BULETINUL INSTITUTULUI POLITEHNIC DIN IAŞI
Publicat de
Universitatea Tehnică „Gheorghe Asachi” din Iaşi
Volumul 68 (72), Numărul 2, 2022
Secţia
CONSTRUCŢII DE MAŞINI
DOI:10.2478/bipcm-2022-0018

# STUDY UPON CYLINDRICAL ROLLER BEARING PROFILE OPTIMISATION USING FEA

BY

**ILHAN MEHMET[1], DRAGOŞ FLORIN CHITARIU[2,*] and CĂTĂLIN GABRIEL DUMITRAŞ[2]**

[1]Kombassan Insaat Tarim Ve Sanayii Isletm, Konya, Turkey
[2]“Gheorghe Asachi” Technical University of Iaşi,
Faculty of Machine Manufacturing and Industrial Management, Iaşi, Romania

Received: June 16, 2022
Accepted for publication: June 29, 2022

**Abstract.** The bearings represent an important element in the mechanical structures, ensuring the proper functioning of the moving / dynamic elements. An important feature of the bearing is its durability, which is necessary for long-term use. This characteristic is influenced by the contact between the rolling element / ring and the rolling elements / balls or rollers. This paper proposes a comparative study of the influences of different types of cylindrical roller profiles proposed in the literature and by the author on the durability characteristic. There are 3 types of profiles, standard, Fujiwara and a profile calculated by the author based on the solid of equal strength.

**Keywords:** bearing, roller profile, optimisation, finite element analysis.

*Corresponding author: *e-mail*: dragos-florin.chitariuiu@academic.tuiasi.ro
© 2022 Ilhan Mehmet et al.
This is an open access article licensed under the Creative Commons Attribution-NonCommercial-NoDerivatives 4.0 International License (CC BY-NC-ND 4.0).

## 1. Introduction

The bearings serve to support the shafts and axles that rotate in space, providing them with the opportunity to rotate or roll freely and take over the loads that act on them. In addition to axles and shafts, the bearings can support the parts that rotate around fixed axles, for example drive wheels, sprockets, etc. (Rabinovci, 1977). There are many types of bearings but the ones that we are focus on are the radial bearings with cylindrical rollers.

The radial bearing with short rollers (Fig. 1a) is intended for large radial loads, its load capacity being equal on average greater than 1.7 times the load capacity of the radial bearing on a single row of balls. This bearing is easily disassembled in the axial direction, allows some axial movement of the rings and is therefore advantageous in case of large thermal deformations of the shafts or when axial self-adjustment of the shafts is necessary, for example shafts on which V-toothed wheels are fixed. It is made with stamped cage (Fig. 1b) or, in case of high speeds, with massive cage (Fig. 1a) the maintenance of the equipment is established by a program which is followed, the only problem is the necessary budget for the maintenance and its obtaining. The next approach is defined by reducing the costs in terms of maintenance and rules of the safety of the equipment. The assessment of the equipment and the adaptation plan of the maintenance with respect to the necessary improvements to the equipment. This is the situation were the maintenance operations present a well-established role (Rabinovici, 1972).



a) b) c)

Fig. 1 – Radial bearings with cylindrical rollers type NU and NJ - catalogue URB.

One of the most important characteristics of the bearing is the maximum durability in the above mention working conditions. This can be achieve due to the increase of the tension in the cylindrical rollers. In order to calculate the distribution and the values of the tensions in rollers we have to take into the account the contact problem.

Two rotating bodies in contact can make a point or linear contact depending on their curves in the contact area. Under the action of a compressive force, the two bodies will deform elastically or plastically, creating a contact surface.

The analytical determination of the stresses and deformations that appear in the contact area is made by applying the theory of Hertz H. and subsequent developments.

The simplifying hypotheses used are: the bodies in contact are considered homogeneous and isotropic; materials will deform elastically; the force acting on the bodies is perpendicular to the contact surfaces; tangential stresses are not considered; the area of the contact surface is reduced compared to the dimensions of the bodies in contact; the contact surface is flat (Rabinovici *et al*., 1977).

