June 4, 1968 W. J. VAN ROSSEM 3,386,648

ROTARY VANE TYPE PUMP

Filed Jan. 31, 1967 3 Sheets-Sheet 1

FIG. 1.

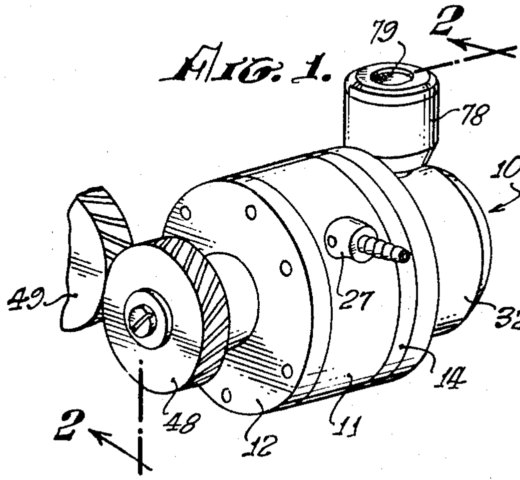



FIG. 8.

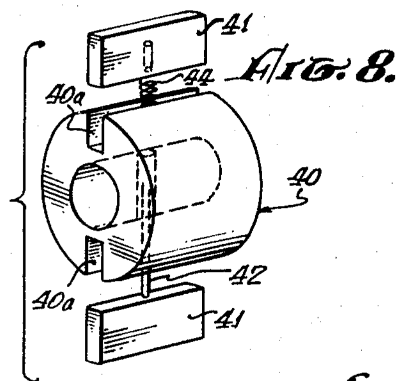
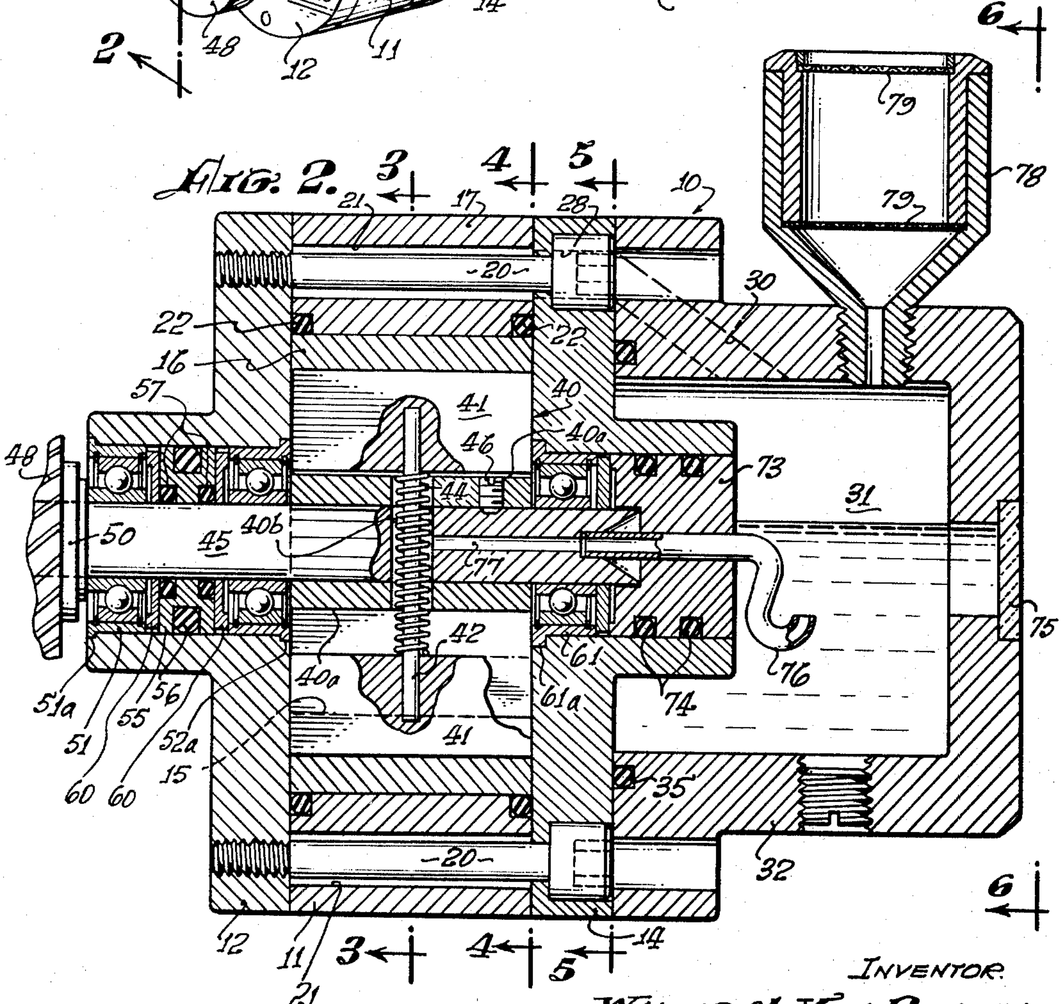



FIG. 2.

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ROTARY VANE TYPE PUMP

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Fig. 3.

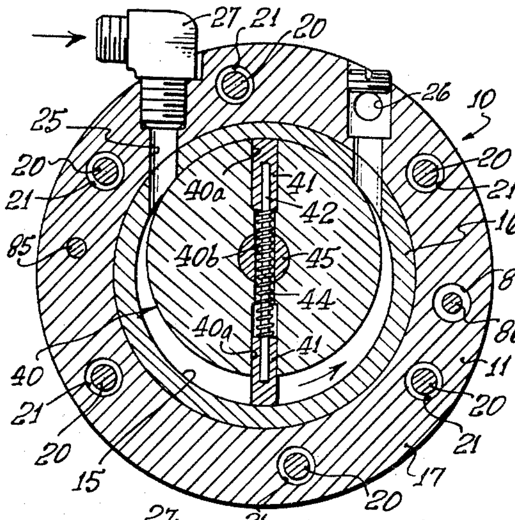

Fig. 4.

