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COMPARISON OF BEARING LIFE THEORIES AND LIFE ADJUSTMENT FACTORS

Tom Stella

SEi Companies
88 Black Falcon Ave, Suite 210
Boston, MA 02210
tstella@seicompanies.com

Abstract

The life predication theories of Weibull, Lundberg-Palmgren, Ioannides-Harris and Zaretsky will be compared to determine the one that most accurately predicts the life of a modern ball bearing. The effect of the Weibull slope on bearing life prediction shows that Lundberg-Palmgren best predicts life for 9th power stress life common with air melted steels. Zaretsky's Equations best predicts life for the modern steel's, 12th power stress life. Since Lundberg-Palmgren's life theory was adopted by ANSI, ISO and ABMA as the standard, steel quality has improved. Life adjustment factors are needed to fine-tune the life formulas towards more accurate results. Open problems in the field have two objectives, improve bearing life and improve life predication. This is being accomplished with better steels, hybrid bearings and coatings of steel bearings.

1. Introduction

The subject of this paper is to compare the various ball bearing life prediction theories and the life factors that influence a bearing's life. The scope of this paper is limited to ball bearings, but all bearings will have similar governing equations and life factors. Roller bearing technology has been around for over 4000 years and has evolved from the ancient Assyrians moving stones on rollers to modern turbine engines that are use to propel aircraft [5]. Ball bearings are used in everything from pump and fan assemblies to automobiles and aircraft or most other mechanical devices with rotating components [7,8,9].

The purpose of a ball bearing is to transmit a load between two structures while providing rotational and position freedom. Ball bearings are desirable when to compared other bearings, such as fluid film bearings, for example when low starting torque and low friction is required, temperature can be controlled, there are changes in the lubrication conditions, and combined radial and thrust loads are present [6].

Ball bearing failure has numerous origins, including steel composition and properties, steel cleanliness, hardening treatments, lubrication, dirt, mounting, manufacturing methods and surface roughness. If a bearing passes all of the above mentioned it will fail, like all other mechanical equipment from metal fatigue. Fatigue is by definition, the failure of material caused by repeated cycles well below the yield stress. In bearings, fatigue is seen in the area of contact been the ball and races. This failure is physically manifested in bearings by the phenomenon spall. Spall is the propagation of cycle dependent loading cracks that occurs in the region of maximum shear stress [5,12].

In any mechanical fatigue test, there will be a wide dispersion in the specimen lifetimes. This is due to the inherent differences in material composition, strength and other factors from the manufacturing process. In addition, the atomic structure is not fully homogenous causing some areas that are weaker or stronger than the surrounding materials. Due to this lack of uniformity, bearing samples tested under identical condition in a laboratory will exhibit wide-ranging dispersion in bearing life tests. For this reason, statistical analysis is needed to characterize the fatigue life of all mechanical parts, including ball bearings [5].

There are four major life theories that attempt to predict the life of a ball bearing assembly. The first was the Weibull distribution that was derived by W. Weibull in the late 1930s. He stated that the dispersion in experimental fatigue life could be expressed according the ratio of stresses. In the 1940s and 1950s, Lundberg and Palmgren extending the work of Weibull to rolling element bearings relating shear stress, cycles and maximum critical shear stress depth to fatigue life. Their combined work formed the groundwork for the American Bearing Manufactures Association (ABMA), International Organization for Standardization (ISO) and American National Standards Institute (ANSI) standards for the load ratings and life of ball bearings in the early 1950s. The next variation is the theory by Ioannides and Harris in 1985 which proposed that bearing steel had a fatigue limit, or loading where the specimen life is infinite (greater than one million cycles). The most recent work has been done by NASA [1960 to present] and researcher Erwin Zaretsky specifically. He has proposed a Weibull-based life theory that centered on more accurately accounting for exponents in the equations for critical shear stress [14,15].

The prediction of bearing life is an important part of any mechanical design and especially those where the bearing is a critical component, such as an aircraft engine. Therefore, the life of the bearing must be know with a high degree of accuracy and precision.

