Research Article

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The use of FEM to evaluate the influence of logarithmic correction parameters of roller generators on the axle box bearing life

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Abstract: In the roller-raceway contacts of the radial cylindrical roller bearing used in the axle boxes of a railway bogie, pressure accumulation may occur, reducing the fatigue life. This accumulation can be eliminated by applying logarithmic correction of generators and in particular varieties of the modified logarithmic correction. The correction parameters should be adapted to the operating conditions of the bearing. This article presents a comparison of the predicted fatigue life of an axle box bearing on correctly selected correction parameters with bearing life, in which correction of roller generators was used, typical for cylindrical roller bearings of general application. The finite element method was used to determine the subsurface stress distributions necessary to calculate the fatigue life.

Keywords: rolling bearing; correction of rolling elements generators; fatigue life; finite element method

1 Introduction

At the ends of the contact lines of bodies with different curvatures and rectilinear generators, there is pressure accumulation and, as a consequence, the accumulation of subsurface stresses determines the fatigue life. Such a phenomenon may occur in the contact of rollers with raceways in rolling bearings. The only way to eliminate or reduce stress accumulation is by giving a proper shape to generators of the mating elements. In modern cylindrical roller bearings, the logarithmic correction of roller generators is used. In the second half of the past century, its simplified equivalent, the chord-arch (ZB) correction, was widely used [1].

The problem of determining the shape of a roller generator, ensuring the pressure distribution closest to the distribution for the Hertzian linear contact, was first undertaken by Lundberg [2]. Aiming to achieve an even distribution of pressures along with the generator, Lundberg proposed the logarithmic shape of the generator of a rolling element using the following formula:

$$h\left(x\right)=\frac{4{p_K}^2}{E'^2\Sigma\rho_K}ln\frac{1}{1-\left(2x/l_K\right)^2} \qquad (1)$$

where

$$\frac{1}{E'}=\frac{1}{2}\left(\frac{1-{\nu_1}^2}{E_1}+\frac{1-{\nu_2}^2}{E_2}\right) \qquad (2)$$

$$\Sigma\rho_K=\frac{2}{D_r}\pm\frac{2}{d_b} \qquad (3)$$

In formula (1), p_K is the pressure for which the correction is selected and l_K is the length of the corrected generator.

From formula (1), it follows that at the ends of the generator the amount of correction $h(x)$ tends to infinity. For this reason, Lundberg, with simplifying assumptions, determined the finite maximum value of the correction coordinate, i.e. the correction $h(x)$ at the ends of the contact line as follows:

$$h_L=\frac{4{p_K}^2}{E'^2\Sigma\rho_K}\left(0.5-ln\frac{4p_K}{l_KE'\Sigma\rho_K}\right) \qquad (4)$$

Logarithmic correction ensures advantageous pressure distribution along the contact line for only one pressure value p_K. To ensure the effectiveness of the correction for cases in which the contact pressure differs from the p_K value, Krzemiński–Freda [3, 4] proposed a simultaneous increase in the maximum correction coordinate and the exponent present in the logarithmic function. He called this correction the modified logarithmic correction. The generator profile corresponding to the modified logarithmic correction is described by the formula:

$$h\left(x\right)=\frac{4{p_K}^2}{E'^2\Sigma\rho_K}\frac{h_m}{h_L}ln\frac{1}{1+e^{-g_K}-\left(2x/l_K\right)^{2\varepsilon_K}} \qquad (5)$$

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where

$$g_K = 0.5 - ln \frac{4p_K}{l_k E' \sum \rho_K}. \tag{6}$$

For general-purpose radial cylindrical roller bearings, Krzemiński-Freda [3, 4] proposed the following parameter values in the modified logarithmic correction formula:

– correction selection pressure: p_K = 2100 MPa; value corresponding to the maximum pressure in the roller-raceway contact for a bearing loaded with radial force, which is equal to 30–40% of the dynamic bearing load rating (F_r = 0.3–0.4°C);
– relative maximum correction coordinate: h_m/h_L = 3;
– exponent in the correction formula: ϵ_K = 3.

