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Katayama

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[54] **VARIABLE-DISPLACEMENT TYPE FLUID PUMP OR MOTOR**

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[52] **U.S. Cl.** **91/483,** 91/501

[51] **Int. Cl.** **F01b 13/04**

[58] **Field of Search** 91/483, 501

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Primary Examiner—William L. Freeh
Attorney—Eric H. Waters et al.

[57] **ABSTRACT**

A variable-displacement type fluid pump or motor comprises, essentially, a rotor having ball pistons movable in the axial direction, a pair of cam members against which the ball pistons abut, ports for supplying hydraulic fluid into and discharging the same out of spaces between the ball pistons within cylindrical bores, a valve sleeve having ports for determining the timing of supply and discharge of the hydraulic fluid, and means for transmitting the rotation of the cam member to the valve sleeve. One of the pair of cam members is fixed, while the other is rotatably supported to a casing. The valve sleeve is also rotatably supported and is rotated by an angle equal to one-half of the rotational angle of the movable cam member in the same direction by the rotation of the movable cam member transmitted by way of the rotation transmitting means.

7 Claims, 10 Drawing Figures

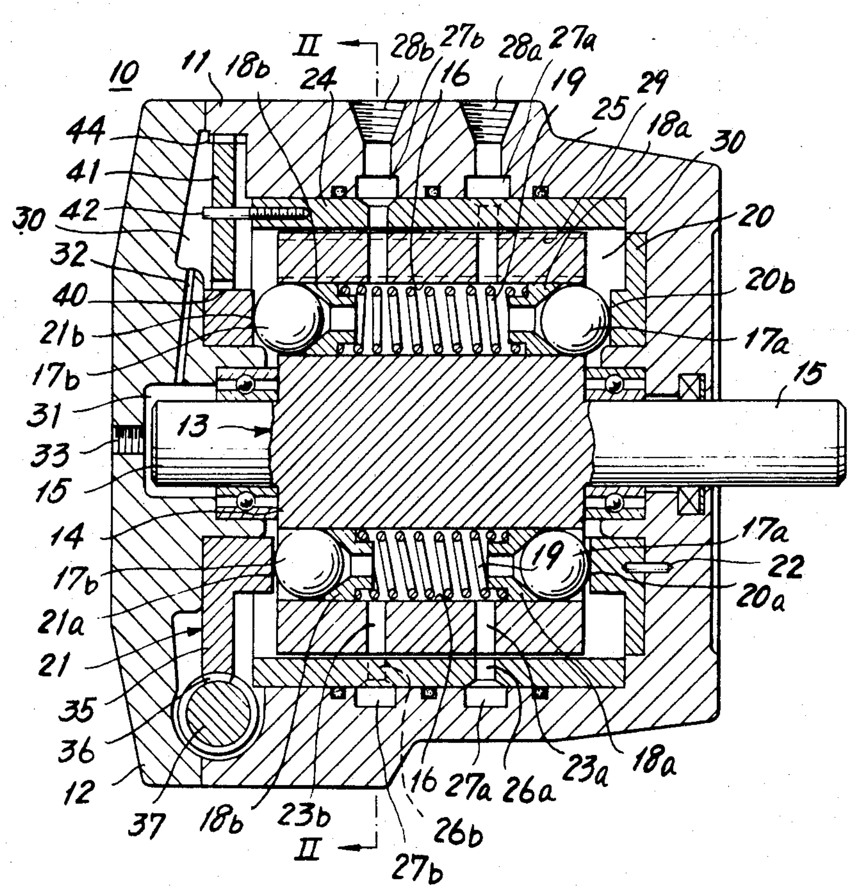
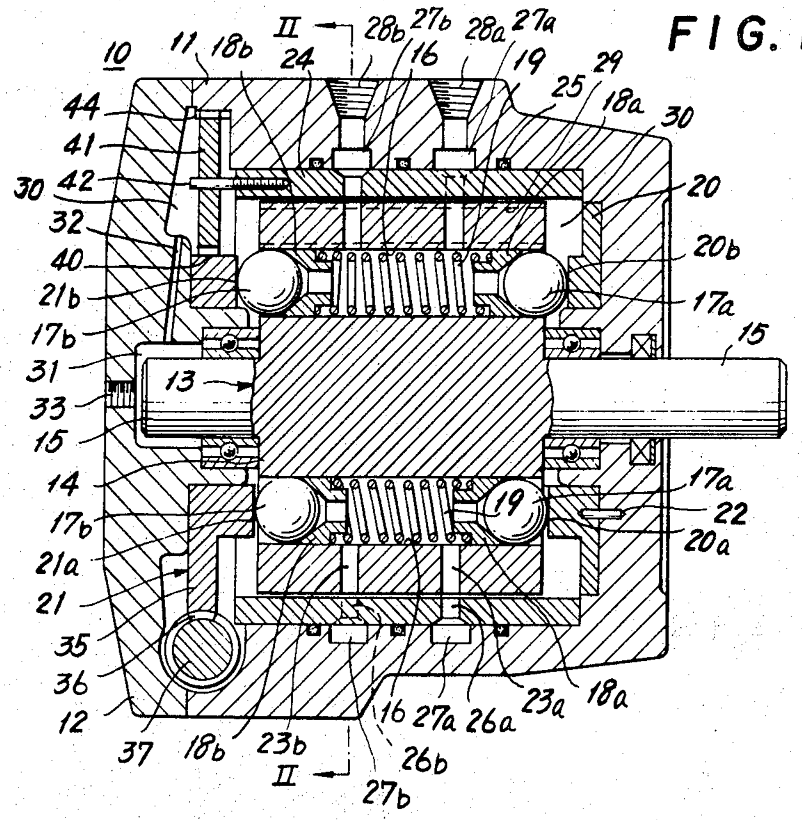


FIG. 1


10
11
18b
24
II
28b
27b
16
28a
27a
19
29
25
18a
30
44
41
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30
32
40
21b
17b
31
13
33
15
14
17b
21a
21
35
36
12
37
18b
23b
II
27b
26b
16
27a
26a
23a
18a
20
20b
17a
15
17a
19
22
20a