In order to achieve the ideal linear contact, the lengths of the two bodies must be equal. In this case, the normal stress distribution has the shape of a semi-elliptical cylinder as in Fig. 2. The value of the maximum normal stress for the linear contact can be determined:

$$\sigma_{max} = 2Q/\pi lb\,. \tag{1}$$

and the normal stress at any point in the contact area can be determined:

$$\sigma = \sigma_{max}\sqrt{1-(y/b)^2}\quad . \tag{2}$$

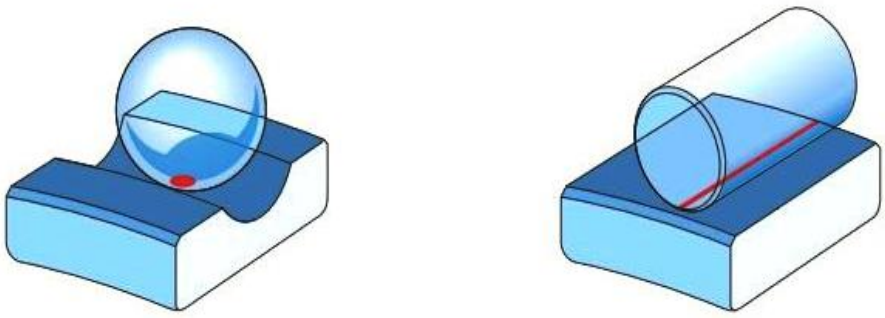


Fig. 2 – Considered contact cases for the bearings.

The half-width of the contact surface can be determined:

$$B = 3{,}33\ x\ 10^{-6}\sqrt{Q/l\sum\rho}\,. \tag{3}$$

The corresponding contact deformation can be determined:

$$\delta = 1{,}504\frac{x10^{-10}Q^{0.9}}{l^{0.8}}\quad . \tag{4}$$

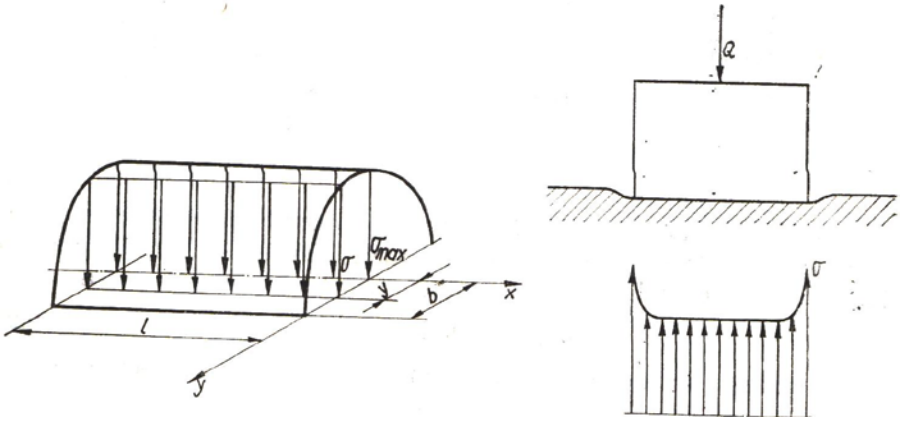



Fig. 3 – Stress distribution for linear contact.

In Fig. 3 one can observe the stress distribution on the cylindrical roller. This distribution lead to poor durability at the ends of the roller and it is of a great importance to determine a profile that reduce the amount of stress at the end of the roller. In the literature are presented several profile that intend to achieve this goal.

The aim of the present article is to identify the optimum profile which lead to an increase of the durability.

## 2. Analytical Approach on the Cylindrical Roller Profile

Following the documentary analysis, the main types of roller profiles used in practice by bearing manufacturers and studied by researchers in the field were identified:

1. standard profile according to ISO / TS 16281;
2. logarithmic profile combined with cylindrical profile;
3. modified logarithmic profile;
4. convex profile;
5. profile with partial bulging;
6. profile with two tangential rays;
7. profile with bevels on the extremities.

Standard logarithmic profile according to ISO / TS 16281- Roller bearings. Methods for calculating the modified modified reference service life for bearings with universal loads (**, 2008).

