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ABSTRACT

In the Revolving Vane (RV) compressor being introduced recently (Teh and Ooi, 2006), it has been shown that there remains significant friction occurring at the vane sides although the overall frictional loss in the compressor is expected to be small. In order to further improve the mechanical efficiency of the RV compressor, this paper presents a design improvement, named 'Improved Revolving Vane (RV-i) compressor', in which the wear and friction at the vane sides are effectively reduced by rigidly attaching the vane to the rotor. This is because in doing so, the vane side contact forces are relieved of the pressure forces acting across the vane. The frictional losses of the new compressor are formulated and analyzed, where results have shown that the friction are reduced mainly to that at the bearings, achieving a 30 % reduction in frictional losses over its predecessor.

1. INTRODUCTION

In order to improve the mechanical efficiency of rotary type compressors, the Revolving Vane (RV) compressor was introduced two years ago by Teh and Ooi (2006) whom had theoretically assessed the friction characteristics in the new compressor. It has been found that due to the use of a rotating cylinder, the relative velocities between sliding components in the compressor mechanism are effectively reduced, thereby decreasing the friction therein. However, because of significant contact forces at the vane sides which are to a large extent caused by the pressure differential across the vane, there is still substantial frictional loss at that region. Furthermore, due to the reciprocatory motion of the vane in the vane slot, boundary friction dominates which results in wear of the vane sides. Such a situation is similar to that in the existing rolling piston compressor as reported by Yanagisawa and Shimizu (1985). Although the vane side wear can be alleviated by passive methods such as surface hardening, it restricts the operating conditions of the compressor such that the pressure force across the vane is kept to within safe limits. This implies that the RV compressor, alike the rolling piston compressor, may not be suitable for use with carbon dioxide as a working fluid.

In order to overcome such limits on the RV compressor and to further improve its mechanical efficiency, an improved design of the RV compressor is presented in this paper.

2. DESIGN AND WORKING PRINCIPLE

Figure 1 shows the basic construction of the 'Improved' Revolving Vane (RV-i) compressor which mainly consists of a rotor, a vane rigidly fixed to (or as a part of) the rotor, and a cylinder. A vane slot on the cylinder which features a convergent-divergent geometry capable of providing a two-degree of freedom motion connects the vane to the cylinder. The rotor and cylinder are supported individually and concentrically on bearing pairs, where their respective axes of rotation are separated by a distance such that a virtual line contact always exists between the two components (at φ = 0 rad). During operation, the rotation of the rotor revolves the vane which in turn rotates the cylinder. The motion causes the volumes trapped within the rotor, vane and cylinder to vary, resulting in suction, compression and discharge of the working fluid.

