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OPTIMIZATION PROCEDURE OF ROLLER ELEMENTS GEOMETRY WITH REGARD TO DURABILITY OF SPHERICAL ROLLER BEARINGS

The article deals with an optimization procedure of roller elements geometry with regard to durability of spherical roller bearings. The aim of the article is to examine the impact of change of the roller elements inner geometry on durability and reliability of spherical roller bearings; the contact strain along a spherical roller by means of the Finite Element Method at contact points of components of a spherical roller bearing by means of designed 3D parametric models. The most appropriate shape of roller elements inner geometry of a bearing from the standpoint of calculated durability was determined based on results of the contact analyses.

Keywords: rolling element, contact strain, spherical roller bearing

1 Introduction

Roller bearings are an inseparable part of most machines and devices in which there takes place a rotation movement or a linear motion. There are different requirements on roller bearings. Production machines need bearings, which are able to work in high revolution, in power engineering bearings have to carry heavy loads, etc.

Development, or rather rolling bearing optimization, is conditioned by an increase of technical parameters in machines and devices. This fact refers especially to an increase of input parameters such as power and revolution, weight and volume reduction, noise level reduction, etc. However, the most important parameters requiring optimization are the bearing lifetime and reliability.

Development of new technologies introduces also new construction materials, new production techniques of semi-finished products and bearing components or new installation methods. It is important not to overlook the bearing construction. Here it is possible to perform geometry adjustment optimization. This adjustment applies especially to geometry adjustment of raceways and rolling elements in the spherical roller bearings.

2 Spherical roller bearings durability

The double-row angular spherical roller bearing has a raceway spherically ground on the outer ring. The bearing is able to accommodate very high radial loads, as well as heavy axial loads in both directions. The high radial load capacity is caused by the great number of rolling elements, the so-called spherical rollers and their close contact on the inner ring raceways [1].

Roller bearings durability depends on a revolution number, which the bearing can perform until fatigue of any of their components takes place. A peeled material is a sign of the component fatigue. Fatigue is a basic and natural way of bearing damage. It is demonstrated by the presence of small cracks under the bearing raceway surface. The depth of these cracks is usually about 0.05-0.3 mm depending on the surface curve radii of rolling elements and the bearing rings raceways. The crack depth allows the material changes, which are caused by the slide pulsating strain. This process leads to a gradual crack formation under the surface. It can take quite a long time until it is visible on the surface in a form of the peeled off material, the so-called pitting [2-3].

3 Contact strain along spherical roller in a spherical roller bearing

It is possible to calculate the intensity of the contact pressure and the size of the contact surface - effective length l_{ef} and width $2b$ from the contact pressure distribution at the most strained point in the bearing inner ring. Figure 1 shows the course projection (the curve) of the contact pressure along the contact surface l_{ef} of the contact ellipse on the bearing inner ring. The contact strain curve has been calculated using the finite element method [4].

4 Optimization of geometry of spherical roller bearing

The bearing model was simplified by axial symmetry. the bonds between the individual parts of the bearing were

