EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention of the grant of the patent: 26.03.1997 Bulletin 1997/13
(51) Int.Cl.: F04C 2/344, F01C 21/08
(86) International application number: PCT/US91/03766

(21) Application number: 91911935.4
(22) Date of filing: 31.05.1991

(54) ROTARY VANE MACHINE WITH SIMPLIFIED ANTI-FRICTION POSITIVE BI-AXIAL VANE MOTION CONTROL
DREHFLÜGELZELLENMASCHINE MIT VEREINFACHTER REIBUNGSARMER POSITIEVER BI-AXIALER STEUERUNG DER FLÜGELBEWEGUNG
MACHINE A PALES ROTATIVES A COMMANDE DE MOUVEMENT DE PALES BIAxIAL POSITIF SIMPLIFIE ANTI-FROTTEMENT

(84) Designated Contracting States: DE ES FR GB IT
(30) Priority: 07.06.1990 US 534542
(43) Date of publication of application: 24.03.1993 Bulletin 1993/12
(73) Proprietor: EDWARDS, Thomas C.
Rockledge, FL 32955 (US)

(72) Inventor: EDWARDS, Thomas C.
Rockledge, FL 32955 (US)

(74) Representative: Altenburg, Udo, Dipl.-Phys.
Patent- und Rechtsanwälte
Bardehe . Pagenberg . Dost . Altenburg
Frohwitter . Geissler & Partner,
Postfach 86 06 20
81633 München (DE)

(56) References cited:
FR-A- 2 507 256
US-A- 2 345 561
US-A- 2 469 510
US-A- 4 859 163
GB-A- 2 192 939
US-A- 3 101 076
US-A- 4 958 995

Note: Within nine months from the publication of the mention of the grant of the European patent, any person may give notice to the European Patent Office of opposition to the European patent granted. Notice of opposition shall be filed in a written reasoned statement. It shall not be deemed to have been filed until the opposition fee has been paid. (Art. 99(1) European Patent Convention).
Description

This invention is related to a non-contact vane-type fluid displacement machine according to the precharacterizing part of claim 1.

Conventional sliding rotary vane machines are distinguished from virtually all other fluid displacement machines in their remarkable simplicity. On the other hand, such machines exhibit relatively poor operating efficiency. This poor energy efficiency is rooted directly in machine friction, both mechanical and gas dynamic. As is well known, the predominant source of mechanical friction in conventional production non-guided vane rotary machines occurs at the intense rubbing interface of the tip of the sliding vane and inner contour of the stator wall. Furthermore, governing the motion of the vane by the stator wall contour necessarily and greatly inhibits the area through which gas can enter or exit the machine. This results in increased fluid flow pressure losses in the inlet and outlet port regions of such machines.

In a majority of previous endeavors to grapple with this mechanical problem, attention has been focused upon the use of wheels or rollers pinned to the sides of the vanes wherein these rollers follow inside a circular or non-circular track of the appropriate configuration. The cooperation of the rollers in the roller guide track then produces a means of dictating the radial location of the vane which is pinned to the roller follower and hence determines the position of the tip of the vane.

However roller wheels cannot provide positive bi-axial radial motion without having to reverse their rotational direction. That is, vanes constrained by rollers can accommodate geometric displacement in only an outward or inward direction at any one time.

Other proposals include the use of sliding arc segment tethers in place of vane rollers. In such prior art instances, the arc segment tethers are captured within a circular annular groove that may or may not be rotatable. The arc segment vane tether has the outstanding and fundamentally important advantage of being able to deliver both positive inward and outward radial motion to the vane simultaneously. However, in the prior art, vane motion control techniques used arc segment vane tethers which entailed considerable mechanical friction that arises from the sliding of the arc tether surfaces against the circular annular guides, whether or not the guides themselves are rotatable.

GB-A-2 192 939 discloses a non-contact vane-type fluid displacement machine having pins, which are projected on respective vanes peripherally slidably engaging the annular races of retainer rings through a respective sleeve bearing. The sleeve bearing slips over that pin is slidably rotated while being pressed against the outside diameter side by the centrifugal force within the annular race of the retainer rings while the retainer rings follow the sleeve bearing for rotation because the former are in a state to be rotatable by the ball bearings.

It is therefore one object of the invention to provide a vane type fluid displacement machine that accomplishes non-contact vane tip sealing in a particularly simple and energy efficient manner, which can operate over a large range of operational speeds with a wide variety of refrigerants, including those not harmful to the earth's stratospheric ozone layers, and whose vane tips are positioned by the utilization of circular radial vane guides, eliminating the use of costly non-circular vane guides.

This object is achieved according to the invention with a non-contact vane-type fluid displacement machine with the features of claim 1.

Further advantageous developments of the inventive machine are mentioned in claims 2 to 9.

The machine according to the invention not only eliminates the majority of the mechanical sliding friction endemic to previous techniques, but it does so with fewer and simpler components than were required by the prior art. At the same time, the fundamentally important positive bi-axial radial vane motion control necessary for the practical operation of such machines is accomplished. Finally, the machine accommodates the natural motion of the tips of circularly-tethered vanes by providing exceedingly close non-contact vane tip sealing as a result of properly shaping the mating or conjugate interior of the casing wall.

