May 23, 1972 A. D. MILLS ET AL 3,664,776

VARIABLE VOLUME VANE PUMP

Filed Aug. 17, 1970 4 Sheets-Sheet 1

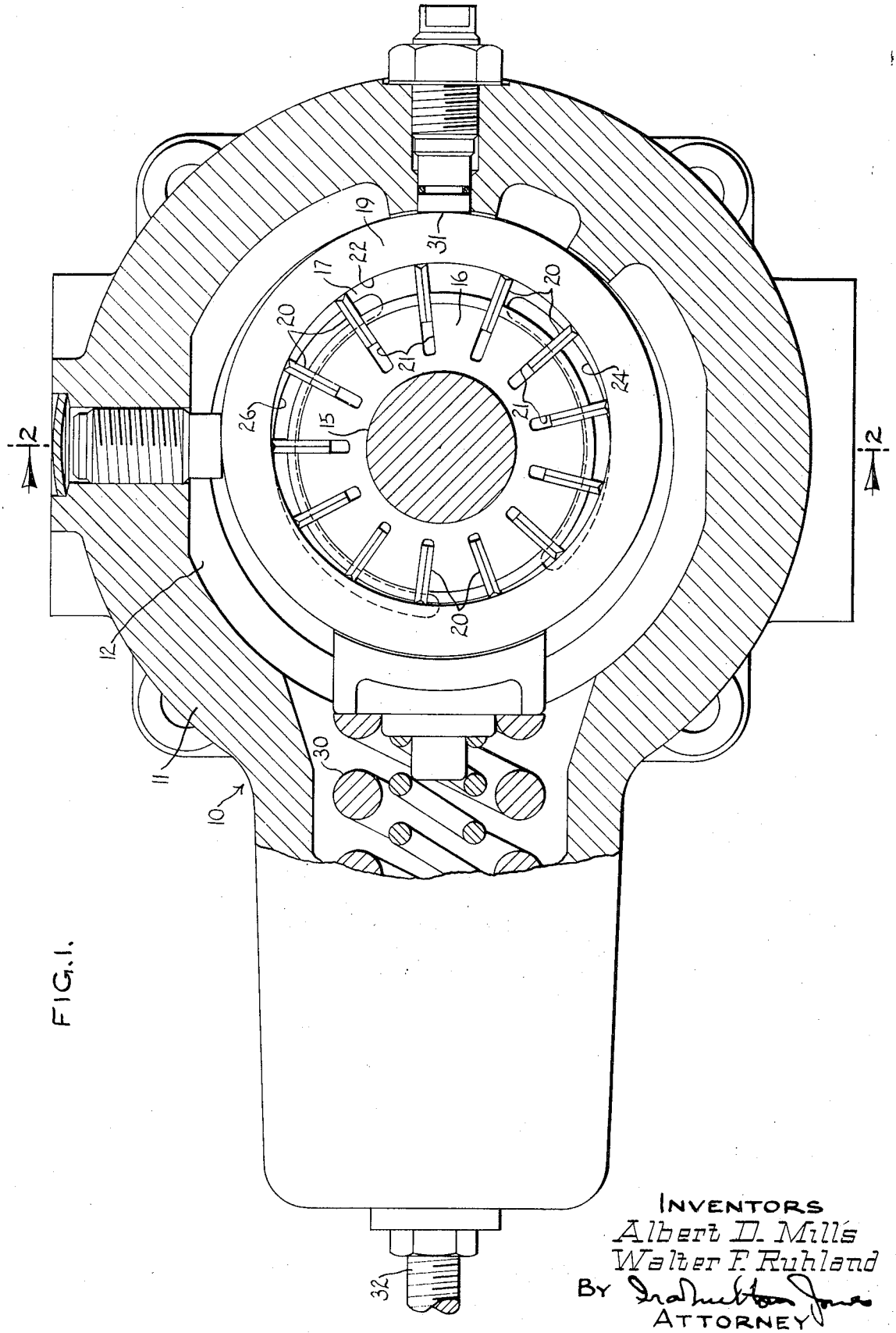


FIG. 1.

