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CYLINDRICAL ROLLER BEARINGS WITH OPTIMIZED PROFILE TAKING INTO ACCOUNT THE MATERIAL STRUCTURE AND OPERATING TEMPERATURE

BY

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Abstract. The present paper proposes a method for an almost uniform pressure distribution between the rolling elements, based on the modification of the roller’s initial straight profile and accounting for material parameters related to the operating temperature. The optimization process is analyzed by including the possible misalignment and/or tilting angles of the roller due to operating conditions. A computer program was developed in the Borland C++ programming language. The contact pressure distribution is evaluated by using a three-dimensional contact analysis model based on the Boussinesq force-displacement relations for a half-space combined with the flexibility approach. For a fast analysis, the algorithm uses a Conjugate Gradient Method in deriving the contact pressures and the real contact area.

Keywords: profile optimization; material structure; skew; misalignment; temperature influence.

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1. Introduction

The contact problem in roller bearings needs a three-dimensional approach, due to the presence of edge-stresses and a contact surface shaped as a "dog-bone".

The famous logarithmic profile proposed by Lundberg (Lundberg, 1939) gives excellent results for almost uniform contact pressures along roller generatrix, but at the ends of the contact high pressures, spikes occur due to the finite length of the roller. Also, the profile is not easy to manufacture and is only optimum for a particular load; any load change will require a new logarithmic function. In practice, rollers can suffer from a tilting moment and a skew angle due to operating conditions. Many solutions have been proposed by various researchers, (Moyer and Neifert, 1963; Hartnett, 1979; Johns and Gohar, 1981; So and Gohar, 1983; Torstenfelt, 1983; Torstenfelt and Fredriksson, 1984; de Mul *et al.*, 1986; Lösche, 1987; Reussner, 1987; Horng *et al.*, 2000; Urata, 2000; Chen *et al.*, 2001; Kamamoto *et al.*, 2001; Hiroki *et al.*, 2009; Kumar *et al.*, 2009; Kabus *et al.*, 2012; Zhu *et al.*, 2012; Cui and He, 2015), either modifying the original logarithmic profile or developing various roller profiles based on a linear profile rounded at the edges with one or two crowning radii and even cylindrical rollers with the profile made by only crowning radii (2 or 3, usually). At the same time, the roller profile has been optimized in order to increase bearing fatigue life, but ignoring the skew and tilt angles that occur in real operating condition. The operating temperature is also neglected, but is has a great importance on the material parameters (Young's modulus, Poisson's coefficient, yield limit) for different material structures (SAE 52100 bearing steel can be obtained with martensitic, bainitic or martensite and retained austenite structure).

2. Constitutive Material Models

Fatigue life of rolling bearings is in relation with the possible development of sub-surface plastic strains. In roller bearings, due to the presence of the edge stresses higher than the central contact pressures, even at moderate loads, this phenomenon can lead to premature failure. In order to avoid any plasticity effect, the rollers have a modified profile, in order to achieve a so-called "uniform pressure distribution" along the whole contact length.

This paper proposes a derivation of roller profile based on the limitation of the maximum contact pressure beyond the value that gives only elastic stresses and strains. In order to properly derive the material parameters, several uniaxial compressive experiments were performed on the Instron™ test device equipped with a climate cell. The samples were cylindrical, 7 mm long with a diameter of 3 mm, and made of three variants of bearing steel structure SAE 52100.

The bearing steel SAE 52100 is widely used in bearing industry. Usually, the most common variant has a martensitic structure with some percentage of

retained austenite (7 to 13%), the 100Cr6 steel; but they are also variants with martensitic structure (with retained austenite less than 2%) or bainitic structure (without retained austenite). Their elastic behaviour is quite similar; differences are related to the yield stress and plastic behaviour. Since the purpose of this work is to accurately describe the elastic behaviour, experiments were performed at ambient temperature (20°C). In most cases, the bearings are working at elevated temperatures; assuming there is a steady-state operating temperature, the values of 65°C and 100°C, respectively, were selected. It is known that at high temperature, the Young's modulus and Poisson's coefficient are different than those at room temperature (Thuvander, 1998). The obtained stress-strain curves and corresponding material parameters are given in Table 1.

