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Research Article

Tolerance Analysis of Cylindrical Roller Bearing under Combined Radial and Axial Loads

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In order to analyze the influence of tolerance values of key parameters for cylindrical roller bearings under combined axial and radial loads, a coupled model, incorporating a dynamic model of cylindrical roller bearings, contact model, and fatigue life model, is developed to investigate the effect of flange angle, roller-end radius, interval of roller length gauge, and roller profile on contact performance and fatigue life. The results show that the grouping design of flange angle and roller-end radius in the tolerance range was helpful for reducing contact ellipse truncation. The difference of the roller length would change the axial load distribution of the bearings. For the longest roller located in the bottom position, the bigger the difference, the bigger the roller tilt angle and carried-axial load. The (0, +2.5 μm) tolerance range of the crown drop can limit the difference of the fatigue life within 20% in the current analysis.

1. Introduction

Cylindrical roller bearings can support axial loads if they are used as semi-locating or locating bearings. These axial loads are transferred via flanges and roller-end faces. Therefore, the design of the flange and roller-end geometry is very important which directly affects the contact condition and determines the axial load carrying capacity of the bearing. Many bearing manufacturers [1–3] and scholars [4–6] have tried to improve the design. Some geometrical optimization of flange-roller end contact truly improves the axial load carrying capacity. However, the manufaturing tolerance is inevitable. The results from reference [4] have shown that the location of flange-roller end contact points was sensitive to the variation of flange angle and roller-end spherical radius, and the location of contact points would affect sliding velocity and roller tilt, further affecting heat generation and fatigue life. In addition, the interval of roller length gauge and roller profile tolerance also affects the axial load distribution and anti-tilt capacity. Therefore, tolerance analysis around the optimal design would be significant in predicting the life of the bearing.

Studies [7–15] investigating the effect of manufaturing error on bearing performance have been widely conducted. Chen et al. [7, 8] analyzed the effect of off-sized roller and its arrayed order on load distribution in a cylindrical roller bearing (CRB). It was shown that one off-sized roller has a significant influence on load distribution. Tong and Hong [9] examined the effect of distributed roller diameter error on the fatigue life of tapered roller bearings. The results showed that the mean value of the fatigue life exhibits a significant decrease with increasing roller diameter deviation. Yu et al. [10, 11] found that the amplitude and order of the roundness error in the inner and outer raceways have a significant impact on the motion of the CRB. Ono and Takahasi [12, 13] analyzed the effect of outer ring waviness on ball bearing vibrations. Fujiwara and Yamauchi [14] defined the tolerance of logarithmic crowning applicable to cylindrical bearing rollers. Liu et al. [15] proposed an analytical method for calculating the friction torque of needle roller bearings with roundness error.

In summary, some research efforts were made to relate the manufacture error with bearing contact and dynamic performance. However, little work has been done on the

CRBs under combined axial and radial loads, and most of the work focused on a single tolerance factor. In practice, different kinds of rolling bearing have their specific concerns. For example, for CRBs under combined axial and radial loads, the match between flange angle tolerance and roller-end radius tolerance requires additional attention. Recently, the authors have developed a dynamic simulation model of CRB under combined axial and radial loads [16], which incorporates a contact solver with skew angle consideration. This model will be further developed in the current article to further include the evaluation of fatigue life. The authors hope that the effect of tolerance on the fatigue life of CRB under axial and radial loads would be thoroughly analyzed by the extended model. It is also hoped that this theoretical analysis can provide some insights into the CRB design for engineers.

2. Brief Description of the Model

The model used in the current analysis is extended from the model developed by Wang et al. [16]. Some key features of the previous model are as follows. (a) A dynamic simulation model of CRBs under combined axial and radial loads includes the friction factor and can be applied to evaluate roller skew angle. (b) A contact solver with the consideration of skew angle was developed to analyze contact pressure distribution. Here, in order to investigate the effect of manufacture tolerance on the fatigue life of CRB, the fatigue life model of CRB is added. The numerical solution scheme is shown in Figure 1, and for more numerical details, the readers are referred to [16].