The described embodiments are ideally suited for use as an automotive air conditioning compressor. A major aspect of the present invention is comprised of two principal embodiments, both of which center upon simple, anti-friction, easily-producible, economical, and motion-positive means of insuring the accurate transfer of radial movement from the circular radial vane guide to the vane. The cooperation of either of these means of precise anti-friction vane motion control with a special internal casing profile, having the shape of an envelope resulting from the path of the vane tips being guided by circular annuli which are eccentric with the rotor axis, results in maintaining an excellent sealing but non-contact, and thus, minimum friction relationship between the tips of the vanes and the internal casing contour. Such a condition yields a simple vane type fluid handling device of high volumetric and energy efficiency.

The first of these vane motion control embodiments involves the use of plain arc segment vane tethers that are pinned pivotally to the vanes and that ride directly upon freely-rotating retained roller bearings that roll inside the internal surface of circular, non-rotating radial vane endplate guides. The second of these embodiments involves vane tether elements resembling roller skates, also pivotally-pinned to the vanes, that ride on non-rotating circular vane guides located in the endplates of the device.

Further advantages will be apparent from the following explanation of the invention by means of embodiments with reference to the accompanying drawings, in which

Figure 1 presents an elevation view of an embodi-
ment with one endplate removed so as to reveal the rotor equipped with the tethered sliding vanes and an accompanying annular vane guide.

Figure 1a illustrates a break-out of one of these tether/vane assemblies.

Figure 2 is a side elevation of a primary embodiment of my invention, offering a cross-sectional view of certain vanes with their tethers in the tether annuli in opposing endplates.

Figure 2a illustrates a break-out of the sideview of a typical vane/tether assembly.

Figure 3 shows a face view of the rotor with a corresponding set of tethered vane assemblies depicted in exploded relationship out of their respective rotor slots. This figure also reveals the broken lines the annular surfaces located in the end plates that serve to guide the vane tethers.

Figure 4a presents enlarged details of the construction of one of the embodiments of this invention that utilizes a freely-rotating caged bearing friction minimizing means, with plain positive outward radial motion control.

Figure 4b presents details of the construction of another embodiment of a tether in which the tether features trunnion rollers and plain positive outward radial motion control.

Figure 4c illustrates an embodiment using freely-rotating retained bearings operating on both the inner and outer peripheries of a plain arc segment vane tether.

Figure 4d shows an arc segment vane tether equipped with trunnion rollers in the outside arc region which interface with a freely-rotating retained roller bearing disposed on the inner periphery of the annular surface of the radial vane guide.

Figure 4e shows the combination of a caged freely-rotating retained roller bearing on the outside periphery of the arc segment vane tether, but revealing that the vane tether is equipped with trunnion rollers on its inner periphery.

Figure 4f portrays a vane tether equipped with trunnion rollers on both its inner and outer peripheries.

Figure 5 shows details of the stator contour geometry required for functional operation of the invention as a gas compressor or the like.

Detailed Description

In order to understand further the function and operation of the non-contact vane-type fluid displacement machine in accordance with a first embodiment of this invention, reference is first made to Figure 1 which illustrates many of the principal elements of my invention. These elements include the casing which is equipped with an internal profile contoured specifically to tangentially mate in a sealing but non-contact relationship with the actual controlled motion of the tips of the vanes as they are carried within the rotor. This cooperation thus maintains a sealing but non-contact relationship there between. I prefer to refer to this internal conforming profile as a conjugate or conformal profile, and the precise technique by which this conjugate profile is determined is explained in detail hereinafter.

With continuing reference to Figure 1 it will be noted that rotor 14 is disposed in an eccentric relationship to the internal conforming profile 12 of the casing 10, with center point 16 denoting the axis about which rotor 14 rotates. Although I am not to be limited to any particular number of vanes to be carried by the rotor 14, for purposes of illustration I have shown in Figure 1 vanes 20, 22, 24 and 26 which, for all intents and purposes, can be regarded as being identical to each other. Further, it can be seen that these vanes are equipped with what I prefer to call vane tethering means, these being denoted as 20a, 22a, 24a and 26a, respectively. These vane tethers can themselves also be considered, for all intents and purposes, identical to each other and to cooperate with the vanes through means such as pins 30, 32, 34, and 36. The vanes 20, 22, 24, and 26 may be seen clearer and in more detail in Figure 3.

As will be understood by those skilled in this art, fluid to be compressed is admitted through the port denoted INLET in Figure 1, and the compressed fluid is delivered out of the port captioned OUTLET.

In Figure 1a I have shown details of a typical vane and its corresponding tether. As noted, this vane is captioned as vane 22, and its tether 22a, and is further equipped with a carefully located circular arc vane tip, indicated in this Figure as T. In accordance with this invention, the vane tip T is intended to travel immediately within the tangentially conforming inner wall 12 of the stator 10 in an exceedingly close yet substantially frictionless non-contacting relationship.

A means in accordance with this invention by which precision vane motion can be accomplished with a minimum of mechanical friction can be seen by referring to Figures 1a, 1, 2a and 2a. It is to be understood that vane tethers 20a, 22a, 24a, and 26a have identical companion tethers equipped with trunnion rollers utilizing the opposing side of each of the respective vanes through the action of corresponding tether pins, and it is therefore sufficient to describe only a single set of tethers associated with each vane. Visible in Figure 2a are tethers 24a and 24aa of vane 24 with tip T. These and the other sets of vane tethers, operating in conjunction with certain endplate annuli and anti-friction means to be described in more detail hereinafter, are responsible for each vane tip T moving in the aforementioned desired exceedingly close yet substantially frictionless relationship to the inner conjugate profile 12 of the casing 10.