2. Life Theories

2.1 Weibull Equation

The Weibull Equation was the first step into predicting the life of mechanical samples. The use of statistical analysis is used to determine the properties of solids. Weibull said that the dispersion or spread in the material properties, namely strength, of a control group of test specimens could be expressed in the following equation:

$$\ln\ln\left(\frac{1}{S}\right) = e\ln\left[\frac{\sigma}{\sigma_B}\right] \qquad (1)$$

with σ = stress, σ_B = characteristic stress and S being the probability of specimen survival.

According to Weibull, Equation 1 can be used in predicting bearing life by rewriting into the following:

$$L_{10} = A\left(\frac{1}{V}\right)^{\left(\frac{1}{e}\right)}\left(\frac{1}{\tau}\right)^{\left(\frac{c}{e}\right)} \approx \frac{1}{S_{\max}^{n}} \qquad (2)$$

L = life in hours or cycles, A = material constant, V = stressed volume, S_{MAX} = maximum Hertz stress, τ = critical shear stress, e = Weibull slope, n = maximum Hertz stress-life exponent and c = critical shear life exponent. From Hertz theory it can be shown that shear stress is proportional to maximum Hertz stress and volume is directly proportional the maximum Hertz stress squared [2,11,12].

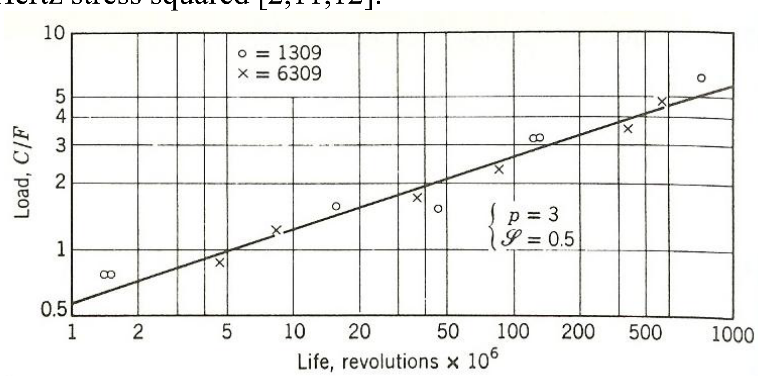

Figure 1: A Typical Weibull Plot for ball bearings [5]

2.2 Lundberg-Palmgren Equation

The Lundberg-Palmgren equation is a derivative of the Weibull equation. The original equation was adapted to fit both bearing geometry and statistical life data to the

Weibull parameters. The major change was that the maximum critical shear stress depth, z_o was introduced into the equation, which was assumed to be the orthogonal shear stress on the sample.

This was done because "it is possible that changes in condition of the material also depend on the depth below the surface of the volume element considered. This assumption is necessary for the treatment of fatigue in rolling bearings"[10]. The other major improvement was that Weibull works from the assumption that the first crack leads to the failure, however when dealing with fatigue in metals it has been shown that this is not the case. Most cracks never even reach the surface of the specimen or cause the failure. This in turn lead to the assumption that the probability of a fatigue failure is dependent on the above mentioned maximum critical shear stress depth at which the highest and therefore most dangerous stresses are seen and shown below in Equation 3 [11,12].

$$L = A\left(\frac{1}{V}\right)^{\left(\frac{1}{e}\right)}\left(\frac{1}{\tau}\right)^{\left(\frac{c}{e}\right)}(z_o)^{\left(\frac{h}{e}\right)} \approx \frac{1}{S_{\max}^{n}} \qquad (3)$$

2.3 Ioannides-Harris Equation

Ioannides and Harris worked from both Weibull and Lundberg-Palmgren, adding in the fatigue-limiting stress, $\tau_{u,}$ and integrated the life of the elemental stress volume to predict bearing life. As can be seen below, Equation 4 is identical to Equation 3 with the exception of the added fatigue-limiting stress term.

