Oct. 27, 1964 T. H. FIKSE 3,153,984

VARIABLE-VOLUME FLUID MOTOR

Filed March 30, 1962 3 Sheets-Sheet 1

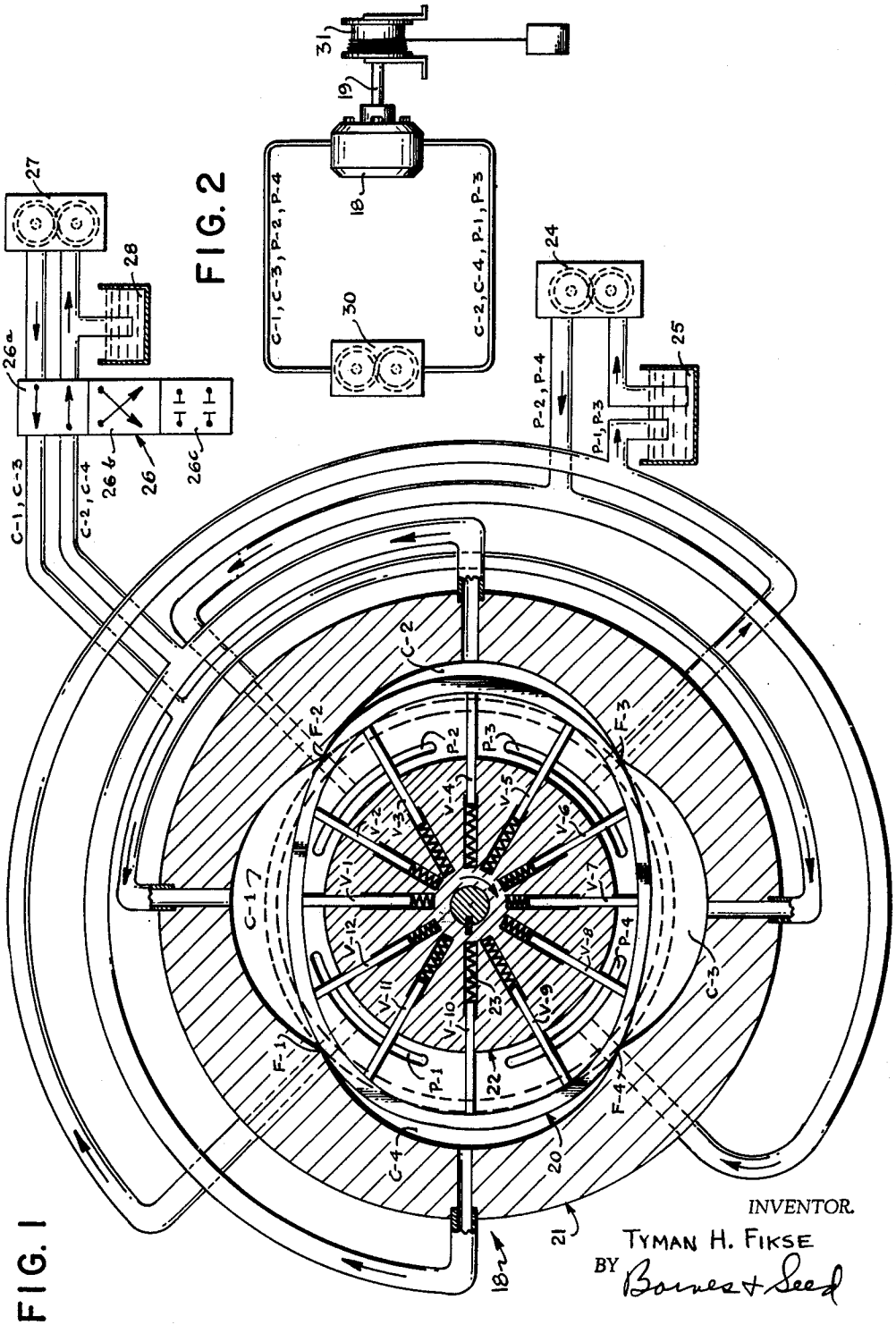