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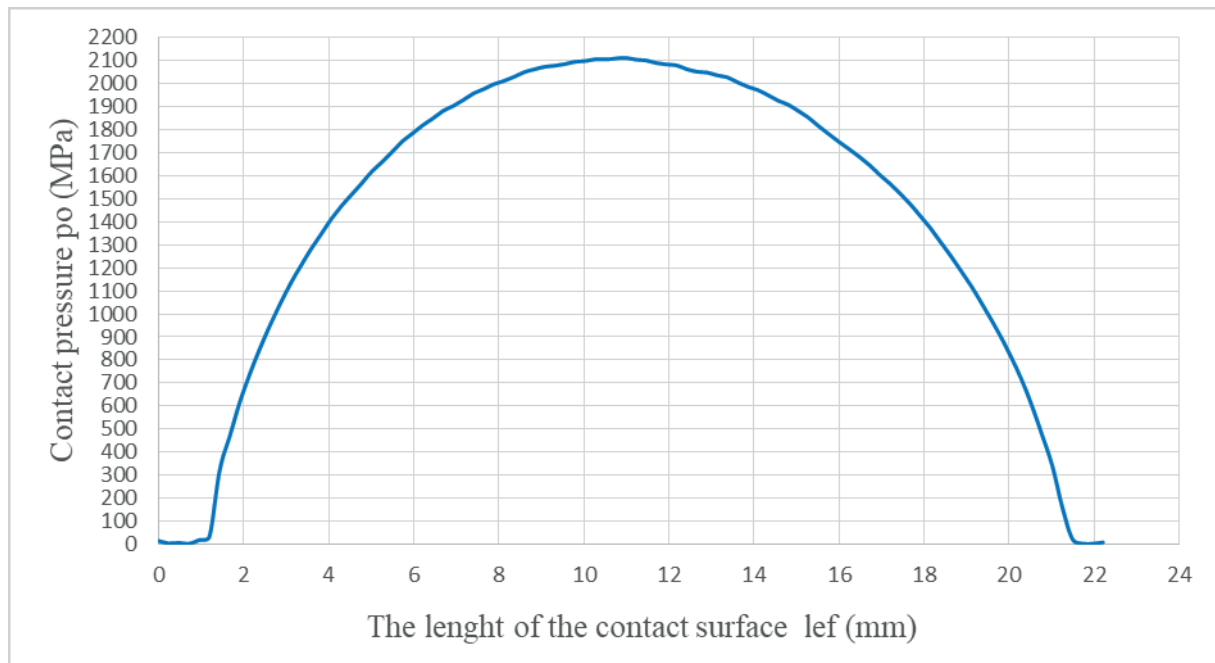