FIG. 2

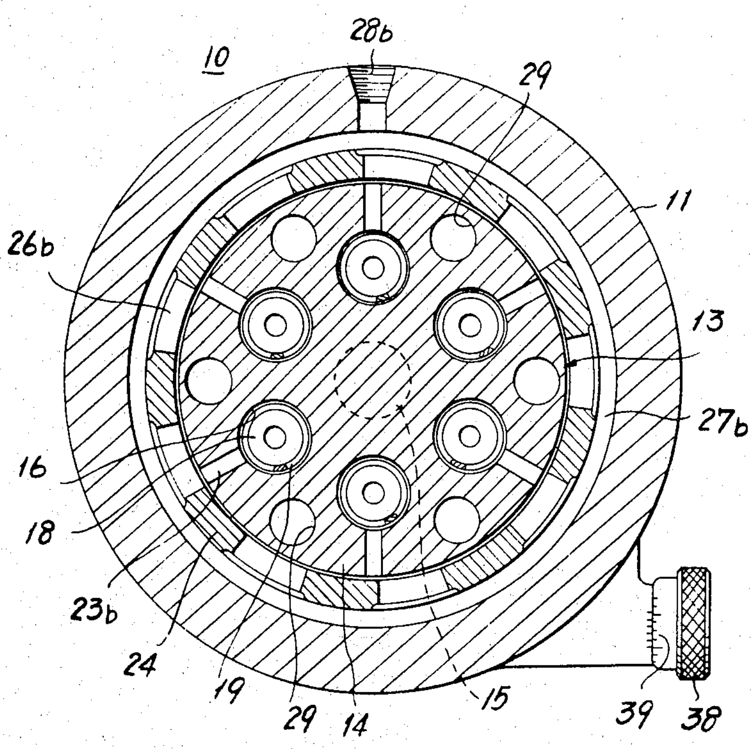
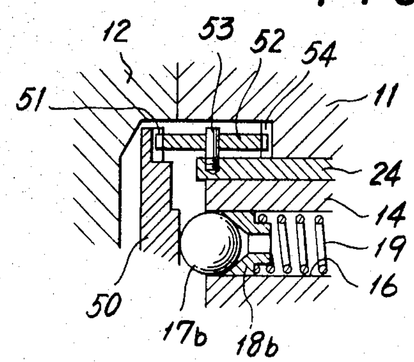

28b
10
29
11
26b
13
27b
16
18
23b
24
19
29
14
15
39
38

FIG. 6


12
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52
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11
24
14
19
16
50
17b
18b

