Wear and Tribodynamics of a Rolling Piston Rotary Compressor

S. K. Padhy
General Electric Company

G. O. Scheldorf
General Electric Company

Follow this and additional works at: http://docs.lib.purdue.edu/icec

http://docs.lib.purdue.edu/icec/960
Wear and Tribodynamics of a Rolling Piston Rotary Compressor

Sisir K. Padhy and Gary O. Scheldorf
General Electric Company, Appliance Park 5-2N, Louisville 40225

ABSTRACT
Rolling piston compressors are widely used in air conditioning and refrigeration equipment. The compressor is of hermetic type and is subjected to complex loading and dynamic motions. The lubricant viscosity varies due to the compression of the gas. These factors crucially govern the wear rate of the vane and the roller and hence the reliability of the compressor. As the compressor is a hermetic one, any contamination due to wear particles and chemical breakdown of the lubricant lead to system failure. The objective of this paper is to study the effect of the dynamic behavior on the elastohydrodynamic lubrication at the roller and the vane interface, to present normalized wear results, material and geometry effects on wear, and to characterize the wear profile with the lubrication conditions.

1. INTRODUCTION
Fujimoto et al. [Fujimoto 84] carried out a tribological analysis of the compressor involving the temperature calculation at the roller and vane interface, and effective lubrication in the cold start and defrosting cycles of the refrigerator. However, no analysis of wear is carried out with respect to the dynamics and material effect. Bhargava [Bhargava 91] in his preliminary work reported that based on the experimental available wear data, the mode is fatigue wear. There is no work reported for the vane side wear. The friction coefficient at vane and roller interface is experimentally determined by Yanagisawa et al. [Yanagisawa 82]. Hess and Soom [Hess 90] also reported an empirical equation for the friction coefficient for hydrodynamic lubrication. However these results are specific to material combination and operating environment. Nakagawa et al. [Nakagawa 90] presented a new oxynitriding plus steam treatment process, that allowed a superb oil retaining property to the surface. This improves the lubricating ability in case of oil break down.

No work is reported on the tribodynamics of rotary considering the effect of dynamics on wear. In this paper a clear role of dynamics is established with reference to wear. The effect of dynamics on oil film thickness at the vane and roller interface is presented.

2. WEAR OF COMPRESSOR
The main components of a rotary compressor mechanism consist of a cylinder, a shaft, an eccentric on the shaft, a roller, two bearing plates and a vane. Among these components, the moving parts are the shaft, the roller and the vane. However, the dynamics of these three parts are extremely complex and coupled. The variation of pressure in the compression chamber, the presence of oil in the chamber, the condition of lubrication at the interface between the roller and vane, the existing interface as well as the bulk temperature at vane and roller interface, the effect of surface roughness, and the vibrations associated with the motions of these parts make the analysis of the wear of the rotary compressor very difficult. The main areas of wear in a rotary compressor are at vane and roller interface, at vane sides and cylinder, and at the journal and thrust bearings. As the journal and thrust bearing wear are somewhat understood, the present paper will focus on the vane and roller interface wear.

In a rotary compressor the vane and roller interface experiences a unsteady sliding and reversal of motion. The motion is asymmetric and the contact point is subjected to load and temperature variation. The relative velocity of the roller with respect to a point on the vane nose changes its direction in each cycle. That means each point on the vane nose is visited twice by the roller in one crank rotation. The vane nose is treated with a TiN coating of 160 μin. The compressor life expectancy is greater than thirteen years which translates to about 80000 hours of operation with a run time of 70%. In 80000 hours, a point on the vane nose experiences $3.3 \times 10^{10}$ cycles with a shaft rpm of 3450. The constraints are that the roller outer diameter wear must be less than 2500 μin. and vane wear be less than 160 μin. Based on these numbers the wear coefficients are calculated to be in the order of $10^5$ for the roller and $10^{10}$ for the vane [Bhargava 91]. Another observation is that the wear modes for the vane may be different than the roller. Hence a proper selection of material is very crucial to the success of the compressor. As the compressor is lubricated, chemical wear, fatigue wear and adhesive wear are the main wear modes present in a rotary compressor. The very low vane wear rate and low wear coefficient suggest that a fatigue based mechanism of wear be predominant for the vane.

