Fatigue Life Prediction of Rolling Bearings Based on Modified SWT Mean Stress Correction

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Title page

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ORIGINAL ARTICLE

Fatigue Life Prediction of Rolling Bearings Based on Modified SWT Mean Stress Correction

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Abstract: Mean stress has a great influence on fatigue life, commonly used stress-based life prediction models can only fit the test results of fatigue life under specific stress ratio or mean stress but cannot describe the effect of stress ratio or mean stress on fatigue life. Smith, Watson and Topper (SWT) proposed a simple mean stress correction criterion. However, the SWT model regards the sensitivity coefficient of all materials to mean stress as 0.5, which will lead to inaccurate predictions for materials with a sensitivity coefficient not equal to 0.5. In this paper, considering the sensitivity of different materials to mean stresses, compensation factor is introduced to modify the SWT model, and several sets of experimental data are used for model verification. Then, the proposed model is applied to fatigue life predictions of rolling bearings, and the results of proposed method are compared with test results to verify its accuracy.

Keywords: Fatigue life prediction, Mean stress correction, Modified SWT model, Rolling bearings

1 Introduction

Rolling bearings are important parts in many rotational mechanisms and are tremendously used in various industries. The rapid development of modern society, especially the manufacturing sector, requires functional life of rolling bearings under extremely harsh conditions, such as heavy load, high speed and high temperature, etc.

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Therefore, the reliability, service life and performance of rolling bearings are even important.

Typical life prediction models for rolling bearings are based on statistics, include Lundberg-Palmgren (L-P) model, Ioannides-Harris (I-H) model, and Zaretsky model [1–4]. Currently, researches related have been carried out. For instance, An experimental procedure using vibration modal analysis to predict the fatigue life of each individual rolling element bearing separately presented by Yakout [5]. Wang et al. [6] proposed two novel mixed effects models for rolling element bearings' prognostics, each model was proved that is able to simultaneously model Phases I and II of the bearing degradation process. Cui et al. [7] established a Switching Unscented Kalman Filter method for remaining useful life prediction of rolling bearings. Ahmad et al. [8] introduced a reliable technique for health prognosis of rolling element bearings, which infers a bearing's health through a dimensionless health indicator (HI) and estimates its remaining useful life (RUL) using dynamic regression models. Wang et al. [9] proposed a method for life prediction of industrial rolling bearing based on state recognition and similarity analysis.

The mentioned models are mainly based on artificial intelligence and statistical regression methods, and those require sufficient experimental data for model training. The modelling process of the mentioned methods ignore failures of rolling bearings. Accordingly, scholars turned to investigate theoretical methods based on mechanical models of rolling bearing fatigue crack damage. Warda et al. [10] introduced a fatigue life prediction method of radial cylindrical roller bearings, in which the influence of bearing geometric parameters were considered. Shi et al. [11] presented a calculation method of relative fatigue life considering surface texture on high-speed and heavy-load

ball bearing.

Yang et al. [12] discussed the mechanical properties of double-row tapered roller bearings through expanding the mathematical model of three degrees of freedom, and analysed the contact load and fatigue life of bearings under different loads. Quagliato et al. [13] predicted the life of roller bearings by accelerated testing approach and finite element (FE) models. He et al. [14–15] analyzed the fatigue life of the wind turbine slewing bearing through finite element simulation and FE-SAFE.

