Elastohydrodynamic Lubrication Properties of Polyol Ester Lubricants-R134a Mixtures

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ELASTOHYDRODYNAMIC LUBRICATION PROPERTIES OF POLYOL ESTER LUBRICANTS-R134a MIXTURES

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ABSTRACT

A modern screw compressor contains several highly loaded metal-to-metal contacts lubricated with oil, including, rolling element bearings, gears, and the rotors. These types of contacts are normally referred as EHD or EHL contacts (Elasto-Hydro-Dynamic-Lubrication), due to the elastic deformation of the surfaces. The expected life of such bearings is to a large degree related to the lubricant film thickness in the bearings. The properties of the lubricant that determine the film thickness in an EHD-contact are normally the dynamic viscosity, \( \eta \), and the pressure-viscosity coefficient, \( \alpha \). In a refrigeration system, both those parameters are affected by the presence of refrigerant.

A falling ball viscometer has been designed that can be pressurized up to 34MPa in order to measure the viscosity and pressure-viscosity of oil-refrigerant mixtures. Eight different polyolesters mixed with R-134a were evaluated and the data presented. The results show that the degree of branching has a large impact on the pressure viscosity coefficient.

An empirical model has been developed that can be used to predict the pressure-viscosity coefficient based on viscosity and amount of branched acids in the lubricant.

INTRODUCTION

A modern screw compressor contains several highly loaded metal-to-metal contacts including rolling element bearings, gears and the rotors. These types of contacts are normally referred to as Elasto-Hydro-Dynamic- (EHD) contacts, due to the elastic deformation of the surfaces. The maximum pressure in such contacts can exceed 4 GPa.

The most important task for a lubricant in these compressors is the formation of a film that separates the interacting surfaces. The expected fatigue life of a ball bearing is to a large degree related to the film thickness in the bearing due to the risk of asperity contact. This is very relevant in a refrigeration compressor since the working viscosity is often very low due to dilution of refrigerant. Contact between the surfaces will cause locally high stresses resulting in fatigue of the bearing material. Asperity contact may also result in wear of the bearing surfaces. The rate of this wear is found very dependent on the type of oil and refrigerant used. Jacobson [1] has tested angular contact bearings lubricated with oil-refrigerant mixtures in a bearing cavity simulator. He found that prevention of wear of the bearing surfaces typically requires 50% higher viscosity when lubricated with HFC-134a/polyol ester mixture than for HCFC-22/mineral oil lubricated bearings.

The properties of the oil that determine the film thickness in EHD-contact are normally the dynamic viscosity, \( \eta \), and the pressure-viscosity coefficient, \( \alpha \). In a refrigeration system, both those parameters are affected by the presence of refrigerant. Speaker and Spauschus [2] present reviews of various methods of measuring viscosity of oil-refrigerant mixtures. Jonsson and Högłund [3] used a concentric cylinder viscometer to measure viscosity and pressure-viscosity coefficient of oil refrigerant mixtures.
A method for determining the viscosity of oil refrigerant mixtures by means of film thickness interferometry measurements has been used by Jonsson and Höglund [4]. This method was later developed further by Akei et al [5, 6] to estimate the viscosity and the pressure viscosity coefficient of oil refrigerant mixtures, including a polyolester with R-134a. They used the Eyring equation, Eq.1, to estimate pressure-viscosity coefficient of oil refrigerant mixtures.

\[
\alpha_{\text{mix}} = \frac{s_{\text{refr}} \left( \alpha_{\text{refr}} - \alpha_{\text{lubr}} \right)}{s_{\text{refr}} (m-1) + 1} + \alpha_{\text{lubr}} \quad (1)
\]

In Eq. 1, \( \alpha_{\text{mix}} \) is the pressure-viscosity coefficient of the mixture, \( \alpha_{\text{refr}} \) and \( \alpha_{\text{lubr}} \) are the pressure viscosity coefficient for pure refrigerant and lubricant respectively. \( s_{\text{refr}} \) is the mass fraction of refrigerant and \( m \) is the molecular weight ratio of the lubricant and refrigerant. Akei and Mizuhara used data for the pressure-viscosity coefficient for R12 measured by Babb and Scot [7] to fit their data.