In many devices, rolling bearings work at significantly lower loads. This occurs in the case of the axle box roller bearing of the railway carriage. Figure 1 shows a cross-section of a typical rolling bearing assembly of a two-axle Intercity passenger carriage bogie. The bearing assembly is made of two FAG radial cylindrical roller bearings, one of the type WJ 130 × 240-TVP and the other of type WJP 130 × 240-P-TVP [5, 6]. The bearings are positioned in such a way that an integral flange of the WJ bearing inner ring and a loose flange ring of the WJP bearing are located on the outer sides of the bearing system.

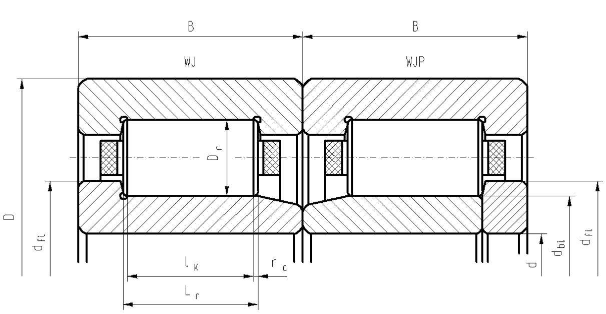

Figure 1: Axle box bearings of a two-axle Intercity passenger carriage bogie.

According to the studies of the FAG company [6], the load of a single bearing of the assembly occurring during operation is equal to F_r = 47,000 N. This load is an equivalent load taking into account dynamic loads, but not the axial load. The dynamic load capacity of the WJ 130 × 240-TVP bearing is C = 540,000 N, which means that each axle box bearing carries a load equal to $F_r \approx$ 0.09 °C. It follows that the maximum pressure occurring in the roller-raceway contact will be much smaller than in general-purpose bearings. The consequence of this fact is the need to select the correction parameters of roller generators in such a way as to ensure the maximum capacity of the contact, and thus the greatest fatigue life of the axle box bearing.

Warda [7] presents the results of analyses of the influence of modified logarithmic correction parameters on the fatigue life of the WJ 130 × 240-TVP axle box bearing. The computer simulations carried out showed that the largest forecasted bearing fatigue life is ensured by the following correction parameters:

– correction selection pressure: p_K = 900 MPa;
– relative maximum correction co-ordinate: h_m/h_L = 2;
– exponent: ϵ_K = 3.

For calculations, a computer program was used to determine the load distribution on radial cylindrical roller bearings (ROLL1), a program for calculating predicted bearing fatigue life (ROLL2), and a program that allows finding pressure and subsurface stress distributions in roller contacts with raceways by solving the Boussinesq problem for elastic half-space (ROLL4). The programs were formulated based on the methodology described by Warda [8] and Warda and Chudzik [9].

This study presents the results of numerical analysis of pressure distribution and the corresponding stress distribution. Numerical calculations were obtained using modern computational techniques. The paper presents a comparison of the results of the predicted fatigue life of a bearing with correction typical for general-purpose bearings, whose parameters were determined for p_K = 2100 MPa, and bearing life working at a smaller load, for which a correction with parameters determined for the correction selection pressure p_K = 900 MPa is more suitable.

Determination of fatigue life of a radial cylindrical roller bearing in which correction of rolling elements generators has been applied is possible only with the use of computer technology. In this case, the method of predicting durability requires finding local values of subsurface stress on which the bearing life depends.

The most accurate information about the nature of the pressure distribution and the corresponding subsurface stress distribution is obtained by the finite element method (FEM). For this purpose, a solid FEM model of the bearing should be built that accurately reflects the internal geometry of the bearing, taking into account the bearing clearances and the relative position of the bearing rings and rollers resulting from the load, as well as resulting from assembly errors or shaft deflection.

Searching for load distribution on bearing rolling elements using FEM, while taking into account contact phenomena occurring in the contacts, is time-consuming.