Referring now specifically to Figure 2, it will there be noted that the casing 10 is revealed to be bounded on its left and right sides by the endplates 40 and 42 which, for the purposes of this explanation, are substantially identical except that the rotor shaft 44 protrudes
through the right endplate. These endplates are secured to the casing 10 by any conventional means, such as through-bolts, and such details are of no particular concern to this invention.

As is clear to those skilled in this art, volumetric changes can be brought about with rotor rotation because of the eccentric relationship between the axis of the rotor 14 with its attending set of vanes 20 through 26, the supporting opposing endplates, and the internal conforming profile 12 of the casing. This is, of course, brought about in such a way that pumping or compression of fluids entering through the INLET can be accomplished and discharged through the OUTLET, as was previously mentioned. However, for compression and/or pumping to be accomplished efficiently, the periphery 15 of rotor 14 must sealingly engage the internal casing profile in region 13.

It can be further noted from Figure 2 that rotor 12, which is rotatably supported in the endplates 40 and 42 by the use of the shaft 44, may be considered either to be integral with the shaft, or to be engaged with the shaft in a close axial sliding fit, having a zero relative rotation. Suitable bearings are utilized in the endplates in order that the rotor shaft 44 and rotor 14 can freely rotate, and it is to be understood that the left and right faces of the rotor 14 are operatively disposed in a contiguous sealing relationship with the inner walls of the endplates. Suitable lubrication is provided at this interface and in other locations within the machine, in accordance with well-known techniques.

It can be noted in Figure 2 that I have opened portions of the drawing in order to reveal the presence in each illustrated endplate of the earlier-mentioned circular annuli, with annulus 50 being located in endplate 40, and annulus 52 being located in endplate 42. It can also be noted that the center of these annuli are coincident with the geometric center of interior casing of the conforming profile 12. It is quite important to observe that because these annuli are circular rather than non-circular, manufacturing costs are minimized by this aspect of my technique. Further savings in manufacturing costs and increases in machine performance can be derived from employing annulus which can be produced separately from the endplate itself and then joined with the endplate during assembly as shown in Figure 1.

In order to facilitate the utilization of one friction minimizing means in the annuli, I prefer, as indicated above, to dispose a hardened steel ring 60 in annulus 50, and a substantially identical hardened steel ring 62 in annulus 52. It is in annular ring 60 that the tethers 20a, 22a, 24a and 26a travel as seen in Figure 1, whereas their companion vane tethers travel in annular ring 62 shown in Figure 2 as the rotor 14 rotates in the casing 10.

Although my invention would be operactive without friction minimizing means utilized with the conjugate internal casing profile 12, I find it greatly preferable to utilize a freely-rotating caged roller bearing inside each of the hardened steel rings, with Figure 2 revealing that bearing 54 is utilized in ring 60 located in annulus 50, whereas bearing 56 is utilized in annulus 52.

Continuing with Figure 2, it will be seen by the utilization of this centrally-disposed cross-sectional view, that I have exhibited that the roller bearings 54 and 56 are arranged to ride inside the hardened steel rings 60 and 62, respectively, in order to provide a minimal friction guide means for tethers 20a, 22a, 24a and 26a. These aforementioned vanes 20, 22, 24, and 26, with a like condition occurring on the opposite side of the machine.

Because of the advantageous techniques I utilize, the tethers, in traveling inside the caged roller bearings disposed in the interior of the respective annuli, will not only experience minimal friction directly, but will also guide vane tips T in a minimal friction relationship with the internal conjugate sidewall 12. Thus, this embodiment of my invention elegantly achieves the paramount goal of yielding a substantially frictionless yet highly effective sealing relationship between the tips T of the vanes and the corresponding conjugate interior surface 12 of casing 10 that can be easily manufactured. The specific means by which the interior surface 12 is developed in accordance with the teachings of this invention will be set forth in detail hereinafter.

As this juncture, however, it is advantageous to realize that the foregoing description can be interpreted in such a fashion as to consider the rings 60 and 62 as behaving as the outer races of conventional roller bearings but with the inner races actually consisting of a plurality of independent circular segments which happen to be pinned to the vanes and thus behave as vane tethers. The caged roller bearings 54 and 56 therefore function in much the same fashion as conventional caged bearing assemblies. Certain of the rollers or roller bearings of the additional tether embodiments in accordance with this invention will be understood to experience both sliding and rolling, much as do the rollers in both full-compliment and retained or caged roller bearings.

As emphasized hereinbefore, an important and basic objective of this invention is to insure positive radially inward vane motion control as well as positive outward vane motion control. This fundamentally important machine function is provided elegantly as shown in Figure 2 by the plain outer diametral surfaces 70 and 72, each being respectively the inner peripheral surfaces of annuli 50 and 52, themselves respective of endplates 40 and 42. The circular peripheral surfaces 70 and 72 serve, through their cooperation with the inner peripheries of the vane tethers, to positively limit the inward radial travel of the vanes. Thus, the combined action of the outward-motion-limiting freely-rotating bearings 54 and 56, operating in conjunction with respective hardened steel rings 60 and 62 with the inner peripheries of inner annular surfaces 70 and 72, serve to positively define the radial motion of the vane tethers moving therebetween. Thus it is to be seen that this arrangement uniquely defines the path of travel of the vane tips, with
vane tip T of vane 22 being shown, for example, in Figure 1a.