Table 1
Material elastic parameters from uniaxial compression tests

SAE 52100 steel structure	Temperature [°C]	Young's modulus [MPa]	Poisson's coefficient [-]	Yield limit in compression [MPa]
Martensite and 13% retained austenite (100 Cr6)	20	214000	0.281	1400
	65	212000	0.2816	1350
	100	210000	0.2825	1150
Martensite (retained austenite < 2%, Ma)	20	214000	0.281	1900
	65	212000	0.2816	1850
	100	210000	0.2825	1540
Bainite (Ba)	20	214000	0.281	2100
	65	212000	0.2816	2040
	100	210000	0.2825	1720

From the values in Table 1, we can observe that there is a significant difference between the yield stress at room temperature and the values at 100°C, for all three steel variants investigated. Also, a general relation was derived between the initial yield limit and that at operating temperature, valid for the range 20°C<T<100°C, for all three steel variants, Eq. (1):

$$\sigma_{yield}(T)=\left(0.948+3.585\cdot10^{-3}\cdot T-4.934\cdot10^{-5}\cdot T^{2}\right)\cdot\sigma_{yield20} \quad (1)$$

where: σ_{yield} is the yield limit at a given temperature and $\sigma_{yield20}$ is the yield limit at ambient temperature (20°C).

Regarding the strength of rolling contacts, previous numerical static contact simulations showed that material microstructure is of great importance. Considering the rolling contact fatigue generated by the development of sub-surface plasticity, a previously developed algorithm (Popescu *et al.*, 2006) was used to obtain the maximum contact pressures that allow a bearing to operate in the elastic regime; *i.e.* no sub-surface plasticity is initiated, Table 2.

Table 2
Maximum operating contact pressures for different bearing steel hardened variants

SAE 52100 steel structure	Temperature [°C]	Contact pressure, Pmax [MPa]
Martensite and 13% retained austenite (100Cr6)	20	2550
	65	2130
	100	1810
Martensite (retained austenite < 2%, Ma)	20	3000
	65	2920
	100	2430
Bainite (Ba)	20	3050
	65	2960
	100	2500

Regardless of the temperature range, the maximum contact pressures that do not generate plasticity can be related to the shear yield limit by the formula:

$$P_{max}=a\cdot k \tag{2}$$

where P_{max} is the maximum contact pressure, k is the shear yield strength and the constant a has the value 3.155 for SAE 52100 with martensite and retained austenite, 2.735 for the variant with martensitic structure and 2.515 for the variant with bainitic structure. Values for contact pressures that initiate plasticity are in agreement with the results obtained by other researchers (Lorösch, 1982; Zwirlein and Schlicht, 1982; Johnson, 1985).

3. General Considerations

In case of concentrated contacts (*i.e.* ball bearings), the analytical solutions of Hertz are widely used, since they are accurate (Johnson, 1985). But in case of line contacts, as in roller bearings, the separation between cylindrical surfaces cannot be computed with analytical relation. Since the roller geometry involves several crowning radii, an interesting algorithm was proposed by (Natsumeda, 1999) to evaluate the separation between the bodies in the common coordinate system:

$$h(x,y)=z_1(x_1,y_1)-z_2(x_2,y_2)+d1(x_1)+d2(x_2) \tag{3}$$

where z_1 is the roller surface, z_2 is the ring surface (inner or outer), $d1(x_1)$ and $d2(x_2)$ are the crowning profiles. If the roller is misaligned, Fig. 1, the Eq. (3) becomes:

$$h(x,y)=z_1(x_1,y_1)-z_2(x_2,y_2)+d1(x_1)+d2(x_2)+x_1\cdot\tan(\beta) \quad (4)$$