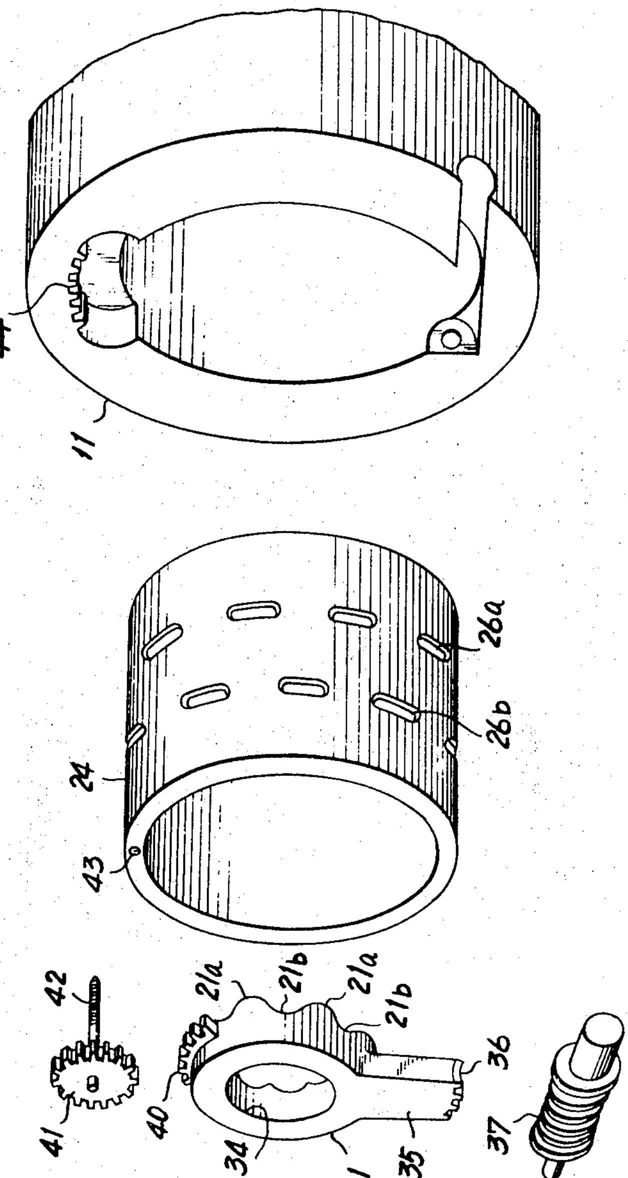

FIG. 3

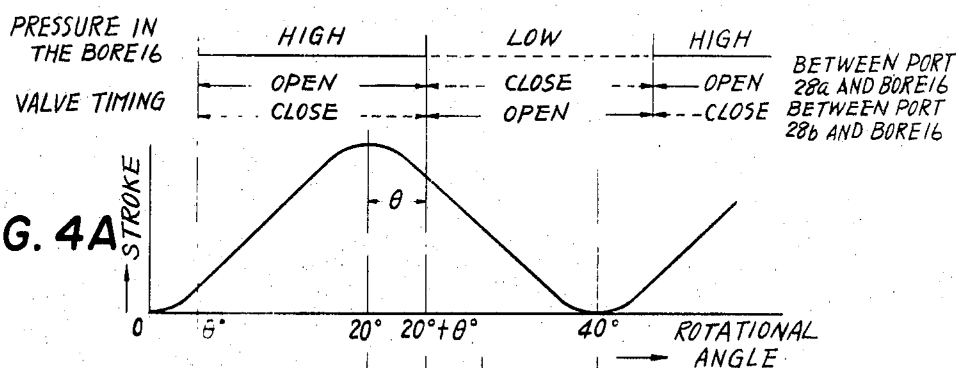

FIG. 4A

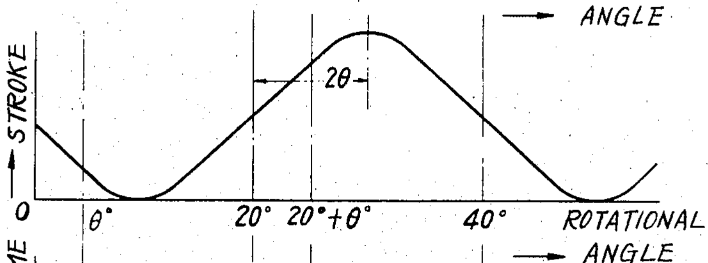

FIG. 4B

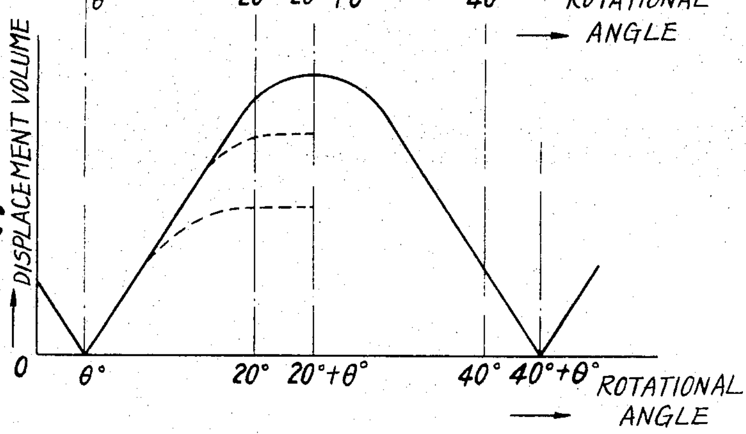

FIG. 4C

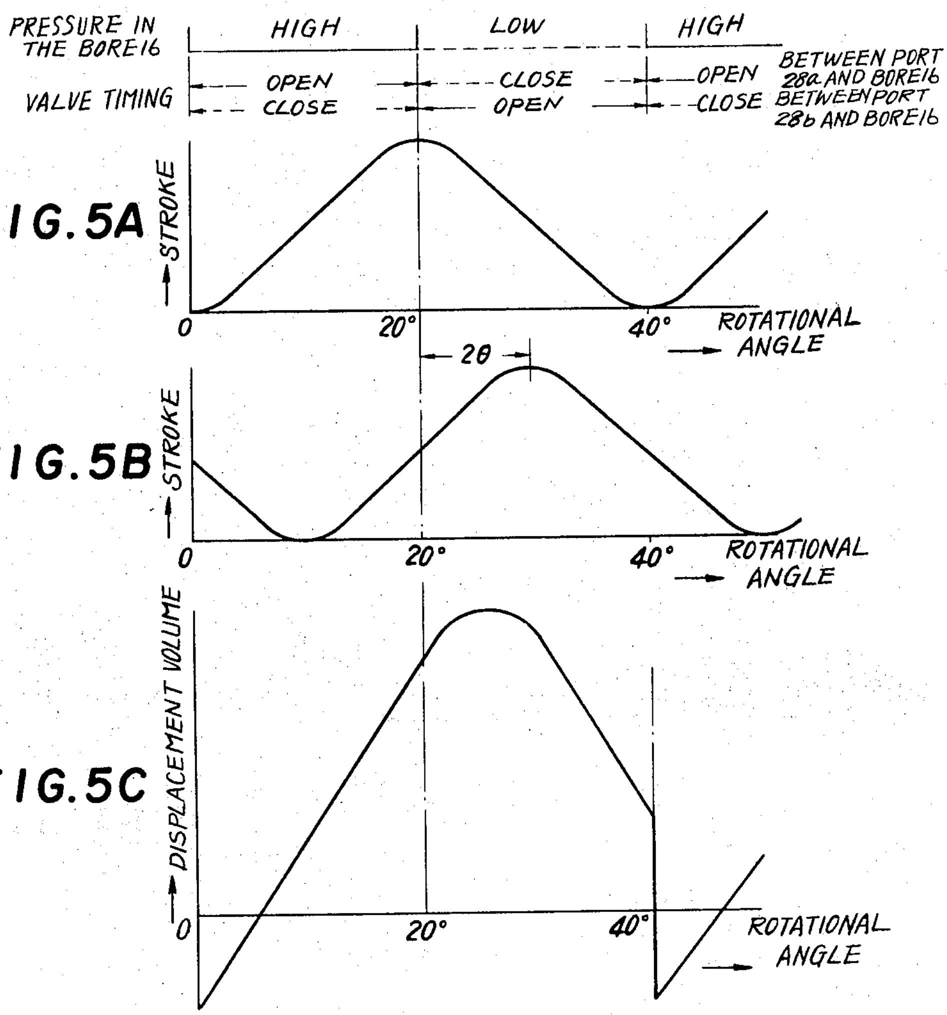

FIG.5A

FIG.5B

FIG.5C

VARIABLE-DISPLACEMENT TYPE FLUID PUMP OR MOTOR

BACKGROUND OF THE INVENTION

This invention relates to a fluid pump or motor of variable-displacement type and more particularly to a new and advanced fluid pump or motor of this type wherein the relative opposed positions of two cam plates can be varied to vary the displacement volume of pistons and thereby to vary the capacity of the fluid pump or motor.