Figure 3 is presented to further elucidate the relationships arising among the rotor 14, the rotor slots 200, 202, 204, and 206 and their corresponding vanes 20, 22, 24, and 26 which are shown radially separated from their actual locations within the rotor slots. The radially outwardly disposed governing surface 208 and the radially inwardly governing surface 210 of the annular vane tethery guide are shown in broken lines in Figure 3 in their proper relationship to the rotor center 16. Point 17 is the coincident center of both the circular annulus and the internal stator casing profile 12. It is these surfaces which enclose the vane tethery and the anti-friction bearing means interposed therebetween, and thus dictate the circular anti-friction path of the vane tethers.

Attention should now be given to Figure 4a which presents yet additional detail regarding the anti-friction radial vane guide embodiment discussed in the foregoing. Note especially that this drawing illustrates the construction and cooperation among the outer radial vane guide race 60, the freely-rotating caged bearing 54, and, for example, vane 20, and the inner peripheral annular surface 70. The face end of vane tethery pin 90 is shown here that pivotally connects vane tethery 20a with vane 20.

It is to be understood in Figure 4a that a slight clearance exists, in accordance with embodiments revealed herein, between the underside peripheral surface of the arc segment vane tethers and the circular peripheral surfaces 70 and 72 of annulus 50 and 52. This clearance is important for two reasons.

One reason is because contact with these internal annular surfaces is ordinarily not needed or wanted because the radially positive centripetal forces on the vane assembly during machine operation are usually sufficient to maintain positive outward radial vane motion. Another reason, which is more subtle, arises when my invention is used as a vapor compressor in an air conditioning system. At start-up or during off-design operating conditions, it is not uncommon for a certain amount of liquid refrigerant to occasionally enter the INLET (shown in Figure 1a) of the machine. This occurrence is known as liquid *slugging.*

If no inward radial slack is available to the vane, extreme pressures can sometimes arise within the compression region of the device and potentially cause significant damage to the device. Thus, the interface clearance between the inner annular surfaces 70 and 72 and the underside peripheries of the vane tethers also provide, in the case outlined here, a built-in "safety valve." The amount of clearance required to prevent damage from liquid slugging is relatively slight, being only on the order of 0.2 or 0.2 mm and therefore functions in harmony with the embodiments herein described.

Attention is now directed to Figure 4b, where the second and preferred basic vane tethery assembly is presented. In the case of the vane tethery depicted here, the vane tethery frame 80, which is attached to vane 100 via tether pin 90, is fitted with trunnioned rollers 110. The trunnions 112 of trunnioned roller 110 ride within the circular bottom bearing slots 120 of the vane tethery frame 80. In this arrangement, the freely-rotating retained needle bearing assembly shown previously is eliminated and effectively replaced by the trunnioned rollers residing within the vane tethery frame 80.

Figure 4c portrays yet another combination bi-axial radial vane motion control embodiment. In this case, the peripheries of vane tethery 170 are plain on both the inner and outer surfaces. Both of these outside peripheral tethery surfaces then ride between the outer and larger freely-rotating retained roller bearing assembly 172 and the inner smaller freely-rotating bearing assembly 174. The outer caged freely-rotating bearing 172 thus rides inside bearing race 178 and the inner caged freely-rotating bearing 174 rides over the inner bearing race 178. Such an arrangement as portrayed here also insures positive anti-friction control of both inward and outward radial motion of the vane/vane tethery assemblies.

Figure 4d shows still another positive bi-axial anti-friction radial vane motion control arrangement. In this combination of elements, the outer periphery of the arc segment vane tethery frame 160 is again equipped with rollers 110 whose trunnions 112 engage trunnion slots 120. Again, the trunnioned rollers 110 ride rollingly inside outer bearing race 162. The inner periphery of this tethery segment 160 then engages the inner freely-rotating retained roller bearing 164 which, in turn, rides upon the inner annular bearing race 166.

Shown in Figure 4e is yet another combination positive bi-axial radial vane tethery motion control system. In this embodiment, the vane tethery frame 180 is equipped with trunnioned rollers 110 on its inner periphery. These inner trunnioned rollers then roll over the outer annular peripheral surface 182. However, as seen in the previous embodiments, the outer peripheral surface of tethery frame 180 rides upon the freely-rotating retained roller bearing assembly 184 which, again in turn, rides upon outer annular race 186. Once more, an embodiment is shown that provides positive bi-axial anti-friction radial vane motion.

Figure 4f shows still another double-acting or bi-axial anti-friction vane tethery frame embodiment. In this case, frame 140 is equipped with trunnioned rollers 110 whose trunnions 112 engage outer peripheral trunnion slots 120 and inner trunnion slots 130. Such an arrangement can also be used when positive bi-axial motion is preferred using anti-friction means. In such a case as shown here, the inner trunnioned rollers 110 ride upon the inner peripheral surface of bearing ring race 142. Such particular means is well equipped to handle especially heavy inward radial loads.

