

Article

Research on Load Distribution and Fatigue Life Under Elliptical Deformation of Four-Point Contact Slewing Bearing Rings for Excavators

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Abstract

Excavators are critical equipment in mining, construction, and other fields. The four-point contact slewing bearings used in their slewing mechanisms operate under harsh conditions such as heavy loads and impacts. Furthermore, the bearing rings are prone to elliptical deformation after installation, making them susceptible to premature failure. To address this issue, this paper establishes a mechanical bearing model to investigate the load distribution among balls and the fatigue life of the bearing under elliptical deformation of the rings. It systematically analyzes the influence of key design parameters. The research finds that elliptical deformation of the rings leads to contact angle deviation and a reduction in load-bearing balls, resulting in severe degradation of bearing fatigue life; therefore, its occurrence must be strictly controlled. Designing with a groove curvature radius coefficient within the range of 0.51 to 0.52 achieves an optimal balance between fatigue life and the four-point contact geometry of the balls. There exists an “optimal clearance” that maximizes bearing fatigue life; when considering significant elliptical deformation, the clearance design should be appropriately increased. Increasing the design contact angle enhances load capacity and helps mitigate the effects of elliptical deformation. However, an excessively large contact angle can cause ellipse truncation in the raceway contact zone; thus, the contact angle should be designed based on practical conditions. Increasing the number of balls can improve the influence of ovality on load distribution and enhance the bearing’s fatigue life. This study provides a theoretical reference for the design of high-reliability slewing bearings for excavators.

Keywords: four-point contact slewing bearing; elliptical deformation; load distribution; bearing fatigue life



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1. Introduction

Excavators are heavy-duty engineering machinery widely used in mining, construction, road building, and other fields. Their slewing mechanism is a critical component that enables operational flexibility and high efficiency. The four-point contact slewing bearing, as the core component of an excavator, directly influences the machine’s overall load-bearing performance and reliability. The bearing rings are susceptible to elliptical deformation caused by installation errors, which alters the load distribution among the balls and can lead to premature failure. Therefore, investigating the load distribution and fatigue life of the balls under elliptical deformation, as well as the coupled effects of bearing

structural parameters, is of great significance for optimizing the design of excavator slewing bearings and enhancing their fatigue life.

Regarding the influence of design parameters on the load distribution and load-carrying capacity of four-point contact slewing bearings, numerous scholars have conducted research. He [1], addressing the demand for high load-carrying capacity in slewing bearings, proposed a collaborative design method for ball diameter and quantity based on dual-objective optimization of static load capacity and fatigue life under fixed dimensional constraints. This method enhances the load-carrying efficiency of the bearing system. Wang [2] established a static load model for slewing bearings, solved for the contact force distribution characteristics under axial, radial, and combined loads, and investigated the influence of contact angle and groove curvature radius coefficient on raceway contact load and the overall bearing load capacity. Li [3], with the optimization objective of enhancing the load-carrying capacity of double-row four-point contact ball slewing bearings, discovered through research that minor perturbations in design parameters such as axial clearance, groove curvature radius coefficient, and contact angle can alter the distribution characteristics of the dynamic load-carrying capacity. Kania [4] investigated the accelerated raceway edge wear problem in four-point angular contact bearings caused by significant contact angle variations, revealed the issue of abnormal contact angle increase, and proposed a geometric correction method for the raceway profile. Tu [5], focusing on the pitch bearing—a double-row four-point contact ball slewing bearing in wind turbine systems—established a multi-load coupled mechanical model based on Hertzian elastic contact theory. Their study indicated that an appropriate negative clearance can effectively enhance the bearing's load capacity.

Chen [6] established a nonlinear static load model for slewing bearings and analyzed the dynamic influence mechanism of geometric parameters on the bearing's contact load distribution and actual contact angle under different operating conditions. Liu [7], aiming to solve the load distribution problem of single-row four-point contact slewing bearings, constructed a simplified finite element model based on nonlinear springs on the Abaqus platform. Furthermore, they designed a static load experiment using circumferential strain gauges on the inner ring to obtain load distribution data, thereby validating the established model. Aguirrebeitia et al. [8] considering operating conditions involving axial load and overturning moment, analyzed the ball load distribution and provided a calculation method for the bearing's limiting load.

Regarding the fatigue life calculation of four-point contact slewing bearings, numerous scholars have also conducted extensive research. Zhu [9], targeting the high-reliability design requirements of large four-point contact ball bearings, established a collaborative optimization model for raceway spacing and curvature radius coefficient based on finite element simulation, and analyzed the computational results of the stress-life model. Guo et al. [10] developed a quasi-static life prediction framework integrating Hertzian contact theory and multi-body contact mechanics. They further analyzed the dynamic evolution of the ball-raceway contact angle, the distribution of the deformation field, and the time-varying patterns of contact stress. Gao [11] analyzed the coupled influence mechanism of raceway geometric parameters on the load capacity and life of four-point contact slewing bearings. The research indicated that enhancing static load capacity requires increasing the curvature ratio and ball diameter, while extending fatigue life necessitates optimizing the sectional clearance and initial contact angle. Wang et al. [12] employed a quasi-static method to analyze the contact mechanics state of high-speed, heavy-duty angular contact ball bearings. They discovered the regulatory effects of radial load and eccentric load angle on the bearing contact stress distribution and fatigue life. Gao [11], focusing on the coupled influence of raceway geometric parameters on the load capacity and life of

four-point contact slewing bearings, revealed the variation patterns of maximum contact load and maximum stress, and proposed a parameter optimization method. R. Potočník et al. [13] first calculated the maximum contact force in slewing bearings using the analytical expressions of Hertzian contact theory, and then calculated the fatigue life using the strain-life method based on surface stresses obtained from finite element analysis.

Regarding the influence of ring deformation in four-point contact slewing bearings, some scholars have conducted research, yet the related investigations remain insufficiently in-depth. Zupan [14] constructed a set of five-element nonlinear static equilibrium equations for slewing bearings and studied the coupling mechanism by which key bearing parameters—conformity and clearance—significantly affect the load transmission path and the deformation coordination relationship between rings by regulating the actual contact angle. Scholars such as Amasorrain [15], Smolnicki [16], and Olave [17] revealed the contact load distribution characteristics of slewing bearings through mathematical models and the finite element method: the internal load distribution deviates from the traditional cosine curve theory, exhibiting a non-uniform “hard-spot effect” phenomenon accompanied by significant deformation of the bearing rings. Zhao [18], addressing the load capacity issue caused by stress concentration in the raceway of traditional single-row ball slewing bearings under large overturning moments, proposed a non-circular raceway optimization design method driven by the dual objectives of static equilibrium equation constraints based on the vector method and maximizing fatigue life. It is important to note that the influence of elliptical deformation of the rings caused by installation errors on load distribution and load-carrying capacity has not been addressed in the existing literature. Furthermore, the underlying mechanism of how ring elliptical deformation affects bearing fatigue life still requires deeper investigation.