79
2.1 Dynamics

In any quantitative wear calculation, a detailed knowledge of the contact stresses, contact geometry, load, and the motion occurring at the contact are required. The dynamics determines the loading variation, the possible type of lubrication, and the type of relative motion exist at interface. The relative velocity between the roller and the vane at the contact point given as follows [Padhy 94]:

\[ u = e\omega \cos(\theta + \alpha) + R_{rol}\omega_p + \dot{x}\sin\alpha \]

where \( R_{rol} \) is the roller radius, \( e \) is the shaft eccentricity, \( \alpha \) is the angle extended by the roller and vane contact point with the X-axis at vane center, \( \theta \) is the crank angle, and \( \dot{x} \) is the vane velocity. The terms in the right hand side of the equation represent sliding, rotational and vane component velocities of the roller respectively. Among these three components the vane component velocity is very small and has little effect on the overall velocity behavior. The sliding component purely depends on geometry. Increasing the eccentricity, the magnitude of the sliding component can be increased. The rotational velocity of the roller is governed by the friction coefficient at vane and roller interface, load, lubricant viscosity between the shaft and the roller, and the friction condition between the roller and the bearing plates. This velocity is very sensitive to the viscosity of the lubricant and the clearance between the roller and the shaft. With a high viscosity oil and low clearance the rotational velocity can be increased significantly. It is found that the friction behavior at the vane and roller interface plays a crucial role in the dynamics of the roller. With higher friction coefficient, the relative velocity decreases. Figure 1 describes different velocities and the total velocity at the roller and vane interface. As the total velocity experiences both positive and negative magnitudes, the velocity goes through zero value. At this instant the lubricant film thickness becomes zero and metallic wear occurs. Increasing the rotational velocity can make the total velocity only positive and thus eliminating the zero velocity condition. Figure 2 presents the velocity variation with operating pressure of the compressor and different thickness of roller. It is found that with a thinner roller (lower inertia of mass) the velocity approaches zero and remains at zero value for a while in one rotation of the crank. However this is delayed to more higher pressure with a thicker roller. This indicates that at high pressures, with a thinner roller the wear can be reduced due to rolling of the roller (zero velocity).

The roller dynamics modes can be sliding with no reversal, rolling, skidding and sliding with reversal. Sliding with no reversal is manifested when the roller moves in one direction with respect to a fixed reference frame. Sliding with reversal occurs when the roller moves both forward and backward. Rolling occurs when there is no relative velocity between the vane and roller. Skidding is present when the same point on the roller contacts the vane nose during the motion of the roller. Depending on the geometry and shaft speed, the roller can experience one or a combination of dynamics modes. The velocity behavior at the interface can be analyzed using a dragster concept (Table 1).

The vane slapping motion in the vane slot is determined from the orientation of the reaction forces between the vane sides and the vane slot. It is found that the vane nose end is in contact with the suction side and the spring end is in contact with the discharge side from the shaft rotation from 22 to 360 degrees [Padhy 92] resulting vane suction side wear. Experiments were carried out with vanes sputter coated with aluminum to produce wear scars in a short time. After 100 hours of run time the vanes were taken out and examined for wear. The experimental results agreed with the theoretical predictions.

![Figure 1. Various components of roller velocity with crank angle](image-url)
2.2 Lubrication

The vane and roller interface experiences boundary lubrication and elastohydrodynamic (EHD) lubrication. The boundary lubrication is probably the primary one. The boundary lubrication is chemical in nature and very complicated to model. It is reasonable to say that the additives in the lubricant are the key factor in eliminating severe wear in boundary lubrication conditions. The lubrication at the vane and roller contact is modeled as elastohydrodynamic lubrication. Although, the vane and roller contact is not purely EHD, the EHD analysis can give an estimate for the minimum oil film thickness at the vane and roller contact. The theory of Dowson and Higginson [Dowson 69] is used for the calculation of the minimum oil film thickness for line contact. Experimental correlations are adapted [Pate 91] to consider the viscosity variation of oil with pressure, temperature and refrigerant concentration in oil.