The logarithmic profile was developed using mathematical tools as well as finite element analysis computer programs. Logarithmic profile name comes from the mathematical expression used to generate the profile. Regardless of the loading conditions, the logarithmic profile uses a larger contact area / generator longer than the incomplete bulging profile and the full bulging profile.

In the case of coaxiality deviations, the load tends to avoid tension concentrations on the ends of the roller.

The default values of the standard are used where a logarithmic profile for rollers and an unmodified track profile is used.

The calculation relationship for the roller profile / discharge at the end of the roller can be expressed by the relation:

$$P(x_k) = 0.00035 \cdot D_w \cdot ln\left[\frac{1}{1-(2\cdot\frac{x_k}{L_{we}})}\right] \quad (5)$$

Where with

$P(x_k)$ - discharge at the end of the roller - the difference between the rectilinear and the logarithmic generating profile for the point $x_k$;

$x_k$ - note the position relative to the center of the roller;

$D_w$ - roller diameter;

$L_{we}$ - the length of the roll.

The general shape of a logarithmic profile roll is shown in the Fig. 4.

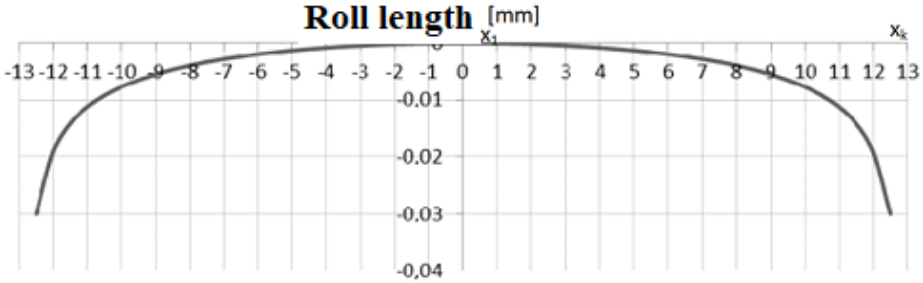



Fig. 4 – Logarithmic profile detail.

However, when this equation is applied, performing calculations using the technique described can in some cases lead to tension on the ends of the roller. This trend becomes more remarkable in the case of deviations from parallelism / coaxially.

Making a rectilinear profile of the generator for cylindrical or conical rollers are preferable for easy processing. The equation does not allow the configuration of a section with profile on rectilinear generators.

To solve this problem Fujiwara, Kawase modified the relationship presented above are introduced in the Johns-Gohar function three parameters, K1, K2 and zm. They have the following meanings: K1: load factor, K2: the ratio between the length of the bulge and the actual contact length (Fujiwara *et al*., 2009).

$$y_m = \frac{1}{2}\sqrt{\frac{1-\exp(-zm\frac{\pi lE'}{2K_1Q})}{1-0{,}3033K_m b/a}} \tag{6}$$

In Fig. 5 we present the profile obtained with the help of the above equation. In Fig. 6 we present a graphic obtained by calculating in established points (Fujiwara *et al*., 2010).

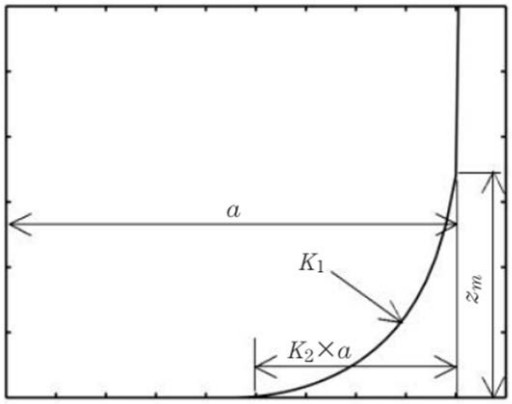



Fig. 5 – Graphical representation of the parameters proposed by Fujiwara and Kawase.

