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NUMERICAL STUDY OF THE ROLLING TENDENCY

SORINEL TODERAȘ SIRETEAN, ILIE MUSCĂ, LUMINIȚA IRIMESCU

Abstract: *Rolling motion is very common in mechanical devices. During this motion, rolling friction occurs and affects the efficiency of the mechanical transmissions. A limit situation of this motion is represented by the pre-rolling phase, when the rolling torque is lower than the rolling friction torque and the body has only a rolling tendency. This paper present a numerical study, based on finite element method, of a concentrated contact with rolling tendency, in order to better understand how rolling motion start and how it is affected by the friction. Some conclusions concerning stress, strains and contact area are presented too.*

Key words: *finite element, rolling, contact mechanics*

1. Introduction

It is well known that rolling movement is present in most machines and mechanisms. Rolling friction, as complex physical-chemical process, has a negatively influence for the rolling movement and mechanical transmission efficiency. A particular case is represented by pre-rolling phase *when the rolling torque is lower than the rolling friction torque and the body has only a rolling tendency.* This article presents, based on the finite element method, a model of a concentrate contact between a ball and an outer ring of a bearing. Stress distributions and their evolution under the action of normal appropriation and the bearing ring rotation is also analysed.

2. Theoretical aspects concerning rolling friction coefficient

Considering a ball, freely positioned on the bearing outer ring raceway, according with Figure 1, when the ring does not rotate, the centers of rotation of the ball and the bearing ring center are located on the same vertical axis. The ball have a roll tendency, if the bearing ring start to rotate. So that, when the rolling torque is less than friction torque, the ball having just a tendency to roll, Figure 2 determine that in the ball center act a traction force, *F*, that will cause appearance in the point of contact, a shear force, *T*. Under these conditions the equilibrium equations of the system, are, [2]:

$$\begin{cases} T - F = 0 \\ N - G = 0 \\ F \cdot r = 0 \end{cases} \tag{1}$$

where *r* is the radius of the ball.

From the equilibrium equation of the moments, we have $F = 0$, but experimental results shows that the system can be in equilibrium even in the case where $F \neq 0$, while the value of the force *F* does not exceed a certain value. Ring rotation with a controlled speed causes contact friction to act as a traction effect on the ball which will be moved out of static equilibrium so that the center of the ball occupies a position on a direction that forming an angle α with the vertical [3].

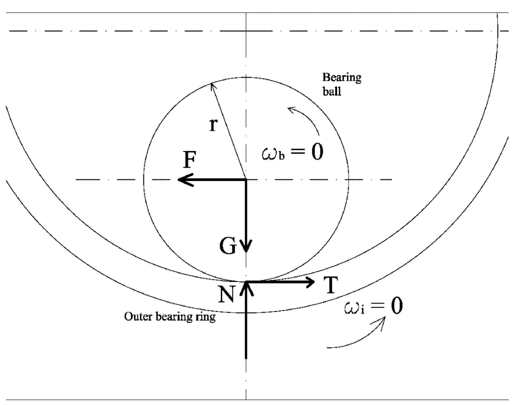

Fig. 1 *Free contact of the ball and the ring.*

Due to the ball rolling tendency there is an asymmetry of the pressure distribution near the point of contact which results in the movement of the application point of the normal reaction *N*, by a distance *s* in the direction of ball rolling. The value of this movement, *s*, is usually called rolling friction coefficient, Figure 2 [4].

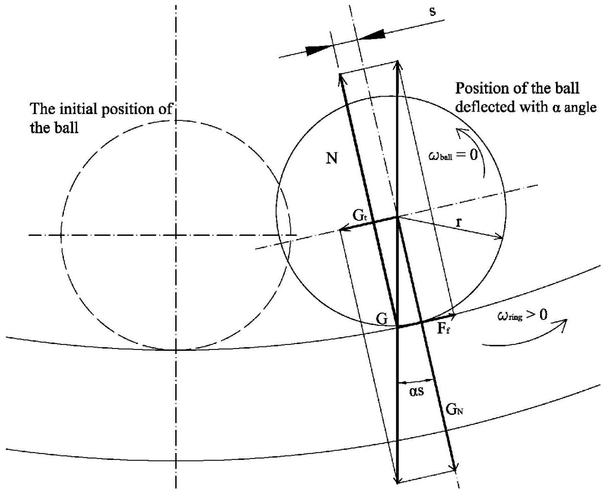

Fig. 2 *Ball-ring contact during the pre-rolling case.*

From the equilibrium of the forces and moments acting on the ball in new position, results, [2]:

$$\begin{cases} G_t - F = 0 \\ N - G_N = 0 \\ M_g - M_{fr} = 0 \end{cases} \tag{2}$$

where:

$$G_t = m \cdot g \cdot \sin\alpha \tag{3}$$

is the tangential component of the gravity force acting in the center of the ball, α is the deviation angle, between the vertical axis and the direction determined by the center of the ring and ball rotation center in the new position, as it can be seen in Figure 2,

$$G_N = m \cdot g \cdot \cos\alpha \tag{4}$$

it is the normal component of the force of gravity acting in the center of the ball.

$$M_g = F \cdot r \tag{5}$$

r is radius of the ball.

$$M_{fr} = N \cdot s \tag{6}$$

is the rolling friction torque of the ball, and *s* is the rolling friction coefficient.

Substituting *N* and *F* into the equation of moments from (2), we have the following equation for the rolling friction coefficient [2]:

$$r \cdot m \cdot g \cdot \sin\alpha = s \cdot m \cdot g \cdot \cos\alpha$$
$$s = r \cdot \frac{\sin\alpha}{\cos\alpha}$$
$$s = r \cdot tg\alpha \tag{7}$$

This relationship shows that the deviation angle corresponding to the position of the ball relative equilibrium is directly correlated with the rolling friction coefficient [1].

3. Numerical modeling with finite element method for the contact between a ball and a bearing ring

Numerical modelling with finite element method, in this case, involves to model the geometry of bodies in contact, meshing, setting the boundary conditions, loads, defining contact parameters, analysis of the created model and results processing.

For the contact model analysis, will use the FEMAP (Finite Element Modeling And Postprocesing) Version 10.3 product of Siemens Product Lifecycle Management Software Inc. [7].

In order to allowing to compare experimental and numerical results, it was considered a ball freely positioned on a raceway of a bearing outer ring. Bodies are isotropic and made of steel, with the following characteristics, Young's modulus $E_1=E_2=E=2{,}1\cdot10^{11}$ *Pa*, Poisson's ratio $v=0{,}3$ and density ρ =7850 kg/m^3. We consider also the *y*-axis in the vertical direction and the ring rotate arround the *z*-axis.

Also consider that the ball has a displacement in the opposite direction of the y-axis and the ring revolve around the *z*-axis after the displacement of the ball on the *y*-axis.

To diminish the computation time based on the fact that the contact stresses have largest values near the contact and less otherwise, it is considered that only a portion of the ball and the ring can be modelled. These parts are obtained by cutting the sphere with four parallel planes at a distance of 0.5 mm from the center of the sphere parallel with the y-axis and the ring model is obtained by cutting the ring with two parallel planes located at a distance of 1mm from the center of the ring, that is located on the y-axis.

Meshing was performed using hexahedral elements after previously was used "Size on solid" function to create fixed size elements, uniform throughout the bodies volumes [6].

Constraints are applied on surfaces, curves and independent nodes, RBE2, rigidly connected to bodies nodes. Thus, the nodes located on the sides of the two bodies can not be rotate in the *x*-axis and y-axis. The nodes located on the ball central axis in the *z* –axis direction can not travel in the direction of x-axis, Figure 3. Nodes of faces ring the *x*-axis direction are rigidly connected to node from the geometric center of the ring and can only have rotation by *z*-axis.

