A new fatigue damage accumulation rating life model of ball bearings under vibration load

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Abstract

Purpose – Bearings in electric machines often work in high speed, light load and vibration load conditions. The purpose of this paper is to find a new fatigue damage accumulation rating life model of ball bearings, which is expected for calculating fatigue life of ball bearings more accurately under vibration load, especially in high speed and light load conditions.

Design/methodology/approach – A new fatigue damage accumulation rating life model of ball bearings considering time-varying vibration load is proposed. Vibration equations of rotor-bearing system are constructed and solved by Runge–Kutta method. The modified rating life and modified reference rating life model under vibration load is also proposed. Contrast of the three fatigue life models and the influence of dynamic balance level, rotating speed, preload of ball bearings on bearing's fatigue life are analyzed.

Findings – To calculate fatigue rating life of ball bearings more accurately under vibration load, especially in high speed and light load conditions, the fatigue damage accumulation rating life model should be considered. The optimum preload has an obvious influence on fatigue rating life.

Originality/value – This paper used analytical method and model that is helpful for design of steel ball bearing in high speed, light load and vibration load conditions.

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Keywords Ball bearings, Fatigue rating life, Optimum preload, Vibration load

Paper type Research paper

1. Introduction

Continuous research is required by the bearing manufacturer to promote the bearing performances. The relationship between the design, manufacture, material and bearing fatigue life have been studied by many scholars and bearing manufacturers (Ebert, 2010; Forster *et al.*, 2009). Electric machines are extensively used and are at the core of most engineering processes. The bearings in the electric machines often work in high speed and light load (Magdun *et al.*, 2010). Meanwhile, the vibration of the machine causes the bearings usually under variable vibration load condition. In the electric machine system, unbalance load of the rotor is the most important source of excitation (Liu *et al.*, 2019). Fatigue failure of bearings occurs when the value of damage accumulation comes to a certain extent. A method for calculating fatigue damage rating life of bearings under vibration load should be founded.

Early work of rolling bearing's life prediction was done by test. The first model of rolling bearing fatigue life was founded by Lundberg and Palmgren (1949). L-P theory gives the relationship between the number of repeated load cycles experienced by the bearing raceway and the survival probability. Zaretsky (1987) derived fatigue life of a whole system based on the life of all components:

$$L_s = \left[(L_1)^{-e} + (L_2)^{-e} + \cdots + (L_n)^{-e}\right]^{-1/e} \tag{1}$$

Where, L_1, L_2,L_n are life of the components and e is Weibull distribution coefficient.

International Organization for Standardization (ISO) simplified L-P theory and obtained basic rating life for rolling bearing L_{10}. However, in many cases, the calculated life was found to be significantly different from the actual experimental life. The lubrication and material should be considered.

Deutsches Institut für Normung (DIN)/ISO 281 was proposed considering material and lubrication factor, bearing fatigue life solved from DIN/ISO 281 was much greater than the basic rating life L_{10} (Trojahn *et al.*, 1999). Subsequently, the modified rating life model was proposed (ISO/TS 281, 2007):

$$L_{nm} = a_1 a_{iso} L_{10} \tag{2}$$

Where, n stands for failure probability, a_1 is reliability factor, a_{iso} is life modification factor.

Although material, lubricant and contamination factors can be considered in ISO/281. However, in order to consider the

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influence of misaligned bearings and bearing clearance during operation, modified reference rating life model was proposed based on each rolling element's contact load (ISO/16281, 2008). The model is used widely for bearing fatigue life prediction.

However, under variable load condition, friction status and contact stress will be changed due to vibration load (Nierlich and Gegner, 2012). Therefore, variable contact load between the rolling elements and rings should be considered in the calculation of bearing fatigue life. In this case, cumulative damage models can be adapted to calculate the fatigue life. Miner (1945) proposed the linear damage rule, which was written as equation (3):

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \cdots = 1 \cdots \text{or} \sum \frac{n_i}{N_i} = 1 \tag{3}$$

Where, n_i is cycles number. N_i is cycles number to failure under σ_i.

Pavlou (2002) demonstrated that loading sequence has a large effect on the failure life. However, Miner's theory cannot consider evolutionary history of the material in different load sequence. Kwofie and Rahbar (2012) proposed a fatigue cumulative damage formula considering load sequence as equation (4):

$$\sum \frac{n_i}{Ni}\left(\frac{\ln(N_i)}{\ln(N_0)}\right) = 1 \tag{4}$$

Vibration of rotor-bearing system was studied widely, and the system often shows nonlinear vibration characteristics. Kappaganthu and Nataraj (2011) found the dynamic characteristics of angular contact ball bearings. The system's vibration influence the mechanical behaviors of the supporting bearing. Yessine *et al.* (2015) studied the relationship between fatigue failure of bearing and vibration properties. They found the modal parameters can change the fatigue process of bearings. Jacobs *et al.* (2016) investigated the influence of variable loads on deep groove ball bearing life through experiment. Results showed that the dynamic load could result in polishing of the raceways caused by sliding. Sola *et al.* (2017) used damage parameters to predict the laminates fatigue life under different loads. They assumed that the parameters are directly proportional to fatigue damage. Li *et al.* (2018) proposed a new damage evolution model for rolling contact fatigue of cylindrical roller bearing. Crack initiation and crack propagation were simulated based on finite element software. The results obtained were consistent with L-P model. Yang and An (2018) studied the fatigue life of spherical roller bearing of mixing truck considering drum's vibration based on the Miner's rule. Results showed that the fatigue life when considering mix drum's vibration was shorter than that without considering vibration.

This paper will present a new fatigue damage accumulation rating life model of ball bearings considering vibration load. Vibration equations of the bearing system are constructed to obtain bearing contact loads. Stability of the system is solved by Floquet multipliers. The influence of dynamic balance level, rotating speed, preload on bearing's fatigue life are analyzed.