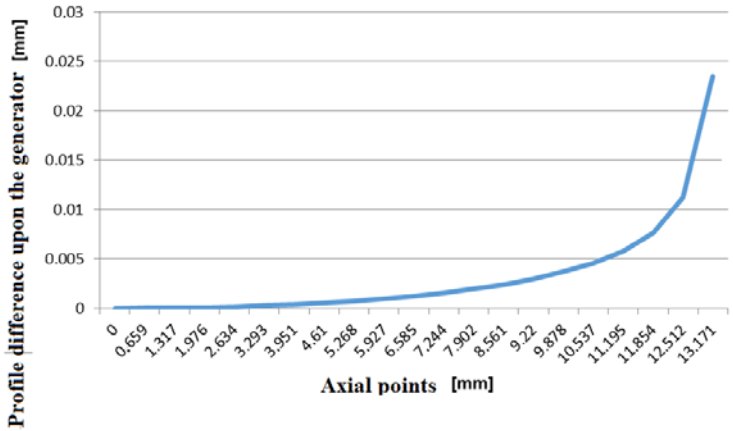



Fig. 6 – Representation of the Fujiwara profile on a section.

We propose a profile which will take into account the equal stress solid body (Mehmet *et al*., 2021). The equation which will define this profile is a hyperboloid of rotation. In Fig. 7 we present the profile calculated using the same points as in Fig. 6 but using the new determined equation.

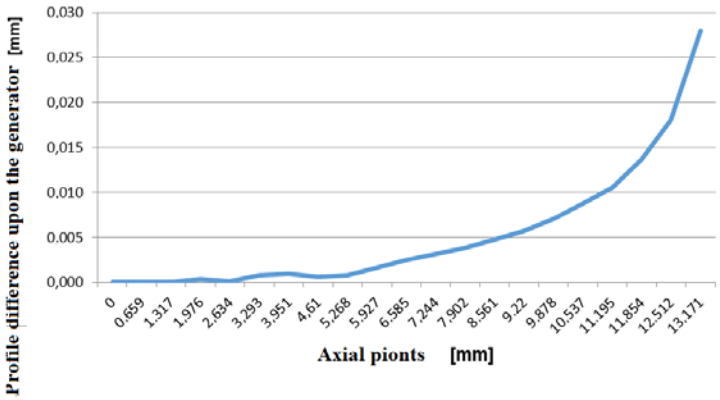



Fig. 7 – Representation of the profile on a section based on its author mathematical model.

In Fig. 8 one can observe the difference between calculated profiles.

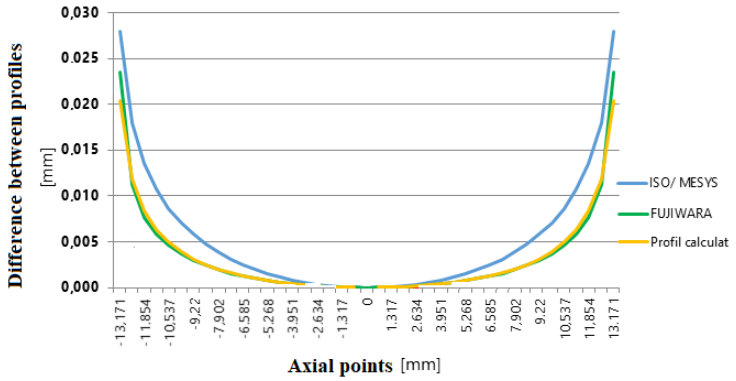



Fig. 8 – Different representation of the profiles.

It can be seen from Fig. 8 that the profile which is proposed by the author have almost the same shape as the profile proposed by Fujiwara.

## 3. The Finite Element Model

We present the finite element model of the bearing NU318 used in order to study the state of stresses and deformations.

In order to perform the finite element analysis, a series of steps must be completed, which requires the choice and introduction of input parameters in the finite element analysis program.

During the stage regarding the elaboration of the 3D models of the cylindrical rollers $\phi$ 30x28 a special attention must be paid for the correct realization of the profile. Following the calculations performed in the above

section, the x, y coordinates of the profile generator can be determined, which is the bulging of the roller with precision of hundreds of micrometers. During the manufacturing process these different profiles for generators - bulges can be obtained / materialized with an accuracy of 0.2 μm.

The finite element analysis program chosen is Ansys V15.0 Workbench module, and the type of analysis used was Static Structural. The pre-processing stage is presented below. First we create the CAD model of the bearing, Fig. 9.

