A New Method of Calculating the Attainable Life and Reliability in Aerospace Bearings

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Abstract—The aviation industry made significant progress improving reliability, efficiency and performance throughout the last decades. Especially aircraft engines and helicopter transmission systems contributed significantly to these improvements. The kerosene consumption decreased by 70% and the CO₂ emissions due to air transport decreased by 30% per passenger kilometer within the last 20 years.

Simultaneously, the flight safety was increased with aircraft engine in-flight-shut-downs as low as 1 ppm and „unscheduled engine removals“ as low as 4 ppm. Flight safety is equal to the reliability of the systems in service. Failure of these systems directly leads to exposure of human life.

Among the most critical aviation systems are aircraft engines including the rolling element bearings which support the rotors. A serious damage to the aircraft engine main shaft bearings during flight requires shut-down of the engine to avoid a further damage escalation subsequently leading to engine fire. Today, it is a requirement for aircraft to operate with one engine shut down. However, each in-flight-engine-shut-down typically is connected with flight diversion or abort and immediate landing. Inflight-shut-downs translate into increased risk for passengers and crew and substantial on cost. Therefore, rolling element bearings for aircraft engines are developed – similar to other aircraft engine components – targeting a reliability of nearly 100% over an operation time of more than 10 000 hours prior to overhaul. To achieve this requirement despite the extreme operating conditions such as high speed and temperatures occurring in gas turbines, special high-performance materials are used for the rolling bearing components which are partially integrated in surrounding engine parts like shafts and housings. These special conditions - deviating from conventional industrial rolling element bearing applications - are currently not sufficiently considered in the standardized method of calculating the bearing life per ISO 281. A new method of calculating the attainable life of rolling elements bearing in aerospace applications is presented. This method considers the special aerospace conditions and materials and thus enables a higher reliability of the theoretical analysis and life prediction.

II. ROLLING ELEMENT BEARING LIFE CALCULATION

A. The “Classical” Rolling Bearing Life Calculation

Today’s calculation method of rolling element bearing life is based on Lundgren and Palmgren [1] theory, published in 1947. Preconditions at that time were constant material properties and the exclusive influence of external loads on the following basic approaches:

i. The failure distribution follows a linear curve in the double-logarithmic Weibull-diagram with a slope of 10/9 for point contact and 9/8 for line contact.

ii. The number of load cycles at the same failure probability is indirect proportional to the contact load by exponent ε: Z ~ P⁻ε

Based on these approaches, the classic life equation for rolling element bearings was developed:

\[ L10 = (C/P)\frac{1}{\varepsilon} \]  \hspace{1cm} (1)

B. The Modified Rolling Bearing Life Calculation

The experience made during the decades after introduction of equation (1) showed that rolling element bearing operating conditions, especially the lubrication conditions, have significant influence on the bearing service life, leading to the modified bearing lifetime calculation [2], [3].

Further research in the 1980’s demonstrated that rolling element bearings – similar to other machine elements – can achieve endurance life under certain operating conditions [3], [4]. Based on that, the standardized calculation method per ISO 281 was developed [5, 6], which is currently used for industrial rolling bearing applications.

This method assumes that

i. classical rolling bearing steels are used for all over-rolled bearing components,

ii. no reduction of the material strength due to increased temperatures,
iii. lifetime is limited by a spalling of one of the raceways, i.e. a raceway spalling occurs prior to a potential spalling of the rolling element.

The high requirements for reliability and service life for aerospace bearing applications translated – starting already in 1950’s – into developments of steels with increased hot hardness and high temperature strength and fracture toughness, such as AISI M50 and AISI M50NiL. The properties of these bearing materials were considered in the aerospace calculation methods developed in the 1990’s [7, 8], which are based on the equations derived in [2, 3].

A further performance and reliability increase can be achieved by using special aerospace bearing materials such as 32CDV13, Cronidur30 [9], balls and rollers made of ceramic material [10, 11] and special heat treatment procedures such as Duplex Hardening [12]. Further high-strength aerospace bearing materials are currently under development, expected to be introduced within the next 5 to 10 years.