2. Contact load model of ball bearings considering vibration load

For a rigid rotor system supported by a pair of angular contact ball bearings, variable loads causing by unbalance load of the rotor system are considered. θ_j is azimuth angle of the jth ball, which is shown in Figure 1(a). Figure1(b) gives curvature center position of raceways and the jth ball. O_j, O_j' are initial position and loading position of the jth ball. O_i, O_i' are initial position and position after loading of curvature center of inner raceway. O_e is position of curvature center of outer raceway. δ_a is axial preload displacement of bearing.

Contact deformation and contact angles between inner raceway, outer raceway and the jth ball can be solved from the geometric relationship in Figure 1:

$$\delta_{ij} = \sqrt{[A\sin\alpha + \delta_a]^2 + [A\cos\alpha + (X - x_j)\sin\theta_j + (Y - y_j)\cos\theta_j]^2} - A - h_{ij} \tag{5}$$

$$\delta_{ej} = \sqrt{[B\sin\alpha]^2 + [B\cos\alpha + x_j\sin\theta_j + y_j\cos\theta_j]^2} - B - h_{ej} \tag{6}$$

$$\alpha_{ij} = \arcsin\left(\frac{A\sin\alpha + \delta_a}{A + \delta_{ij}}\right) \tag{7}$$

$$\alpha_{ej} = \arcsin\left(\frac{B\sin\alpha}{B + \delta_{ej}}\right) \tag{8}$$

Where, $A = (f_i - 0.5)D_w$, $B = (f_e - 0.5)D_w$. D_w is diameter of the ball. α is initial contact angle. f_i, f_e are curvature coefficient of inner and outer ring. h_{ij}, h_{ej} are oil film thickness between inner ring, outer ring and the jth ball (Harris, 2007).

Then, contact loads Q_{ij}, Q_{ej} between inner ring, outer ring and the jth ball can be calculated. (Harris, 2007). The friction coefficients between balls and raceways are determined by an experimentally verified five-parameter lubricant rheology model (Wang, *et al.*, 2004):

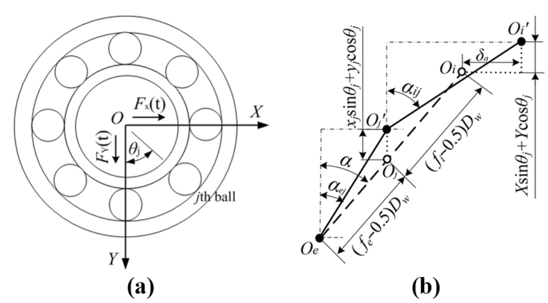
$$\mu = (A + Bs)e^{-Cs} + D \tag{9}$$

Where, s is slide-to-roll ratio. Coefficients A, B, C, D are related to lubricant parameters, which can be determined by reference (Wang, *et al.*, 2004).

Then, frictions between balls and inner ring, outer ring f_{ij}, f_{ej} can be obtained by contact loads and the friction coefficients. Figure 2 gives the forces on the jth ball.

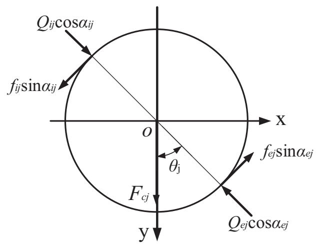
Vibration equations of the rigid rotor-bearing system are given as:

Figure 1 Bearing geometric relationship



Notes: (a) Ball bearing under radial load; (b) position of raceways and the jth ball

Figure 2 Forces of the *j*th ball



$$M\ddot{X}+C\dot{X}+2\sum_{j=1}^{zb}(Q_{ij}\cos\alpha_{ij}\sin\theta_j-f_{ij}\sin\alpha_{ij}\cos\theta_j)=F_X(t) \tag{10}$$

$$M\ddot{Y}+C\dot{Y}+2\sum_{j=1}^{zb}(Q_{ij}\cos\alpha_{ij}\cos\theta_j+f_{ij}\sin\alpha_{ij}\sin\theta_j)=F_Y(t) \tag{11}$$

$$\begin{aligned} &m_b\ddot{x}_j+c_b\dot{x}_j+Q_{ej}\cos\alpha_{ej}\sin\theta_j-f_{ej}\sin\alpha_{ej}\cos\theta_j \\ &-Q_{ij}\cos\alpha_{ij}\sin\theta_j+f_{ij}\sin\alpha_{ij}\cos\theta_j=0 \end{aligned} \tag{12}$$

$$\begin{aligned} &m_b\ddot{y}_j+c_b\dot{y}_j+Q_{ej}\cos\alpha_{ej}\cos\theta_j+f_{ej}\sin\alpha_{ej}\sin\theta_j \\ &-Q_{ij}\cos\alpha_{ij}\cos\theta_j-f_{ij}\sin\alpha_{ij}\sin\theta_j=F_{cj} \end{aligned} \tag{13}$$

Where, M,C are mass and damping of rotor system. m_b,c_b are mass and damping of a ball. $m{\cdot}e$ is product of unbalance mass and eccentric. F_{cj} is centrifugal force of the jth ball. ω is rotating speed. $\boldsymbol{z}_b$ is number of balls.

$$F_X(t)=(m\cdot e)\omega^2\sin\omega t$$

$$F_Y(t)=F_r+Mg+(m\cdot e)\omega^2\cos\omega t$$

$$\theta_j=\frac{2\pi}{z_b}(j-1)+\frac{1}{2}\omega\left(1-\frac{D_w}{D_{pw}}\cos\alpha\right)\times t \tag{14}$$

Where, D_{pw} is pitch diameter of the bearing.

The nonlinear vibration equations are solved by Runge–Kutta method. To ensure safe operation, working temperature of the bearing cannot exceed allowable temperature value. Local friction in bearing can be calculated based on the contact status results. Then, total power consumption can be given by the sum of the local heat sources in the bearing. Then, temperature distribution of the bearings can be calculated (Cui and Zhang, 2019).