In response to the issues mentioned above, this paper takes the four-point contact slewing bearing for excavators as the research object. A mechanical model incorporating elliptical deformation of the outer ring is established to investigate the influence of the groove curvature radius coefficient, axial clearance, design contact angle, and number of balls on the load distribution and fatigue life of the four-point contact slewing bearing under the condition of outer ring elliptical deformation. It should be noted that the research methodology is the same for cases where both the inner and outer rings exhibit elliptical deformation and for cases where only the outer ring is deformed. To simplify the problem and highlight the core influences, this paper will focus solely on a detailed analysis and discussion of the model with elliptical deformation of the outer ring.

2. Establishment of a Mechanical Model for the Outer Ring Elliptical Deformation in Four-Point Contact Slewing Bearings for Excavators

2.1. Bearing Dimensions

Taking a specific type of four-point contact slewing bearing for excavators as an example, analysis and calculation are conducted. The main structural parameters of the bearing are listed in Table 1. In the table, the groove curvature radius coefficient is defined as the ratio of the radius of curvature of the raceway generatrix to the diameter of the ball.

Table 1. Dimensional parameters of excavator slewing bearing.

Parameters	Value
Inner diameter	1502 mm
Outer diameter	1855 mm
Inner and Outer Groove Curvature Radius Coefficient	0.52
Axial Clearance	0.2 mm

Table 1. *Cont.*

Parameters	Value
Design Contact Angle	45°
Ball Diameter	60 mm
Number of Balls	75

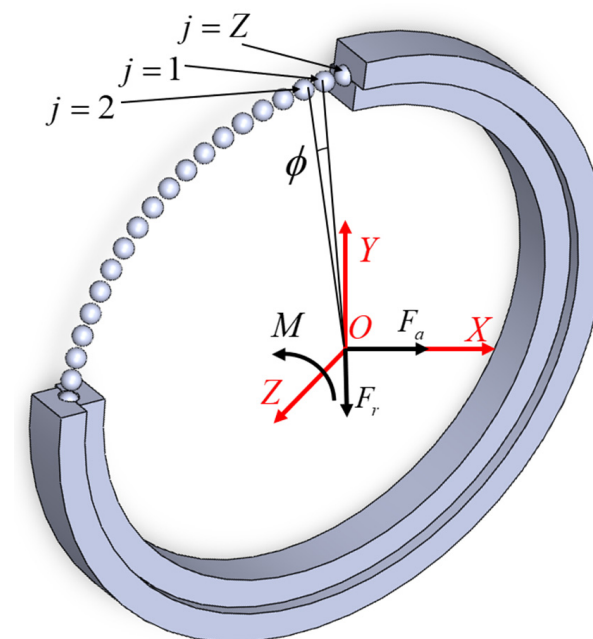
The four-point contact slewing bearing of this excavator model is subjected only to axial force and overturning moment during operation. Table 2 lists its commonly used operating condition parameters.

Table 2. Excavator slewing bearing operating condition parameters.

Operating Condition	Axial Force/kN	Radial Force/kN	Overturning Moment/kNm	Rotational Speed/(r/min)	Proportion
1	600	0	1000	12	20%
2	700	0	1600	12	60%
3	800	0	2200	12	20%

2.2. Structural Characteristics and Modeling Analysis of the Bearing

Taking a four-point contact slewing bearing for a specific excavator as an example, and based on the established bearing load capacity analysis model that accounts for the influence of outer ring elliptical deformation, this study focuses on analyzing the impact of bearing structural parameters—such as the groove curvature radius coefficient, axial clearance, and design contact angle—on the bearing's load capacity. To analyze the internal forces of the four-point contact slewing bearing under external loads, a global coordinate system with the bearing center O as the origin, as shown in Figure 1, was established. The external loads are defined as the axial force F_a , radial force F_r , and overturning moment M . In the figure, Z represents the number of balls, j denotes the index number of a specific ball, and the azimuth angle ϕ of a ball is given by $\phi = 2\pi(j - 1)/Z$.

**Figure 1.** Schematic of the bearing coordinate system and ball azimuth angles.

The physical structure of the four-point contact slewing bearing used in the excavator is shown in Figure 2. The bearing mainly consists of an outer ring, an inner ring, and balls. Its inner ring is connected to the reduction gearbox via internal teeth, while the outer ring directly bears the external loads. As seen in the bearing cross-sectional view in Figure 3, both the inner and outer raceways employ a typical “gothic arch groove” design. This design allows a single ball (ball) to simultaneously form four contact points with the inner and outer raceways. These four contact points are arranged in two opposing pairs, constituting two independent contact pair groups, labeled as Contact Pair Group 1 and Contact Pair Group 2, respectively.

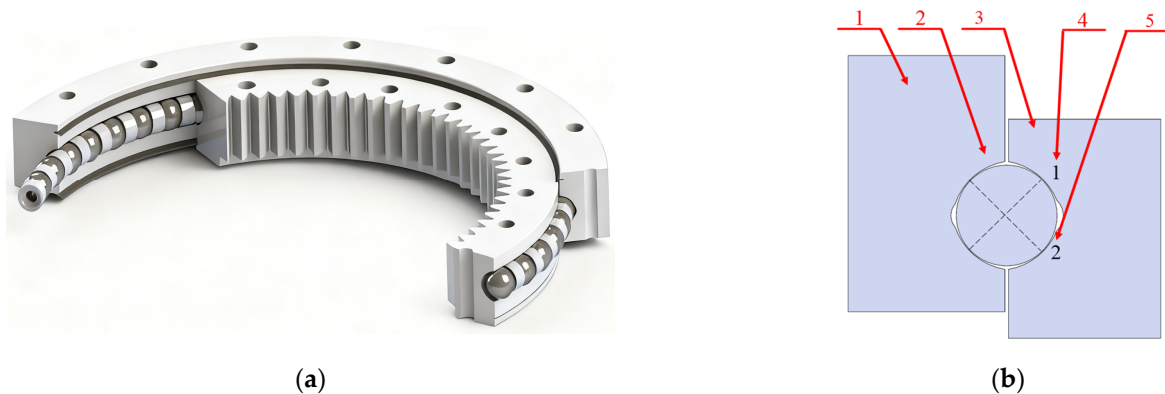


Figure 2. Structural schematic of the four-point contact slewing bearing: (a) physical photograph of the excavator slewing bearing; (b) cross-sectional view—1: outer ring; 2: ball; 3: inner ring; 4: Contact Pair Group 1; 5: Contact Pair Group 2.