Rolling bearings are subjected to alternating loads during operation, the load amplitude and mean stress continuously change with different working conditions and together determine the fatigue life [16–21]. Various researches on the effect of mean stress on fatigue life have been carried out. For instance, Barbosa et al. [22] proposed an artificial neural network prediction method considering the influence of average stress on the fatigue life of metallic materials. Zhang et al. [23] analyzed the influence of mean stress and phase angle on multiaxial fatigue behavior of TiAl alloy and established a life model. Benedetti et al. [24] developed a new fatigue criterion based on strain-energy-density (SED) to illustrate the influence of mean stress and plasticity on the uniaxial fatigue strength. Kalombo et al. [25] used an artificial neural network to predict the fatigue life of an all-aluminum alloy 1055 MCM conductor considering different mean stresses. Li et al. [26] established a new fatigue model based on the effect of mean stress on high-cycle fatigue performance and compared it with some other models. Laszlo et al. [27] introduced a numerical fatigue assessment method for composite plates, which considered mean stress correction and multiaxial fatigue failure criterion inspection. A fatigue life prediction method considering the effects of casting defect and mean stress was presented by Duan [28]. Rolling contact fatigue is the most important failure mode of bearings [29–30]. The contact load and stress at each contact point of bearing are cyclically pulsating and belong to asymmetric cyclic load. Therefore, the influence of mean stress needs to be considered when predicting the fatigue life of rolling bearings.

In this paper, we attempt to consider the sensitivity of different materials to mean stress and propose a modified life prediction model based on the SWT correction criterion. The proposed model is used to predict the fatigue life of rolling bearings. The rest of this paper is organized as follows. Section 2 introduces common stress-based life prediction models, and then gives the procedures of establishing improved models. Section 3 develops the

fatigue life prediction model of rolling bearings based on the proposed model. Section 4 performs model validation using the experimental data of 1Cr11Ni2W2MoV, GH4133 and GCr15, and verifies the applicability of proposed model for the prediction of the fatigue life of rolling bearings. In Section 5, some conclusions are drawn based on the current investigation.

2 Modified Fatigue Life Prediction Model Based on SWT Criterion

2.1 Stress-based Life Prediction Methods

The stress-based life prediction method is the most widely used prediction method in engineering. Its theoretical basis is the S-N curve, which is generally expressed by the Basquin formula, as shown in Eq. (1).

$$\sigma N_f^{-b} = A, \quad (1)$$

where N_f represents fatigue life, A represents the fatigue strength constant, which is an inherent property of the material, b is the material constant.

Some test results show that the specimen can withstand countless stress cycles without breaking under a stress lower than a certain critical stress amplitude, and the fatigue life tends to be infinite. Basquin formula fails to reflect the existence of fatigue limit and the influence of fatigue limit on fatigue in the long-life zone, and it is difficult to fit the fatigue test results in the long-life zone. The complete stress-fatigue life curve is shown in Figure 1. Considering the influence of the fatigue limit, the relationship between fatigue life, fatigue limit stress and stress range is established by Weibull [31], as shown in Eq. (2).

$$N_f = C_f (\sigma_a - \sigma_{ac})^\beta, \quad (2)$$

where σ_{ac} is the endurance-limit stress, C_f and β are material constants determined by experiments.

The components are often subjected to asymmetric cyclic loads during service, as shown in Figure 2. The magnitude of the load and the mean stress together determine its fatigue life. Because the Weibull formula can only fit the test results of fatigue life under a certain stress ratio or mean stress, it cannot show the effect of stress ratio or mean stress on fatigue life. Therefore, it is necessary to modify the mean stress of the life model.

Figure 1 The complete stress-fatigue life curve

Figure 2 The asymmetric cyclic load

Because the Walker criterion considers the sensitivity of different materials to the mean stress, it has a good correction effect on all materials [32]. The Walker correction model is shown in Eq. (3). However, the mean stress sensitivity coefficient γ in the Walker criterion requires a lot of fatigue tests, which is not only inefficient, but also not economical. Considering that the mean stress sensitivity coefficient is difficult to obtain, Smith, Watson and Topper [33] proposed a simple form of mean stress correction criterion, whose expression is shown in Eq. (4). Hence, the SWT criterion is used to modify Eq. (2) to obtain a fatigue life prediction model considering the mean stress effect, as shown in Eq. (5).

