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A case study on relation between roughness, lubrication and fatigue life of rolling bearings

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Abstract. A spherical roller bearing under high radial loading, constant speed and imposed roughness for the contacting surfaces was chosen as case study. Different lubrication regimes were obtained by varying oil viscosity through the operating temperature. For bearings with especially machined contacting surfaces, λ-ratio is firstly determined and its value is used to estimate the particular value of the lubrication parameter κ. Using the λ-ratio approach the paper reveals the relationship between roughness amplitude and the modified rating life of rolling bearings. The roughness values corresponding to good manufacturing practice are possible to be determined for each particular case. Three groups of random Gaussian roughness were generated with the same values for the *Ra* parameter as used in the modified lives investigations. For medium and especially high radial loads, the contacts between rough surfaces develop, inside the shallow layer, von Mises equivalent stresses higher than the fatigue limit stress. For condition of lack of lubricant or starved lubrication, these findings explain the initiation of the rolling contact fatigue in the shallow layer, close to contacting surfaces.

1. Introduction

Methods of calculating the basic dynamic load rating of rolling bearings manufactured from *high quality* hardened steel are based on conventional design regarding *profiles* of rolling contact surfaces and *good manufacturing practice* regarding roughness of contacting surfaces (International Standard ISO 281:2007 [1]).

It is well known the competition between surface and subsurface origin of contacting surfaces fatigue [2]. Surface roughness may significantly affect the lubrication performance, film thickness and contact severity by creating, in the shallow layer close to surface, high values for von Mises stresses. The severity of the asperity contact appears to be vital to lubrication breakdown and surface failure.

A numerical simulation program has been developed in order to evaluate the pressures distributions due to various surface roughness under low, medium and high loading. Also the in-depth pressure distribution and the film thickness were evaluated.

2. EHL film thickness, roughness and fatigue life

In the EHL process of rolling bearings, the microgeometry of the surface has two main effects: a slight increase in the local clearances due to elastic deformation of surface micro-geometry, and the generation of pressure fluctuations that might have an effect on life.

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2.1. EHL film thickness

Relative motion between two contacting surfaces causes a hydrodynamic lubricating film to be generated which modifies the pressure distribution to a certain extent. The opposing surfaces within the contact are almost parallel and planar and film thickness is often described in this regional film by the central film thickness h_c. The lubricant experiences an abrupt rise in viscosity as it enters the contact followed by an equally sharp decline to ambient viscosity level at the exit of the contact. To maintain continuity of flow and compensate for the loss of lubricant viscosity at the contact exit, a constriction is formed close to the exit. The effect is known as the *horse-shoe* constriction.

The minimum film thickness h_0 is found at the constriction. For the case of point contact and a thick EHL film, that means 1µm or more, the minimum film thickness is found at both ends of the constriction and at this location the film thickness is only about 60% of its central value, [3]. For very thin EHL films, that mean less than 50 nm, the film thickness does barely show any side constriction.

The minimum film thickness is a very important parameter since it controls the likelihood of asperity interaction between the two surfaces. Asperity interaction occurs when the minimum film thickness becomes unusually low, or when the surface roughness is high. Rougher surfaces combined with poor lubrication conditions would generally result in shorter lives.

2.2. Pressure distribution

A large pressure peak is generated next to the constrictive on the upstream side, and downstream the pressure rapidly declines to less than dry Hertzian value.

2.3. Lambda ratio approach

Local film variation as a function of local surface roughness is usually characterized by so called *lambda ratio* proposed by Tallian *et al.* [4] and defined as the ratio of the minimum film thickness h_0 to the composite roughness of two surfaces in contact:

$$\lambda = \frac{h_0}{\sqrt{\sigma_1^2 + \sigma_2^2}} \tag{1}$$

where σ_1, σ_2 are RMS surface roughness of the raceway and rolling element, respectively.

Tallian *et al.* [4], Tallian [5] and Popinceanu *et al.* [6] performed electrical rezistance measurements, on two disks rigs but also on rolling bearings, to correlate percentage of metal-to-metal contact to the calculated penetration of roughness's asperities into the lubrication film and performance of the rolling contact.

Durability tests carried out on:

- tapered roller bearings (Skurka [7], and Danner [8]),
- a number of 2025 ball bearings grouped in 81 test lots (Popinceanu *et al.* [6, 9] figure 1),
- a number of 500 cylindrical roller bearings grouped in 20 test lots (Creţu [10], Balan and Creţu [11] figure 2),

brought a solid confirmation from bearing manufacturers regarding the relation between the lubrication quality expressed by λ ratio and bearings rating life.

The life tests carried out on a number of 2300 tapered roller bearings grouped in 93 test lots (Moyer and Bahney [12]) pointed out the following important features: (a) – the improved finishes do not significantly affect life when bearings were operated at large λ ratios; (b) – at low λ ratios, the enhanced–finish bearings have greater fatigue lives than the standard–finish bearings, figure 3.

Using the equations of partial EHL theory, Liu *et al.* [13] evaluated the available works performed in the field of endurance tests of rolling bearings and found a quite good agreement between the rating lives, even with their specific spreading, and the prediction of the λ ratio model, figure 4.

