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SIMPLIFIED COMPUTATION METHODOLOGY FOR CONTACT FORCES ON TAPERED ROLLING BEARING WITH FLEXIBLE PARTS

Summary. Durability calculations of bearings take in account the distribution of forces on rollers. Calculation of these forces in flexibly supported rings is the aim of the paper. We use simplified finite element (FE) models of bearings, which are integrated into the external geometry. This approach can consider the stiffness of the surrounding structure as well as the clearance of the bearing rings, the misalignment of bearings, shaft deflections, and the forces of crowning rollers. The presented results show an influence of the initial radial interference of the outer ring with housing on the distribution of forces in bearing rollers. As the radial stiffness of the housing is close to the stiffness of the outer ring

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interference, this causes considerable deformation of the housing. The presented approach is also able to calculate the distribution of contact pressures over any rolling element.

Keywords: FE analysis; contact force; tapered bearing; ANSYS

1. INTRODUCTION

Tapered rolling bearings are among the most important components of conveyor belts, working machines or various rotary mechanical applications. Therefore, the demand trend in terms of their production, accuracy, load-bearing capacity and reliability is increasing.

Their application is found in all branches of industry, from the production of heavy-duty machinery in mining and stamping, to all modes of transport and the application of miniature bearings to high-speed dental drills.

Due to their multiple uses, emphasis is increasingly placed on their development and improvement, because, in many cases, the bearings do not provide the required durability according to the calculation values [8,9]. There may be many reasons for this, such as overload, the load being lower than required, inadequate lubrication, ineffective sealing, inadequate overfitting or impact load [4].

With the development of tapered rolling bearings, it has been necessary to consider the relevant mathematical and physical facts, as well as start using computer technology to simulate the virtual prototype of the bearings and obtain optimal results for individual factors, such as load, body contact, service life, deformation and prediction [3,5,6,10,12].

For bearing applications involving slow to moderate rotational speeds, the effects of ball and roller inertial loading on internal load distribution can be neglected [7]. For these cases, it is possible to use static equilibrium equations to obtain the load distribution over the circumference of the rings and along the length of the rollers. The roller raceway contact lamina concept is widely used in calculations of the contact force distribution and fatigue life of the bearings. Several researchers have advanced Hertz' contact solution for non-ideal situations [1,9,11]. More general models for solving local fields in contact problems can be found in [2,13].

2. METHODOLOGY DESCRIPTION

We use FE models of inner and outer rings of a bearing in our approach. This gives us the possibility to include the effects of ring stiffness, the tolerances of the inner and outer ring arrangements, and the stiffness of the construction onto which the bearings are mounted. The rolling elements are simplified by the net of truss elements with gaps and constraint equations. These gaps approximate to the circumferential and longitudinal shape of the rolling element. An additional advantage of this model is that it eliminates the requirement to solve the contact between the rolling elements and raceways, which significantly reduces the computation time and the memory demands of the analysis, because the total number of elements is greatly reduced. A reduction in calculation time is also achieved by the fact that, in this model, it is possible to apply the prescribed load in one load step. The proposed simplified bearing model allows the submodelling technique to compute the distribution of contact pressures on selected rolling elements (highly loaded ones).

The methodology is implemented via ANSYS APDL macros, which generate bearings with simplified geometry that are integrated into the external geometry. Each type of bearing is described by a set of a geometrical parameters.

External geometry is meshed, constrained and loaded in the ANSYS Workbench. Local coordinate systems are used to locate bearings in the external geometry. Named selections are used to couple the bearings with the external geometry. Coupling is made by contact elements. The GUI workbench can be used to verify the values and directions of applied loads in each load step, while a large series of load steps can be imported via Excel sheets. This feature simplifies the transfer of load steps from external programs.

From the point of view of calculating bearing life, it is first necessary to identify the most loaded rolling elements for different load variants, which appear in the operation time of the investigated device. The number of these variants can run into dozens. The aforementioned procedure allows a series of calculations to be performed in one large task.

3. NUMERICAL EXAMPLES

Fig. 1 shows a hollow shaft with housing into which two identical single-row tapered bearings are going to be mounted. The bearings are in a back-to-back arrangement (Fig. 3). The dimensions of bearings are: inner ring bore diameter - 1,720 mm, total bearing width - 270 mm, outer diameter - 2,200 mm, contact angle - 45°. There are 51 rolling elements in each bearing. Bearing 1 lies closer to the coordinate system origin. The logarithmic correction is applied to the profile of the tapered roller, which is defined by the following relation:

$$y = 0{,}000675 \cdot d \cdot \ln \frac{1}{1-\left(2 \cdot x / l_a\right)^2} \quad , \tag{1}$$

where y is the actual value of correction in position x, d is the diameter of rolling element, x is the distance measured from the centre of the rolling element in the longitudinal direction, and l_a is the roller effective length.