$$\sigma_{ar} = \sigma_{\max}^{1-\gamma} \sigma_a^\gamma = \sigma_{\max} \left(\frac{1-R}{2} \right)^\gamma, \quad (3)$$

$$\sigma_{ar} = \sqrt{\sigma_{\max} \sigma_a} = \sigma_{\max} \sqrt{\frac{1-R}{2}}, \quad (4)$$

$$N_f = C_0 (\sigma_{ar} - \sigma_{ac})^{\beta_0} = C_0 \left(\sigma_{\max} \sqrt{\frac{1-R}{2}} - \sigma_{ac} \right)^{\beta_0}, \quad (5)$$

where σ_{ar} is equivalent stress amplitude, $\sigma_{\max}$ is maximum stress, σ_a is stress amplitude, R denotes stress ratio. γ denotes the mean stress sensitivity coefficient, and its value is between [0 1], the larger γ , the less sensitive the material to mean stress, and vice versa. C_0 and β_0 are material constants.

2.2 Proposed Model

SWT criterion is a special form of Walker criterion, which regards the sensitivity of different materials to the mean stress as the same constant, that is, $\gamma=0.5$. Hence, for the material whose γ is close to 0.5, the life prediction corrected by the SWT criterion has good prediction accuracy, while for the material whose γ deviates from 0.5, the prediction result will have a large error.

Dowling [34] found that γ has a certain relationship to fatigue performance parameter yield limit of materials. For the same type of material, as the yield limit increases, γ decreases. In this paper, a compensation factor α is introduced to consider the sensitivity of the material to the mean stress. Substituting α into Eq. (5), a modified fatigue life prediction model based on SWT criterion is proposed, as shown in Eq. (6).

$$N_f = C_1 \left(\alpha \sigma_{\max} \sqrt{\frac{1-R}{2}} - \sigma_{ac} \right)^{\beta_1}, \quad (6)$$

$$\alpha = \frac{2\sigma_b}{\sigma_b + \sigma_0}$$

where C_1 and β_1 are material constants. σ_b indicates the yield limit, σ_0 represents the yield limit of similar materials when $\gamma=0.5$.

3 Fatigue Life Prediction Model of Rolling Bearings Based on Proposed Model

3.1 Load and Stress Distribution of Rolling Bearings

The main performance parameters of rolling bearings, such as deformation, contact stress between rolling elements and rings, stiffness, and fatigue life, can only be calculated after the load distribution is determined. Therefore, the calculation of load distribution is the primary step of rolling bearings performance investigation.

The load acting on the bearing is transmitted from one ring to the other through the rolling elements, so the bearing capacity is determined by the rolling element load. The rolling bearing under the action of radial load is shown in Figure 3. The load distribution of the rolling elements

can be computed as [35]:

$$Q_\psi = Q_{\max} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^{1.5}, \quad (7)$$

where ψ denotes angular position, Q_ψ is the rolling element load at an angle ψ . $Q_{\max}$ is the maximum rolling element load, and T demonstrates the load zone parameter.

According to the rule of force balance,

$$\begin{aligned} F_r &= \sum Q_\psi \cos \psi \\ &= \frac{\sum \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^{1.5} \cos \psi}{Z} Q_{\max} \end{aligned} \quad (8)$$

The load distribution integral J_r is introduced,

$$\begin{aligned} J_r &= \frac{1}{2\pi} \int_{-\psi_0}^{+\psi_0} \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^{1.5} \cos \psi d\psi \\ &\approx \frac{\sum \left[1 - \frac{1}{2T} (1 - \cos \psi) \right]^{1.5} \cos \psi}{Z} \end{aligned} \quad (9)$$

And then,

$$Q_{\max} = \frac{F_r}{Z J_r}, \quad (10)$$

where, Z demonstrates the number of rolling element. ψ_0 denotes the bearing range, and $\cos \psi_0 = 1 - 2T$.