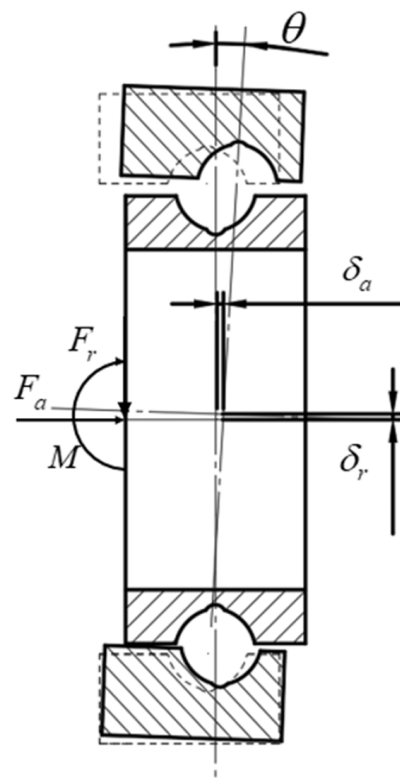


Figure 3. Schematic of outer ring displacement.

2.3. Load and Deformation Analysis of the Bearing

During operation of this specific type of excavator four-point contact slewing bearing, the outer ring rotates while the inner ring remains fixed. Under combined loading, the outer ring undergoes an axial displacement δ_a , a radial displacement δ_r , and an angular displacement θ , as shown in Figure 3.

Under combined loading, the outer ring of the four-point contact slewing bearing displaces, causing a change in the distance between the outer raceway curvature centers. The positions of the inner and outer raceway curvature centers, along with the variation in contact angles at azimuth angle ϕ before and after loading, are illustrated in Figure 4. C_{1i} and C_{2i} represent the positions of the inner raceway curvature centers before and after loading, respectively. Since the inner ring is fixed, the positions of its curvature centers do not change. C_{1e} and C_{2e} denote the positions of the outer raceway curvature centers before loading, while C'_{1e} and C'_{2e} denote their positions after loading. O and O' represent the geometric centers of the ball before and after loading, respectively. α_{01} and α_{02} denote the design contact angles of the bearing before loading, whereas $\alpha_{1\phi}$ and $\alpha_{2\phi}$ denote the working contact angles after loading.

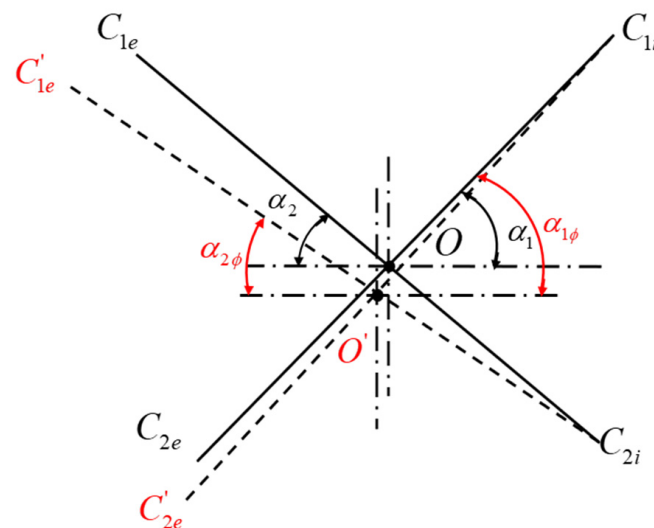


Figure 4. Schematic of the relationship between bearing curvature centers and the variation in contact angles at azimuth angle ϕ .

Before the bearing is loaded, the distance between the inner and outer raceway curvature centers for contact pair group t is denoted as A_t .

$$A_1 = r_i + r_e - D_w - \frac{1}{2}G_a \sin \alpha_{01} \quad (1)$$

$$A_2 = r_i + r_e - D_w - \frac{1}{2}G_a \sin \alpha_{02} \quad (2)$$

where r_i , r_e are the curvature radii of the inner and outer raceways, respectively; D_w is the ball diameter; G_a is the axial clearance of the bearing.

Under external loading, relative displacement occurs between the outer and inner rings, causing a change in the distance A_t (where $t = 1, 2$) between the raceway curvature centers of the two Contact Pair Groups in the bearing. As shown in Figure 5, after loading, the axial deformation at any azimuth angle ϕ caused by the axial displacement is δ_a , the radial deformation component caused by the radial displacement is $\delta_r \cos \phi$, and the axial

deformation caused by the angular tilt displacement is $R_e\theta\cos\phi$. Here, R_e is the radius of the locus of the outer raceway curvature center, and their respective expressions are

$$R_e = \frac{1}{2}d_m - (r_e - 0.5D_w) \cos \alpha_{01} + \frac{1}{4}G_a \sin \alpha_{01} \quad (3)$$

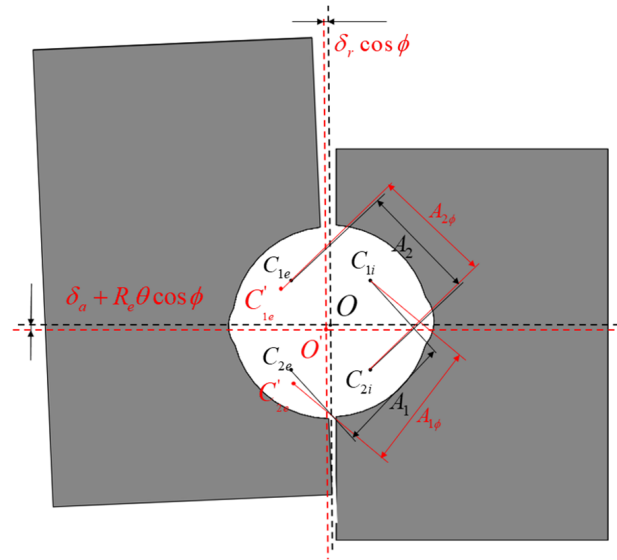


Figure 5. Variation in the distance between bearing curvature centers at azimuth angle ϕ .

After loading, the distance $A_{t\phi}$ between the raceway curvature centers of the Contact Pair Group t at the azimuth angle ϕ can be expressed as

$$A_{1\phi} = [(A_1 \sin \alpha_{01} + \delta_a + R_e \theta \cos \phi)^2 + (A_1 \cos \alpha_{01} + \delta_r \cos \phi)^2]^{1/2} \quad (4)$$

$$A_{2\phi} = [(A_2 \sin \alpha_{02} - \delta_a - R_e \theta \cos \phi)^2 + (A_2 \cos \alpha_{02} + \delta_r \cos \phi)^2]^{1/2} \quad (5)$$

The contact angle $\alpha_{t\phi}$ of Contact Pair Group t between the ball and the raceway at the azimuth angle ϕ also changes, and the altered contact angle is given by

$$\alpha_{1\phi} = \arcsin\left(\frac{A_1 \sin \alpha_{01} + \delta_a + R_e \theta \cos \phi}{A_{1\phi}}\right) \quad (6)$$

$$\alpha_{2\phi} = \arcsin\left(\frac{A_2 \sin \alpha_{02} - \delta_a - R_e \theta \cos \phi}{A_{2\phi}}\right) \quad (7)$$

2.4. Establishment of Bearing Force Equilibrium Equations

After loading, the total contact deformation between the ball and the inner and outer rings at any azimuth angle ϕ is equal to the difference between the distance $A_{t\phi}$ of the raceway curvature centers after loading and the initial center distance A_t before loading. That is

$$\delta_{t\phi} = A_{t\phi} - A_t (t = 1, 2) \quad (8)$$

according to the *Hertz* point contact theory, the relationship between the normal contact load $Q_{t\phi}$ and the contact deformation $\delta_{t\phi}$ for Contact Pair Group t at the azimuth angle ϕ is given by

$$Q_{t\phi} = \begin{cases} K_n \delta_{t\phi}^{1.5} (\delta_{t\phi} \geq 0) \\ 0 (\delta_{t\phi} \leq 0) \end{cases} \quad (9)$$

where K_n is the load–deflection coefficient for Contact Pair Group t .