INVENTORS
Albert D. Mills
Walter F. Ruhland
BY
ATTORNEY

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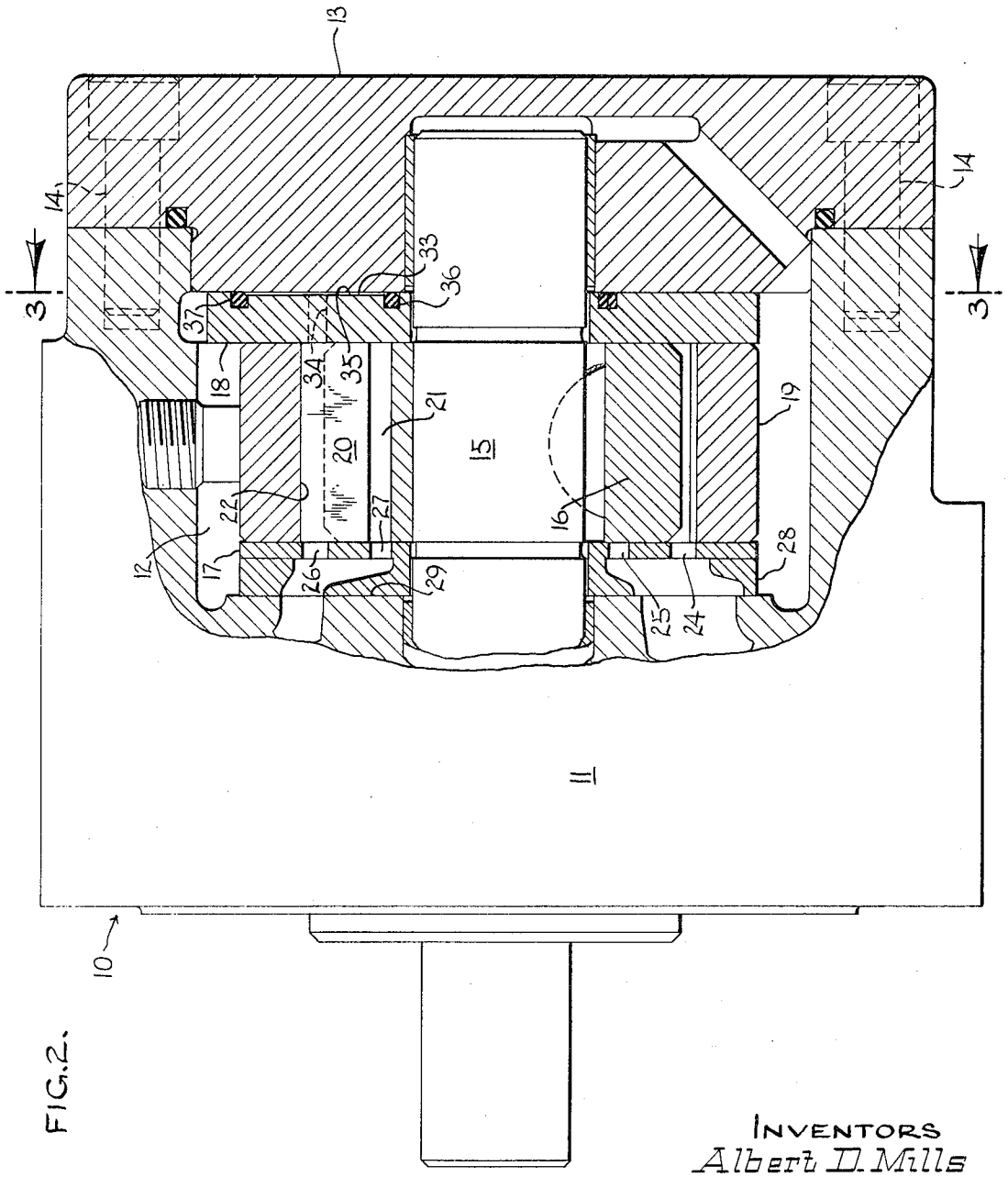


FIG. 2.