$$L = A\left(\frac{1}{V}\right)^{\left(\frac{1}{e}\right)}\left(\frac{1}{\tau-\tau_u}\right)^{\left(\frac{c}{e}\right)}(z_o)^{\left(\frac{h}{e}\right)} \approx \frac{1}{S_{\max}^{n}} \qquad (4)$$

If the term τ_u is equal to zero then Equation 3 and 4 are identical. The major contribution of their combined work is the theory that bearings have a fatigue limiting stress, meaning a critical stress less than the fatigue stress would produce a bearing with an infinite life. The Ioannides-Harris relation has been shown to significantly over predict bearing life when comparing the calculated results to field-testing [11,12].

2.4 Zaretsky Equations

As shown above, both Weibull and Lundberg-Palmgren relate the critical shear stress-life exponent, c, to the Weibull slope, e. Therefore, ratio of c/e becomes a separate term in and of itself, which is commonly referred to as the effective critical shear stress-life exponent. This suggests that bearing life dispersions are dependent on the critical shear stress life exponent. Zaretsky's research found that the for a wide-ranging set of conditions, most stress-life exponents vary between 6 and 12. He therefore, concluded that the exponent was independent of the scatter in the data. Zaretsky's Equation shown below, varies from Lundberg-Palmgren in the exponent of τ and it does not take into account the critical stress depth.

$$L = A\left(\frac{1}{V}\right)^{\left(\frac{1}{e}\right)}\left(\frac{1}{\tau}\right)^{c} \approx \frac{1}{S_{\max}^{n}} \qquad (5)$$

Conceptually Zaretsky makes the survival probability dependent on the local stress in the stressed volume, like Ioannides-Harris, but differs from Lundberg-Palmgren, which only used one critical stress. Zaretsky also does not propose that there is a fatigue limiting stress, but has stated in numerous papers that he does not exclude it either [11,12].

2.5 Comparisons

The four theories can be broken down into two subset of analysis. Ioannides-Harris is the same as the Lundberg-Palmgren theories, the only difference being that Ioannides-Harris has a fatigue limiting term. The other subset is Weibull-Zaretsky in which the exponents are slightly different. Lundberg-Palmgren has shown to be an overly conservative life predictor. Ioannides-Harris assumption of a fatigue limiting stress distorts the stress life exponents and therefore over predicts bearing life. Ioannides-Harris and Zaretsky produced statistically similar results. The graphical difference in the results can be seen in Figure 1 below.

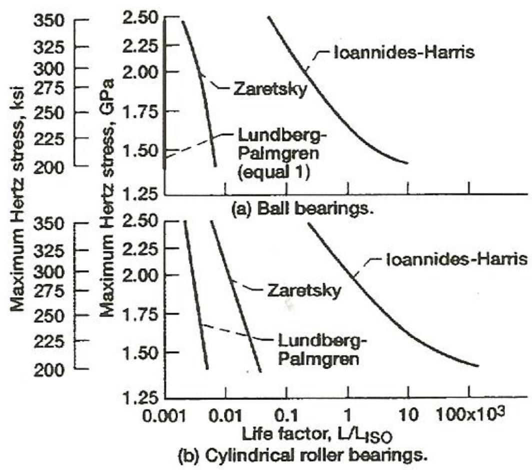

Figure 2: Comparison of Life Theories for rolling element bearings [11]

The major differences in the values comes not only from the different equations but can also be traced back to the n exponent on the S_{max} term. The n term can be pulled out the above mentioned equations with some work and set equal to the other exponents in varying forms depending on which of the four methods used. The results can be seen in Table 1, which shows the maximum hertz stress-life exponent with respect to the Weibull slope. The Weibull slope is shown with three different numbers. The slope usually varies between one and two, with the standard number used being 1.1.