Under the combined action of axial load, radial load, and overturning moment on the bearing outer ring, the equilibrium equations for the outer ring are

$$\sum_{\phi=0}^{2\pi} (Q_{1\phi} \sin \alpha_{1\phi} - Q_{2\phi} \sin \alpha_{2\phi}) - F_a = 0 \quad (10)$$

$$\sum_{\phi=0}^{2\pi} (Q_{1\phi} \cos \alpha_{1\phi} + Q_{2\phi} \cos \alpha_{2\phi}) \cos \phi - F_r = 0 \quad (11)$$

$$\frac{1}{2} d_m \sum_{\phi=0}^{2\pi} (Q_{1\phi} \sin \alpha_{1\phi} - Q_{2\phi} \sin \alpha_{2\phi}) \cos \phi - M = 0 \quad (12)$$

where $Q_{t\phi}$ represents the normal contact load of Contact Pair Group t ($t = 1, 2$); F_a , F_r , M denote the external axial force, radial force, and overturning moment applied to the bearing, respectively; $\alpha_{t\phi}$ denotes the contact angle of Contact Pair Group t ($t = 1, 2$); the subscript ϕ indicates the azimuth angle of the ball; d_m is the bearing pitch diameter.

2.5. Calculation Method for Bearing Fatigue Life

For this type of four-point contact slewing bearing, there are two rings (inner and outer), each containing two raceways. The calculation of its fatigue life must comprehensively consider the condition of all four raceways.

The dynamic load ratings for the inner and outer rings are [19]

$$Q_{ci(e)} = 98.1 \lambda \eta \left(\frac{2f_{i(e)}}{2f_{i(e)} - 1} \right)^{0.41} \frac{(1 \mp \gamma)^{1.39}}{(1 \pm \gamma)^{1/3}} \left(\frac{D_w}{d_m} \right)^{0.3} Z^{-1/3} \times 3.647 D_w^{1.4} \quad (13)$$

In the equations above, the upper sign applies to the inner ring, and the lower sign applies to the outer ring. f_i and f_e are the groove curvature radius coefficients for the inner and outer raceways of the ball, respectively. λ and η are correction factors for the ball bearing. In the equations, λ is taken as 0.95, and η is taken as 1. γ is the dynamic load rating reduction coefficient, expressed as

$$\gamma = D_w \cos \alpha_{i(e)} / d_m \quad (14)$$

In the formula, α_i is the contact angle between the ball and the inner raceway, and α_e is the contact angle between the ball and the outer raceway.

As the outer ring rotates relative to the applied load on the bearing, the equivalent dynamic load for a ball in the outer ring is

$$Q_{eet} = \left(\frac{1}{Z} \sum_{j=1}^Z Q_{t\phi}^3 \right)^{1/3} \quad (15)$$

As the inner ring is stationary relative to the applied load on the bearing, the equivalent dynamic load for a ball in the inner ring is

$$Q_{eit} = \left(\frac{1}{Z} \sum_{j=1}^Z Q_{t\phi}^{10/3} \right)^{3/10} \quad (16)$$

Consequently, the rated fatigue life for each raceway can be calculated as

$$L_m = \left[\left(\frac{Q_{ci(e)}}{Q_{ei(e)t}} \right) \right]^3 \quad (17)$$

In the equation, L_m (where $m = 1, 2, 3, 4$) denotes the rated fatigue life of the raceway corresponding to index m .

Consequently, the rated fatigue life of the complete bearing, L_b (in million revolutions), can be obtained as

$$\left(\frac{1}{L_b}\right)^{\frac{10}{9}} = \left(\frac{1}{L_1}\right)^{\frac{10}{9}} + \left(\frac{1}{L_2}\right)^{\frac{10}{9}} + \left(\frac{1}{L_3}\right)^{\frac{10}{9}} + \left(\frac{1}{L_4}\right)^{\frac{10}{9}} \quad (18)$$

In practical calculations, it is customary to express the bearing fatigue life in operating hours (unit: hour), denoted as L_h

$$L_h = \frac{10^6}{60n} L_b \quad (19)$$

where n denotes the rotational speed of the bearing.

2.6. Consideration of Outer Ring Elliptical Deformation

Considering that the outer raceway is elliptical, the degree of elliptical deformation of its curvature center trajectory can be expressed by the following formula:

$$\lambda = \frac{a}{b} - 1 \quad (20)$$

In the formula, λ denotes the degree of elliptical deformation, where a and b represent the lengths of the semi-major axis and semi-minor axis of the ellipse, respectively. Given that the bearing is large and the variation in the lengths of the two semi-axes is relatively small when elliptical deformation occurs, for the convenience of study, the length of the semi-minor axis b is taken to be equal to the radius R_e of the circular trajectory of the outer raceway's curvature center during calculations.

After elliptical deformation of the outer ring, the equation for the trajectory of the outer raceway's curvature center, denoted as R_ϕ , can be expressed as

$$R_\phi = R_e \sqrt{(\lambda^2 + 2\lambda) \cos^2(2\phi - 2n) + 1} \quad (21)$$

In the equation, R_e is the radius of the circular trajectory of the outer raceway's curvature center prior to elliptical deformation. n denotes the azimuthal orientation of the semi-major axis of the outer raceway's elliptical deformation. To simplify the model, the subsequent analysis assumes that the semi-major axis aligns with the direction of the overturning moment, $n = 0$.

Following elliptical deformation of the outer ring, the ball contact angles $\alpha_{01\phi}$ and $\alpha_{02\phi}$ are expressed as

$$\alpha_{01\phi} = \arctan\left(\frac{A_1 \sin \alpha_{01}}{A_1 \cos \alpha_{01} + R_\phi - R_e}\right) \quad (22)$$

$$\alpha_{02\phi} = \arctan\left(\frac{A_2 \sin \alpha_{02}}{A_2 \cos \alpha_{02} + R_\phi - R_e}\right) \quad (23)$$

In the equation, A_0 is the distance between the inner and outer raceway curvature centers prior to elliptical deformation. α_{01} and α_{02} are the design contact angles.