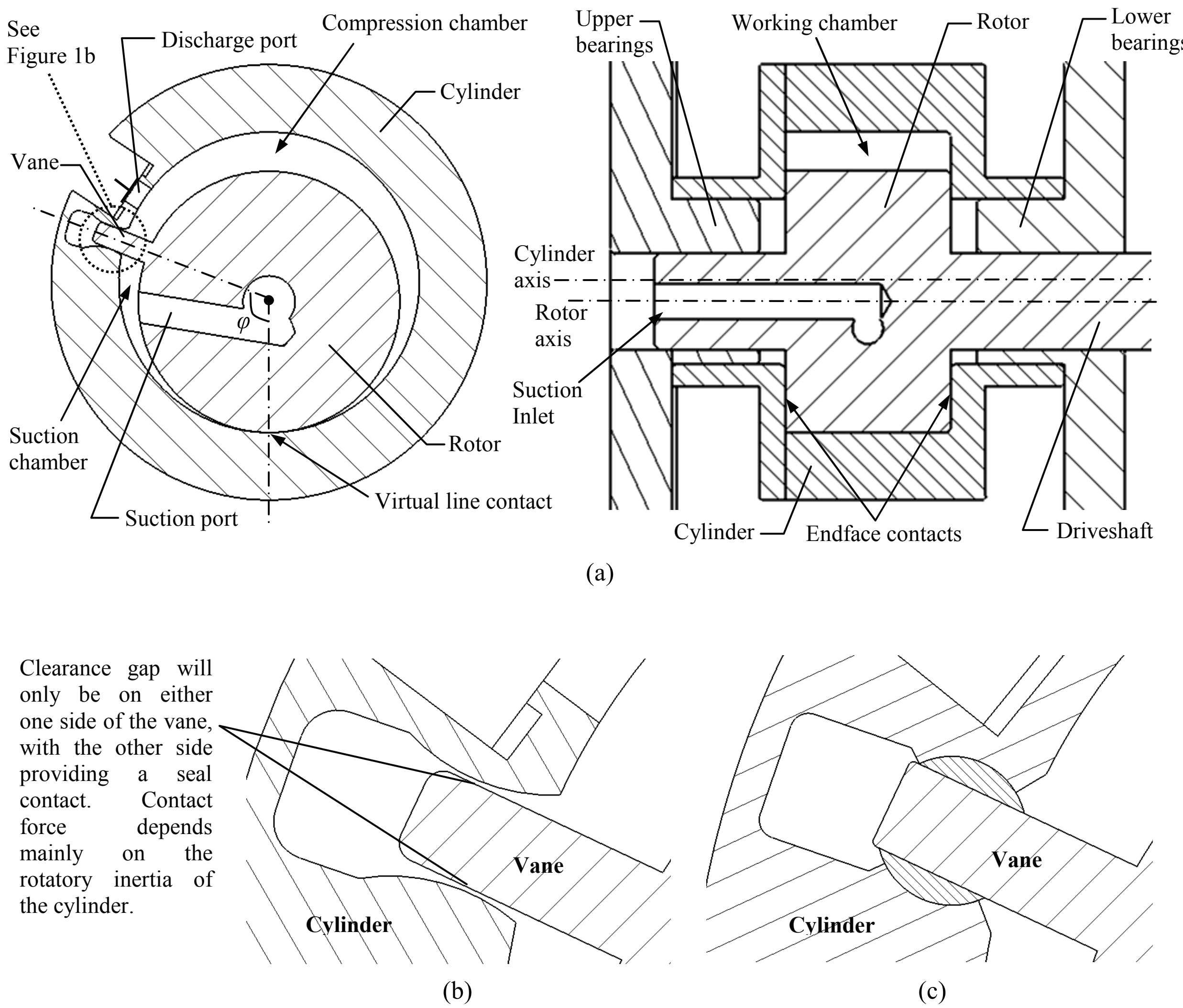

Figure 1: Schematics of the RV-i compressor: (a) Front and side sectional views; (b) Detailed view of 'clearance' joint; (c) Alternative rotary-sliding joint

Comparing the RV-i and RV compressors, it is noticed that the two designs are largely similar to each other with differences existing only at the vane connections. This modification, although seemingly minor, affects the performance of the compressor to a significant extent, as demonstrated later in a friction analysis. In order to allow the simultaneous rotating and sliding motion of the vane relative to the cylinder, a 'clearance joint' at the cylinder wall is proposed mainly for the reason of simplicity. Figure 1b shows its basic geometry which incorporates a clearance fit within a convergent-divergent slot to allow the vane to swivel and slide at the same time. Although a conventional rotary-sliding joint shown in Figure 1c can be used, additional parts are needed which complicates the assembly. Furthermore, the conventional joint is foreseen to cause difficulties when the compressor is being miniaturized. The 'clearance joint', on the other hand, only requires an accurate machining process to be made. Nevertheless, further investigation is necessary to determine the exact geometry of such a new feature, which is not discussed in this paper.

3. FRICTIONAL LOSSES

There are a total of three contact regions in the RV-i compressor at which friction occurs, namely, a) at the contact between the cylinder and the vane, b) at the journal bearings, and c) between the endfaces of the rotor and the cylinder. Amongst these three regions, the endface friction can be expected to be comparatively insignificant due to small differences in the angular velocities of the rotor and cylinder, as well as the absence of substantial contact loads at that region. Nevertheless, the analyses of all three sources of friction shall be presented. In this investigation, the RV-i compressor is designed for air-conditioning uses, where its operating specifications and main dimensions are given in Table 1.