Figure 1 The contact pressure course on the inner bearing ring raceway

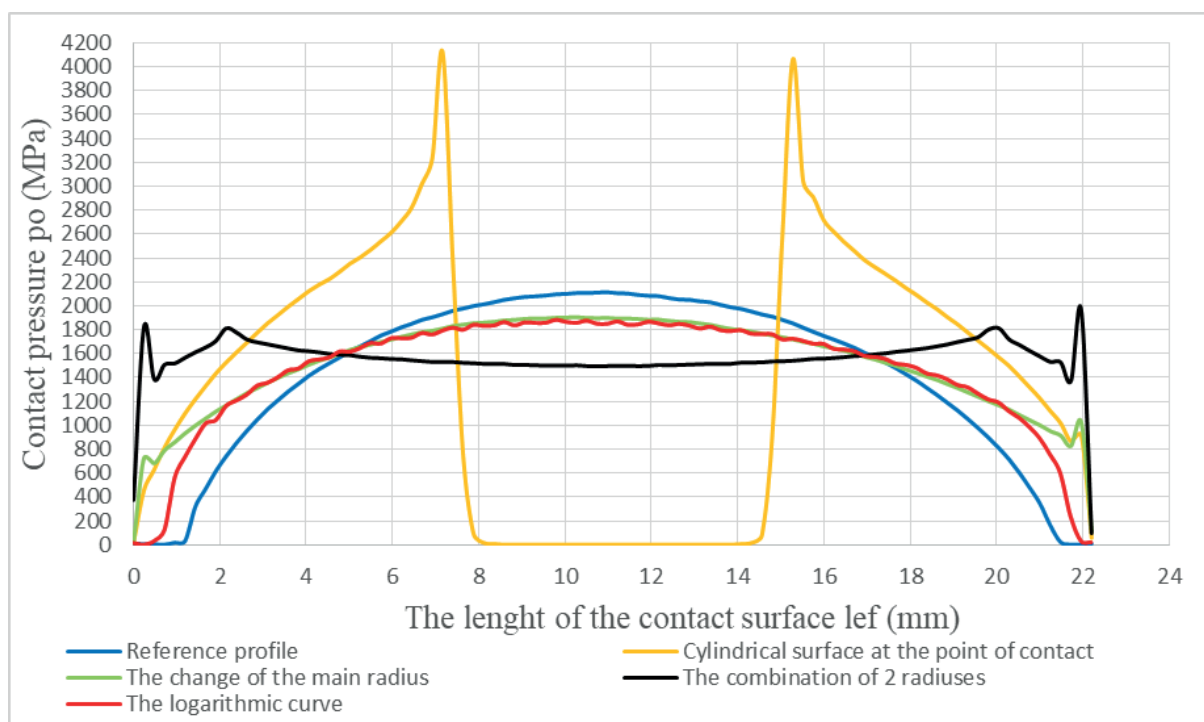


Figure 2 Comparison of the contact pressure courses in the bearing raceway of the spherical roller bearing inner ring for the rolling element analyzed geometries

replaced by contacts. Degrees of freedom were taken from the outer ring of the bearing. The rolling element load calculation is based on the Hertz theory and the reference bearing calculation data. It is possible to distribute the bearing load to individual rolling elements. The load on the selected part of the bearing was calculated to be 9 kN. The contact volumes were meshed with 0.08 mm hexahedrons. The other parts were meshed with 0.56 mm tetrahedrons. The transition edges were meshed at 0.24 mm [5-6].

The aim of optimization was a decrease of contact pressure that acts at the point of contact of the rolling element with the outer and the inner ring. The profile of a rolling element was optimized and the contact strain between the rolling elements and bearing rings was calculated, as well [7].

The three new geometries of the rolling element for spherical roller bearings were designed that were consequently compared with the reference profile. Selection of the most appropriate design of the new geometry of the

