Design and Friction Analysis of the Revolving Vane Compressor

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ABSTRACT

Over the years, rotary compressors have gained significant popularity and widespread usage because of their compact nature and good vibration characteristics. Although possessing many advantages over the reciprocating compressors, one of the performance limitations of the rotary machines is their inferior mechanical efficiency due to the existence of large frictional losses, which occurs as a result of the high relative velocities between components in sliding contact. In efforts to effectively improve the mechanical efficiencies of rotary compressors, a new compressing mechanism has been developed which engages a radical concept of operation that is anticipated to produce a substantial reduction of the existing friction. A friction analysis taking into consideration the kinematics and dynamics of the novel machine is being conducted, where simulated results have revealed a superior mechanical efficiency as compared to other existing designs.

1. INTRODUCTION

One of the crucial factors affecting the performance of a compressor is its mechanical efficiency. For example, the reciprocating piston-cylinder compressor exhibits good mechanical efficiency, but its reciprocating action results in significant vibration and noise problems. To negate such problems, rotary type compressors have been developed and have since gained much popularity due to their compact nature and good vibration characteristics. However, as their parts in sliding contact generally possess high relative velocities, frictional losses are predominant and have thus limited the efficiency of the machines. For instance, in Rotary Vane compressors, the slider vane tips rub against the cylinder interior at high velocities, resulting in enormous frictional losses (Kruse, 1982 and Ma et al., 2002). Similarly, in Rolling-Piston compressors, the high relative speeds between sliding components such as the eccentric and the rolling piston also result in significant losses (Yanagisawa and Shimizu, 1985). It is therefore believed that if the relative velocities of the rubbing components in rotary compressors can be effectively reduced, their overall performance can be improved substantially.

This paper presents the theoretical work of such an attempt to improve the mechanical efficiency of rotary type compressors by introducing a newly designed compression mechanism, named the Revolving Vane compressor.
2. DESIGN AND WORKING PRINCIPLE

In its most basic form, the Revolving Vane (RV) compressor is comprised of similar elements as that of the Rotary Vane compressor, as shown in Figure 1. A sliding vane is housed in a rotor which rotates about a first axis of rotation. The tip of the sliding vane is connected via a pin-type joint to a cylinder housing of a larger diameter, where the latter is allowed to rotate about a second axis of rotation. There exists an offset distance between the first and second axes of rotation such that a line contact is made between the rotor and the cylinder. In the present configuration, a drive shaft is integrated with the rotor concentrically which is to be coupled to a prime driver. During operation, the rotation of the rotor component spins the sliding vane which in turn sets the cylinder in rotary motion. The sliding vane in rotation causes the volumes of the suction and compression chambers to vary, resulting in suction, compression and discharge of the working fluid.

![Diagram of RV Compressor with Key Components](image)

Figure 1: (Left) Front sectional view of RV compressor; (Right) Side sectional view of RV compressor with journal supports shown

One who is familiar with the operation of the Rotary Vane compressor would find significant similarities between the former and the RV design. In fact, the only modification made was that the cylinder housing is made to rotate instead of being fixed. The reason of doing so lies in the need to effectively reduce the drastic friction between the sliding vane/s and the stationary cylinder housing in the Rotary Vane compressor (Kruse, 1982 and Ma et al., 2002), which in the RV machine was almost non-existent. Although this friction seems to be virtually eliminated at first glance, it is not quite as simply because an additional two bearings are required to support the rotating cylinder in the RV design, thus resulting in additional journal losses. However, as the friction at the journal bearings can be more adequately controlled with careful bearing design and a good lubrication system, it is expected that overall frictional losses should decrease. The purpose of this paper is to provide a theoretical assessment of the friction in the RV compressor in order to validate this conjuncture.

As it can be observed that all the primary components of the RV compressor are rotating, the suction and discharge ports would also be inevitably set in motion. In this paper, the RV concept is being applied to a hermetic air-conditioning compressor which is designed to be of a high-side configuration, where its specifications are shown in Table 1. The suction inlet is located along the axis of rotation of the rotor which is exposed directly to the suction pipe. The discharge port is positioned on the cylinder housing and therefore the discharged gas is contained within the hermetic shell before being delivered out of the compressor unit. A discharge valve is thus required. As the valve is rotating together with the cylinder at high speeds, centrifugal effects on the valve component cannot be neglected. Such a situation has been modeled by Teh and Ooi (2006) whom had found that under proper design the reliability and performance of the valve can be enhanced by the centrifugal loads. As such, although radical, rotating inlet and discharge manifolds are anticipated to be of no detrimental nature to the compressor’s operation.
Table 1: Design specifications and main dimensions

