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Stiffness analysis of slewing bearings

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Abstract. The article deals with analysing the stiffness of a special type of bearings called slewing bearings. The purpose of the analysis is to examine the stiffness of the bearing steel. This type of bearings is most often designed for operation in which the rolling elements and hence the rings themselves are loaded in the axial direction. However, the aim of the work is not only to examine the stiffness but to identify possible ways that would favourably affect the stiffness, achieving the maximum lifetime or the maximum hours of operation. Stiffness measurements will be performed in the Ansys Workbench calculation program. On the basis of the measured values, manual calculation will be performed on the basis of which possible inaccuracies that arise from the bearing production itself will be corrected.

1. Introduction

Slewing bearings also knowns as slewing rings are large sized rotary elements usually used in applications in which large rotational functional elements are involved, such as tower cranes, wind turbine generators, boring machines, etc. There are many different types of slewing bearings and they can be divided in depending on the number of rows and also on the type of rolling elements. There are bearings with one, two or three rows of rolling element and we distinguish between balls and rollers. Figure 1 points to a typical arrangement of four contact point slewing bearing. Figure 1 also shows the usual load system acting on bearing in axial or radial direction.

It has been studied that all angular contact ball bearings have similar features in consideration of geometry, mechanism and structure [5]. The stiffness of angular four contact point ball bearings has a very important influence on the dynamics of a rotating shaft and the machine system [6], the life and the rolling contact fatigue can be determined by full scale bearing tests. This type of tests are released on test rigs and that is very expensive and time consuming solution. Bearing designers would like to understand the impact of four variables like Subsurface residual stress, Ball material density, Gradient in yield strength with depth Raceway surface hardness that are thought to affect spall propagation [7].

Mullick [8] researches radial stiffness of a radial and ball bearings with angular contact using the John Harris method and also the finite element method. For solving systems of nonlinear equations, we are using Newton Raphson's method, while in contact analysis uses finite elements method. The results show that the displacement and of the bearing rings depend with high probability on the combined loads and centrifugal force. Antoine et al. [9],[10] propose two methods for determination the contact angle between inner and outer ring of the bearing depending on the preload and also speed for special cases of elastic preload. There are many methods based on the Hertz contact theory. They

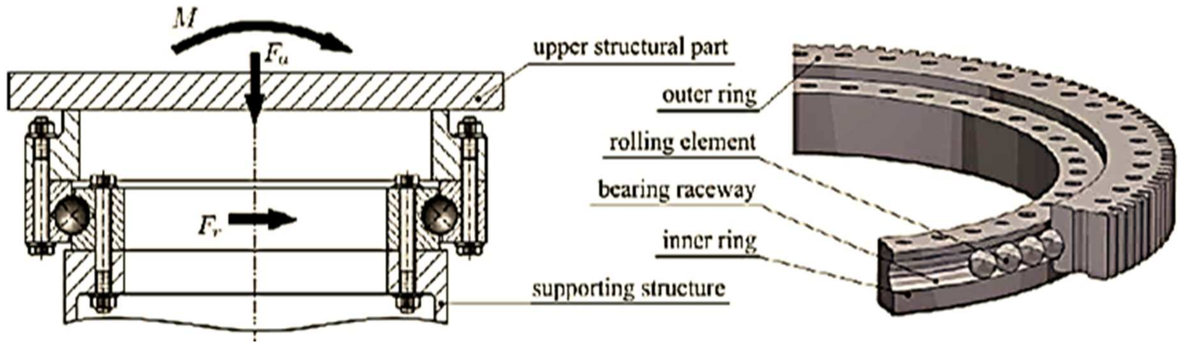

Figure 1. Typical arrangement of four contact point slewing bearing.

are based on the assumptions that the force of preload constant and doesn't affect on the change of contact angle or speed. In solving the system of equations, it is admitted that a certain speed, for a preload comes only to the revoke of the axial clearance, and that there is no axial deformation.

Sun M. K. at al. [11] examine relationship between contact deformation, change of stiffness and clearance of beating. The results gained by experimental and analytical propose determining the elastic deformation of bearing elements must be taken into account radial and axial clearances in the bearing. Wei L. at al. [12] examined the influence of centrifugal force, gyroscopic moment and preload on the bearing stiffness. They concluded that if the value of contact angle of raceway and ball exceed 8.9°, the value of the radial stiffness decreases with increasing speed. By the experiments have been shown that the radial stiffness decreases more than 20% if the contact angle is 40° and revs are 15,000 rpm. They also noted that the increasing of the temperature affects an increasing of preload of bearing and increase the frequency of oscillation.