FIG. 2

FIG. 1

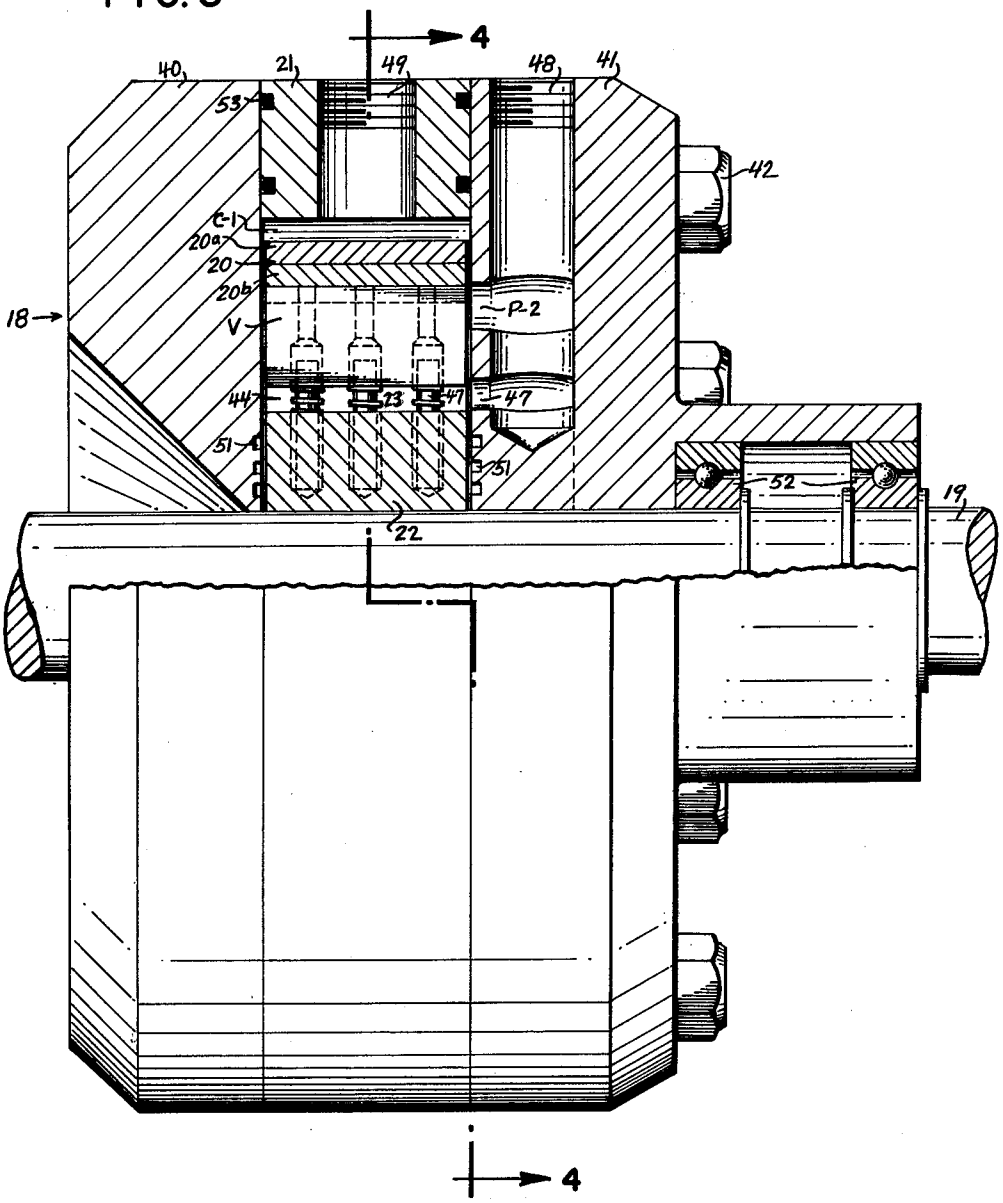
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FIG. 3