2.1. Fatigue Life Model. According to [17, 18], the L_{10} fatigue life of a roller-raceway line contact subjected to normal load Q may be estimated by

$$L_{10} = \left(\frac{Q_c}{Q}\right)^4, \tag{1}$$

where L_{10} is in millions of revolutions and Q_c is the basic dynamic load rating of inner or outer ring.

$$Q_c = 552\frac{(1\mp\gamma)^{29/27}}{(1\pm\gamma)^{1/4}}\gamma^{2/9}D^{29/27}L^{7/9}Z^{-1/4}, \tag{2}$$

where D is the roller diameter, $\gamma = D/dm$, and dm is the pitch diameter of the bearing. The upper signs refer to the inner ring, and the lower signs refer to the outer ring.

The basic of load rating of a slice is expressed as follows:

$$q_{c\,i,o} = Q_c\left(\frac{1}{k}\right)^{7/9}. \tag{3}$$

The equivalent load of a slice is calculated as

$$q(i)^{i,o} = \left(\frac{1}{Z}\sum_{j=1}^{z}\left[\left(\frac{p_{ij}}{271}\right)^2 D(1\mp\gamma)\frac{L_{we}}{n_s}\right]^4\right)^{1/4}, \tag{4}$$

where p_{ij} denote the maximum contact pressures at the i_{th} slice and j_{th} roller of inner or outer race, which are available from the pressure analysis.

Thus, the basic reference rating life, L_{10r}, is

$$L_{10r} = \left\{\sum_{i=1}^{k}\left[\left(\frac{q_{ci}}{q(i)^i}\right)^{-4.5} + \left(\frac{q_{co}}{q(i)^o}\right)^{-4.5}\right]\right\}^{-8/9}. \tag{5}$$

It should be pointed out that the fatigue life introduced in reference [17] is applied in the conditions of time-invariant load distribution. Here, the internal load distribution of cylindrical roller bearings varies periodically due to tolerance. The Palmgren–Miner linear cumulative damage theory can be used to evaluate fatigue life of cylindrical roller bearings [9].

$$L'_{10r} = \frac{\sum_{t=1}^{N}(L_{10r})}{N}, \tag{6}$$

where N denotes the number of load cases.

3. Results and Discussion

In this section, the authors attempt to design flange angle tolerance, roller-end radius tolerance, interval of roller length gauge, and roller profile tolerance for a NJ213ECP cylindrical roller bearing. The basic geometry and operation parameters are shown in Table 1, which are identical to those in reference [16]. Here the flange angle υ and roller-end radius R_s are assumed to be $30'$ and 650 mm, respectively, which are considered to be in the reasonable range. These values would be used as the basic values in this article to design tolerance range. Some principles are set for optimizing the tolerance range. First, no contact ellipse truncation and edge loading exist for any dimension values in the tolerance range. Second, the scatter of evaluated fatigue life is not more than 20% for any dimension values in the tolerance range in order to improve the reliability of the bearings.

3.1. Tolerance Analysis of Flange Angle and Roller-End Radius. From reference [16], it can be found that the values of flange angle and roller-end radius directly affect the location of contact points and roller tilt angle, which further affect the contact ellipse and contact pressure. Thus, the effect of tolerance of flange angle and roller-end radius on contact ellipsis and fatigue life is discussed in this section. Here, the flange angle varies from 25 minutes to 35 minutes, the roller-end radius varies from 550 mm to 750 mm, and the flange height is 3.0 mm. In addition, the half width of the contact ellipse is nearly 1.0 mm in current working conditions. Thus, the location of the contact point should be in the range of 1 mm to 2 mm in order to avoid contact ellipse truncation. Figure 2 shows the effect of flange angle and roller-end radius on the location of contact point. From Figure 2, it can be found that when the flange angle is fixed, the location of contact point decreases with increasing roller-end radius and vice versa. Moreover, the effective values of flange angle