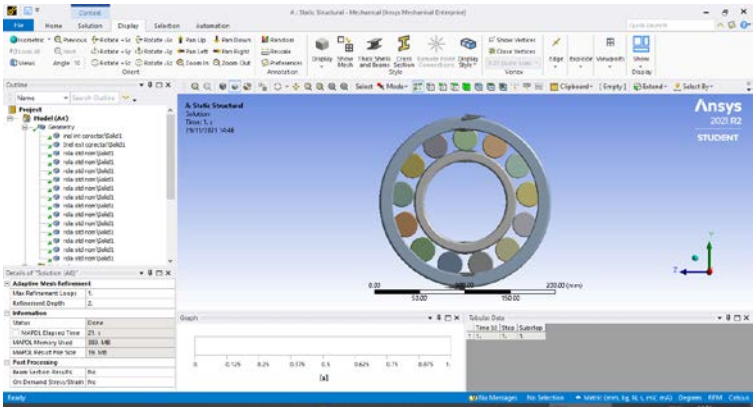



Fig. 9 – The CAD model of the NU 318 bearing.

We create, based on the calculus mention above, the model of 3 types of cylindrical roller:

- external cylindrical profile ϕ28,
- theoretical profile proposed by author,
- profile based on Fujiwara equation.

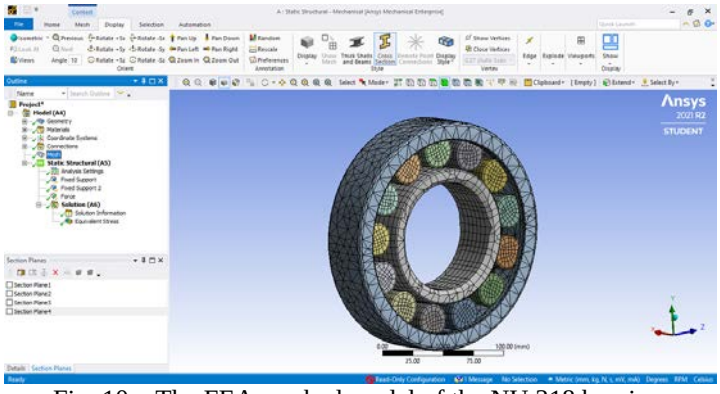



Fig. 10 – The FEA meshed model of the NU 318 bearing.

In the meshed stage of the model, 58763 nodes between elements and 24153 elements were used, Fig. 10.

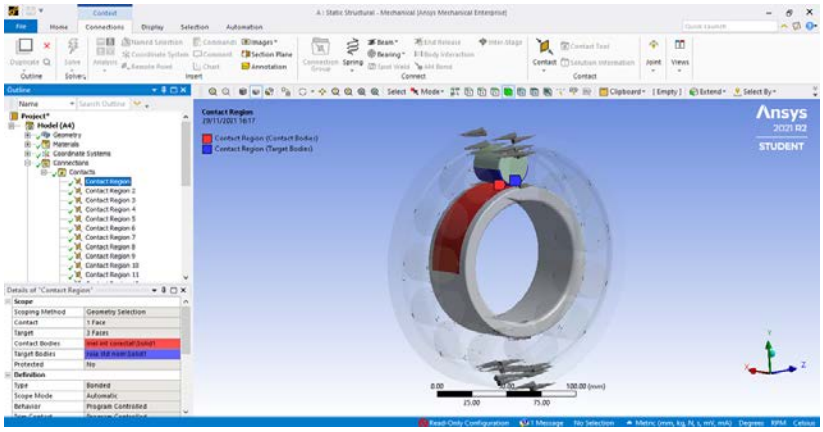



Fig. 11 – The contact settings between the inner ring and the cylindrical roller.

In Fig. 11 one present the 26 contact zones between the inner ring and the cylindrical roller.

## 4. Results of the Finite Element Analysis

After running the simulation we obtain the following results.
In Fig. 12 we present the results for the standard cylindrical roller case.