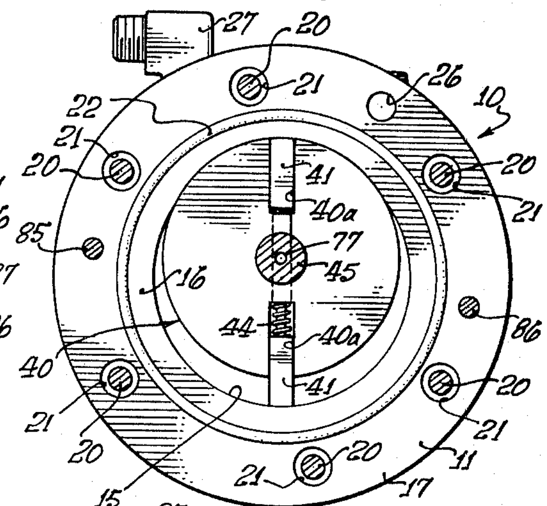

Fig. 5.

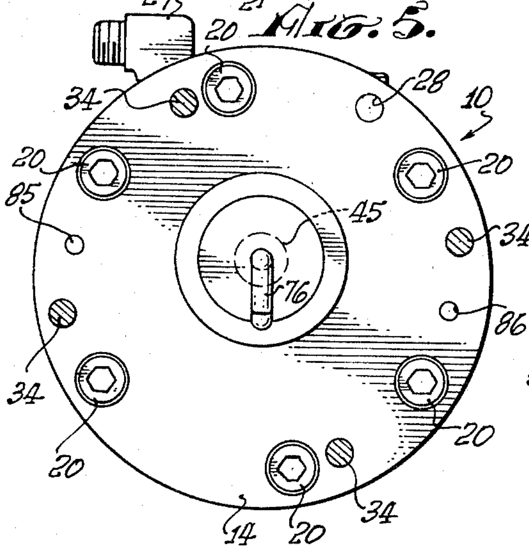

Fig. 6.

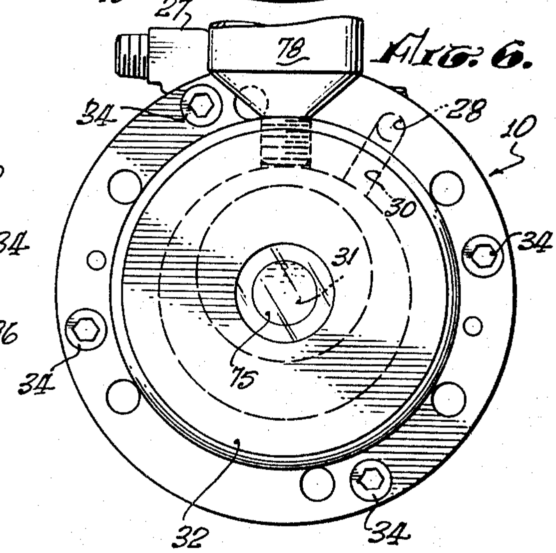

Fig. 7.

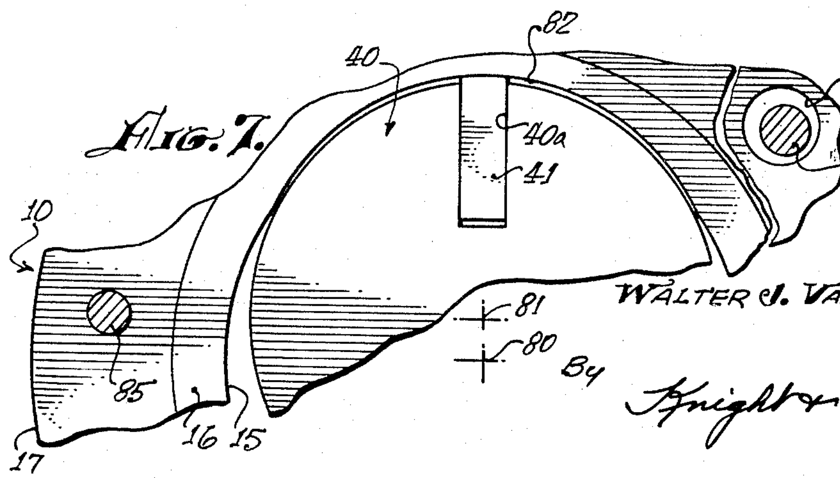

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FIG. 9.

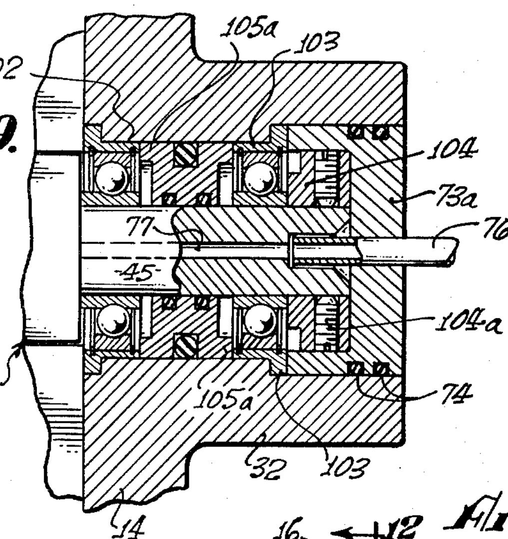



FIG. 10.

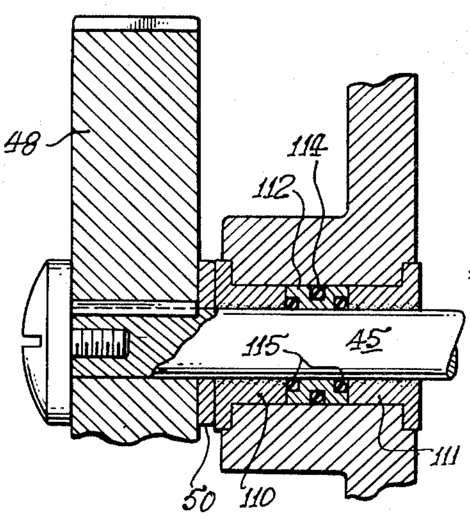



FIG. 11.

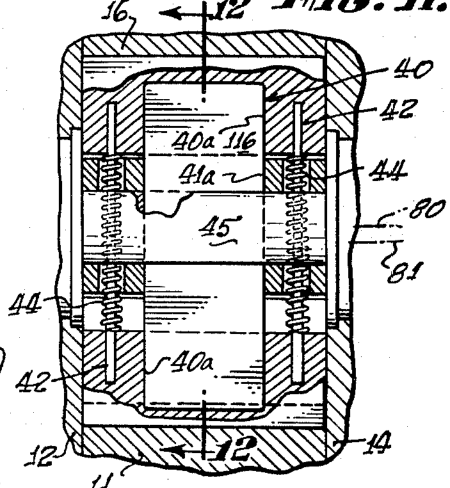



FIG. 12.