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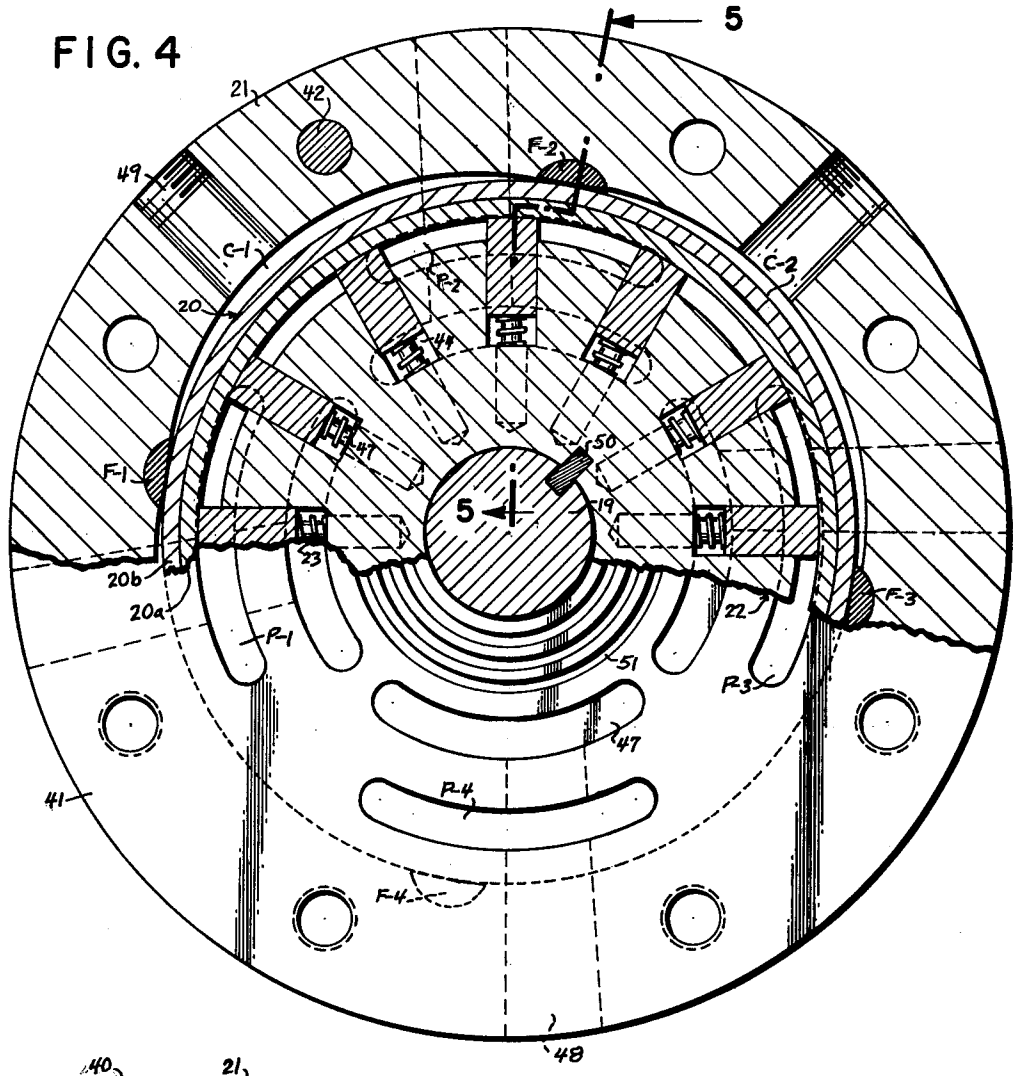
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FIG. 4



