21** and the valve plate **15**.

With he unit working as a motor, fluid introduced to high pressure port **14** is conducted through space **33** and drilled passage **32** to kidney shaped port **29** and cylinder bore **17**. The resultant hydrostatic forces, developed in the cylinder bores **17**, maintain cylinder barrel **16** against flat face of the rotary valve **15**, which is kept by the same force in abutment with working surface of the housing **10**. The non-rotating cylinder barrel **16**, equipped with spherical surface **70** working in contact with ring **71**, is capable of aligning itself to the surface of the rotary valve plate **15** containing the timing slots and therefore of working at minimum leakage level. The fluid under pressure, contained in the cylinder bore **17**, reacts against projected area of piston **18**. This creates a force which is transmitted through universally mounted connecting rods **20** to the reaction plate **19**, which is stabilized against rotation by the bearing **27** engaging slot **28**, and through series of balls **22** to the cam plate **21**. The sum of the hydraulic forces, transmitted from the pistons subjected to pressure, acting on the inclined plane of the cam plate **21**, will induce rotation. The rotary motion of the cam plate **21** is transmitted to the rotary valve plate **15** by the displacement changing mechanism. The exhaust fluid from the cylinder bores **17**, at low pressure, is displaced through kidney shaped ports **29**, open slot **35** and the series of passages **36** provided in the rotary valve plate **15**, to chamber **37** and low pressure port **13**. The cam plate **21** and valve plate **15** revolve under action of the torque generated on the inclined surface of the cam plate. During rotation the fluid under pressure is sequentially introduced by the rotary valve plate **15** to cylinder bores **17** of stationary self-aligning cylinder barrel **16**. The exhaust oil is conducted by the above described passages contained in the rotary valve plate to low pressure port **13**. The actual torque developed in the motor will depend on the magnitude of the pressure of the fluid introduced to the cylinder bores **17** and on the angular relationship between the inclined surface of the cam plate **21** and the rotary valve plate **15**. With the principal axis of the cam passing through the highest and lowest points of the inclined surface, coinciding with the axis dividing rotary valve plate **15** into the high and low pressure zones, the maximum torque is generated, the displacement of the unit per one revolution being at its greatest. When the angular relationship between these two axes is changed from this maximum position in either direction, a gradual reduction both in torque and displacement of the unit will result until a point is reached when with axis angularly disposed by 90° where both the torque output and effective displacement of the motor will become zero. Further angular rotation of the axis past the 90° position will gradually increase the torque output and the displacement of the motor while changing the direction of rotation of the output shaft. The present invention is mainly concerned with a mechanism which will change the angular relationship between the cam plate and the valve plate and which is responsive to a controllable signal whereby the motor displacement, torque output and direction of rotation can be controlled.

The mechanism transmitting the driving torque and controlling the angular relationship between the rotating cam plate **21** and valve plate **15** is based on the principle of inclined plane where any relative rotation between these two members must be accompanied by proportional relative axial movement within the mechanism. The ball nut **66** engages the ball screw **67** through the series of balls **68** working at minimum friction level so that the unit is reversible. With the ball nut **66** secured to cam plate **21** the ball screw **67** slidably engages with cylindrical pins

64, slots 63, provided in the control cylinder 55 which is secured to the rotary valve plate 15. The controlled axial movement of the ball screw 66 will produce a proportional change in the angular relationship between the cam plate 21 and the valve plate 15, thus changing the effective displacement and the torque output of the fluid motor.

Since the stroke changing mechanism must transmit torque equivalent to frictional resistance of the rotary valve plate 15, an axial force must be supplied to maintain the ball screw 67 in any specified position. This force must be equal to the reaction of the transmitted torque multiplied by mechanical advantage of the control ball screw 67. With the control mechanism at rest the control spring 69 maintains the control ball screw 67 in its extreme position against a stop, provided by the control cylinder 55, as shown in FIGURE 1. In this position the angular relationship between the cam plate 21 and valve plate 15 is such that the fluid motor is capable of developing maximum torque at maximum effective displacement for rotation of the cam plate 21 in one direction. The preload in the control spring 69 and therefore the force exerted on the ball screw 67 is so selected that the maximum torque can be supplied to the valve plate 15, without producing a relative rotation of the control mechanism.