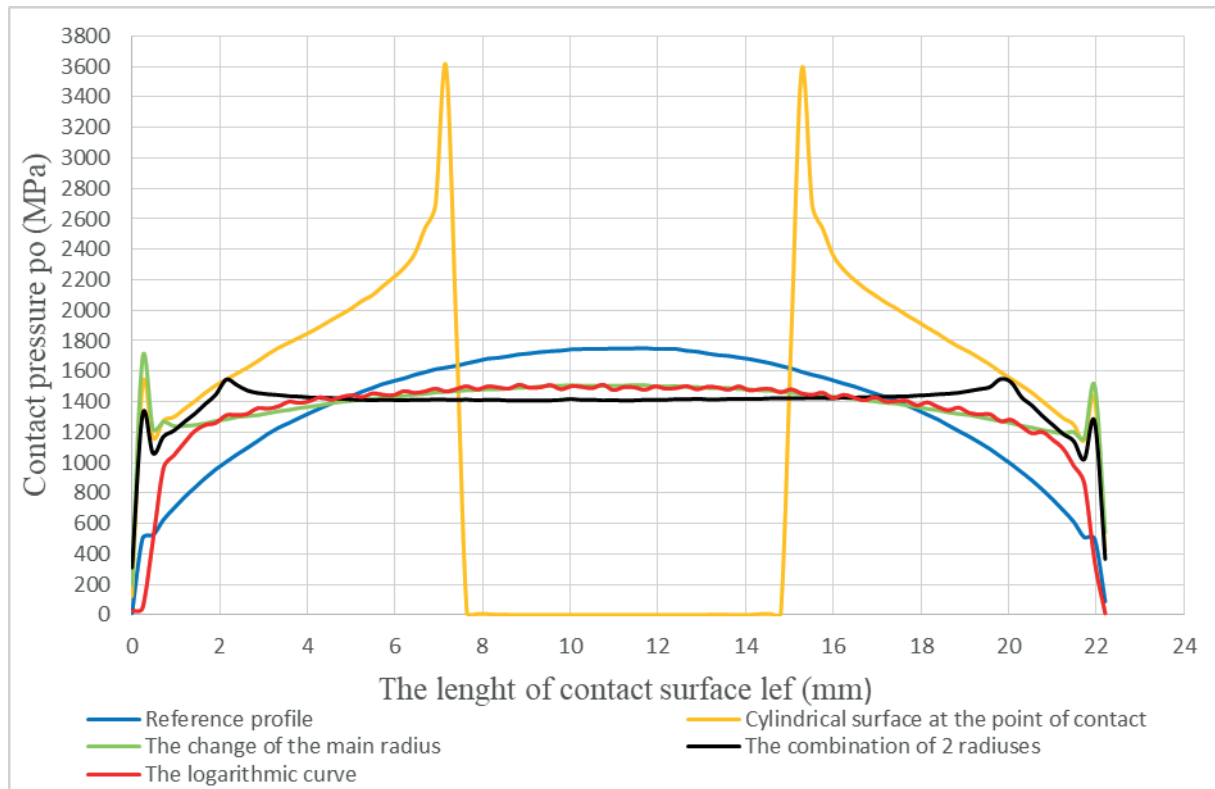


Figure 3 Comparison of the contact pressure courses in the bearing raceway of the spherical roller bearing outer ring for the rolling element analyzed geometries

Table 1 Evaluation of results of analyses in the spherical roller bearing inner ring

| Design nr. | Title | l_{ef} (mm) | b_{ef} (mm) | p_o (MPa) | σ_{max} (MPa) |
|-------------------|---|---------------|---------------|-------------|----------------------|
| Reference bearing | Profile | 21 | 0.96 | 2106.1 | 1426 |
| 1 | Cylindrical surface at the point of contact | 22.14 | 1.21 | 4022.1 | 3019.1 |
| 2 | The change of the main radius | 22.13 | 0.96 | 1921.5 | 1319.9 |
| 3 | The combination of 2 radii | 22.14 | 0.64 | 2106.1 | 1315.3 |
| 4 | Logarithmic curve | 21.9 | 0.63 | 1948.3 | 1302.6 |

Table 2 Evaluation of results of analyses in the spherical roller bearing outer ring

| Design nr. | Title | l_{ef} (mm) | b_{ef} (mm) | p_o (MPa) | σ_{max} (MPa) |
|-------------------|---|---------------|---------------|-------------|----------------------|
| Reference bearing | Profile | 21.8 | 0.8 | 1758.4 | 1426 |
| 1 | Cylindrical surface at the point of contact | 22.13 | 0.99 | 3658.6 | 3019 |
| 2 | The change of the main radius | 22.14 | 1.21 | 2048.7 | 1319.9 |
| 3 | The combination of 2 radii | 22.14 | 1.18 | 1719.7 | 1315.3 |
| 4 | Logarithmic curve | 22 | 0.64 | 1576.9 | 1302.6 |

rolling element was based on comparison of the contact pressures on bearing raceways of the inner and outer bearing rings.

The comparison of curves of the rolling elements contact pressures depends on the length of the contact surface l_{ef} . Figure 2 (inner ring) and Figure 3 (outer ring) show the curves' shape. A decrease of contact pressure on

bearing raceways of bearing rings was obtained in all the designs of a new geometry of the rolling element [7-8].

As shown in Figures 2 and 3, the lowest contact pressure acts between the two bearing rings and the rolling element with new geometry 4. At the same time, the contact pressure that acts between the bearing rings and the rolling element does not produce maximum strain