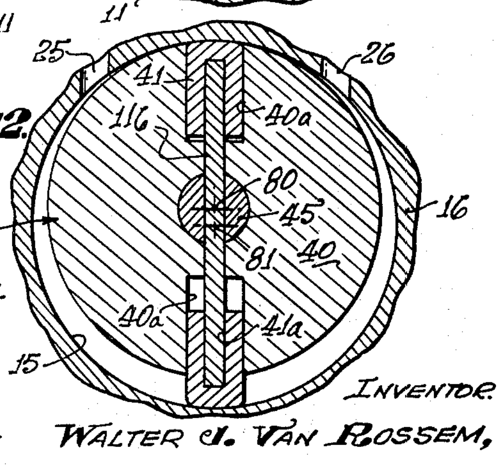
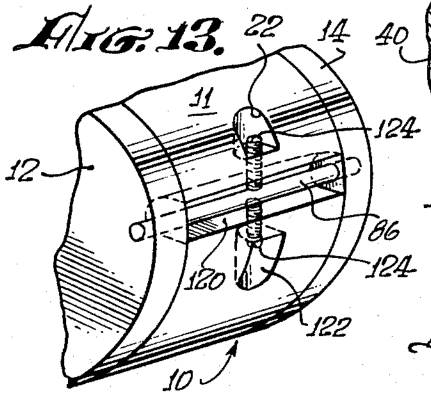



FIG. 13.

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United States Patent Office

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# 3,386,648
# ROTARY VANE TYPE PUMP

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Filed Jan. 31, 1967, Ser. No. 612,844
19 Claims. (Cl. 230—153)

## ABSTRACT OF THE DISCLOSURE

The rotor is eccentric with respect to the axis of the rotor chamber. A housing section is shiftable relative to the end plates to regulate the minimum spacing between the rotor and housing wall and the degree of fluid seal at that location. The shaft bearings are radial type bearings arranged to transmit axial thrust to the housing to relieve the rotor and vanes of end loading; and the bearings, together with the fluid seals carried on a removable collar, are all easily removed from the housing for servicing.

*Background of the invention*

The present pump is a rotary pump of the sliding or guided vane type and is disclosed in an embodiment designed especially as a vacuum pump; but the same pump can be caused to operate as a compressor. Generally speaking, the invention is independent of the fluid pumped. A typical use for a pump of this type is for investment of wax patterns in the "lost wax" technique of casting when it is desired to produce a vacuum of the order of one millimeter of mercury; although the same pump may be used advantageously for many other purposes where a lesser vacuum is satisfactory.

In pumps of this type, the rotor turns about an axis eccentric with respect to the axis of the rotor chamber, and the spacing between these two axes is fixed in known designs of pumps. A minimum clearance between the rotor and the sliding vanes, on the one hand, and the wall of the rotor chamber on the other, is desired in order to reduce air leakage between the inlet and the outlet to a minimum. This is particularly important in the zone of closest approach of the rotor to the periphery of the chamber; and a good fluid seal in this zone increases the degree of vacuum that the pump can produce.

Generally speaking, the clearance between the moving members is desired as large as possible in order to reduce wear. The fact that a close fit increases wear on the moving parts, causing the pump life and efficiency of the pump to fall off as the rate of wear increases, imposes a conflicting design requirement. In the case of a pump handling a non-lubricating fluid such as air, oil is introduced into the chamber to provide a lubricating film between moving parts.

The present invention seeks to solve this problem of wear versus clearance between the rotor and the housing by providing means for adjusting the radial clearance between the rotor and the housing. This arrangement not only makes it possible to compensate for wear on the rotor or housing during the life of the pump, but also to increase the clearance, thereby reducing the wear when only a moderate vacuum is required. In practice, the vanes need replacing only at long intervals. The vanes wear in and provide a better seal during most of their life, but eventually may need replacement, using the same shaft

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and rotor. In the absence of any adjustment between the housing and rotor, replacement of the vanes is necessary much sooner.

Another practical deficiency in known designs of pumps of this type is the fact that the wear on the air seals renders them inefficient and eventually requires replacement; but known designs of pumps are such that servicing them is difficult. Likewise, shaft bearings become worn and require replacement, often a substantial service problem in known pump designs. Because of this difficulty and the expense involved, service is often deferred longer than it should be with the result that the pumps do not perform satisfactorily.

The present invention seeks to solve this problem by a novel and advantageous arrangement of the bearings rotatably supporting the shaft and the associated air seals, rendering the bearings and seals easily removable for replacement or servicing, as may be required throughout the life of the pump. This insures that air-tight seals and high efficiency of the pump are easily maintained.

*Summary of invention*

The present invention is embodied in a vacuum pump having a housing that includes a central body with parallel end faces at opposite sides of the body and a pair of end plates of which one engages each of the two opposite end faces of the body. The end plates cooperate with the body to define in the housing a rotor chamber of substantially circular cross-section. Clamping means, typically in the form of bolts or machine screws, are provided to clamp the end plates to the body and maintain the assembled parts in fixed, fluid-tight relationship.

A rotor in said chamber has two or more radial vanes mounted in slots in the rotor, the vanes reciprocating as the rotor turns about an axis eccentric with respect to the axis of the rotor chamber. The rotor is mounted on a shaft rotatably supported in bearing assemblies at each side of the rotor. Each bearing assembly includes seal means engaging both the housing and the rotating shaft to insure a fluid-tight seal with these parts to prevent the entry of air into the rotor chamber. The bearings are typically radial type bearings but are provided with flanges whereby axial thrust on the shaft may be transmitted through the bearing to the housing, thereby eliminating end-loading on the rotor and vanes with consequent undue wear. The bearings and seal carriers all have a sliding fit within bores in the end plates and are easily removable for servicing and replacement.