As emphasized throughout the foregoing, the geometric shape of the inner wall 12 of the stator casing 10 shown in Figure 1 is critical to the efficient function of
my invention. Appreciation for this governing fact can be seen in Figure 5. Shown here is a magnified view of the special conjugate or mating internal casing profile that is demanded of this invention. In this Figure, the variance of the contour 12 from a pure circle becomes quite apparent. It can be seen that the vane tip T actually recedes significantly inside the path of a true circular contour as the vanes rotate and reciprocate with the rotor.

The reason for this geometric effect is due to the fact that, although the vane tether pin follows a true circle, the necessary rotor-to-stator eccentricity (offset) causes the vanes to tilt at a constantly-varying but cyclic angle with respect to the slope of the inner stator contour. Further, the point or line of tangency at the vane tip T to internal conjugate casing profile 12 continuously changes location with the motion of the vanes. The complex and subtle vane motion thus describes a contour that resembles a circle that is compressed about its equator.

Recall that a fundamental assignment of machines such as disclosed here is to efficiently compress gases or pump liquids. This can be achieved only if the distance between the line of tangency of curved vane tip T and the inner stator contour 12 of casing 10 is very small; on the order of only a few hundredths of a millimeter. Thus, my invention can function at high efficiency only if contour 12 takes on this very special and non-circular shape. If a true circular stator contour was used, and as can be seen in Figure 5, large leakage gaps develop between the vane tip and stator housing wall. The development of such leakage gaps using a true circular stator interior is many times larger than would be acceptable for efficient performance. Therefore, very close attention must be brought to bear in determining the unique shape of the interior stator wall.

With continuing reference to Figure 5, the required geometrical condition can be seen for the vane tip to remain tangent to the inner stator contour 12 at all angular locations of the rotor/vane assembly. I have found that the precise point of tangency of the vane tip with contour 12 can be determined by constructing a line from the geometric center Os of the vane guide ring (which is also the geometric center of the conjugate internal casing contour 12) to the center of the radius of the vane tip, Pvt.

If this special line is extended to intersect the radial contour of the vane tip, this point of intersection (shown in Figure 5 as Pvt) is exactly the location of the corresponding point required to define the conjugate casing interior contour 12. I have used this insight in the creation of the required conjugate stator profile employed in accordance with this invention, the details of which are now presented.

Knowing now the precise geometrical condition required to accurately define the conjugate internal casing contour 12, algebraic and trigonometric relationships can be applied to compute the entire locus of points that define this special contour. A direct computation algorithm for the required internal casing contour can be capitalized as follows in connection with Figure 5:

A. Set initial extended angular location of the vane.
B. Locate the coordinates of the vane pivot pin, Pp, from a knowledge of the vane angle and the radius of the circular radial vane guide.
C. Compute the corresponding angle from the horizontal axis of the stator to the line from the stator center, Os, and the vane tip radius center Pvt, from a knowledge of the dimensions of the vanes and trigonometric functions.
D. Locate the coordinates of the vane tip radius center from the angle found in C above and the linear dimensions of the vane.
E. Finally, locate the coordinates of tangency point Pvt from a knowledge of the vane tip radius and the angle to the center of this vane tip radius from the stator center.
F. Repeat the calculations as needed by incrementing the angular location of the vane to generate the entire locus of points of the required internal conjugate casing contour.

The specific mathematical relationships which code the foregoing are next presented, also in reference to Figure 5:

I. Definition of Initial Nomenclature:

\[ Rg = \text{Radius of annular vane tether guide} \]
\[ Rr = \text{Radius of rotor} \]
\[ Rs = \text{Vertical semi-minor axis of internal stator profile} \]
\[ Rt = \text{Distance from tether pin center to center of vane tip radius} \]
\[ rt = \text{radius of vane} \]
\[ e = Rs - Rg; \text{Rotor eccentricity} \]
\[ Ar = \text{Rotor/vane input angle as measured from the horizontal and repeatedly incrementable to generate locus of conjugate stator profile points} \]

II. Algebraic and Trigonometric Relationships:

1. Cartesian coordinates of vane tether pin centers as measured from the coincident center Os of the conjugate stator profile and the annular vane tether guides -

\[ xg = Rg[\cos(Ar)] \]
\[ yg = Rg[\sin(Ar)] \]

where \( \cos \) and \( \sin \) each represent the trigono-
metric cosine and sine functions, respectively;

2. Angle $A_g$ of line from rotor center through vane tether pin center $P_p$ and through vane tip radius center $P_{v tc}$ as measured from the horizontal rotor axis -

$$A_g = \text{atan}[y_g/x_g]$$

where atan signifies the trigonometric arc tangent function;

3. Radius $R_p$ from rotor center to tether pin center -

$$R_p = \sqrt[x_g^2 + y_g^2]$$

where sqrt signifies the mathematical square root and $^2$ signifies the mathematical square;