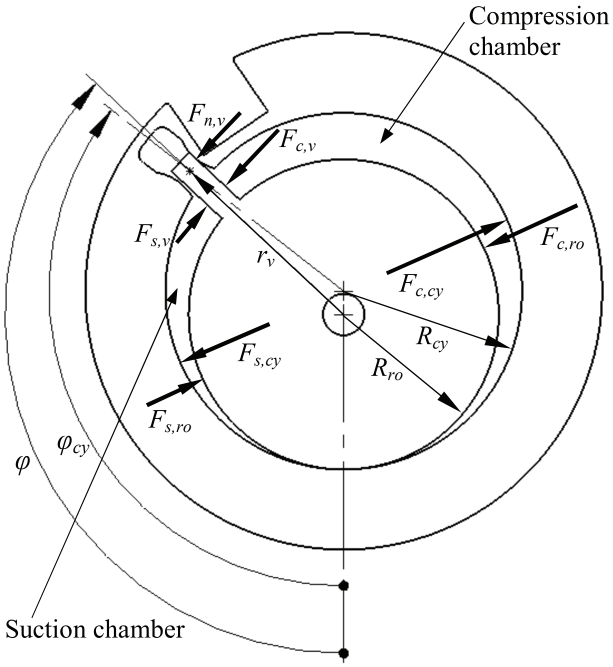

Figure 2: Forces in the RV-i compressor

Table 1: Operating specifications and main dimensions of RV-i compressor

Operating specifications	
volumetric displacement, V_c	32.46 cm^3/rev
operational speed, ω_{ro}	2875 rev/min
working fluid	R22
suction pressure, P_s (abs)	0.63 Mpa
discharge pressure, P_d (abs)	2.17 Mpa
Main dimensions	
rotor radius, R_{ro}	30 mm
cylinder radius, R_{cy}	34.5 mm
moment of inertia, I_{cy}	0.2 mkg m^2
compressor axial length, L_c	35.6 mm
rotor bearing radius, $R_{b,ro}$	11 mm
total length, $L_{b,ro}$	56 mm
radial clearance, $\delta_{b,ro}$	13 µm
cylinder bearing radius, $R_{b,cy}$	17.5 mm
total length, $L_{b,cy}$	34 mm
radial clearance, $\delta_{b,cy}$	13 µm
endface clearance, δ_{ef}	7.5 µm
lubricant dynamic viscosity, μ	3.4 mPa s

3.1 Friction at the Vane Sides

During operation, the vane slides within the vane slot in the cylinder. Due to the reciprocatory nature of the sliding motion, boundary friction is anticipated to occur for most part of the contact. With reference to Figure 2, the frictional loss at this region can be calculated by the equation:

$$P_{f,vs} = \eta_{vs} \left| F_{n,v} \dot{r}_v \right| \tag{1}$$

Due to the nature of the vane side contact being similar to that at the vane tip in the rolling piston compressor, the kinetic friction coefficient, η_{vs}, is assumed a constant value of 0.15 proposed by Yanagisawa *et al*. (1982). Assuming that the vane swivels about a point close to the inner surface of the cylinder, the sliding velocity of the vane can be found to have the following expression:

$$\dot{r}_v = \frac{dr_v}{dt} = \frac{d\varphi}{dt}\frac{dr_v}{d\varphi} = \omega_{ro}\frac{d}{d\varphi}\left\{ R_{cy}\left[\sqrt{1-\left(1-\frac{R_{ro}}{R_{cy}}\right)^2 \sin^2\varphi} - \left(1-\frac{R_{ro}}{R_{cy}}\right)\cos\varphi \right] \right\} \tag{2}$$

When the vane rotates the cylinder, a normal contact force, $F_{n,v}$, arises on either one side of the vane mainly due to the inertia of the cylinder as a result its angular velocity being constantly varying. Neglecting the frictional resistance at the cylinder bearings which is comparatively small, $F_{n,v}$ can be given by:

$$F_{n,v} = \frac{I_{cy}\ddot{\varphi}_{cy}}{R_{cy}\cos\left(\varphi - \varphi_{cy}\right)} \tag{3}$$

in which the angular acceleration of the cylinder has the formulation:

$$\ddot{\varphi}_{cy} = \frac{d^2\varphi_{cy}}{dt^2} = \left(\frac{d\varphi}{dt}\right)^2 \frac{d^2\varphi_{cy}}{d\varphi^2} = \omega_{ro}^2 \frac{d^2}{d\varphi^2}\left\{\cos^{-1}\left[\frac{R_{cy}^2 + \left(R_{cy} - R_{ro}\right)^2 - r_v^2}{2R_{cy}\left(R_{cy} - R_{ro}\right)}\right]\right\} \tag{4}$$

