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ABSTRACT

Rolling element bearings are used extensively in air, gas, and refrigerant compressor applications because of their low friction, long life, and high reliability. Compressors having rolling bearings can have higher efficiency and reduced lubrication requirements compared to those using hydrodynamic bearings. The New Life Method for the selection and evaluation of rolling bearings has been developed and successfully applied to compressor applications. The compressor lubrication and cleanliness conditions, and the bearing fatigue load limit are considered by the New Life Method. The methodology is proposed for inclusion in an updated ISO Standard. The work of Jacobson to consider the influence of the refrigerant / lubricant mixture on bearings life is applied with the New Life Method. This paper describes the application of the New Life Method for the selection of rolling bearings in compressor applications.

INTRODUCTION

Rolling bearings are used to support the rotating components of air, gas, and refrigerant compressors of all types. They have successfully replaced hydrodynamic bearings in compressors on account of their lower friction and their ability to operate with reduced oil lubrication. Rolling bearings are selected for compressors on the basis of providing adequate service life, satisfactory positioning and running accuracy of the rotating components, low friction, stiffness, and minimum deflection.

A rolling bearing is comprised of inner and outer rings, rolling elements and cage. The rolling elements can be annular (balls) or rollers. In a compressor, the inner ring is mounted on the rotor(s) and the outer ring is fitted into the housing. The radial, axial, or combined loads are supported by the bearings. The rotation establishes elastohydrodynamic (EHD) film lubrication in the Hertzian contact of the rolling elements on the raceways. The high pressures in the Hertzian contact causes the bulk viscosity of the lubricant to increase significantly. This increase in viscosity with pressure, in a well lubricated situation, prevents direct contact between the components, all the while elastic deformation of the components takes place. The lubricant increase in viscosity with pressure is characterized by the pressure-viscosity coefficient, $\alpha$. Each passage of the rolling element into the load zone of the bearing results in cyclic stresses in the materials. In a well lubricated bearing the highest stresses are subsurface. As the EHD film thickness decreases the highest stresses approach the rolling surfaces. The bearing rings and rolling elements are made of high strength bearing steel, having ultra high cleanliness, heat treated for high fatigue strength. High cyclic stresses can cause material fatigue. Surface damage from the manufacturing process, damage due to insufficient EHD
lubrication, and damage caused by the rolling over of solid particle contaminants cause surface stress concentrations that can result in fatigue.

The estimate of life in rolling bearings is based on the prediction of subsurface and surface initiated metal fatigue of the bearing components. Lundberg and Palmgren[1] established that the life of a rolling bearing could be reliably predicted given the applied load, rotational speed, and knowledge of the bearing internal geometry. Using Weibull analysis they combined the Hertzian contact parameters and the bearing internal geometry parameters to determine the bearing load carrying capacity and a simplified method for determining the bearing rating life, \( L_{10b} \).

\[
L_{10b} = [\frac{C}{P}]^p \frac{16667}{n} \quad \text{Eq. 1}
\]

Where

- \( L_{10b} \) = basic rating life ( 90 % reliability ), hrs
- \( C \) = bearing dynamic load rating, N
- \( P \) = bearing equivalent dynamic load, N
- \( p \) = load-life exponent
- \( n \) = bearing rotational speed, rpm

The basic rating life, \( L_{10b} \) was established as the period of time a group of identical bearings could predictably operate without subsurface initiated fatigue or spalling. Shown on a log-log verses log scale this basic rating life defined a finite life for a given set of operating conditions, Figure 1. This prediction of bearing life has been used as the basis of standardization by ISO (and ANSI/ABMA ) and by general industry for the selection of bearings for applications, including compressors.