The presence of a skew angle requires the use of a third coordinates system (Johnson, 1985), Fig. 2, in which the separation of the bodies can be expressed as:

$$h(x,y)=A\cdot x^2+B\cdot y^2+d1(x_1)+d2(x_2)+(x\cdot\cos\theta-y\cdot\sin\theta)\cdot\tan(\beta) \quad (5)$$

where:

$$\begin{cases} A=\frac{1}{4}\cdot\left[\frac{1}{R_{11}}+\frac{1}{R_{21}}\right]-\frac{1}{4}\cdot\sqrt{\frac{1}{R_{11}^2}+\frac{1}{R_{21}^2}+\frac{2\cdot\cos(2\alpha)}{R_{11}\cdot R_{21}}} & (a)\\ B=\frac{1}{4}\cdot\left[\frac{1}{R_{11}}+\frac{1}{R_{21}}\right]+\frac{1}{4}\cdot\sqrt{\frac{1}{R_{11}^2}+\frac{1}{R_{21}^2}+\frac{2\cdot\cos(2\alpha)}{R_{11}\cdot R_{21}}} & (b)\\ \theta=\frac{1}{2}\cdot\tan^{-1}\left(\frac{\frac{1}{R_{21}}\cdot\sin(2\alpha)}{\frac{1}{R_{11}}+\frac{\cos(2\alpha)}{R_{21}}}\right) & (c) \end{cases} \quad (6)$$

In the above relations, R_{11}, R_{12}, R_{21} *and* R_{22} are the principal radii of curvature of the surfaces at the origin.

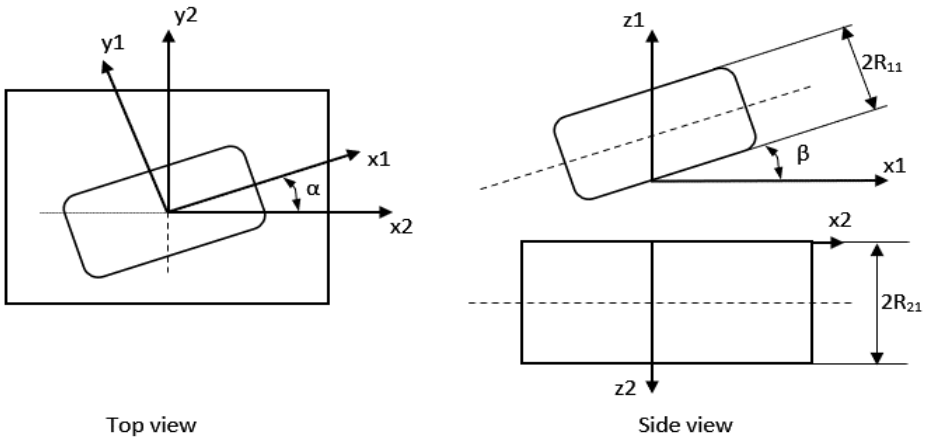


Fig. 1 – Geometry of roller-inner raceway contact.

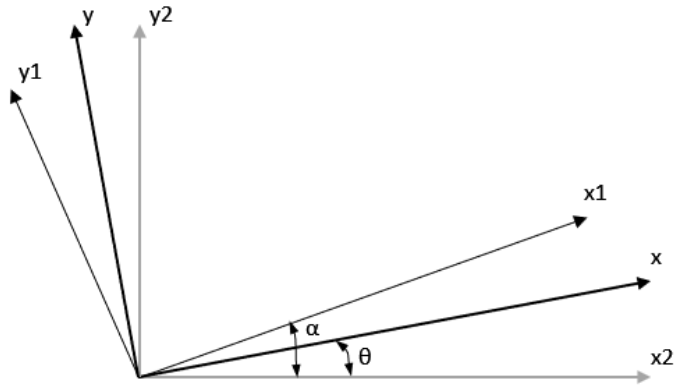


Fig. 2 – Coordinates systems used in the analysis.

If we are working in the common coordinates system, since the curvature radii are much larger than the bodies radii, an acceptable approximation is to consider *that* $d1(x_1) \approx d1(x)$ and respectively $d2(x_2) \approx d2(x)$. This is valid since the skew angle is small than ± 1.5°.

In case of only misalignment occurs, then is no need for the third coordinates system, and more, the pressure distribution is symmetric with respect to the x-axis. In case of skewing, the pressure distribution remains symmetric in the common coordinates system, while in the system attached to ring it becomes anti-symmetric.