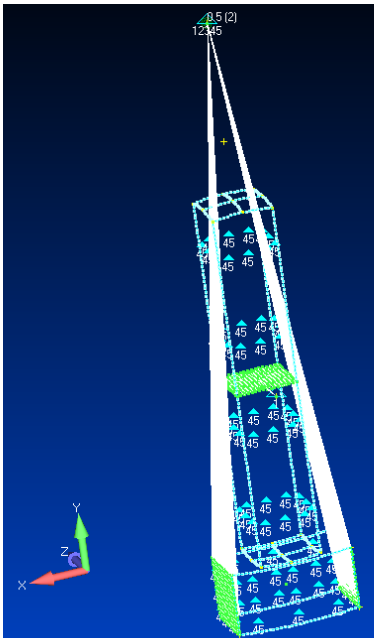

Figure 3 *Model constraints setting (1,2,3 constrained translations on axes O*x*, O*y*, O*z *and with 4,5,6 constrained rotations around the axes* O*x*, O*y*, O*z*).*

In this analysis was used a type of load by displacement, in reverse direction of the *y*-axis, simulating the force of the ball gravitational force and moment around the *z*-axis, applied in RBE2 node from geometric center of the bearing ring to simulate ring rotation.

The analysis of the model was carried out in 100 steps, for a period of 2 s with a step of 0,02s. The displacement is linear increased from 0 to the maximum value from the start to the time of 1s, and it is maintained at the maximum value until the time of 2s.

Rotation moment is 0 from the beginning until the time of 1s, then linear increase from 0 to a maximum value of up to 1.8s and is maintained at maximum value from 1.8s to 2s time.

The bodies surfaces coming into contact are defined as deformable and "master" for the ball and "slave" type for the ring, imposing also a constant coefficient of friction between the surfaces.

Model analysis is carried out in 100 steps, by running the subprogram "Advanced Nonlinear Transient".

4. Numerical results

The below figures shows the spatial distribution of the stress sy and equivalent Von Mises stress, for the contact between a ball and a bearing ring: Figures 4-5 for time 1s (when on the ball acts just load due to the displacement inverse to the *y* axis direction), and in Figures 6-7 for the time 2s when the two bodies are subjected to both charges (displacement on *y*-axis and rotation around the *z*-axis).

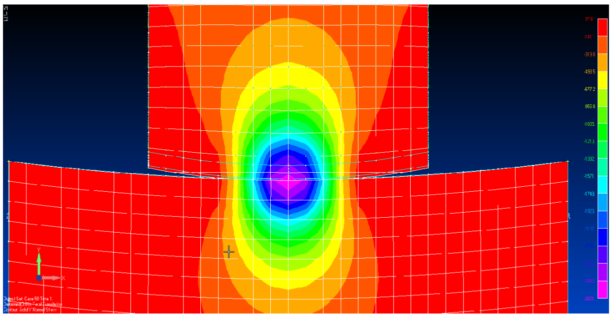

Figure 4. *sy stress distribution when on the ball acts just loading due to the displacement (section plan xOy through initial contact point).*

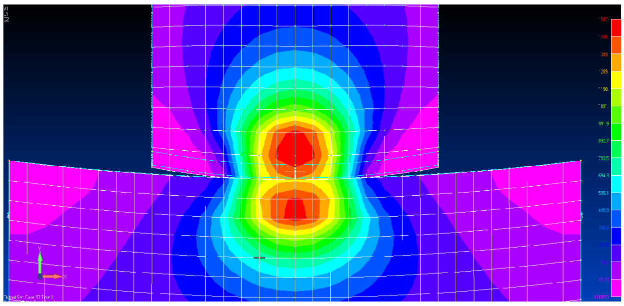

Figure 5 *Von Mises stress distribution, when on the ball acts just loading type displacement (sectional plane xOy through initial contact point).*

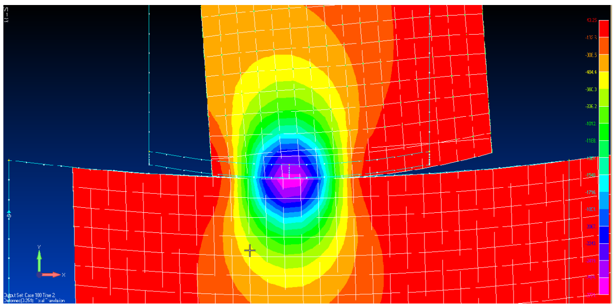

Figure 6 *sy stress distribution, when the two bodies are subjected to both charges (sectional plane xOy through initial contact point).*

For plotting the stresses values evolution, values are extracted from the midline of the nodes on the contact surface of the ball, on the *x*-axis direction. Figures 8-10 presents the results obtained in the two cases, when on the ball acts just loading type displacement (time 1s) and the situation when on the ball and the ring are applied simultaneously displacement and torque (time 2s).