The EHD lubrication thickness at the vane nose and roller interface is compared against the experimental wear profile at the vane nose. Excellent agreement is found. As the roller moves against the vane nose, it experiences a sliding mode with reversal. At point A (Figure 3) the relative velocity changes from positive to negative and goes through a zero velocity. At this point the thickness of the oil film is zero and wear is adhesive in nature and is severe. After this point there is again oil film thickness exists and wear reduces. After the roller reached the end of the vane nose it traverses back and goes through another reversal at point B resulting severe wear again. After this point the relative velocity is practically zero and a rolling condition is reached. As the wear is minimum at rolling condition, the wear profile shows a reduced wear from point B to point C. At point C the dynamics mode is changed from rolling to sliding and wear is increased again. After that the oil film thickness is maintained and wear is reduced.

It is found that the suction side portion of the vane nose wears more than that of the discharge side. This is attributed to the fact that the suction side of the vane nose experience a poor lubrication. Due to high pressure in the discharge side, there is a clearance between the vane and vane slot and oil is injected from the oil sump to the discharge side. However, as the suction side is in contact with the vane slot, the oil injection is practically zero. This is observed in roller dynamics experiments. Another factor that governs the suction side vane nose wear is the higher loading on suction side than that of the discharge side.

By increasing the vane radius, the oil film thickness at the vane nose decreased slightly. However the contact stress is also decreased. The net result can be confusing. It should be noted that the lubrication state is not clearly understood as it is assumed to be EHD. Hence the contact stress will dominate in determining the wear. This will lead to a reduction of wear. This prediction is supported by the experimental results.

2.3 Interface Temperature

The Interface temperature at vane nose and roller interface plays an crucial role in the boundary lubrication as well as EHD lubrication regime. The oil breakdown is possible if the interface temperature exceeds the critical temperature of the oil. The temperature is calculated employing the scheme as outlined by Rabinowicz [Rabinowicz 65]. The equation is given as follows.
where $T_i$ is the total temperature at asperity, $k_1$ and $k_2$ are thermal conductivity of the mating materials, $F$ is load, $n$ is the number of asperity contact, $H_o$ is the ambient hardness of the softer material, $T_o$ is the sink temperature and $\eta$ is the friction coefficient. In general the number of asperity contact is about 5 [Rabinowicz 92]. The friction coefficient with some lubrication can be about 0.2 for steel and cast iron combinations. Other methods are also used and the results are compared against this result [Padhy 92].

With an ambient temperature of 200 degree Fahrenheit, the asperity temperature can reach in between 350 to 450 degree Fahrenheit. If the compressed gas temperature is added to this temperature, a temperature around 700 to 800 degree Fahrenheit can be attained at the asperity contact and oil breakdown is possible. This is substantiated by the black deposits on the vane nose and discharge valve.

2.4 Material

In this paper, compressor wear for three stages of design are analyzed. They describe from the initial choice of materials to the final choice of materials. The following tables present a minor portion of the data from a large data base of wear that was generated over seven years of extensive testing.

Some results are presented in figures 4, 5 and 6. Figure 4 shows the dramatic effects of material selection on vane and roller wear. For configuration-I and condition-I test (holding the roller dynamics velocity profile constant), a tenfold improvement in roller wear was achieved. The TiN coating on the vane dramatically improved the vane wear.

In figure 5, both the configuration-I and II have the same mode of roller dynamics (sliding with reversal) at condition-II, but configuration-II has a higher peak velocity. As material changed from stage-I to stage-III, the impact on roller wear was more pronounced for configuration-II, even though no change in roller dynamics velocity profile had
Table 2. Material and lubricant at various stages of compressor design

<table>
<thead>
<tr>
<th>Material and Lubricant</th>
<th>Stage-I</th>
<th>Stage-II</th>
<th>Stage-III</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vane</td>
<td>Sintered Ni-alloy steel</td>
<td>M2 tool steel</td>
<td>M2 tool steel with TiN coating at vane nose</td>
</tr>
<tr>
<td>Roller</td>
<td>Sintered Ni-alloy steel</td>
<td>Alloy cast iron</td>
<td>Alloy cast iron</td>
</tr>
<tr>
<td>Lubricant</td>
<td>3GS +ZDDP</td>
<td>AB+PAO+ZDDP</td>
<td>AB+PAO+ZDDP</td>
</tr>
</tbody>
</table>