INVENTORS
Albert D. Mills
Walter F. Ruhland
BY Fred Hilton Jones
ATTORNEY

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VARIABLE VOLUME VANE PUMP

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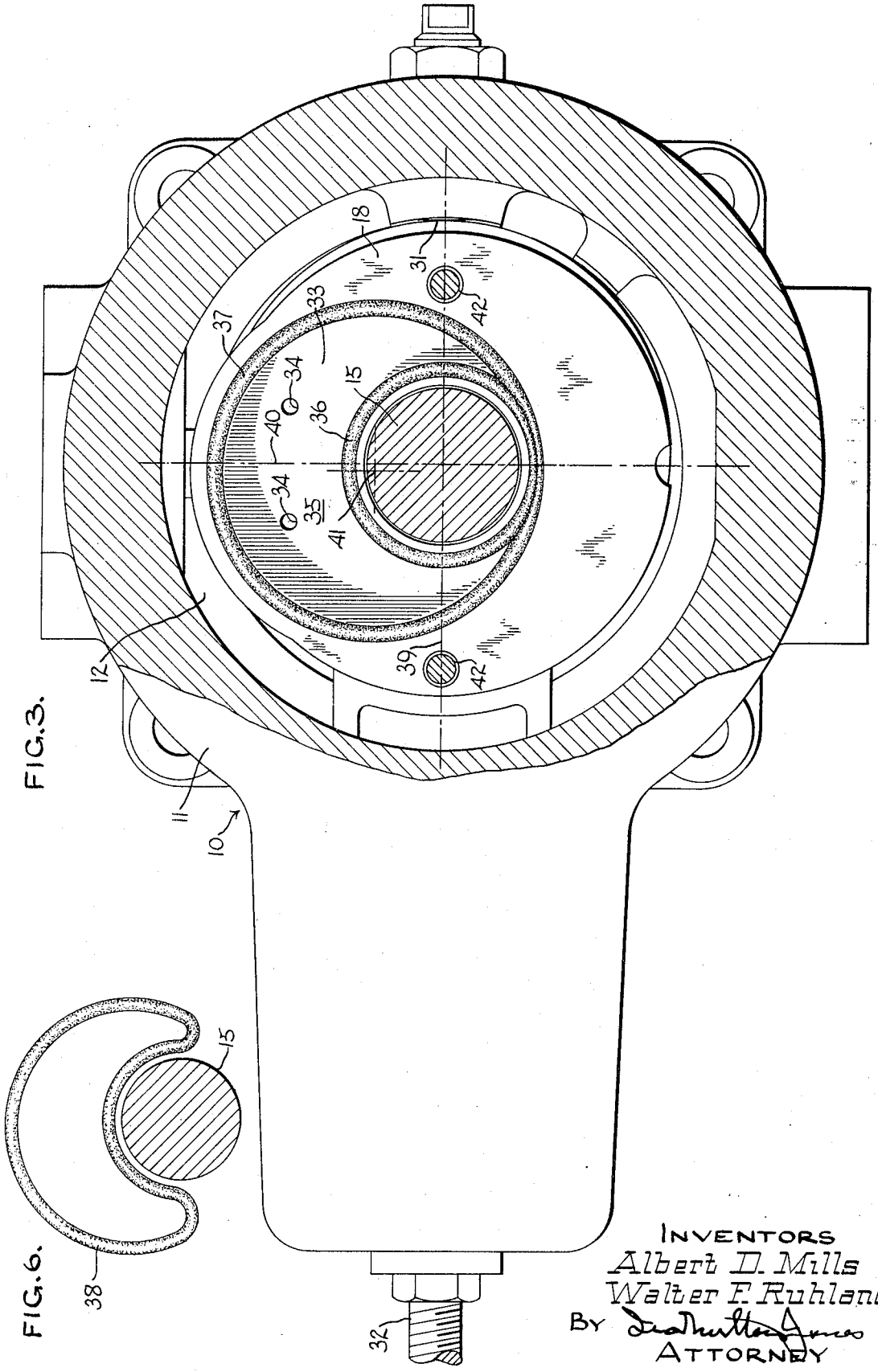


FIG. 3.

FIG. 6.

INVENTORS
Albert D. Mills
Walter F. Ruhland
BY
ATTORNEY

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VARIABLE VOLUME VANE PUMP

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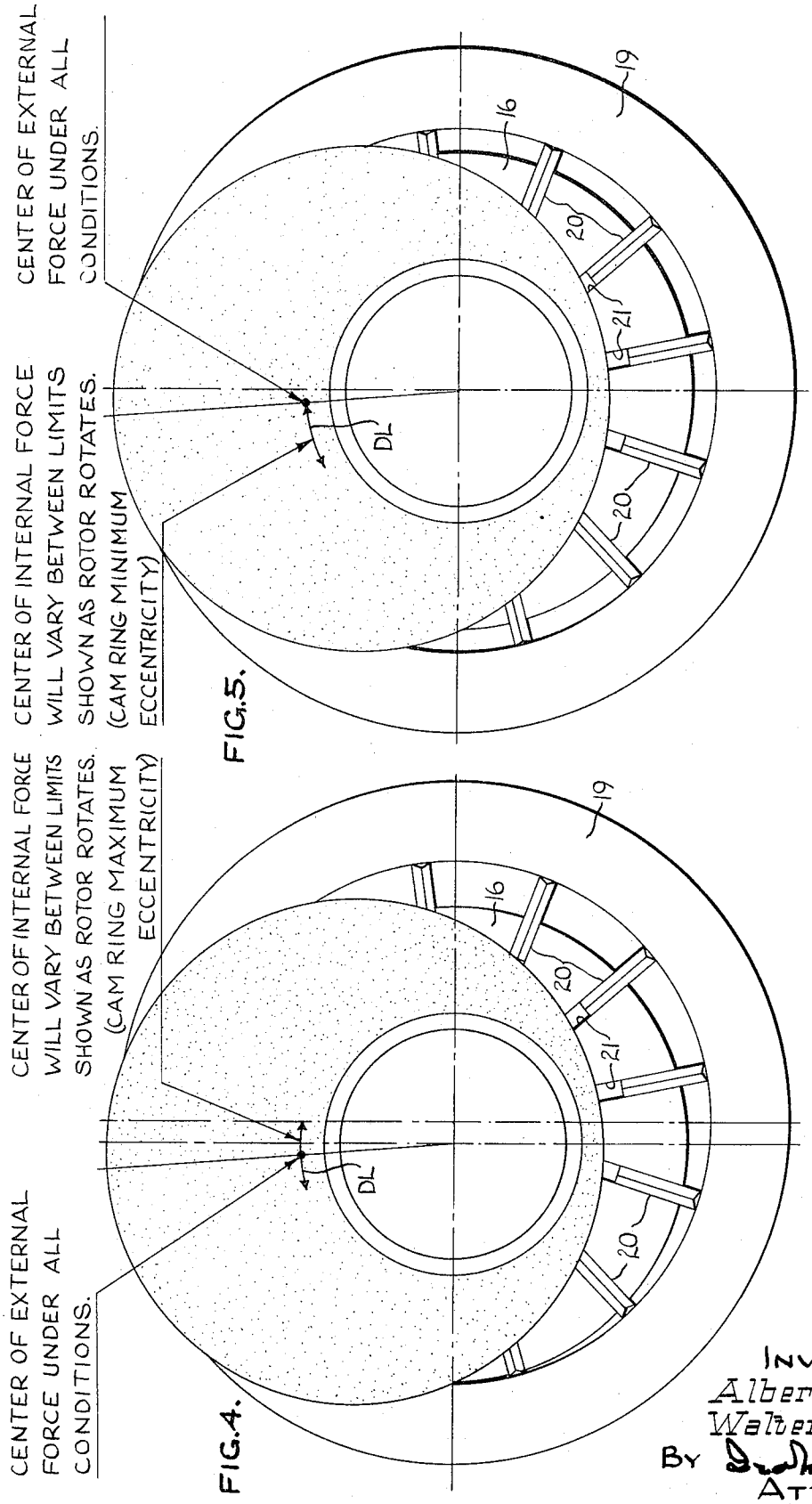