When elliptical deformation occurs in the outer ring, the trajectory of the outer raceway's curvature center changes from a circle with radius R_e to the elliptical form R_ϕ (Equation (21)). Consequently, the design contact angles for the balls transform from the original α_{01} and α_{02} to $\alpha_{01\phi}$ and $\alpha_{02\phi}$ (Equations (22) and (23)). Based on this foundation, a model is established to solve for the load distribution and fatigue life of the bearing under the condition of outer ring elliptical deformation.

2.7. Model Validation

To validate the reliability of the analytical model established in this study, computational results from the commercial software Romax 2023 were used for comparison. Romax software is widely applied in the fields of transmission and bearing analysis. Its bearing analysis module features mature modeling and calculation methods and has undergone substantial validation in simulations for industries such as wind power and engineering machinery. Since the Romax 2023 model does not account for elliptical deformation of the bearing outer ring, the comparative analysis is confined to the condition where the outer ring is free from elliptical deformation, ensuring a consistent comparison baseline. Using the structural parameters listed in Table 1 and the operating conditions described in Table 2, modeling and calculations were performed in both Romax 2023 and the model presented herein. The comparative results for the ball load distribution and calculated fatigue life under three operating conditions are shown in Figure 6. In the figure, Q_1 represents the contact load for contact pair group 1, and Q_2 represents the contact load for contact pair group 2. It is noted that the total number of balls in the analyzed bearing is odd. Consequently, the maximum contact load for contact pair group 1 occurs at Ball 1, while that for contact pair group 2 occurs at Balls 38 and 39.

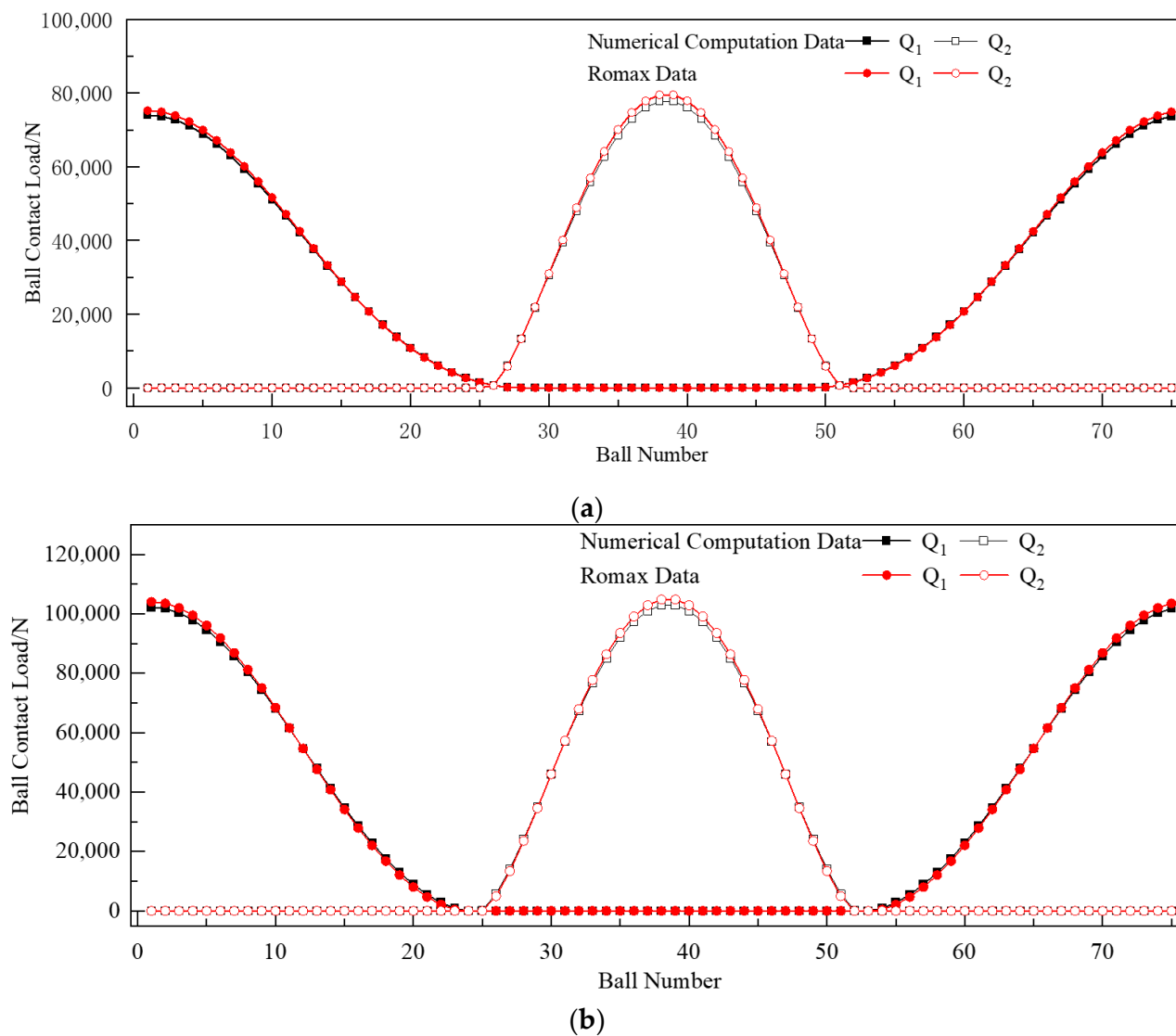


Figure 6. Cont.

