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Modeling and Optimization of Bearing Life Using a Statistical Approach

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Abstract The bearing life problem is a global issue that concerns all manufacturing companies, especially those using rotating machines in which bearings play an important role in guiding rotating shafts. Bearing life is traditionally evaluated by analytical approaches that have become obsolete in large-scale production. In recent years, reliability-based statistical methods have been introduced to evaluate bearing life. The work presented in this article consists, on the one hand, of studying and synthesizing the various existing life models, and on the other hand, based on the Palmgren-Lundberg model, of proposing an optimized bearing life model. This article has shown that in the case of operation with significant shocks, the weighted (optimized) life model proposed for ball and roller bearings allows for a greater increase in life compared to the corrected model that had been proposed to better predict the fatigue endurance of bearings.

Keywords: *lifespan, maintenance, modeling, reliability, rolling*

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1. Introduction

Industrial production operates in an increasingly stringent environment regarding quality, safety, and noise. Rotating machinery therefore requires effective maintenance. Indeed, unexpected catastrophic breakdowns result in both lost production and very high repair bills. Machine downtime for repairs is undesirable for manufacturers, who, despite all available maintenance methods, are faced with failures related to machine components. Among these failures, bearing failures represent a significant proportion observed in rotating machinery (approximately 69% in synchronous machines [1]).

A rolling bearing is defined as a body formed by two concentric rings called an inner and outer ring, comprising raceways between which are rolling elements uniformly spaced by a cage. This cage is placed between the rings and serves to reduce friction by preventing the elements from rubbing against each other [2]. Like all mechanical components, the causes of bearing defect are numerous. Most commonly, these are chipping, seizure, and corrosion. All these defects have in common the fact that they sooner or later result in the loss of metal fragments. Bearings are integral components of our daily lives, linked to what transmits movement in the fields of transportation and rotating machinery.

Analyzing the various characteristics of bearings and their limitations allows us to establish selection criteria [3,4]. The challenge is to design and manufacture bearings with a sufficiently long service life. It is in this context that several researchers initially attempted to model bearing service life using an analytical approach. This made it possible to evaluate service life by taking into account parameters such as stress or geometry in the design. However, this approach has limitations when it comes to optimizing service life. More recently, statistical approaches have been used to model bearing service life. In this work, we will synthesize analytical and statistical approaches to lifespan modeling. Drawing on these approaches, we will attempt to propose an optimized lifespan model. The objective of this work is twofold:

- first, to synthesize the various existing lifespan models;
- second, to build on these models to propose an optimized lifespan model.

2. Life Modeling

2.1. Bearing Fatigue Life Modeling

From an analytical perspective, the bearing life is defined by the number of revolutions or hours required by the designer at a given constant speed before the bearing shows the first signs of material fatigue [5].

$$L_h = \left(\frac{C}{P}\right)^p \frac{10^6}{60.N} \quad (1)$$

Where:
Lh: Nominal operating life in hours;
C: Basic dynamic load rating, N;
P: Equivalent dynamic load rating, N;
p=3: For ball bearings; p=10/3: For roller bearings;
N: Number of revolutions per minute.

2.2. Corrected Fatigue Life Model

The standard (ASA) was not in effect for long before changes became necessary. During the 1960s, with the advent of improved cleanliness, such as commercial ball steels, Consequently, the following equation was proposed to better predict bearing fatigue life:

$$L_{na} = a_1 a_2 a_3 \left(\frac{C}{P}\right)^p \quad (2)$$

a_1 : The life adjustment factor for the selected reliability level;

a_2 : The life adjustment factor for the material;

a_3 : The life adjustment factor for the operating conditions;

L_{na} : The corrected bearing life at $10^{\wedge 6}$ cycles.

For survival probabilities S, the values of the reliability factor a_1 are given in Table 1.

Table 1. Reliability factor [6]

S	0,90	0,95	0,96	0,97	0,98	0,99
a_1	1	0,62	0,53	0,44	0,33	0,21

2.3. Damage-Based Life Modeling

Bearing damage is sometimes very difficult to predict, given the many causes and mechanisms involved. However, it remains a key issue in the certification of parts that are also subjected to stress over a very large number of cycles. Damage to rolling elements can be divided into two categories: • First, there is direct damage, in which parts may be damaged during manufacturing, transport, assembly, or during operation when an incident occurs; and • Second, there is indirect damage, i.e., damage resulting from a long process of fatigue and wear of the parts. While the first type of damage is often easy to detect, the second is more based on the probability of critical events occurring during operation and therefore requires the use of models or criteria to be predicted [5].