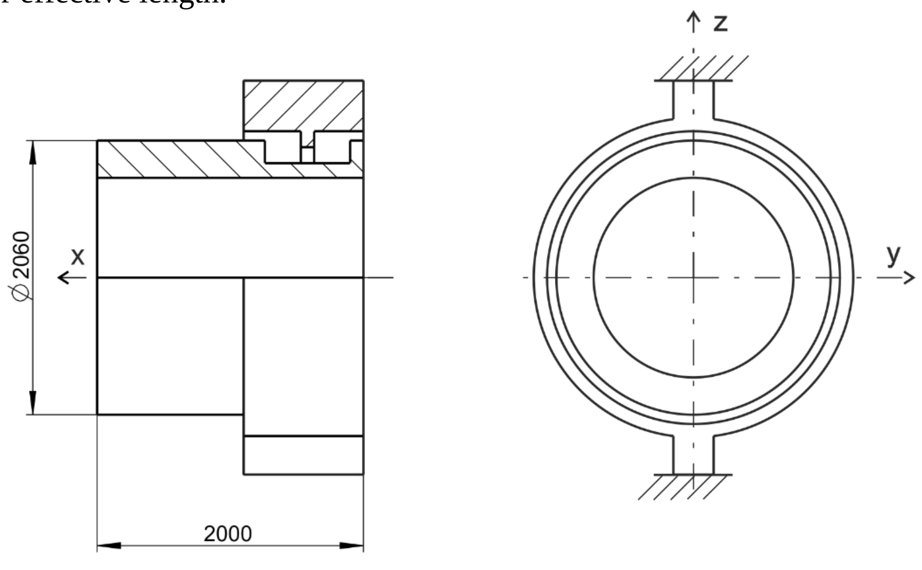


Fig. 1. Housing with a hollow shaft

The ends of the vertical ribs are fixed. The applied loads are: axial force $F_x=2\cdot10^6$ N, radial force $F_y=4\cdot10^6$ N, moment $M_z=1\cdot10^3$ Nm (Fig. 2). Forces and moment are applied on the left shaft's face, while the inner rings are connected to the shaft by bonded contact and the outer rings are connected to the housing by rough contact. Rough contact is a variant of frictional contact with a coefficient of friction whose value is approaching infinity.

There are two load cases. The only difference between the load cases is the radial interference, which is 0.6 mm of the outer ring with housing in the second load case. The value of radial interference affects the internal clearance of the bearings and the distribution of contact forces.