The analysis must be performed in the coordinates system (x2, y2) attached to the raceway in order to assess the influence of the skew angle on the shape of the contact area and corresponding pressure distribution. In this case, the initial separation between cylindrical bodies is given by the relation (Natsumeda, 1999):

$$h(x,y)=\frac{y_1^2}{2\cdot R_{11}}+\frac{y_2^2}{2\cdot R_{21}}+d1(x_1)+d2(x_2)+x_1\cdot\tan(\beta) \tag{7}$$

After some modification, including the possible presence of a skew angle, the above relation becomes:

$$h(x,y)=\frac{(x_2\cdot\sin\alpha+y_2\cdot\cos\alpha)^2}{2\cdot R_{11}}+\frac{y_2^2}{2\cdot R_{21}}+d1(x_2\cdot\cos\alpha-y_2\cdot\sin\alpha)+ \\ +d2(x_2)++(x_2\cdot\cos\alpha-y_2\cdot\sin\alpha)\cdot\tan(\beta) \tag{8}$$

In case of zero value for the skew angle, Eqs. (5) and (8) are identical.

4. Computational model and results

The numerical contact program developed in C++ performs the pressure evaluation and the roller optimized profile derivation in the following steps:

– for a bearing with a known geometry and material structure, the maximum operating contact pressure is selected;

– for the applied load, on the possible contact area a mesh is created (with more than 2000 elements), with uniform elementary contact surfaces. The positive contact pressures are calculated based on a Conjugate Gradient Method (Polonsky and Keer, 1999);

– in the contact area, in each elementary contact stripe along rolling direction, if the elementary contact pressures are higher than Pmax, the pressure is limited to Pmax in the centre of the contact patch and redistributed based on a parabolic shape along the rolling direction, by verifying the equilibrium between the initial and final elementary load applied to the stripe; if necessary, the contact surface is extended along rolling direction;

– as final step, the equilibrium between the total applied load and the computed load is verified;

– the next step is to derive the roller's axial profile correction from the difference between contact surface normal displacements based on the initial and modified contact pressures. That allows us to obtain a roller profile made of two crowing radii: a very large one for the central part of the contact and a smaller one for the end of the roller;

– the previous steps are repeated until no significant change (within a reasonable relative error) in crowning radii is noticed.

A special subroutine was implemented in order to check if the elements located along the border of the contact domain have zero-pressures. If not, that means the initial contact is underestimated and all the process is automatically restarted with an increased contact domain. The procedure is repeated until the real contact area is fully contained in the initial estimation.

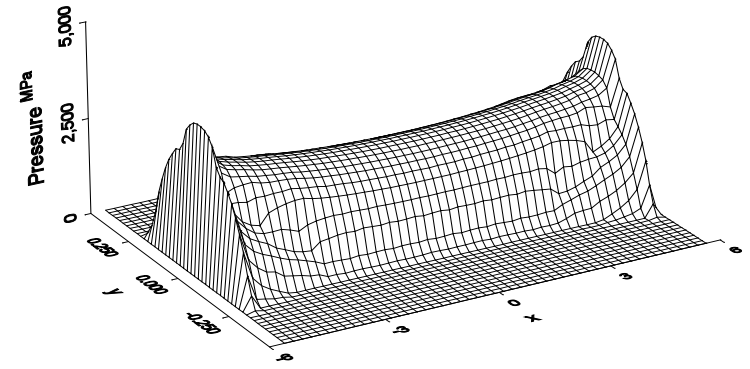
Another major problem what needs to be solved is the interference between the roller and the raceway, due to misalignment and/or skewing. This is physically impossible and if occurs, the program evaluates the maximum penetration and the separation between bodies is modified according to this value.

A cylindrical roller bearing NU 306E type was selected. The roller is cylindrical, with a diameter of 11(mm) (R_{11} = 5.5 mm) and with straight profile ($R_{12} = \infty$). The roller's length is 12 (mm) and the ends are crowned with a radius of 0.5 (mm). The raceways have also straight profile ($R_{22} = \infty$), with the radius R_{21} = 20.25 (mm) for the inner raceway and R_{21} = -31.25 (mm) for the outer raceway respectively. This type of bearing presents edge stresses even in case of moderate loads.