Two periodic excitations, unbalance force and nonlinear contact force in the bearing. To solve periodic solutions, interval of time period kT_ω is defined as (Choi and Noah, 1992):

$$\left|\frac{kT_\omega}{T_{VC}}(\mathrm{mod}\,1)-1\right|<\varepsilon \tag{15}$$

where, T_ω is rotor rotation cycle, T_{VC} is cycle of ball bearing varying compliance. $k=1,2,3,\ldots$

$$T_{VC}=\frac{4\pi z_b}{\omega\left(1-\frac{D_w}{D_{pw}}\cos\alpha\right)} \tag{16}$$

To solve bifurcation behaviors and stability of the system, Floquet multipliers are used.

3. Fatigue rating life models of ball bearings under vibration load

3.1 Modified reference rating life model under vibration load

Under variable loading condition, contact loads of bearing vary with the variable loading. In order to calculating modified reference rating life under the variable load, a modified reference rating life model under vibration load is proposed based on ISO/TS 16281 (2008).

According to ISO/TS 16281 (2008), the equivalent value of the contact loads in one cycle should be calculated. Harris gives the equivalent contact loads based on Miner theory (Harris, 2007):

$$Q_{ij_{equ}}=\left(\frac{1}{T}\int_0^T Q_{ij}{}^3\,dt\right)^{1/3} \tag{17}$$

$$Q_{ej_{equ}}=\left(\frac{1}{T}\int_0^T Q_{ej}{}^3\,dt\right)^{1/3} \tag{18}$$

Where, $Q_{ij_{equ}}$,$Q_{ej_{equ}}$ are the equivalent contact loads in one cycle. $T=kT_\omega$.

The dynamic equivalent rolling element load for inner ring, outer ring can be expressed as:

$$Q_{ei_{equ}}=\left(\frac{1}{z_b}\sum_{j=1}^{z_b}Q_{ij_{equ}}{}^3\right)^{1/3} \tag{19}$$

$$Q_{ee_{equ}}=\left(\frac{1}{z_b}\sum_{j=1}^{z_b}Q_{ej_{equ}}{}^{10/3}\right)^{3/10} \tag{20}$$

The basic dynamic load ratings for inner rings and outer rings Q_{ci}, Q_{ce} are obtained by ISO/TS 16281 (2008). The modified reference rating life model under vibration load can be written as:

$$L_{nmr}=a_1a_{iso}\left[\left(\frac{Q_{ci}}{Q_{ei_{equ}}}\right)^{-10/3}+\left(\frac{Q_{ce}}{Q_{ee_{equ}}}\right)^{-10/3}\right]^{-9/10} \tag{21}$$

Similarly, the modified rating life can also be obtained based on the ISO/281 and equivalent load solution method.

Table 1 Bearing parameters

Parameters	Value
Inner diameter	50(mm)
Outer diameter	80(mm)
Width	16(mm)
Ball diameter	9.525(mm)
Ball number	18

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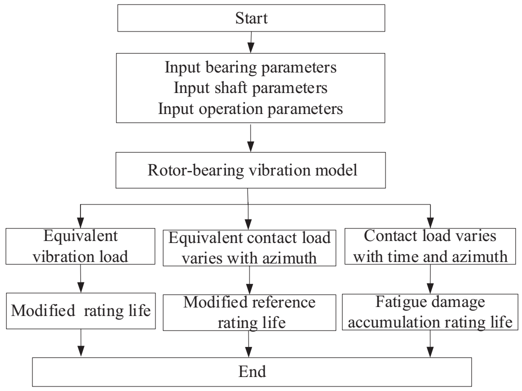
Figure 3 Calculation flow chart



3.2 Fatigue damage accumulation rating life model

To consider fatigue damage accumulation effect in one variation cycle, the fatigue damage accumulation rating life model is proposed.

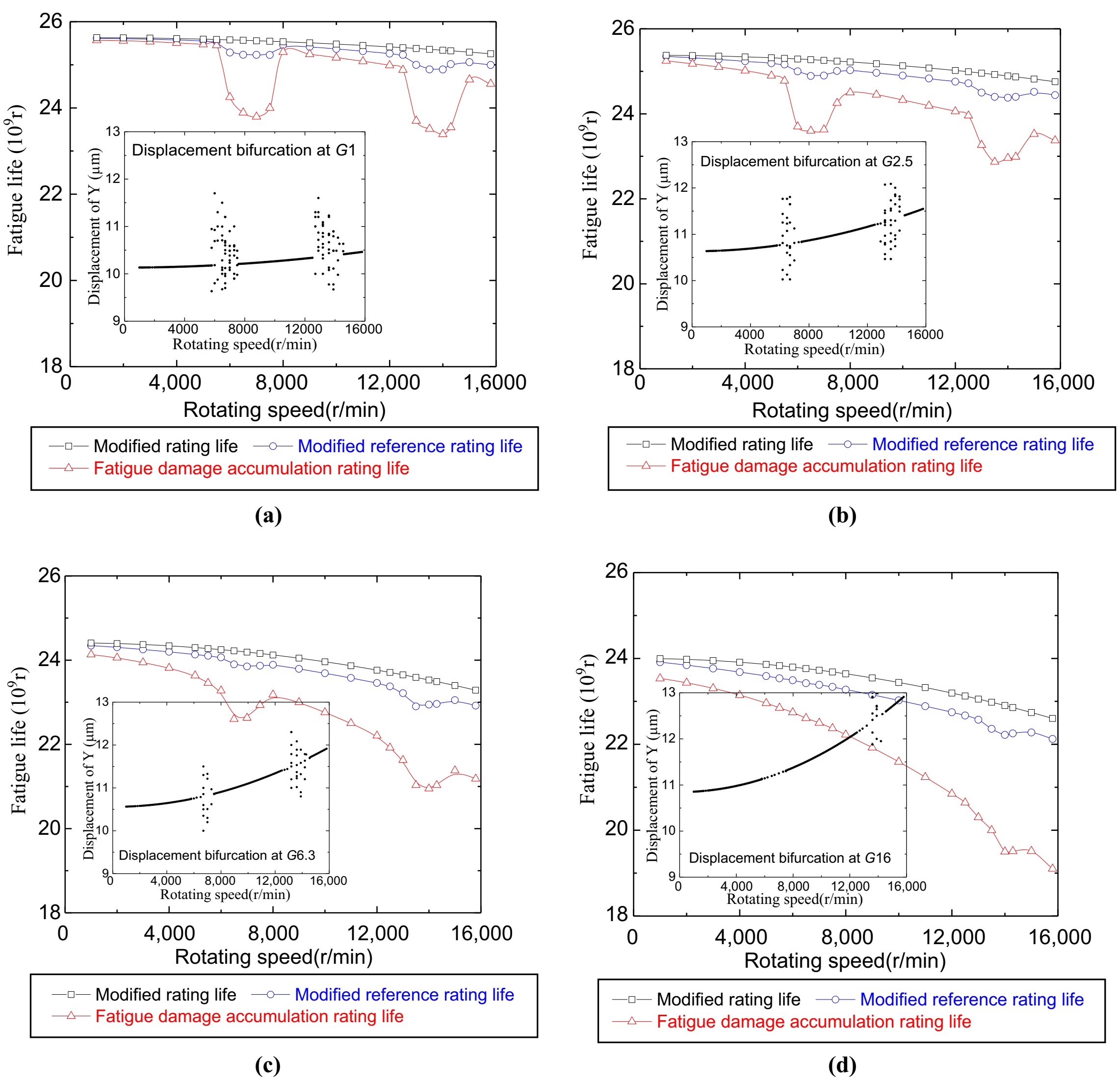
According to fatigue cumulative damage theory (Kwofie and Rahbar, 2012), equation (22) should be met when fatigue damage occurs at the contact point between inner ring and the *j*th ball:

$$\frac{L_{ij}}{T}\int_0^T \frac{1}{N_{t_{ij}}}\left(\frac{\ln(N_{t_{ij}})}{\ln(N_{0_{ij}})}\right)dt = 1 \tag{22}$$

Where, $N_{t_{ij}} = (Q_{cij}/Q_{ij})^3 \times 10^6 \times 60/(T \times \omega)$, $N_{0_{ij}} = (Q_{cij}/Q_{ij0})^3 \times 10^6 \times 60/(T \times \omega)$. Q_{ij0} is contact load between inner ring and the *j*th ball at the initial time in a cycle.

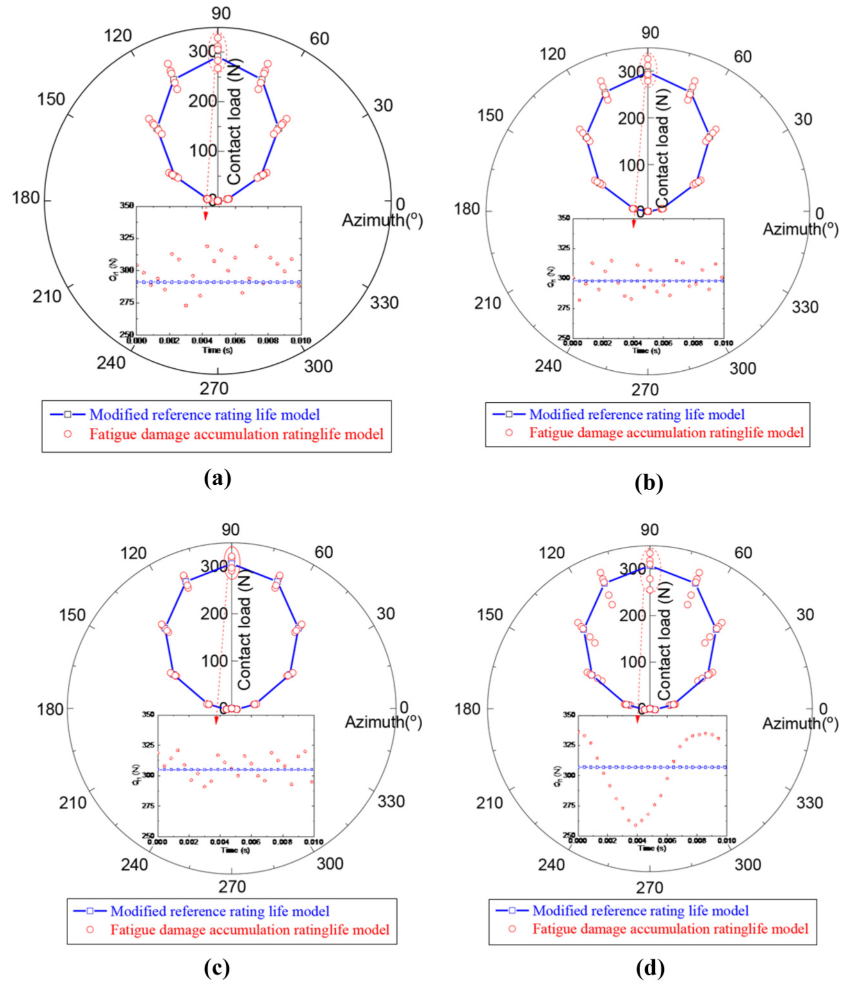
Fatigue life of inner ring at the contact point between inner ring and the *j*th ball can be derived as:

Figure 4 Fatigue life and bifurcation of bearing at different dynamic balance level



Notes: (a) Fatigue life varies with speed at *G*1; (b) fatigue life varies with speed at *G*2.5; (c) fatigue life varies with speed at *G*6.3; (d) fatigue life varies with speed at *G*16

Figure 5 Contrast of bearing contact loads at 7000r/min



Notes: (a) Contact loads at $G1$; (b) contact loads at $G2.5$; (c) contact loads at $G6.3$; (d) contact loads at $G16$

$$L_{ij} = \frac{T}{\int_0^T \left[\left(\frac{Q_{ij}}{Q_{cij}}\right)^3 \frac{ln\left(\left(Q_{cij}/Q_{ij}\right)^3 \times 10^6 \times 60/(T \times \omega)\right)}{ln\left(\left(Q_{cij}/Q_{ij0}\right)^3 \times 10^6 \times 60/(T \times \omega)\right)} \right] dt} \tag{23}$$

Where, Q_{cij} is the basic dynamic load ratings between the jth ball and inner ring.