2.4. Richard Stribeck's Model

Ball and roller bearings' use in industrial applications can generally be attributed to the late 19th century. With their increased use and the growing sophistication of mechanical design, their endurance capabilities began to be predicted. The first formulation of the internal load associated with rolling bearings is attributed to R. Stribeck, who established the equation for the most loaded ball in a radial ball bearing with zero radial internal clearance and subjected to a simple radial load [5]:

$$Q_{\max} = 4,7 * F_r / Z \quad (3)$$

Where (Fr) is the applied radial load and (Z) is the number of balls. Stribeck also identified his equation with the static load capacity (CS) as [5]:

$$C_S = KZD^3 / 5 \quad (4)$$

Where (D) is the ball diameter and (k) is a constant associated with the level of permanent deformation to be determined empirically. These relationships were to prove useful in later establishing the fatigue endurance of bearings [5].

2.5. John Goodman Model

It was J. Goodman who initially identified the phenomenon of rolling contact fatigue, which was later used as the basis for evaluating the endurance of ball and roller bearings. At the time, consistently producing balls of uniform size for a given bearing was not easily achievable. It was assumed that, in a given ball complement, the larger balls would wear until the entire ball set consisted of balls of the same diameter; after such wear, the bearing would operate smoothly. J. Goodman was the first to specify the spalling condition, i.e., pitting of the rolling contact surfaces, as the endurance limit. Goodman specified the following equation for the load capacity of a ball on the raceway [6]:

$$Q_S = \frac{KD^3}{n_S d + AD} \quad (5)$$

Where (d) is the raceway diameter, (n_S) is a function of the bearing's rotational speed.

Around 1919, the Swedish SKF company introduced the following equation in its catalog for the maximum permissible load (Cn) for the proper operation of ball bearings [6]:

$$C_n = f_{cn} \frac{iZD^3}{5} \quad (6)$$

Where (i) is the number of ball rows and (fcn) is a function of speed. The similarity to Stribeck's equation is apparent; the relationship to rotational speed was derived from Goodman. In 1921, SKF introduced a similar equation for roller bearings [6]:

$$C_n = f_{cn} \frac{iZlD^3}{5} \quad (7)$$

Where (l) is the total length of the bearing and (D) is the bearing diameter. The length and diameter of the bearing were usually denoted by a subscript (w), for example (lw) and (Dw), where (w) stands for walzlager [6].

2.6. Arvid Palmgren's Model In 1924

In 1924, A. Palmgren published the first equation for evaluating the life of ball bearings:

$$L_{14} = \left(\frac{C_1}{P - P_u}\right)^3 - C_2 \quad (8)$$

Where C_1 load capacity, C_2 minimum life constant, P applied load, Pu limit load, and L_{14} fatigue life for which 14% of bearings would not survive.

Thus, A. Palmgren introduced the concept of probability of failure into a bearing life assessment and into the life prediction equation. He also defined a minimum applied load for which, if not exceeded in a bearing application, the bearing was not considered vulnerable to fatigue failure. Unfortunately, ball bearings from this period, endurance tested under high loads to minimize test times, did not show the theoretical tendency toward infinite life, and the concept was not put into practice. However, the idea of the probability of failure associated with a given bearing operating under a given load was confirmed. Ball and roller bearings became the first machine components to be classified according to their probability of failure. In 1928, SKF (*Bearing Manufacturing Compagny*, www.skf.com) introduced a safety factor of 1 associated with a fatigue life of 200 hours. According to A. Palmgren, this represented "an appropriate compromise between the requirements of reliability and economy."

Therefore, it was necessary at that time to empirically develop life indices for ball and roller bearings through long-term endurance tests that could not logically be extrapolated beyond the test parameters. The effect of lubrication on the fatigue life of bearings was also known only empirically. It was generally stated that lubrication should be sufficient to achieve the expected lifetimes [6].