4. Radius $R_{tc}$ from rotor center to center of vane tip radius -

$$R_{tc} = R_p + R_t$$

5. Cartesian coordinates of vane tip radius center as measured from the stator profile center -

$$x_{tc} = R_{tc}[\cos(A_r)]$$

$$y_{tc} = R_{tc}[\sin(A_r)] + e$$

6. Angle $A_t$ from stator center to vane tip radius center as measured from the stator horizontal axis -

$$A_t = \text{atan}[y_{tc}/x_{tc}]$$

7. Radius $R_{tc}$ from stator profile center to center of vane tip radius -

$$R_{tc} = \sqrt[x_{tc}^2 + y_{tc}^2]$$

8. Extended radial distance $R_{tt}$ from the stator center to the corresponding point of tangency $P_{vt}$ between the vane tip and the conjugate internal stator contour -

$$R_{tt} = R_{tc} + r_t$$

9. Cartesian Coordinates of vane tip/stator wall tangency point $P_{vt}$ -

$$x_{tt} = R_{tt}[\cos(A_t)]$$

$$y_{tt} = R_{tt}[\sin(A_t)]$$

The combination of angle $A_t$, found in 6 and the extended tangency radius $R_{tt}$, found in 8 defines the polar coordinates of the required conjugate stator profile 12 while the Cartesian coordinates of this same conjugate stator contour are found in 9 as the rotor/vane angle $A_r$ is incremented over 360 angular degrees.

It is to be understood that the very small continuous gap between the vane tip and the conjugate profile in an actual machine is created either by shortening the vane tip in relation to the desired magnitude of this small interface gap or by adding this constant gap width to the conjugate contour itself. That is, in the first case, the actual distance, $R_t$, between the vane tether pin and the center of the vane tip radius, is $R_t$ diminished by a small clearance, say 0.025 mm: $R_t = R_t - 0.025$ mm. Of course, the actual conjugate profile 12 is computed and manufactured on the basis of $R_t$.

In the second case, the distance $R_t$ would remain physically the same, but the profile 12 would be computed and produced on the basis of $R_{tt}$ increased by the small desired gap: $R_{tt} = R_t + 0.025$ mm. Both methods can satisfactorily generate the sealing but non-contact condition between the vane tip and the conjugate stator profile required for efficient operation by this invention.

The present invention can be embodied in ways other than those specifically described here, which were presented by way of non-limitative example only. Variations and modifications can be made without departing from the scope of the invention herein described which are to be constructed and limited only by the following appended claims.

**Claims**

1. A non-contact vane-type fluid displacement machine comprising a casing (10) having around its interior an internal profile (12), said casing being secured between two opposing endplates (40, 42), each endplate (40, 42) containing in its interior a circular annulus (50, 52), the center of each annulus (50, 52) being coincident with the geometric center of said internal casing profile (12), a rotor (14) supported by said endplates (40, 42) and mounted for rotation within said interior of said casing (10) in a matching eccentric relationship with said internal casing profile (12), said rotor (14) having ends operationally disposed in a close fitting relationship.
with said opposing endplates (40, 42), said rotor (14) being equipped with at least one substantially radially disposed slot (200, 202, 204, 206), in each slot (200, 202, 204, 206) being contained a substantially rectangular vane (20, 22, 24, 26) having an arcuate configured tip (T) maintained in an exceedingly close but non-contact relationship with said internal profile (12) of said casing (10), said internal profile (12) having the shape of an envelope resulting from the path of the vane tips (T) being guided by said circular annuli which are eccentric with the rotor axis (44), each end of each vane (20, 22, 24, 26) remote from said vane tip (T) being equipped with a pivotally-mounted tether (20a, 22a, 24a, 26a), each vane tether (20a, 22a, 24a, 26a) having inner and outer peripheries, anti-friction means (54, 62, 56) disposed in each annulus (50, 52) serving as a guide for the respective tethers (20a, 22a, 24a, 26a) and, therefore, for the tips (T) of said vanes (20, 22, 24, 26), characterized in that said anti-friction means are freely rotatable caged roller bearings (54, 56) or trunnioned bearings (110) which engage at least the outer periphery of each tether (20a, 22a, 24a, 26a, 140, 160, 170, 180) during operation of the machine and are in direct contact with the outer periphery of said tethers (20a, 22a, 24a, 26a),

2. Machine in accordance with claim 1, in which said anti-friction means are in direct contact with and engage both the inner and the outer periphery of each tether (170) and include freely-rotating caged roller bearings (172) on the outer periphery freely-rotating caged roller bearings (174) on the inner periphery of each tether (170).

3. Machine in accordance with claim 1, in which said anti-friction means are in direct contact with and engage both the inner and the outer periphery of each tether (160) and include, trunnioned bearings (110) on the outer periphery and freely rotating caged roller bearings (164) on the inner periphery of each tether (160).

4. Machine in accordance with claim 1, in which said anti-friction means are in direct contact with and engage both the inner and the outer periphery of each tether (180) and include freely rotating caged roller bearings (184) on the outer periphery and trunnioned bearings (110) on the inner periphery of each tether (180).

5. Machine in accordance with claim 1, in which said anti-friction means are in direct contact with and engage the inner and the outer periphery of each tether (140) and include trunnioned bearings (110) on the outer and on the inner periphery of each tether (140).

6. Machine in accordance with claim 3, characterized in that the trunnions (112) of said trunnioned bearings (110) are installed within the outer peripheries of said vane tethers (160, 140) said trunnioned roller bearings being rollingly engaged with the outer periphery of said annulus (50, 52) within said end plates (40, 42).