In general, a fluid pump or motor (hereinafter referred to simply as "fluid device") has been developed as a fluid device of the bent-axis type or the cam-plate type. In a fluid device of this type, plunger type pistons slide over cam plates and undergo reciprocating motion. For this reason, the piston motion is limited to a harmonic motion of one stroke per one revolution of a rotor. Consequently, conventional fluid devices of the axial-piston type have had the disadvantage of substantially large fluctuations of discharge pressure when used as pumps or of output torque when used as motors. Accordingly, in order to reduce the period of these fluctuations, it has been necessary to use these fluid devices at rotational speeds within the range of medium and high speeds. Furthermore, since the reaction force of the hydraulic fluid cannot be fully balanced, quantities such as the rigidity of the rotary shaft and the capacity of the bearings are required to be greater than those of fluid devices of other types. This requirement has given rise to various difficulties such as an adverse effect on the serviceable life and performance of these fluid devices.

As an axial-piston type fluid device of a new type wherein these difficulties can be overcome, a fluid device of a so-called axial ball-piston type has been proposed. A fluid device of this type has a plurality of pairs of ball pistons, which pairs successively undergo stroke motion.

In some cases, however, there arises the necessity of varying the discharge rate when the fluid device is operating as a pump or of varying the rotational torque when it is operating as a motor. These variations can be effected by varying the mutually relative positions of the two opposed cam plates thereby to vary the displacements of the stroke motions of the ball pistons.

Axial ball-piston fluid devices of this variable-displacement type have been proposed, one example being that disclosed in the specification of U.S. Pat. No. 3,508,465. In this proposed fluid device, two cam plates are simultaneously rotated in mutually opposite directions by one bevel gear thereby to vary the relative opposed positions of the two cam plates. However, since the two cam plates are being continually subjected to reaction forces due to the ball pistons and have a movable construction, errors easily develop in the positions of the cam plates. This has been a difficult problem in this type of fluid device. For this reason, it is desirable that the cam plates have a fixed construction as much as possible.

SUMMARY OF THE INVENTION

Accordingly, it is a general object of this invention to provide a novel and useful variable-displacement fluid pump or motor wherein the above described difficulties accompanying the proposed fluid device are overcome.

Another object of this invention is to provide a fluid device of the above stated type wherein one of two opposed cam plates is of fixed construction, while the other is adapted to be movable, and the displacement volume is varied by the rotation of the movable cam plate. In this fluid device, a valve sleeve rotates, and the timing of port switching is also varied together with the rotation of the movable cam plate. Since one of the two cam plates is stationary, there is very little possibility of error arising in the relative opposed positions of the two cam plates, whereby the displacement volume can be varied and adjusted with high accuracy. Futhermore, it is possible to use, in the fluid device of this invention, parts which are common to fluid devices of the fixed capacity type.

A further object of this invention is to provide a fluid device wherein fluid which has leaked within the rotor can be conducted to the outside, and, at the same time, the moment of inertia of the rotor is reduced. By applying a back pressure to the discharge outlet of the leakage fluid, the rotor can be caused to a free wheeling.

Further objects and features of this invention will be apparent from the following detailed description with reference to the accompanying drawings, in which like parts are designated by like reference numerals.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. 1 is a side elevation, in vertical section, showing essential parts of one embodiment of a variable-displacement type fluid device according to this invention;

FIG. 2 is a cross section taken along the plane indicated by line II — II in FIG. 1;

FIG. 3 is an exploded perspective view showing essential parts of a mechanism for adjustably varying capacity in the fluid device shown in FIGS. 1 and 2;

FIGS. 4A through 4C are graphical representations for a description of the variations of the strokes and displacement volume of a pair of ball pistons with respect to the rotational angle of the rotor in the case where one cam plate is rotated by an angle 2 θ from the other cam plate, and, at the same time, the valve sleeve is rotated through an angle θ ;

FIGS. 5A through 5C are graphical representations similar to FIGS. 4A through 4C in the case where the valve sleeve is fixed, and one cam plate is rotated by an angle 2 θ from the other cam plate; and

FIG. 6 is a fragmentary side elevation, in vertical section, showing another embodiment of the ball piston part of the fluid device illustrated in FIG. 1.

DETAILED DESCRIPTION

In one embodiment as illustrated in FIGS. 1, 2 and 3 of a variable-displacement fluid device according to this invention, a fluid device generally designated by reference numeral **10** is operable as a fluid pump or as a fluid motor. The fluid device **10** has an outer casing made up of a housing **11** formed in the shape approximately of a hollow cylinder with one closed end and an opposite open end and an end cover plate or end cap **12** covering the open end of the housing **11**. The end wall of the closed end of the housing **11** and the end cap **12** on their inner sides respectively support bearings which are coaxially aligned on an axis perpendicular to the end wall and the end cap. These bearings rotatably support a rotor shaft **15** integrally and coaxially formed with a cylindrical rotor block or rotor cylinder 14 to constitute a rotor 13. Accordingly, the rotor cylinder 14 can rotate freely within the casing. This rotor cylinder 14 is provided therein with a plurality (six in the instant embodiment) of cylindrical bores 16 of circular cross-section bored completely therethrough with axes parallel to and equidistant from the axis of the shaft 15 with equal circumferential spacing therearound and constituting cylinders for operation therein of ball pistons as described hereinafter.

While the rotor cylinder 14 and the rotor shaft 15 of the rotor 13 in the instant embodiment are formed integrally from the same material stock, the rotor may also be made by fabricating the rotor cylinder and the rotor shaft separately but with mutually mating spline grooves and ridges and then assembling the rotor by fitting the shaft into the rotor cylinder.