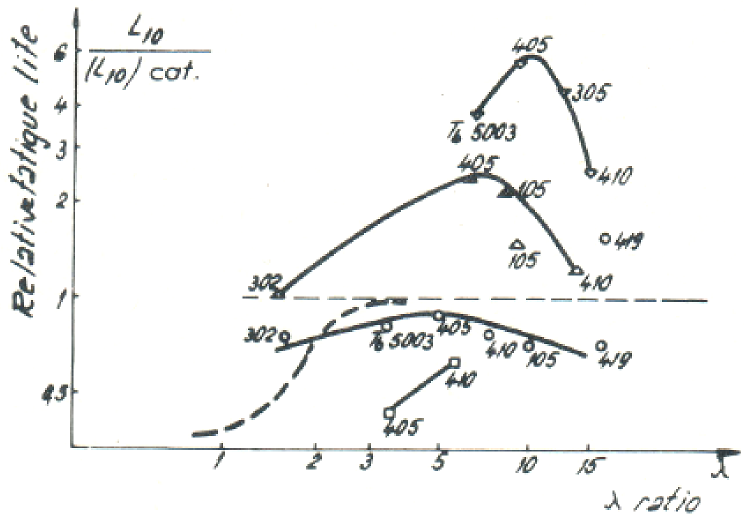

Figure 1. L_{10} life vs. λ-ratio (ball bearings endurance tests, Popinceanu *et al.* [9]).

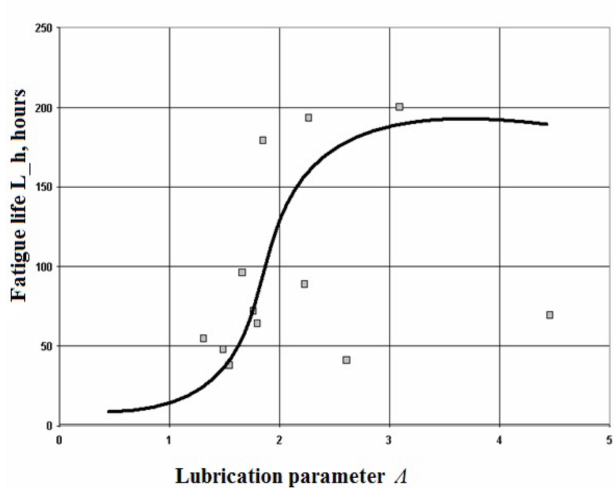

Figure 2. L_{10} life vs. λ-ratio, (cylindrical roller bearings endurance tests, Creţu and Balan [10,11]).

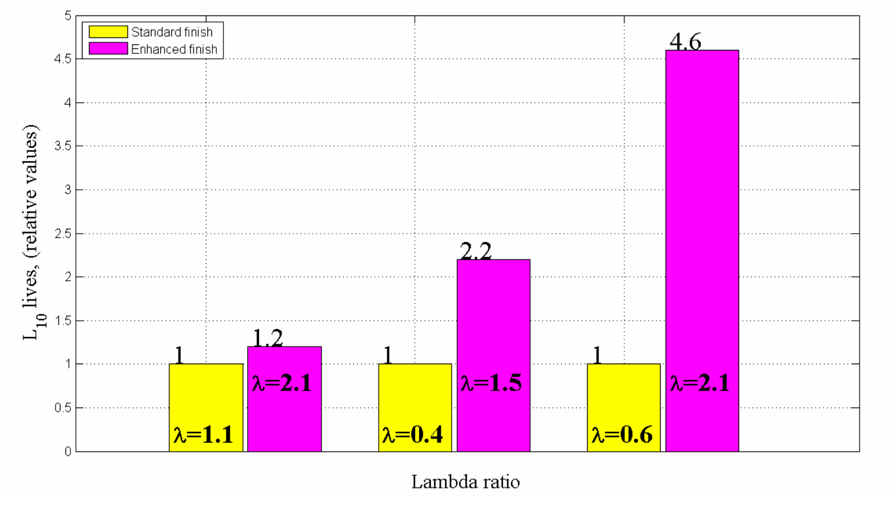

Figure 3. Effect of enhanced-finish on L_{10} life (tapered roller bearings tests, Moyer and Bahney [12]).

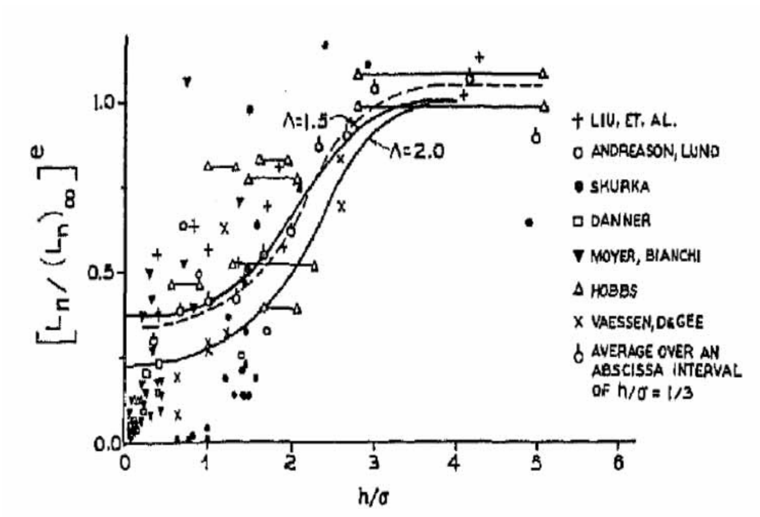

Figure 4. Experimental L_{10} rating lives vs. λ-ratio (Liu *et al.*[13]).