The trend towards ultra-lightweight aircraft requires integration of bearing rings with other aircraft engine parts such as housings, elastic fixations, shafts, sealings, etc. This often translates into different materials used for the components within one bearing, i.e. different treatments for the inner raceway, outer raceway and rollers or balls are required.

For a realistic calculation of the bearing life or its reliability, the various materials and their combination for inner raceway, outer raceway and rollers or balls must be considered.

The “new method of calculating the attainable life and reliability in aerospace bearings as described in the following sections is based on the fundamental Lundberg-Palmgren theory. The new method applies adjustment coefficients to consider the operating conditions and reliability based on ISO 281 [5], but refined to consider the special requirements of aerospace bearing applications. This includes the individual consideration of the components inner ring raceway, outer ring raceway and particularly the ball/roller set, the high rotational speed, the elevated operating temperatures, the manufacturing quality, the lubricant properties, the system reliability, and much else.

III. CALCULATION OF THE MODIFIED ATTAINABLE LIFE IN AEROSPACE BEARING APPLICATIONS

The modified life of the entire bearing is calculated by the modified life of the components inner ring raceway, outer ring raceway and set of rolling elements per equation (2)

\[ L10_{AC} = (L10_{ACi} - \epsilon + L10_{ACe} - \epsilon + L10_{ACr} - \epsilon) \frac{1}{\epsilon} \]  

(2)

The modified life of the components inner ring raceway, outer ring raceway and set of rolling elements is calculated from the basic rating life multiplied with the life modification factor \( a_{AC} \) of the corresponding component.

\[ L10_{ACi} = L10_{i} \cdot a_{ACi} \]  

(3.1)

\[ L10_{ACe} = L10_{e} \cdot a_{ACe} \]  

(3.2)

\[ L10_{ACr} = L10_{r} \cdot a_{ACr} \]  

(3.3)

The basic rating life of the components inner ring raceway, outer ring raceway and rolling element is determined by using systematically the Lundgren and Palmgren-theory [1] in correlation with ISO 281:2007, ISO T/R 1281-1:2008 and ISO/TS 16281:2008 [5, 6, 13].

For that, the basic dynamic load rating of the entire bearing \( C \) is dissolved into the prime element basic dynamic load rating on the inner and outer ring raceway contacts, respectively, \( Q_{ci/en} \). This is the contact load acting on a contact element with a diameter of 1 mm attaining 10⁶ load cycles or rolling contacts with 90 % reliability.

Point Contact:

\[ Q_{ci/en} = K_e \cdot (1 + \gamma) \frac{2}{2R_{i/en} - D_w} \cdot \frac{f_{Dw}}{D_w} \cdot D_{w}^{3/10} \cdot 0.5^{1/3} \]  

(4)

Line Contact:

\[ Q_{ci/en} = K_e \cdot (1 + \gamma) \frac{35}{2R_{i/en} - D_w} \cdot L_{w}^{7/9} \cdot D_{w}^{35/27} \cdot 0.5^{1/4} \]  

(5)

The upper signs refer to the rolling element / inner raceway contact and the lower signs refer to the rolling element / outer raceway contact.

Based on the prime element basic dynamic load rating \( Q_{ci/en} \) the elementary basic dynamic load rating for a specific raceway contact diameter per 10⁶ load cycles is determined per equations (6) and (7).

Point Contact:

\[ Q_{ci/ci} = K_e \cdot \frac{(y_{ci} / \cos \alpha)}{2^{1/4}} \cdot (1 + \gamma) \cdot D_{w}^{29/35} \cdot L_{w}^{7/9} \cdot 0.5^{1/4} \]  

(6)

Line Contact:

\[ Q_{ci/ci} = K_e \cdot \frac{(y_{ci} / \cos \alpha)}{2^{1/4}} \cdot (1 + \gamma) \cdot D_{w}^{29/35} \cdot L_{w}^{7/9} \cdot 0.5^{1/4} \]  

(7)

Again, the upper signs refer to the rolling element / inner raceway contact and the lower signs refer to the rolling element / outer raceway contact.