7. Machine in accordance with claim 4, characterized in that the trunnions (112) of said trunnioned bearings (110) are installed within the inner peripheries of said vane tethers (140, 180) said trunnioned roller bearings (110) being rollingly engaged with the inner periphery of said annulus (50, 52) within said end plates (40, 42).

8. Machine in accordance with claim 1, further characterized in that at least one of the peripheral surfaces of said annuli (50, 52) of said end plates (40, 42) is fitted with separate hardened precision races (60, 62) to accommodate the bearing loads exerted by said vane tethers (20a, 22a, 24a, 26a).

9. Machine in accordance with claim 1, further characterized in that a small distance is maintained between the inner peripheries of said vane tethers (20a, 22a, 24a, 26a) and the inner periphery (70, 72) of said annulus (50, 52) of said end plates (40, 42), said small distance providing inward radial slack in the radial position of said vane (20, 22, 24, 26) in order to provide a purposeful leakage path between said vane tips (T) and said internal casing profile (12) for said compressed fluid in the event of inadvertently high pressure development inside said machine.

**Patentansprüche**

1. Kontaktfreie Drehflügelzellenmaschine für Fluid mit einem Gehäuse (10), das in seinem Inneren ein Innenprofil (12) aufweist, wobei das Gehäuse zwischen zwei gegenüberliegenden Endplatten (40, 42) befestigt ist und jede Endplatte (40, 42) in ihrem Inneren einen kreisförmigen Ring (50, 52) umfaßt, wobei die Mitte eines jeden Ringes (50, 52) mit der geometrischen Mitte des inneren Gehäuseprofils (12) übereinstimmt, einem Rotor (14), der von den Endplatten (40, 42) getragen und zwecks Rotation innerhalb des Inneren des Gehäuses (10) in einer exzentrischen Anpassungsbeziehung mit dem inneren Gehäuseprofil (12) montiert ist, wobei der Rotor (14) Enden aufweist, die im Betrieb in enger Anpassung an den gegenüberliegenden Endplatten (40, 42) angeordnet sind, wobei der Rotor (14) mit wenigstens einem im wesentlichen radial ange-
ordneten Schlitz (200, 202, 204, 206) ausgestattet ist, wobei in jedem Schlitz (200, 202, 204, 206) ein im wesentlichen rechteckiger Flügel (20, 22, 24, 26) aufgenommen ist, der eine bogenförmig gestaltete Spitze (T) aufweist, welche in einer äußerst nahen, aber kontaktfreien Beziehung zu dem Innenprofil (12) des Gehäuses (10) gehalten ist, wobei das Innenprofil (12) die Form einer Ummantelung aufweist, die sich aus dem Weg der Flügelspitzen (T) ergibt, die von den kreisförmigen Ringen geführt sind, welche exzentrisch zur Rotorachse (44) sind, wobei jedes Ende eines jeden Flügels (20, 22, 24, 26), das von dem Flügelende (T) entfernt ist, mit einem schwenkanlen montierten Halsteil (20a, 22a, 24a, 26a) ausgestattet ist, jedes Flügelhalsteil (20a, 22a, 24a, 26a) einen Innen- und einen Außenumfang aufweist, wobei Antifrikionsmittel (54, 56, 56), die in jedem Ring (50, 52) angeordnet sind, als eine Führung für die entsprechenden Halteile (20a, 22a, 24a, 26a) und deshalb auch für die Spitzen (T) der Flügel (20, 22, 24, 26), dienen, dadurch gekennzeichnet, daß die Antifrikionsmittel frei drehbar ummantelte Rollenlager (54, 56) oder Zapfenlager (110) sind, die mindestens mit dem Außenumfang eines jeden Halteile (20a, 22a, 24a, 26a, 140, 160, 170, 180) während des Betriebes der Maschine im Eingriff stehen und in direktem Kontakt mit dem Außenumfang der Halteile (20a, 22a, 24a, 26a) sind.

2. Maschine nach Anspruch 1, wobei die Antifrikionsmittel mit dem Innen- und Außenumfang eines jeden Halteile (170) in direktem Kontakt und in Eingriff stehen und sich frei drehende, ummantelte Rollenlager (172) an dem Außenumfang und sich frei drehende Rollenlager (174) an dem Innenumfang eines jeden Halteile (170) umfassen.

3. Maschine nach Anspruch 1, wobei die Antifrikionsmittel mit dem Innen- und dem Außenumfang eines jeden Halteile (160) in direktem Kontakt und in Eingriff stehen und Zapfenlager (110) an dem Außenumfang und frei drehbar gelagerte Rollenlager (164) an dem Innenumfang eines jeden Halteile (160) umfassen.

4. Maschine nach Anspruch 1, wobei die Antifrikionsmittel mit dem Innen- wie auch dem Außenumfang eines jeden Halteile (180) in direktem Kontakt und im Eingriff stehen und frei drehbar gelagerte Rollenlager (184) an dem Außenumfang und Zapfenlager (110) an dem Innenumfang eines jeden Halteile (180) umfassen.

5. Maschine nach Anspruch 1, wobei die Antifrikionsmittel mit dem Innen- und Außenumfang eines jeden Halteile (140) in direktem Kontakt und im Eingriff stehen und Zapfenlager (110) an dem Außenumfang eines jeden Halteile (140) umfassen.