To evaluate the effect of λ ratio on the film thickness and lubrication quality of common rolling bearings, Tedric Harris [2] proposed the diagram presented in figure 5, where the ordinate *percent film* is a measure of time during which the contacting surfaces are fully separated by the oil film.

The mean curve in figure 6 is frequently used to estimate the effect of λ ratio on bearings rating life:

- if $\lambda \geq 4$, fatigue life can be expected to exceed standard $\boldsymbol{L_{10}}$ estimates by at least 100%;
- if $\lambda < 1$, the rolling bearing is working in the mixed lubrication regime and will not attain calculated life.

A critical review of the subject regarding the λ-ratio as a lubrication quality indicator was done by Cann *et al*. [16].

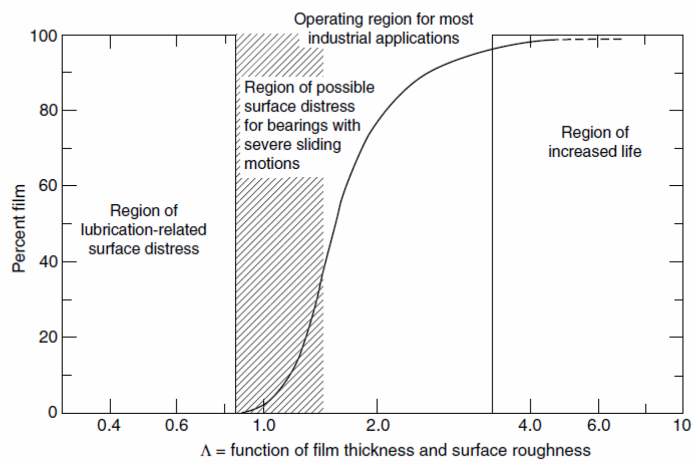

Figure 5. The dependency percent film – λ ratio, (from Harris and Kotzalas [2]).

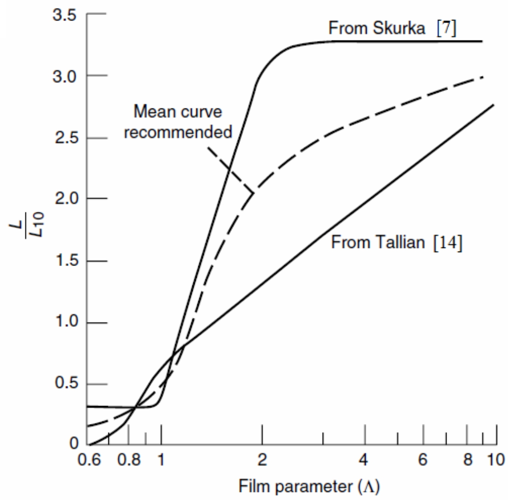

Figure 6. Lubrication–life factor vs. λ–ratio (from Harris and Kotzalas [2]).

2.4. The κ parameter approach

2.4.1. Viscosity ratio and film parameter approach. The value of the basic dynamic load rating *C* reported in bearing catalogues and the computing algorithm presented in ISO-281 are frequently used for theoretical estimation of the modified rating life, L_{10m}:

$$L_{10m} = a_{ISO} \cdot L_{10} = a_{ISO} \cdot \left(\frac{C}{F}\right)^{10/3}. \quad (2)$$

The modification factor, a_{ISO} , was derived from the following equation:

$$a_{ISO} = f\left(\frac{e_c \cdot C_u}{P}, \kappa\right) \quad (3)$$

where: the function f is presented in ISO 281:2007, (equations (34) … (36) [1]), C_u is the fatigue load limit, and e_c is the contamination factor.

According ISO 281:2007, for mineral oil lubrication and bearing raceway surfaces machined with *good manufacturing quality*, the condition of lubricant separation is described by a complex lubrication parameter κ, defined as the ratio of the actual kinematic viscosity ν to the reference kinematic viscosity ν_1 (Ioannides E *et al*. [17]):

$$\kappa = \frac{\nu}{\nu_1}. \quad (4)$$

The parameter λ-ratio depends on two variables only: the film thickness and roughness, while κ, being only indirectly connected to film thickness, allows for more flexibility as the introduction of other rolling bearing-specific life related safety factors as starvation and inlet shear heating effects. Morales-Espejel *et al.* [17] shown that the current definition of κ (ISO 281 [1]) still lacks the consideration of several effects as: a modern film thickness equation and roughness deformation.

For a more detailed estimation of the κ value, e.g. for especially machined surfaces λ is firstly evaluated as the ratio between the minimum thickness of the EHD oil film, $h_{EHD\min}$, formed between the most loaded roller and the composed roughness of the involved surfaces. This is further converted to κ forcing the approximation:

$$\kappa = \lambda^{1.3}. \quad (5)$$

3. Role of roughness on modified rating life L_{10m}

3.1. Spherical roller bearing and operating conditions

The comparative study was conducted on 24038 spherical roller bearing with main catalogue data: inner diameter d = 190 mm, outer diameter D = 290 mm, width B = 100 mm, inner ring chamfer radius r_{ch_i} = 3 mm, basic dynamic load rating C_r = 978 kN, basic static load rating C_0 =1800 kN, fatigue load limit C_u = 163 kN, reference speed n = 1300 rpm, limiting speed $n_{\lim}$ = 1500 rpm.