Table 3. Operating conditions

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Condition-1</th>
<th>Condition-2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Cabinet life test</td>
<td>Compressor only test</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>110 F</td>
<td>90 F</td>
</tr>
<tr>
<td>Suction pressure</td>
<td>2.3 psig</td>
<td>15 psig</td>
</tr>
<tr>
<td>Discharge pressure</td>
<td>185 psig</td>
<td>225 psig</td>
</tr>
<tr>
<td>Compressor temperature</td>
<td>198 F</td>
<td>270 F</td>
</tr>
<tr>
<td>Run-time</td>
<td>90%</td>
<td>100%</td>
</tr>
</tbody>
</table>

Table 4. Configurations

<table>
<thead>
<tr>
<th>Parameters at -10/130 F (ASHRAE standard condition)</th>
<th>Configuration-I</th>
<th>Configuration-II</th>
<th>Configuration-III</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity</td>
<td>800 Btu/h</td>
<td>1200 Btu/h</td>
<td>670 Btu/h</td>
</tr>
<tr>
<td>Displacement</td>
<td>0.32 cu. in.</td>
<td>0.474 cu. in.</td>
<td>0.26 cu. in.</td>
</tr>
<tr>
<td>Energy efficiency ratio (EER)</td>
<td>4.6 Btu/watt-h</td>
<td>4.7 Btu/watt-h</td>
<td>4.4 Btu/watt-h</td>
</tr>
</tbody>
</table>

Figure 4. Vane and roller wear at 8000 hours for different stages

Figure 5. Roller wear at 8000 hours for different configurations

Figure 6. Roller wear at 8000 hours for different conditions
occurred. It was found that the wear was greater for configuration-II in stage-I design and wear greater for configuration-I in the stage-III design.

In figure 6, the effect of dynamics mode on wear is presented. Configuration-I and configuration-III have similar sliding with reversal roller dynamics at condition-I. However, at condition-II, configuration-III changes the dynamics mode to sliding and rolling without any reversal. This new mode is favorable for roller wear reduction in stage-III materials. From condition-I to condition-II the roller wear for configuration-I more than doubles whereas the roller wear for configuration-II increases only slightly.

3. CONCLUSION

The tribodynamics of a rotary compressor is described in this paper. The effect of material, lubrication, dynamics and operating conditions are presented. The wear profile and dynamics correlation is analyzed. From the wear data it is found that the Stage-III design has proven to be very reliable. The wear problem is fixed by proper selection of material, surface coating and lubrication. The proper choice of material is very crucial to the reliability of the compressor and it plays the most important role. Fundamental analysis of dynamics is very helpful in analyzing the effects of various dynamics parameters on reliability. Depending on the dynamics mode (type of motion) the wear can be influenced either adversely or favorably. It is also found that roller dynamics and wear relationships are material dependent. Dynamics can be improved to reduce wear. By proper geometry, oil viscosity and clearances the roller relative velocity can be made only positive without any reversal or can be made zero, i.e. rolling condition. The vane and roller interface lubrication is also governed by the dynamics. A very good agreement is found between the wear profile and EHD lubrication film thickness. Increasing the vane radius resulted an decrease in wear and this is attributed to the rolling condition and decrease in Hertzian stress at interface. Quantification of wear is difficult from dynamics alone, and extensive testing is required in a similar condition to produce the empirical factor for the wear equation.

ACKNOWLEDGMENT

The authors wish to thank Mr. Bert Stocker and Mr. Carl Cordy of General Electric for their help and interest in preparation of this paper. The authors appreciate the permission of General Electric Company to publish this paper.

REFERENCES

[Yanagisawa 82] Yanagisawa, T., Shimizu, T., Chu, I. and Ishijima, K., Motion Analysis of Rolling Piston Rotary Compressor, Proc. of International Compressor Engineering Conference, pp. 185-192, Purdue University,West Lafayette, IN, 1982.