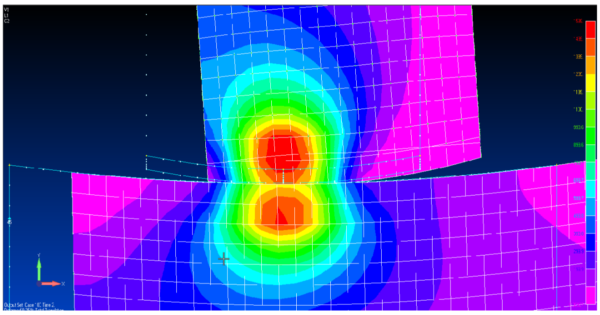

***Figure** 7 Von Mises stress distribution when the two bodies are subjected to both charges (sectional plane xOy through initial contact point).*

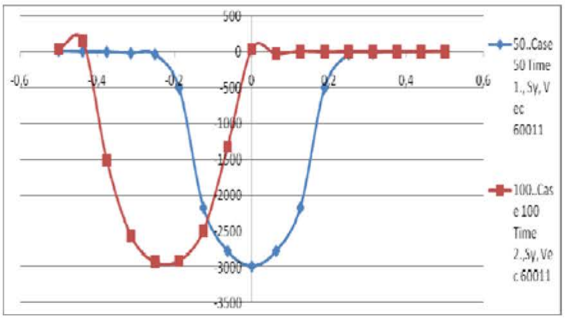

***Figure 8** sy stress distribution in the nodes on the midline contact surface in xOy plan, in x-axis direction*

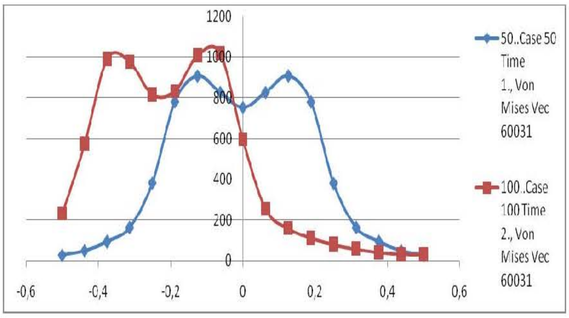

***Figure 9** Von Mises stress distribution in the nodes on the midline contact surface in xOy plane, in x-axis direction*

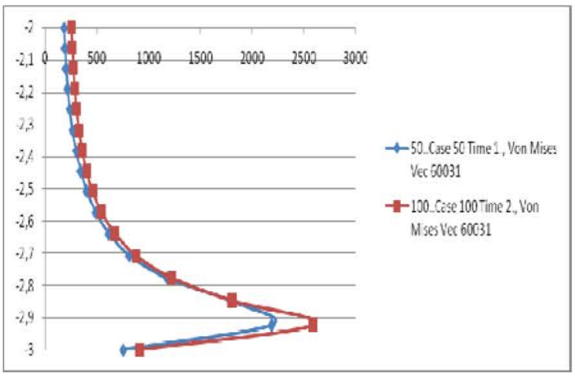

***Fig. 10** Von Mises stress distribution in nodes on y-axis in contact point*

5. Conclusions

In this paper was presented a numerical model based on the finite element method using FEMAP software (Finite Element Modeling And Postprocesing), of a contact between a ball and an outer ring of a ball bearing.

From this analysis it can seen that the obtained results are in good agreement with Hertz theory.

S_y component values along at x-axis is sensitive to the presence of the tangential force due to the rotation of the ring.

Rolling friction has an influence on the maximum value of the Von Mises stress.

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Contact:
Department of Mechanical and Technologies
The University "Ştefan cel Mare" of Suceava,
Addresss: Str. Universităţii 13, 720229
Suceava, România
E-mail: sorin.t.siretean@gmail.com