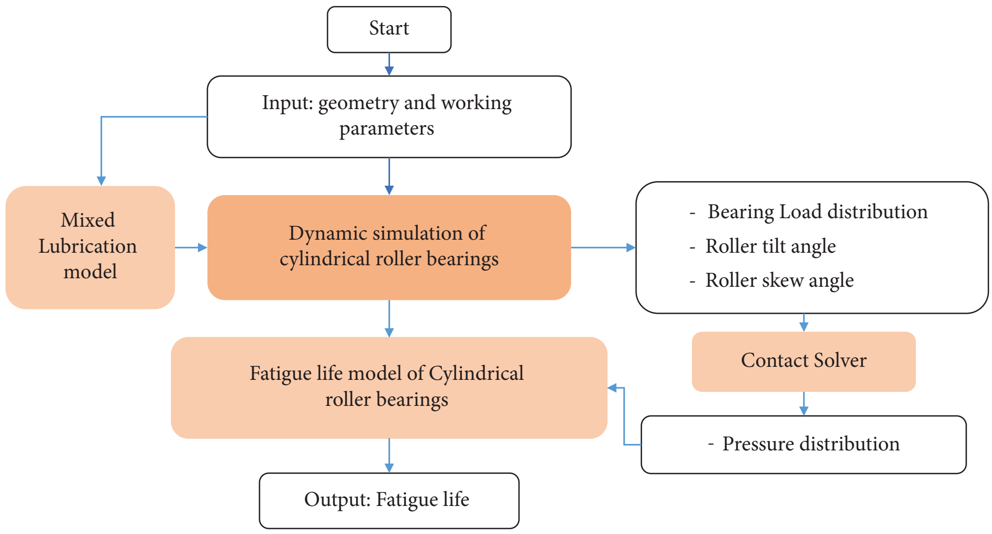

FIGURE 1: Flowchart of the extended model.

TABLE 1: Dimension and working parameters of NJ213ECP.

Roller length L_r	15 mm
Roller radius R	7.5 mm
Number of rollers Z	16
Roller chamfer dimension r_c	0.5 mm
Outer raceway radius R_o	54.25 mm
Inner raceway radius R_i	39.25 mm
Bearing pitch radius R_m	46.75 mm
The axial load F_x	2440 N
The radial load F_z	12200 N

and roller-end radius are in the dotted box. Generally, manufacturing error of flange angle and roller-end radius can be controlled in the 5 minutes and 100 millimeters range, respectively. However, the set of these tolerance values may lead to contact ellipse truncation. Two methods can be applied to eliminate this risk. First, the manufacturing precision should be further improved. For example, the error of flange angle and roller-end radius can be controlled in the 2.5 minutes and 50 millimeters range. In this situation, the tolerance range of flange angle and roller-end radius can be set as $(30', 32.5')$ and (650 mm, 700 mm), respectively. Apparently, this method put more challenges on machining and process control. The other method is grouping the roller-end radius and flange angle. For example, the tolerance range of flange angle can be sorted into two groups, $(30', 32.5')$ and $(32.5', 35')$, and the roller-end radius can be sorted into two groups, (600 mm, 650 mm) and (650 mm, 700 mm), as well. When assembling a CRB, flange in the group of $(30', 32.5')$ should pair with rollers in the (650 mm, 700 mm) group, and flange in the group of $(30', 32.5')$ should be pair with rollers in the (650 mm, 700 mm) group.

Figure 3 gives the relative fatigue life of CRBs in the groups above. Here, the fatigue life at the flange angle 30 minutes and roller-end radius 650 mm is set as the benchmark. From Figure 3, it can be found that the difference between the maximum and the minimum fatigue life in each group is approximately 35% in current working conditions, and the average fatigue life is about 82.5% of the benchmark. The minimum fatigue life is achieved at larger flange angle and larger roller-end radius. This is because the location of the contact point is relatively low which can cause roller tilt. Therefore, the results in Figures 2 and 3 hint that for the CRBs, the values of flange angle and roller-end radius have significant influence on the location of contact point. In order to avoid the contact ellipse truncation, the flange angle and roller-end radius should be sorted and assembled according to the detailed dimension gap. Furthermore, the finer the grouping, the less the discreteness of the performance.

3.2. Analysis on the Interval of Roller Length Gauge. For some specific application requirements, the cylindrical rollers assembled in a bearing should be sorted in specific length gauges. According to ISO 12997 [19], the interval of roller length gauge is 6 μm for the roller with diameter < 40 mm and length < 48 mm. That is to say, the length of each roller is different in a CRB, which would affect roller-end play and further affect roller tilt and axial load distribution. In this section, analysis is based on the severest situation for improving reliability. The length of the most bottom roller is the longest, and others are identical in length. Figure 3 shows the effect of the length variation on radial and axial load distribution, where 0 μm represents that the length of all rollers is identical, 2 μm represents that the bottom roller is two microns longer than the others, and so on. From Figure 4, it can be found that the radial load distribution is nearly unchanged with the increase of the length of the bottom roller, while the axial load distribution changed. Although the number of loaded rollers remains the same, the axial load of the bottom roller increases from 372 N to 710 N. This is because the longest roller restricts the movement of the ring to some extent and avoids other rollers' early contact. For the bottom roller, it can be deduced that the contact ellipse would increase and bending force of the flange would increase which may cause crack formation of the flange or the fracture of the flange. Furthermore, the tilt angle of the bottom roller increases from