<table>
<thead>
<tr>
<th>Design Specifications</th>
<th>Main Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric Displacement</td>
<td>Suction temp 338.3 K</td>
</tr>
<tr>
<td>Cooling capacity 5.3 KW</td>
<td>Discharge temp 399.3 K</td>
</tr>
<tr>
<td>Operational speed, ( \omega_d ) 2875 rpm</td>
<td>Suction pressure 0.63 MPa</td>
</tr>
<tr>
<td>Working fluid R22</td>
<td>Discharge pressure 2.17 MPa</td>
</tr>
<tr>
<td>Rotor radius, ( r_r ) 25 mm</td>
<td>Axial length, ( L_w ) 38 mm</td>
</tr>
<tr>
<td>Cylinder radius, ( r_{cy} ) 30 mm</td>
<td>Vane length, ( L_v ) 21 mm</td>
</tr>
</tbody>
</table>

### 3. FRICTIONAL LOSSES

There are mainly three regions of contact interaction in the RV compressor where frictional losses are significant, namely at the vane sides and the rotor and cylinder bearings. There also exists relative motion between the rotor and cylinder at the end faces, but however as both components are rotating, the sliding velocity is low and the absence of axial loads makes the friction at the planar contact relatively insignificant. The same can be expected for the pin joint contact between the vane and the cylinder where extremely small rubbing velocities renders the friction comparatively negligible, despite having some contact load. The main sources of friction are therefore familiar to most which methods of analysis have been well-established. The following presents the modeling of the existent friction in the RV compressor.

#### 3.1 Friction Loss at Vane Sides

The sliding vane rubs with the vane slot in the rotor component during operation. Significant contact forces arise due to the pressure differential across the vane which results in sliding resistance. With reference to Figure 2, the frictional loss at this region is given by

\[
P_v = \eta \left( |R_1| + |R_2| \right) v_v
\]

where the dynamic friction coefficient, \( \eta \), is assumed a constant value of 0.15 (Yanagisawa et al., 1982). The sliding velocity of the vane, \( v_v \), can be found by the law of cosines to give the following relation:

\[
v_v = \frac{dr}{dt} = \omega_d \frac{d}{d\phi} \left( r_{cy} \left[ \sqrt{1-(1-a)^2 \sin^2 \varphi} - (1-a) \cos \varphi \right] \right)
\]

The forces at the contact points, \( R_1 \) and \( R_2 \), can be obtained by solving the force and moment equilibrium equations on the vane, given by:

\[
R_2 - R_1 = F_{e,v} - F_{s,v} + F_{l,sv} \cos \gamma + F_{k,v}
\]

\[
R_1 \left( \frac{L_v}{2} \right) - R_2 \left( r_v - r_r - \frac{L_v}{2} \right) = \left( F_{e,v} - F_{s,v} \right) \left( \frac{L_v - r_v + r_r}{2} \right) + \left( F_{l,sv} \cos \gamma \right) \left( \frac{L_v}{2} \right)
\]

The reaction force acting on the vane pin joint, \( F_{l,sv} \), occurs mainly due to the inertia of the cylinder as a result of its angular velocity being constantly varying (Figure 4). Neglecting the friction at the cylinder bearings and considering a small offset distance \( (r_{cy} - r_r) \), this force can be approximated by

\[
F_{l,sv} \approx \frac{I_{cy} a_{cy}}{r_{cy}}
\]

where the angular acceleration of the cylinder, \( a_{cy} \), can be formulated by the law of cosines to give
\[
\alpha_{cy} = \frac{d^2 \phi_{cy}}{dt^2} = \alpha_{cy} \frac{d^2}{d\varphi^2} \left[ \cos^{-1} \left( \frac{r_{cy}^2 + (r_{cy} - r_i)^2 - r_i^2}{2r_{cy}(r_{cy} - r_i)} \right) \right]
\]

(6)

Figure 3 shows the variation of the vane side reaction forces and the sliding velocity for one complete working cycle. The contact forces peak in the vicinity of \(\pi\) radians when the most part of the vane is exposed to the large pressure differential across the component. The sliding velocity of the vane follows a sinusoidal profile as expected.

3.2 Friction Loss at Journal Bearings

The friction at the journal bearings is modeled using a simple combination of the long and short bearing approximation method developed by Reason and Narang (1982) which is subsequently modified by Hirani et al. (1997). The friction force is found to have the relation:

\[
F_f = \frac{\mu \omega_b r_b^2 L_b \pi}{\delta_b \sqrt{1 - \varepsilon^2}} \left( \frac{2 + \varepsilon}{1 + \varepsilon} \right) + \frac{\delta_b \varepsilon}{2r_b} \sqrt{F_{blx}^2 + F_{bly}^2} \sin \Phi
\]

(7)

where the journal load components, \(F_{blx}\) and \(F_{bly}\), for the respective rotor and cylinder bearings can be obtained from force balances in the horizontal and vertical directions to give

\[
F_{blx} = (R_1 - R_2) \cos \varphi - F_{ex,x} - F_{ex,x} \quad \text{and} \quad F_{bly} = (R_2 - R_1) \sin \varphi - F_{sy,y} - F_{sy,y}
\]