The fatigue life of the jth ball at the contact point between the jth ball and inner ring under variable vibration load can be given as:

$$L_{bij} = \frac{T}{\int_0^T \left[\frac{D_{pw}\left(1-\gamma_{ij}{}^2\right)}{2D_w} \left(\frac{Q_{ij}}{Q_{bij}}\right)^3 \frac{\ln\left(\left(Q_{bij}/Q_{ij}\right)^3 \times 10^6 \times 60/\left(\frac{T\omega D_{pw}\left(1-\gamma_{ij}{}^2\right)}{2D_w}\right)\right)}{\ln\left(\left(Q_{bij}/Q_{ij0}\right)^3 \times 10^6 \times 60/\left(\frac{T\omega D_{pw}\left(1-\gamma_{ij}{}^2\right)}{2D_w}\right)\right)} \right] dt} \tag{24}$$

Similarly, fatigue life of outer ring at contact point between outer ring and the jth ball L_{ej}, fatigue life of the jth ball at contact point between the jth ball and outer ring L_{bej} can also be derived. The fatigue damage accumulation rating life of ball bearings under vibration load can be expressed as

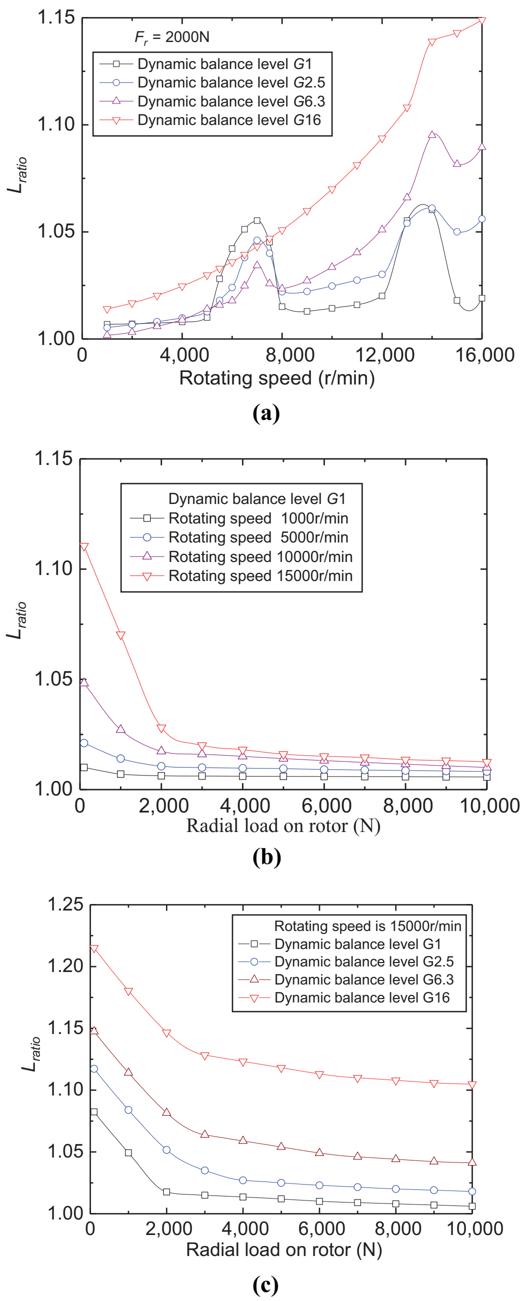
$$L_{nfr} = a_1 a_{iso} \left[\frac{1}{z_b} \left(\sum_{j=1}^{z_b} L_{ij}{}^{-\frac{10}{9}} + \sum_{j=1}^{z_b} L_{ej}{}^{-\frac{10}{9}} \right) + \sum_{j=1}^{z_b} L_{bij}{}^{-\frac{10}{9}} + \sum_{j=1}^{z_b} L_{bej}{}^{-\frac{10}{9}} \right]^{-\frac{9}{10}} \tag{25}$$

4. Results and discussions

Taking a rigid rotor supported by a pair of spindle ball bearing in electric machine as an example, the bearings parameters are shown in Table 1.

Initial contact angle of the bearing is designed as 15°. Material of rings and balls is GCr15 [GB/T18254, 2016], which is corresponding to the material Acronym of American Iron and Steel Institute/Society of Automotive Engineers 52100 and DIN 100Cr6. (Li, *et al.*, 2016). Assumed the bearing is lubricated by grease Svenska Kullager-Fabriken lubrication grease low temperature 2, which can be used in high speed condition. The maximum allowable temperature of the bearing is 80°C.

Figure 6 Fatigue life ratio vary with operating parameters



Notes: (a) Vary with rotating speed; (b) vary with radial load and rotating speed; (c) vary with radial load and dynamic balance level

Bearing fatigue rating life programs is developed by MATLAB software. Figure 3 gives flow chart for calculation of bearing fatigue rating life. Firstly, equivalent vibration loads acting on bearing are calculated, then modified rating life can be obtained. Secondly, variable contact loads in bearing considering vibration are solved by vibration model of rotor-bearing system. Then, equivalent contact loads are calculated, and modified reference rating life is obtained. Finally, fatigue damage accumulation rating life is solved by using time-vary contact loads.

According to the rotor balance standard (ISO/1940, 2003), dynamic balance level is defined as

$$G = \frac{\omega \times (m \cdot e)}{10^{-3} \times M} \tag{26}$$

Dynamic balance levels $G1$, $G2.5$, $G6.3$ and $G16$ are considered. Assuming M is 20Kg, radial load acting on the rotor is 2000 N. Modified rating life, modified reference rating life and fatigue damage accumulation rating life of the bearing are calculated and compared, and the influence factors on bearing's fatigue life are analyzed.