The concentrated contacts achieved inside the bearing are described by the following geometrical data:

- for spherical roller: roller diameter D_w = 24.5 mm, radius of roller profile R_{w2} = 133 mm, roller length L_w = 39.4 mm, roller end chamfer R_{w_ch} = 1.0 mm, number of spherical rollers Z_w = 2 × 26 and contact angle $\alpha = 11.4^0$;
- curvature radii of the inner raceway: in transverse plane $r_{ci\_1} = 111$ mm, and in axial plane $r_{ci\_2} = 136$ mm;
- curvature radii of the spherical outer raceway: $r_{co\_1} = r_{co\_2} = 135$ mm.

A value of 300 kN has been adopted as the radial force supported by rolling bearing.

Lubrication: ISO VG32 oil with external cooling and a 10 microns filter (contamination factor e_C = 0.8).

For smooth surfaces, the pressures distribution achieved between the most loaded roller and inner raceways is presented in figure 7.

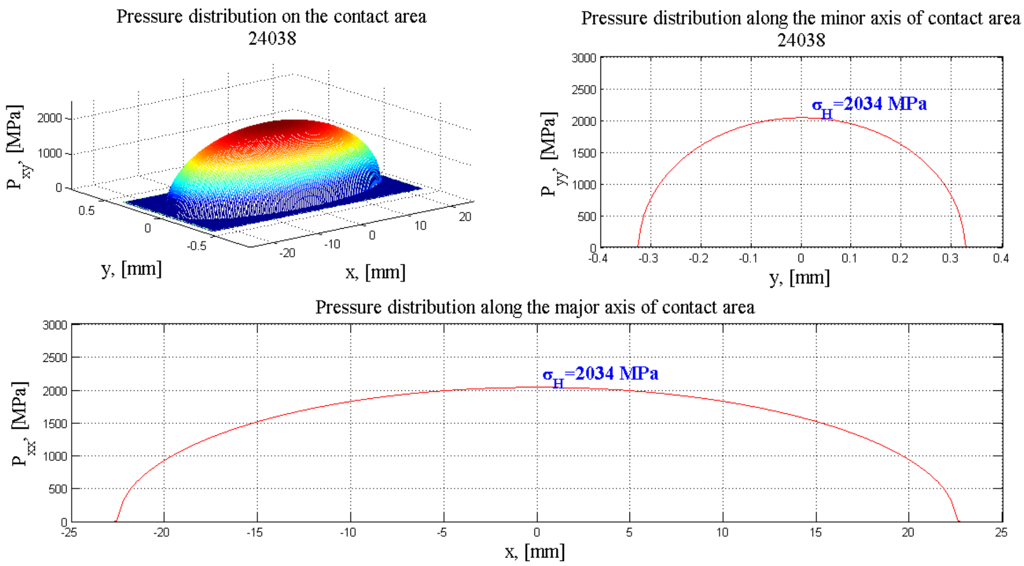

Figure 7. Pressures distribution achieved by the most loaded roller on the inner raceway.

3.2. Minimum film thickness
For the operating conditions attained in the contact between the most loaded roller and inner raceway, the figure 8 depicts the dependence of the kinematic viscosity and minimum film thickness on operating temperature.

3.3. Modified rating lives
3.3.1. Roughness considered in the comparative lubrication study. Three groups were considered for the roughness of the contacting surfaces: roughness corresponding to good manufacturing practice: Ra_i = 0.16 µm on inner raceway, Ra_w = 0.12 µm on roller surface, composed Ra_Σ = 0.25 µm; coarser roughness: Ra_i = 0.25 µm, Ra_w = 0.134 µm, Ra_Σ = 0.35 µm; finer roughness: Ra_i = 0.12 µm, Ra_w = 0.08 µm, composed Ra_Σ = 0.18 µm.

3.3.2. Modified rating lives. For coarse roughness case, figure 9 presents the theoretical values for the modified rating lives evaluated with the two methods. For an operating temperature t = 60^{0}C the theoretical values of the modified rating lives resulted as: $h_{90-ISO-\frac{\nu}{\nu_1}} = 2482$ hours, when κ was evaluated using the viscosity ratio method (implicitly involving a hypothetical good manufacturing quality), and $h_{90-ISO-\lambda} = 750$ hours, when the film λ-method was used (involving coarse roughness).

Figure 10 depicts the same estimations for the modified rating lives provided by the $\kappa = \nu/\nu_1$ and λ-ratio approaches when the composite roughness is Ra_Σ = 0.225 µm.

If more advanced lapping technologies, able to provide finer roughness, are involved greater values for the modified rating life are possible to be obtained. Figure 11 reveals this for the case: $Ra_{raceway}$ = 0.12 µm and Ra_{roller} = 0.08 µm. For the same operating temperature of 60 ^{0}C the theoretical value of the modified rating life increased to $h_{90-ISO-\lambda}$ = 3100 hours.