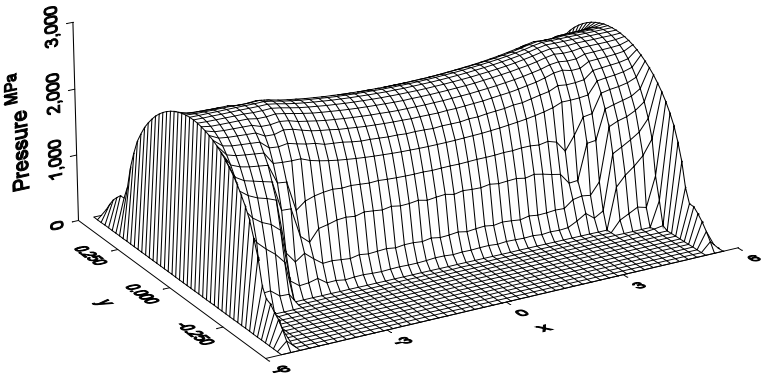
The applied load is 8000 [N] and for all three bearing steel SAE 52100 variants results were obtained at room temperature and at 100°C. Further, the

combined action of skewing and tilting of the roller was investigated in comparison with the original profile.

Fig. 3 presents the contact pressure distribution for the original contact geometry and the modified pressure distribution for a maximum contact pressure of 2550 [MPa] (100Cr6 steel variant), with material parameters at room temperature.



a)



b)

Fig. 3 – Contact pressure distribution between straight roller and inner ring a) original profile, b) pressures limited at 2550 [MPa], modified roller's profile.

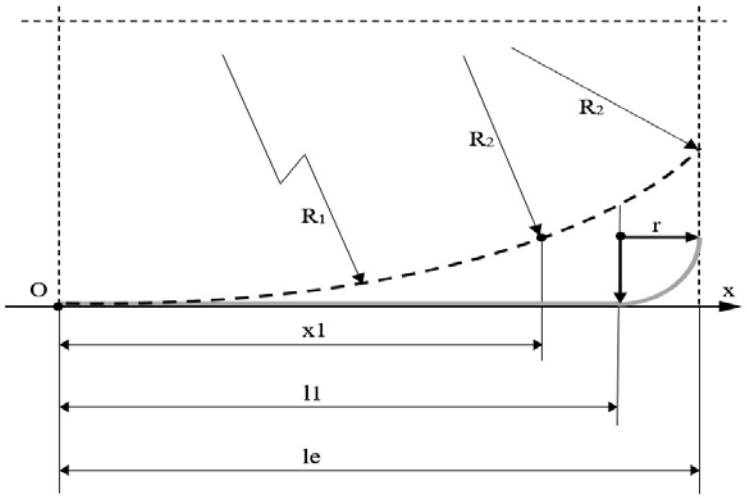


Fig. 4 – Roller with optimized profile, the continuity of the axial profile (same tangent) is imposed at x1 coordinate; l1- half-roller linear length, r=0.5 [mm], le - half-roller length, 6 [mm].

Numerical values for the crowning radii R1 and R2; and axial coordinate x1 are presented in Table 3. Since the hertzian contact pressure is 2387 [MPa], the steel variant 100Cr6 cannot be used at 100°C because of the lower value of the maximum contact pressure that assures safe-operation (1810 [MPa]); in this case only martensitic and bainitic steel structure variants can be used to derive optimized profiles.

Table 3
Optimized roller profiles for different bearing steel hardened variants and operating temperature

Temperature [°C]	SAE 52100 steel structure	Contact pressure, Pmax [MPa]	Roller profile		
			R1 [mm]	R2 [mm]	x1 [mm]
20	100Cr6	2550	13830	537	4.0673
	Ma	3000	62026	179	4.7685
	Ba	3050	807303	113	4.9088
100	100Cr6	1810	-		
	Ma	2430	13455	390	4.2075
	Ba	2500	12574	512	4.0673

Figs. 5 to 10 present the axial contact pressures in the coordinate system attached to the inner raceway in case of possible skew (α angle) and misalignment (β angle) of the roller, for the same applied load.