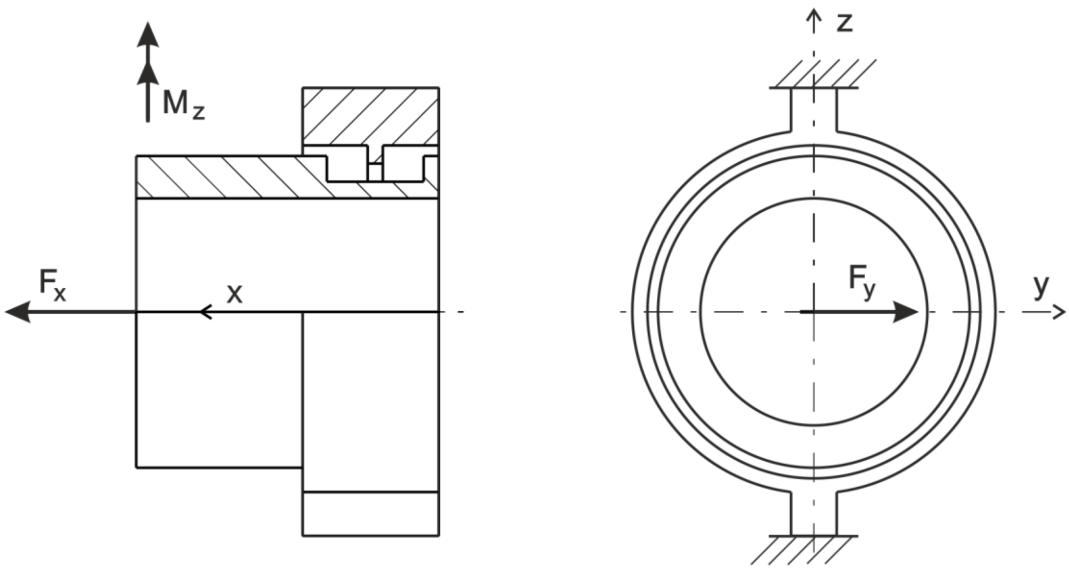


Fig. 2. Boundary conditions

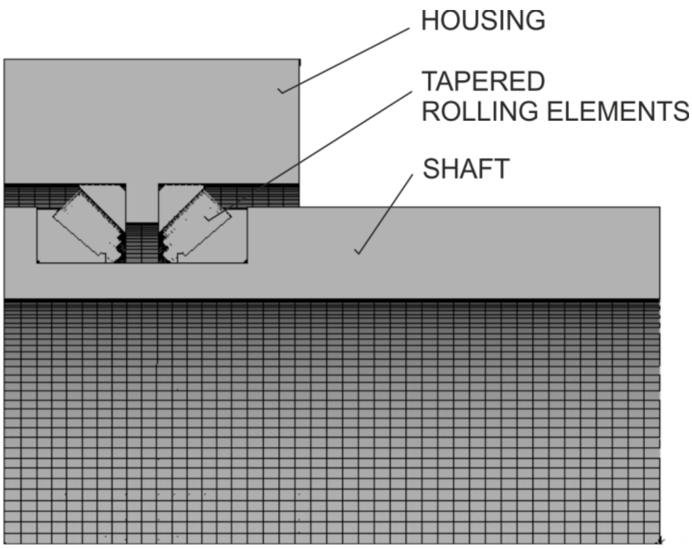
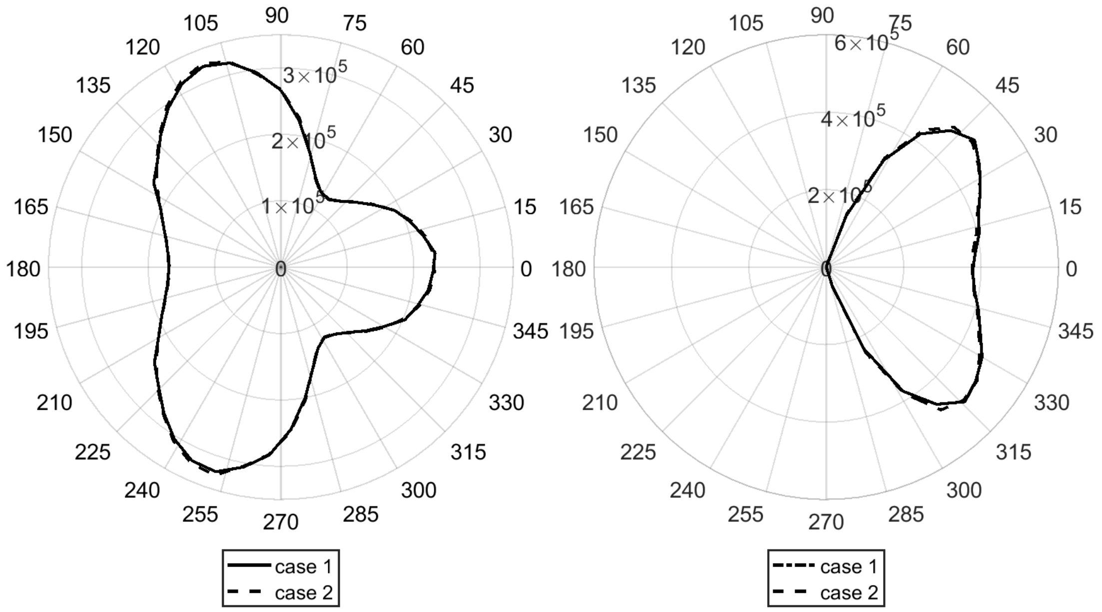


Fig. 3. Assembly of the FE model



a) b)

Fig. 4. Roller raceway contact forces [N] for two load cases on: a) Bearing 1 and b) Bearing 2

4. CONCLUSIONS

The presented methodology confirms the capacity to include the stiffness of the surrounding construction in the calculation of contact forces. Two maximums in the contact forces on Bearing 1 (Fig 4a) at an angular position of 110° and 250° are the result of a local increase in the stiffness of the housing affected by two ribs. The maximums are shifted from the vertical position (90° and 270°) by moment load M_z. The smaller third maximum is a consequence of the hollow shaft and housing deformation, with the applied force in the y direction.

The small change in the distribution of the contact forces (Fig. 4), caused by radial interference of the outer rings and housing, is a consequence of the low stiffness ratio of the housing to the outer ring. Stiffer housing will cause a higher level of applied preload caused by radial interference and a more uniform distribution of load on rollers.

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