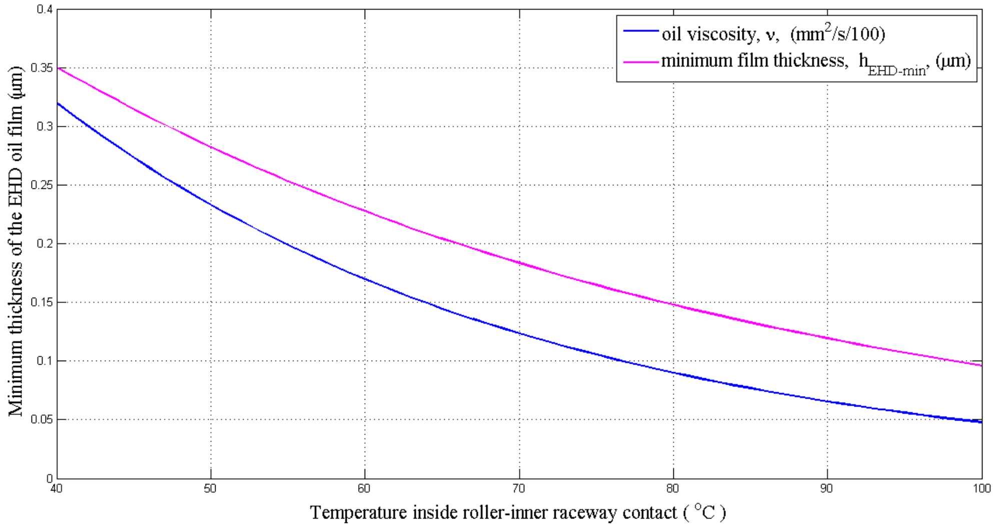

Figure 8. Evaluation of kinematic viscosity and minimum film thickness.

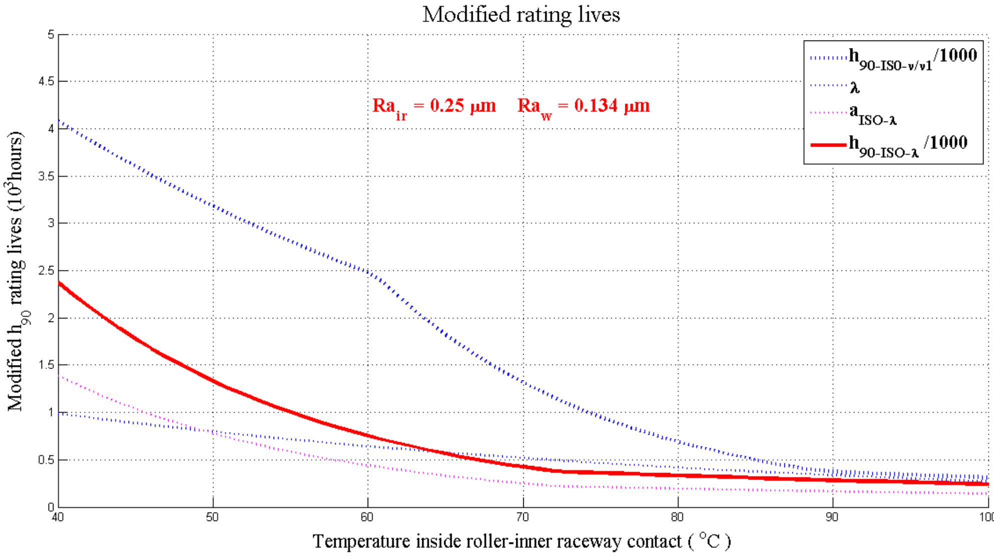

Figure 9. Modified rating lives estimations: $\kappa = \nu / \nu_1$ approach vs. λ-ratio approach, for coarse roughness ($Ra_{raceway}$ = 0.25 μm, Ra_{roller} = 0.134 μm, Ra_{Σ} = 0.35 μm).