SKF [8] and Jacobson [9] suggest that the viscosity used in calculations of bearing life is modified to compensate for the reduction in pressure-viscosity coefficient compared to mineral oil. The adjusted viscosity, \( \nu_{\text{adj}} \), for use in the evaluation of bearing lubrication can be determined as follows: \( \nu_{\text{adj}} = \nu_{\text{mix}} \left( \frac{\alpha_{\text{mineral}}}{\alpha_{\text{mix}}} \right)^{0.72} \) where: \( \nu_{\text{mix}} \) and \( \nu_{\text{adj}} \) are the actual kinematic viscosity and pressure-viscosity coefficient of the oil-refrigerant mixture at the location of the bearing and \( \alpha_{\text{mineral}} \) is the pressure-viscosity coefficient of mineral oil that is used as a reference.

Today, the oil manufacturers normally supply pressure-temperature-concentration data and viscosity-temperature concentration data for their products. This type of data is also available in the literature, however, detailed information on the composition of the lubricants is usually not disclosed. Pressure-viscosity coefficient data is seldom available due to the complexity of the measurements required to obtain the data. It is important that pressure viscosity coefficients of lubricant/refrigerant pairs be available for new methods of bearing life calculation. This work aims at making such data available in a format that can be used in the refrigeration industry.

**TESTED LUBRICANTS**

Eight different lubricants were evaluated with R-134a. The lubricants were all refrigeration grade basestocks containing less than 100 ppm water. They were selected to give a broad representation of commercial polyolesters for industrial refrigeration applications. Polyolester properties can vary significantly, depending on the structure of the acids used in their preparation. In refrigeration application, branched acids impact properties dependent on interaction between refrigerant and the lubricant such as solubility, miscibility and viscosity-concentration dependency. To better understand this phenomenon, esters with varying degrees of branching were examined. Table 1 lists the lubricants used in this study and outlines their essential properties and structural features. The last two columns lists the lowest temperature a 10 respectively 20% oil-refrigerant mixture will form a single phase.

<table>
<thead>
<tr>
<th>Name</th>
<th>Polyol</th>
<th>% Branched acids</th>
<th>Kinematic viscosity, cSt</th>
<th>Viscosity index</th>
<th>10% misc. °C</th>
<th>20% misc. °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ester-A</td>
<td>Tech-PE</td>
<td>0</td>
<td>31</td>
<td>140</td>
<td>ND</td>
<td>ND</td>
</tr>
<tr>
<td>Ester-B</td>
<td>Mono-PE</td>
<td>28</td>
<td>30</td>
<td>129</td>
<td>ND</td>
<td>ND</td>
</tr>
<tr>
<td>Ester-C</td>
<td>Mono-PE</td>
<td>67</td>
<td>32</td>
<td>102</td>
<td>&lt;-60</td>
<td>&lt;-60</td>
</tr>
<tr>
<td>Ester-D</td>
<td>TMP</td>
<td>70</td>
<td>31</td>
<td>116</td>
<td>-40</td>
<td>-30</td>
</tr>
<tr>
<td>Ester-E</td>
<td>TMP</td>
<td>100</td>
<td>46</td>
<td>94</td>
<td>-48</td>
<td>-28</td>
</tr>
<tr>
<td>Ester-F</td>
<td>Tech-PE</td>
<td>70</td>
<td>65</td>
<td>114</td>
<td>-7</td>
<td>5</td>
</tr>
<tr>
<td>Ester-G</td>
<td>Tech-PE</td>
<td>85</td>
<td>82</td>
<td>97</td>
<td>-25</td>
<td>-20</td>
</tr>
<tr>
<td>Ester-H</td>
<td>Di-PE</td>
<td>100</td>
<td>385</td>
<td>105</td>
<td>20</td>
<td>40</td>
</tr>
</tbody>
</table>

The refrigerant used in this work (R-134a, 1,1,1,3-tetrafluorethane) was obtained from a commercial source and was used as received.
EXPERIMENTAL METHOD

The viscosity was measured with a falling ball viscometer at 40 and 80°C at pressures of 2, 17 and 34 MPa for each concentration. A few tests were made at 60°C to confirm the viscosity-temperature relation as published in ASTM D341. The concentration of dissolved refrigerant was varied between 0 and 40% in steps of approximately 10% to cover the range found in most applications.