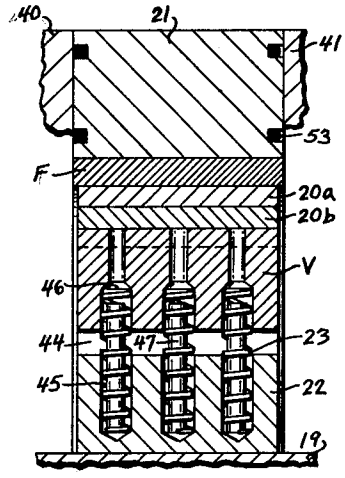


FIG. 5

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United States Patent Office 3,153,984 Patented Oct. 27, 1964

3,153,984

VARIABLE-VOLUME FLUID MOTOR

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Filed Mar. 30, 1962, Ser. No. 183,820
8 Claims. (Cl. 91—75)

The present invention relates to a variable-volume fluid motor of a type in which the fluid volume per revolution is determined by the controlled deformation of a spring ring surrounding and constantly engaged by vanes on the rotor of the motor.

The invention aims to provide an improved variable-volume fluid motor of particularly simple and economical construction whose output and direction of rotation can be readily controlled, and particularly remotely controlled by pressurized fluid.

A further object is to provide an improved fluid motor-winch arrangement in which the torque of the motor automatically adjusts to winch load.

With these and still additional objects and advantages in view, the invention consists in the novel construction and in the adaptation and combination of parts hereinafter described and claimed.

In the accompanying drawings:

FIGURE 1 is a schematic view of a preferred embodiment of my invention shown in transverse vertical section and illustrating a remote control fluid layout.

FIG. 2 is a schematic showing my improved fluid motor-winch arrangement.

FIG. 3 is a side elevational view partly in longitudinal vertical section of the motor.

FIG. 4 is a transverse vertical sectional view taken as indicated by line 4—4 of FIG. 3.

FIG. 5 is a fragmentary longitudinal sectional view taken along the line 5—5 of FIG. 4.

Attention is first directed to the schematic of FIG. 1 in which the fluid unit 18 has a spring ring 20 confined between four equally spaced fulcrums F–1 to F–4. These fulcrums separate four chambers C–1 to C–4 in a housing 21. Opposite sets of these chambers may be supplied with pressurized fluid to deform the ring 20 to a generally elliptical shape. For example, when chambers C–1 and C–3 are pressurized the ring 10 takes the elliptical form shown in FIG. 1 with its minor axis between such chambers and its major axis between chambers C–2 and C–4. A rotor 22 is journaled in the housing 21 and has radially movable vanes V–1 to V–12 each of which is yieldingly urged as by springs 23 against the ring 20 to divide a gap between the rotor and the ring into pockets. These pockets consequently communicate with fluid ports P–1 to P–4 in the housing which are centered relative to respective of the fulcrums F–1 to F–4. Ports P–1 and P–3 always have fluid under one common pressure whereas the fluid at ports P–2 and P–4 is under a second pressure. For practical purposes the effects of these pressures on the spring ring can be ignored because their resultant forces act through the respective fulcrums. For example, the pressure on the ring from fluid at port P–2 is centered on fulcrums F–2 and is equal and opposite to the pressure on the ring from fluid at P–4, the latter being centered on fulcrum F–4.