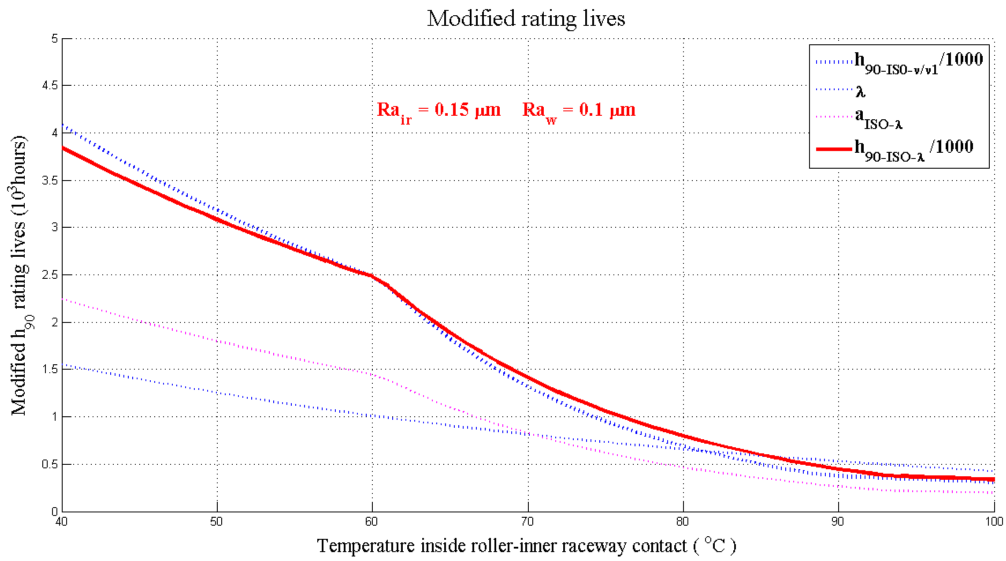

Figure 10. Modified rating lives estimations: the $\kappa = \nu / \nu_1$ approach vs. λ-ratio approach, roughness defined as good manufacturing ($Ra_{raceway}$ = 0.15 µm, Ra_{roller} = 0.10 µm, Ra_{Σ} = 0.225 µm).

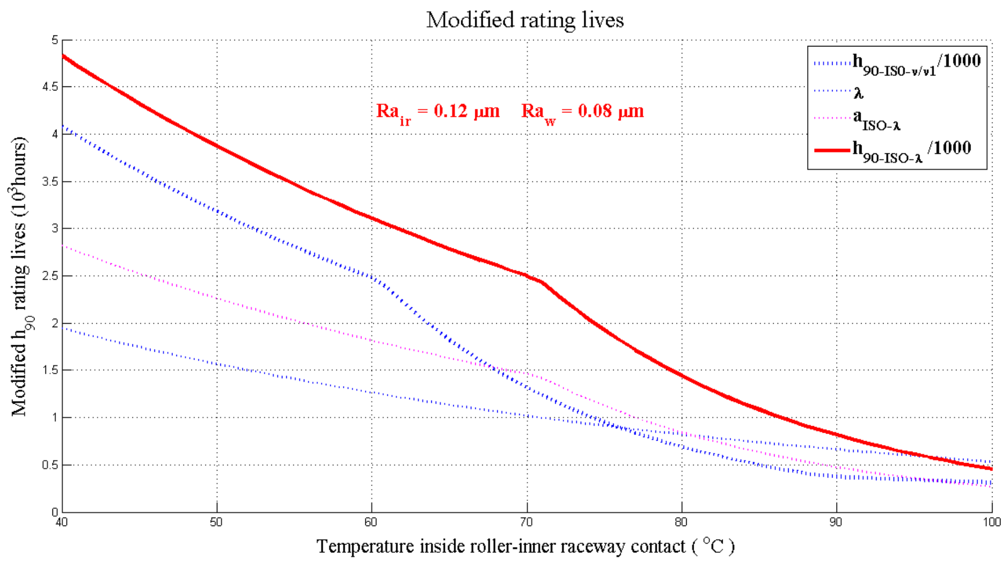

Figure 11. Modified rating lives estimations: the $\kappa = \nu/\nu_1$ approach vs. λ-ratio method, finer roughness ($Ra_{raceway}$ = 0.12 µm, Ra_{roller} = 0.08 µm, Ra_{Σ} = 0.18 µm).

4. Depth distribution of the equivalent stress

4.1. Initial data

High radial loading ($F_r = 0.3 \cdot C_r$) applied in the experimental tests was extended to another two cases: medium and low loading having $F_r = 0.1 \cdot C_r$ and $F_r = 0.05 \cdot C_r$ respectively.

Three groups of random Gaussian roughness were generated with the imposed values for the *Ra* parameter as those used in the modified lives investigations.

The algorithms used to generate the random Gaussian roughness as well as the semi-analytical method involved to solve the concentrated rough contacts are exposed in [19] to [22].

4.2. Numerical results and discussion

For reasons of comparisons, the depth evolutions of equivalent von Mises stresses were referring to: rough surfaces, smooth surfaces, smooth surfaces and the fatigue load limit as radial load ($F_r = C_u$). The results graphically summarized in figure 12 were obtained with data normalized to values corresponding to the smooth surfaces case.

For the same loading level, the depth distribution of von Mises stress for rough surfaces is close to the distribution found for smooth surfaces, except the shallow layer. Details of the von Mises stresses inside the shallow layer are pointed out in figure 13.

For medium and especially high radial loads, the contact between rough surfaces develops, inside this shallow layer, von Mises equivalent stresses higher than the fatigue limit stress. For condition of poor lubrication, these findings explain the initiation of the rolling contact fatigue in the shallow layer close to contacting surfaces.