The viscometer used was a modified Hoppler viscometer. The working principle is shown in Figure 1.

![Schematic diagram of the falling ball viscometer](image)

Figure 1 Schematic diagram of the falling ball viscometer

The viscosity was determined by the time it took the ball to travel a defined distance in a tube containing the test liquid. Two pairs of sapphire windows were used to time the ball. The separating piston is used to pressurize the test fluid during measurements. The test fluid and the pressurizing fluid are separated by two O-rings on the piston. The volume between the O-rings is ventilated through the piston rod to prevent contamination of the test fluid in case of a leakage at the O-rings. The viscometer was always pressurized above the vapor pressure of the test fluid. A more detailed description of the viscometer can be found in Ref. [10]. Concentration of refrigerant was determined by taking a sample through a capillary tube. The sample valve is an angular pattern valve with no dead volumes ensuring a well-defined sample. A spreadsheet program was used to calculate the viscosity for each test point. The program compensates for effects of pressure and temperature on the dimensions of the ball and the bore. Eq (1) is used to fit experimental data of pressure-viscosity coefficient vs concentration of refrigerant.

RESULTS AND DISCUSSION

Figure 2 shows the variation of dynamic viscosity, \( \eta \), with refrigerant concentration at 40 and 80°C. The data show that increased branching results in a more rapid decrease in mixed viscosity at a given refrigerant concentration. This particularly stands out in lubricants that contain 100% branched acids (Esters E and H), but can also be seen in the other combinations. Increased branching will also result in a higher solubility resulting in a further decreased viscosity for a fixed temperature and pressure.

Figure 3 shows the pressure viscosity coefficient variation with refrigerant concentration. Several generalities can be seen in this data. Within the ISO VG 32 lubricants (esters A-D), the \( \alpha \)-value shows a strong correlation to % branching, with higher branching resulting in higher \( \alpha \)-values. The undiluted \( \alpha \)-values for esters E and H are the extreme examples of this. Ester E has a higher \( \alpha \)-value than esters F and G despite significantly lower viscosity, suggesting that in some cases branching afford extremely high \( \alpha \)-values. The data also suggests that the reduction in \( \alpha \)-value with increased refrigerant concentration is larger as branching increases in the lubricant structure. This is particularly apparent in esters which are 100% branched (esters E and H). This data suggest that caution should be exercised when using undiluted pressure-viscosity coefficients to predict EHD lubrication-properties of lubricant/refrigerant pairs.
Figure 2  Viscosity at 40 and 80°C of the tested lubricants vs concentration

Figure 3  Pressure-viscosity coefficient at 40 and 80°C vs concentration

The data shown in Figure 3 is fitted with Eq. 1 in two steps. The first step was used to estimate an effective pressure-viscosity coefficient of pure R-134a, \( \alpha_{\text{efr}} \). The resulting effective refrigerant pressure-viscosity coefficient was 11 GPa\(^{-1} \). This was then inserted into Eq (1) to obtain a two-parameter model. The resulting coefficients are listed in Table 2.