The end plates have planar faces engaging the end faces of the housing body, thereby permitting the body to be shifted parallel to said faces and relative to the end plates of the rotor in a direction to increase or decrease the minimum clearance between the rotor and the housing at the zone of closest approach of these two elements. In the preferred embodiment, this means includes at least one aligning pin extending between the end plates and passing through the body to provide a pivot about which the body can move with respect to the rotor and end plates to obtain the said adjustment. Preferably, a second aligning pin is provided which maintains alignment of the two end plates but has some clearance with the body, thereby permitting the desired range of movement of body about the first pin. After the body is in adjusted position, the clamp means already mentioned are tightened to clamp the body between the two end plates. The oil reservoir is mounted entirely on an end plate to move therewith so that the adjustment of the rotor spacing does not interfere with lubrication.

*Brief description of the drawing*

FIGURE 1 is a perspective view of a pump embodying the present invention;

FIGURE 2 is a longitudinal median section through the pump of FIGURE 1 on line 2—2 of FIGURE 1;

FIGURE 3 is a transverse cross section through the rotor and housing on line 3—3 of FIGURE 2;

FIGURE 4 is a transverse section through the pump at the end of the rotor on line 4—4 of FIGURE 2;

FIGURE 5 is a transverse section through the housing at the outer surface of one of the end plates on line 5—5 of FIGURE 2;

FIGURE 6 is an end elevation of the pump as indicated by line 6—6 in FIGURE 2;

FIGURE 7 is an enlarged fragmentary diagrammatic view illustrating the movement of the body to effect adjustment of the minimum clearance between the body and the rotor;

FIGURE 8 is an exploded view of the rotor and vane assembly removed from the pump housing;

FIGURE 9 is a fragmentary longitudinal section showing a variational bearing arrangement at one end of the shaft;

FIGURE 10 is a fragmentary longitudinal section showing a further variational bearing arrangement;

FIGURE 11 is a longitudinal section through a rotor showing a variational construction;

FIGURE 12 is an end view of the rotor of FIGURE 11; and

FIGURE 13 is a fragmentary perspective of an adjustment screw.

*Description of preferred embodiment*

Referring now to the drawing, and more particularly to FIGURE 2 thereof, it will be seen that the pump illustrated comprises a housing or case, indicated generally at **10**, which includes a hollow central body section **11** between the two end plates **12** and **14**. Body section **11** has parallel end faces, each of which is engaged by a planar face on one of the end plates **12** or **14**. Body section **11** is annular in shape and has a generally circular interior space which is closed at its end by the two end plates to define within the housing rotor chamber **15**, as shown in FIGURES 3 and 4.

Since it is preferred to make end plates **12** and **14** and as much of body **11** as possible out of lightweight metals, such as aluminum alloys, while the rotor chamber must have a wear-resistance circumferential surface, the body is provided with a liner **16** of ferrous or other hard metal alloy, for example "Inconel," within which rotor chamber **15** is located. The liner and outer casing **17** of the body are maintained in the desired relative alignment or position by assembling the two with a press or shrink-fit which insures that the two parts are rigidly mounted with respect to each other.

The annular body member **11** is held clamped between the two separate and removable end plates **12** and **14** which respectively engage one of the two parallel end faces of the body member. Any suitable clamping means may be provided; but a preferred arrangement comprises a plurality of through bolts **20**. The bolts are screwed into tapped holes in one of the end plates, plate **12** in the drawing, and the bolt heads bear against the other end plate to clamp the body member rigidly between the two plates when the bolts are tightened. The bolt heads are accessible externally of the housing. For reasons which will become more apparent, the openings **21** in the body through which bolts **20** pass are slightly larger in diameter than the shanks of the bolts, providing some clearance between these openings and the bolts, as indicated at **21** in FIGURES 2 and 3.

In order to prevent leakage of air from the exterior into the rotor chamber, suitable seals are provided at each joint or interface between the annular body and the end plates. These seals may conveniently take the form of O-rings **22**, each located in an annular groove cut in an end face of body **11** and bearing against the opposing planar face of the adjoining end plate. Alternatively, O-rings **22** may be in grooves cut in the faces of end plates **12** and **14**. This arrangement of the sealing means effects a fluid-tight seal between the body and the end plates while also allowing a limited relative radial movement between the body and the end plates for purposes which will be further explained.

Pump housing **10** is provided with air inlet and outlet passages **25** and **26**, respectively, in body **11**, as shown in FIGURE 3. Inlet passage **25** is preferably threaded near its outer end to receive a suitable fitting **27** for attachment thereto of a hose or other conduit (not shown) connected to the article to be evacuated. Outlet passage **26** is an angular passage with one section opening to the periphery of rotor chamber **15**, as shown in FIGURE 3, and another intersectiong section extending substantially at right angles thereto. The latter section terminates at the interface between body **11** and end plate **14** where outlet passage **26** continues on by way of an aligned opening **28** which extends through the end plate. This opening **28** through the end plate, in turn, communicates with a diagonally drilled passage **30** leading to the interior of oil reservoir **31** that is provided inside cap **32** which is fastened to and exteriorly of end plate **14**. Thus, outlet passage **26**, through wihch air is exhausted from the rotor chamber, leads the discharged air into the interior of cap **32**. Thus, any oil carried by the discharged air from the rotor chamber is returned to the interior of cap **32**. Cap **32** provides an oil reservoir holding a body of oil which lubricates the pump. Cap **32** is rigidly secured to end plate **14** by a plurality of machine screws **34** which are received in tapped holes in end plate **14**. Thus, the end plate forms a part of the reservoir enclosure. A fluid-tight seal between the reservoir cap and the outer face of end plate **14** is provided by means of O-ring **35**, or any other suitable sealing means.

The rotor assembly illustrated in FIGURE 8 comprises a rotor **40** carrying a pair of diametrically disposed radial vanes **41**. The vanes are slidably mounted in longitudinally and radially extending slots **40a** in the rotor which open to the rotor periphery in order for the vanes to extend beyond the rotor and engage the wall of chamber **15**. A guide pin **42** is slidably and snugly received at each end in bores in the two vanes **41** and also passes through a radial bore **40b** centrally of the rotor. Around guide pin **42** and laterally supported thereby is helical spring **44** which is compressed between the two vanes, the spring bearing at its two ends against the two vanes. Thus, spring **44** yieldingly urges the vanes outwardly against the wall of the rotor chamber. This outward pressure on the vanes is deisred in order to pump fluid until sufficient centrifugal force is developed that the vanes are held by centrifugal force out in engagement with the circumferential wall of chamber **15**.