2.7. Model By G. Lundberg and A. Palmgren

The most important publication on rolling bearing load and life was produced by Lundberg and Palmgren in 1947. Based on the research of W. Weibull, they postulated that fatigue cracking begins at weak points in the material beneath the contact surfaces. The quantities and types of weak points in raceways and rolling elements depend on the material's chemical composition, metallurgical structure, and homogeneity. The greater the volume of material stressed, the greater the risk of fatigue failure, according to W. Weibull. In 1949, W. Weibull published a specific work on his statistical theory of fatigue failure in solids. Based on Weibull theory, Lundberg and Palmgren developed the following equation which underlies all their subsequent developments. This basic equation relates the fatigue life to the number of stress cycles experienced by the first chip, the applied stress and the probability of survival:

$$\ln\left(\frac{1}{S}\right) \propto \frac{\left(N^e \tau_0 V\right)}{Z_0^h} \tag{9}$$

Where S is the survival probability, N is the number of stress cycles experienced, τ0 is the maximum orthogonal shear stress below the contact surface, Z0 is the surface depth at the location of τ0, and V is the volume of the stressed material. The exponent (e) is called the Weibull slope or shape parameter. As shown in Figure 1, (e) is the slope of the failure distribution plotted on a Weibull coordinate graph paper.

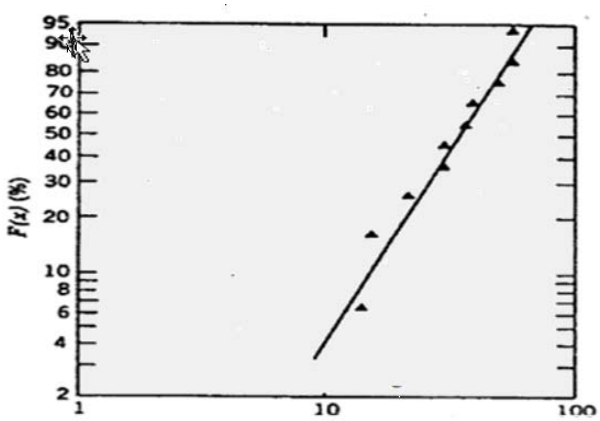


Figure 1. Fatigue life plot (action) on Weibull coordinates [5]

According to G. Lundberg and A. Palmgren, e = 10/9 for point contact bearings and e = 9/8 for line contact bearings. Exponents (c) and (h) are determined empirically; they depend on the value of the Weibull slope (e). The main equation resulting from the effort of G. Lundberg and A. Palmgren, derived from equation (10), relates the fatigue life of the bearings to the applied load [6]:

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

Where L10 = 90% reliability, extending the fatigue life to $10^{\wedge 6}$ cycles. C: basic dynamic capacity, P: equivalent load, p: life exponent.

Equation (10), like equation (8), assumes that not all bearings subjected to the applied load will survive the fatigue life. Rather, only 10% of bearings are expected to survive. (L), also referred to as (L10), is called the life. G. Lundberg and A. Palmgren designated bearing types based on point contact and line contact. Point contact occurs if two bodies, such as a ball on a raceway, contact a single point under zero load; line contact occurs if two bodies, such as a cylindrical roller on a raceway, touch along a line under zero load. The life exponent (p) was experimentally determined by G. Lundberg and A. Palmgren and takes the value 3 for ball bearings. The same value is also assumed for roller bearings operating in point contact, e.g., lightly loaded spherical roller bearings (SRBs). For line contact roller bearings, (p = 4). Table 2 gives the values of the exponents (e, c, h, p). It can be seen that the exponents (c) and (h) are identical for point and line contact; they are considered material constants [6].