Usually, the authors of papers regarding the determination of load and stress distributions in roller bearings used two-dimensional FEM models [10–12], and thus did not allow for taking into account the shape of the mating elements. Often, to simplify the three-dimensional bearing model, rollers were replaced by so-called superelements. Superelements were used by Golbach [13] and Kania [14] in their research, which allowed them to consider the effect of corrections of rolling element generators on load and stress distribution in bearings. In a few cases, the authors used the three-dimensional FEM model of the entire bearing but did not include the complex profile of the generators. This is what Blanusa et al. [15, 16] and Ke et al. [17] did in their work on thermal phenomena occurring in bearings of the axle box. On the other hand, whole bearing models are often used for ball bearing analysis [18].

Researchers generally used numerical methods other than the FEM to determine load distribution on rolling elements [19–21], while solid FEM models of a bearing fragment were used to analyze phenomena occurring in a single contact [22–26].

The authors of this article used a combination of two methods in their earlier work on the prediction of fatigue life of radial cylindrical roller bearings – the FEM for a bearing section to find stresses and the numerical solution of equilibrium equations of rings and rollers to determine load distribution on rolling parts [8, 27–29]. This approach significantly reduced the computational time and allowed them to be carried out on standard computer equipment.

2 Fatigue life of a radial cylindrical roller bearing

The fatigue life of the axle box bearing was determined using the method described by Warda [8] and Warda and Chudzik [9] and also used in papers [27–29]. This method uses the basic assumptions of the fatigue life prediction model developed by Lundberg and Palmgren [30, 31].

The durability of the inner ring rotating relative to the load is calculated from the formula determining the inverse logarithm of the probability of durability $\zeta_i$as:

$$ln\frac{1}{\varphi_o} = Au_i^e L_{10i}^e 2\pi \left[\frac{1}{2\pi} \int_0^{2\pi} \left(\int_0^{l_K} r_{bix} \sigma_{ix\psi}^c Z_{ix\psi}^{1-h} dx \right)^{1/e} d\psi \right]^e , \tag{7}$$

while the durability of the stationary outer ring from the formula is:

$$ln\frac{1}{\varphi_o} = Au_o^e\, L_{10o}^e \int_0^{2\pi} \int_0^{l_K} r_{box}\, \sigma_{ox\psi}^c\, Z_{ox\psi}^{1-h}\, dx\, d\psi \tag{8}$$

The fatigue life L_{10} (for $\zeta_i = \zeta_o = 0.9$) of the entire bearing, expressed in millions of revolutions, is determined from the formula:

$$L_{10} = \left(L_{10i}^{-e} + L_{10o}^{-e}\right)^{-1/e} \tag{9}$$

In the above formulas, u is the number of load cycles per revolution, $\sigma_{x\psi}$ the maximum subsurface stress determined in accordance with the von Mises hypothesis, $Z_{x\psi}$ the depth at which the maximum stress occurs, and r_{bx} is the local bearing raceway radius. The values of the exponents found in both formulas are $c = 31/3$, $h = 7/3$, $e = 9/8$. The value of the material constant $A = 7.1 \times 10^{-40}$ was assumed for the axle box bearing.

3 Determination of subsurface stresses using FEM

To determine the values of subsurface stress in the tested bearing, a numerical 3D solid model mapping of the bearing's internal geometry was built introducing modified logarithmic correction of roller generators. Calculations were carried out for comparative purposes for rollers under load during bearing operation when $p_K = 900$ MPa and $p_K = 2100$ MPa. Using the symmetry conditions, FEM analysis was performed for half the roller and the corresponding part of the raceway of the inner or outer ring.

To analyze the subsurface stress in the tested model, it is extremely important to properly model the contact zone of the roller with the raceway. Non-linear phenomena appear during bearing operation. For this reason, the creation of contact pairs and the appropriate setting of contact parameters is a key element to obtain the correct calculation results.

In the numerical analysis, the results of calculations mainly depend on the number of elements resulting from the division of the tested model into finite elements. The contact model was used for calculations in the contact zone: the raceway surface was adopted as the target surface (TARGE type), the roller surface was adopted as the contact surface (CONTA type). The contact surface was modeled using contact elements CONTA175. The target surface was modeled using elements TARGE 170. For selected roller volumes in the contact zone, the roller contact edge with the raceway is divided into equal parts with a length of 0.05 mm. The edge of selected volumes of contact between the raceway and the roller is divided into equal parts with a length of 0.1 mm.