Figure 3 shows the variation of the frictional loss at the vane sides and its components for one complete shaft revolution. The normal contact force can be observed to vary accordingly to the rate of change of the cylinder velocity shown in Figure 4a. The range of the contact force magnitude in this case is reasonably small, which can be expected to increase with a larger rotatory inertia of the cylinder. Therefore, the vane friction can be reduced by using a lighter cylinder. As the contact force changes polarity at $\varphi = 0, \pi$ rads, the contact point on the vane can be expected to change sides during those two positions. The sliding velocity varies in a similar manner as the contact force which results in the vane side friction to have an analogous trend. For comparison, the vane side friction of the RV compressor having the same specifications given in Table 1 is shown in the same figure. It can be clearly seen that a significant improvement has been achieved in the RV-i compressor. Most importantly, the contact forces and friction at the vane side are no longer dependent on the large pressure differential across the vane.

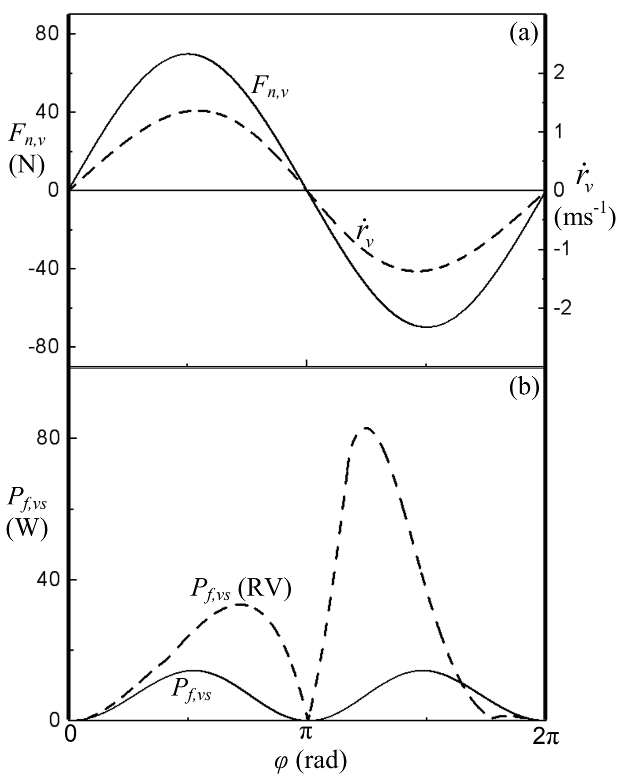

Figure 3: (a) Normal contact force at vane side and sliding velocity; (b) Frictional loss at vane sides

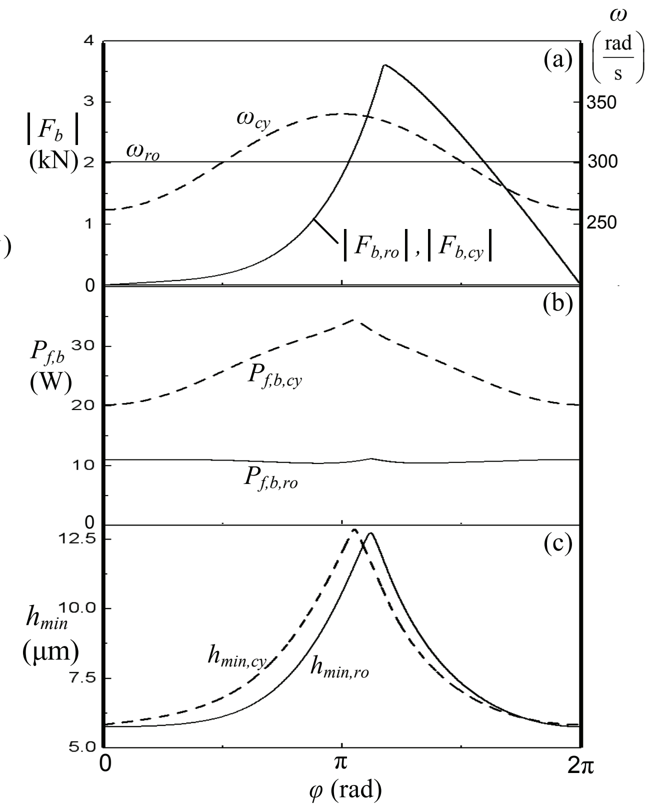

Figure 4: (a) Bearing forces and angular velocities; (b) Frictional loss at journal bearings; (c) Variation of minimum oil film thickness