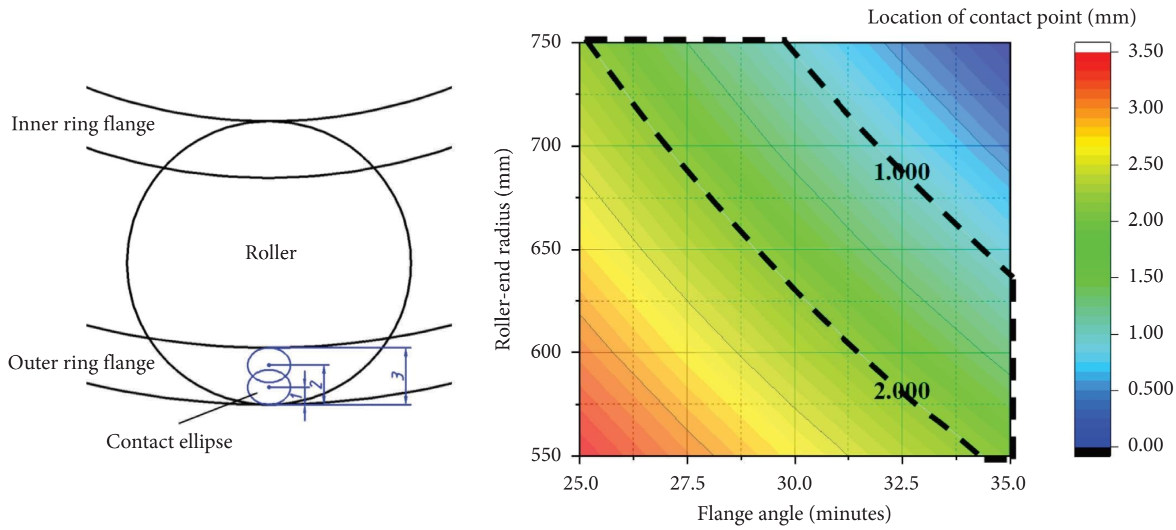

Figure 2: The effect of flange angle and roller-end radius on contact point.

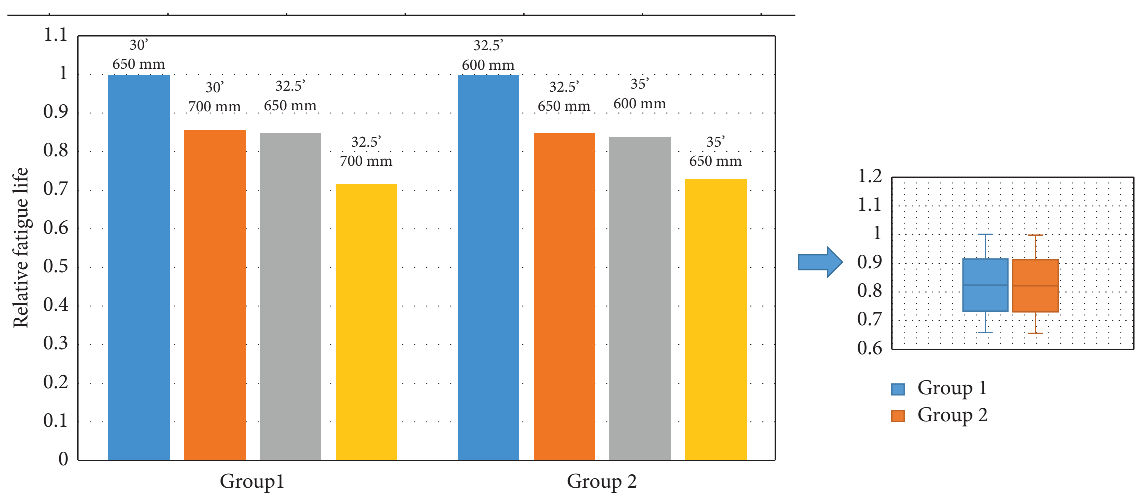

Figure 3: The relative fatigue life in different groups.