FIG. 5.

FIG. 4.

INVENTORS
Albert D. Mills
Walter F. Ruhland
BY
ATTORNEY

United States Patent Office 3,664,776 Patented May 23, 1972

3,664,776

VARIABLE VOLUME VANE PUMP

Albert D. Mills, Minneapolis, and Walter F. Ruhland, Shakopee, Minn., assignors to Continental Machines, Inc., Savage, Minn.

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Int. Cl. F01c 19/08, 21/16; F04c 15/00
U.S. Cl. 418—16 8 Claims

ABSTRACT OF THE DISCLOSURE

In a sliding vane pump having end plates a rotor with its vanes and a movable cam ring which eccentrically surrounds the rotor during pumping. One of the end plates is hydraulically urged toward the other by fluid at pump outlet pressure exerted on its outer side at a zone thereof axially opposite the pump outlet zone. The area on which such fluid exerts inward force is crescent shaped, defined by two seal rings, one encircling the rotor shaft, the other larger and encircling the smaller one.

BACKGROUND OF THE INVENTION

This invention relates to variable volume sliding vane pumps and refers more particularly to pumps to that type that have pressure loaded end plates.

A vane pump of the type with which this invention is concerned has a rotor which rotates on a fixed axis within a radially shiftable cam ring, between a pair of parallel end plates. The vanes are radially slidably mounted in slots in the rotor that open to its periphery, and they project outwardly beyond the rotor to engage the inner surface of the cam ring. The inner surfaces of the end plates are contiguous to the opposite axial ends of the cam ring, the rotor, and the vanes carried by the rotor; and the end plates cooperate with the rotor and the cam ring to define a pump assembly.

As long as the cam ring is eccentric to the rotor, the pumping chambers between the outwardly projecting end portions of the vanes are caused to alternately increase and decrease in volume as the rotor rotates. The chambers progressively increase in volume for one-half a revolution of the rotor, during which they travel through an arcuately elongated inlet zone at one side of the rotor axis, and they then progressively decrease in volume for the next half revolution of the rotor, during which the chambers travel through an arcuately elongated outlet zone at the opposite side of the rotor axis. Fluid to be pumped is sucked into the expanding chambers in the inlet zone, and fluid at elevated pressure is expelled from the contracting chambers in the outlet zone.

The cam ring is shiftable toward and from what is commonly referred to as a deadhead position substantially concentric to the rotor axis, to vary the volumetric output of the pump. While the invention is especially useful in variable volume, pressure compensated vane pumps wherein the outlet pressure is automatically maintained at a substantially constant level even though the volumetric demands of the system are varying it is by no means limited thereto. Those skilled in the art will recognize that the invention applies equally as well to variable displacement vane pumps with other types of volumetric control such as manual, servo controlled, hydraulic pilot controlled, etc.

Ordinarily, one of the end plates constitutes a port disc, having arcuately elongated inlet and outlet ports located at diametrically opposite sides of the rotor axis. Generally speaking, these ports define the inlet and outlet zones, respectively, of the pump assembly.

In all vane type pumps of this nature, the internal or

system fluid pressure of the pump assembly, in the outlet zone of the pump, imposes force on the end plates tending to spread them apart and away from the adjacent axial ends of the rotor and the cam ring encircling it. When this occurs, a leakage path is created along which high pressure fluid escapes from the pump assembly past the ends of the cam ring, into the space in the interior of the pump housing surrounding the assembly. Such leakage of high pressure fluid, which is commonly referred to as case drain leakage, if excessive, objectionably lowers the efficiency of the pump.

In the past, shims were frequently placed behind at least one of the end plates to maintain the end plates snugly engaged against the axial ends of the cam ring, the rotor and the vanes carried by the rotor, hopefully without interfering with rotor rotation and/or radial shifting of the cam ring toward and from concentricity with the rotor. This expedient, however, required shimming of the pump assembly to a specific operating clearance.