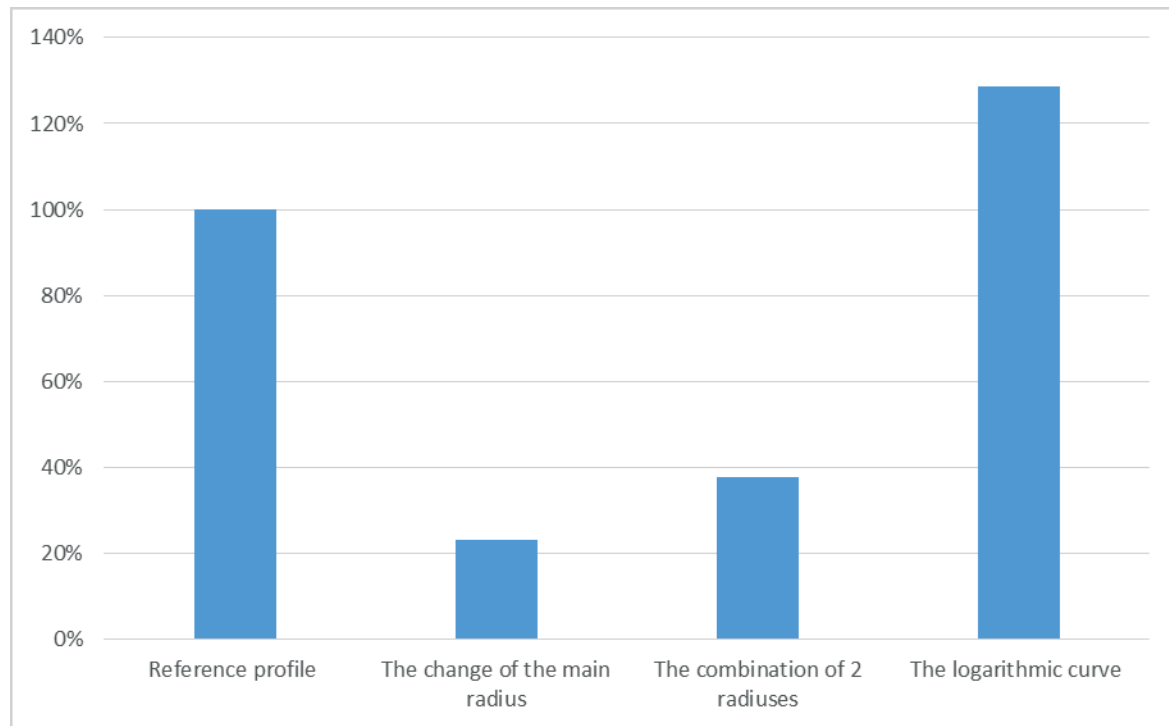


Figure 4 Comparison of calculated lifetimes of analysed geometries

values that negatively affect the bearing durability. The new geometry of the spherical roller bearing formed by the logarithmic curve is the most appropriate for optimization of the spherical roller bearing regarding its durability and lifespan [8].

Evaluation and selection of the most appropriate design of the new geometry are shown in Table 1 and Table 2, respectively.

For the better evaluation of analysed geometries durability of individual analysed geometries was calculated according to the Lundberg-Palmgren theory:

$$\ln \frac{1}{s} \approx A * \frac{N^p * \tau_0^c * V}{z_0^h}, \quad (1)$$

$$\tau_0 \approx 0.256 * p_0, \quad (2)$$

$$z_0 \approx 0.25 * 2b, \quad (3)$$

where: S is the probability of survival,

N is the number of load cycles,

V is the stressed volume,

e, c, h, A are material constants defined by experiments,

p_0 is the pressure present at the contact point,

$2b$ is the minor axis of the ellipse [9].

Calculation of a total lifetime of the bearing was based on partial lifetimes of bearing rings. As far as a logical comparison is concerned, 100 % is assigned to the reference geometry.

In a new design 1 (Cylindrical surface at the point of contact), the calculated pressures are well above those of the reference bearing. Therefore, the analysis of the new design 1 is excluded from further comparison.

Comparison of the calculated lifetimes of analysed geometries is shown in Figure 4 [10].

5 Conclusions

Spherical roller bearings can be optimized by modification of the geometry of the rolling element, i.e., the spherical roller. The most appropriate geometry seems to be the one formed by the logarithmic curve after a comparison of lifetimes of the bearing with the new geometry of the rolling element (Figure 4). The logarithmic curve is described by equations, while in this case a parameter of a loss of the logarithmic curve profile, i.e., a modified surface of the spherical roller. The optimal value of the parameter is 0.00035 mm which is similar as in the case of the rolling bearing with the logarithmic profile.

The new geometry of the rolling bearing composed by the logarithmic curve increases the total carrying capacity and thus bearing durability by more than 25 %. This new geometry does not form the strain peaks that negatively affect the total bearing durability.

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