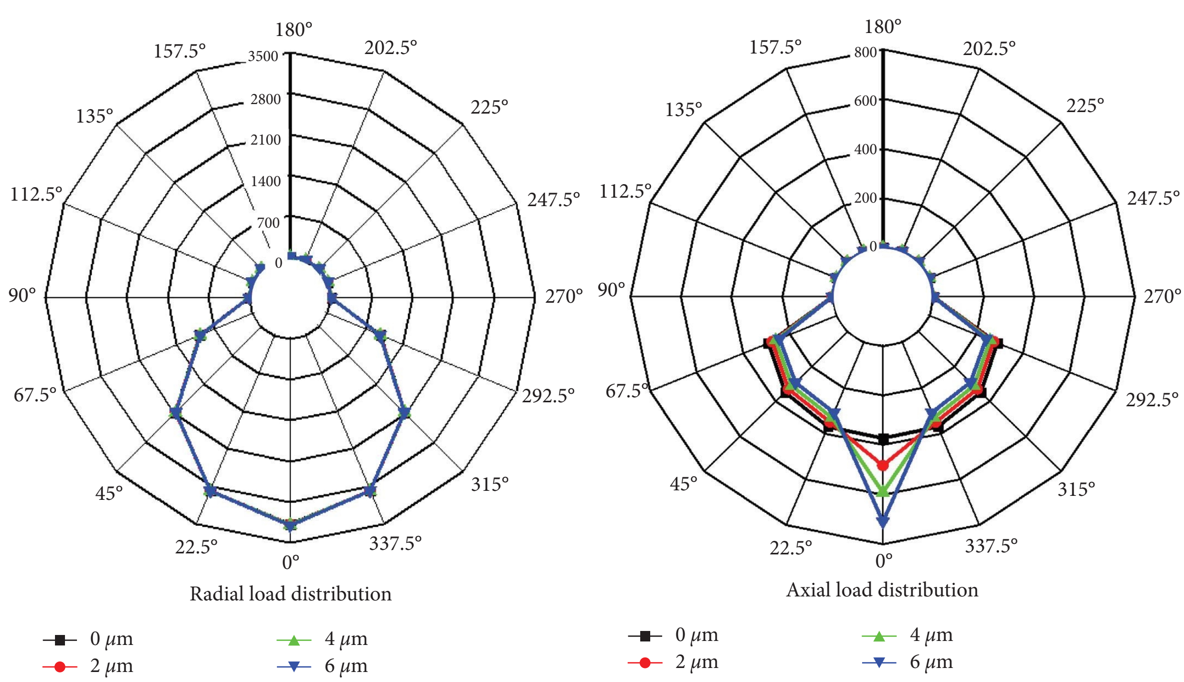

Figure 4: The effect of the length variation on radial and axial load distribution.