As is well known, the use of shims to prevent objectionable leakage of high pressure fluid from the pump assembly left much to be desired, and was at best only a partial solution to the leakage problem. The fixed clearance provided by shims could be at an optimum value for only one pump output pressure, as determined, in a variable volume pressure compensated pump, by the adjustable spring force resisting shifting of the cam ring toward concentricity with the rotor.

Attempts have been made to utilize hydraulic force derived from pressure fluid at the pump outlet, to force one of the end plates toward the other and thus counteract the internal pressure that heretofore tended to separate the plates and cause leakage past the ends of the cam ring. These past attempts have not been successful, and in some cases necessitated the provision of a fixed spacer ring encircling and radially spaced from the exterior of the cam ring, to limit relative motion of the end plates toward one another with a view to preventing undue frictional forces on the rotor and cam ring and the consequent interference with radial shifting of the cam ring.

SUMMARY OF THE INVENTION

It is the purpose of this invention to solve the leakage problem discussed hereinabove, in a variable volume pressure compensated vane pump wherein the end plates are held against separation from the cam ring and the rotor and its vanes as a consequence of the imposition of hydraulic force on predetermined areas of one of the end plates.

More particularly, it is an object of the invention to provide for the imposition of fluid pressure force, derived from the high pressure zone of the pump assembly, upon the outer side or face of one of the end plates, but only at an area thereof which is remote from the inlet zone and axially opposite the outlet or pressure zone of the pump assembly.

In a more specific sense, it is another purpose of the invention to define the shape, size and location of the hydraulically pressurized area on the outer side of the pressure loaded end plate by means of a pair of seal rings, one of which surrounds the other and both of which are confined between the pressure loaded end plate and a wall of the housing adjacent thereto; and which seal rings also serve to bias the pressure loaded end plate against the cam ring and the rotor when the pump is not in operation, to prevent excessive leakage between the cam ring and the thrust plate when the pump is started up.

Still another object of the invention is to achieve the desired pressure loading of one of the end plates in a way that automatically minimizes leakage of fluid during full volume operation of the pump while assuring suffi-

cient case drain leakage for adequate cooling when the pump is operating in its deadhead condition.

With these observations and objectives in mind, the manner in which the invention achieves its purpose will be appreciated from the following description and the accompanying drawings, which exemplify the invention, it being understood that such changes in the specific apparatus disclosed herein may be made as come within the scope of the appended claims.

The accompanying drawings illustrate one complete example of the embodiment of the invention constructed according to the best mode so far devised for the practical application of the principles thereof, and in which:

FIG. 1 is a cross sectional view through a variable volume pressure compensated vane pump embodying this invention;

FIG. 2 is a view partly in elevation but mainly in section taken on the line 2—2 in FIG. 1;

FIG. 3 is a cross sectional view taken on the line 3—3 in FIG. 2;

FIGS. 4 and 5 are diagrammatic views similar to FIG. 3, illustrating the difference in positional relationship between the centers of internal and external force acting upon the thrust plate when the operation of the pump is at full volume output and at deadhead, respectively; and

FIG. 6 is a fragmentary view showing a modified embodiment of certain details of the pump structure.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the accompanying drawings, the numeral **10** generally designates the housing of the pump of this invention. The housing is comprised of a body **11** having a cavity **12** therein opening to one end of the body, and a cover **13** which is secured to said end of the body by screws **14** to close the body cavity **12**.

The pump shaft **15** extends lengthwise through the body and is rotatably journalled in bearings carried by the body and the cover **13** for rotation on a fixed axis. The body cavity **12** accommodates a pump assembly comprising a rotor **16** mounted on the shaft **15** between a pair of end plates **17** and **18** that are centrally bored to receive the opposite ends of the shaft, and a cam ring **19** which encircles the rotor and is also axially confined between the end plates **17** and **18**. As will be described later the inlet and outlet ports of the pump are in the end plate **17**. Hence this end plate is usually referred to as a port plate, and the other as the thrust plate.

The pump described thus far is similar to that disclosed in the Whitmore et al. Patent No. 3,252,423, issued May 24, 1966. As in that patent, the rotor **16** also carries a number of vanes **20** which are radially slidably received in circumferentially spaced slots **21** that open to the periphery of the rotor. The vanes project outwardly beyond the rotor to have their outer edges track on the inner surface **22** of the cam ring **19** and their axial edges, which are adjacent to the end plates **17** and **18**, are flush with the opposite axial ends of the rotor.

The rotor construction described provides a series of pumping chambers around the interior of the pump assembly, between adjacent vanes, each bounded by the periphery of the rotor, the port and thrust plates **17** and **18**, and the inner surface **22** of the cam ring **19**. Due to the eccentricity of the cam ring the pumping chambers are caused to progressively increase in volume during rotation of the rotor through first and second quadrants, and then are caused to progressively decrease in volume as the rotor rotates through third and fourth quadrants to complete one full revolution of the rotor.