Table 2. Values of the exponents e, c, h and p as determined by G. Lundberg and A. Palmgren [6]

Exponent	E	c	h	P
Contact Point	10/9	31/3	7/3	3
Contact Line	9/8	31/3	7/3	4

2.8. Ioannides and Harris Model

Ioannides and Harris developed a new model to predict bearing life and overcome some of the shortcomings of the Lundberg-Palmgren model. Their first assumption is based on the discretization of material volumes with different survival probabilities. The survival probability is

obtained for each portion of the material volume, and the overall material failure risk is obtained by integrating the survival probabilities of the volume portions. Furthermore, Ioannides and Harris recognized the existence of a fatigue limit below which failure does not occur. They established a valid relationship for σ > σu between the material survival probability S and the bearing life obtained from equation (11) [7].

$$\ln\frac{1}{S} \propto AN^{e}\frac{(\sigma-\sigma_u)}{z^{h}}dV \tag{11}$$

σ is the stress at depth z, σu is the fatigue limit of the material and A is an empirical constant. The stress σ, which is no longer limited to the orthogonal shear stress, can be extended to the shear stress, the Von Mises equivalent stress. The other parameters (c, e and h) are material parameters obtained from a series of tests. The service life L10 is then obtained with relation 12 [7]:

$$L_{10} = \frac{A}{\left[1-\left(\frac{P_u}{P}\right)^{(c/e)}\right]}\left(\frac{C}{P}\right)^{p}, P > P_u \tag{12}$$

With Pu, the load corresponding to the limiting stress σu. Figure 2 presents a comparison between the Lundberg-Palmgren and Ioannides-Harris lifetime models and highlights the notion of infinite lifetime [7].

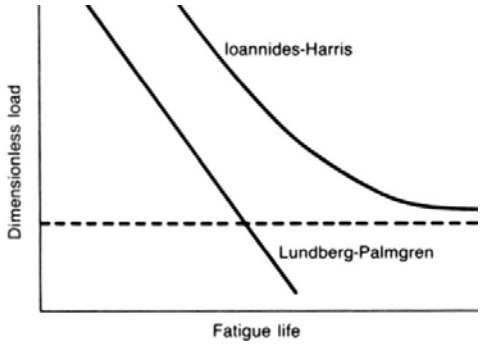


Figure 2. Comparison between the Ioannides-Harris and Lundberg-Palmgren theories [6]

2.9. Dahl Model

Dahl developed his model to simulate control systems with friction. Dahl's starting point was to conduct several experiments on friction in servo systems with ball bearings. One of his discoveries was that rolling friction behaves like that of solids. Dahl's model is inspired by the characteristics of the stress-strain curve in solid mechanics [8].

This curve is modeled by the following differential equation [8]:

$$\frac{df(\dot{x})}{d_x} = \sigma_0\left(1-\frac{f(\dot{x})}{\mu_c}\mathrm{sgn}(\dot{x})\right)^{\alpha} \tag{13}$$

Where x, σ_0, f ($\dot{x}$), μ_c and $\dot{x}$ represent respectively the displacement, the friction force, the Coulomb coefficient, the stiffness coefficient (rigidity) and the velocity. The parameter α determines the shape of the stress-tension curve (usually α= 1) given in figure 3.

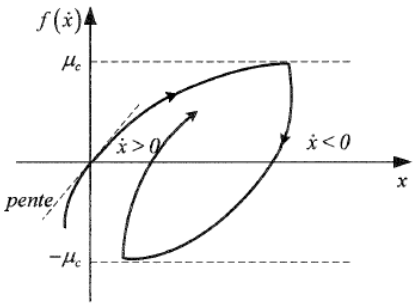


Figure 3. Dahl model [7]

2.10. Palmgren and Lunlberg Life Optimization

Our proposed approach is derived from the models presented above. Since the models of John Goodman and Arvrid Palmgren predicted significantly shorter fatigue lives than actually occurred in some steel applications, we propose a corrected and weighted fatigue life model. For survival probabilities S, the values of the reliability factor a1 are given in Table 1. We know that the bearing fatigue life L_{10} is a function of the applied load $L_{10} = \left(\frac{C}{P}\right)^{p}$ p. The fatigue life L_{10} = 90% reliability, which increases the fatigue life to $10^{\wedge 6}$ cycles, with $a_1 = 1$. Thus, Equation 2, which was proposed to better predict the fatigue life of bearings, becomes:

$$L_{na'} = a'_1\, a'_2\, a'_3\left(\frac{C}{P}\right)^{p} \tag{14}$$

With:

L_{na} : The adjusted bearing life at;

C: the basic dynamic load, N;