3.2 Friction at the Journal Bearings

The journal bearings are an important element of a compressor which affects its reliability and performance to a large extent. As such, both the rotor and cylinder bearings must be properly sized so that full hydrodynamic lubrication is maintained throughout the entire shaft revolution, but yet not causing unnecessarily large friction. At each bearing, the frictional loss is calculated by the expression:

$$P_{f,b,J} = \frac{\mu \omega_J^2 R_{b,J}^3 L_{b,J} \pi}{\delta_{b,J} \sqrt{1-\varepsilon_J^2}} \left(\frac{2+\varepsilon_J}{1+\varepsilon_J} \right) + \frac{\omega_J \delta_{b,J} \varepsilon_J}{2} \sqrt{F_{bx,J}^2 + F_{by,J}^2} \sin \Phi_J \tag{5}$$

where subscript $J = ro, cy$ for the respective rotor and cylinder bearing. Due to the dynamic nature of the bearing forces which constantly change in magnitude and direction , the eccentricity ratio, ε_J, and the attitude angle, Φ_J, for each bearing are found using a method of solution proposed by Hirani *et al*. (1999) for dynamically loaded finite length journal bearings. With reference to Figure 2, the journal load components, F_{bx} and F_{by} on the respective rotor and cylinder journals can be obtained from force summations in the horizontal and vertical directions to give:

$$\begin{aligned} F_{bx,ro} &= F_{sx,ro} - F_{cx,ro} + \left(F_{c,v} - F_{s,v} \right) \cos\varphi + F_{n,v} \cos\varphi - \eta_{vs} \left| F_{n,v} \right| \operatorname{sgn}\left(\dot{r}_v \right) \sin\varphi , \\ F_{by,ro} &= F_{sy,ro} - F_{cy,ro} - \left(F_{c,v} - F_{s,v} \right) \sin\varphi - F_{n,v} \sin\varphi - \eta_{vs} \left| F_{n,v} \right| \operatorname{sgn}\left(\dot{r}_v \right) \cos\varphi \end{aligned} \tag{6}$$

$$\begin{aligned} F_{bx,cy} &= F_{cx,cy} - F_{sx,cy} - F_{n,v} \cos\varphi + \eta_{vs} \left| F_{n,v} \right| \operatorname{sgn}\left(\dot{r}_v \right) \sin\varphi, \\ F_{by,cy} &= F_{cy,cy} - F_{sy,cy} + F_{n,v} \sin\varphi + \eta_{vs} \left| F_{n,v} \right| \operatorname{sgn}\left(\dot{r}_v \right) \cos\varphi \end{aligned} \tag{7}$$

Figure 4 shows the variation of the bearing loss and its components for one shaft revolution. In Figure 4a, it is observed that the magnitudes of the rotor and cylinder journal loads are equal to each other exactly, where they reach a maximum of 3.6 kN in the later half of the shaft revolution due to a large pressure differential. The exact similarity of the journal load magnitudes implies the absence of unbalanced forces in the compressor; it has been shown by Teh and Ooi (2008) that the respective bearing forces remain in opposite directions with equal magnitudes throughout the entire shaft revolution, which indicates a perfect dynamic balance. Therefore, from this observation, the RV-i compressor can be expected to exhibit a lower vibration.

In the same figure, it is shown that for a constant rotational velocity of the rotor, the angular velocity of the cylinder is of a sinusoidal profile which magnitude of variation depends on the offset distance between the bearing centers. Comparing Figures 4a and 4b, it is observed that the frictional loss of each bearing follows a similar trend to its angular velocity, indicating that the bearing friction is largely unaffected by load variations. This is only possible if the bearing eccentricities are adequately small such that the minimum film thickness of the lubrication oil is always maintained above a safe value, as shown in Figure 4c. Each bearing is comfortably operated within a maximum eccentricity ratio of 0.6. However, due to the larger radius of the cylinder bearing, it is clearly noticed that the cylinder bearing friction is substantially larger than that of the rotor bearing, despite providing a similar load capacity. Therefore, it is preferred that the radii of both bearings be minimized while having a longer length. It is important to optimize the bearing design as the bearing friction is the only significant frictional loss in the RV-i compressor.