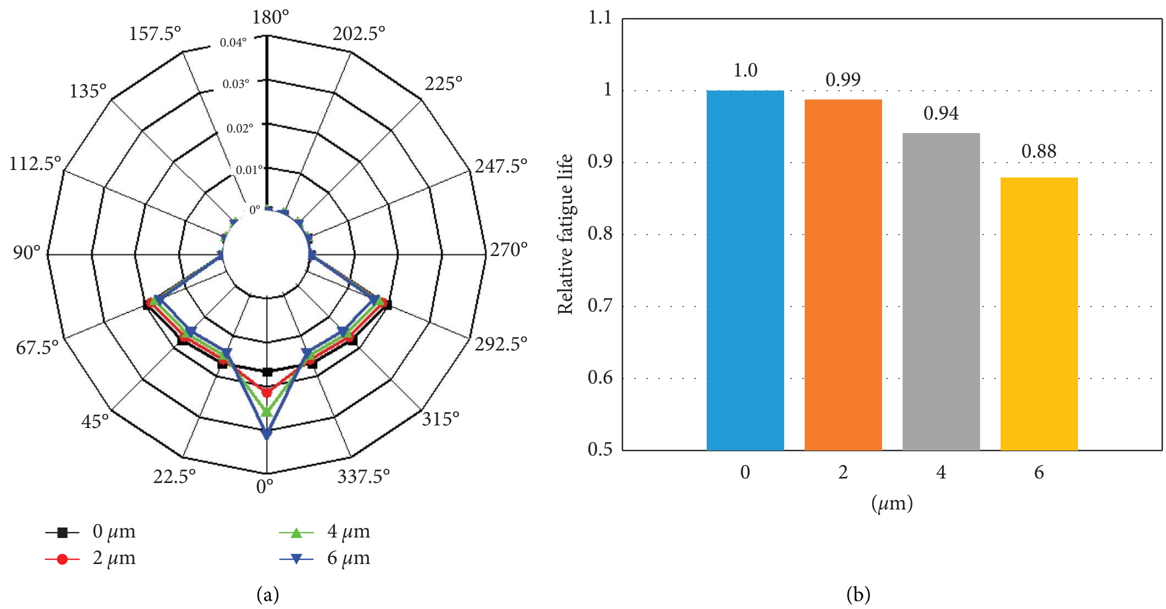

Figure 5: The effect of the length variation on: (a) tilt angle of roller. (b) fatigue life of roller bearings.

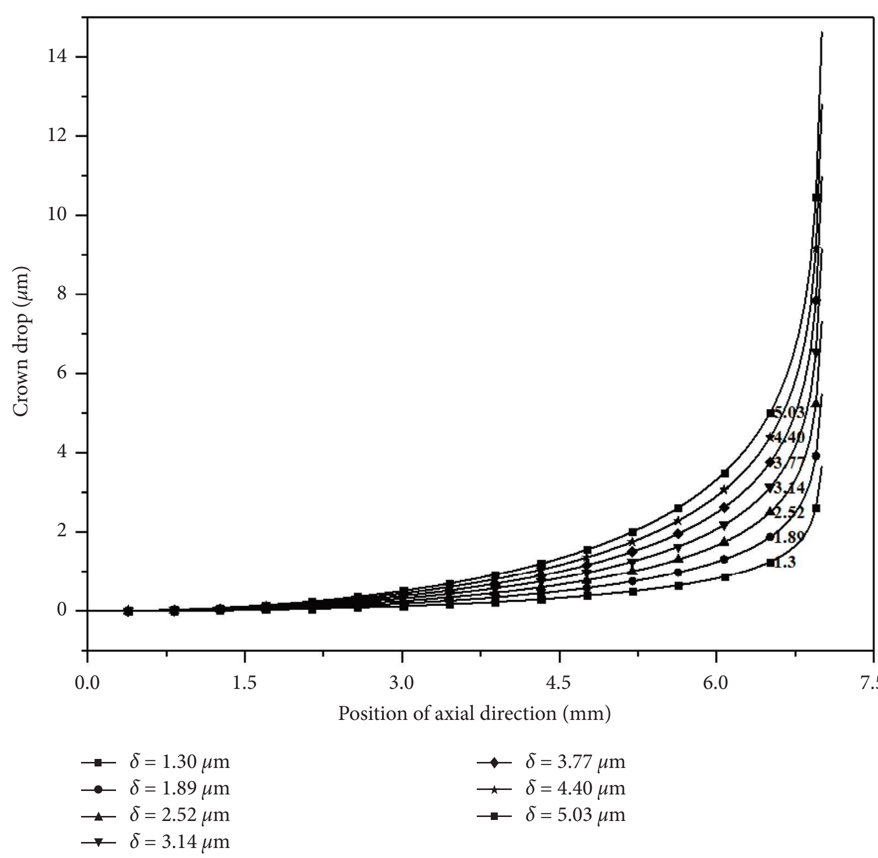

Figure 6: The logarithmic-profiled roller with different crown drops.

0.016 degrees to 0.036 degrees, as shown in Figure 5(a), due to the increase of the misalignment moment. This would lead to the occurrence of the edge loading and further reduce the fatigue life of CRBs, as shown in Figure 5(b). From Figure 5(b), it can be found that the fatigue life decreases as the interval of roller length gauge increases. Thus, for the CRBs under combined axial and radial loads, the interval of roller length gauge should be reduced. For the current working conditions, the 4 μm interval is sufficient to guarantee that the fatigue life is not less than 90% of the 0 μm interval.

