Contact pressure analysis of slewing rings

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Abstract. The paper deals with creation of functional parametric 3D model of slewing bearing. The slewing bearing is loaded by the radial force that acts from the centre of the bearing and it is also perpendicular to the axis of bearing. 3D fully parametric model of slewing ring was designed in Creo Parametric 3 and it is controlled by the ball diameter. To calculate several geometric bearing modifications, the 3D model from Creo was imported to the Ansys Workbench computational program. Based on many analysis in Ansys, it is possible to evaluate the size of contact pressures, determine the most loaded place and compare values between individual bearing modifications. By transferring the measured values of the contact pressures to the Microsoft office, it was possible to construct a graphical dependence between the values of measured stresses and the magnitude of the radial force acting on the slewing bearing.

1. Introduction
The well-known fact about large diameter slewing bearing is that they are primary designed for applications where large amount of axial load is transferred through thin-walled rolling rings to rolling elements, which usually are balls or rollers. Inner ring is evenly surrounded by rolling elements located on its orbits [1]. This also applies to the outer ring, with one difference and it is that the rolling elements are located on its inner side because there is the raceway located [2].

The stiffness of slewing ring is one of most important thing if we talk about them. They have significant influence on the dynamics of a rotating shaft and also on the precision of the machine systems [3]. Life of rolling elements and the rolling contact fatigue can be determined by full-scale endurance tests. This tests that are realized in bearing test rigs are expensive and time consuming [4]. Lin analyses the displacement and coefficient of radial stiffness with angular contact [5]. For the analysis John Harris’s method based on Hertz-contact theory has been used. In order to compare the results obtained by the mentioned method and Palmgren’s empirical relations the finite element method has been used [6]. Research about the radial stiffness of the radial bearing was realised by Mullick and it was also based on John Harris’s finite element method [7]. For solving a nonlinear equations the Newton Raphson’s method has been used, while in contact analysis uses finite elements method. In this study the gyroscopic moment and the influence of sliding was neglected [8]. The results shows that the relative displacement and stiffness of the bearing rings depend on the centrifugal force and combined loads [9]. Antoine et al. propose two new approximate, methods for determination the angle of contact depending on the preloaded and speed or in special cases for elastic preload [10]. Duval et al. concluded that the treatment thickness, in the investigated depth range of 3,2 mm to 4,5 mm, has no real influence because critical points from computations being at a depth of less than 3,2 mm. Increasing the initial contact angle from 45 degrees to 50 degrees does not affect the level of damage compared to the chosen reference case [11]. However, this increase of the angle makes appear a second critical zone, at the track border, with a non-negligible probability of failure also the reduction of conformity compared to the
reference is penalizing in term of fatigue damage. On the other hand, in case a surface cracking is
favoured, compared to the under layer cracks predicted in other cases [12]. The most influential
parameter is undoubtedly the diametric gap. The introduction of a gap with respect to the references
case (null gap), strongly reduces the fatigue strength and increases the risk of cracking in the track
border. This parameter can be used as an indicator in the context of damage monitoring on the slewing
ring [13].

Radial stiffness depends on the elastic deformation (deflection) of the bearing under load and can be
expressed as a ratio of load to deflection. However, since the relationship between deflection and load
is not linear, only guideline values can be provided [14].

The main task of this paper is to select concrete type of bearing then create virtual 3D model
parameterize it and then to modify this model by several parameters direct from the environment of
Ansys workbench calculation program while the model is under the load [15].

Methodology of solving this task includes selection of critical parameters for creation of the
parametric bearing model. In this case we introduce two main parameters like ball diameter and contact
angle. There are also minor parameters like bearing pitch diameter which has significant influence on
the amount of used rolling elements. If we change number of rolling elements directly from Ansys then
after we update model and value of elements will be increased we lost defined boundary conditions of
bearing. So we will keep the same value of pitch diameter during all measurements.

A methodology based on finite element method has been developed to calculate accurately the value
of ball forces around slewing bearing and its deformation. This methodology can use the non-linear
springs to simulate contact. It is necessary to use this method when stiffness of support structure is not
uniform.

In the end we need some calculations like calculation of semi-axis, contact pressures, and the depth
at which the maximum tension is applied. If we have all of this we can move forward and based on many
calculations of the deformations performed in the Ansys we can compare influence of ball diameter and
contact angle. Based on analysis graphs were constructed.

2. Creation of basic geometry of four-point contact slewing bearing and its parameterization
Slewing bearing or ring, is large diameter bearing, usually loaded by axial force, small radial force and
tilting moment at the same time see figure 1. Parts from which the bearing and rolling element consist
are shown in figure 2. Dimensions that are needed for parameterization are showed on figure 3 and also
Table with dimensions and relations between them is shown on figure 4.

Slewing bearing considered in this paper is four point contact type. So if we introduce radial force
only, the whole load will be transmitted through the 4 contact points see figure 5.