Figure 3 Force distribution of rolling bearings under the radial load

When the centrifugal force is not considered, the contact load Q_{ij} between the inner ring and the rolling elements and the contact load Q_{oj} between the outer ring and the rolling elements are equal. Considering the centrifugal force F_c of the rolling element, the radial force balance equation of the rolling element can be expressed as:

$$Q_{oj} - Q_{ij} - F_c = 0, \quad (11)$$

$$F_c = \frac{\pi}{12} \rho D_b^3 D_m \omega_b^2, \quad (12)$$

where D_m is the bearing circle diameter, D_b is the rolling element diameter, ρ denotes the density of the rolling elements, ω_b represents the revolution angular velocity of the rolling element.

To predict bearing's life, it is necessary to obtain the contact stress and deformation of a bearing. When two curved objects are pressed against each other under a load, a certain contact zone is generated at the contact point. Since the rolling contact between raceway and rolling elements is a curved body, the Hertz elastomer contact theory can be used to calculate the contact stress and deformation in the rolling bearing.

For the ball bearing, the contact zone between a ball and a raceway is elliptical based on the Hertz contact theory. The semi-major axis of the ellipse is represented by a , and the semi-minor axis is represented by b , as shown in Figure 4. The contact stress in the contact zone is distributed as an ellipsoid, as shown in Figure 5. When the contact load is Q , the contact stress at any point (x, y) in the contact zone can be expressed as follows [36]:

Figure 4 Contact ellipse

Figure 5 Stress distribution in the contact zone

$$\sigma = \frac{3Q}{2\pi ab} \left[1 - \left(\frac{x}{a} \right)^2 - \left(\frac{y}{a} \right)^2 \right]^{1/2}, \quad (13)$$

The basic equations for calculating the contact stress are derived by the Hertz contact theory and are shown as follows:

$$a = \mu_3 \sqrt{\frac{3}{E} (1 - \lambda^2) \frac{Q}{\sum \rho}}, \quad (14)$$

$$b = \nu_3 \sqrt{\frac{3}{E} (1 - \lambda^2) \frac{Q}{\sum \rho}}, \quad (15)$$

$$\sigma_{\max} = \frac{3Q}{2\pi ab}, \quad (16)$$

where $\sigma_{\max}$ represents the maximum contact stress, μ and ν are elliptic integrals related to the curvature function $F(\rho)$, E and λ are the elastic modulus and Poisson's ratio of the material, respectively, and Q denotes the rolling element load. $\sum \rho$ demonstrates the sum of the principal curvatures at the contact.

3.2 Fatigue Life Prediction Model of Rolling Bearings

The life prediction model proposed by Lundberg and Palmgren (L-P model) is commonly used in engineering to

predict the fatigue life of rolling bearings. But the biggest flaw is that it cannot take the microstructure of the material into account, which affects the universality of the model. The fatigue life prediction method proposed in this paper is based on the S-N curve of the material, which can describe the fatigue characteristics of the material well.

The life obtained by Eq. (6) is the number of stress cycles. The number of stress cycles of a rolling bearing refers to the number of times that a certain point on the raceway of a bearing is subjected to stress within a certain number of rotations of the bearing, it is related to the number of loaded rolling elements in the load zone. The contact load and stress of the rotating ring and rolling elements change when they pass through the various points in the load zone, the contact load and stress at each contact point are characterized by a pulsating cycle; they are not loaded in the unloaded zone. The contact load and stress distribution at each point of the rotating ring are shown in Figure 6. The load and stress of each point of the fixed ring are unequal, and the contact load and stress at each load point show the same characteristics of pulsation cycle, but the amplitude value is different. The contact load and stress distribution at each point of the fixed ring are shown in Figure 7.

