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MODERN APPROACH TO ROLLING BEARING SERVICE LIFE PREDICTION

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Abstract

Capability to accurately predict rolling bearing service life is necessary to the optimal maintenance of rolling bearing performance in terms of operational quality and reliability. Over the last several decades, the life factor-based model, based on the theory of Lundberg and Palmgren, has been widely accepted both for industry standards and in manufacturers catalogs in the bearing industry. However, modeling the increasingly demanding operating conditions of modern machinery, suggested equation does not reflect actual bearing life for all operating conditions. Hence, a new and improved method for the prediction of bearing service life is proposed to include all the conditions which influence rolling bearing service life by converting these conditions to an integreted material stress field. This paper presents a review of rolling bearing service life prediction methods, with the emphasis on modern approach to this impotrant problem.

Keyword: *rolling bearings, life prediction theories, strees-life method*

1. INTRODUCTION

There are many different types of ball and roller bearing design and many different application. The selection of rolling bearings for a definite purpose assumes the possession of the corresponding procedure for determining the type and size of the bearings of different manufacturers. The selection of type and size of rolling bearings is carried out on the basis of operational conditions and also required service life. Since modern bearing applications incrisingly demand more compact bearings with greater load-carrying capability, to select optimum size bearing it is necessary to predict bearing service life as accurately as possible.

It is entirely unacceptable and expensive to determine service life of particulare types of bearings for a given purpose only by test. That is why analytical methods were proposed, which are most often result of theoretical and experimental investigation.

2. BEARING LIFE THEORIES

The first generally accepted method to predict rolling bearings service life was published in 1947 by Lundberg and Palmgren. That method focused on the most appropriate bearing failure mode of those time, e.g. subsurface originated fatigue. The fatigue theory of Lundberg and Palmgren presumes that a fatigue crack is initiated at a distance below the surface in rolling contact, consistent with the simultaneous occurrence at that distance of a maximum orthogonal shear stress and a weak point in the material. Such weak points are presumed to be stochastically distributed throughout the material. The entire subsurface stress distribution is assumed to be a function only of the Hertz stresses acting normal to the rolling element and raceway surfaces. This does not mean that Lundberg and Palmgren overlooked the effects of non Hertzian stresses. According to reference [**1**], the omission of such parameters relates only to the calculation tools available at the time of development.

The basic life equation developed by Lundberg and Palmgren states that for a bearing ring subjected to a number of cycles N of repeated concentrated stress, the probability of survival from subsurface initiated fatigue is given by:

$$ln\left(\frac{1}{S}\right) \approx \frac{\tau_0^{\,c} N^e V}{z_0^{\,h}} \quad (1)$$

in wich τ_0 is the maximum orthogonal shear stress under the contact surface and z_0 is the depth in wich it occurs. Furthermore, they assumed that volume in fatigue risk V was proportional to the volume bounded by the semi-major axis of the Hertzian contact elipse a, the depth z_0 and the lenght of running track of the race l, that is:

$$V = alz_0$$

This proportionality is satisfactory only if the surfaces are geometrically perfect and a simple normal stress occurs between contacting surfaces. In the event of a significant surface shear stress, for example, the strong tendency for surface initiated fatigue failure is completely ignored. Since then bearings have been continuously improving in terms of design, manufacturing precision, steel integrity and heat treatment. Moreover, cleanliness of modern steels has significantly reduced the occurrence of subsurface initiated bearing failure, and failure now frequently commences at the rolling element and raceway surfaces.

It must further be understood that bearing fatigue lives calculated using Eq. (1), as inffered in reference [**1**], pertain to bearings which fail due to failure of bearing raceways. Particulary, for ball bearings, Lundberg and Palmgren assumed that the probability of ball failure was much less than that of the raceway. Thus, the incidence of rolling element failure is not included in Eq. (1).

The principal equation derived from Eq. (1), relates the bearing fatigue life to applied load:

$$L_{10} = \left(\frac{C}{P}\right)^p$$

where L_{10} is fatigue life in milions of revolutions which 90 percent of bearings will achive, C is basic dynamic capacity, P is the applied equivalent radial load and p is the load-life exponent. Accepting this criterion means that the bearing users is willing, in principle, to accept that 10 percent of a group of bearings will fail before this time.

However, the improvements of the bearing quality in all domains, have conditioned the production of bearings with much longer life than the calculated one, which led to the introduction of new magnitudes into the calculation. To accomodate various levels of reliability, as well as various steels from which bearings are manufactured and effect of lubrication on service life, the modified life equation retained the calculation of the L_{10} life, but complemented by life adjustment factors,

$$L_{na} = a_1 a_2 a_3 L_{10}$$

containing three factors:

- a_1 life adjustment factor for the reliability,
- a_2 life adjustment factor for non-conventional material,
- a_3 life adjustment factor for non-conventional operating conditions,
- L_{na} adjusted bearing fatigue life in milions of revolutions.

Values for each of these factors were provided, for example, in rolling bearing manufacturers catalogs [**2**].

Unfortunately, the modified life equations are not able to predict the bearing fatigue life for any given bearing over a complete range of operation. For example, at very small loads, in the absence of lubrication contamination, incorrect or misaligned mounting, the occurrence of rolling contact fatigue is non existent. Hence it appeared that, similar to other mechanical components fatigue endurance, bearing fatigue life is also influenced by fatigue limit stress. This means that, if a component in rolling contact does not experience stresses in excess of the fatigue limit for its material, fatigue failure will not occur.