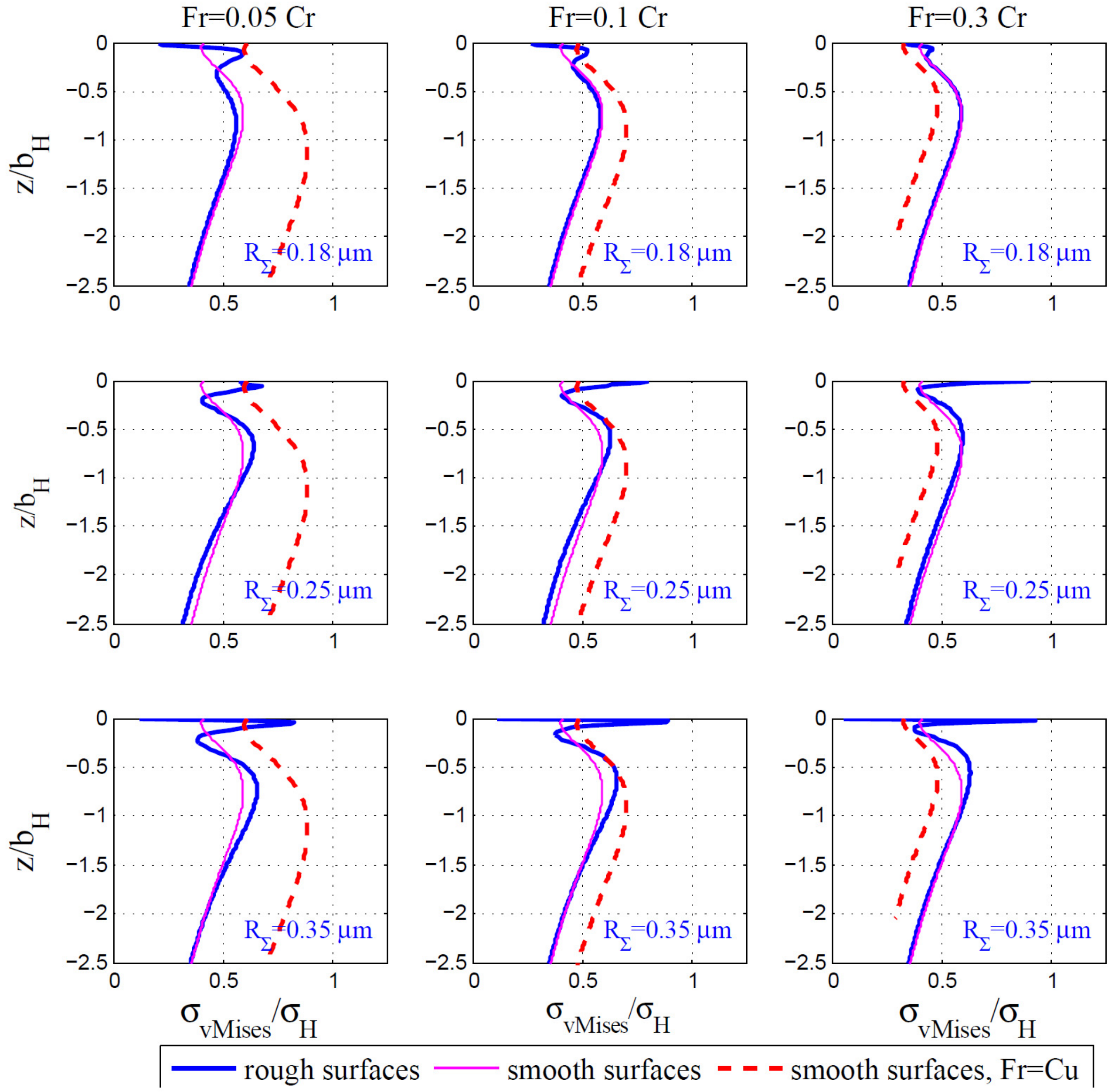

Figure 12. von Mises stress distribution in depth vs. loading and roughness parameter *Ra*, [23]

5. Conclusions

ISO 281:2007 describes the condition of lubricant separation by a complex lubrication parameter κ, defined as the ratio of the actual kinematic viscosity ν to the reference kinematic viscosity ν_1. For bearings with raceway surfaces machined with good manufacturing quality the reference viscosity is established analytically as a function of the pitch diameter and rotation speed. There are no specification regarding the roughness values corresponding to good manufacturing practice for a particular bearing type and dimension. For bearings with especially machined contacting surfaces, λ-ratio, defined as the ratio of the minimum film thickness to the composite roughness of two surfaces in contact, is firstly determined and its value is used to estimate the particular value of κ.