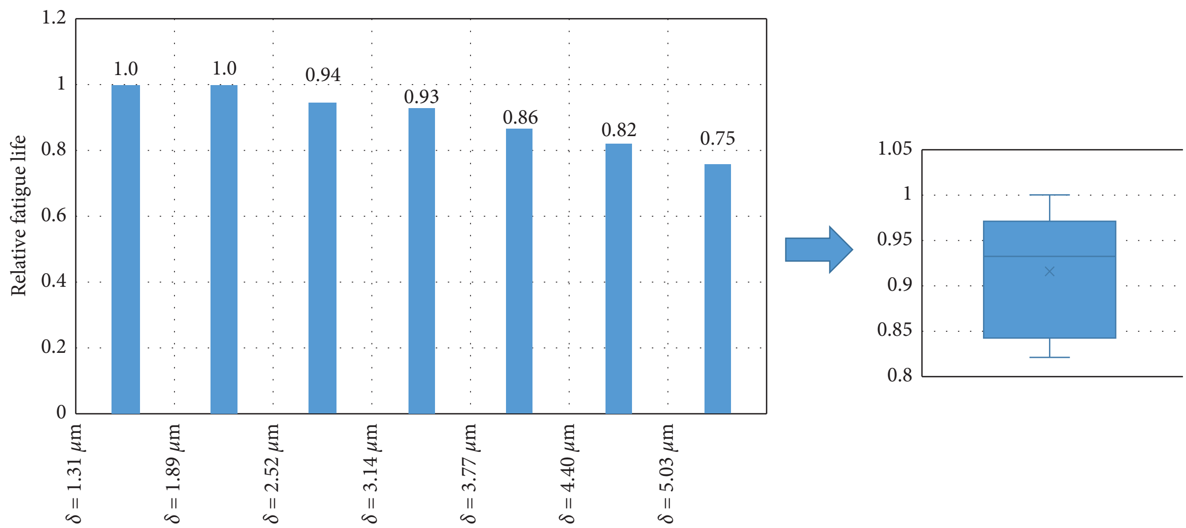

Figure 7: The fatigue life with the different roller profiles.

3.3. Tolerance Analysis of Roller Profile. Although lots of studies have shown that the roller profile has significant influence on bearing performance and given the optimization method of the roller profile, few works discussed the tolerance of roller profile required in engineering application. Fujiwara and Yamauchi [14] investigated the effect of crown drop on von Mises stress based on single roller-raceway contact and discussed the tolerance range of crown drop under the condition of a rate of stress <10%. Here, the relationship between the crown drops and fatigue life of CRBs was directly established. In order to define the difference between logarithmic-profiled rollers, the crown drop at the 0.47 l is set to be gauge points, where l is the effective length of the roller. The crown drop varies from 1.30 μm to 5.03 μm, as shown in Figure 6. From Figure 7, it can be found that when the crown drop is 1.31 μm and 1.90 μm, the fatigue life is the longest and basically identical, and the fatigue life decreases with increasing crown drop. If the fatigue life is set as at least 80% of the longest fatigue life, the benchmark of crown drop can be set as 1.90 μm, and the tolerance range is (0, +2.5 μm). In this situation, the average fatigue life can be 90% of the longest fatigue life. It should be noted that although when the crown drop is 1.31 μm, the fatigue life is the longest in this analysis, the crown drop here is chosen from 1.90 μm since smaller crown drop may cause edge loading.

4. Conclusions

In this study, tolerance analyses of cylindrical roller under combined axial and radial loads were conducted to evaluate the influence of tolerance range of key parameters, such as flange angle and roller-end radius, on contact and bearing performances. Some conclusions drawn from this study are as follows:

(1) The location of the contact point is very sensitive to the choice of flange angle and roller-end radius. When the flange angle and roller-end radius are controlled in the 5 minutes and 100 millimeters range, respectively, the contact ellipse of some sets would be truncated under the provided working conditions. Grouping the flange angle and roller-end radius and selective assembling would improve the contact performance.

(2) The interval of roller length gauge significantly affects the axial load distribution of CRBs and roller tilt angle. When the longest roller is located in the bottom position, the longer the roller length, the larger the axial load and the tilt angle of the bottom roller.

(3) The crown drop of logarithmic-profiled roller is a key parameter affecting the fatigue life of CRBs. In the presented analysis, when the benchmark of crown drop is 1.90 μm and the tolerance range is (0, +2.5 μm), the difference of the fatigue life is limited within 20%.

Data Availability

All data generated or analyzed during this study are included within this article. The code used during this study is available from the corresponding author on reasonable request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Acknowledgments

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References

[1] X. Ai, *Roller Bearing with Enhanced Roller-End and Flange Contact*, The Timken Company, North Canton, OH, USA, 2019.