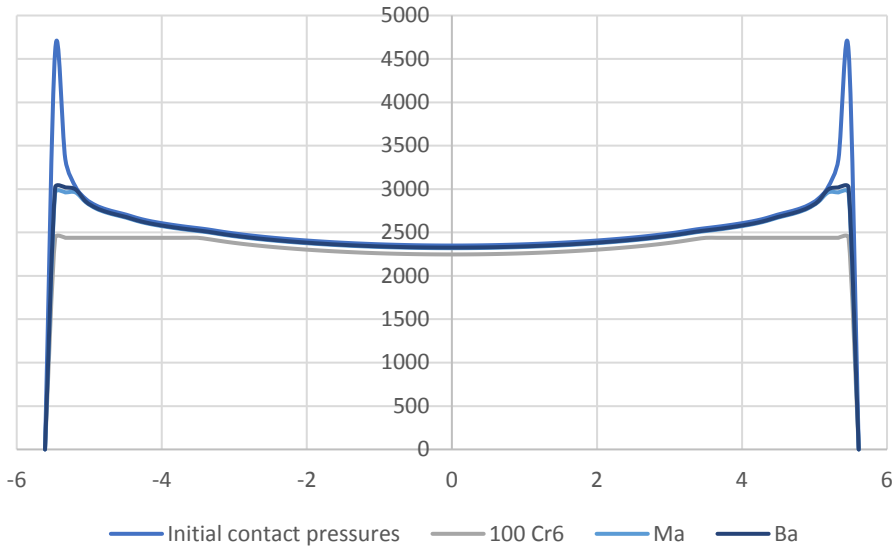


Fig. 5 – Axial contact pressures for original and optimized profiles, operating at 20°C.

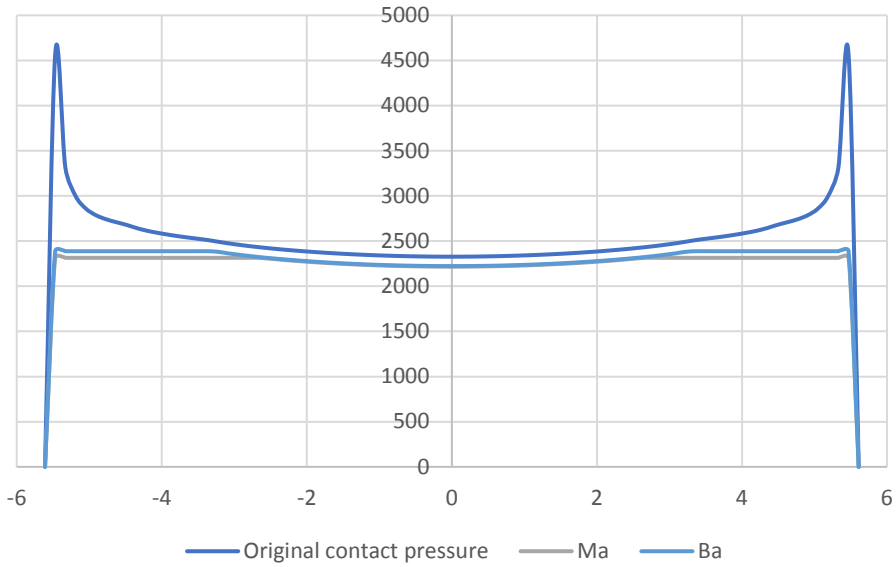


Fig. 6 – Axial contact pressures for original and optimized profiles, operating at 100°C.