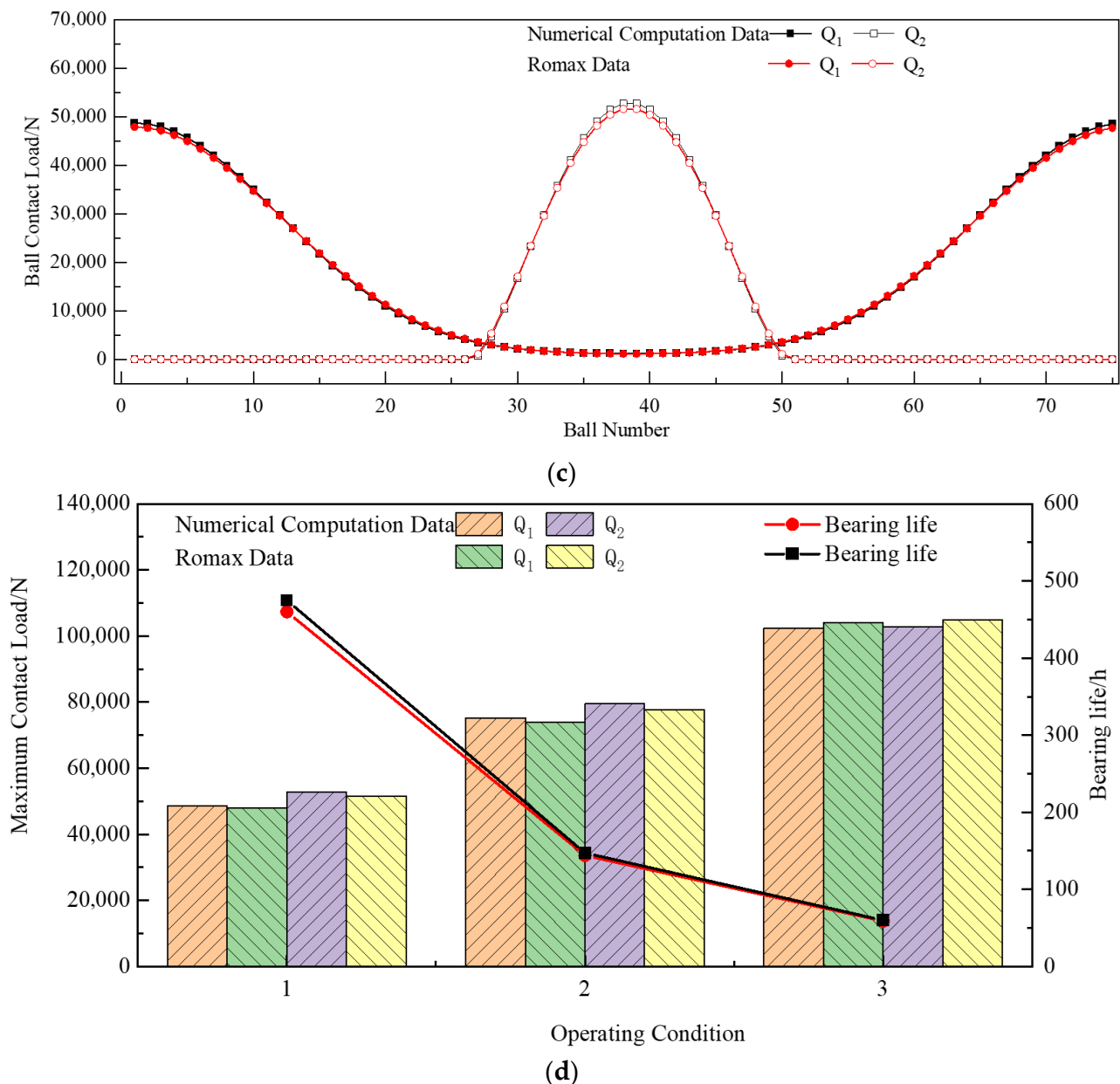


Figure 6. Comparison results between the present model and Romax 2023 software: (a) load distribution comparison under operating condition 1; (b) load distribution comparison under operating condition 2; (c) load distribution comparison under operating condition 3; (d) comparison of maximum contact loads and fatigue life.

As can be seen from Figure 6, the model proposed in this paper shows good agreement with the results from the commercial software Romax 2023 in terms of the calculated ball load distribution and bearing fatigue life. Table 3 lists the maximum differences and discrepancies for Q_1 , Q_2 , and the bearing fatigue life. The maximum discrepancy for the ball contact load does not exceed 2.2%, and that for the bearing fatigue life does not exceed 3%. This demonstrates that the proposed model is correctly established and possesses sufficient accuracy. Consequently, this model can be extended to incorporate calculations for load distribution and fatigue life under the condition of outer ring elliptical deformation.

Table 3. Differences and discrepancies in the comparison results between the present model and Romax software.

Operating Condition	1		2		3	
	Difference	Tolerance	Difference	Tolerance	Difference	Tolerance
$Q_{1\max}$	834	0.0171	1239	0.0165	1780	0.0174
$Q_{2\max}$	1153	0.0219	1805	0.0227	1953	0.019
Bearing life	14.5	0.0299	3.4	0.0231	1.2	0.02

3. Results and Analysis

The four-point contact slewing bearing analyzed in this paper operates under Operating Condition 2 for the longest duration during its working cycle. Therefore, the subsequent analysis uniformly adopts the load parameters from Operating Condition 2 for calculation. Specifically, the bearing is subjected to an axial load of 700 kN, a radial load of 0 kN, an overturning moment of 1600 kNm, and operates at a speed of 12 r/min. In this bearing configuration, the semi-major axis of the elliptical deformation corresponds to azimuth angles of 0° and 180° . These angular positions coincide with the location of Ball 1 and the mid-region between Balls 38 and 39, respectively. Given that this geometric relationship does not precisely align with a single ball index, for the sake of clarity and consistency in subsequent analysis and discussion, this paper consistently uses the azimuth angles of 0° and 180° to refer to these two specific load-bearing orientations.

3.1. Impact of Outer Ring Elliptical Deformation on Bearing Load Distribution and Fatigue Life

Figure 7 illustrates the variation patterns in the bearing's contact load distribution characteristics under different degrees of outer ring elliptical deformation. Figure 7a shows the contact load distribution for Contact Pair Group 1. It can be observed that when no elliptical deformation occurs in the outer ring, the contact load exhibits only a single peak at the 0° position. However, with the onset of elliptical deformation, a second peak gradually forms in the load distribution at 180° . As λ increases, the load values at both the 0° and 180° peaks rise continuously, while the load-bearing zone of the balls progressively narrows. Figure 7b depicts the contact load distribution for Contact Pair Group 2. When $\lambda = 0$, the contact load has only one peak located at 180° . When $\lambda = 1.0002$, another peak begins to appear at the 0° position. With further increase in λ , the contact load at the peak locations likewise increases, and the load-bearing zone of the balls continues to shrink. Synthesizing the results from both figures, an increase in elliptical deformation leads to a significant concentration of contact load at specific angular positions (0° or 180°), accompanied by a progressive reduction in the extent of the balls' load-bearing zone. The fundamental cause of this phenomenon lies in the fact that elliptical deformation disrupts the internal geometric compatibility of the bearing. This causes loads to become highly concentrated along the direction of the semi-major axis (0° and 180°), while inducing unloading of the balls along the semi-minor axis direction, resulting in the shrinkage of the load-bearing zone.

As can be seen from Figure 8, as λ increases, both $Q_{1\max}$ and $Q_{2\max}$ show a significant upward trend. Correspondingly, the bearing's fatigue life exhibits a continuous downward trend. Therefore, an increase in λ not only intensifies the local concentration of contact loads, directly leading to a reduction in the bearing's overall load-carrying capacity, but also significantly accelerates the degradation of the bearing's service life.