The basic life of the inner raceway and outer raceway contacts per 10⁶ load cycles is determined from the loading of the individual contacts of the inner ring raceway \( F_{in} \) and outer ring raceway \( F_{en} \), respectively (equation (8)).

\[ L10_{i/e} = \left[ z \cdot \left( \frac{1}{2} \sum_{n=1}^{N} \frac{F_{i/en}}{Q_{ci/en}} e^{-1/\epsilon'} \right) \right]^{1/\epsilon'} \]  

(8)

The exponent \( \epsilon' \) equals 1, if each volume element of the raceway experiences all loads \( F_{ci/en} \), i.e. the load rotates relative to the raceway.

If the ring stands still relative to the load, the entire volume is not loaded uniformly. Therefore, the average loading is compiled from the different loaded volume elements. For this case, \( \epsilon' = \epsilon = 10/9 \) for point contact and \( \epsilon' = \epsilon = 9/8 \) for line contact, applies.

Also based on prime element basic dynamic load rating
$Q_{w/e}$ (equations (4) and (5)), the elementary basic dynamic load rating per $10^6$ load cycles for the rolling elements is determined per equations (9) and (10).

**Point Contact:**

$$Q_{w/e} = K_e \cdot (1 \mp \gamma) \frac{165}{100} \cdot \left( \frac{2R_{i/e}}{K_{i/e} \cdot D_w} \right)^{41/100} \cdot f_{Dw} \cdot 0.5^{1/3} \quad (9)$$

**Line Contact:**

$$Q_{w/e} = K_e \cdot (1 \mp \gamma) \frac{35}{27} \cdot \frac{2}{D_w} \cdot D_w^{29} \cdot 0.5^{1/4} \quad (10)$$

Again, the upper signs refer to the rolling element / inner raceway contact and the lower signs refer to the rolling element / outer raceway contact.

One rolling element (ball or roller) is loaded during its 360° orbit by all contact loads acting on the inner and outer ring raceway contacts. Analogous to equation (8), the basic reference life for one rolling element set $L_{10}$, is yielded by:

$$L_{10r} = \left\{ \left[ \frac{1}{2} \sum_{n=1}^{\infty} \left( \frac{P_{tu}}{Q_{cri}} \right) \right]^{1/e} + \frac{1}{2} \sum_{n=1}^{\infty} \left( \frac{P_{tu}}{Q_{cri}} \right) \right\}^{-1/e} \quad (11)$$

It is known from comprehensive rig testing and practical field experience, that the calculated basic life of rolling element bearings per equation (1) is exceeded multiple times in real applications and endurance strength can be achieved depending on the true material stressing due to contact load, the lubrication and contamination conditions ([14] to [17]) for a realistic stressing.

The factors $f_d$, $S_{AC}$, and $f_1$ are intensifiers of the nominal contact stress for consideration of specific operating conditions. The method for determination of these factors will be identical for both the point contact and the line contact because the basic life of discrete stressed material in rolling contacts results in minor difference only when calculated either for a point contact or a line contact (Fig. 2).

**A. The Endurance Strength at Operating Temperature $P_{0tu}$**

The endurance strength of bearing steels is reduced at elevated operating temperature depending on their hot hardness. This reduction is considered by constant $K_t$ and equation (13):

$$P_{0tu} = P_{0u} + K_t \cdot (t - 100) \quad (13)$$

$P_{0u}$ is valid for temperatures up to 100 °C, i.e. $K_t$ is applicable only for operating temperatures above 100 °C.