The most appropriate way to determine axial or radial stiffness of four point contact ball bearing is to create 3D model with cad software programme like for example creo 3.0 parametric. In the next step, it is necessary to convert native file from 3d cad software into a suitable format that could be meshed in program such as ansys workbench. After the network is created, calculations can be made. In an ideal case, experimental test will be performed to verify the calculations and confirm the hypothesis.

2. Hertz contact theory

The Hertz contact theory assumes the contact of two bodies with curved surfaces, which are pushed against each other by the force Q. Each of the two bodies is characterized by a curvature in main planes that are perpendicular to each other, and in which the maximum and minimum curvature is

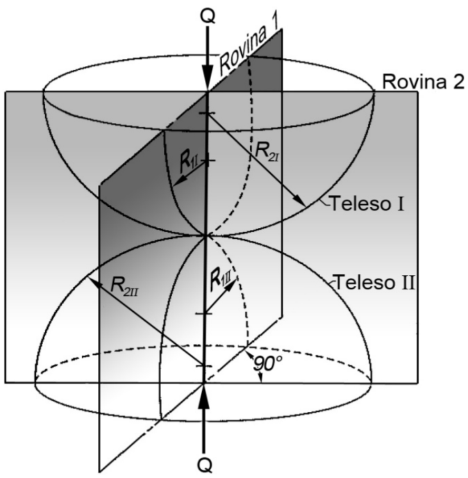

Figure 2. Radii of curvature located in main planes.

located. Curvature ρ is defined as the inverted value corresponding to the radius of curvature r. Curvature ρ is positive if the centre of curvature is located inside the body (convex curvature$+\rho$), and negative if the centre is located outside the body (concave curvature $-\rho$).

$$\cos\tau = \left|\frac{\rho_{1I}-\rho_{1II}+\rho_{2I}-\rho_{2II}}{\sum\rho}\right| \tag{1}$$

where:

$$\sum\rho = \rho_{1I} + \rho_{1II} + \rho_{2I} + \rho_{2II} \tag{2}$$

In accordance with Herts derived relations, the coefficients μ, ν and $2K/\pi\mu$ can be determined as a function of $\cos\tau$. The values of the coefficients depending on the function of $\cos\tau$ are shown in Figure 3.

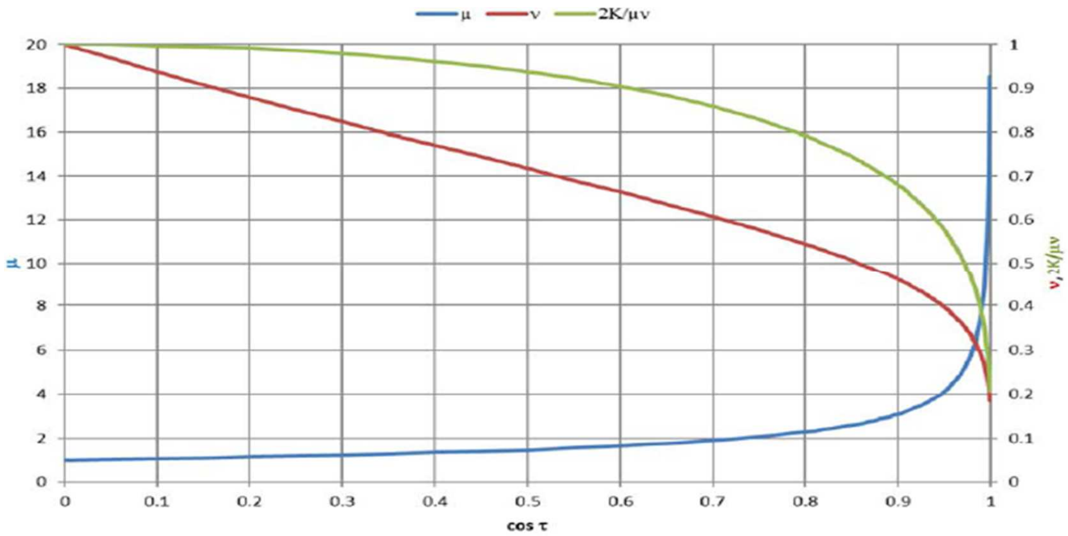

Figure 3. Coefficients μ,ν and 2K/πμ.