By the 1980’s this original methodology for the prediction of bearing life was found to underestimate the actual longer lives observed in well maintained and lubricated bearings in practice and in tests. Longer bearing lives were attributed to improvements in the bearing steel manufacturing processes, improvements in bearing internal geometry, and the influence of lubrication on bearing life. An adjustment factor, \( a_{23} \) was established by ISO as a modifying factor for an adjusted rating life, \( L_{10ah} \) as a function of lubricant viscosity ratio, \( \kappa \) (Kappa):

\[
\kappa = \frac{\nu}{\nu_1} \quad \text{Eq. 2}
\]

Where

- \( \nu \) = lubricant viscosity, cSt at bearing operating temperature
- \( \nu_1 \) = minimum required lubricant viscosity, cSt

The minimum required lubricant viscosity, \( \nu_1 \) is defined as follows:

\[
\nu_1 = \frac{4500}{\sqrt{n}d_m} \cdot 3 \cdot \sqrt{\frac{1060}{n}} \quad \text{Eq. 3a}
\]
Where \( \nu_1 = \frac{4500}{\sqrt{nd_m}} \) for \( n > 1000 \) rpm \hspace{2cm} \text{Eq. 3b} \\

For well lubricated situations, \( \kappa \) is greater than 1. \( \kappa \) less than 1 is less favorable lubrication, indicating a reduction in bearing life.

Despite this consideration of lubrication on bearing life using an \( a_{23} \) factor, the adjusted rating life, \( L_{10\text{ah}} \) still did not account for the shorter lives observed in bearings due to surface initiated fatigue (i.e., particle denting) when operated in contaminated conditions or the finding of longer lives for bearings operating in clean conditions. Moreover, endurance tests of bearings in clean and well lubricated conditions showed the existence of a threshold value of stress below which fatigue did not occur and infinite bearing life was observed. Zaretsky[2] suggests that Palmgren may have recognized the existence of a fatigue stress limit as well as the applicability of the basic rating life concept to surface initiated fatigue in addition to subsurface fatigue in his original 1924 work.

NEW LIFE METHOD

In 1985 Ioannides and Harris[3] published their new model for bearing life prediction which included a fatigue stress limit and the concept of a variable stress field within the bearing. This new model could then be used to consider the influence of solid particle contamination. This was adopted as the New Life Method [4]. The same methodology is presently proposed, by an ISO working group, for inclusion into an updated ISO Standard. The working group is comprised of bearing users and manufacturers. The proposal is awaiting final comments from ISO members. The New Life Method rating life, \( L_{10\text{ah}} \) defines that for loads below a fatigue load limit in clean and well lubricated conditions, bearings can expect to have infinite life. See Figure 1. Furthermore, the influence of solid particle contamination on bearing life can be predicted. This is applied using an adjustment factor, \( a_{XYZ} \):

\[
L_{10\text{ah}} = a_{XYZ} L_{10h} \hspace{2cm} \text{Eq. 4}
\]

Where \( L_{10\text{ah}} = \) bearing rating life (90% reliability) considering the New Life Method.

Figure 2 shows, in principle the interdependence of lubrication (\( \kappa \)), the applied load (\( P \)), the fatigue load limit (\( P_u \)), and the contamination (\( \eta_c \)). For well lubricated bearings (\( \kappa > 1 \)), having light loads (\( P < P_u \)) and good cleanliness (\( \eta_c \to 1.0 \)) the \( a_{XYZ} \) factor approaches a 50 times improvement to the basic rating.

The use of the New Life Method allows a more accurate prediction of bearing life. Bearings can now be selected considering the inherent cleanliness in compressor lubricant systems. Smaller size, lower cost bearings can be employed with satisfactory reliability and a reduction in power loss. The importance of lubrication system filtration can also be appraised.
LUBRICATION

In compressor applications it is usual for the bearings to be lubricated by synthetic lubricants. Depending on the pressure-viscosity characteristic of the synthetic lubricant compared to that of mineral oils, the bearing life can be increased or shortened. Bearings operating with lubricants having lower viscosity increase with pressure (i.e., lower pressure-viscosity coefficient, \( \alpha \)) will have thinner EHD film thickness and therefore shorter lives. To consider the influence of the synthetic lubricant on bearing life using the New Life Method, an adjusted viscosity, \( \nu_{\text{adj}} \) is used:

\[
\nu_{\text{adj}} = \nu \left( \frac{\alpha_{\text{synthetic}}}{\alpha_{\text{mineral}}} \right)^{0.72}
\]

Eq. 5

Where

\( \alpha_{\text{mineral}} \) = pressure-viscosity coefficient of mineral lubricant, MPa\(^{-1}\)

\( \alpha_{\text{synthetic}} \) = pressure-viscosity coefficient of synthetic lubricant, MPa\(^{-1}\)

This adjusted viscosity, \( \nu_{\text{adj}} \) is used in Eq. 2 to determine the adjusted, \( \nu_{\text{adj}} \) which intern is used to determine the New Life Method adjustment factor, \( a_{\text{XYZ}} \).

