

Analysis of Contact Stress between Cylindrical Roller and Outer Ring Raceway with Taper Error Using ANSYS

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Abstract

Taking a cylindrical roller bearing as a research object, the contact stress between the cylindrical roller and the outer ring raceway with a taper error is studied by means of ANSYS in order to obtain the allowable value of the upper bound of the taper error. The results show that the given load corresponds to the suitable upper bound of the taper error of the outer ring raceway and with the increasing taper error, the contact stresses between the cylindrical roller and the outer ring raceway increase observably and the distribution of the contact stresses presents more complex asymmetry and nonuniformity.

Keywords: contact stress, cylindrical roller bearing, outer ring raceway, taper error, ANSYS

1. Introduction

The rings are the important parts of a rolling bearing and their quality has an important impact on the bearing capacity, operation life, and performance reliability due to the additional contact stresses generated by the machining errors (Chandrasekara et al., 1983; Chiu et al., 1987; Sun et al., 2008; 2010; Wang et al., 2011; Xia et al., 2012, 2012a; Xia, 2012). The taper error of the outer ring raceway as one of the most common machining errors can lead to an abnormal contact stress distribution between the roller and the outer ring raceway. This brings about a bad rolling stability of the roller. But, it is little reported how the machining errors influence the bearing contact stresses (Andrey et al., 2012; Demirhan et al., 2008; Hanson et al., 2010; Slack et al., 2010). Therefore, taking the cylindrical roller bearing coded by N1015 as a research object, this work investigates the relationship between the contact stresses and the taper error of the outer ring raceway with the help of the finite element analysis software ANSYS to establish the finite element model of the cylindrical roller bearing, to analysis the static contact between the cylindrical roller and the outer ring raceway with the taper error, and to determine the allowable value of the upper bound of the taper error.

2. Finite Element Model of Cylindrical Rolling Bearing

2.1 Technical Parameter of Bearing

The technical parameters of the cylindrical roller bearing N1015 are shown in Table 1.

Table 1. Technical parameter

Parameter	Parameter value
Bore diameter of bearing, d /mm	75
Outside diameter of bearing, D /mm	115
Thickness of ring, T /mm	5
Width of ring, W /mm	20
Diameter of roller, Φ /mm	10
Length of roller, L /mm	11
Number of roller, N	22
Poisson ratio of bearing element material, ν	0.3
Elastic modulus of bearing element material, E /GPa	208

2.2 Meshing

In order to reduce computing time, on one hand, the model of one quarter roller at the bottom of the bearing that bears the maximum load is created; on the other hand, the chamfers of the inner and outer rings and the radial clearance are ignored. The Solid 45 is adopted to mesh the model. The meshes of the contact part of the roller and the inner and outer rings are refined to enhance the computing accuracy. The length of the grid along the axial direction is 0.07 mm. Because the side length of the finite element model mesh is, particularly, less than half of the size of the minor semi-axis, the calculation results are precise enough. The half-width size of the contact area is 0.15 mm and the size of the finite element model is suitable, as shown in Figure 1.

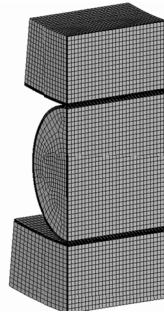


Figure 1. Finite element model

2.3 Creation of Contact Pair

The creation of the contact pair and the setting of the contact parameters are crucial issues on contact analysis (Andrey et al., 2012; Demirhan et al., 2008; Hanson et al., 2010; Slack et al., 2010; Xia et al., 2012, 2012a). Considering the inner and outer rings whose surfaces are bigger and stiffness is higher than the roller, the inner and outer rings are set as target surfaces and the roller is set as a contact surface. Then, the contact pairs are created respectively. Both the contact stiffness coefficient and the tolerance of penetration are key contact parameters. The smaller contact stiffness coefficient is favorable to be convergent. However, the bigger contact stiffness coefficient is favorable to improve the precision. By many times of calculation, the contact stiffness coefficient is set to 1.5 and the tolerance of penetration is set by a default value.