In the numerical model of the roller bearing part not involved in contact with rolling elements, 10-node solid elements of the TET187 type were used to divide into finite elements. Due to the large dimensions of the rolling elements of the tested bearing, the adopted model differs from the models used by the authors in previous works, in which 8-node SOLID 185 solid elements were used. As a result, the examined numerical model was divided into 803,145 solid finite elements. Possible shape errors have been eliminated by adjusting the distance angle between the raceway nodes and the roller. To reflect the actual operating conditions of the bearing, the possibility of displacement in all directions of the outer surfaces of the outer and inner race was taken. The lateral surfaces of the outer and inner races are deprived of the possibility of displacement in the direction of the z-axis. The symmetry conditions of the model were determined relative to the y–z plane. The contact stiffness factor FKN = 1.5 was used for the calculations. The augmented Lagrange method was used to calculate the contact phenomena. Numerical calculations were carried out using the ANSYS program.

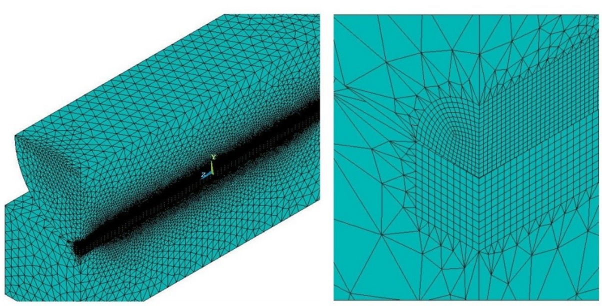

Figure 2: FEM solid model (3D) of the roller-raceway contact.

4 Parameters of the tested axle box bearing

The main parameters of the analyzed axle box bearing are shown in Table 1.

The profile of the roller generator with modified logarithmic correction can also be described by the formula:

Table 1: Parameters of the WJ 130 × 240-TVP (WJP 130 × 240-P-TVP) axle box bearing [5].

Dynamic load rating	$C = 540{,}000$ N
Bearing bore diameter	$d = 130$ mm
Bearing outside diameter	$D = 240$ mm
Bearing width	$B = 80$ mm
Diameter of the outer ring raceway	$d_{bi} = 157$ mm
Diameter of the inner ring raceway	$d_{bo} = 211$ mm
Roller diameter	$D_r = 27$ mm
Roller length	$L_r = 48$ mm
Roller chamfer dimension	$r_c = 1$ mm
Number of rollers in the bearing	$Z_r = 17$

$$h(x) = A_K ln \frac{1}{1 + e^{-1/C_K} - \left(x/B_K\right)^{E_K}} \tag{10}$$

where

$$A_K = \frac{4{p_K}^2}{E'^2 \Sigma\rho_K}\frac{h_m}{h_L}, \quad B_K = l_K/2, \quad C_K = 1/g_K\,, \quad E_K = 2\varepsilon_K \tag{11}$$

Table 2 summarizes the values of the parameters A_K, B_K, C_K, and E_K for the analyzed correction cases. For the calculation, the following values were taken: Young's modulus $E = 2.08 \times 10^5$ MPa, Poisson's ratio $\nu = 0.3$, and curvature sum $\Sigma\rho_K = 8.704 \times 10^{-2}$ mm^1.

5 The distributions of radial loads

The first part of the calculations determining the predicted fatigue life of the axle box bearing was to determine the load distributions for individual bearing rollers. The calculations were carried out assuming the bearing load radial force $F_r = 47{,}000$ N, and the value of radial clearance in the bearing $g = 0.025$ mm. This clearance value corresponds to the load distribution angle $\psi_\epsilon \approx\ 65°$ and the force acting on the most loaded roller equal to $Q_{rmax} = 5\,F_r/Z_r$. Bearing manufacturers do not provide the value of the radial clearance occurring during bearing operation but recommend the use of bearings with an increased radial clearance of 0.120–0.160 mm [6]. The radial clearance that occurs during the operation of the bearing is much smaller and difficult to determine. On its reduction, it affects both the tight fit of the bearing inner rings and the air stream cooling of the outer rings during travel which leads to a further reduction in radial clearance.