Figure 6 The contact load and stress distribution at each point of the rotating ring

Figure 7 The contact load and stress distribution at each point of the stationary ring

Since the life unit of a rolling bearing is usually expressed in revolutions, it is necessary to convert the life obtained by Eq. (6). When the inner ring rotates and the outer ring is stationary. Under the radial load, the contact stress of the outer ring raceway is the largest at the 0° angular position of the load zone, which is the place where the outer ring is most likely to fail. The number of stress cycles and the life of the outer ring at this point can be expressed as Eq. (17). The amplitude of the stress on a point on the inner ring raceway is different every time due to the rotation of the inner ring, and it changes periodically. According to Miner's damage accumulation theory, when the inner ring rotates for one revolution, the damage to the inner ring is expressed as Eq. (18), and the life of the inner ring can be expressed as Eq. (19).

$$L_e = \frac{N_e}{u_e}, \quad (17)$$

$$D = \sum_{j=1}^{u_i} \frac{n_j}{N_j}, \quad (18)$$

$$L_i = \frac{1}{D} = \frac{1}{\sum_{j=1}^{u_i} \frac{n_j}{N_j}}, \quad (19)$$

where N_e denotes the number of stress cycles at the maximum stress position on outer raceway, and N_i denotes the number of stress cycles of inner raceway under the contact stress σ_j , they can be obtained by Eq. (6). n_j is the number of contact times of the inner ring raceway under the stress σ_j . L_i and L_e represent the life of inner ring and outer ring, respectively. u_i and u_e are the number of rolling elements passing through a certain point of the inner and outer rings when the inner ring rotates one revolution.

For a ball bearing with an outer ring speed of n_e and an inner ring speed of n_i , the contact angle is zero under pure radial force. When the cage rotates once concerning the inner or outer ring, Z rolling elements are passing through a certain point of the inner or outer ring. u_i and u_e can be expressed as Eq. (20) and Eq. (21).

$$u_i = \frac{Zn_{ci}}{n_i}, \quad (20)$$

$$u_e = \frac{Zn_{ec}}{n_e}, \quad (21)$$

where Z represents the number of rolling elements. n_{ci} denotes the rotation speed of the cage relative to the inner ring, and n_{ec} denotes the rotation speed of the outer ring relative to the cage. They can be calculated as follows [37].

$$n_c = \frac{1}{2} \left[n_i \left(1 - \frac{D_b}{D_m} \right) + n_e \left(1 + \frac{D_b}{D_m} \right) \right], \quad (22)$$

$$n_{ci} = n_c - n_i = \frac{1}{2} (n_e - n_i) \left(1 + \frac{D_b}{D_m} \right), \quad (23)$$

$$n_{ec} = n_e - n_c = \frac{1}{2} (n_e - n_i) \left(1 - \frac{D_b}{D_m} \right), \quad (24)$$

where n_c is the rotation speed of the center of roll-ing element. D_b is the diameter of rolling element and D_m is the mean diameter of bearing.

Substituting Eqs. (20) - (24) into Eqs. (17) - (19), the fatigue life of the inner and outer rings of a rolling bearing can be obtained. Then, the overall life of ball bearing is obtained by Eq. (25) [35].

$$L = \left[(L_i)^{-10/9} + (L_e)^{-10/9} \right]^{-9/10}, \quad (25)$$

4 Case Study

4.1 Validation of Modified Fatigue Life Prediction Model Based on SWT Criterion

Several sets of fatigue experimental data from materials 1Cr11Ni2W2MoV, GH4133 and GCr15 [38–40] under different stress ratios are used to determine the material constants and validate the prediction accuracy of proposed model. The material properties of 1Cr11Ni2W2MoV, GH4133 and GCr15 are provided in Table 1. The life experimental data of 1Cr11Ni2W2MoV under the condition of $R=-1$ is shown in Table 2. Table 3 shows the life experimental data of GH4133 under the conditions of $T=250^\circ\text{C}$ and $R=0.44$. Table 4 and Table 5 respectively show the life data of the contact fatigue test at $R=0$ and the life data of the torsion fatigue test at $R=-1$ of GCr15.