When the unit acts as a hydraulic motor to drive the rotor shaft 19, ports P–2 and P–4 are supplied with pressurized fluid from a constant volume pump 24 while ports P–1 and P–3 dump to a reservoir 25. In such a case, vanes V–4 and V–10 are urged clockwise by the pressures in ports P–2 and P–4, respectively, while vanes V–1 and V–7 are urged counterclockwise by said pressures. The rest of the vanes, being subjected on opposite sides to pressure from only one of the ports, are balanced. Since, the

distance from the rotor to the ring at vanes V–4 and V–10 (the major axis) is greater than at vanes V–1 and V–7 (the minor axis), the total force acting on vanes V–4 and V–10 in the clockwise direction is greater than the total force acting on vanes V–1 and V–7 in the opposite direction. Accordingly, there is a clockwise net torque exerted on the rotor which is directly proportional to the difference between the major and minor axes of the deformed ring 20. This difference is in turn proportional to the excess of the pressure in chambers C–1 and C–3 over that in chambers C–2 and C–4. If these pressures are reversed, thereby reversing the major and minor axes of the deformed ring, the net torque acting on the rotor will be in the counterclockwise direction.

As indicated in FIG. 1, the unit can be connected hydraulically with a four-way pilot valve 26. One position 26*a* of this valve pressurizes chambers C–1 and C–3 from a constant volume pump 27 and communicates chambers C–2 and C–4 with the sump 28. The second position 26*b* of the valve pressurizes chambers C–2 and C–4 and connects chambers C–1 and C–3 with the sump, and the third position 26*c* blocks off the four chambers. Thus, position 26*a* gives clockwise rotation, and position 26*c* is neutral when ports P–2 and P–4 are connected with the supply 24 of pressurized fluid and ports P–1 and P–3 communicate with the reservoir 25.

Another application of the unit 18 as a motor is shown in FIG. 2, in which ports P–2 and P–4 and chambers C–1 and C–3 are all connected with the output side of a constant volume hydraulic pump 30 while the input side of the latter is connected with the other ports and chambers, The rotor shaft 19 of the unit is coupled to a winch drum 31. With this arrangement the load on the winch resists clockwise turning of the rotor and this resistance causes the pressure to rise at ports P–2 and P–4 and hence to also rise in chambers C–1 and C–3. Such a rise results in deformation of the ring 20 as before described, and hence results in a clockwise torque which continues to increase until it overcomes the load on the winch. As the motor 18 then begins to turn its speed will be universely proportional to the torque requirement because the torque output is a direct function of the displacement of the hydraulic motor.

Continuing to a description of a more detailed embodiment of the hydraulic motor, and directing attention particularly to FIGURES 3–5 it is seen that the rotor 22 is housed at its ends by front and back plates 40–41 between which a fulcrum ring 21 is held by bolts 42 in radially spaced circumscribing relation to the rotor. Inner and outer flexible spring rings 20*a*–20*b* function as a single spring ring 20 in the gap between the rotor 22 and the fulcrum ring 21. The inside face of the spring ring 20 is engaged by the vanes V of the rotor while the outside face of the spring ring 20 bears against four equally spaced rockers, F–1 through F–4. These rockers F are segmental in transverse section and seat in correspondingly shaped grooves formed along the inside face of the fulcrum ring 21. This inside face is not cylindrical but has its quadrants struck from respective centers which are offset a common distance toward the quadrants from the center axis of the fulcrum ring. As a result the chambers C–1 through C–4 are formed. In its relaxed state the spring ring 20 is circular with its outer diameter slightly greater than the distance between diametrically opposite of the rockers F. Accordingly, the spring ring is force fitted into operative position.

The rotor vanes V are mounted for radial sliding movement in pockets 44 which extend the length of the rotor and each have a series of wells 45 to complement wells 46 in the inner face of the vanes. These wells receive the springs 23 which are mounted on cylindrical guides 47 and urge the vanes outwardly against the spring ring 20.

The outer end face of the vanes is transversely rounded somewhat at its longitudinal edges so that the inside face of ring 20 will not be scored thereby. In a manner to be later explained, the springs 23 are aided by pressurized fluid constantly supplied to the ends of the pockets 44.