Each of the above mentioned cylindrical bores 16 at its two ends accomodates respective ball pistons 17*a* and 17*b* slidably disposed therein to constitute a pair with outer parts projecting out of the cylindrical bore. These ball pistons 17*a* and 17*b* in each cylindrical bore 16 are seated and held in the seats of respective seal rings 18*a* and 18*b* disposed slidable within the cylindrical bore 16 inward of the ball pistons. The pair of seal rings 18*a* and 18*b* are continually urged in axial directions outward from the cylindrical bore 16 by the elastic force of a compression coil spring 19 disposed within the bore 16 between the seal rings 18*a* and 18*b*. Accordingly, the outward force of the spring 19 is transmitted through the seal ring 18*a* and 18*b* to the ball pistons 17*a* and 17*b* to urge them in their outward projecting directions in each cylindrical bore 16. The ball pistons 17*a* and 17*b* at their outermost parts abut against the cam surfaces of a pair of mutually opposed cam plates 20 and 21 and are thereby limited in their outward displacement.

The cam plate 20 is fixed to the inner end surface of the housing 11 by a dowel pin 22, while the cam plate 21 is rotatably supported on the inner side of the end cap 12. The cam plates 20 and 21 respectively have circular cam surfaces, each having a plurality (nine in the instant embodiment) of cam crests 20*a* and 21*a*, against which the outer parts of the ball pistons 17*a* and 17*b* abut, and over which they roll along the cam paths of the cam plates as the rotor 13 rotates. As a result, the ball pistons 17*a* and 17*b*, in following the cam surface, are displaced in the axial direction of the cylindrical bore 16 in accordance with the projection of the cam crests 20*a* and 21*a* from the cam valleys 20*b* and 21*b*. The cam curves of the cam surfaces of the cam plates 20 and 21 are so determined that, in the case where the fluid device 10 is used as a motor, the total sum of the driving torques of all cylindrical bores 16 becomes a constant value. In the case where the fluid device 10 is used as a pump, the cam curve is so determined that the pump discharge pressure becomes constant. Furthermore, the cam surface is so designed that, ordinarily, in one half cycle of the cam curve, there exist three constant and continuous periods, namely, an acceleration period, a constant velocity period, and a deceleration period of the torque. The displacement volume of the fluid device 10 is determined by the distances of projections of the cam crests 20*a* and 21*a*. Furthermore, the number of the cam crests may be selected as desired. Specifications of the fluid device 10 such as the range of operational rotational speed and output torque can be changed by suitably interchanging the cam plates 20 and 21 and a valve sleeve 24 described hereinafter.

The rotor cylinder 14 is further provided between its peripheral cylindrical surface and the cylindrical bores 16 with hydraulic fluid supply (or suction) ports 23*a* and exhaust (or discharge) ports 23*b* extending outwardly in radial directions. A valve sleeve 24 having the shape of a hollow cylinder with open ends is fitted rotatably within and in sliding contact with the inner cylindrical surface of the housing 11 and rotatably accommodates therewithin the rotor cylinder 14. For oil-tight sealing, O-rings 25 are provided between the housing 11 and the valve sleeve 24. The valve sleeve 24 is provided with ports 26*a* and 26*b* formed therein in positions to confront the ports 23*a* and 23*b*, respectively, as indicated in FIG. 1. These ports 26*a* and 26*b* are each of a number equal to the number (nine in the instant embodiment) of the crests of the cam plates 20 and 21, and each of these ports has a certain angular width as viewed in the axial direction as in FIG. 2. Although not shown in FIG. 2, the ports 26*a* and 26*b* are formed in mutually staggered or offset positions as viewed in the axial direction so that the both ports 26*a* and 26*b* will not simultaneously confront the ports 23*a* and 23*b* of the rotating rotor 13 as described hereinafter.

The inner cylindrical surface of the housing 11 is provided with annular grooves 27*a* and 27*b* formed therein around the entire inner circumference at positions to confront the ports 26*a* and 26*b* of the valve sleeve 24. The housing 11 is further provided with a hydraulic fluid supply (or suction) port 28*a* and an exhaust(or discharge) port 28*b* communicating the annular grooves 27*a* and 27*b* respectively with the outside.

The rotor cylinder 14 is further provided with through holes 29 formed therethrough parallel to the cylindrical bores 16. In the instant embodiment, the through holes 29 are of the same number (six) as the cylindrical bores 16. The two ends of each of the through holes 29 are communicative with spaces 30 formed between the housing 11 and the two ends of the rotor cylinder 14. The end cap 12 has on its inner side a central cavity 31 accommodating one end of the rotor shaft 15 and a fluid passage hole 32 communicating the cavity 31 to one of the above mentioned spaces 30. The end cap 12 is further provided with an aperture 33 for controlling leakage fluid which communicates the cavity 31 with the outside. In the case where, during the operation of the fluid device, hydraulic fluid leaks out of the rotor cylinder 14 through interstices such as gaps between the seal rings 18 and ball pistons 17*a* and 17*b* and the cylindrical bores 16, the fluid which has thus leaked collects in the spaces 30 and it then discharged out of the fluid device through the through holes 29, fluid passage hole 32, the cavity 31, and the fluid control aperture 33.

When hydraulic fluid at a pressure higher than the normal fluid supply pressure is supplied through the control aperture 33, the ball pistons 17*a* and 17*b* are subjected to this pressure from the side of the spaces 30 and are pushed inward in the cylindrical bores 16, overcoming the force of the springs 19. Consequently, the ball pistons 17*a* and 17*b* are released from contacting with the cam plates 20 and 21, and the rotor 13 assumes a state wherein it can rotate freely, that is, a state wherein it can freewheel. This state corresponds to that of the fluid device in cases such as that wherein it is used, for example, as the motor of a winch and is caused to lower a load abruptly after the load has been hoisted.

The above described through holes **29** have, in addition to being passage holes for fluid which has leaked, the function of also reducing the mass of the rotor cylinder **14** thereby to decrease the moment of inertia thereof and to improve the rotational responsiveness. In the case where this fluid device **10** is used as a servomotor operated in cooperation with a servo-valve, its control performance, particularly, is remarkably improved. The shape of the cross section of the through holes **29** is not limited to a circle but may be any suitable shape such as a circular sector or an ellipse. By selecting the optimum cross-sectional shape in each case, the moment of inertia of the rotor **13** can be reduced even more effectively.