4.1 Influence of rotating speed on fatigue life

Figure 4 gives influence of different dynamic balance level on bearing fatigue life. Fatigue life of the three models both decrease when rotating speed increases.

As the dynamic balance level decreases, bearing's fatigue life decreases obviously and the difference of bearing fatigue life between the three models is gradually increasing. The displacement bifurcation figures demonstrate that low dynamic balance level is beneficial to weaken nonlinear characteristic of bearing system. In the non-periodic vibration range, bearing's fatigue life reduces significantly for fatigue damage accumulation rating life model, while it is not obvious for modified reference rating life model due to equivalent calculation method of contact loads. However, the modified rating life model cannot reflect the reduction of bearing fatigue life due to non-periodic vibration. Furthermore, the difference between modified rating life and modified reference rating life increases as the speed increases and the dynamic balance level decreases.

Figure 5 shows contrast of equivalent contact loads and variable contact loads at 7000r/min. Equivalent contact loads in modified reference life model increase from $G1$ to $G16$ and keep constant with time. Variable contact loads in fatigue damage life model show non-periodic motion characteristics and the variation amplitude of the contact loads decrease from $G1$ to $G6.3$, then enter into periodic motion at $G16$.

It can be concluded that modified rating life model is not suitable for the variable load condition. To illustrate the applicability under vibration load condition for the modified reference rating life model and fatigue damage accumulation rating life model, bearing's fatigue life ratio is defined as:

$$L_{ratio} = L_{nmr}/L_{nfr} \tag{27}$$

Figure 6 gives fatigue life ratio vary with operating parameters. It can be seen from Figure 5(a) that fatigue life ratio increase with rotating speed and decrease with radial load. L_{ratio} may increase greatly when the system enters into non-periodic motion. Figure 5(b) and 5(c) shows that fatigue life ratios vary with radial load, rotating speed and dynamic balance level.

Under vibration load condition, fatigue damage accumulation rating life will be significantly shorter than modified reference rating life in high speed and light load conditions, especially in low dynamic balance level condition.

4.2 Influence of preload on bearing fatigue life

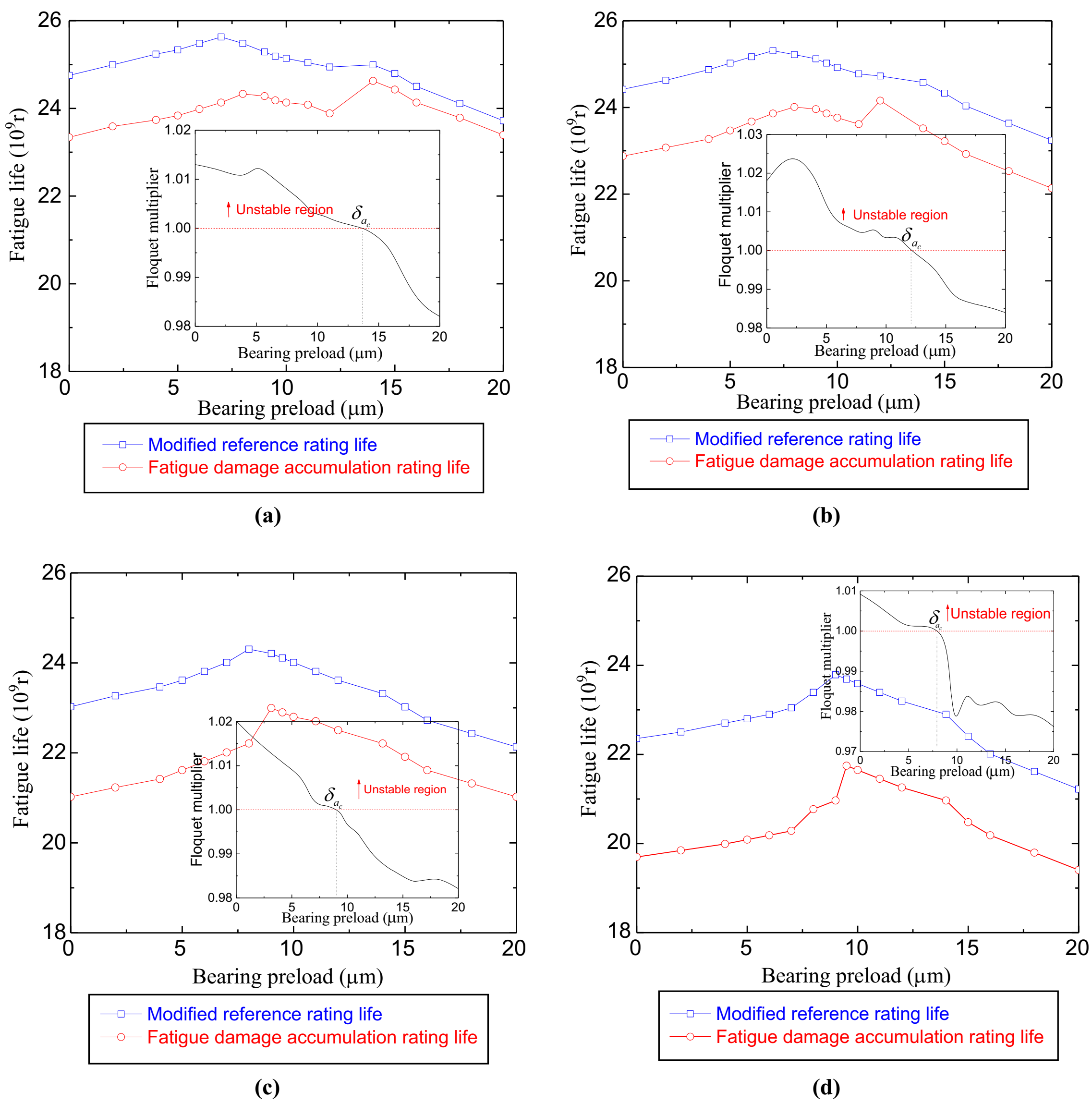
Preload can extend bearing's fatigue life. There exists an optimum bearing preload corresponding to the maximum bearing's fatigue life. The optimization objective is to maximize the modified reference rating life:

$$\max[f(\delta_{a_m})] = \max(L_{nmr}) \tag{28}$$

$$\max\left[f\left(\delta_{a_f}\right)\right] = \max(L_{nfr}) \tag{29}$$

Where, $f(\delta_{a_m})$, $f\left(\delta_{a_f}\right)$ represent the objective function, $0 \leq \delta_{a_m} \leq 20\ \mu$m, $0 \leq \delta_{a_f} \leq 20\ \mu$m. $\delta_{a_{mopt}}$, $\delta_{a_{fopt}}$ are defined as the optimum preloads corresponding to the maximum modified reference rating life and the maximum fatigue damage accumulation rating life. By increasing δ_{a_m} from 0 to 20 μm, $\delta_{a_{mopt}}$ can be found when the maximum of modified reference rating life occurs. Similarly, the optimum preload $\delta_{a_{fopt}}$ can also be obtained. In addition, there also exists a critical bearing

Figure 7 Fatigue life under preloads at 14000r/min



Notes: (a) Dynamic balance Level $G1$; (b) dynamic balance Level $G2.5$; (c) dynamic balance Level $G6.3$; (d) dynamic balance Level $G16$

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preload δ_{a_c} corresponding to critical motion status from instability to stability (Cui and Zheng, 2014).

Assuming rotating speed remains 14000r/min, Figure 7 gives fatigue life under different preloads. Floquet multipliers are calculated to judge the stability of periodic motion.

Figure 7 shows that fatigue life increases first and then decreases with increase of preloads. Fatigue damage accumulation rating life has an obvious increase when the bearing changes from non-periodic to periodic motion, while it is not obvious for the modified reference rating life. Table 2 gives the optimal preloads and critical preloads at 14000r/min.

Table 2 Preloads at 14000r/min

Dynamic balance level	$\delta_{a_{mopt}}$ (μm)	$\delta_{a_{fopt}}$ (μm)	δ_{a_c} (μm)
*G*1	7	14	14
*G*2.5	7	12	12
*G*6.3	8	9	9
*G*16	9	9.5	8

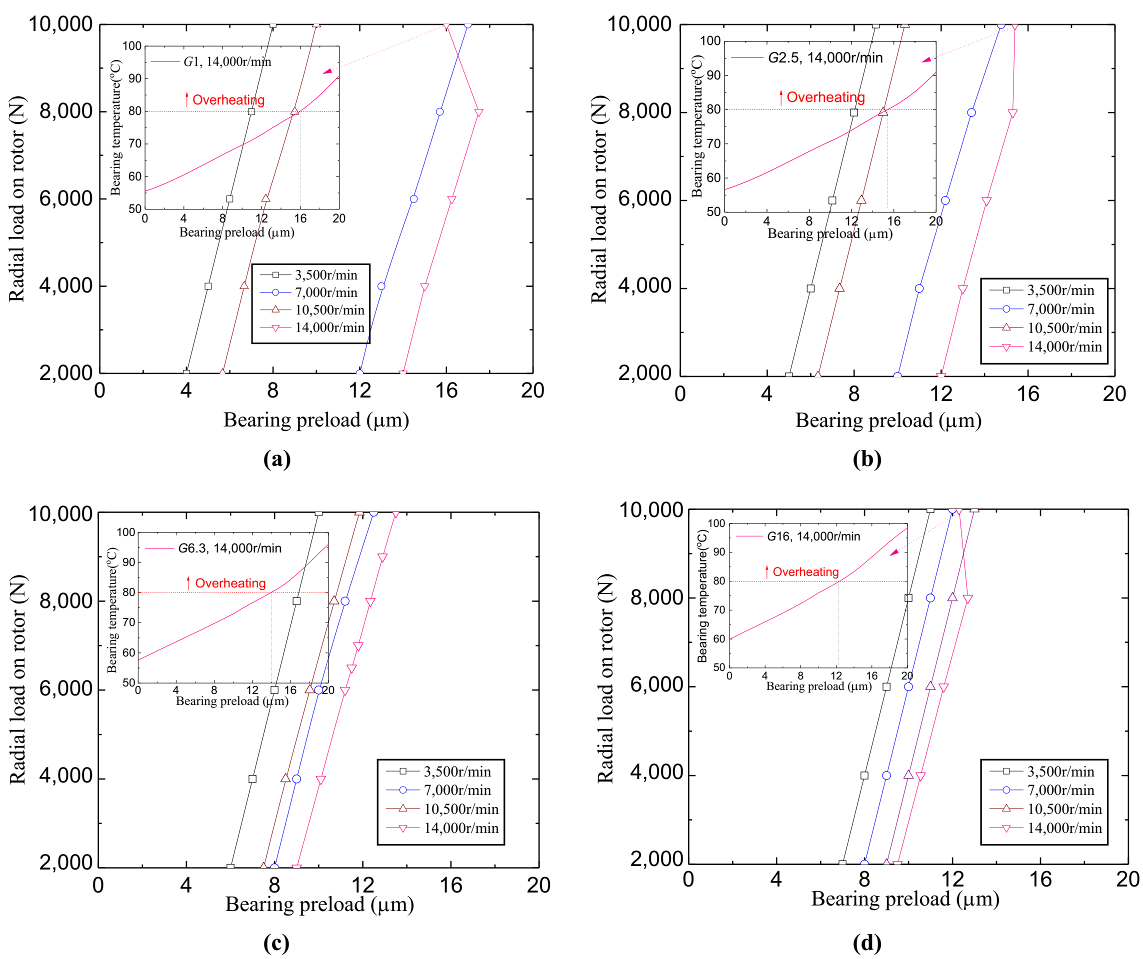
Comparing with modified reference rating life model, bearing optimum preloads should be larger when consider time-varying vibration load in fatigue damage accumulation rating life model. The optimum preload has an obvious influence on bearing's fatigue life. To extend bearing fatigue life, the optimum bearing preload should be considered.