6. Maschine nach Anspruch 3, dadurch gekennzeichnet, daß die Zapfen (112) der Zapfenlager (110) innerhalb des Außenumfanges der Flügelhalteile (160, 140) angeordnet sind, wobei die Zapfenlager in rollendem Eingriff mit dem Außenumfang des Ringes (50, 52) innerhalb der Endplatten (40, 42) stehen.

7. Maschine nach Anspruch 4, dadurch gekennzeichnet, daß die Zapfen (112) der Zapfenlager (110) innerhalb des Innenumfanges der Flügelhalteile (140, 180) angeordnet sind, wobei die Zapfenlager (110) rollend mit dem Innenumfang des Ringes (50, 52) innerhalb der Endplatten (40, 42) im Eingriff stehen.

8. Maschine nach Anspruch 1, dadurch gekennzeichnet, daß mindestens eine der Umfangsflächen der Ringe (50, 52) der Endplatten (40, 42) auf separate, gehärtete Präzisionslaufringe (60, 62) aufgepaßt ist, um Lagerbeanspruchungen aufzunehmen, die von den Flügelhalteile (20a, 22a, 24a, 26a) ausgelöst werden.

9. Maschine nach Anspruch 1, dadurch gekennzeichnet, daß ein geringer Abstand zwischen dem Innen- und Außenumfang der Flügelhalteile (20a, 22a, 24a, 26a) und dem Innenumfang (70, 72) des Ringes (50, 52) der Endplatten (40, 42) aufrechterhalten ist, wobei der geringe Abstand nach innen gerichteten radialen Schlupf in der radialen Position des Flügels (20, 22, 24, 26) schafft, um im Falle einer unbeabsichtigten hohen Druckentwicklung innerhalb der Maschine einen zweckmäßigen Leckageweg zwischen der Flügelspitze (T) und dem Gehäuseinnenprofil (12) für das komprimierte Fluid zu schaffen.

Revidierungen

1. Dispositif de déplacement d’un fluide du type à pales sans contact comportant un carter (10) ayant autour de sa partie intérieure un profil intérieur (12), ledit carter étant fixé entre deux plaques d’extrémité opposées (40, 42), chaque plaque d’extrémité (40, 42) comportant à l’intérieur de celle-ci un anneau circulaire (50, 52), le centre de chaque anneau (50, 52) coïncidant avec le centre géométrique dudit profil de carter intérieur (12), un rotor (14) supporté par lesdites plaques d’extrémité (40, 42) et monté pour être mis en rotation dans la partie intérieure dudit boîtier (10) selon une relation excentrée complémentaire dudit profil intérieur (12) de carter, ledit rotor (14) ayant des extrémités agencées de ma-
nière opérationnelle selon un agencement proche desdites plaques d'extrémité opposées (40, 42), ledit rotor (14) étant muni d'au moins une fente agencée à peu près radialement (200, 202, 204, 206), dans chaque fente (200, 202, 204, 206) étant agencée une palette à peu près rectangulaire (20, 22, 24, 26) ayant un bout configuré de manière arquée (T) maintenu d'une manière excessivement proche dudit profil intérieur (12) dudit carter (10) mais sans contact avec celui-ci, ledit profil intérieur (12) ayant la forme d'une enveloppe résultant du trajet des bouts de palette (T) guidé par lesdits anneaux circulaires qui sont excentrés par rapport à l'axe du rotor (44), chaque extrémité de chaque palette (20, 22, 24, 26) éloignée dudit bout de pale (T) étant munie d'un amarrage agencé de manière pivotante (20a, 22a, 24a, 26a), chaque amarre de palette (20a, 22a, 24a, 26a) ayant des périphériques intérieure et extérieure, des moyens d'antifriction (54, 62, 56) agencés dans chaque anneau (50, 52) servant en tant que guide des amarres respectives (20a, 22a, 24a, 26a) et, par conséquent, des bouts (T) desdites palettes (20, 22, 24, 26), caractérisé en ce que lesdits moyens antifriction sont des roulements à rouleaux (54, 56) à cage librement rotative ou des paliers à pivots (110) qui viennent en contact avec au moins la périphérie extérieure de chaque amarre (20a, 22a, 24a, 26a, 140, 160, 170, 180) pendant le fonctionnement de la machine et sont en contact direct avec la périphérie extérieure desdits amarres (20a, 22a, 24a, 26a).

2. Dispositif selon la revendication 1, dans lequel lesdits moyens antifriction sont en contact direct avec les périphériques intérieure et extérieure de chaque amarre (170) et en prise avec celles-ci et comportent des roulements à rouleaux (172) à cage librement rotative sur la périphérie extérieure, et des roulements à rouleaux (174) à cage librement rotative sur la périphérie intérieure de chaque amarre (170).

3. Dispositif selon la revendication 1, dans lequel lesdits moyens antifriction sont en contact direct avec les périphériques intérieure et extérieure de chaque amarre (160) et en prise avec celles-ci et comportent des paliers à pivots (110) sur la périphérie extérieure et des paliers à rouleaux (164) à cage librement rotative sur la périphérie intérieure de chaque amarre (160).

4. Dispositif selon la revendication 1, dans lequel lesdits moyens antifriction sont en contact direct avec les périphériques intérieure et extérieure de chaque amarre (180) et en prise avec celles-ci et comportent des roulements à rouleaux (184) à cage librement rotative sur la périphérie extérieure et des pa-