P: the equivalent dynamic load, N;

p=3: for ball bearings;

p=10/3: for roller bearings;

a'_1 : the life adjustment factor for the weighted reliability level is given by equation (15):

$$a'_1 = \frac{a_1}{a_1} \tag{15}$$

a'_2 : The life adjustment factor for the weighted material is given by equation (16):

$$a'_2 = \frac{a_2}{a_1} \tag{16}$$

a'_3 : The life adjustment factor for weighted operating conditions is given by equation (17):

$$a'_3 = \frac{a_3}{a_1} \tag{17}$$

3. Results and Discussion

To better understand the difference between the models, we use a simple example by calculating the service life for a few models. Consider, for example, the probability S = 99%, which gives $a_1 = 0.21$.

The coefficient a_2 is a function of the bearing steel property, and a3 is a coefficient of lubricant film efficiency in the connecting rod raceway contact during separation of the contact surface.

Assume that $a_2 = 1$ and $a_3 = 1$. For the static load case, the basic static load Co is defined by the pressure at the center of contact between the inner or outer ring raceway and the most heavily loaded ball.

The pressure values are given by SKF (www.skf.com) for various bearings:

- 4,600 MPa for self-aligning ball bearings;
- 4,200 MPa for all ball bearings;
- 4,000 MPa for other roller bearings.

The equivalent static load P0 is given by equation (18) as:

$$P_0 = X_0\, Fr\ + Fa \tag{18}$$

Where:

P0 = equivalent radial static load, N.

X0 = radial static load factor

Y0 = axial static load factor.

Fr = radial component of the load, N.

Fa = axial component of the load, N.

The relationship between Co and Po is given by equation (19), which is expressed as [5]:

$$C_o = S_o * P_o \tag{19}$$

With So: static safety factor given in Table 3. Knowing Co, we can calculate Po and vice versa. To simplify the example, we consider the static case, for ball and roller bearings. Three operating cases are possible (table 3):

Table 3. Ball bearing geometry (source, SKF)

Operating mode	Static Bearing	
	ball	roller
Regular/no vibration	0,4	0,8
Normal	0,5	1
Good/Severe shocks	1	2

Before proceeding with the calculations for the comparison of different approaches, we summarize some values of the geometry of the bearings in a table (4 and 5) with their references or codes thanks to the SKF catalog (Bearing Manufacturing Company) [23,24,25].

Table 4. Ball bearing geometry (source, SKF)

Code Bearing	Ball			
	d(mm)	D(mm)	B(mm)	dr (mm)
6003ZZ	17	35	10	6

Table 5. Cylindrical roller bearing geometry (source, SKF)

Code Bearing	Cylindrical rollers				
	d(mm)	D(mm)	B(mm)	dr (mm)	Z
204ET	20	47	14	6	8

The life of bearings or also called the nominal life does not only take into account the loads and rotation speeds to which they are subjected, but also the various external influencing factors such as the material quality factor, lubrication factor, etc. These factors allow us to obtain a more realistic estimate of the actual life.

Table 6. Static values for the 6003ZZ type bearing

Ball bearing (p= 3)					
Operating mode	S_0	C_0 (N)	P_0 (N)	Lna (cycles)	Lna' (cycles)
Regular/no vibration	0,4	791280	1978200	0,01	1,5
Normal	0,5	791280	1582560	0,03	2,8
Good/Severe shocks	1	791280	791280	0,2	22,7

Table 7.Static values for cylindrical roller bearing type 204ET

Roller bearing (p= 10/3)					
Operating mode	S_0	C_0 (N)	P_0 (N)	Lna (cycles)	Lna' (cycles)
Regular/no vibration	0,8	2688000	3360000	0,099887	10,8
Normal	1	2688000	2688000	0,21	22,7
Good/Severe shocks	2	2688000	1344000	2,11178247	228

Before performing the calculations of the corrected and weighted life, we looked for data on the static load of the Co bases (i.e. the contact pressure between the rolling elements) provided by the Bearing Manufacturing Company (SKF). The values are presented in MPa, then converted to Newton in order to perform the calculation of the corrected and optimized life.