(8)

\[
F_{bx,xy} = R_{cy} \sin \varphi + F_{1,xy} \cos \varphi_{cy} - F_{sx,xy} - F_{ex,xy} \quad \text{and} \quad F_{by,xy} = R_{cy} \cos \varphi - F_{1,xy} \sin \varphi_{cy} - F_{sy,xy}
\]

(9)

The radial contact force, \(R_{cy}\), can be found by a force balance on the vane in the radial direction:

\[
R_{cy} = F_{cx,xy} - F_{1,xy} \sin \gamma - \eta \left| R_1 \right| + \left| R_2 \right| - m_y \dot{v}_y
\]

(10)

The losses at the rotor and cylinder bearings can be subsequently be calculated respectively from

\[
P_{r} = F_{br,xy} \omega_r r_{br,xy} \quad \text{and} \quad P_{cy} = F_{b_{cy,xy}} \omega_{cy} r_{b_{cy,xy}}
\]

(11)
Figure 4 shows the variation of the journal loads and the angular velocities for one working cycle. The magnitudes of the journal forces at both the rotor and cylinder bearings follow a similar trend as they are geometrically similar and exposed to the same pressure variation. The rotor bearing load is slightly larger than that of the cylinder because of the vane contact forces, $R_1$ and $R_2$, acting on the former. In the right diagram of the same figure, it is shown that for a constant driver velocity, $\omega_d$, the angular velocity of the cylinder is of a sinusoidal profile averaging at the driver velocity.

![Diagram](image1)

Figure 4: (Left) Load magnitudes on journal bearings; (Right) Rotational velocities of rotor and cylinder

### 4. RESULTS AND COMPARISON

Figure 5 shows the variation of the frictional losses at the vane sides and the journal bearings for a complete operating cycle. It is observed that large friction occurs at the vane sides when the sliding velocity is large. The frictional loss at this region takes up a significant portion of the total loss averaging at about 40 W for the current configuration. In the lower diagram of the same figure, it is shown that the friction at the cylinder journal substantially exceeds that of the rotor journal. This is due to the difference in the radiuses of the respective journals as can be inspected from Figure 1. Although the angular velocities are comparable, the frictional loss is proportional to the cubed power of the bearing radius, thus resulting in the large disparity. In order to reduce both the vane side loss and the journal losses, it is proposed that axial length of the compressing mechanism, $L_m$, and the vane length, $L_v$, be maximized. Further details are given in the accompanying paper entitled ‘Geometrical Optimization of the Revolving Vane Compressor’.

![Diagram](image2)

Figure 5: (Upper) Friction loss at vane side contacts; (Lower) Friction loss at journal bearings

![Diagram](image3)

Figure 6: Comparison of total frictional losses in the Revolving Vane and Rolling-Piston compressors

For comparison purposes, the total frictional loss in the RV compressor and that of a Rolling-Piston (RP) compressor with the same capacity employing a similar working fluid are shown in Figure 6. It is observed that for the RV compressor its frictional loss averages at about 86 watts, achieving more than 20 % reduction over the RP
design. It is anticipated that future development will be able to further increase the mechanical efficiency of the RV compressor.

5. CONCLUSION

In this paper, a new design of a compression mechanism has been introduced which employs a simple but radical modification of the existing Rotary Vane compressor. Theoretical studies have shown a higher mechanical efficiency of the design as compared to the popular Rolling-Piston compressor. Considering the compactness of the design and the simple geometries of its components, it is believed that the new compressor has indeed the potential for contention with today’s dominant compressor designs. Further development is currently in progress.

NOMENCLATURE

\( \alpha \) radius ratio, \( r/r_{cy} \)  
\( F_c \) force due to compression pressure  
\( F_{cf} \) centrifugal force  
\( F_i \) inertia force  
\( F_k \) coriolis force  
\( F_s \) force due to suction pressure  
\( I \) second moment of inertia  
\( L \) length  
\( L_m \) axial length of compressing mechanism  
\( m \) mass  
\( P \) friction loss  
\( R \) reaction force  
\( r \) radius  
\( r_v \) radial distance from rotor centre to vane pin joint  
\( v \) velocity

\( a \) angular acceleration  
\( \gamma \) angle between rotor and cylinder angles, \( \varphi - \varphi_{cy} \)  
\( \delta \) radial clearance  
\( \varepsilon \) bearing eccentricity ratio  
\( \eta \) dynamic friction coefficient  
\( \mu \) dynamic viscosity of lubricant  
\( \varphi \) driver/rotor angle  
\( \varphi_{cy} \) cylinder angle  
\( \Phi \) bearing attitude angle  
\( \omega \) angular velocity

Subscripts

\( B \) of bearing  
\( c \) compression  
\( cy \) of cylinder component  
\( d \) of drive shaft  
\( r \) of rotor component  
\( s \) suction  
\( v \) of sliding vane

REFERENCES

Yanagisawa, T. et al., 1982, Motion Analysis of Rolling Piston in Rotary Compressor, Proceedings of the Purdue Compressor Technology Conference, p. 185-192