Lubrication Limits of Rolling Element Bearings in Refrigeration Compressors

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LUBRICATION LIMITS OF ROLLING ELEMENT BEARINGS IN REFRIGERATION COMPRESSORS

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ABSTRACT

A bearing test apparatus has been designed to test axial and radial rolling element bearings under conditions similar to those to be found in full-scale refrigeration compressors.

Four angular contact ball bearings have been tested using two different lubricants. Three tests were made to study the influence of refrigerant concentration. The fourth test aimed at comparing the influence of lubricant structure on bearing wear rate. SEM-images of the tested bearings showed that the contact surface between the ball and race was chemically attacked in bearings operating with a refrigerant dilution of 23% R-134a. The lubricant that was formulated with shorter acids gave the highest wear rate.

A new toroidal roller bearing, CARE™, was tested with and without a cage separating the rollers in an ammonia-mineral oil environment. It was found that it can be used in ammonia compressors if it is fitted with a cage separating the rollers.

INTRODUCTION

The rapid change from chlorinated to non chlorinated and natural refrigerants has put great demands on refrigeration compressor companies to develop new equipment designed for these environmentally friendly refrigerants. In recent years, lubrication of bearings in compressors using these alternative refrigerants has been a major issue. For example, the combination of a HFC refrigerant and a polyolester lubricant gives the lubricant very poor anti wear properties. The oxide layer normally present in a rolling element bearing is not maintained and the bearing will experience high wear rates if the lubricant film that separates the surfaces is too thin.

Komatsuzaki and Homma [1] used a four ball tester to evaluate the EP properties of pure refrigerants and oil refrigerant mixtures. They found that R-134a in Polypropylene glycol gave higher rates of wear than R-12 and R-22 in mineral oil. R-134a in polypropylene glycol gave only slightly lower wear rates than the samples tested in air. Mizuhara et. al. [2] studied the anti wear properties of R-12, R-22 and R-134a in mineral oil and PAG and showed that chloride and fluoride were present on the surface after wear tests. Randles and Heavers [3] carried out a systematic study on how the structure of ester lubricants influences their lubricating properties. They found that the wear rate in Falex and four-ball tests increased with increased content of branched acids.

The most important property of a lubricant is to form an oil film that separates the surfaces and prevents metal to metal contact. The expected life of a ball bearing is largely related to the film thickness in the bearing. A thin film will increase the number and severity of asperity contacts between the race and the roller or ball. Contact between the surfaces will cause locally high stresses, increasing the risk of fatigue in the bearing material. Asperity contacts can also lead to wear of the bearing surfaces. New methods for bearing life calculations developed by Ioannides et al [4] consider these effects. Figure 1 shows the relationship between the life adjustment factor, \( a_{\text{def}} \), and the viscosity ratio, \( \kappa \). The curve describes a complex relation between the bearing fatigue limit, \( P_m \), the load, \( P \) and the contamination level, \( \eta_c \). The solid curve is valid for the contamination-level/load ratio, \( \eta_c P_u / P = 0.1 \) at normal operating conditions.
Bearings operating under normal conditions  
Bearing lubricated with POE/R-134a mixtures

**Figure 1** Ball bearing life adjustment factor vs. lubricant viscosity ratio. From [4].

The wear behavior of rolling element bearings has also been found to depend upon the oil-refrigerant combination used. Jacobson [5, 6, 7] tested angular contact ball bearings lubricated with 20 and 30% R-134a in an ISO VG 68 polyol ester at 2000 and 6000 rpm and found that only the bearing operating with 20% dilution at 6000 rpm gave an acceptable wear rate. By measuring the contact resistance between the inner and the outer race, he also found that wear of the bearing could occur even if there was no electrical contact between the rings. Based on this work, he suggested that the decrease in anti-wear properties of polyol ester/R-134a mixtures could be compensated for by multiplying the minimum required viscosity, \( v_1 \), by a factor depending on the oil-refrigerant combination used. Preliminary results by SKF [11] indicates that Jacobson's correction factor should be around three for bearings lubricated with polyol ester/R-134a mixtures. The impact of this on the bearing life is illustrated with the dotted line in Figure 1.