Equations	Weibull Slope	Stress-Life Exponent
		n
Weibull	1.11	11.1
	1.5	8.2
	2	6.2
L-P	1.11	9.0
	1.5	6.7
	2	5.0
I-H	1.11	9.0
	1.5	6.7
	2	5.0
Zaretsky	1.11	10.8
	1.5	10.3
	2	10.0

Table 1: Maximum Hertz Stress-Life Exponents vs. Weibull Slope [11]

The table above shows how all of the relations have a strong dependence on the Weibull slope. If the slope was factored into the equations as is the case with first three methods the load-life exponent decreased rapidly and no longer becomes accurate when compared to actual testing data. The fourth equation does not have the same dependence on the exponent and has a much more consistent maximum Hertz stress-life exponent over the range of Weibull slopes.

The value of the Hertz stress-life exponent is general taken to be 9 for air-processed steels, such as those analyzed by Lundberg and Palmgren. For new and cleaner steels, this number is taken as 12. The results in Table 1 demonstrate that for the 9^{th} power Hertz stress-life exponent Lundberg-Palmgren best predicts life in ball bearings. However, for the newer steels using 12^{th} power relations, Zaretsky's equation best predicts life [11,12].

3. Life Adjustment Factors

As the quality of steels improved after World War II the Lundberg-Palmgren Equations were shown to be under predicting the life of ball bearings by a greater and greater amount. To counter this without having to have a wholesale change in the current standards, which were based on Lundberg-Palmgren, ASME published adjustment factors to account for the differences. The numerous factors have varying effects on the life of a ball bearings life. They include reliability needed, the material hardness, material processing and lubrication. There are many others factors such as operating conditions, operating speed and surface conditions, which exist but are less used. All of the factors can be combined to adjust the predicted life of the bearing. An example of the equation is below.

$$L = A_1 A_2 A_3 A_4 A_X \left(\frac{C}{C_p} \right)^P \qquad (6)$$

L is the predicted life of the bearing, A is the Life Adjustment Factor, C is stress, C_P is the characteristic stress and p being the stress-life exponent [6,7].

3.1 Reliability

Reliability is the simplest of the life adjustment factors. The engineer uses the factor after determining how consistent the bearing must be over its service life. Notice how an increase in reliability from 90% to 96% decreases the factor by almost half. Most bearings are selected using the 90% reliability. See the table below for common life adjustment factors for reliability [7].

Reliability (%)	Factor
90	1
95	0.62
96	0.53
97	0.44
98	0.33
99	0.21

Table 2: Life Adjustment Factors for Reliability [7]

3.2 Processing

The type of material and the way in which it is made is also important in determining the life a bearing. The most commonly used steels are AISI 52100 (90% of all bearing) and M-50 (high-temperature applications). Other steels have been developed and are utilized due to a wider variety of operating conditions, such as high and low temperature and corrosive environments. Other materials, along with 52100 and M-50, are listed below in Table 3 with the corresponding material factors [5,6].

Material	Factor
52100	2
M-1	0.6
M-2	0.6
M-50	2
T-1	0.6
WB-49	0.6

Table 3: Life Adjustment Factors for Material [5]

As can be seen above some of the factors will increase the bearings life, while others will decrease it. This can be traced back to how the material was manufactured.

The first type of manufacturing process was a case-hardening of the steel. This worked, however it did not produce a lot of homogeneity between samples and over the surface of the bearing itself. This was improved by using a through hardening process.

In this process the hardness is achieved by cold working or heat treating, rather than by a strictly surface modification such as carburizing or nitrating [6].

The melting practices are also important. The early techniques were air melting. This lead to undesirable inclusions in the steel such as sulfides, oxides and silicates that can increase local stresses in the sample. These local stresses can reach critical levels under the repeated loading of the bearing and cause a fatigue spall or pit resulting in failure [6].