Table 1 depicts values for $P_{0u}$ and $K_t$ applicable for currently used aerospace bearing materials.

| Material                  | $P_{0u}$ (MPa) | $K_t$ (SigmaT) |
|---------------------------|----------------|----------------|
| 100Cr6 (AMS 6440/6444)    | 1500           | -5.0           |
| M50 (AMS 6491)            | 1800           | -2.0           |
| M50 DH                    | 1900           | -2.0           |
| M50NiL (AMS 6278)         | 1900           | -2.0           |
| M50NiL DH                 | 2000           | -2.0           |
| 440C                      | 1400           | -5.0           |
| Cronidur 30               | 1900           | -4.0           |
| Pyrowear 53               | 1800           | -2.0           |
| RBD                       | 1800           | -2.0           |
| 32CDV13 (AMS 6481)        | 1800           | -2.0           |
| SAE 9310                  | 1500           | -5.0           |
| Ceramics (Si$_3$N$_4$)    | 2500           | -1.0           |

**B. The Influence of Lubricating Film Thickness**

ISO 281 considers the lubricating conditions by means of the viscosity ratio $\kappa$, the relation of operating viscosity of the lubricant $\nu$ and the velocity-dependent reference viscosity $\nu_1$. As this ratio is only influenced by the rotational speed and the viscosity of the lubricant and assumes...
standardized values for curvature ratios, load and surface topography, the true lubricating conditions are represented incompletely.

A more realistic evaluation of the lubricating conditions can be achieved by use of the film thickness parameter $\lambda$, i.e. the ratio of the minimum film thickness [18, 19, 20] and the composite roughness of ring raceway and rolling element per [7].

Referring to $\lambda$, the factor $f_\lambda$ having inverse effect on the stress in equation (12) can be expressed by the following function:

$$f_\lambda = \left( 1 - \frac{k_3}{(\lambda + k_3)} \right)^4$$

(14)

The slope shown in Fig. 3 is determined by $k_3$ which considers the material behavior of mixed-friction conditions for typical aerospace bearing materials.

The values for $k_3$ currently in use for aerospace bearing materials are shown in Table II.

**TABLE II: VALUES FOR $k_3$ FOR COMMON AEROSPACE BEARING MATERIALS**

| Material | $k_3$ |
|----------|------|
| 100Cr6 (AMS 6440/6444) | 0.2 |
| M50 (AMS 6491) | 0.1 |
| M50 DH | 0.06 |
| M50NiL (AMS 6278) | 0.1 |
| M50NiL DH | 0.06 |
| 440C | 0.2 |
| Cronidur 30 | 0.1 |
| Pyrowear 53 | 0.1 |
| RBD | 0.1 |
| 32CDV13 (AMS 6481) | 0.1 |
| SAE 9310 | 0.2 |
| Ceramics (Si$_3$N$_4$) | 0.05 |

**C. The Influence of the Lubricating System Cleanliness**

Per ISO 281, the cleanliness factor for classical rolling bearing steel 100Cr6 (SAE 52100) is exponentiated by 1/3 for point contact (ball bearings) and by 0.4 for line contact (roller bearings).

Likewise, for the new calculation method presented herein, the cleanliness level $S_{AC}$ is exponentiated by $e_c$, the applicable individual aerospace bearing material exponent. Note that $e_c$ is multiplied with a factor of 1.2 for line contact and 1 for point contact. The exponent $e_c$ considers therefore the sensitivities of the various aerospace bearing materials on hard particle contamination.

Table III depicts values for $e_c$ for the various aerospace bearing materials.

**TABLE III: EXPOERNT $e_c$ FOR VARIOUS AEROSPACE BEARING MATERIALS**

| Material | $e_c$ |
|----------|------|
| 100Cr6 (AMS 6440/6444) | 0.33 |
| M50 (AMS 6491) | 0.33 |
| M50 DH | 0.3 |
| M50NiL (AMS 6278) | 0.33 |
| M50NiL DH | 0.3 |
| 440C | 0.33 |
| Cronidur 30 | 0.3 |
| Pyrowear 53 | 0.33 |
| RBD | 0.33 |
| 32CDV13 (AMS 6481) | 0.33 |
| SAE 9310 | 0.33 |
| Ceramics (Si$_3$N$_4$) | 0.3 |

The cleanliness level is determined per Fig. 4, depending on filtration rate, filter quality, bearing configuration and rolling element size.