Vanes **41** are made preferably of a softer metal than liner **16**, thereby concentrating wear on the vanes and increasing their efficiency. Annealed half-hard brass is a very satisfactory material. The rotor is designed for maximum life and is made of a hard, wear-resistant metal. Also, resistance to acid fumes is sometimes required. All these requirements can be met by using aluminum-silicon-bronze for the rotor.

As will be later explained further, rotor **40** turns about an axis eccentric with respect to the axis of chamber **15**. Consequently, vanes **41** reciprocate radially in slots **40a** as the rotor turns. When fully extended, as at the lower vane position in FIGURE 3, the air pressure on the leading face of the vane causes the trailing face of the vane to press against the rotor body. This pressure is concentrated at the outer edge of the slot and creates a tendency for the vane to tilt from a truly radial position. Resistance to this tilting and better vane positioning are provided by the guide pin **42** which fits snugly, but slidably, into a bore in each of the vanes. As a result, less wear occurs on the rotor and pump efficiency is maintained longer by adding guide pin **42**. This advantage is gained in addition to the action of pin **42** in aligning and stabilizing spring **44** against bending.

Rotor **40** is non-rotatably attached to drive shaft **45** in any suitable manner, as for example by set screw **46**. Drive shaft **45** extends out of the pump housing in one direction only, this being the drive end of the shaft; and on this projecting end of the shaft, there is mounted a driven helical gear **48** which meshes with and is driven by drive gear **49**, also helical, as shown in FIGURE 1. Drive gear **49** is driven by an electric motor or other drive means, not shown. Helical teeth are preferred, because of their silence and small back lock, for gears **48** and **49** but are not essential as spur gears, a chain, a belt, or a direct drive may be used. The teeth of the helical gears are inclined in a direction that directs axial thrust on shaft **45** inwardly as the shaft is driven; and means are provided as described below to transmit this axial thrust to the housing.

The drive end of shaft **45** extends through and beyond end plate **12**; and this end of the shaft is rotatably supported in the end plate by a pair of spaced ball bearings **51** and **52**. Each of these bearings comprises inner and outer rings or races, as may be seen in the drawing. The inner races engage and rotate freely with shaft **45**. The outer races are stationary and are in engagement with the end plate. The shaft and bearings are typically stainless steel.

Bearings **51** and **52** have outwardly extending flanges **51*a*** and **52*a***, respectively, on the outer races, each engaged by a shoulder on end plate **12** to resist axial displacement of the bearing in one direction. These flanges are so arranged that the thrust tending to move one bearing toward the other bearing is resisted so that axial thrust received by either bearing is not transmitted to the other one.

Shoulder means, on the drive shaft, as for example bearing washer **50** on shaft **45**, is provided. This washer is interposed between gear **48** and the inner race of bearing **51**. Thus, any axial thrust on shaft **45** produced by the drive means and directed inwardly of shaft **45** is transmitted by the shoulder on washer **50** to the inner race of bearing **51** and thence to the outer race of the same bearing and finally through flange **51*a*** to the housing, as represented by end plate **12**. Thus, bearing **51** acts both as a radial and as a thrust bearing for shaft **45**. This arrangement of the bearings resists the endwise thrust on the shaft and allows rotor **40** and vanes **41** to turn freely within the pump housing and without excessive pressure and consequent wear against the end wall of the rotor chamber as represented by the opposing face of end plate **14**.

In order to develop a high degree of vacuum within the pump, the rotor chamber is sealed against leakage of air from outside the pump. In the case of drive shaft **45**, this is accomplished by novel sealing means around the shaft between the two bearings **51** and **52**. This sealing means preferably comprises a collar **55** which slides over shaft **45** and is also a sliding fit within the cylindrical bore through end plate **12** in which are located bearings **51** and **52**. Collar **55** is provided with an external annular groove in which is located O-ring **56** or other suitable seal, that engages the seal carrier and the end plate to effect a fluid-tight seal between these two members. The seal carrier also serves to position a second air seal, for example one or more O-rings **57**, each located in a recess cut in one end face of the carrier adjacent the central opening. At this location, each O-ring **57** engages the carrier and also the periphery of shaft **45** to effect a fluid-tight seal between these two members.

There is preferably added at each side of seal carrier **55** a washer **60** which has a raised rim on the face away from the seal carrier, as shown in FIGURE 2. The washers serve to confine O-rings **57** at one side while the raised rims face outwardly away from the seal carrier in order to bear against the outer races of bearings **51** and **52**. The purpose of this arrangement is two-fold. Primarily, it is desired that the inner races of the two bearings turn with shaft **45** and not come in contact with any stationary members which introduce friction; and this is accomplished by relieving the central portions of the washers **60** adjacent bearings **51** and **52**. Also, the raised rims bear against the outer races and thus introduce a frictional engagement between the outer races and carrier **55** which holds all the outer elements of the bearing assembly stationary and also serves to transmit any axial loads through the bearing assembly between the outer races of the bearings and thereby avoid loading the inner races, to obvious advantage.

The end of shaft **45** opposite the drive end, referred to herein as the tail end, is rotatably mounted in end plate **14** by a ball bearing assembly **61**, which is preferably a substantial duplicate of the bearing assembly **51** already described. This assembly, like ball bearing assembly **51**, has a stationary outer race provided with a flange **61*a*** engaging a shoulder on the end plate to resist axial displacement of the bearing. The two end plates **12** and **14** are substantial duplicates of each other and consequently have aligned bores of equal diameter extending through them in which the several bearings are located. However, in the case of end plate **14**, the drive shaft does not extend beyond the end plate and instead the bore is closed beyond the end of the drive shaft by a stationary plug **73**. Plug **73** carries suitable seals, such as Q-rings **74** in annular peripheral grooves to provide a fluid-tight seal between the end plate and the plug, thus closing the rotor chamber at this side thereof to prevent entry of air or oil from the interior of cap **32**.

As mentioned earlier, cap **32** is utilized to form an oil sump or reservoir **31** holding a body of oil for lubricating the rotor and the vanes. A sight glass **75** may be provided, if desired, so that the level of the oil in the reservoir can be determined visually without removing any of the parts. Oil from the reservoir is conducted to the vanes by a short length of tubing **76** passing through plug **73**. The tubing is suitably shaped so that the inlet of the tubing is always below the normal level of oil in the reservoir.