Figure 1. Slewing bearing loaded by tilting moment, radial and axial forces.
Figure 2. Parametric model of one bearing part (left), design of rolling element (right).

Figure 3. Parametric model of rolling element.

Figure 4. Relations between the geometry dimensions.

Figure 5. Distribution of load in four point contact.
3. Basic calculations

All calculation are based on Hertz-contact theory that is most suitable for this type of task. First of all we can find the curvature radii of inner and outer raceway.

\[
\rho_{1I} = \frac{1}{r_{we}}; \quad \rho_{2I} = \frac{1}{r_{we}}
\]

\[
\rho_{1II} = \frac{1}{r_1}; \quad \rho_{2II} = \frac{1}{r_{pw}}
\]

From obtained values of radii of curvatures it is possible to obtain sum of values. Then we calculated the value of \(\cos\tau\).

\[
\cos\tau = \left| \frac{\rho_{1I} - \rho_{1II} + \rho_{2I} - \rho_{2II}}{\Sigma \rho} \right|
\]

In the next step we can calculate distance between the centres of raceways.

\[
A = B \cdot D_{we}
\]

After the distance between centres has been calculated, we move forward and we calculate the size of contact area.

\[
2_a = 2 \cdot \mu \cdot \frac{1}{\sqrt{E}} \cdot (1 - m^2) \cdot \left( \frac{3Q}{\Sigma \rho} \right)
\]

\[
2_a = 2 \cdot \vartheta \cdot \frac{1}{\sqrt{E}} \cdot (1 - m^2) \cdot \left( \frac{3Q}{\Sigma \rho} \right)
\]

After we obtain values of both contact zones, then it has been calculated the depth in which the maximum pressure acts.

\[
2z_0 = 2b \cdot 0,25
\]

For calculation of stiffness between raceway of inner and outer ring and surface of rolling element it has been needed to calculate approximated parameter of ellipticity.

\[
\kappa_{i,o} = \frac{\pi}{a_{i,o}^2}
\]

Where the value of \(a_{i,o}\) can be calculated by following equation.

\[
a_{i,o} = \frac{2f_{io}D_{pw}}{D_{we}(D_{pw} - D_{we})(2f_{io} - 1)}
\]

Also in next step it was necessary to calculate stiffness between inner ring contact surface and ball. Then it was also calculated for outer ring and the ball.

\[
K_{i,o} = \pi \cdot \kappa_{i,o} \cdot E' \cdot \sqrt{\frac{2\vartheta R_{io}}{9\vartheta^3}}
\]

After the calculation of both stiffness parameters we are able to calculate combined stiffness.

\[
K_n = \left[ \left( \frac{1}{K_i} \right)^{\frac{1}{n}} + \left( \frac{1}{K_o} \right)^{\frac{1}{n}} \right]^{-n} \rightarrow n = \frac{3}{2}
\]

Then we calculated the maximum of contact pressure by equation bellow.

\[
p_0 = \frac{1.5}{\pi \mu \nu} \cdot \sqrt{\frac{E}{(1-m^2)}} \cdot \left( \frac{3Q}{\Sigma \rho} \right)^2 \cdot Q
\]
4. Boundary Conditions and Simulation

After introduction of 3d model from Creo parametric to Ansys Workbench boundary conditions has been set as you can see on figure 6. First of all, we set the conditions for point contacts because only by this way the right conditions can be achieved. For removing of movements and rotations there was used the boundary conditions like Displacement for movements and Remote control for movements and rotations.

Figure 6. Boundary conditions of bearing part.

Before calculations were performed the mesh method in contact zones has been set to hexahedral and their size to one tenth of millimeter. For achieving a fast calculations the size of elements in less important locations need to be set to highest value as possible. Mesh created on rolling element is shown on figure 7. For better computing the hexahedral fine mesh has been created on places where the maximum of contact pressure acts. On elements such as bearing rings was created coarse mesh because it has not a significant influence to the calculations see figure 8.

Figure 7. Mesh created on rolling element.  
Figure 8. Mesh on bearing part.
After the boundary conditions have been set, we start the solve mechanism in Ansys and after a few hours we obtain the results. Concrete results and load distribution of contact pressure in contact surfaces see figure 9.

The graph of dependence between force and the contact pressure has been created from the results obtained in Ansys Workbench figure 10.

**Figure 9.** Distribution of contact pressure.

**Figure 10.** Graph of dependency between contact pressure and force.

### 5. Conclusions

Creation of 3D calculation model is very helpful if large number of calculations are needed for different geometric variants. By this method, it is possible to change the geometry parameters and determine the relationships them. After the model of bearing is imported to the Ansys workbench, it is necessary to set boundary conditions and create finite element mesh. Main rule for accurate calculations is that the mesh in most loaded location needs to be created by hexahedral mesh method.

By measuring of contact pressure at different geometry parameters, it has been proved that the relationship between the applied force and maximum value of contact pressure is evenly growing, which means that when the value of force is higher, also the maximum of contact pressure will be greater, but there are also shear losses.
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