4.3 Influence of speed and radial load on bearing optimal preload

Figure 8 gives effects of speed and radial load on optimum preload $\delta_{a_{fopt}}$. Temperature of the bearing increases as increase of the preload and decrease of dynamic balance level.

It can be seen that bearing optimum preloads increase as the radial load increases, however, the optimum preloads decrease with increase of the radial load when bearing temperature exceeds 80°C. The optimum preloads increase as the speed increases and dynamic balance level decreases when bearing is working in periodic motion status. In the

Figure 8 Influence of speed and radial load on bearing optimum preloads



Notes: (a) Dynamic balance Level *G*1; (b) dynamic balance Level *G*2.5; (c) dynamic balance Level *G*6.3; (d) dynamic balance Level *G*16

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non-periodic motion region, the optimum preload may become very large due to nonlinear characteristics. In order to obtain accurate optimum preload, nonlinear vibration and temperature rising should be considered.

Figure 9 High-speed ball bearings test bench

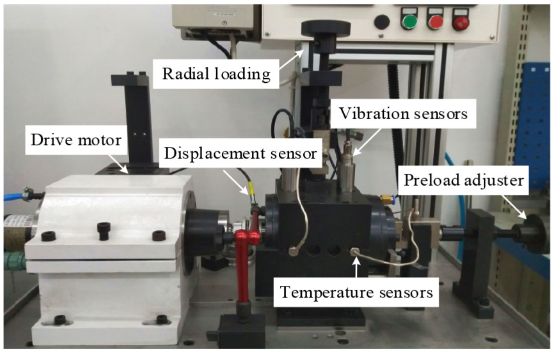
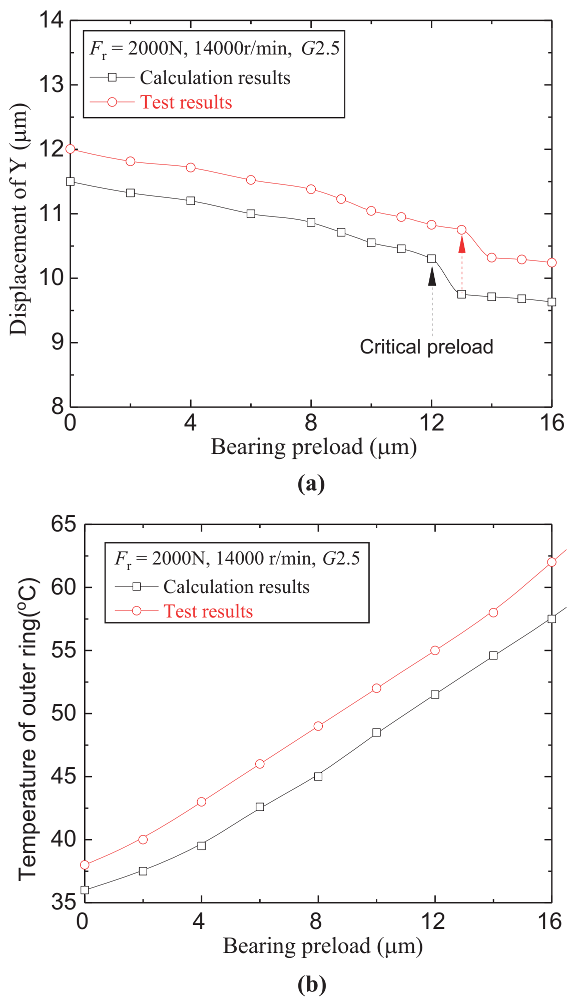


Figure 10 Contrast of test results and calculation results



Notes: (a) Contrast of bearing displacement; (b) contrast of bearing temperature

5. Experimental verification

Figure 9 illustrates test bench of high-speed ball bearings rotor system, the structure is simulated from the high speed electrical machine. The main shaft is driven by a high-speed drive motor and the maximum rotating speed is 24000r/min. Two spindle ball bearings shown in Table1 are used, initial contact angle is 15°, steel balls are used. Radial load and preload can be applied to the test bearings. Bearing temperature is measured by thermos-couple sensors. Displacement of the bearing system is measured by a non-contact eddy current sensor. Dynamic balance level of the rotor-bearing system is $G2.5$, which is measured by an on-site dynamic balance testing instrument.

Radial load on the system is set as 2000 N, bearing preload increases from 0 N to 16 μm. Rotating speed is set as 14000r/min. Displacement of the bearing system are measured. Average of the maximum value in 10 cycles is recorded as one test data.

Figure 10 gives contrast between experimental results and theoretical results from the model in this paper.

Figure 10(a) shows that displacement of the bearing system decreases gradually with increase of preload. The critical preload of the test bearing at 14000r/min is 13 μm, which is close to theoretical result 12 μm. The maximum difference between calculation results and test results is 9.5%. Figure 10(b) demonstrates that bearing temperature increases with preload. The maximum difference is 9.2%. Figure (10) demonstrates that the calculation results are close to the experimental results.

6. Conclusions

- Bearings in electric machines often work in high speed, light load and vibration load conditions. A new fatigue damage accumulation rating life model is proposed. Considering vibration load due to dynamic unbalance, fatigue damage accumulation rating life is significantly shorter than modified reference rating life in high speed and light load conditions, especially in low dynamic balance level condition. Fatigue damage accumulation rating life reduces significantly when bearing enters from periodic motion into non-periodic motion.
- The optimum preload for ball bearing is defined. When considering time-varying vibration load in fatigue damage accumulation rating life model, the optimum preload of bearing is larger than that without considering time-varying vibration load. The optimum preload has an obvious influence on fatigue life. To obtain accurate optimum preload, nonlinear vibration and temperature rising of bearings should be considered.
- To calculate fatigue rating life of ball bearings more accurately under vibration load, especially in high speed and light load conditions, the fatigue damage accumulation rating life model should be considered.

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