Table 2: Modified logarithmic correction parameters for the assumed correction selection pressure p_K.

Correction selection pressure p_K	2100 MPa	900 MPa
Lundberg correction co-ordinate h_L	0.0202 mm	0.0043 mm
Relative maximum correction co-ordinate h_m/h_L	3	2
Maximum correction co-ordinate h_m	0.0606 mm	0.0086 mm
ϵ_K	3	3
A_K	0.011695 mm	0.001432 mm
B_K	23 mm	23 mm
C_K	0.193	0.166
E_K	6	6

Table 3: Radial load distributions on WJ 130 × 240-TVP bearing rollers.

F_r = 47,000 N, g = 0.025 mm, ψ_ϵ = 66.8°	
Roller (j)	Q_{rj} [N]
1	13,490
2	11,831
3	7,215
4	877
5	0

The results of the load distribution calculations obtained using the ROLL1 program [9] are shown in Table 3. As the table shows, due to the symmetry of the distribution, radial load carries 7 out of 17 bearing rollers simultaneously.

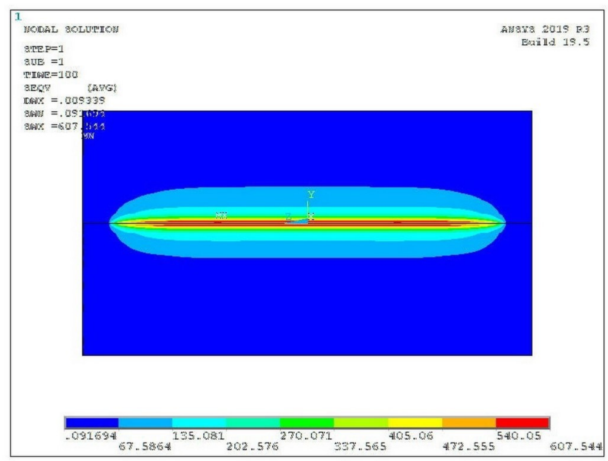

Figure 3: Distributions of von Mises stresses under the most loaded roller-inner ring raceway contact surface, p_K = 2100 MPa.

6 Distribution of maximum subsurface stress and their depth of occurrence

The second stage of calculations, which is necessary to predict the bearing's fatigue life, is to determine the maximum von Mises stress distribution σ and the depth of their occurrence Z. The calculations were performed using the FEM in accordance with the assumptions described in Section 3. It has been assumed that the rollers and bearing rings are made of elastic-plastic material with material properties that were adopted to determine the correction parameters of roller generators. Figures 3 and 4 show von Mises stress maps for the most loaded axle box bearing roller in its contact with the inner ring raceway for both analyzed correction cases.

The basic package of the ANSYS program allows to create graphs showing the maximum stress distributions along the roller contact line with the raceway; however, it is not possible to obtain the depth distributions of these stresses. But it is possible to determine the distribution of subsurface stress in a plane perpendicular to the axis of the roller set by the user at any point along the contact line. By analyzing changes in subsurface stresses in selected sections, the distribution of the depth of occurrence of maximum subsurface stresses can be determined. The method of determining distributions has been described by Chudzik and Warda [28, 29].

Combining the characteristics of stress distribution σ along the contact line (x coordinate) obtained from the ANSYS program with the Z depth distribution patterns, the graphs presented in Figures 5–8 were obtained. Values of σ and Z along the line of contact are shown in Table 4. Each of the drawings illustrates stress distribution σ and depth Z for both the considered cases of forming corrections.