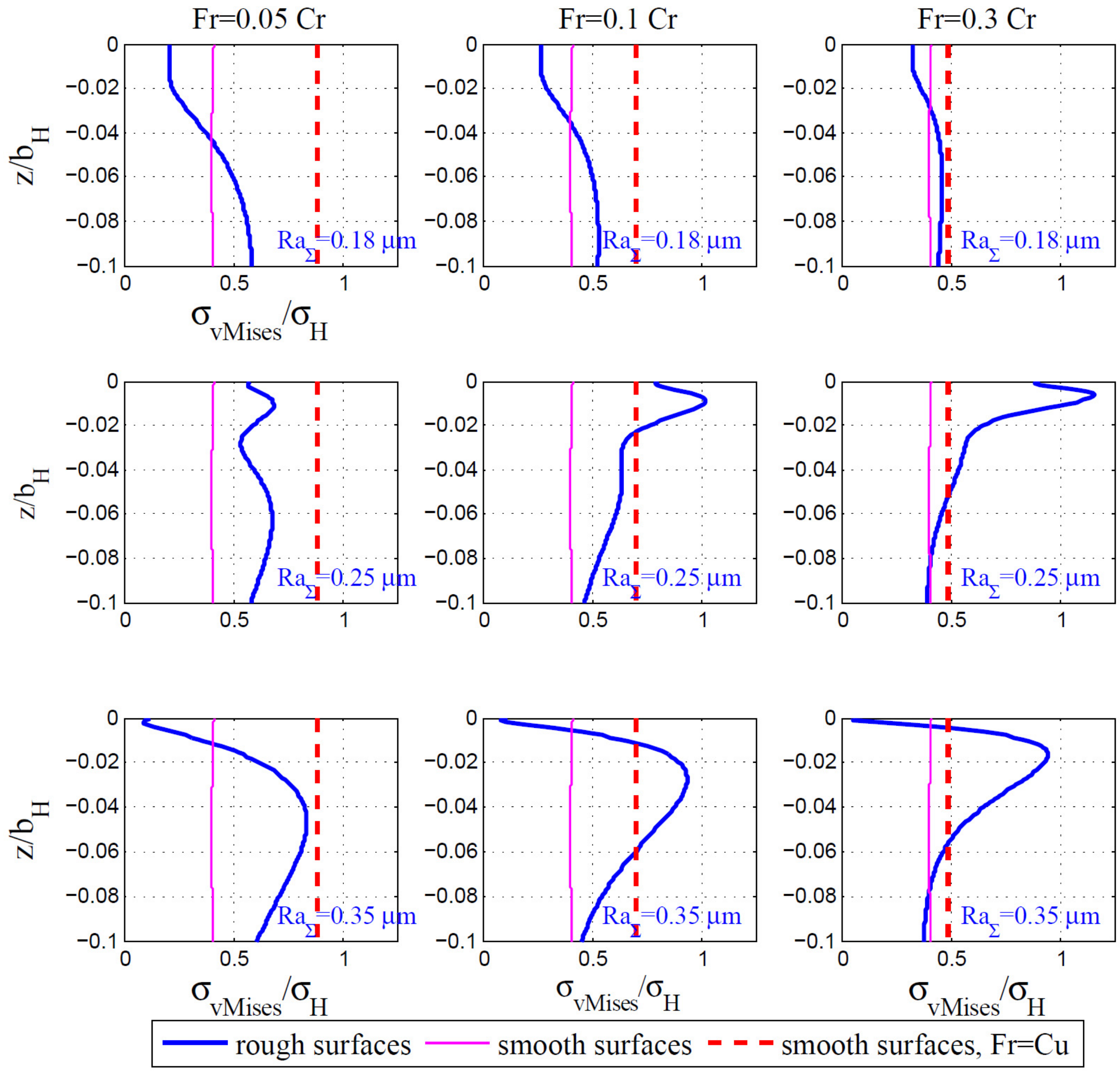

Figure 13. Sub-surface distribution of von Misses stresses presented in figure12, [23].

Using the λ-ratio approach, the paper reveals the relationship between roughness amplitude and the modified rating life of rolling bearings. The roughness values corresponding to good manufacturing practice are possible to be determined for each particular case.

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Numerical analysis shown that for the same loading level, the depth distribution of von Mises stress for rough surfaces is close to the distribution found for smooth surfaces, except the shallow layer close to the contacting surfaces.

For medium and especially high radial loads, the contacts between rough surfaces develop, inside the shallow layer, von Mises equivalent stresses higher than the fatigue limit stress. For condition of lack of lubricant or starved lubrication, these findings explain the initiation of the rolling contact fatigue in the shallow layer, close to contacting surfaces.

Nomenclature

a_H	major half-axis of the hertzian contact area, in millimeters
a_{ISO}	life modification factor, based on a system approach of life calculation
a_1	life modification factor for the reliability
b_H	minor half-axis of the hertzian contact area, in mm
C_r	basic dynamic radial load rating, in N
C_u	fatigue load limit, in newtons
D_w	roller diameter, in mm
e_c	contamination factor
F_r	bearing radial load, in N
h_c	central film thickness, in mm
h_0	minimum film thickness, in mm
$h_{EHD\min}$	minimum film thickness in an elasto-hydrodynamic lubrication, in mm
h_{90-ISO}	modified rating life, in hours
i	number of rows of rolling elements
L_{nm}	modified rating life, in million revolutions
L_{we}	effective roller length, in mm
L_{10}	basic rating life, in million revolutions
n	speed of rotation, in rot/min
P	dynamic equivalent load, in N
Ra	arithmetic average of the surface roughness heights, in μm
Ra_Σ	arithmetic average of the composed roughness, in μm, $Ra_\Sigma = 1.25 \cdot \left(Ra_{raceway}^2 + Ra_{roller}^2\right)^{1/2}$
S	radial operating clearance, in mm
Z_w	number of spherical rollers
z	depth
α	nominal contact angle of the bearing, in degrees
κ	viscosity ratio, $\kappa = \nu / \nu_1$
λ	film parameter, evaluated as the ratio between the minimum film thickness and composed roughness of involved surfaces
σ_H	maximum pressure in a hertzian distribution
σ_{vMises}	von Mises equivalent stress
ν	actual kinematic viscosity at operating temperature, in mm²/s
ν_1	reference kinematic viscosity required to obtain adequate lubrication, in mm²/s

Subscript

e	outer ring
i	inner ring
w	roller

n probability of failure

6. References

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