SKF [8] suggest that the viscosity used in calculations of bearing life in refrigeration compressors should be adjusted to compensate for the change in pressure-viscosity coefficient compared to mineral oil. The adjusted viscosity, \( v_{adj} \), for use in the evaluation of bearing lubrication can be determined as:

\[
v_{adj} = v (\alpha/\alpha_{mineral})^{0.72}
\]

Where: \( v \) is the actual kinematic viscosity, \( \alpha \) is the pressure-viscosity coefficient of the oil-refrigerant mixture and \( \alpha_{mineral} \) is the pressure-viscosity coefficient of the mineral oil. The pressure-viscosity coefficient for polyol ester lubricant/R-134a mixtures can be determined using data from Jonsson and Lilje [9].

The new toroidal roller bearing, CARB™, invented by Kellström [10], is suggested for use as a radial bearing in refrigeration compressors [11]. The self-guiding property of the rollers in this bearing type makes it very attractive for use without a cage guiding the rollers to obtain the highest load carrying capacity possible. The high sliding velocity between the rollers in full complement bearing increases the risk of high wear rate in refrigerant atmosphere. Previous field experience with ammonia compressors fitted with taper roller bearings motivates the need for lab tests to be performed.

This paper presents a study of the influence of refrigerant and lubricant structure on the wear of rolling element bearings in refrigeration compressors. A new test rig has been developed that can test radially or axially loaded bearings in a refrigerant atmosphere. Results from four tested angular contact bearings and two toroidal roller bearings are presented.

**EXPERIMENTAL METHOD**

The test apparatus used consisted of a high-speed spindle with an integrated electrical motor as shown in Figure 2. The rig was designed to operate with a single radially loaded bearing or with a pair of axially loaded bearings. Axial load was generated by twelve helical springs mounted between the test bearings. The maximum axial load was 11.5 kN. Radial loads were generated by a hydraulic cylinder and a link rod acting on the test bearing housing. The maximum radial load was 10 kN.
The end cap was mounted on spacers that were slightly longer than the test bearing housing. The test bearing housing was sealed with axially compressed O-rings. This design allowed small radial movements of the housing so that a radial load can be applied to the bearing. The shaft was sealed from the spindle using a mechanical face seal. The speed of the motor could be controlled between 1500 and 12000 rpm using a variable frequency drive.

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Figure 2  Design of the bearing test apparatus

A lip seal is fitted between the seal housing and the test bearing prevented heated oil from the seal cavity influencing the running condition for the test bearing. The test rig was connected to a sealed circulation system. The oil was filtered through a 3μm, β_{ratio}=75, glass fiber filter which could be bypassed if wear particles were to be analyzed during the test.

Lubricants and Bearings

The mineral oil used was an ISO VG 68 solvent refined naphthenic mineral oil. The polyolester lubricants was mixed acid ISO VG 68 pentaerythritol esters. Ester-A was a technical grade pentaerythritol formulated with approximately 70% branched C9 and the remainder consisting of C7 to C10 linear acids. Ester-B was a pentaerythritol formulated with approximately 70% branched C9 and the remainder consisting of mixed acids ranging from C5 to C7. Ester-A has a characteristic miscibility temperature of -7°C. Ester-B is miscible down to -40°C.

The angular contact test bearings were SKF 7210 BEP and were used as received. The toroidal roller bearings were delivered off the shelf as full complement SKF CARB™ C 2210 V bearings. One of the bearings was disassembled and fitted with a laser-cut prototype cage made from unfilled polyamide PA6G barstock. Serial production caged C 2210 were not available at the time of the tests.

Test Procedure

The system was filled with the test lubricant and evacuated before adding the refrigerant. The lubricant-refrigerant mixture was circulated for at least 24 hours before the starting the spindle in order to trap any particle contamination in the filter. All tests were performed with a bearing oil inlet temperature of 40±0.5°C. Temperature variation during the tests was less than ±0.5°C. A summary of the test conditions is given in Table 1 and Table 2

<table>
<thead>
<tr>
<th>Test</th>
<th>Lubr.</th>
<th>Conc., %</th>
<th>Viscosity, v, cSt</th>
<th>Press.-Visc. coeff., α, GPa(^{-1})</th>
<th>Visc. ratio, k</th>
<th>Test time, h</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ester A</td>
<td>14.7-31.9</td>
<td>16.8-4.5</td>
<td>17.7-15.2</td>
<td>1.57-0.38</td>
<td>240</td>
</tr>
<tr>
<td>2</td>
<td>Ester A</td>
<td>17.3-15.6</td>
<td>13.3-15.5</td>
<td>17.6-17.2</td>
<td>1.44-1.22</td>
<td>100</td>
</tr>
<tr>
<td>3</td>
<td>Ester A</td>
<td>22.8-22.4</td>
<td>8.4-8.7</td>
<td>16.4-16.3</td>
<td>0.77-0.74</td>
<td>100</td>
</tr>
<tr>
<td>4</td>
<td>Ester B</td>
<td>23.0-22.4</td>
<td>8.1-8.5</td>
<td>16.8-16.7</td>
<td>0.76-0.72</td>
<td>100</td>
</tr>
</tbody>
</table>