The chambers travel through an arcuately elongated inlet zone at one side of the rotor axis as their volume increases, to thus suck in fluid through an arcuate inlet port **24** in the port plate **17**. At that time inlet fluid also is drawn into the inner ends of the slots **21** from another inlet **25** in the port plate, to allow the vanes to move outwardly into contact with the inner surface of the cam ring.

The chambers thereafter travel through an arcuately elongated outlet zone at the opposite side of the rotor axis, as their volume decreases, so that fluid can be expelled from the chambers, through an outlet port **26** in the port plate, and from the inner ends of the slots **21** through another outlet port **27**.

A second or outer port plate **28**, confined between the plate **17** and the bottom **29** of the cavity **12**, cooperates with the inner plate **17** to provide a port plate assembly. However, it will be appreciated that a single port plate equal in thickness to the combined thickness of the plates **17** and **28** would serve equally as well as the two plate assembly.

Although the arcuately elongated zones are usually thought of as being the high pressure areas and the suction areas of the pumps, it is to be understood that the high pressure forces which tend to separate the side plates from the rotor do, in fact, extend beyond the arcuately shaped high pressure zone. Some high pressure fluid finds its way radially outwardly and inwardly of the arcuately shaped high pressure zone. Thus it will be assumed that when the high pressure zone is hereinafter referred to it shall be understood to include all of those areas subjected to a positive pressure higher than inlet pressure; and when the term "center of internal force" is used hereinafter it shall be understood to mean the net effective center of all axially acting internal forces generated by the above described pressures.

Maximum volumetric output of the pump is developed when the cam ring is in a position of maximum eccentricity to the rotor, as seen in FIG. 1. A governor spring **30** yieldingly resists radial motion of the ring toward concentricity with the rotor, and tends to maintain the ring engaged with a diametrically opposite stop **31** which adjustably defines the position of maximum eccentricity of the ring. The spring force imposed upon the ring is also adjustable, as by means of a screw **32**.

The cam ring is automatically movable toward concentricity with the rotor as the demand for pump output fluid decreases and tends to create a higher output pressure. The aforementioned Whitmore et al. patent can be referred to for a more detailed discussion of the action of the cam ring and of the forces which control its degree of eccentricity with respect to the rotor axis. It is sufficient to note here that the cam ring must never be held so tightly between the port and thrust plates that the resulting friction forces inhibit its movement toward and from a deadhead position of little or no eccentricity to the rotor. It should also be observed that the cam ring is constrained to move back and forth, toward and from concentricity with the rotor, along a path which extends substantially lengthwise of the inlet and outlet ports **24** and **26**, respectively.

If they were not opposed, the internal forces exerted on the end plates **17** and **18** by pressure fluid in the pumping chambers at the outlet or high pressure zone of the pump assembly would separate the end plates from the axial ends of the cam ring and of the rotor and its vanes. However, in accordance with this invention, one of the end plates, in this case the end plate **18** which is the thrust plate, is externally pressure loaded, i.e. urged toward the port plate **17**, with a force that counterbalances the internal pressure fluid forces acting upon the end plates. Such loading of the thrust plate **18** is accomplished by pressurizing a chamber **33** at its outer side with high pressure fluid from the outlet or high pressure zone of the pump, which reaches the chamber **33** through holes **34** in the thrust plate **18**.

The chamber **33** can comprise a shallow depression or concavity in either the underside of the cover **13** or in the outer side of the thrust plate, as shown, as long as one wall of the chamber is provided by a surface **35** of substantial area on the outer side of the thrust plate. In

the present pump the concavity is in the thrust plate and its bottom provides the surface **35**. Two different sized seal rings **36** and **37** which are set into grooves in the thrust plate **18** and engage the underside of the cover **13** define the boundaries of the surface **35** and coact with the thrust plate and the cover **13** to form the chamber **33**. The seal rings can be of any suitable type, but in the present case they are O-rings, which being made of a resilient rubber-like material, such as buna-N, are maintained under axial compression and thereby preclude excessive leakage when the pump is started up. The smaller one of the O-ring seals, designated **36**, encircles the pump shaft **15**. The other O-ring seal **37** eccentrically encircles the shaft and the smaller O-ring **36**, and has a substantially large diameter. Its eccentricity is such that it cooperates with the smaller O-ring to define a crescent shaped area between them, which area provides the surface **35** to be subjected to pump output pressure.

While the O-rings are tangent to one another in the embodiment of the invention illustrated, they need not be; and in fact the two rings could be replaced by a single appropriately shaped ring **38** as shown in FIG. 6.

It is important to note that the surface **35** is located substantially entirely at the side of the roor axis remote from the inlet **24**, and axially opposite the outlet port **26** and the high pressure or outlet zone of the pump. With reference to FIG. 3, the crescent shaped area of the surface **35** will be seen to be nearly entirely above a horizontal line **39** through the axis of the rotor. It is not exactly centered with respect to a vertical line **40** through the rotor axis, but instead it is offset slightly to the left of the vertical line **40** through the rotor axis, so as to lie at the side of said line that is remote from the cam ring stop **31** as indicated by the center line **41** of the O-ring **37**. The importance of orienting the pressure loaded surface in the manner described above will be explained shortly.