3. Theoretical calculations

According to Hertz relationship we can calculate the elastic deformation in order to analytically express the ball slewing ring stiffness. Elastic deformation δ_n is calculated, with respect to the Hertzs theory, separately for each point of contact. Which means that the calculations are separately for rolling element - inner race contact and also separately for rolling element – outer race. For two rolling elements with point contact that are compressed against each other by the compression force $Q_n(N)$, Hertz compiled the calculation as [13]

$$\delta_n = 1.5\left(\frac{2K}{\pi\mu}\right)\sqrt[3]{\left(\frac{1-u^2}{E}\right)^2\left(\frac{\sum\rho}{3}\right)Q_n^2} \tag{3}$$

Provided that the effective elastic modulus is [13]

$$E' \frac{E}{(1-u^2)} \tag{4}$$

The resulting formula can be written as:

$$\delta_n = 1.5\left(\frac{2K}{\pi\mu}\right)\sqrt[3]{\left(\frac{1}{E'}\right)^2\left(\frac{\sum\rho}{3}\right)Q_n^2} \tag{5}$$

If we introduce constant K from equation 3 :

$$K = 1.5\sqrt[3]{\frac{1}{3}} \tag{6}$$

We obtain a simplified adjusted relationship

$$\delta_n = K\left(\frac{2K}{\pi\mu}\right)\sqrt[3]{\left(\frac{1}{E^{'}}\right)^2\sum\rho Q_n^2} \tag{7}$$

The total elastic deformation δ_n for rolling or slewing rings is equal to the summation of the elastic deformation δ_o between the outer race and the rolling element and deformation δ_i between the inner race and rolling element.

$$\delta_n = \delta_i + \delta_o \tag{8}$$

Then after formal adjustments we obtain :

$$\delta_n = 2K\left(\frac{1}{E^{'}}\right)^{\frac{2}{3}}(Q_n)^{\frac{2}{3}}\left[\frac{2K}{\pi\mu_i}(\sum\rho_i)^{\frac{1}{3}} + \frac{2K}{\pi\mu_o}(\sum\rho_o)^{\frac{1}{3}}\right] \tag{9}$$

By adjusting the magnitude of the constant K to

$$K = 2 * 1.5\sqrt[3]{\frac{1}{3}} \tag{10}$$

We can than adjust equation 10 if we want to express the magnitude of the normal force Q_n acting on:

$$Q_n = \delta_n^{\frac{3}{2}}\frac{E^{'}}{K^{\frac{3}{2}}\left[\frac{2K}{\pi\mu_i}(\sum\rho_i)^{\frac{1}{3}}+\frac{2K}{\pi\mu_o}(\sum\rho_o)^{\frac{1}{3}}\right]^{\frac{3}{2}}} \tag{11}$$

If we introduce the constant c_δ for a double point contact of three bodies with spherical surface

$$c_\delta = \frac{E^{'}}{K^{\frac{3}{2}}\left[\frac{2K}{\pi\mu_i}(\sum\rho_i)^{\frac{1}{3}}+\frac{2K}{\pi\mu_o}(\sum\rho_o)^{\frac{1}{3}}\right]^{\frac{3}{2}}} \tag{12}$$

We obtain the relationship for the normal force Q_n in the following well known form [13]

$$Q_n = c_\delta c_n^{\frac{3}{2}} \tag{13}$$

If we take into account only the axial force F_a because we are dealing with axial type of bearing then, the relationship between axial deformation and normal deformation can be expressed by the goniometric function

$$\sin\alpha = \frac{\delta_a}{\delta_i+\delta_o} = \frac{\delta_a}{\delta_n} \rightarrow \delta_n = \frac{\delta_a}{\sin\alpha} \tag{14}$$

Then the the final relationship between the normal force and the axial deformation of slewing ring will be:

$$Q_n = c_\delta\left(\frac{\delta_a}{\sin\alpha}\right)^{\frac{3}{2}} \tag{15}$$

4. Basic model for FEM analysis

Geometrical parameters of the slewing bearing Figure 4. that are necessary to build 3D cad model, that will be the basic element needed for structural analysis in ansys workbench or another software.

For analysing the slewing ring we must divide the whole bearing into a smaller units Figure 5 for achieving more accurate results and much better computing mesh that is one of the most important elements of this analysis.