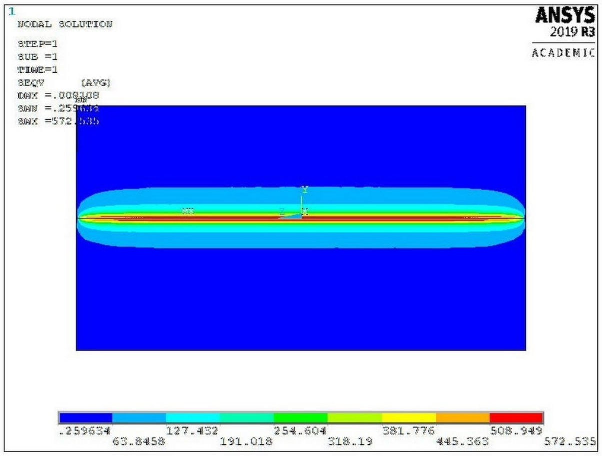

Figure 4: Distributions of von Mises stresses under the most loaded roller-inner ring raceway contact surface, p_K = 900 MPa.

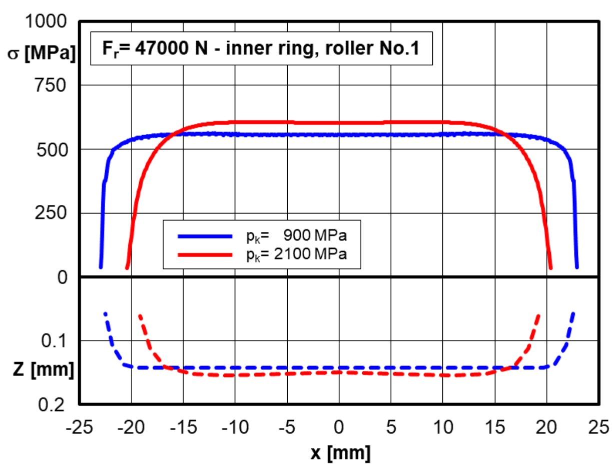

Figure 5: Distributions of von Mises stresses σ and the depth of their occurrence Z along the line of contact of roller No. 1 with inner ring raceway.

As can be seen from the graphs, the value of the correction selection pressure p_K has a significant impact on the nature of the stress distribution under the contact surface. Adoption for bearing operating under low load ($F_r \approx\ 0.09\ ^\circ$C) of correction parameters typical for general-purpose bearings means that the length of the contact field is much smaller than the length of the roller generator, and the working surface of the roller is not fully used. The correction parameters properly selected for the bearing operating under specific conditions allow the full exploitation of the working surface of the rollers, and at the same time do not allow pressure accumulation.

Regardless of the generator shape, each of the rollers is loaded with the same radial force Q_{rj}, resulting from the load distribution. Incomplete use of the working surface of

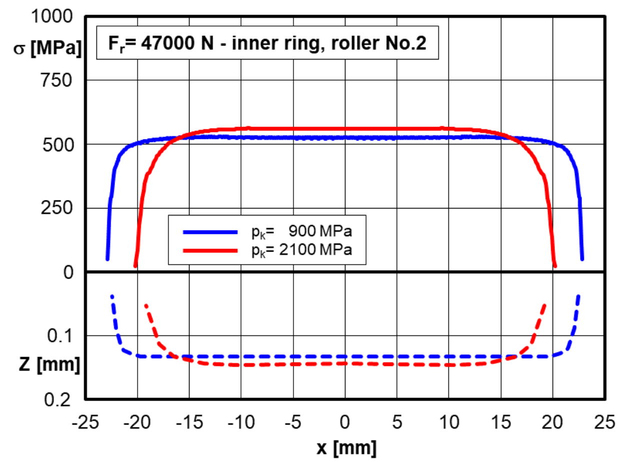

Figure 6: Distributions of von Mises stresses σ and the depth of their occurrence Z along the line of contact of roller No. 2 with inner ring raceway.