2.4 Loading

Before the load and the constraint are applied, the nodal coordinate systems belonging to the middle plane of the roller are all converted to the cylindrical coordinates and the displacement of nodes of the middle plane of the roller along the circumferential direction are all constrained. Furthermore, a symmetry constraint is applied to the cross-section of the roller and all the degrees of freedoms of the outer surface of the outer ring nodes are constrained. The load is applied to the finite element model after the radial freedom of the inner surface of the inner ring is coupled.

3. Analysis of Results

Through modeling, meshing, constraining, loading, and solving, the results of the surface stress of the roller are obtained by finite element analysis. The normal direction contact stress produced by the roller and the outer ring raceway is used to analyze the effect of taper error of the outer ring raceway on the stress. The normal roller loaded 1/5 of the dynamic load rating is analysed, as shown in Figure 2. When the abscissa x_1 takes the value 5.4 mm that corresponds to the surface of the middle part of the outer ring raceway, the contact stress takes the largest value 1235.6 MPa. When the value of the abscissa x_1 is greater than 5.4 mm or less than 5.4 mm, the contact stress gradually decreases. Obviously, the stress is distributed uniformly at the middle of the outer ring raceway.

According to the different values of the taper error of the outer ring raceway, finite element models are recreated. The taper error of the outer ring raceway is from 1 μm to 5 μm and the results of contact stresses are given in Figures 3-7.

It can be seen from Figures 2-7 that the maximum stress of the roller surface changes with the taper error of the outer ring raceway along the axial direction. The larger the taper error, the larger the maximum stress. If the taper error exceeds 3 μm , the value of the maximum stress increases sharply, as shown in Figure 8, showing the unusually additional contact stress generated by the machining error.

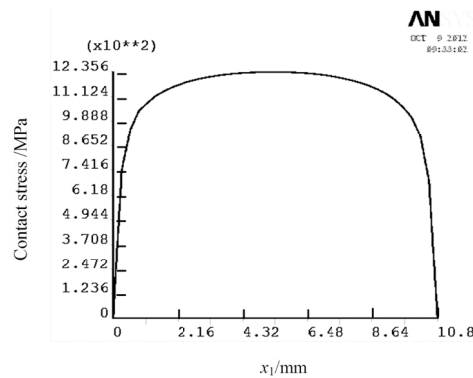


Figure 2. Contact stress between outer ring raceway with 0 μm taper error and roller

In addition, when the taper error of the outer ring raceway is 0 μm , the distribution of the contact stress is the most uniform, i.e., the maximum stress appears at $x_1=5.4$ mm that corresponds to the surface of the middle part of the roller. With the increasing taper error, the location of the maximum stress gradually deviates from the middle part of the roller. If the taper error exceeds 3 μm , the location of the maximum stress sharply deviates from the middle part of the roller, as shown in Figure 9, showing the abnormal contact stress distribution generated by the machining error.

According to contact mechanics and rolling bearing theories, the unusually additional contact stress and abnormal contact stress distribution generated by the machining error belongs to an unfavorable condition. If the rolling bearing runs at great speed under such a condition, the axis of rotation of the roller would appear skewed and the roller can therefore get stuck in two raceways, resulting in a safety incident.

As a result, in the process of manufacturing the raceway of the outer ring, the taper error must be controlled within an appropriate range, such as 0 μm -3 μm for the cylindrical roller bearing studied in this work, which is called the suitable upper bound of the taper error of the outer ring raceway.

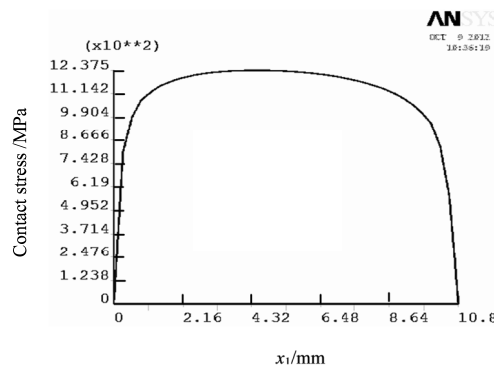


Figure 3. Contact stress between outer ring raceway with 1 μm taper error and roller