The latest technique is the vacuum-melting process, which reduces or eliminates the inclusions in the steel as well as any gases and trace elements in the metal. This process results in a much better quality, cleaner, steel. The four main types are vacuum induction melting, consumable-electrode melting, eletroslag melting and vacuum induction melting – vacuum arc remelting. Over the last 30 years, these processes have had the largest impact in improving bearing life. These processes have the same cost as air-melted steels at a substantially higher quality, making them the present day standard in ball bearings [5,6,7].

3.3 Material Hardness

The hardness of a material is also very important. The minimum recommend hardness for a ball bearing is a Rockwell Hardness C of 58. A decrease in the hardness due to high operating temperature or a flaw will cause a faster time to failure. An increase in hardness over 58 will leads to an increase in predicted life. It has been found in laboratory tests that the optimum combination of ball and races with respect to fatigue life is when the balls were one to two points higher than the races. The adjustment factor equation is shown below [5,6,7].

$$A = \left(\frac{R_c}{58}\right)^{10.8} \qquad (7)$$

3.4 Lubrication Factor

The lubrication factor depends on the separation distance of the balls and races. The smaller the separation, the more surface peaks that interact and an increase in the shear stress from the direct metal to metal contact. This leads to a shorter life. With a larger separation distance, the fatigue life is greatly extended. Equation 8 below, shows film parameter of lubrication, Λ

$$\Lambda = \left(\frac{h}{\sqrt{\left(f_r^2 + f_b^2\right)}}\right) \qquad (8)$$

with h = film thickness, f_b & f_r = rms surface of the ball and race respectively. This value can be used in conjunction with Figure 3 to find the lubrication factor. As can be seen in the figure a film parameter greater than three yields little improvement as the separation distance is adequate to separate all but the most extreme peaks on the ball and races [5,6,7].

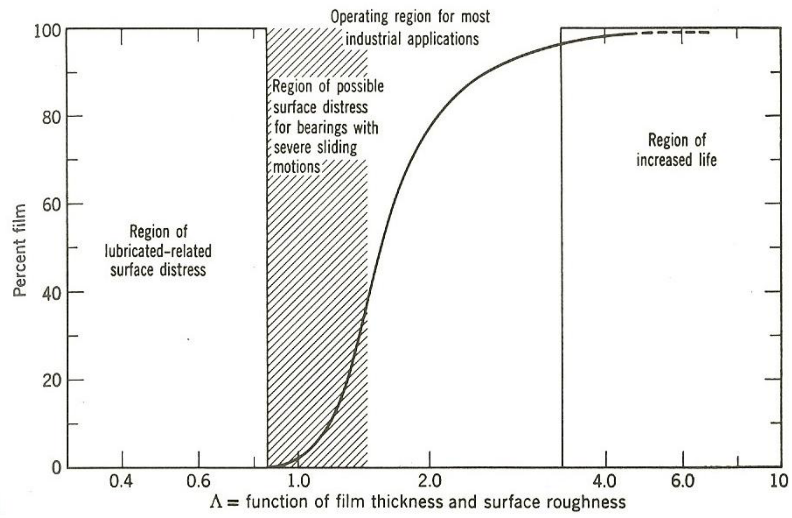

Figure 3: Film Parameter vs. Surface Roughness [7]

4. Open Problems in the Field

There are of course many open problems in this large, but very specific field. The only objectives are to increase the life of a ball bearing and be able to predict exactly when it will fail. Computers have helped increase the accuracy of the calculations, using finite-element analysis to calculate the exact stress on an elemental volume below the surface of a sample. Steel manufactures are always being pressed to make the steels cleaner and cleaner, leading recently to double vacuum induction melting. Engineers and scientists have experimented with making the steels harder to increase the life of the bearing; this is mainly done with better heat treatments and with coatings, such as by the high velocity oxyfuel coating process. Hybrid bearings have also increased in popularity in recent years. A hybrid bearing is one that has different materials for the balls and races. Most of have tests and calculation performed were done using a hot-pressed silicon nitride. In some tests hybrid bearings have exhibited longer fatigue lives than all steel bearings [1,2,6,11,12,14].

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6. Appendix