REFRIGERANT CONDITIONS

In an air atmosphere, a protective oxide layer is formed on the bearing surfaces which aids in the protection of the metallic surfaces in the rolling contact. A similar protective layer is formed on bearings operating with lubricants having extreme pressure (EP) additives. The chlorine molecules of the now banned CFC-12 refrigerant provided a protective surface layer much like an EP additive. Jacobson [5] found that the reduced chlorine of the HCFC-22 refrigerant and the absence of chlorine in the HFC-134a refrigerant significantly increases the viscosity requirements for rolling bearing lubrication. He estimated that two times greater operating viscosity is need for a HCFC-22 / mineral oil lubricated bearing and three time greater viscosity is needed for a HFC-134a / polyol ester (POE) lubricated bearing compared to an air / mineral oil lubricated bearing. This is represented by the following equations:

\[
\begin{align*}
\text{HCFC-22} & \quad \nu_{\text{adj}} = 2 \nu_1 \\
\text{HFC-134a} & \quad \nu_{\text{adj}} = 3 \nu_1
\end{align*}
\]

Eq. 6a

Eq. 6b

Where \( \nu_{\text{adj}} \) = adjusted minimum lubricant viscosity, cSt considering the refrigerant / lubricant mixture

These adjustment factors to the minimum required viscosity are referred to as the Jacobson Numbers.
The adjusted minimum viscosity, $v_{1 \text{adj}}$ and the effective viscosity based on the refrigerant dilution (Figures 3) can be used to determine the adjusted $k_{\text{adj}}$ for use in the New Life Method to consider the influence of the refrigerant/lubricant mixture. The curve for the HFC-134a/POE mixture of Figure 3 also considers the difference in the pressure-viscosity coefficient, $\alpha$ of the POE oil. This laboratory work compares very favorably with field experience. More work is necessary to further refine the work of Jacobson and to include other refrigerant/lubricant mixtures.

**DUTY CYCLE**

To select the appropriate bearing for a compressor application it is best to consider that the applied load as well as the lubricant viscosity may vary during the life of the compressor operation. Using only the heaviest loading condition or extremes of viscosity, etc. can result in an oversized and costly bearing arrangement. Excessive power loss and skidding damage due to operation with too light applied load can result. It is recommended that the rating of the bearing be made considering a duty cycle calculation. Computer programs are available to estimate bearing rating life using a duty cycle calculation and which also include the New Life Method.

**CONCLUSIONS**

The New Life Method for bearing selection has been successfully applied in compressor applications. The same basic methodology is proposed for inclusion in an updated ISO Standard. Computer programs are available to estimate bearing rating life using the New Life Method. These consider the fatigue load limit, $P_u$, the cleanliness of the system, $\eta_c$, and the lubrication condition, $\kappa$. The adjusted viscosity, $v_{\text{adj}}$ and adjusted minimum viscosity, $v_{1 \text{adj}}$ can be used to determine the adjusted $k_{\text{adj}}$ for use in the New Life Method depending on the lubricant type and the refrigerant/lubricant mixture. A duty cycle calculation of load and lubricant conditions is an appropriate method for selection of bearings. Additional work is need to further refine the Jacobson Numbers. Comparisons to field operation of bearings in compressors indicates very favorable correlation.

**REFERENCES**

2 Zaretsky, E., A. Palmgren Revisited-A Basis for Bearing Life Prediction, Lubrication Engineering, February 1998
4 SKF General Catalogue, 3000E, 1995
5 Internal SKF Interoffice Letter, Jacobson, B., Lubrication selection and life calculation in refrigeration conditions, November 18, 1996
Figure 1  New Life Method having fatigue load limit

Figure 3  Adjusted viscosity for refrigerant dilution