Table 1 Material properties of 1Cr11Ni2W2MoV, GH4133 and GCr15

Material	E (GPa)	Poisson's ratio	σ_b (MPa)	Density (g/cm ³)
1Cr11Ni2W2MoV	180	0.277	979	7.8
GH4133	223	0.36	878	8.21
GCr15	207	0.3	1617	7.81

Table 2 Experimental data for 1Cr11Ni2W2MoV at $R=-1$

σ_{max} (Mpa)	Nt (cycles)
1183	2590
1083	2912
987	4078
919	9402
839	23440
817	26124

Table 3 Experimental data for GH4133 at $R=0.44$

σ_{max} (Mpa)	Nt (cycles)
935	2535
933	2191
905	3026
905	2930
887	2508
865	2879
863	3720
862	4276
841	5320
806	5610
777	8461
756	17710
749	12855
744	12141
741	14352

Table 4 Contact fatigue experimental data for GCr15 at $R=0$

σ_{max} (Mpa)	Nt (cycles)
5000	12.8802e+7
5300	11.4798e+7
5600	8.5787e+7
5900	7.5976e+7
6200	7.1121e+7

Table 5 Torsion experimental data for GCr15 at $R=-1$

σ_{max} (Mpa)	Nt (cycles)
500	5.95e+7
630	3.12e+6
760	8.43e+5
800	7.56e+5
950	7e+4
1000	4.68e+4

The experimental values are compared with the predicted results of the proposed model and the SWT model, as shown in Figures. 8 - 11, respectively.

Figure 8 Predicted life vs. the tested life of 1Cr11Ni2W2MoV **Figure 9** Predicted life vs. tested life of GH4133 **Figure 10** Predicted life vs. tested life of GCr15 under contact test **Figure 11** Predicted life vs. tested life of GCr15 under torsion test

Note from Figure 8 for 1Cr11Ni2W2MoV and Figure 9 for GH4133, it can be clearly observed that the prediction results of the proposed model is more consistent with test data than the prediction results of the SWT model. Almost all the predicted results of the proposed model are within the ± 2 scatter band, but many of the prediction results of the SWT model are outside the ± 2 scatter band and have a large scatter. Note from Figure 10 for GCr15 under contact test, a good agreement for the proposed model and SWT

model can be found that all the predicted results are within the ± 2 scatter band, but the results of the proposed model are closer to the diagonal and have a smaller scatter. Note from Figure 11 for GCr15 under torsion test, the predicted results of the proposed model and SWT model are quite similar, while only one prediction result of the proposed model is outside the ± 2 scatter band, and the predicted lives of the SWT model shows much more predicted results are outside the ± 2 scatter band than that provided by the proposed model.

In order to obviously show the differences between the proposed model and the SWT model, the predicted life deviation between the logarithmic predicted life and experimental life is used to describe the estimation errors, as shown in Eq. (26), is utilized to indicate the accuracy of these models [41–42].

$$P_{error} = \lg(N_p) - \lg(N_t), \quad (26)$$

The estimated errors of proposed model and SWT model on the four sets of experimental data are shown in Figures 12 - 15.

Figure 12 Estimation error of 1Cr11Ni2W2MoV

Figure 13 Estimation error of GH4133

Figure 14 Estimation error of GCr15 under contact test

Figure 15 Estimation error of GCr15 under torsion test

Overall, it can be observed from Figures 8-15 that the proposed model can provide accurate and reasonable predictions.

4.2 Life Prediction of the Proposed Model for Rolling Bearings

In this section, a deep groove ball bearing (code 6206) is used in the experiment. Its material is GCr15 and its specification is shown in Table 6. Under the conditions of radial force of 5KN and a rotation speed of 12000rpm, the distribution of rolling element load is obtained through quasi-static analysis. Then, according to the Hertz contact theory, the contact stress of the rolling bearing is analyzed. Finally, at the current conditions, the maximum contact stress of the inner raceway is 2924.6MPa, and the maximum contact stress of the outer raceway is 2567.9MPa. The rolling element load at different angular position is shown in Figure 16, and the contact stress of inner raceway distribution at different angular position is shown in Figure 17.