The action of the control spring 69 is opposed by a hydraulic force acting on the control piston 59. Fluid under controlled pressure is supplied from port 58 and longitudinal passage 50 to chamber 51 which is in direct communication with bore 60 of the control cylinder 55. When the hydraulic force, induced by control pressure acting on the control piston 61, will balance either a sum or difference of axial force torque component and preload in the control spring 69, depending on the direction of rotation, the control mechanism will be in a state of equilibrium. Any further increase in the control pressure beyond this point will move the control ball screw 67 from its extreme position, compressing the control spring 69.

Still further increase in the control pressure will produce a movement of the control ball screw 67 and corresponding increase of the phase shift between the cam plate 21 and the valve plate 15, the magnitude of the increase depending on the rate of the control spring 69. With a continuing increase in control pressure level, at a phase shift of 90°, effective displacement and theoretical torque output of the fluid motor will become zero. Further movement of the control ball screw 67 will reverse the direction of rotation of the motor, gradually bringing its effective displacement and torque to the maximum value when reaching 180° phase shift. With the change of rotation of the fluid motor at phase shift of 90° the axial component of the friction torque, transmitted through the control mechanism, will change its direction, helping instead of opposing the force exerted by the control spring 69. This will result in a higher increase of control pressure per unit length of travel of the control ball screw 67. The change in direction of rotation will always take place with fluid motor at rest, thus eliminating shock loads. In this way torque output, effective displacement and direction of rotation of the motor can be controlled by a variable pressure signal.

The influence of the control pressure on the hydrostatic balance of the valve plate 15 is eliminated by the introduction of fluid at control pressure to chamber 52, the effective area of the enlarged end of the hollow balancing member 41 being the same as the area exposed to pressure in space 52. The effective areas of the hollow balancing plunger 41, the transfer sleeve 39 and the smaller end of the balancing piston 44 subjected to motor inlet pressure and pressure gradient on the sealing surfaces are so arranged that the hydraulic forces cancel each other, valve plate 15 remaining in state of perfect equilibrium, with its sealing surfaces hydrostatically balanced. An introduction of a device in form of a mechanical link spaced in the longitudinal passage 50 and engaging the control piston 59 will produce a mechanically operated control, where the position of control ball screw 67 can be changed at will.

While I have illustrated and described a preferred form of my invention and some modifications thereof it will be apparent to those skilled in the art that the device is capable of various additional modifications without departing from the scope and spirit of the invention as defined in the following claims.

What is claimed is:

1. An energy translating fluid pressure device comprising a housing defining a fluid chamber therein, a first supply port on said housing opening into said chamber, a cylinder barrel nonrotatably mounted within said chamber and having a plurality of cylinder bores extending axially therein, a cam plate journaled in said housing within said chamber for rotation about an axis parallel to said cylinder bores, a piston within each of said cylinder bores, means operable by said cam plate to progressively reciprocate said pistons, a valve plate mounted in said housing for rotation about the axis of said cam plate, said valve plate being mounted between the end of said cylinder barrel opposite said cam plate and said housing, said cylinder barrel being free for limited movement toward and away from said valve plate and free to align itself with said end of said cylinder barrel against said valve plate, said cylinder barrel being shaped so that pressure in said cylinder bores produces a hydrostatic force urging said cylinder barrel against said valve plate with a force which is a direct function of the pressure in said cylinder bores, said valve plate having first and second ports arranged to be sequentially moved into registration with said cylinder bores as said valve plate is rotated, said first valve plate port being in communication with said fluid chamber and said first supply port, drive means interconnecting said cam plate and said valve plate, said driving means including an antifriction screw and nut assembly, said nut being fixedly secured to said cam member, said screw being connected to said valve plate by a spline allowing said screw to move axially with respect to said valve plate member and said nut and rotatable with said valve plate, whereby axial movement of said screw changes the phase relationship between said valve plate and said cam plate to control the effective displacement of said device, an expansible chamber control motor including a cylinder member carried by said valve plate and a control piston slidable within said cylinder member and engageable with said screw to shift said screw toward said cam plate, spring means to shift said screw away from said cam plate, an axial passage in said valve plate, said passage being connected to the other of said valve plate ports, a second supply port on said housing, a longitudinal passage in said housing co-axial with the axis of rotation of said valve plate and extending between said valve plate and said second supply port, a sleeve slidable in said axial passage and adapted to make sealing connection between said second supply port and the axial passage of said valve plate whereby fluid may flow between said second valve plate port and said second supply port, a transfer tube mounted within said axial housing passage and said valve plate passage, said transfer tube making sealing contact with said valve plate to admit control fluid to said control motor, and control port means on said housing adapted to admit fluid pressure into said transfer tube, the hydrostatic pressures on said valve plate being substantially balanced.