The main purpose of pressure loading, of course, is to minimize leakage of fluid past the ends of the cam ring and from the high pressure zone of the pump assembly to the low pressure zone thereof, past the axial ends of the rotor vanes. Leakage can be minimized by force exerted on the thrust plate in a way that holds the rotor and cam ring substantially snugly between the end plates of the pump assembly. Such force, however, must not be so great as to interfere with rotation of the rotor or with movement of the cam ring toward and from concentricity with the rotor. Nor must the pressure loading force be so great as to preclude case drain leakage in a volume sufficient to keep the pump cool.

With proper sizing and orientation of the pressure loaded surface of the thrust plate, the external force exerted on the thrust plate will never reach a value at which rotor rotation or movement of the cam ring is objectionably impeded. Hence, there will be no need to provide fixed stops for the thrust plate, as was sometimes necessary in the past, to prevent clamping of the rotor and the cam ring too tightly between the end plates.

Moreover, a pressure balanced thrust plate is achieved by the application of fluid pressure force to only that portion of the thrust plate **18** which is axially in line with the high pressure zone in the interior of the pump assembly. Such pressure balance assures that the internal fluid pressure force in the interior of the pump assembly, at the outlet or high pressure zone thereof, will be counterbalanced by the external force exerted on the thrust plate in consequence of pressurization of the chamber **33**; while the absence of substantial external pressure loading force on that portion of the thrust plate which is at the inlet zone assures that the force differential at opposite sides of the thrust plate at that zone will be substantially the same as obtains at opposite sides of the thrust plate at the pressure loaded zone thereof. The balanced forces on the thrust plate thus prevent any tendency for it to cant or tilt when the cam ring is in its positions of high eccentricity and the pump is operating at or near its maximum volumetric output.

Complete balancing of the opposing pressure forces on the thrust plate at high volume pump outputs is assured by having the chamber **33** so located that the center of the pressure loaded surface **35** is a close as possible to being directly opposite the center of the high pressure zone of the pump assembly when the cam ring is at maximum eccentricity to the rotor. This condition is depicted in FIG. 4.

As indicated by the dimension line DL in FIGS. 4 and 5, the center of the internal force oscillates during pump operation towards and from alignment with the center of fluid pressure force acting on the surface **35**.

In the deadhead position of the cam ring (least eccentricity) the conditions diagrammatically illustrated in FIG. 5 prevail, and a small amount of canting or tilting of the thrust plate **18** is intentionally permitted. This follows from the fact that at this time the center of the internal force tending to push the thrust plate **18** away from the rotor is offset slightly from the center of the opposing external thrust force. With the thrust plates **18** thus slightly tilted away from the rotor under the influence of the internal thrust thereon, the leakage of fluid out of the interior of the pump assembly into the interior of the pump housing is increased, which is desirable and essential for cooling of the pump when it is in its deadhead condition.

It should be observed that because of the pressure balance of the thrust plate **18**, and the particular location of the pressure loaded surface on its exterior, the deadhead pressure setting of the pump can be varied while the amount of clearance between the components of the pump assembly is maintained at an optimum value. This follows from the fact that the pressure on the exterior of the thrust plate varies in accordance with the deadhead pressure setting, giving the advantage of automatic maintenance of proper clearances at all times; whereas a fixed clearance established by shimming, as mentioned previously, can be at an optimum value for only one pressure setting.

There is another advantage in locating the pressure loaded surface on the exterior of the thrust plate opposite the outlet zone of high pressure in the pump assembly. Such pressure balancing eliminates the tendency for the thrust plate to cock on pins **42** (shown in FIG. 3) to an extent that would restrict its ability to float in directions parallel to the rotor axis, and therefore the thrust plate can position itself to accommodate unequal thermal expansion of the components of which the pump assembly is comprised. As a result, there is no tendency toward scoring between the rotor and the end plates.

While the pressure in the loading chamber **33** can be kept the same as pump output pressure, it should be observed that a reduced pressure could be maintained in the chamber provided the area of the pressure loaded surface **35** was increased accordingly. This is mentioned because of the possible desirability of effecting a flow of pump output fluid through the chamber **33**, by means of a restricted outlet for it. The fluid thus discharging from the chamber could be used for pump cooling.

It is also to be observed that suction and pressure porting could be formed in the thrust plate **18** if desired, but suitable seals around the ports would then be necessary. Likewise, it would also be possible to pressure load the port plate **17** according to the principles of this invention. This could be done instead of having the plate **18** pressure loaded; or both end plates could be pressure loaded in accordance with the principles of this invention.