The discharge end of the tubing is located concentrically of and within the end of shaft **45**. The end of the shaft is preferably counter-bored to receive telescopically a straight terminal section of tube **76**, the tubing thereby communicating with an axial oil passage **77** formed in the tail end of shaft **45** and extending inwardly to intersect the bore **40*b***. This bore terminates at its outer end in the slots **40*a*** holding the two vanes **41**. This arrangement permits oil passing through tube **76** and passage **77** to be distributed to the two vanes and to lubricate the vane surfaces at their contacts with rotor **40** and the walls of chamber **15**.

It will be evident that the paths of oil circulation and air circulation coincide in part. Oil from the rotor chamber is carried with the discharged air stream through outlet passage **26**, **28**, and **30** and re-enters the reservoir **31**. In the reservoir, the oil and air discharged from the rotor chamber separate by gravity; and the air leaves reservoir **31** through exhaust stack **78** which is preferably provided with air cleaning means, such as a fine wire screen **79**. The screens allow exhaust air to pass through while retaining droplets of oil which then drain back into the oil reservoir. Added filter means may be placed between screens **79**.

As will be understood by those in the art, rotor chamber 15, when viewed in transverse cross-section, has a substantially circular peripheral wall against which the ends of vanes 41 bear. The rotor is mounted a turn about the axis of shaft 45, which is eccentric with respect to the center of curvature of the major portion of chamber 15. This relationship is shown in the diagram constituting FIGURE 7 in which the center of curvature of the rotor chamber wall is indicated at 80, while the center of rotation of the rotor, that is the axis of shaft 45, is indicated at 81. For simplicity of description, center 81 will be assumed to be above center 80. In order to obtain a better seal between the moving rotor and the chamber wall at the location of closest approach of the rotor to the chamber wall, a section of the chamber wall indicated at 82 departs from the true circle about center 80 and instead conforms in curvature to the periphery of rotor 40, i.e. has its center of curvature at 81. This change in curvature of the rotor wall allows a very close approach of the rotor to the chamber wall over an arc of substantial length located between inlet 25 and outlet 26. For practical purposes, this zone of close approach may be considered to extend completely between the inlet and the outlet.

A close approach between the rotor and the chamber wall over this zone is desired in order to prevent fluid leakage between the inlet and outlet which reduces the effectiveness of the vacuum pump and thus serves to limit the degree of vacuum which it can achieve.

Under some conditions, it may be desired to develop the maximum vacuum possible; and, under this condition, the rotor and the chamber walls should approach each other as closely as possible. However, there are other situations in which the same pump may be used in which it is not desired to develop the maximum possible vacuum, but instead it may be desired to limit the vacuum to some lower value, for example 300–400 mm. of mercury. Under these circumstances, a somewhat greater minimum clearance between the rotor and the chamber wall can be advantageously utilized. Accordingly, means are provided for adjusting the clearance between the rotor and the chamber wall in the zone 82 between the inlet and the outlet to the rotor chamber.

To permit adjustment of this clearance, end plates 12 and 14 are provided with planar opposing surfaces which are maintained parallel with each other. These opposing surfaces engage the end faces of annular body 11 at opposite sides thereof, as described above. This arrangement permits body 11 to be adjusted through a limited range of movement in a direction parallel to the faces of plates 12 and 14 and normal to the axis of rotation of shaft 45. This movement is permitted by the clearance at 21 previously mentioned around through bolts 20.

Alignment of the end plates during such movement of housing body section 11 relative to rotor 40 is maintained by a pair of dowel pins 85 and 86. Dowel 85 fits snugly within aligned openings in end plates 12 and 14 and also in body section 11 with the result that dowel 85 forms a pivot about which housing body 11 can swing with respect to the end plates and shape 45. Dowel 85 is so located it is removed approximately 90° around the axis of shaft 45 from the center of chamber wall arc 82.

The second dowel 86 is opposite pin 85. It passes through two aligned sections in end plates 12 and 14 which receive the dowel snugly, but pin 86 passes through an opening 87 in body section 11 which is of larger diameter than dowel 86 thereby allowing some clearance between the pin and the wall of opening 87. With this arrangement, dowels 85 and 86 maintain the end plates and shaft bearings in fixed alignment. Clearance of body 11 around dowel 86 allows the swinging movement of the body around the axis of dowel 85 to adjust the clearance between the rotor and the chamber wall.

With this arrangement, adjustment of this clearance can be accomplished merely by loosening through bolts



20, even while the pump is running, shifting annular body section 11 to pivot it around dowel 85 to achieve the desired clearance with rotor 40, and then tightening bolts 20 to again clamp body section 11 between the end plates 12 and 14 in the adjusted position.

It will be realized that the range of radial movement of the housing body is limited since the clearance between the rotor and the chamber wall is never great. The actual clearance will vary with several factors, including rotational speed, the vacuum desired, the viscosity of the fluid pumped, and others; and may be reduced to actual contact of the rotor with the body wall. Of course, wear is reduced on the chamber wall as the clearance is increased, within limits.

*Description of variational structures*

The above-described design of pump provides particularly for ease and economy in manufacture, ease of removal of parts for repair and servicing, and reliability of operation over a long period of time. All of these features can be retained in various optional designs described below which adapt the pump to particular conditions.

FIGURE 9 illustrates a bearing construction for the tail end of the drive shaft which is a substantial duplicate of the bearing construction previously described for the drive end, thus providing a symmetrical bearing arrangement. This bearing arrangement of FIGURE 9 is advantageous when the axial thrust on the shaft is in a reverse direction from that previously described, that is the shaft has an endwise thrust applied to it directed toward the left in FIGURE 9. Of course, this same arrangement is desirable for bi-directional thrust conditions.

In the construction shown in FIGURE 9, the tail end of drive shaft 45 is mounted in a pair of spaced ball bearings 102 and 103 which are substantial duplicates of the ball bearing assemblies 51 and 52 already described. Each outer race of the two ball bearing assemblies is provided with an outwardly projecting annular flange which seats against a shoulder in the housing to resist endwise displacement. Particularly, flange 103*a* on bearing assembly 103 is designed to transmit directly to the housing any thrust on shaft 45 directed towards the left in FIGURE 9. Collar 104 is non-rotatably affixed to the end of shaft 45 in such a way as by set screws 104*a*, that it turns with the shaft and bears against the inner race of bearing 103 to transmit endwise thrust on the shaft through the bearing to flange 103*a* and thence to the housing, more particularly to the end plate 14.