The concentration range for tests 1-4 represents the concentration measured at start-up and before stopping the rig. The reduction of refrigerant concentration during the test was caused by samples being removed from the system and also leakage.

The viscosity, v, the pressure-viscosity coefficient, α, and the viscosity ratio, k=\(v_{ad}/v_1\), are calculated as follows:

The minimum required viscosity, \(v_1\), for a 7210 bearing operating at 3000 rpm is 9.8 cSt [12]. The pressure-viscosity coefficient for mineral oil, α_{mineral}, was assumed to be 20 GPa\(^{-1}\). The viscosity and pressure-viscosity coefficient of Ester A was taken from the raw data used in Ref. [9]. The density was estimated using data from

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The kinematic viscosity for Ester B was obtained from the manufacturer. The $\alpha$-value for pure Ester B was estimated as 22 GPa$^{-1}$ based on the viscosity at 40°C and an effective amount of branched acids of 85% as input to the diagram developed by Jonsson and Lilje [9]. The $\alpha$-value at 23% dilution with R-134a was estimated as 17 GPa$^{-1}$. The adjusted viscosity, $\nu_{adj}$, was calculated using Eq. 1.

Table 2  Test conditions for CARB™ C 2210 in mineral oil/ammonia.

<table>
<thead>
<tr>
<th>Test</th>
<th>Bearing type</th>
<th>Speed, rpm</th>
<th>Pressure, MPa</th>
<th>Duration, h</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>C 2210 V</td>
<td>3000</td>
<td>0.3-0.7</td>
<td>192</td>
</tr>
<tr>
<td>6</td>
<td>C 2210</td>
<td>3000-12000</td>
<td>0.3-0.8</td>
<td>165</td>
</tr>
</tbody>
</table>

The pressure range represents the gas pressure in the system measured at the tank.

Test 1 was used to establish the concentration at which accelerated wear began to take place in angular contact bearings. The refrigerant concentration was increased in daily steps shown in Figure 3. Oil samples were taken every 24 hours. The results from this test were then used to determine the running conditions for Tests 2-4. The filter was bypassed during the test so that particles generated by the bearing were not trapped in the filter.

Tests 2 and 3 were run under constant conditions with the aim of verifying the results from Test 1. Test 4 aimed at investigating the influence of lubricant structure on bearing wear rate.

Test 5 was carried out in order to investigate whether full complement CARB™ bearings could be used in ammonia compressors.

Test 6 investigated the performance of a CARB™ bearings operating in an ammonia environment at high speed. The initial speed was 3000 rpm and the refrigerant pressure increased from 0.1MPa to 0.8MPa during the first 40 hours. The speed was then increased in steps so that the last 24 hours were run at 12000 rpm.

The axial load for the angular contact bearings was 11.5 kN for all tests. This equals a load rating/load ratio, $C/P$, of 5.95 giving an ISO rating life, $L_{10}$, of 1170 hours. The radial load for the toroidal roller bearings was 4.6 kN giving a $C/P$ ratio of 21.3 for the full complement bearing and 18.4 for the caged bearing. The radial load was limited by the support bearings in the test apparatus.

RESULTS AND DISCUSSION

Angular Contact Bearings Lubricated with Polyolester/R-134a Mixtures

The diagram and microscope images in Figure 3 show the results from Test 1. The graph shows refrigerant concentration and the metal content in the lubricant vs. time. The result indicates that wear started to occur when the refrigerant concentration was increased above 18%. The test was continued to a concentration level of 32%.

![Graph showing refrigerant concentration and metal content vs. time](image)

Figure 3  Iron and refrigerant content in lubricant vs. time and optical microscope images of the bearing race from Test 1.