[2] S. Ijuin, Yamamoto, and Takashi, *Cylindrical Roller Bearing*, NSK LTD, Tokyo, Japan, 2003.

[3] H. Sakoda, S. Natsumeda, Murai, and Takashi, *Radial Roller Bearing*, NSK LTD, Tokyo, Japan, 2003.

[4] G. Prisacaru, I. Bercea, N. Mitu, and S. Cretu, "Load distribution in a cylindrical roller bearing in high speed and combined load conditions," *European Journal of Mechanical and Environmental Engineering*, vol. 40, pp. 19–25, 1995.

[5] A. Chudzik and B. Warda, "Fatigue life prediction of a radial cylindrical roller bearing subjected to a combined load using FEM," *Eksploatacja i Niezawodnosc - Maintenance and Reliability*, vol. 22, no. 2, pp. 212–220, 2020.

[6] S. R. Brown and S. Y. Poon, "The lubrication of the roller-rib contacts of a radial cylindrical roller bearing carrying thrust load," *A S L E Transactions*, vol. 26, no. 3, pp. 317–324, 1983.

[7] G. Chen, F. Mao, and B. Wang, "Effects of off-sized cylindrical rollers on the static load distribution in a cylinder roller bearing," *Proceedings of the Institution of Mechanical Engineers - Part J: Journal of Engineering Tribology*, vol. 226, no. 8, pp. 687–696, 2012.

[8] G. Chen, B. Wang, and F. Mao, "Effects of raceway roundness and roller diameter errors on clearance and runout of a cylindrical roller bearing," *Proceedings of the Institution of Mechanical Engineers - Part J: Journal of Engineering Tribology*, vol. 227, no. 3, pp. 275–285, 2013.

[9] V. C. Tong and S. W. Hong, "Statistical investigation into the effects of distributed roller diameter error on the fatigue life of tapered roller bearings," *Journal of Mechanical Science and Technology*, vol. 31, no. 12, pp. 5977–5985, 2017.

[10] Y. J. Yu, G. Chen, J. Li, and Y. Xue, "Influence of geometric error of rollers on rotational accuracy of cylindrical roller bearings," *Xibei Gongye Daxue Xuebao/Journal of Northwestern Polytechnical University*, vol. 37, no. 4, pp. 774–784, 2019.

[11] Y. J. Yu, G. Chen, J. Li, Y. Xue, and B. Pang, "Prediction method for the radial runout of inner ring in cylindrical roller bearings," *Mathematical Problems in Engineering*, vol. 2017, Article ID 6584561, 13 pages, 2017.

[12] K. Ono and K. Takahasi, "Theoretical analysis of shaft vibration supported by a ball bearing with small sinusoidal waviness," *IEEE Transactions on Magnetics*, vol. 32, no. 3, pp. 1709–1714, 1996.

[13] K. Ono and Y. Okada, "Analysis of ball bearing vibrations caused by outer race waviness," *Journal of Vibration and Acoustics*, vol. 120, no. 4, pp. 901–908, 1998.

[14] H. Fujiwara and K. Yamauchi, "Tolerance design of logarithmic roller profiles in cylindrical roller bearings," *Journal of Advanced Mechanical Design Systems and Manufacturing*, vol. 4, pp. 728–738, 2010.

[15] J. Liu, Z. Yan, and Y. Shao, "An investigation for the friction torque of a needle roller bearing with the roundness error," *Mechanism and Machine Theory*, vol. 121, pp. 259–272, 2018.

[16] Z. Wang, J. Song, X. Li, and Q. Yu, "Modeling and dynamic analysis of cylindrical roller bearings under combined radial and axial loads," *Journal of Tribology*, vol. 144, no. 12, 2022.

[17] Iso, *Rolling Bearing - Methods for Calculating the Modified Reference Rating Life for Universally Loaded Bearings*, p. 16281, Geneva, Switzerland, 2008.

[18] T. A. Harris and T. A. Kotzalas, *Advanced Concepts of Bearing Technology*, CRC Press Taylor & Francis Group, Boca Raton, Florida, 2007.

[19] Iso, *Rolling Bearings - Steel Cylindrical Rollers - Dimensions and Tolerances*, p. 12297, Geneva, Switzerland, 2012.