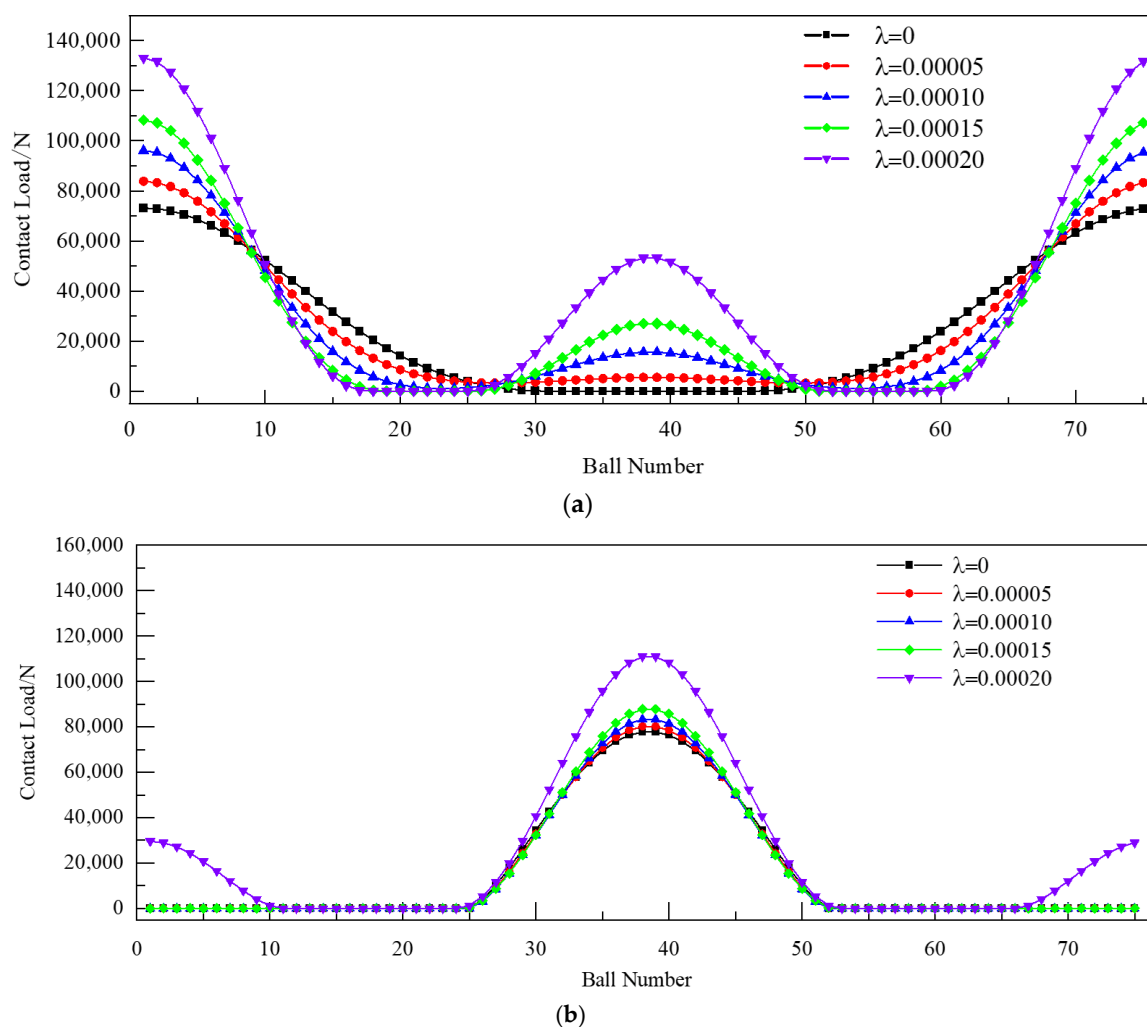


Figure 7. Load distribution of balls under different degrees of elliptical deformation; (a) Q_1 distribution; (b) Q_2 distribution.

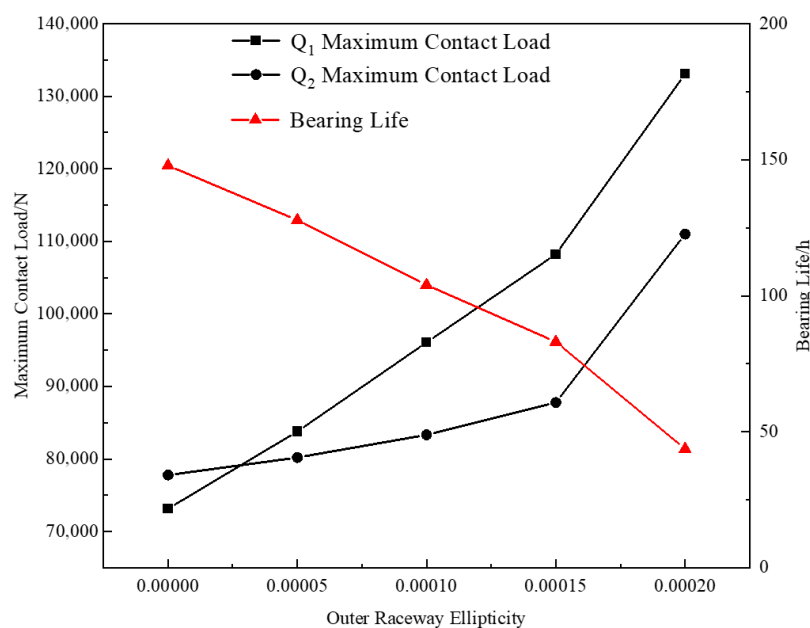


Figure 8. Variation in maximum ball contact loads and bearing fatigue life under different degrees of elliptical deformation.

3.2. Effect of Groove Curvature Radius Coefficient on Bearing Load Distribution and Fatigue Life Under Outer Ring Elliptical Deformation

Figure 9 illustrates the influence of the groove curvature radius coefficient on bearing load distribution under different degrees of outer ring elliptical deformation. As this coefficient increases, $Q_{1\max}$ gradually decreases, and the load-bearing zone of the balls expands accordingly. In contrast, $Q_{2\max}$ continues to rise, with a relatively smaller change in its load-bearing zone. An increase in the groove curvature radius coefficient leads to greater conformity between the raceway and the balls, effectively reducing the local contact stiffness between them. This results in a smaller variation in load for a given amount of deformation. When elliptical deformation occurs in the outer ring, four-point contact forms between the balls and the raceway along the semi-major axis, generating a second load peak. This peak continuously rises with increasing λ , thereby significantly reducing bearing life. For smaller λ values, a larger groove curvature radius coefficient is more sensitive, causing a greater variation in the second peak. However, when λ increases significantly, the sensitivity of a smaller coefficient rises sharply and becomes the dominant factor, while the influence of a larger coefficient weakens.