The inside face of the end plate 41 has four equally spaced arcuate grooves cut therein on a common radius to form the ports P–1 through P–4, and these are complemented by four additional arcuate grooves 47 having a lesser radius such as to intersect the opposing end of the vane pockets 44 to serve as ports for the latter. In each quadrant of the plate 41 the respective ports P and 47 are intersected by a radial passage 48. Similarly each chamber C has a radial supply passage 49 in the fulcrum ring 21. These passages 48 and 49 are either respectively interconnected within the plate 41 and ring 21 in the desired manner or extend to the periphery thereof to be externally interconnected as indicated by the schematics of FIGS. 1 and 2. It will be apparent that the other end plate 40 can also be provided with the porting and passages of the plate 41 in which case the corresponding passages 48 in the two end plates can be connected by longitudinal passages in the fulcrum ring or externally connected.

The rotor is keyed at 50 to an output shaft 19 and the end plates 40–41 are provided with opposed circumferential grooves 51 serving as a labyrinth seal. Bearings 52 for the shaft 19 are mounted in the end plate 41. It will be noted that the end plates 40–41 and fulcrum ring 21 are sealed by O-rings 53 and collectively serve as a housing which can be mounted in any suitable manner.

The two ring components of the spring ring 20 can be made of stainless steel, S.A.E. 4140 for example. Experience has shown that repeated deflections of the spring ring while the rotor and vanes are turning causes its components to rotate a small increment during each flexure cycle. This feature greatly improves the wear life of the spring ring by dispersing the stress and wear patterns. In the schematic of FIG. 1 the deflection of the spring ring 20 has been greatly exaggerated for clarity of illustration. The flexure deflection range in each direction need only be about 3% of the ring diameter in actual practice.

Although the present invention has been described in terms of use as a hydraulic motor it will be apparent that if the shaft 19 is driven the unit will serve as a pump. For example, if the rotor 22 is driven counter-clockwise when viewed as in FIG. 1, and the spring ring 20 is deformed as therein shown due to superior pressures in chambers C–1 and C–3, ports P–2 and P–4 will become the pump outlet and ports P–1 and P–3 the pump inlet. Changing of the valve 26 to position 26*b* will thereupon reverse the pump inlet and outlet, and movement to position 26*c* will inactivate the pump.

It is thought that the advantages of the invention will have been clearly understood from the foregoing detailed description of the illustrated embodiments. Minor changes in the details of construction will suggest themselves and may be resorted to without departing from the spirit of the invention, wherefore it is my intention that no limitations be implied and that the hereto annexed claims be given a scope fully commensurate with the broadest interpretation to which the employed language admits.

What I claim is:

1. In combination, a housing, a rotor journal-mounted in said housing and having a plurality of outwardly projecting and circumferentially spaced vanes which are movable inwardly and outwardly relative to the rotor and are yieldingly urged outwardly, flexible ring means circumscribing said rotor in radially spaced relation thereto and engaged by said vanes, two pairs of diagonally opposite fulcrum means in said housing engaged by said ring means for restraining expansion of the ring means thereat, two pairs of diagonally opposite chambers in said

housing between said fulcrum means exposed to the outer surface of said spring means, two pairs of diagonally opposite ports in said housing located endwise of said ring means and exposed to the space between said rotor and ring means, each said pair of ports being generally centered circumferentially of the rotor relative to a respective pair of said fulcrum means and said ports being spaced apart sufficiently to be isolated from one another by said vanes, means for supplying fluid to one of said pairs of ports and for discharging said fluid from the other pair of ports, and control fluid means in said chambers for exerting a pressure differential as between said pairs of chambers for deforming said ring means between said fulcrum means to a generally elliptical shape, thereby determining the fluid flow per revolution of the rotor.

2. The structure of claim 1 in which said control fluid means includes means for varying said pressure differential to thereby vary said deformation of the ring means and responsively vary the fluid flow per revolution of the rotor.

3. The structure of claim 1 in which said control fluid means includes means for reversing said pressure differential to thereby responsively reverse the direction of rotation of the rotor.