An essential part of the fluid device **10** according to this invention is the capacity or displacement varying machanism, which will now be described with respect to one embodiment thereof. The cam plate **21** has a cam surface which is identical to that of the cam plate **20** and, in assembled state, is in coaxial opposition to the cam surface of the cam plate **20.** The cam plate **21** further has a cylindrical hollow part **34,** as indicated in FIG. 3, which is rotatably fitted onto a hollow boss part of the end cap **12** on the inner side thereof. The cam plate **21** further has an arm **35** extending downward and having a lower end on which a partial (sector) worm gear **36** is formed. This worm gear **36** is meshed with a worm **37** supported on and fixed to a rotatable shaft, which extends out of the casing, and a turning knob **38** for adjustably varying the capacity or displacement volume is fixed to the outer end of this shaft. The rotational angle of the knob **38** is indicated by a calibration scale **39.** By turning the knob **38,** the worm **37** is rotated, and the cam plate **21** is rotated about the boss part of the end cap **12.**

The cam plate **21** is still further provided at its upper peripheral part with a partial (sector) gear **40** formed integrally therewith and meshed with a pinion or gear wheel **41.** The gear wheel **41** is supported on and fixed to a shaft **42,** which is fitted in a hole **43** in the end surface of the valve sleeve **24** in a freely rotatable manner and, moreover, in a manner such as to be rotatable integrally together with the valve sleeve without any play relative to the center of rotation of the valve sleeve. The gear wheel **41** is further meshed with a partial internal gear **44** formed on the inner side of the housing **11** at its end to which the end cap **12** is secured.

The radii and gear ratios of the cam plate **21** and the gear wheel **41** are so selected that, when the cam plate **21** is rotated in one direction by an angle of 2 θ by the worm **37,** the valve sleeve **24** will be rotated in the same direction by an angle of θ.

The fluid device **10** and the capacity varying mechanism of the above described construction according to this invention operate in the manner described below with respect to the case where the fluid device is used as a fluid motor.

High-pressure hydraulic fluid supplied into the supply port **28** a flows successibely through the annular groove **27***a* and ports **26***a*. The position of the valve sleeve **24** is initially so set that, when a pair of ball pistons **17***a* and **17***b*, abutting against the cam surfaces of the cam plates **20** and **21,** assume a state wherein they are pressed into their innermost positions within their cylindrical bore **16**(i.e., the positional state indicated by the lower pair of ball pistons in FIG. **1**), a port **26***a* of the valve sleeve **24** and the pertinent port **23***a* of the rotor **13** become communicative. Furthermore, the relative angular positions of the cam plates **20** and **21** are initially so set that, in order to obtain the maximum displacement volume, the crests of their respective cam surfaces are in exactly opposed alignment.

When the port **26***a* and the pertinent port **23***a* become communicative as mentioned above, the high-pressure hydraulic fluid supplied through the port **28***a* flows through the port **23***a* and is thus supplied into the space between the seal rings **18***a* and **18***b* within the cylindrical bore **16.** As a result, the pressure of the hydraulic fluid thus supplied is transmitted through the seal rings **18***a* and **18***b* to the ball pistons **17***a* and **17***b* to impart outwardly directly forces to the ball pistons **17***a* and **17***b*, that is, a rightward force to the ball piston **17***a* and a left-ward force to the ball piston **17***b* as viewed in FIG. **1.** Consequently, the ball pistons **17***a* and **17***b* are respectively forced against the cam surfaces of the cam plates **20** and **21** and at the same time are forced to rol down inclined cam surfaces toward the bottoms of the valleys between one pair of corresponding cam crests **20***a* and **21***a* to the succeeding pair of cam crests. Since the forces due to the pressure of the hydraulic fluid at this time is divided into respective components in the rotational direction of the rotor **13** and components in the axial direction of the cylindrical bore **16,** the ball pistons **17***a* and **17***b* roll down their respective inclined cam surfaces and, at the same time, impart a torque to the rotor **13** whereby the rotor **13** rotates in one direction.

The angular widths and relative positional relationship of the ports **26***a* and **26***b* are so set that, during the interval from the state when the ball pistons **17***a* and **17***b* contact the cam crests **20***a* and **21***a* to the state when they reach the bottoms of the valleys **20***b* and **21***b* of the cam surfaces after rolling down the inclined surfaces (the latter state corresponding to that of the upper pair of ball pistons in FIG. **1**), the ports **26***a* and **23***a* are in a communicative state, while, at the same time, the ports **26***b* and **23***b* are in a non-communicative state. When the ball pistons **17***a* and **17***b* are at the bottoms of the cam valleys **20***b*, the ports **20***a* and **23***a* become non-communicative, while ports **26***b* and **23***b* become communicative.

At this time, the above described operation is started with respect to the ball pistons **17***a* and **17***b* in another cylindrical bore **16,** whereby the rotor **13** receives a further torque urging it to rotate in the same direction. The rotation of the rotor **13** causes the ball pistons **17***a* and **17***b* which have reached the bottoms of their respective cam valleys to roll up the inclined surfaces leading to the succeeding cam crests and are thereby displaced inward (i.e., respectively leftward and rightward as viewed in FIG. **1**) in their cylindrical bore **16.** Consequently, the hydraulic fluid in the cylindrical bore **16** between the seal rings **18***a* and **18***b* is discharged by way of ports **23***b* and **26***b*, annular groove **27***b*, and port **28***b* to a drain side outside of the fluid device **10.** Thereafter, the above described operation is repeated successively with respect to the remainder of the pairs of ball pistons, whereby the rotor **13** continues to rotate smoothly. The torque thus imparted to the rotor **13** is transmitted to the outside through the rotor shaft **15**, the fluid device **10** thereby operating as a motor.

In the case where the rotational direction of the rotor **13** is to be reversed relative to that described above, the port **28b** is used as the supply port of the high-pressure hydraulic fluid, and the port **28a** is used as the discharge or exhaust port. Furthermore, in the case where the fluid device **10** is to be used as a pump, the rotor shaft **15** is rotated by mechanical motive power supplied from the outside, and the port **28a** or **28b** is used as the suction port, while the port **28b** or **28a** is used as the discharge port.