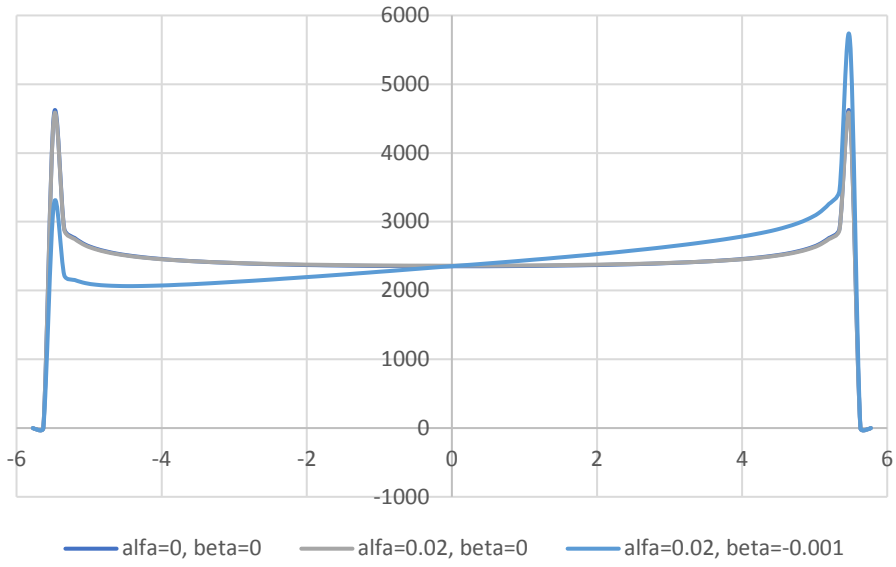


Fig. 7 – Axial contact pressures for the original profile, in case of skew and misalignment angle, 20°C.

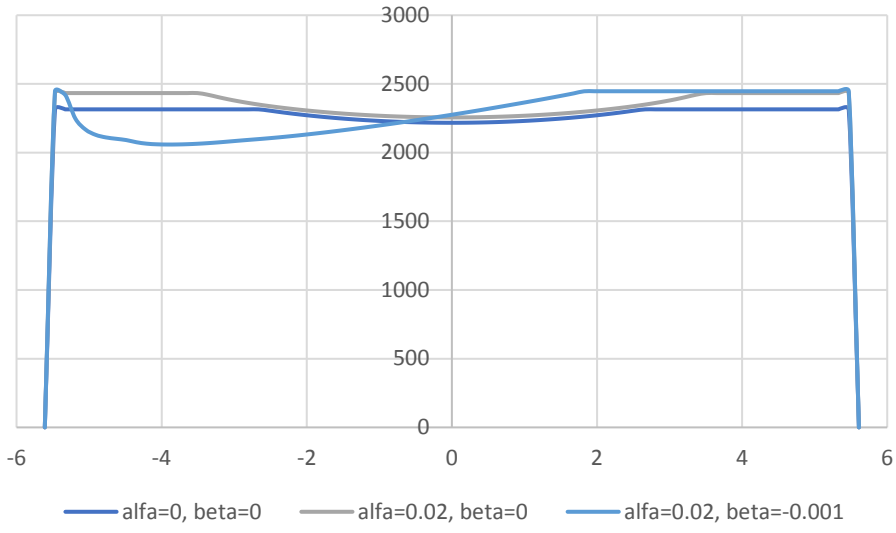


Fig. 8 – Axial contact pressures for the optimized profile, 100 Cr6 variant, in case of skew and misalignment angle, 20°C.