Thrust collar 104 herein is added to the bearing structure of the form of pump illustrated in FIGURE 2; but is functionally similar to thrust washer 50 of the first described embodiment of the invention. This thrust collar 104 may conveniently be set in a recess in plug 73*a* closing the bore in the housing.

Another difference between the bearing arrangement illustrated in FIGURE 9 and that in FIGURE 2 is that the washers 60 in the latter are omitted. Instead, carrier 105 for the O-ring seals is provided on opposite faces with raised rims 105*a* which bear against the outer races of bearing assemblies 102 and 103. The depressed or relieved central portions of the seal carrier end faces thus avoid contact with the inner races of the bearings to eliminate any drag or frictional engagement which would tend to impede free turning of shaft 45. Carrier 105 is metal or other impervious material and so constitutes a part of the fluid seal between the bearings.

Although ball bearing assemblies are preferred for all bearings since they have very well known low friction characteristics, it is possible to substitute journal bearings for the ball bearing assemblies. FIGURE 10 illustrates an exemplary arrangement in which two sleeve bearings 110 and 111 are substituted for the ball bearing assemblies 51 and 52 at the drive end of shaft 45. These two journal bearings are spaced apart and between them is carrier 112 which carries one external O-ring seal 114 and two inner O-ring seals **115**. The former seal is in contact with the housing and with the carrier, while the latter seals are in contact with the carrier and shaft **45**, thus providing, with the carrier, a fluid-tight seal in the space between the two sleeves **110** and **111**. The inner bearing **111** can be lubricated by oil from within the rotor chamber. The seal between the two bearings requires that bearing sleeve **110** be lubricated from an external source or be some permanently lubricated type, as for example a graphite impregnated sintered copper bearing.

Under heavy duty conditions, it may be desired to increase the effectiveness of pin **42** in bracing blades **41** against rocking in the slots of the rotor. This effect can be increased by the construction shown in FIGURES 11 and 12 in which the pin has been elongated and is now a flat pin **116** which extends through a slot in drive shaft **45** and into elongated slots **41*a*** in the two rotor vanes **41**. In order to get the maximum benefit of this construction, it is preferred that pin **116** be sufficiently long that it projects outwardly beyond the periphery of the rotor when at the point of greatest outward movement of each vane **41**, which is the position shown for the bottom vane in FIGURE 12. The opposite vane **41** is then moved inwardly to the maximum extent since it is at the zone **82** of minimum clearance between the rotor and the rotor chamber wall. Pin **116** is but slightly shorter than the maximum distance between the ends of the two slots **41*a*** in the two vanes, having minimum operating clearance.

With this construction, a spring **44** reinforced by a pin **42** is located at each side of center blade **116**.

An advantage of the flat or elongated pin **116** is that the spring loading on it can be added to or obtained solely from bow springs in the bottom of vane slots **41*a*** between the pin and one or both vanes.

FIGURE 13 illustrates an optional arrangement for obtaining a fine adjustment of the minimum clearance between the rotor and the chamber wall in the zone **82** (FIGURE 7). In this construction, dowel pin **86** is exposed where it passes through body **11** by placing it in a slot **120** milled lengthwise in the housing body **11**. At its ends, pin **86** is firmly held in end plates **12** and **14**, as before.

Above and below slot **120**, recesses **122** are cut in the exterior surface of body section **11**; and in each of these recesses is located an adjusting screw **124** threaded into the body of the pump housing and coming into engagement with rod **86**. The two screws **124** bear against rod **86** at opposite sides thereof so that by advancing one screw and backing off the other, housing body **11** can be shifted slightly with respect to the end plates in order to adjust the clearance between it and rotor **40**, as previously described. When the desired adjustment is attained, the two screws **124** may be tightened against the rod.

Such movement of the body with respect to the end plates requires that the bolts **20** be loosened first, as already explained; and after the adjustment is effected, the bolts are tightened in order to clamp the end plates and the body together in the adjusted position. One advantage of the arrangement illustrated in FIGURE 13 is that the adjustment of the clearance can be effected while the pump is in operation since the relative movement of body **11** is always restrained by the adjusting screws and seals **22** are effective to a large extent even when bolts **20** are loosened enough to permit said adjustment.

From the foregoing description, it will be apparent that various changes in the details of construction and arrangement of the component parts constituting the present invention may occur to persons skilled in the art without departing from the spirit and scope of the invention. Accordingly, it is to be understood that the foregoing description is considered as being illustrative of, rather than limitative upon, the invention as defined by the appended claims.

I claim:

**1.** A vane type pump comprising, in combination:
a housing including a central body between a pair of end plates at opposite sides of the body cooperating therewith to define a rotor chamber;
a rotor with radial vanes within the chamber;
a shaft carrying the rotor and rotatably mounted in the end plates for turning the rotor within the chamber about a fixed axis;
a pair of spaced bearings carried by one end plate and rotatably supporting the shaft;
sealing means around the shaft between the bearings including a seal carrier, a first seal on said carrier engaging the shaft and the carrier, and a second seal on the carrier engaging said one end plate and the carrier;
and means holding the body and end plates in fixed relative positions but permitting the body to be shifted relative to the rotor to adjust the radial clearance between the rotor and the body within the rotor chamber.

**2.** A vane type pump according to claim **1** which also comprises:
the outer one of said bearings having inner and outer races and rotatably supporting the shaft, the outer race of the bearing having shoulder means engaging the end plate to resist relative inward movement axially of the shaft;
and means on the shaft engaging the inner race of said bearing to transmit thereto inward axial thrust from the shaft, whereby the rotor is relieved of axial thrust against the end plate.

**3.** A vane type pump according to claim **1**, in which two vanes are slidably mounted in said rotor at diametrically spaced positions, which also includes:
spring means bearing at opposite ends thereof against said two vanes and yieldingly urging the vanes outwardly aganst the wall of the rotor chamber;
and a guide pin passing centrally through the spring and slidably engaging at each end the wall of a socket in a vane to support the vane laterally, said guide pin extending beyond the periphery of the rotor where a vane is at the extreme of its travel out of the rotor.

**4.** A vane type pump comprising, in combination:
a housing including a central body between a pair of end plates at opposite sides of the body cooperating therewith to define a rotor chamber;
a rotor with radial vanes within the chamber;
a shaft carrying the rotor and rotatably mounted in the end plates for turning the rotor within the chamber about a fixed axis;
and means holding the body and end plates in fixed relative positions but permitting the body to be shifted relative to the rotor to adjust the radial clearance between the rotor and the body within the rotor chamber, said means including a pair of pins extending between the end plates and passing through the body at opposite sides of the rotor, one of the pins providing a pivot for rocking movement of the body and the other pin having clearance with the body to allow limited rocking movement of the body about said one pin;
and releasable means clamping the end plates to the body over the range of relative movement.