Table 6 Specification of the tested bearing

Parameter	Value
Ball nominal diameter D_b (mm)	9.525
Pitch diameter of the bearing D_m (mm)	46
Coefficient of inner raceway groove curvature radius f_i	0.515
Coefficient of outer raceway groove curvature radius f_e	0.52
Number of balls Z	9

Figure 16 Rolling element load at different angular position **Figure 17** Contact stress distribution of inner raceway at different angular position

According to the expression of GCr15's contact fatigue life in the proposed model, combined with Eqs. (17) - (19), the fatigue life of inner ring and outer ring are predicted, where $L_i=4.469e+8$ and $L_e=6.998e+8$. Finally, through Eq. (25), the predicted life of rolling bearing is obtained, where $L=2.915e+8$.

The comparison between the experimental life that three test bearings at the same conditions and the predicted life is shown in Figure 18 [43]. It can be clearly identified that the prediction results are agree with the experimental data with a life factor of ± 2 . In general, applying the proposed model for fatigue life prediction of rolling bearings can provide reasonable prediction results.

Figure 18 Predicted life L_p vs. the tested life L_t under $F_t=5\text{KN}$ and $n=12000\text{rpm}$

5 Conclusions

In this paper, a modified model based on SWT criterion that considers the mean stress effect and sensitivity is proposed to predict fatigue life. Several sets of experimental data are used for verifying the proposed method. The conclusions are drawn as follows:

- (1) Considering the sensitivity of different materials to mean stress, a modified model is established based on the SWT criterion to estimate the fatigue life. The experimental data of three materials: 1Cr11Ni2W2MoV, GH4133 and GCr15 were used for model verification. Compared with the SWT model, the applicability of the proposed model is better, and the life prediction results of the three materials are more accurate.
- (2) Combined with the proposed model, the life prediction model of rolling bearings is established. Finally, the fatigue life of ball bearing (code 6206) is predicted under the working conditions on 5KN radial force and 12000rpm, then the predicted result is compared with the experimental results of three test bearings. The comparison result shows that the prediction result is in good agreement with the experimental results, and it is feasible to apply the proposed model to the fatigue life prediction of rolling bearings.

6 Declaration

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Availability of data and materials

The datasets supporting the conclusions of this article are included within the article.

Authors' contributions

HH was in charge of the whole trial; AY wrote the manuscript; YL, HL and YZ assisted with sampling and laboratory analyses. All authors read and approved the final manuscript.

Competing interests

The authors declare no competing financial interests.

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Figures

Figure 1

The complete stress-fatigue life curve

Figure 2

The asymmetric cyclic load

Figure 3

Force distribution of rolling bearings under the radial load

Figure 4

Contact ellipse

Figure 5

Stress distribution in the contact zone

Figure 6

The contact load and stress distribution at each point of the rotating ring

Figure 7

The contact load and stress distribution at each point of the stationary ring

Figure 8

Predicted life vs. the tested life of 1Cr11Ni2W2MoV

Figure 9

Predicted life vs. tested life of GH4133

Figure 10

Predicted life vs. tested life of GCr15 under contact test

Figure 11

Predicted life vs. tested life of GCr15 under torsion test

Figure 12

Estimation error of 1Cr11Ni2W2MoV

Figure 13

Estimation error of GH4133

Figure 14

Estimation error of GCr15 under contact test

Figure 15

Estimation error of GCr15 under torsion test

Figure 16

Rolling element load at different angular position

Figure 17

Contact stress distribution of inner raceway at different angular position

Figure 18

Predicted life L_p vs. the tested life L_t under $F_r=5\text{KN}$ and $n=12000\text{rpm}$