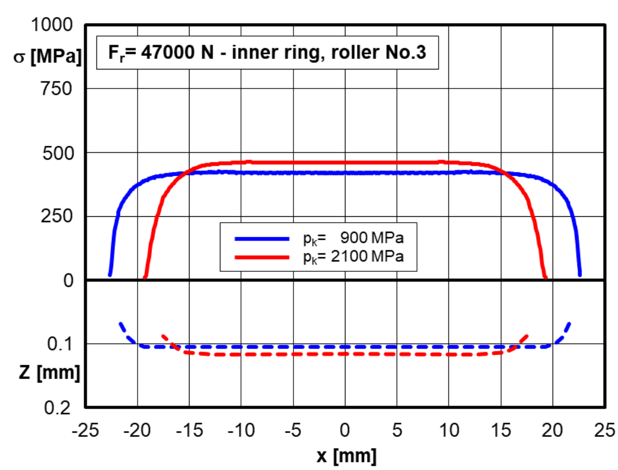

Figure 7: Distributions of von Mises stresses σ and the depth of their occurrence Z along the line of contact of roller No. 3 with inner ring raceway.

Table 4: Values of von Misses stress σ and the depth Z along the line of contact.

F_r = 47,000 N, roller contact with inner ring						
	p_K = 2100 MPa			p_K = 900 MPa		
	x [mm]	σ [MPa]	Z [mm]	x [mm]	σ [MPa]	Z [mm]
Roller No. 1	0.13981	557.89	0.14200	0.148810	604.39	0.14901
	11.88423	557.94	0.14200	10.92539	604.45	0.15408
	17.89625	552.69	0.14196	14.11846	588.22	0.15219
	19.64394	543.08	0.14192	16.81260	532.79	0.14150
	20.76245	524.77	0.13700	18.20957	458.38	0.11150
	21.88097	484.53	0.10700	19.20740	353.56	0.06150
	22.51013	375.73	0.05700			
Roller No. 2	0.20972	524.12	0.13300			
	11.88423	526.61	0.13302	0.24853	559.63	0.14300
	17.68653	517.63	0.13296	10.42656	559.58	0.14596
	19.71384	504.88	0.13291	13.61966	550.46	0.14300
	20.69254	493.54	0.12800	16.51342	514.68	0.13300
	21.32171	479.37	0.12300	17.91040	464.49	0.11300
	21.53143	465.21	0.11800	19.20760	368.44	0.05300
	22.09069	416.79	0.08800			
	22.44023	318.53	0.03800			
Roller No. 3	0.20972	419.27	0.10457			
	12.16386	426.40	0.10457			
	14.68052	419.11	0.10457	0.14874	460.36	0.11550
	16.42820	416.43	0.10457	12.52203	455.45	0.11650
	18.10598	406.09	0.10457	15.41579	418.42	0.11250
	19.29440	389.62	0.10457	16.21406	393.11	0.10750
	20.27310	365.17	0.09850	17.51126	323.63	0.08750
	20.97217	335.37	0.08850			
	21.60134	289.88	0.06850			
Roller No. 4	0.13982	81.96	0.02500			
	7.06080	82.65	0.02500			
	9.78724	82.83	0.02632	0.14868	121.62	0.03650
	11.88451	90.96	0.02695	9.82791	118.94	0.03696
	12.44378	91.80	0.02500	10.82578	114.80	0.03616
	13.28269	88.42	0.02550	12.12299	101.82	0.03526
	14.05169	88.77	0.02499	13.22064	81.232	0.03250
	15.44986	84.39	0.02499	14.41807	39.520	0.01850
	17.26750	73.24	0.02250			
	18.38604	60.15	0.01250			

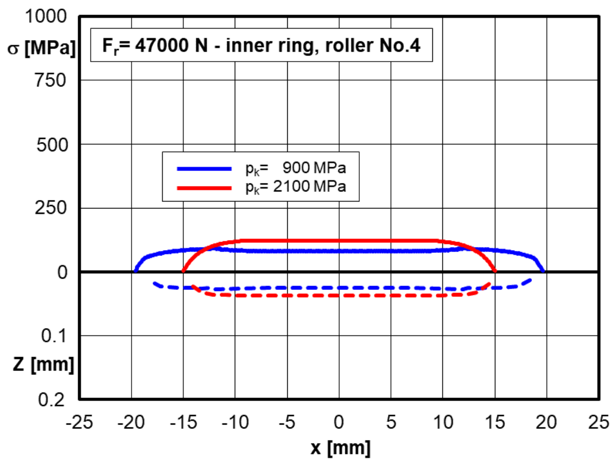

Figure 8: Distributions of von Mises stresses σ and the depth of their occurrence Z along the line of contact of roller No. 4 with inner ring raceway.