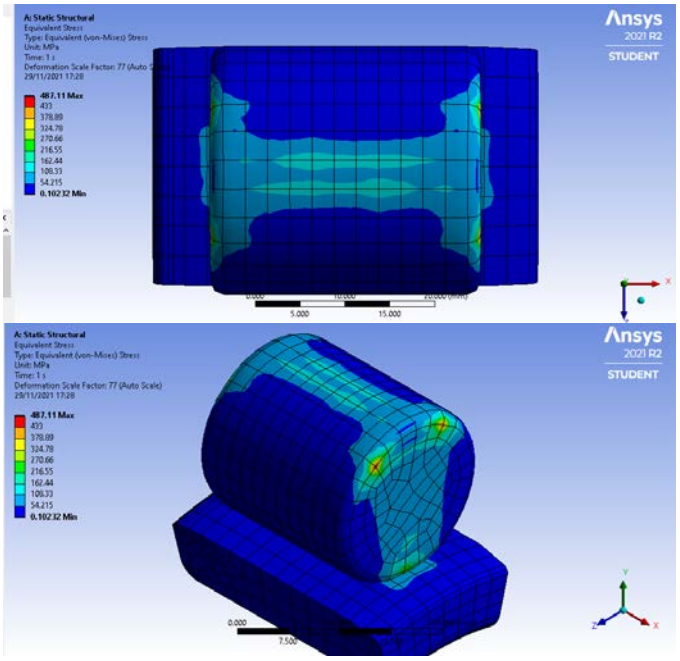



Fig. 12 – FEA results for the cylindrical roller using standard profile.

In Fig. 13 it is shown the behavior of the proposed profile roller.

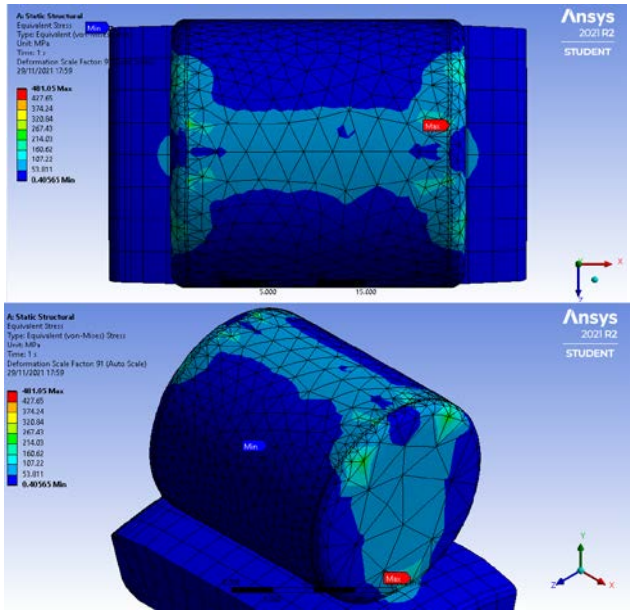



Fig. 13 – FEA results for thecylindrical roller using proposed profile.

In Fig. 14 we have the results of the Von Mises stress for the cylindrical roller with Fujiwara profile.

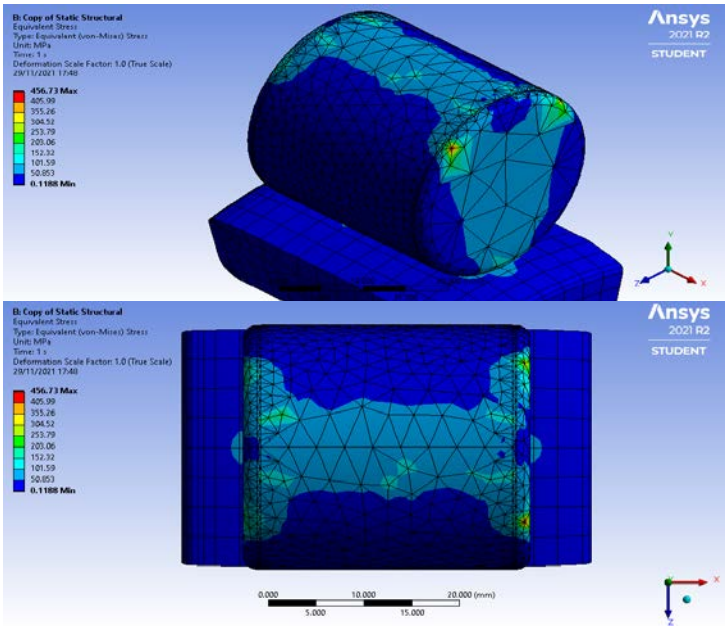



Fig. 14 – FEA results for thecylindrical roller using Fujiwara profile.