**D. The Influence of Tangential Stresses**

The influence of tangential stresses (friction) is considered by factor $f_1$. The factor is evaluated depending on friction intensity of each bearing configuration, e.g. for axial bearings.

The material stressing might be further affected by severe operational effects resulting in particular additive tangential stress due to:

i. interference fit and / or high rotational speed
ii. ring bending of elastically supported races
iii. slip effects (spin-to-roll, $P*V$)
iv. ball excursion of combined loaded ball bearings

These effects together with considerations of the according additional stresses on life have been described and evaluated in [20], [21], [22], [23] for instance.
factor $f_1$ can be estimated accordingly or on the basis of advanced calculation programs.

$$f_1 = 1 - \frac{\text{Additional Stress}}{\text{Nominal Stress}}$$  \hfill (15)

Without consideration of these specific operation effects the factor $f_1=1$ is applied.

IV. THE RELIABILITY FACTOR $a_1$ (FAILURE PROBABILITY)

As discussed before, a reliability of almost 100% is expected from modern aircraft engines. Therefore, a rolling bearing life calculation based on the typical failure probability of 10% is not acceptable.

Currently aircraft operators require one “Unscheduled Engine Removal” (UER) per one million flight hours. As multiple rolling element bearings and other critical components act together with individual reliabilities, the single component reliability must be significantly higher. Therefore, the typical reliability requirement is 99.8% per 10000 operating hours for rolling element bearings in flight critical application.

Per ISO 281, the factor $a_1$ allows for failure probabilities <10%, i.e. reliabilities >90%.

According to the reliability definition discussed, for a required reliability of 99.8% ($L_{0.2AC}$) applies:

$$L_{0.2AC} = L_{10AC} \cdot 0.12$$  \hfill (16)

For a typical operating time of 10000 hours with a reliability of 99.8%, a calculated bearing life $L_{10AC}$ of more than 80000 hours would be required.

Derived from the required system reliability with systems having multiple bearings and other components, a single bearing reliability of almost 100% shall be targeted in order to avoid extreme over-dimensioning with detrimental consequences for bearing operability and performance, e.g. for high-speed bearings.

Experimental investigation results from Tallian [4] show that for reliabilities of more than 99.95% no bearing failures occur.

Based on ISO 281, 100% reliability is achieved if

$$L \geq 0.077 \cdot L_{10}$$  \hfill (17)

| RELIABILITY AND RELIABILITY FACTOR $a_1$ | $L_{0.2AC}$ | $a_1$ |
|-----------------------------------------|------------|-------|
| 90                                      | $L_{10AC}$ | 1     |
| 95                                      | $L_{2AC}$  | 1.12  |
| 96                                      | $L_{4AC}$  | 1.26  |
| 97                                      | $L_{6AC}$  | 1.37  |
| 99                                      | $L_{8AC}$  | 1.50  |
| 99.2                                   | $L_{10AC}$ | 1.22  |
| 99.4                                   | $L_{12AC}$ | 1.19  |
| 99.6                                   | $L_{14AC}$ | 1.16  |
| 99.8                                   | $L_{16AC}$ | 1.12  |
| 99.9                                   | $L_{18AC}$ | 0.93  |
| 99.92                                  | $L_{20AC}$ | 0.878 |
| 99.94                                  | $L_{22AC}$ | 0.080 |
| 99.95                                  | $L_{0.05AC}$ | 0.077 |

V. SUMMARY AND CONCLUSION

The presented method of calculating the attainable bearing life for aerospace applications considers the properties of the individual materials used for inner ring raceway, outer ring raceway and rolling element set under the applicable operating conditions, i.e. lubricating, friction and cleanliness conditions.

The theoretical bearing life is based on the theoretical calculated life of the components inner ring raceway, outer ring raceway and rolling element set.