The most important place where the fine calculation mesh is needed are the rolling elements itself. Before the rolling elements will be divided, we need to know which part of rolling element surface is in the contact with the race ways which are also the part of inner and outer rings. Schematic representation of the model for the ball and the ball raceway contact is shown in Figure 6

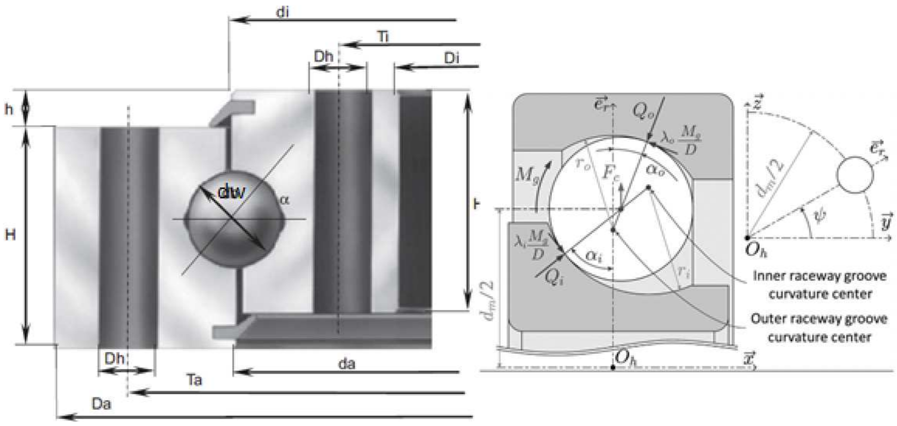

Figure 4. Geometrical parameters of the slewing bearing.

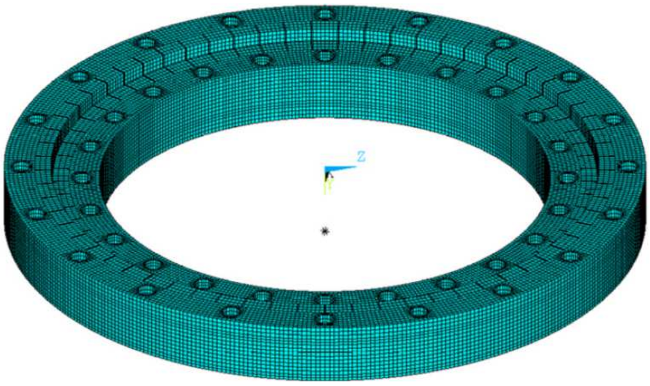

Figure 5. FEM model of slewing bearing.

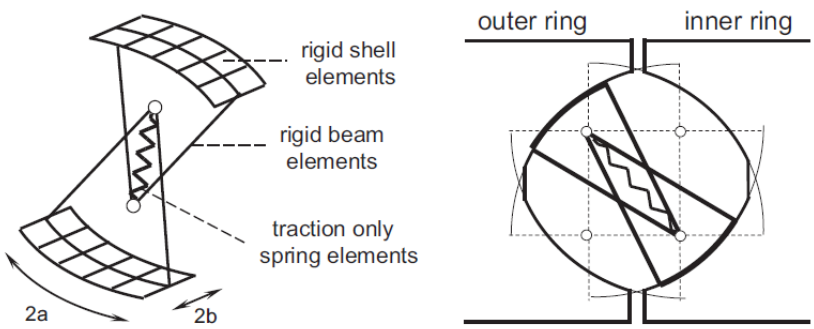

Figure 6. Schematic representation of the model for the ball and the ball raceway contact.

An elliptical shape, being the values of its semi axes a and:

$$a \approx 1.71 * \ 10^{-2}(1-s)^{-0.4091} D_W{}^{\frac{1}{3}} Q^{\frac{1}{3}} \tag{16}$$

$$b \approx 1.52 * \ 10^{-2}(1-s)^{-0.1974} D_W{}^{\frac{1}{3}} Q^{\frac{1}{3}} \tag{17}$$

The maximum pressure in the contact area is found to be

$$P_{MAX} \approx 1.84 * 10^{3}(1-s)^{0.2117} \frac{Q^{\frac{1}{3}}}{D_w^{\frac{2}{3}}} \tag{18}$$

Finally, the elastic contact pressure that causes the static failure of the ball is, according to P_{MAX}.

5. Results and conclusions

The main purpose of this work was to build a 3D cad model of the slewing bearing so that this model could be recalculated using the ansys workbench calculation program, then the stiffness calculations were to be performed to verify the simulated analysis. Due to the lack of structural analysis and the time-consumingness of the solution of this task, a model was developed and the equations based on which the bearing will be counted. Therefore, it can be assumed that the following article will deal with the results of the current work that is in the process of solution.

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