3.3 Friction at the Endfaces

At the upper and lower endface contacts, friction occurs between the endfaces of the rotor and the cylinder due to viscous shear of the oil film between the two components. If the oil flow velocity is small, a full Couette shear dominates which allows the total endface frictional loss to be easily approximated by the expression:

$$P_{f,ef} = 2 \times \frac{\mu A V^2}{\delta_{ef}} \tag{8}$$

As both the rotor and the cylinder are rotating, the motion of the rotor relative to the cylinder is mainly an orbiting motion similar to that of the moving scroll in the scroll compressor. The orbital sliding velocity and area of contact at each endface are respectively defined by:

$$V = \omega\left(R_{cy} - R_{ro}\right), \qquad A = \pi\left(R_{ro}^2 - R_{b,cy}^2\right) \tag{9}$$

In the present case, the endface loss is found to be 3.1 W, which is comparatively insignificant even though the axial clearance at each endface is rather small. It is to be noted that such a small endface clearance is necessary to minimize the leakage loss at the endface region (Teh and Ooi, 2008).

4. COMPARISON OF MECHANICAL EFFICIENCY

From the above analyses, it can be concluded that the frictional losses in the improved design are mainly reduced to that at the bearings, which is an advantageous feature peculiar to the RV-i compressor. In the present configuration, its theoretical mechanical efficiency is 96.5 %, typically higher than those reported of the rolling-piston and scroll compressors. Table 2 shows a comparison of the mechanical efficiencies amongst several refrigeration compressors of various capacities reported in available literatures.

Table 2: Comparison of mechanical efficiencies amongst various compressors

S/n	Compressor Type	Mechanical Efficiency (%)	Investigators
1.	Rolling Piston	85 – 90	Ozu and Itami, 1981
2.	Rolling Piston	93.3	Matsuzaka and Nagatomo, 1982
3.	Rolling Piston	92.5	Wakabayashi *et al.*, 1982
4.	Rolling Piston	92.0 – 92.2	Sakaino *et al.*, 1984
5.	Rolling Piston	91.8	Ishii *et al.*, 1990a
6.	Scroll	92.3 – 92.7	Hayano *et al.*, 1988
7.	Scroll	92.3 – 93.4	Ishii *et al.*, 1990b, 1992, 1994
8.	RV	95.1	present
9.	RV-i	96.5	present

5. CONCLUSIONS

The design of the RV-i compressor and its friction characteristics have been presented in this paper. By the rigid attachment of the vane to the rotor, the contact forces at the vane side are relieved of the pressure differential across the vane, which significantly alleviates the wear and friction at that region. In consequence, the frictional losses in the new compressor have been reduced mainly to that at the bearings, which predictably leads to a higher mechanical efficiency over its predecessor as well as those of existing compressors. Furthermore, the RV-i compressor is also expected to have a high volumetric efficiency as shown by Teh and Ooi (2008). Besides good performance, the RV-i compressor also has the ability to achieve a complete dynamic balance. In further consideration of the low number of parts and simple component geometries, it is believed that the RV-i compressor can be readily employed in various refrigeration/air-conditioning applications and achieve high COP values. Further development is in progress.

NOMENCLATURE

F	force [N]	P_f	frictional loss [W]
h_{min}	minimum oil film thickness [m]	R	prescribed radius [m]
I	second moment of inertia [kg m^2]	r	radius [m]
L	length [m]	r_v	radial distance from rotor center to neck of vane slot [m]
L_c	axial length of chamber volumes [m]		
P	pressure [Pa]	t	time [s]

V_c	displaced volume = $\pi\left(R_{cy}^2 - R_{ro}^2\right)L_c$ [m³]
δ	clearance [m]
ε	bearing eccentricity ratio [-]
η_{vs}	kinetic friction coefficient at vane sides [-]
μ	dynamic viscosity of lubricant [N s m^{-2}]
φ	driver/rotor angle [rad]
φ_{cy}	cylinder angle [rad]
Φ	bearing attitude angle [rad]
ω	angular velocity [rad s^{-1}]

Subscripts

b	of journal bearing
c	compression
cy	of cylinder
d	discharge
ef	at endface
n	normal to
ro	of rotor
s	suction
v	at vane
vs	at vane sides

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