the roller results in a higher pressure on the contact surface and higher stress under the surface. The maximum stress values σ are in the case of the correction designed for p_K = 2100 MPa by about 40 MPa higher than the values of stress occurring in the contact with the correction selected for p_K = 900 MPa. This has an obvious effect on the fatigue life of the bearing.

7 Results of the calculations of predicted fatigue life of the axle box bearing

Graphs similar to those presented in Section 6 were also prepared for roller contacts with the outer ring raceway, and the data contained in the files based on which the charts were made were used to calculate the predicted fatigue life of the axle box bearing. The ROLL2 computer program [9] was used to calculate the predicted bearing fatigue life. Figure 9 shows a comparison of the bearing fatigue life for both types of corrections considered.

Calculations of the predicted fatigue life of the axle box bearing show how much impact the selection of appropriate correction parameters of roller generators has on the bearing durability. The generator profile, which was determined for the correction selection pressure p_K = 900 MPa and relative maximum correction coordinate h_m/h_L = 2, guarantees the bearing fatigue life about 50% greater than the profile that is recommended for general-purpose bearings (p_K = 2100 MPa, h_m/h_L = 3).

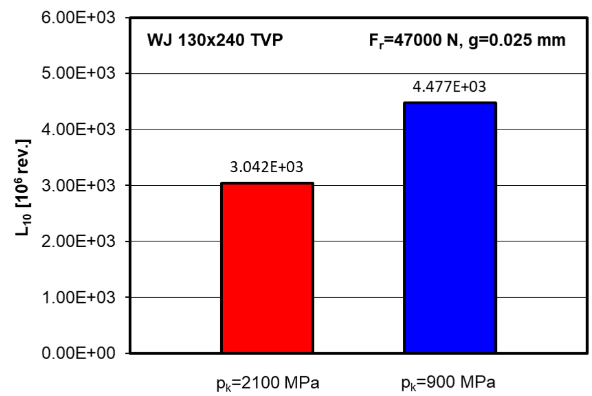

Figure 9: The predicted fatigue life of the axle box bearing with modified logarithmic correction of roller generators with parameters determined in accordance with p_K = 2100 MP and p_K = 900 MPa.

8 Conclusions

1. A modified logarithmic correction can provide a favorable pressure distribution along the contact line of the mating elements for a much wider load range than the logarithmic correction proposed by Lundberg; however, it requires a suitable selection of the parameters. Correction parameters should be different for standard cylindrical roller bearings operating under a load representing 30–40% of the dynamic bearing capacity of the bearing, and different for special-purpose bearings, such as railway carriage axle box bearings.
2. Correctly selected parameters of the modified logarithmic correction should guarantee the greatest possible length of contact between the roller and the raceway while preventing pressure accumulation. Axle box bearing with such correction can achieve a life of 50% longer than a similar bearing with a correction typical for general-purpose bearings.
3. The value that has the greatest impact on the effectiveness of the applied correction of the profile of the rolling element generators is the pressure for which the correction is selected. The value of the correction selection pressure should be close to the maximum pressure values occurring in the contact of the roller with the bearing raceways. What is also important are the other two parameters of the correction, the relative maximum correction coordinate and exponent of logarithmic function, determining the shape of the generator and pressure distribution at the edges of the roller.

4. To determine the predicted fatigue life of a bearing, it is necessary to know the subsurface stresses in the roller contacts with the bearing raceways. Stress distributions are most conveniently determined using the FEM. At the same time, load distribution on rolling elements should be determined. This can be done by performing FEM calculations for the complete bearing solid model, which is time-consuming and requires the use of computers with very high computing power. Sufficiently accurate results can be obtained by combining the two methods: the FEM for a bearing section, and the numerical solution of equilibrium equations of rollers and bearing rings.

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