In 1985, Ioannides and Harris [**3**] proposed a method based on the Lundberg-Palmgren theory, but including fatigue limit stress. Life prediction according to their method could ideally by infinite if the fatigue limit stress is not exceeded.

To accommodate this condition, Eq. (1) was subsequently modified to:

$$ln\left(\frac{1}{\Delta S_i}\right) \approx \frac{\left(\sigma_i - \sigma_{limit}\right)^c N^e \Delta V_i}{z_i^{\,h}} ; \sigma_i > \sigma_{limit} \quad (2)$$

where ΔS_i is the probability of survival of material volume element ΔV_i subjected to initiating stress σ_i.

The use of σ as the failure initiating stress implies that maximum orthogonal shear stress τ_0 does not nave to be used as the failure initiating stress. If $\sigma_i \geq \sigma_{limit}$, volume element ΔV_i is not subjected to fatigue. Thus potential for unlimited life exists if all elements are stressed less than limit stress. When τ_0 is selected as failure initiating stress and $\sigma_{limit}=0$, the Eq. (2) is identical to that Eq. (1).

3. STRESS-BASED LIFE METHOD

The need to update bearing service life prediction methods has been recognized for a number of years. Users require more accurate bearing service life prediction, with improved reliability and improved prediction methods.Modern understanding recognizes that the aforementioned bearing life theories do not reflect bearing service life for all operating conditions. Therefore, it is neccesary to establish an improved and universally acceptable method for predicting the service life of modern rolling bearings.

Recently, the trend has been toward the use of the full subsurface stress field in rolling bearings service life prediction, e.g. stress field-based life models. The challenge is to determine all significant parameters that influence bearing service life and to express them in terms of generalized stress field.

The improved procedure, adopted as a basis for the international standard ISO 281/2 [**4**], replaces the independent multiplying factors with single interdependent factor derived from generalized bearing stress field.

Many different forms of imposed stress combine to form generalized stress field that exists in all bearing aplications. Examples of imposed stresses include those of Hertzian contact, surface shear, interference fits, hoop, residual and contaminant induced. It is the stress field that determines the sequence of events for both subsurface and surface initiated failures.

Many innovative and sophisticated calculation methods, presented in reference [**5**], have been incorporated into the new methodology in orded to define accurately generalized stress field.

The capability of the material to resist these stresses depends on the material chemistry, metallurgy, heat treatment, material cleanliness and the manufacturing and processing of the bearing, which produce a strenght or fatigue limit stress below which no failure would be expected to occur.

To accommodate effect of stresses other than Hertz stresses it is appropriate to use the von Mises stress as a failure initiating stress. Moreover, von Mises stress is appropriate since it is easy to compute, and for general engineering usage, consideration of the ease of computation should also be given in the selection of a stress criterion.

In simple terms, bearing failures occur when the generalized stress exceeds the bearing material strenght. This comparison between generalized stress field and limit stress must be made on a local level at all points of the bearing material and then integrated throughout the material volume to establish bearing life.

For continuity purposes, the concept of basic dynamic capacity C and equivalent load P, originally derived by Lundberg and Palmgren are preserved in the proposed procedure, and the effects of the applied, induced, material residual stresses and the presence of a fatigue limit stress are taken into account by the use of a single stress-life factor a_{SL} in the following equation:

$$L_{SL} = a_1 a_{SL} \left(\frac{C}{P} \right)^p$$

where a_1 is the reliability life factor and a_{SL} is a function of all parameters for which technology is sufficiently well advanced that their effect may be expressed in terms of generalized stress field.

The calculation of the stress-life factor may be acomplished using the method indicated in reference [**6**]. This method utilizes the numerical integration of the effective maximum von Mises stresses at every point from the contact surface through the zone of influential subsurface stresses. As given by [6], the stress-life factor can be estimated from following equation:

$$a_{SL} = \frac{\left\{ \sum_{j=1}^{j=n} c_j \sum_{k=1}^{k=n} c_k \left[\frac{\left(\sigma_{vM,jk} - \sigma_{vM,lim} \right)^c}{\xi_h^{\ k}} \right] \right\}^{1/e}}{\left\{ \sum_{j=1}^{j=n} c_j \sum_{k=1}^{k=n} c_k \left[\frac{\left(\sigma_{vM,jk} \right)^c_{LP}}{\xi_h^{\ k}} \right] \right\}^{1/e}_{LP}} \quad (3)$$

In Eq. (3) the c values are Simpson rule coefficients for numerical integration, while n is the number of increments. Also, ξ is the dimensionless distance z_i /b from the contact surface to the subsurface stress location. Eq. (3) utilizes the von Mises fatigue limit stress $\sigma_{vM,lim}$ in the determination of the numerical integral.

When only simple applied Hertz stress is considered and zero limit stress, $a_{SL}=1$.
The method predicts individual lives for the inner ring, outer ring and rolling element and then combines these individual lives to determin bearing life.

4. CONCLUSION

The foregoing discussion presents a new and improved method that incorporates the effects of many new and significant parameters to improve substantially the rolling bearing service life prediction.
For the first time, it expresses the integrated effects of Hertzian contact, surface shear, residual stresses, interference fits, lubricant rheology, contamination and material strenght. It recognizes that these effects are interdependent and cumulative.
In summary, the new methodology allows prediction of rolling bearing service life with a high accuracy, on a consistent basis in various operating conditions.

5. REFERENCES

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