By examining the table (6 and 7), we can see that the values of the corrected life are lower than those of the weighting for the case of ball and cylindrical roller bearings. We note in both (02) cases, the optimization is proportional (i.e. the optimization has multiplied the corrected life by a very significant factor, up to approximately 108 times for the different bearings) for each type of bearing (balls and rollers). Figure 4 shows the service life for the case of ball bearings for three operating cases (regular/without vibration, normal and Good/significant shocks).

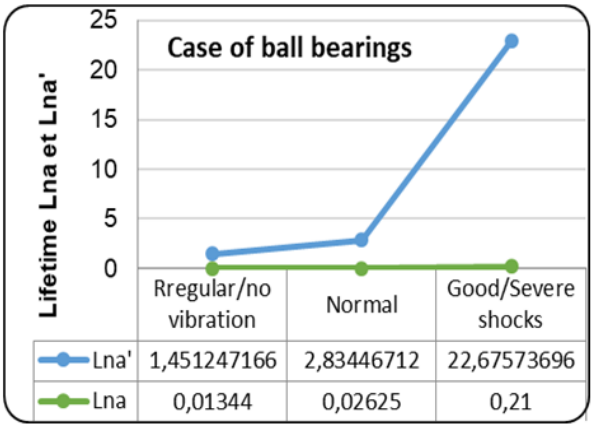


Figure 4. Corrected and weighted (optimized) service life (ball bearing, p=3)

The graph shown in the figure above highlights the following results:

- The corrected service life (Lna) of the ball bearing is low, around 0.01344 million revolutions for smooth operation without vibration, and reaches a maximum of 0.21 million revolutions for operation with significant shocks.
- The weighted service life (Lna') is much higher, with a service life of 22.6557 million revolutions for operation with significant shocks. This highlights a very significant increase compared to the corrected service life. The weighted and therefore optimized service life is even better when the bearings operate under severe operating conditions with significant shocks.

From the two graphs above (in blue and green), it is evident that weighted bearings are significantly more durable and wear-resistant than those without such improvements. This highlights the importance of optimizing ball bearing operating conditions to ensure longer life, reduced costs and increased performance.

Figure 5 shows the service lives for cylindrical roller bearings.

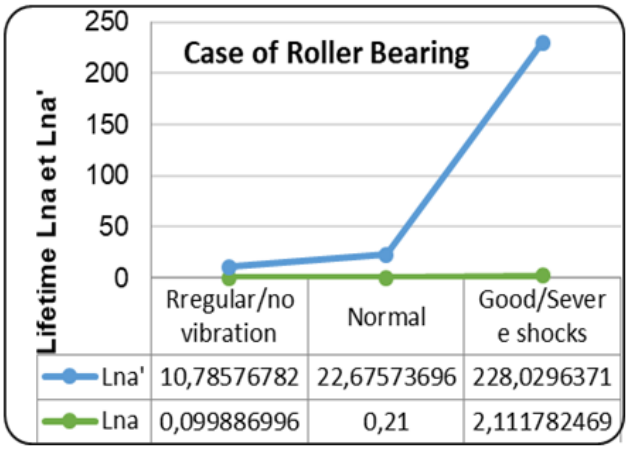


Figure 5. Corrected and weighted service life (cylindrical roller bearing, p=10/3)

In this figure:

- the corrected service life (in green) of cylindrical roller bearings is low at 0.1 million revolutions for smooth, vibration-free operation and increases to approximately 2.1 million revolutions for smooth operation and significant shocks.
- the weighted service life (in blue) of cylindrical roller bearings is much higher and reaches a maximum of 228 million revolutions for operation with significant shocks.

The results obtained show that weighting can optimize bearing service life. This can result in improved performance and reduced long-term maintenance costs.

4. Conclusion

In this work, we synthesized the different bearing life models proposed in the literature and proposed an optimized model based on the Palmgren and Lunlberg bearing life model. Using the basic data of the Bearing Manufacturing Company (SKF), we performed life calculations for two models for comparison. The proposed optimized bearing life model given in equation 14 presents better results compared to the corrected life results given in equation 2 for the different cases of bearings operating under different conditions.

These results show us a significant improvement in bearing life and open new perspectives on bearing parameterization and design. In the case of good or sever chocks, the results are too significant in regards of corrected model.

However, these results could be validated from a full-scale lifespan test.

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