Obviously the boundary of the external balancing area can have many different shapes and can be sealed with many different types of sealing elements as long as the total area enclosed by the sealing elements multiplied by the pressure to which the external balancing chamber is subjected, will produce a force substantially equal in mag-

nitude to the total internal axial force tending to separate the end plates. However, whatever may be the specific shape of the external balancing chamber its location should be such that the net effective equivalent single force location lies on substantially the same radial line from the center of the rotor as does the net effective equivalent single internal force when the cam ring is in its maximum eccentric position, and the distance measured radially outwardly from the center of the rotor to these two single equivalent forces should be approximately equal. With this relationship established, the effective centers of the opposing forces will lie on substantially the same radial line when the cam ring is in its position of maximum eccentricity, and the internal and external forces will be most perfectly balanced in both magnitude and location. But when the cam ring approaches concentricity the internal and external forces, although remaining balanced in magnitude, will not act co-linearly and therefore a slight cocking moment will be produced on the thrust plate tilting it slightly to provide an increased case drain leakage; the pump, therefore, will have less case drain leakage when pumping maximum volume than it will when pumping zero or minimum volume.

Nevertheless, adequate cooling will be had at all times, since the oil flowing through the pump and being discharged at the outlet port when the cam ring is displaced from concentricity provides a cooling effect sufficient to keep the internal pump parts cool when the case drain is reduced as cam ring eccentricity is increased.

From the foregoing description, together with the accompanying drawings, it will be readily apparent to those skilled in the art that this invention provides pressure loading for a variable volume vane pump in a way that is superior to past pressure loading expedients.

The invention is defined by the following claims.

We claim:

1. In a variable displacement vane pump having a housing with a pump assembly therein comprising a rotor with circumferentially spaced radially slidable vanes projecting from its periphery, a shiftable cam ring encircling the rotor with a variable degree of eccentricity, and opposing end plates having inner sides contiguous to the axial ends of the ring, the rotor, and the rotor vanes, there being a high pressure zone in the pump, the effective center of the force produced thereby oscillating between defined limits about the axis of the rotor as the eccentricity of the cam ring varies, the improvement by which leakage of fluid during full volume operation of the pump is minimized while assuring sufficient case drain leakage for adequate cooling when the pump is operating in its deadhead condition, and which improvement comprises:

(A) chamber means at the outer side of one of said end plates defining a pressure loading zone for applying an inwardly directed pressure on said one end plate, said chamber means being fixed against oscillation about the rotor axis so that its center of pressure is stationary and said chamber means being so located that

(1) when the cam ring is at its position of maximum eccentricity the effective center of force of the high pressure zone in the pump is substantially directly opposite the stationary center of the pressure loading zone, and

(2) when the cam ring is at its position of minimum eccentricity, the effective center of force in the high pressure zone in the pump is displaced from the stationary center of the pressure loading zone so that the said one end plate tilts and allows increased case drain; and

(B) means for supplying pressurized fluid to said chamber means.

2. In a variable displacement vane pump, the improvement set forth in claim 1, wherein said means for supplying pressurized fluid to the chamber means comprises at least one hole in said one end plate, opening to the opposite sides thereof adjacent to the high pressure zone in the pump.

3. In a variable displacement vane pump the improvement set forth in claim 1, wherein a wall of said chamber means is provided by part of the surface of the outer side of said one end plate, and wherein the area of said part of the end plate surface is such that the force acting thereon as a result of the pressure fluid in the chamber means is susbtantially equal to the force applied on said one end plate by the high pressure in the pump.

4. In a variable displacement vane pump, the improvement set forth in claim 1, further characterized by:

(A) the pump having a shaft upon which the rotor is mounted and which has a portion projecting through a centrally disposed bore in said one end plate;

(B) a pair of circular seals confined between the outer side of said one end plate and a wall of the housing adjacent thereto,

(1) one of said seals having a small diameter and substantially closely encircling said projecting shaft portion,

(2) and the other seal having a substantially larger diameter and eccentrically encircling said projecting shaft portion and the smaller seal; and

(C) said seals co-acting with the outer side of said one end plate and said adjacent wall of the housing to form said chamber means.

5. In a variable displacement vane pump, the improvement set forth in claim 1, further characterized by:

(A) said one end plate being confined between an end surface of the rotor and a wall of the housing;

(B) and sealing means coacting with the adjacent surfaces of said one end plate and said wall of the housing to define the chamber means.

6. In a variable displacement vane pump the improvement set forth in claim 5, wherein said last named means comprises resiliently yieldable sealing means by which said one end plate is urged into engagement with the adjacent axial ends of the cam ring, and the rotor and its vanes, to prevent excessive leakage of fluid when the pump is starting up and there is no pressure in said pressure loading zone.

7. In a variable displacement vane pump the improvement set forth in claim 1, wherein one wall of said chamber means is provided by the bottom of a shallow concavity in the outer side of said one end plate.

8. In a variable displacement vane pump, the improvement set forth in claim 7 wherein the area of the bottom of said shallow concavity is such that the force acting thereon as a result of the fluid pressure in the chamber means is substantially equal to the force applied on said one end plate by the high pressure in the pump.

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CARLTON R. CROYLE, Primary Examiner

J. J. VRABLIK, Assistant Examiner

U.S. Cl. X.R.

418—27, 30, 133, 135