The displacement volume between the ball pistons **17a** and **17b** in this fluid device is determined by a relative movement stroke wherein the respective movement directions of the ball pistons are also taken into consideration. The relative movement stroke of the ball pistons **17a** and **17b** is determined by the relative opposed positions of the cam crests and valleys of the mutually opposed cam plates **20** and **21**. Furthermore, the normal operation of this fluid device **10** as a motor or as a pump is determined by the variation of the above mentioned displacement volume and by the valve timing of the communicative and non-communicative states of the ports **26a** and **26b** of the valve sleeve **24** and the ports **23a** and **23b** of the rotor cylinder **14** of the rotor **13**.

For example, in an extreme case wherein the relative positions of the cam plates **20** and **21** are so set that the cam crests of one cam plate are mutually opposite the cam valleys of the other, and the cam valleys of the former cam plate are mutually opposite the cam crests of the latter, the ball pistons **17a** and **17b** merely move simultaneously to the right or to the left, while maintaining a constant spacing mutually therebetween, as the rotor **13** rotates. In this case, accordingly, there is no variation of the displacement volume, and suction of hydraulic fluid and discharge thereof are not accomplished when the rotor **13** is forced to rotate compulsorily by power applied from the outside.

In varying the displacement volume, the aforedescribed knob **38** is rotated by a suitable angle, whereupon the worm **37** rotates to rotate the worm gear **36**, whereby the cam plate **21** is rotated. Simultaneously, the rotation of the cam plate **21** causes the gear wheel **41** meshed with gear teeth **40** and **44** to undergo a planetary gear motion whereby it rotates about its own axis and simultaneously revolves about an orbital path. The orbital revolution of the gear wheel **41** causes an integral rotation of the valve sleeve **24**. In this operation, as mentioned hereinbefore, the rotation of the cam plate **21** by an angle 2 θ causes the valve sleeve **24** to rotate in the same direction by an angle θ.

The case where only the cam plate **21** is rotated by an angle 2 θ, and the valve sleeve **24** is held fixed as it is will now be considered. For this purpose, reference is made to FIG. 5A, which indicates the variation of the rightward displacement stroke of the ball piston **17a** on the right side in abutment with the cam plate **20** with respect to the rotational angle of the rotor **13**. FIG. 5B indicates the variation of the leftward displacement stroke of the left ball piston **17b** abutting against the cam plate **21** with respect to the rotational angle of the rotor **13**. As is apparent from a comparison of FIGS. 5A and 5B, the angular position of the peak of curve shown in FIG. 5B is offset by the angle 2 θ from the angular position of the peak of the curve shown in FIG. 5A. However, the valve sleeve **24** is not being rotated relative to the cam plate **20**, and the valve timing determined by the positional relationships of the ports **26a** and **26b** of the valve sleeve **24** and the ports **23a** and **23b** of the rotor **13** are the same as those prior to the rotation of the cam plate **21**. More specifically, the valve timing is as follows. With respect to the curve of FIG. 5A, in the intervals from the bottoms of the valleys to the peaks of the curve (i.e., 0° to 20°, 40° to 60° etc., in the instant embodiment), the hydraulic fluid supplying port **28a** is communicating by way of the ports **26a** and **23a** with the cylindrical bore **16**, whereby the mechanism is in the valve-open state, while the discharge port **28b** is not communicative with the cylindrical bore **16**, and the mechanism is in the valve-closed state, the pressure within the cylindrical bore **16** being at a high value. Furthermore, during the intervals from the peaks to the bottoms of the valleys (i.e., 20° to 40°, 60° to 80°, etc., in the instant embodiment), the states are reversed. That is, the hydraulic fluid supplying port **28a** is not communicative with the cylindrical bore **16**, whereby the mechanism is in the valve-closed state, while the discharge port **28b** is communicative by way of ports **26b** and **23b** with the cylindrical bore **16**, whereby the mechanism is in the valve-open state, and the interior of the cylindrical bore **16** is at a low pressure.

At this time, the curve of variation of the displacement volume determined by the spacing between the right and left ball pistons **17a** and **17b** and by the respective displacement directions of these ball pistons becomes as indicated in FIG. 5C. As is apparent from this figure, the relationship between the variation of the ball pistons and the above described valve timing is such that a negative part in the displacement volume is produced. Moreover, this negative part is produced abruptly at the valve timing changeover point as is best indicated on the right-hand side of the curve shown in FIG. 5C. This negative part of the displacement volume means that while the fluid device **10** is being operated normally as a motor, for example, it suddenly operates as a pump. This phenomenon is undesirable.

Accordingly, the fluid device of this invention is so adapted that the cam plate **21** is rotated by the angle 2 θ, while, at the same time, the valve sleeve **24** is rotated by the angle θ. FIGS. 4A and 4B, similarly as in FIGS. 5A and 5B, respectively, indicate curves of variations of strokes in the rightward and leftward strokes of the right side and leftside ball pistons **17a** and **17b**. Since the cam plate **21** is rotated by the angle 2 θ relative to the cam plate **20**, the peaks of the curves of FIGS. 4A and 4B are mutually offset by the angle 2 θ.

Furthermore, since the valve sleeve **24** is rotated by the angle θ in the same direction as the cam plate **21**, the changeover point of the valve timing is offset by the angle θ from that in the case described above in conjunction with FIGS. 5A, 5B, and 5C. More specifically, the interval wherein the hydraulic fluid supplying port **28a** is in communicative (valve-open) state with the cylindrical bore **16**, and the discharge port **28b** is in non-communicative (valve-closed) state with the cylindrical bore **16** corresponds to the range wherein the rotational angle of the rotor **13** is $\theta°$ to $(20° + \theta°)$, $(40° + \theta°)$ to $(60° + \theta°)$, etc. in FIG. 4A. The interior of the cylindrical bore **16** at this time is at a high pressure. Furthermore, the interval wherein the hydraulic fluid supplying port **28a** is in non-communicative (valve-closed) state with the cylindrical bore **16**, and the discharge port **28**b is in communicative (valve-open) state with the cylindrical bore **16** corresponds to the range wherein the rotational angle of the rotor **13** is $(20° + \theta°)$ to $(40° + \theta°)$, $(60° + \theta°)$ to $(80° + \theta°)$, etc. in FIG. **4**A. The interior of the cylindrical bore **16** at this time is at a low pressure.