The reference or basic life of the individual bearing components is calculated with reference to [5, 6 and 13]. An adjustment factor $a_{AC}$ is introduced which allows for consideration of the properties of each individual bearing component material and the lubricating, friction and cleanliness conditions. Based on the reference or basic bearing life and the factor $a_{AC}$ for the individual bearing components, the modified component life and subsequently the modified bearing life is calculated. The factor $a_{AC}$ is defined similar to $a_{SD}$ in [5].

The method presented in this article allows for incorporation of properties of future bearing materials.

NOMENCLATURE

$L_{10}$ = basic rating life with a failure probability of 10%, in million revolutions

$C$ = basic dynamic radial load rating or the load, under which 10% of a bearing collective have failed after one million revolutions

$P$ = dynamic equivalent radial load

$\varepsilon$ = life exponent, 3 for point contact and 10/3 for line contact

$L_{10AC}$ = modified life of the entire bearing with a failure probability of 10%

$L_{10ACI}$ = modified life of the inner ring raceway with a failure probability of 10%

$L_{10ACe}$ = modified life of the outer ring raceway with a failure probability of 10%

$L_{10ACr}$ = modified life of the rolling element set with a failure probability of 10%

$e = \text{exponent for failure distribution according to Weibull}$

$K_c = \text{constant for point contact: } 121; \text{approximated from } 98.1, b_m = 1.3 \text{ and reduction factor } \lambda = 0.95. \text{Constant for line contact: } 564; \text{approximated from } 551.1, b_m = 1.1 \text{ and reduction factor } \lambda = 0.93 \text{ (the reduction factor of 0.93 instead of 0.83 is applied under the precondition that the determination of the pressure distribution across the roller length per [13] is calculated with highest precision).}$

$\gamma = \text{auxiliary parameter: } D_w \cos \alpha / D_{pw}$ ($D_w$ = rolling element diameter, $\alpha$ = axial contact angle, $D_{pw}$ = pitch diameter)

$R_{iw}$ = curvature radius for inner/outer raceway

$f_{aw}$: from $D_w \cdot 1.8$ for $D_w \leq 25.4$ or from $3.647 \cdot D_w^{1.4}$ for $D_w > 25.4$, see [5]

$l_{eff}$ = effective roller length

$a_{AC}$ = life factor
$P_{0\text{in}}$ = Endurance strength as Hertzian stress at operating temperature

$P_0$ = maximum Hertzian stress at the contact element for the inner ring raceway or outer ring raceway at operating condition

$f_h$ = Stress factor for consideration of insufficient lubricant separation between the contact elements

$f_{d1}$ = Friction factor

$\Delta C_L$ = Cleanliness level

$e_c$ = Contamination exponent

$e_{L}$ = Stress-Life-Exponent, 9 for point contact, 8 for line contact

$\lambda$ = film thickness parameter, ratio of minimum calculated film thickness and the composite roughness of ring raceway and rolling element

$Q_{\text{ideal}}$ = Contact load acting on a contact element with a diameter of 1 mm attaining 10$^6$ load cycles respectively over-rolling contacts with 90 % reliability

$Q_{\text{die}}$ = Load capacity on the inner or outer contact for any contact diameter attaining 10$^6$ load cycles with 90% reliability

$Q_{\text{wie}}$ = Rolling element load capacity on the inner or outer contact attaining 10$^6$ load cycles with 90% reliability.

**REFERENCES**

[1] Lundgren, G. and Palmgren, A., “Dynamic Capacity of Rolling Bearings,” Acta Polytechnica: Mechanical Engineering Series, 1, 1947.

[2] ISO R281, “Rolling Bearings – Methods of Evaluating Dynamic Load Ratings,” 1977.

[3] Brändlein, J., Eschmann, P., Hasborgen, L. and Weigard, K. „Die Wälzlagerpraxis,” Vereinigte Fachverlage GmbH, Mainz, 2002.

[4] Tallian, T., “Weibull Distribution of Rolling Contact Fatigue Life and Deviations Therefrom,” ASLE Transact. 5 No. 1 1962.