Inspection of the bearing after the test showed that the bearing surface was damaged, the ball track having a frosted appearance on both sides of a distinctive bright polished band along the race as shown in Figure 3. It appears from this that wear of the race is strongly dependent on the sliding speed and contact pressure. The balls also had identical bands around the great circle. This is typical for bearings operating at constant load as described by Tallian [14]. This suggests that testing at constant load as done in this study and in the work by Jacobson [5,6] might increase the observed wear rate. The varying load in twin screw compressors is probably beneficial by keeping the bearing wear more uniform thus maintaining the internal geometry of the bearing and avoiding a change in the Hertzian load.
distribution. Since the damage of the bearing is severe it can be concluded that this bearing would suffer a premature failure.

Figure 4 shows SEM-images of the bearings from Test 2-4. The bearing operated in Ester A with 17% refrigerant dilution showed no significant damage. The grinding marks are clearly visible. Higher magnification revealed oriented kinematic wear marks which were smoother than those found in normal applications. The authors suggest that this is due to tribo-chemical interaction between the oil-refrigerant pair and the bearing steel. Based only of the appearance of the surface it is difficult to predict the influence this could have on the life of the bearing.

The bearing operated in Ester A with 23% refrigerant dilution showed similar, but more extensive, damage to that found in the bearing operated in 17% refrigerant. The grinding marks from the manufacturing process were still clearly visible but the surface was covered with shallow dents oriented at a specific angle indicating that they originate from the ball-race interaction. It is difficult to draw any conclusions as to how this damage would affect bearing life. It can, however, be concluded that the life will be shorter than for the bearing operated at 17% dilution and longer than for the bearing operated at 23% dilution in Ester B.

The bearing operated in Ester B with 23% refrigerant dilution showed a different texture. The grinding marks from the manufacturing process are almost totally erased and the surface is covered with shallow pits oriented along the track. The dark area on the upper picture next to the scale bar shows deeper pits than the typical ones shown in the lower picture. The surface features on the race cannot be explained by mechanical wear only. Since all grinding marks are worn away and the surface texture show signs of deep holes, it can be concluded that a bearing operating under these conditions would fail prematurely. The authors believe that the difference between the results from the tests with Ester A and Ester B can be explained by the corrosive nature of the short acids present in Ester B.

A graphical illustration of these results is included in Figure 1. The dots represents the operating viscosity ratio and an estimated influence on bearing life based on the appearance of the bearing surface and the amount of metal worn away during each test. Thees results indicate that it might be necessary to adjust the Jacobson number depending on the oil type. Further work is necessary to fully understand this process and the variables controlling it.

Torodial Roller Bearing in Ammonia/Mineral Oil

The total test time for the full complement CARB™ was 192 hours. The test was ended since the iron content in the lubricant was increasing steadily. The ICP-AES analysis of the lubricant indicated an iron content of 11 ppm when the test was ended. The ICP-AES is only capable of detecting particles smaller than 3μm so it is likely that the actual iron level was much higher. Post test inspection of the bearing showed that the center portion of the rollers was worn down to a cylindrical shape with a width of 4 mm as shown in the left hand picture in Figure 5. The test confirms that high internal sliding speeds must be avoided in ammonia environments even at low load such as the roller-roller contact in a torodial roller bearing.
Figure 5 Roller from CARB™ bearings used with and without cage in an ammonia atmosphere

The CARB™ C 2210 fitted with PA-6 cage was tested for 165 hours. The test was ended after the bearing had operated at 12000 rpm for about 24 hours. The test time at high speed was limited due to concern over the integrity of the non-reinforced prototype cage. This speed corresponds to a characteristic speed, \( n_d \), of 840 000 mm/min.

No significant wear marks could be seen on the rollers or on the races. The loaded zone on the outer race was barely noticeable viewed through a stereo microscope. No wear could be observed on the rollers or at the inner ring. The left hand picture in Figure 5 shows the roller after the test.

CONCLUSIONS

- The wear of the races in a rolling element bearing is to a large extent controlled by chemical reactions between the oil-refrigerant mixture and the bearing surfaces initiated by friction heat. The lubricant structure seems to have a significant impact on the wear rate. The different wear rate between Ester A and Ester B suggest that the lack of long linear acids or the presence of short acids in the lubricant plays a key role for the wear-rate.
- The results suggests that the Jacobson number might need to be adjusted dependent on the anti-wear properties of the lubricant.
- Caged versions of the CARB™ bearings can be used in ammonia compressors operating at very high speeds.

REFERENCES

[7] Private communication, Professor Bo Jacobson, Lund, Sweden. Formerly employed by SKF-ERC, Holland