In Table 1 on synthesis the results in order to indicate the most appropriate result.

**Table 1**

*Results – Von Mises stress*

| Profile type | Applied force [N] | Von Mises stress [MPa] |
|---|---|---|
| standard profile | 31500 | 487 |
| proposed profile | 31500 | 481 |
| Fujiwara profile | 31500 | 456 |

The maximum stress obtained in the contact location, that allow an assessment of the difference between the 3 profiles, indicate a value of the maximum Von Mises stress for the cylindrical roller profile of 487 MPa at a stress on the outer ring of 31500N. The most efficient profile identified is the one proposed by Fujiwara where the maximum contact voltage is 456 MPa.

Finite element analysis also allows verification of the distribution of stresses in addition to the calculation of maximum values.

As the difference between the proposed profiles is of the order of micrometer fractions it is impossible to perform analytical calculations. Only finite element analysis can determine differences in contact voltages and their distribution.

## 5. Conclusions

It is necessary to develop and experiment the known models for cylindrical roller profiles. Generic cases of optimization are presented in the literature but it is necessary to apply the extension of known calculation models and adjust the known parameters for a wide range of bearing types. Also, it is important to verify the relationships identified on other common materials in the bearing industry and correction of parameters. Experimental testing of generated profiles. Numerical modeling research shows that discontinuities along the roller profile generate strong increases in pressure distributions that can lead to slight changes in the roller profile.

As the difference between the proposed profiles is of the order of micrometer fractions it is impossible to perform analytical calculations. Only finite element analysis can determine differences in contact voltages and their distribution.

Using finite element analysis, the maximum stresses that appear at the ring roller contact were identified for the 3 profiles: standard $\phi$ 28, proposed theoretical profile, Fujiwara.

The best profile in the comparative analysis was the one proposed by Fujiwara.

## REFERENCES

Fujiwara Hiroki, Yamauchi Kazuto, *Tolerance Design of Logarithmic Roller Profiles in Cylindrical Roller Bearings*, Journal of Advanced Mechanical Design, Systems, and Manufacturing, Vol.4, No. 4, 2010, pp.728-738.

Fujiwara Hiroki, Kawase Tatsuo, Op*timized logarithmic roller crowning design of cylindrical roller bearings and its experimental demonstration, roceedings of the* ASME/STLE 2009 International Joint Tribology Conference IJTC2009, October 19-21, 2009, Memphis, Tennessee, USA, IJTC2009-15032.

Mehmet I., Dogan G, Chitariu D.F., Dumitraș C., Negoescu F., *Research on advances in roller bearing manufacturing*, ModTech 2021 IOP Conf. Series: Materials Science and Engineering 1182 (2021) 012045, IOP Publishing, doi:10.1088/1757-899X/1182/1/012045.

Rabinovici I. *et al*., Rulmenți, Ed. Tehnică, București, 1972.

Rabinovici I. *et al*., Rulmenți, Ed. Tehnică, București, 1977.

*** Technical specification ISO/TS 16281. Rolling bearings – Methods for calculating the modified reference rating life for universally loaded bearings, 2008.

## STUDIU PRIVIND OPTIMIZAREA PROFILULUI ROLELOR CILINDRICE DE RULMENT UTILIZÂND FEA

(Rezumat)

Rulmenții reprezintă un important element în structurile mecanice, asigurând buna funcționare a elementelor în mișcare/dinamice. O caracteristică importantă a rulmentului este durabilitatea lui, acest lucru fiind necesar pentru o utilizare îndelungată. Această caracteristică este influențată de contactul între elementul de rulare/ inel și elementele de rostogolire/bile sau role. Lucrarea de față propune un studiu comparativ a influențelor diferitelor tipuri de profile de role cilindrice propuse în literatura de specialitate și de către autor asupra caracteristicii de durabilitate. Se propun 3 tipuri de profile, standard, Fujiwara și un profil propus de către autori, bazat pe solidul de egală rezistență.