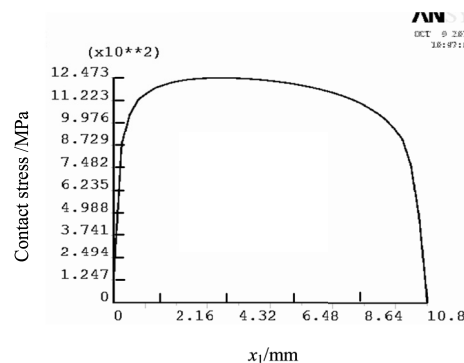


Figure 4. Contact stress between outer ring raceway with 2 μm taper error and roller

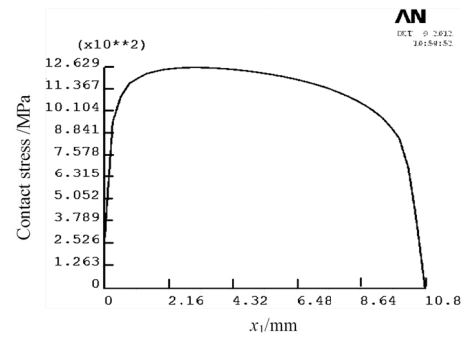


Figure 5. Contact stress between outer ring raceway with 3 μm taper error and roller

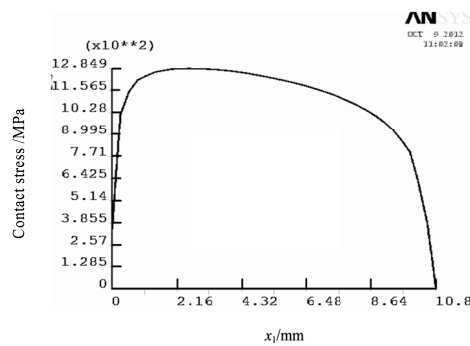


Figure 6. Contact stress between outer ring raceway with 4 μm taper error and roller

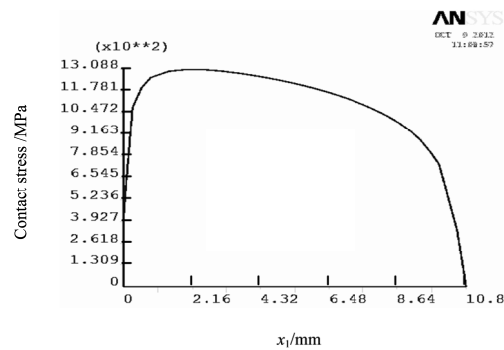


Figure 7. Contact stress between outer ring raceway with 5 μm taper error and roller

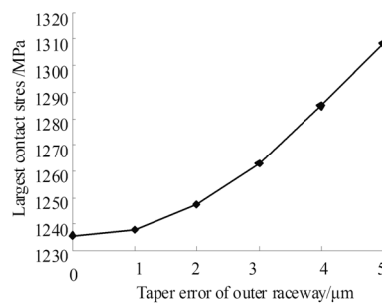


Figure 8. Effect of taper error on largest contact stress

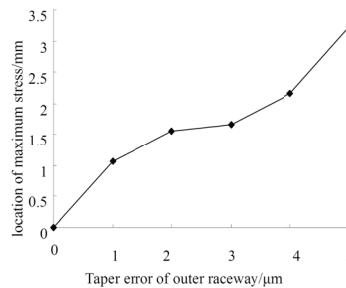


Figure 9. Influence of taper error on location of maximum stress

4. Conclusions

The given load corresponds to the suitable upper bound of the taper error of the outer ring raceway and with the increasing taper error, the contact stresses between the cylindrical roller and the outer ring raceway increase observably and the distribution of the contact stresses presents more complex asymmetry and nonuniformity.

In the process of manufacturing the raceway of the outer ring, the taper error must be controlled within a appropriate range, such as 0-3 μm for the cylindrical roller bearing studied in this work, which is called the suitable upper bound of the taper error of the outer ring raceway.

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