[5] ISO 281:2007, “Rolling Bearings – Dynamic Load Ratings and Rating Life,” 2007.

[6] ISO/TR 1281-1:2008, “Rolling Bearings – Explanatory Notes on ISO 281,” 2008.

[7] FAG Kugelfischer, “A Practical Method of Calculating the Attainable Life in Aerospace Bearing Applications,” FAG Publ. No. FL40134EA, 1989.

[8] Ebert, F. and Poulin, P., “The Effect of Cleanliness on the Attainable Bearing Life in Aerospace Applications,” Tribology Transactions, vol. 38, 1995.

[9] Streit, E., Trojahn, W., Chin, H.A., and Ehlerl, D., “Progress in Bearing Performance of Advanced Nitrogen Alloysed Stainless Steel Conidur 30,” Mat. wiss. u. Werkstoffech., vol. 30, pp. 605-611, 1999.

[10] Ebert, F.-J., “Performance of Silicon Nitride (Si3N4) Components in Aerospace Bearing Applications,” The American Society of Mechanical Engineers, Proceedings of the Gas Turbine and Aeroengine Congress and Exhibition, June 11–14, Brussels, 90-GT-166, 1990.

[11] Gloeckner, P., Martin, M., and Fleurus, M., “Comparison of Power Losses and Temperatures between an All-Steel and a Direct Outer Ring Cooled, Hybrid 133 mm Bore Ball Bearing at Very High Speeds,” Tribology Transactions, 60, pp. 1148–1158, 2017.

[12] Streit, E., Brock, J. and Poulin, P., “Performance Evaluation of “Duplex Hardened” Bearings for Advanced Turbine Engine Applications,” Journal of ASTM International, vol. 3, No. 4, West Conshohocken, 2006.

[13] ISO/TS 16281-1:2008, “Rolling Bearings – Methods for calculating the modified reference rating life for universally loaded bearings,” 2008.

[14] Loroesch H.-K., „Lebensdauer und Dauerfestigkeit von Wälzlagern,” VDI-Report, 549, pp. 109-127, 1985.

[15] Loroesch H.-K., “The life of the rolling bearing under varying loads and environmental conditions,” Schweinfurt, Germany: FAG Kugelfischer, FAG Publication No. EA1981, vol. 1, 17 – 23, 1981.

[16] Loroesch, H.-K., „Einfluss von festen Verunreinigungen auf die Lebensdauer von Wälzlagern,” Antriebstechnik, 23 (10), pp. 63-69, 1984.

[17] Loroesch, H.-K., “Effects of unfavourable environmental conditions on the service life of jet engine and helicopter bearings,” 60th AGARD meeting, San Antonio, Texas, 394, 1985.

[18] Hamrock, B.J. and Dowson, D., “Isothermal Elastohydrodynamic Lubrication of Point Contacts,” ASME J. Lubr. Technol. 99, 1977.

[19] Dowson, D. and Higginson, G.R., “The Isothermal Lubrication of Cylinders,” Proceedings of Institution of Mechanical Engineers, ASME Transaction 8, 1968.

[20] Gloeckner, P., Sebald, W. and Bakolas, V., “An Approach to Understanding Micro Spalling in High Speed Ball Bearings Using a Thermal Elasohydrodynamic Model,” Tribology Transactions, 52, 534-543, 2009.

[21] Coe, H. H., and Zaretsky, E. V., “Effect of Interference Fit on Roller Bearing Fatigue Life,” NASA Technical Memorandum 87165, 1986.

[22] Broszeit, E., and Zvirin, O., “Internal Stresses and Their Influence on Material Stresses in Hertzian Contacts – Calculation With Different Stress Hypotheses,” Journal of Tribology, vol. 108, 1986.

[23] Jones, A. B., “A General Theory for Elastically Constrained Ball and Radial Roller Bearings under Arbitrary Load and Speed Conditions,” ASME J. Basic Eng. 309 – 320, 1960.