**5.** A vane type pump according to claim **4** in which the central body is of annular configuration with parallel end faces and the end plates each have a planar face engaging an end face of the body within the range of said relative shifting;
and which also includes releasable clamping means holding the assembled body and end plates in selected adjusted positions relative to each other.

6. In a vane type pump having a housing, a rotor therein provided with one or more radial vanes, and a drive shaft turning the rotor, the improvement that comprises:

means defining an oil reservoir beyond one end of the shaft;

an oil passage in the rotor and shaft communicating with the vanes and including a section concentric with the shaft and opening to said one end of the shaft;

and a stationary tube communicating with the oil reservoir and having one end telescopingly fitting into one end of the drive shaft concentrically of said oil passage therein for transmission of oil from said reservoir to said vanes in the rotor.

7. In a vane type pump having a housing, a rotor therein provided with one or more radial vanes, and a drive shaft turning the rotor and projecting beyond the housing at one side thereof, the improvement that comprises:

a pair of spaced bearings rotatably supporting the shaft and mounted in the housing;

and sealing means around the shaft between the bearings including an impervious seal carrier, a first seal engaging the shaft and the carrier, and a second seal engaging the housing and the carrier.

8. In a vane type pump, the combination according to claim 7, which also includes:

shoulder means on the shaft engaging one bearing to transmit to the bearing axial thrust from the shaft;

and shoulder means on the one bearing engaging the housing transmitting said thrust to the housing.

9. In a vane type pump, the combination according to claim 7 in which the drive shaft is subject to an axial thrust and in which each of said bearings has an inner race and an outer race, which also includes:

shoulder means carried by the shaft engaging the inner race of one bearing and transmitting said axial thrust thereto;

and the outer race of said one bearing has a flange engaging the housing and transmitting axial thrust to the housing.

10. In a vane type pump having a housing, a rotor therein provided with one or more radial vanes, and a drive shaft turning the rotor and projecting outside the housing, the improvement that comprises:

a pair of spaced bearings rotatably supporting the shaft and mounted in the housing each bearing comprising inner and outer races;

sealing means around the shaft betwen the bearings including an impervious seal carrier, a first seal engaging the shaft and the carrier, and a second seal engaging the housing and the carrier;

and means providing frictional engagement between the seal carrier and the outer races of the bearings and providing clearance at the inner races whereby the inner races turn freely with the shaft.

11. In a vane type vacuum pump having a housing with a removable end plate and a bore therein, a rotor within the housing, and a drive shaft turning the rotor and projecting outwardly of the housing for connection to drive means, the improved bearing means for rotatably supporting the shaft that comprises:

an inner bearing tightly fitted into the bore in the end plate;

an outer bearing spaced from the first bearing and slidably fitted into said bore and having shoulder means engaging the end plate to resist inwardly directed thrust;

sealing means in the bore between the bearings having a sliding fit with the end plate, the shaft having a sliding fit with the sealing means and the bearings;

a shoulder means carried by the shaft applying inwardly directed axial thrust generated in the shaft by the drive means to the outer bearing to hold the bearings and sealing means in assembled position in the end plate yet permitting manual removal of the sealing means and the outer bearing from the end plate after removal of the shaft from the bearings and sealing means.

12. A vane type pump according to claim 11 in which the inner bearing also has shoulder means engaging the end plate to resist outwardly directed thrust and the shaft carries a second shoulder means applying to the inner bearing outwardly directed axial thrust generated in the shaft.

13. A vane type pump comprising, in combination:

a housing having a rotor chamber with inlet and outlet means;

a rotor in said chamber;

bearing means mounting the rotor eccentrically of said chamber;

a pair of diametrically opposed radial vanes slidably mounted in said rotor and bearing against the wall of said chamber;

spring means outside the vanes bearing at its ends against said vanes yieldingly urging them outwardly relative to said rotor against said wall;

and a guide pin passing centrally through the spring and slidably received at each end in a socket in a vane, said pin being a snug sliding fit in each socket to support the associated vane against tilting relative to the pin and extending beyond the periphery of the rotor when a vane is at the extreme of its travel out of the rotor.

14. A rotor construction for a vane type pump, comprising:

a rotor having a pair of oppositely arranged vane-receiving slots with a hole extending through the rotor between the bases of the slots;

a vane slidably mounted in each slot and having a pin-receiving socket, each slot and the vane therein being so dimensioned relatively that more than one-half the vane is in the slot at all times;

a pin passing slidably through said hole in the rotor and received at its ends in the sockets in said vanes, the pin being a snug sliding fit in each socket to support the associated vane against tilting relative to the pin and extending beyond the periphery of the rotor at each vane when the vane is most fully extended beyond the rotor.

15. A rotor construction as in claim 14 which also includes spring means outside of the vanes and bearing against the vanes urging them outwardly in the slots.

16. A rotor construction as in claim 14 in which the pin is a flat member elongated axially of the rotor.

17. A rotor construction as in claim 14 in which the pin is a flat member elongated axially of the rotor, and which also includes spring means at opposite sides of the pin urging the vanes outwardly in their slots.

18. A vane type pump comprising, in combination:

a housing including a central body between a pair of end plates at opposite sides of the body cooperating therewith to define a rotor chamber;

a rotor with radial vanes within the chamber;

a shaft carrying the rotor and rotatably mounted in the end plates for turning the rotor within the chamber about a fixed axis;

and means holding the body and end plates in fixed relative positions but permitting the body to be shifted relative to the rotor to adjust the radial clearance between the rotor and the body within the rotor chamber, said means including a continuous pin extending through the body and into the end plates and having coaxial end sections located one in each end plate, to establish an axis parallel to the rotor axis about which the body can swing relative to both end plates, said pin being located about midway between maximum and minimum clearance of the rotor with the body;

and means to hold the body in an adjusted position relative to the end plates.

19. A vane type pump according to claim 18 in which the last mentioned means includes a rigid member extending between the end plates to retain the end plates in mutual rotational alignment and adjusting means engaging the rigid member to swing the body about the continuous pin to obtain the desired clearance between the rotor and the body.


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