2. An energy translating fluid pressure device comprising a housing defining a fluid chamber therein, a first supply port on said housing opening into said chamber, a cylinder barrel nonrotatably mounted within said chamber and having a plurality of cylinder bores extending axially therein, a cam plate journaled in said housing within said chamber for rotation about an axis parallel to said cylinder bores, a piston within each of said cylinder bores, means operable by said cam plate to progressively reciprocate said pistons, a valve plate mounted in said housing for

rotation about the axis of said cam plate, said valve plate being mounted between the end of said cylinder barrel opposite said cam plate and said housing, said cylinder barrel being free for limited movement toward and away from said valve plate and free to align itself with said end of said cylinder barrel against said valve plate, said cylinder barrel being shaped so that pressure in said cylinder bores produces a hydrostatic force urging said cylinder barrel against said valve plate with a force which is a direct function of the pressure in said cylinder bores, said valve plate having first and second ports arranged to be sequentially moved into registration with said cylinder bores as said valve plate is rotated, said first valve plate port being in communication with said fluid chamber and said first supply port, drive means interconnecting said cam plate and said valve plate, said driving means including an antifriction screw and nut assembly, said nut being fixedly secured to said cam member, said screw being connected to said valve plate by a spline allowing said screw to move axially with respect to said valve plate member and said nut and rotatable with said valve plate, whereby axial movement of said screw changes the phase relationship between said valve plate and said cam plate to control the effective displacement of said device, an expansible chamber control motor including a cylinder member carried by said valve plate and a control piston slidable within said cylinder member and engageable with said screw to shift said screw toward said cam plate, spring means to shift said screw away from said cam plate, an axial passage in said valve plate, said passage being connected to the other of said valve plate ports, a second supply port on said housing, a longitudinal passage in said housing co-axial with said valve plate and extending between said valve plate and said second supply port, a sleeve slidable in said axial passage and adapted to make sealing connection between said second supply port and the axial passage of said valve plate whereby fluid may flow between said second valve plate port and said second supply port, a transfer tube mounted within said axial housing passage and said valve plate passage, said transfer tube having a radial flange adapted to make sealing contact against said valve plate on the side adjacent said cylinder barrel, a control chamber on the end of said housing outward of said second supply port, said transfer tube extending from said flange through said valve plate passage, said sleeve and into said control chamber, seal means to prevent fluid communication between said control chamber and said second supply port, a piston on said transfer tube whereby pressures in said control chamber and said hydraulic cylinder on said transfer tube

are balanced, and control port means on said housing adapted to admit fluid pressure through said transfer tube into said control motor, the hydrostatic pressures on said valve plate being substantially balanced.

3. An energy translating fluid pressure device comprising a housing defining a cavity, a generally nonrotatable cylinder barrel mounted in said housing within said cavity, said cylinder barrel being formed with a plurality of cylinder bores, pistons reciprocable in said bores, cam means mounted to rotate in said housing and operatively associated with said pistons, a low pressure port in said housing open to said cavity, a high pressure port in said housing, a valve plate positioned between one end of said cylinder barrel and said housing drive means connected between said cam means and valve plate operating said valve plate to sequentially connect said cylinder bores to said cavity and to said high pressure port, a control pressure port in said housing, said drive means including a fluid motor connected to said control pressure port operable to change the position of said valve plate with respect to said cam means and thereby change the effective displacement of said device in response to changes in pressure at said control pressure port, a hydrostatically balance transfer tube extending through said valve plate to connect said control pressure port to said fluid motor, the exterior of said transfer tube defining at least part of a passage connecting said valve plate and said high pressure port, said cylinder barrel being mounted in said housing for limited free movement toward and away from said valve plate and so that said end of said cylinder barrel is substantially free to align itself with said valve plate, said cylinder barrel being shaped so that it is hydrostatically urged toward said valve plate by fluid under pressure in said cylinder bores with a force which is a direct function of the pressure in said cylinder bores, said valve plate being hydrostatically balanced so that changes in pressure in said high pressure port and said control pressure port do not produce any substantial resulting forces on said valve plate.

References Cited

UNITED STATES PATENTS

2,969,810	1/1961	Dudley	103—228
3,011,453	12/1961	Budzich	103—173
3,074,345	1/1963	Scheflow	103—173

DONLEY J. STOCKING, *Primary Examiner.*

MARK NEWMAN, *Examiner.*

R. M. VARGO, W. L. FREEH, *Assistant Examiners.*