The curve of variation at this time of the displacement volume determined by the spacing between the right and left ball pistons **17**a and **17**b and by their respective stroke directions is as shown in FIG. **4**C. The strokes of the ball pistons **17**a and **17**b at the point of angle θ in FIGS. **4**A and **4**B are equal. At this point, furthermore, changeover of the valve timing is carried out. For this reason, as indicated in FIG. **4**C, the curve of variation of the displacement volume is always a continuous curve of a normal hump or peak shape which is on the positive side with respect to the vertical coordinate, and there is no possibility of occurrence of negative parts and discontinuous parts in the curve as shown in FIG. **5**C. Accordingly, the fluid device **10** always operates as a normal motor or pump.

The displacement volume varies in accordance with the selection of the value of the rotational angle 2θ of the cam plate **21**, and, with increase in the angle 2θ, the displacement volume decreases in the manner indicated by the dashed lines in FIG. **4**C. In the instant embodiment, the displacement volume becomes zero when the angle 2θ is 40°.

While the cam plates **20** and **21** both receive reaction forces due to the ball pistons **17**a and **17**b, there is no occurrence of deviation in the position one of the cam plates (cam plate **20**) since this cam plate **20** is fixed. Accordingly, by merely constructing only the movable cam plate **21** so that it will not be subject to positional deviation due to reaction force, it is always possible to select accurately the mutual positions of the cam plates **20** and **21** and to accomplish accurate variation and adjustment of the capacity. Furthermore, since the valve sleeve **24** is never subjected to a reaction force due to the rotation of the rotor **13**, there is no necessity of strict specifications in the construction of parts such as the gear wheel **41** and the shaft **42**.

Another embodiment of the mechanism for transmitting the rotation of a cam plate to the valve sleeve is illustrated in FIG. **6**. A cam plate **50** of this embodiment corresponds to the movable cam plate **21** of the preceding embodiment, and has gear teeth **51** on the inner side of its end. These gear teeth **51** are meshed with a gear wheel **52** supported rotatably on a vertical shaft **53** supported on the outer peripheral surface of the valve sleeve **24** in the vicinity of the end thereof. The gear wheel **52** is further meshed with a gear teeth **54** formed in the housing **11**. Rotation of the cam plate **50** causes the gear wheel **52** to rotate about its axis and, at the same time, to revolve about the axis of the valve sleeve **24**, which is caused to rotate. The mechanism of this embodiment is advantageous in that the radius of the gear wheel **52** relative to the radius of the cam plate **50** can be selected at will.

Further, this invention is not limited to these embodiments but various variations and modifications may be made without departing from the scope and spirit of the invention.

What I claim is:

1. A variable-displacement type fluid device operable as a pump and as a motor, comprising: a casing having a hollow space therein; a rotor for rotation in said space and provided with a plurality of cylindrical bores extending in directions parallel to its axis; a pair of ball pistons provided with a space therebetween within said bores; means for continually urging said pistons in opposite directions, outward from said bores; two cam members respectively disposed on opposed sides of said pistons and having respective cam surfaces against which said pistons abut, one of said cam members being supported rotatably in said casing and having first and second gear portions, the other cam member being fixed to said casing; a valve sleeve provided rotatably within said casing in coaxial confrontal relation to a rotational outer peripheral surface of said rotor; ports formed in said casing for selectively supplying and exhausting, as well as for suction and discharge of, hydraulic fluid for motor and for pump operation; ports formed in said sleeve and said rotor, operating in accordance with the rotation of the latter, selectively to place a specific one of said casing ports in communicative and non-communicative state with said space between said pistons; a first gear for meshing with said first gear portion and for rotating said one cam member; a shaft extending through said casing for rotating said first gear; a second gear supported rotatably in said sleeve, for gear meshing with said second gear portion; and a fixed gear part provided on the inner side of said casing; said second gear being rotated about its own axis and revolved in orbit motion with said sleeve by the rotation of said one cam member, and said sleeve being rotated by means of said second gear by an angle which is one half of the rotational angle of said one cam member.

2. The fluid device as defined in claim **1**, wherein said shaft is extended through said casing in the same plane as the rotating plane of said one cam member, and said shaft is provided with a knob at the outermost end thereof.

3. The fluid device as defined in claim **1**, wherein said casing has therein a boss part on which said shaft is journaled, so that said one cam member rotates about said boss part within said casing.

4. The fluid device as defined in claim **1**, wherein said second gear has a rotational shaft and is rotatably supported at the end surface of said sleeve so that said shaft is parallel to the axis of rotation of said rotor, and that said second gear rotates in a plane parallel to the plane of rotation of said one cam member.

5. The fluid device as defined in claim **1**, wherein said second gear has a rotational shaft and is rotatably supported on the outer peripheral surface of said sleeve in the vicinity of one end thereof so that said shaft is perpendicular to the axis of rotation of said rotor, and that said second gear rotates in a plane perpendicular to the plane of rotation of said one cam member.

6. The fluid device as defined in claim **1**, comprising a plurality of through holes formed through said rotor, parallel with said bores, and extending in directions parallel to the axial direction, said holes providing communication between cavity spaces within said casing at the two ends of said rotor for flow of the hydraulic fluid which has leaked, and to reduce the moment of inertia of said rotor.

7. The fluid device as defined in claim **6**, further comprising an outlet for discharging leaked hydraulic fluid, communicating said cavity space on one side of said rotor with the outside of said casing.

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