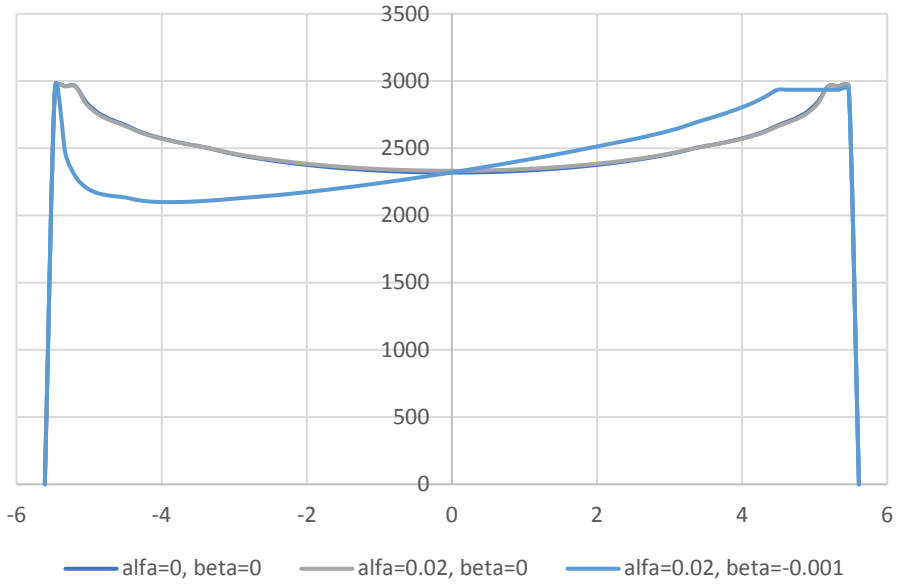


Fig. 9 – Axial contact pressures for the optimized profile, Ma variant, in case of skew and misalignment angle, 20°C.

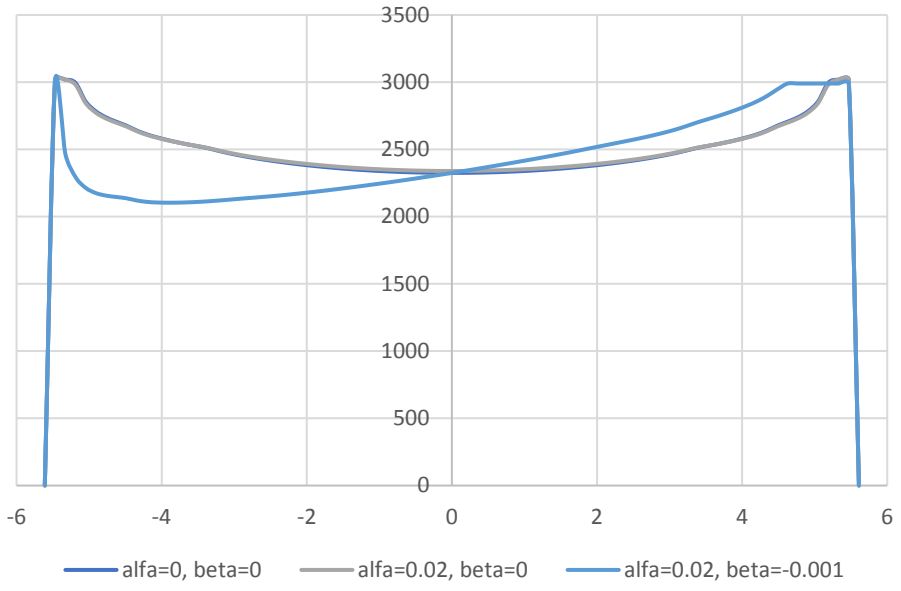


Fig. 10 – Axial contact pressures for the optimized profile, Ba variant, in case of skew and misalignment angle, 20°C.

5. Conclusions

A numerical procedure to increase the roller bearings performance is proposed. Roller optimized profiles have been computed based on the limitation of the contact pressure, taking into account the applied contact load, material structure and operating temperature. They were verified for the cases when skew and misalignment angles occur during operation, showing excellent behaviour in comparison with the original profile. The model also offers valuable information about the limitations for a steel variant, depending on the current working situation; keeping the stress and strain fields in the elastic domain.

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RULMENȚI CU ROLE CILINDRICE CU PROFIL OPTIMIZAT ÎN FUNCȚIE DE STRUCTURA MATERIALULUI ŞI TEMPERATURA DE FUNCȚIONARE

(Rezumat)

Lucrarea de față propune o metodă pentru o distribuție aproape uniformă a presiunii între elementele de rulare, bazată pe modificarea profilului drept inițial al rolei şi luând în considerare parametrii materialului legați de temperatura de funcționare. Procesul de optimizare este analizat prin includerea posibilelor rotiri și/sau unghiuri de

înclinare ale rolei din cauza condițiilor de funcționare. A fost dezvoltat un program de calculator în limbajul de programare Borland C++. Distribuția presiunii de contact este evaluată prin utilizarea unui model de analiză a contactului tridimensional bazat pe relațiile forță-deplasare Boussinesq pentru un semispațiu combinat cu metoda flexibilității. Pentru o analiză rapidă, algoritmul folosește o metodă de Gradient Conjugat pentru a determina presiunile de contact și aria reală de contact.