Table 2  Results from curve fit

<table>
<thead>
<tr>
<th></th>
<th>POE-A</th>
<th>POE-B</th>
<th>POE-C</th>
<th>POE-D</th>
<th>POE-E</th>
<th>POE-F</th>
<th>POE-G</th>
<th>POE-H</th>
</tr>
</thead>
<tbody>
<tr>
<td>40°C ( \alpha_{\text{efr}} )</td>
<td>14.23</td>
<td>16.03</td>
<td>17.63</td>
<td>17.91</td>
<td>24.3</td>
<td>21.51</td>
<td>23.54</td>
<td>30.16</td>
</tr>
<tr>
<td>m</td>
<td>1.42</td>
<td>2.31</td>
<td>3.05</td>
<td>3.03</td>
<td>3.07</td>
<td>3.27</td>
<td>2.73</td>
<td>3.67</td>
</tr>
<tr>
<td>80°C ( \alpha_{\text{efr}} )</td>
<td>12.05</td>
<td>--</td>
<td>13.71</td>
<td>14.5</td>
<td>18.53</td>
<td>17.34</td>
<td>17.86</td>
<td>21.88</td>
</tr>
<tr>
<td>m</td>
<td>0.74</td>
<td>--</td>
<td>0.67</td>
<td>2.23</td>
<td>2.01</td>
<td>2.15</td>
<td>2.01</td>
<td>2.09</td>
</tr>
</tbody>
</table>

Figure 4 illustrates how the pressure-viscosity coefficient varies with temperature for pure lubricants and for lubricant refrigerant mixtures. The diagram shows that the pressure-viscosity coefficient-temperature relationship can be approximated with a linear model.
Figure 4  
Pressure-viscosity coefficient for different lubricant-refrigerant mixtures at 40, 60 and 80°C

LUBRICANT STRUCTURE-PRESSURE VISCOSITY COEFFICIENT MODEL

Figure 3 shows the strong influence that the viscosity and the amount of branched acid has on the pressure-viscosity coefficients of the lubricants. The following approximation, Eq. 2, can be used to estimate the pressure viscosity coefficient of polyolester refrigeration oils at operating temperature. The equation under estimated the α-value of Ester A at 80°C with 17% but for all the typical refrigeration oils included in this work, the estimated α-value is within ±8% of the measured value over the whole temperature range.

\[ \alpha_{\text{pure}} = A + B \cdot \ln T + C \cdot X \]  

(2)

In this model, \( \alpha_{\text{pure}} \) is the predicted α-value for the pure lubricant, \( \eta \) is the undiluted dynamic viscosity and \( X \) is the amount of branched acids in the lubricant structure in percent. A, B and C are constants determined to be 2.611, 3.391 and 0.065, respectively. The reduction in \( \alpha \) with refrigerant dilution can then be calculated using Eq (1) and the constants from Table 2. Temperature corrections can also be made using the graph in Figure 4. Figure 5 provides a graphical representation of the calculation. The undiluted α-value is estimated by straight line extrapolation from the viscosity and the % branching. The diluted α-value is then estimated by following the nearest curved line to the desired dilution and reading back to the y-axis.

Figure 5  
Plot for estimation of diluted pressure-viscosity coefficient at 40°C

Temperature corrections can then be made using the graph in Figure 4. The graph can be used to directly estimate the undiluted α-value from 40 to 80°C. The graph shall not be used to adjust α-values for dilution at other temperatures than 40°C since the α-value vs concentration curves are generated using data for 40°C. The graph is generated using the dynamic viscosity. The error introduced by using it with kinematic viscosity as input can be neglected from a practical point of view.
CONCLUSIONS

The pressure viscosity coefficient and the viscosity of various lubricant/R-134a pairs were investigated using a high-pressure falling ball viscometer. The study shows that the lubricant structure has a strong influence on interactions with the refrigerant. Esters with mainly branched acids show the highest pressure viscosity coefficient. The data also suggests that the reduction in pressure viscosity coefficient with increased refrigerant concentration is larger as branching increases in the lubricant structure. Care should be exercised when using undiluted pressure-viscosity coefficients to predict EHL-properties of lubricant/refrigerant pairs. A general model was developed to predict pressure viscosity coefficients of polyol ester/R-134a pairs based on initial viscosity and percentage branching. The effective pressure-viscosity coefficient for pure R-134a was estimated to $11 \text{ GPa}^{-1}$.

A graphical method was developed that allows pressure-viscosity coefficients of polyolester/R-134a mixtures to be determined.

REFERENCES

[9] Private communication, Professor Bo Jacobson, Lund, Sweden