4. The structure of claim 1 in which each of said fulcrum means comprises a rocker seated in a respective groove in said housing.

5. In combination, a housing, a rotor journal-mounted in said housing and having a plurality of outwardly projecting and circumferentially spaced vanes which are movable inwardly and outwardly relative to the rotor and are yieldingly urged outwardly, flexible ring means circumscribing said rotor in radially spaced relation thereto and engaged by said vanes, two pairs of diagonally opposite fulcrum means engaged by said ring means for restraining expansion of the ring means thereat, two pairs of diagonally opposite ports in said housing located endwise of said ring means and exposed to the space between said rotor and ring means, each said pair of ports being generally centered circumferentially of the rotor relative to a respective pair of said fulcrum means and said ports being spaced apart sufficiently to be isolated from one another by said vanes, means for supplying fluid to one of said pairs of ports and for discharging said fluid from the other pair of ports, and control means for deforming said ring means between said fulcrum means to a generally elliptical shape thereby determining the flow rate per revolution of the rotor.

6. The structure of claim 5 in which said control means comprises pressurized fluid acting on the outer surface of said ring means at diagonally opposite areas between said fulcrum means.

7. In combination, a housing, a rotor journal-mounted in said housing and having a plurality of outwardly projecting and circumferentially spaced vanes which are movable inwardly and outwardly relative to the rotor and are yieldingly urged outwardly, flexible ring means circumscribing said rotor in radially spaced relation thereto and engaged by said vanes, two pairs of diagonally opposite fulcrum means in said housing engaged by said ring means for restraining expansion of the ring means thereat, two pairs of diagonally opposite chambers in said housing between said fulcrum means exposed to the outer surface of said ring means, two pairs of diagonally opposite ports in said housing located endwise of said ring means and exposed to the space between said rotor and ring means, each said pair of ports being generally centered circumferentially of the rotor relative to a respective pair of said fulcrum means and said ports being spaced apart sufficiently to be isolated from one another by said vanes, means for supplying fluid at a constant flow rate to one of said pairs of ports and for discharging said fluid from the other pair of ports, and control fluid means selectively communicating with said pairs of chambers for selectively exerting a pressure differential as between said pairs

of chambers and responsively deforming said ring means between said fulcrum means to a generally elliptical shape thereby determining the fluid flow per revolution of the rotor.

8. In combination, a housing, a rotor journal-mounted in said housing and having a plurality of outwardly projecting and circumferentially spaced vanes which are movable inwardly and outwardly relative to the rotor and are yieldingly urged outwardly, flexible ring means circumscribing said rotor in radially spaced relation thereto and engaged by said vanes, two pairs of diagonally opposite fulcrum means in said housing engaged by said ring means for restraining expansion of the ring means thereat, two pairs of diagonally opposite chambers in said housing between said fulcrum means exposed to the outer surface of said ring means, two pairs of diagonally opposite ports in said housing located endwise of said ring means and exposed to the space between said rotor and ring means, each said pair of ports being generally centered circumferentially of the rotor relative to a respective

pair of said fulcrum means and said ports being spaced apart sufficiently to be isolated from one another by said vanes, means for supplying fluid at a constant flow rate to one of said pairs of ports and for discharging such fluid from the other pair of ports, and means for intercommunicating one of said pairs of chambers with said one pair of ports for creating a pressure differential as between said pairs of chambers for deforming said ring means between said fulcrum means to a generally elliptical shape thereby determining the fluid flow per revolution of the rotor.

References Cited in the file of this patent

UNITED STATES PATENTS

1,190,139	Ford	July 4, 1916
2,673,448	Wheeler	Mar. 30, 1954
2,691,482	Ungar	Oct. 12, 1954
2,842,064	Wahlmark	July 8, 1958
2,846,984	Zwayer	Aug. 12, 1958
3,054,389	Roggenburk	Sept. 18, 1962