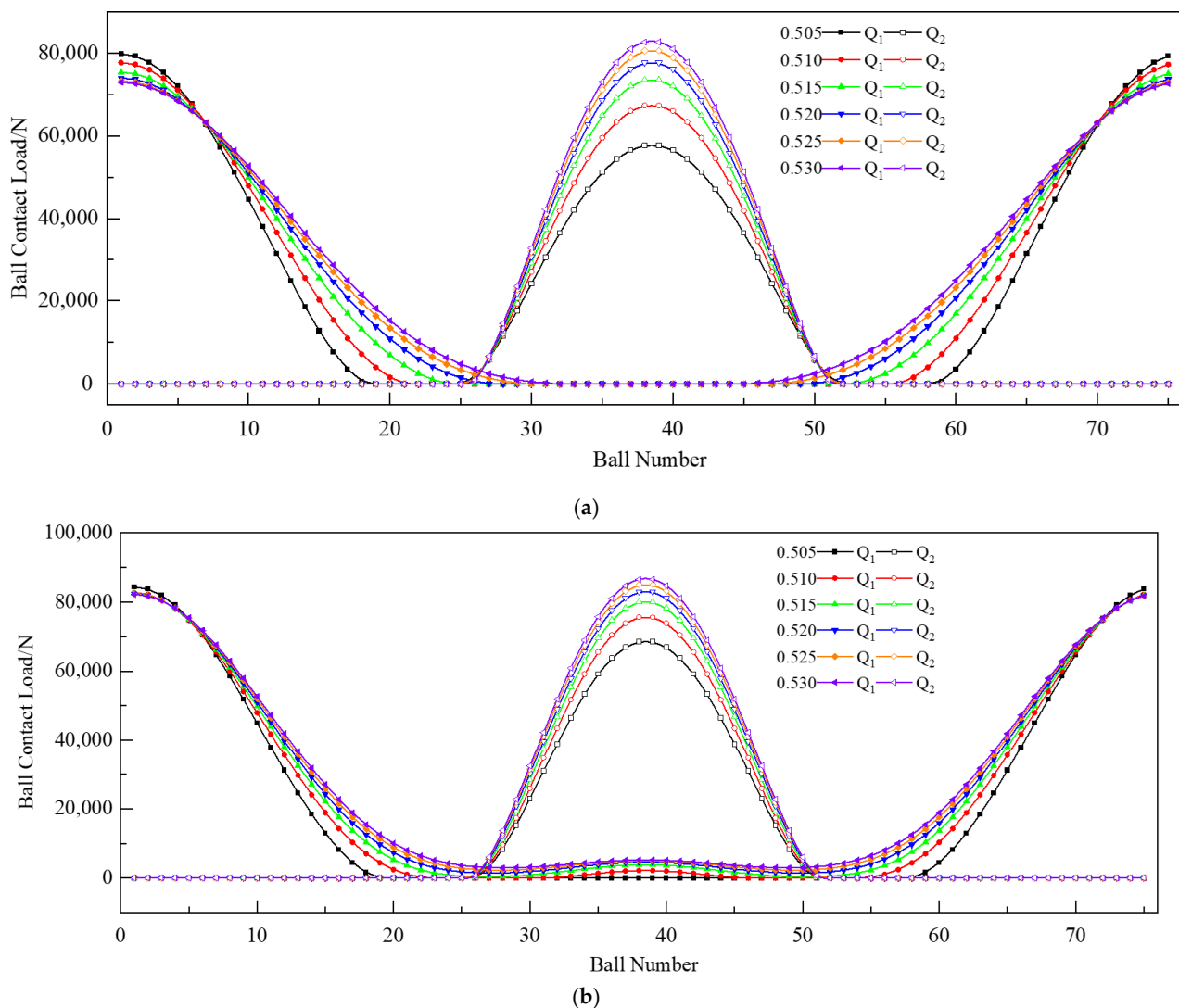


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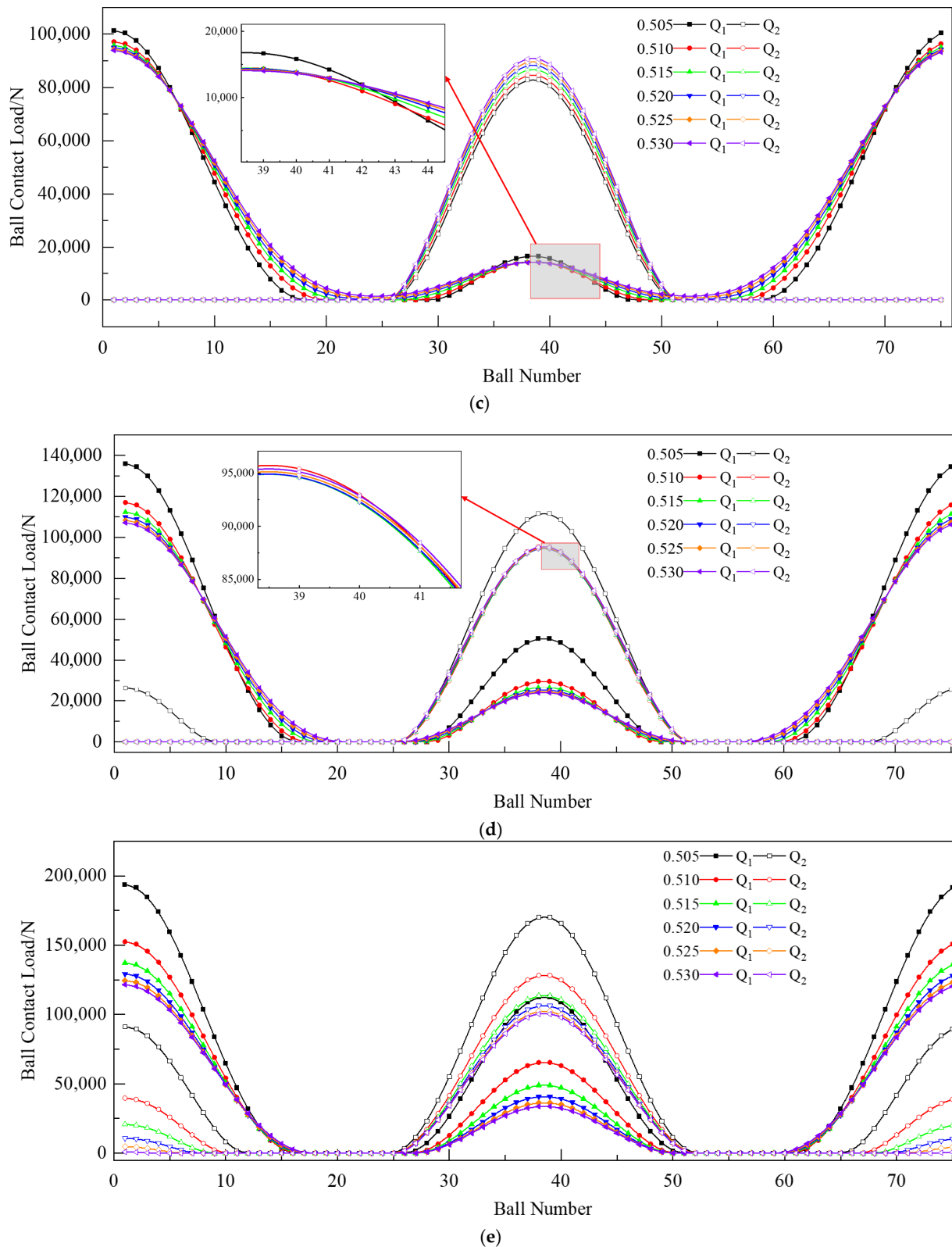


Figure 9. Effect of groove curvature radius coefficient on bearing load distribution under different degrees of outer ring elliptical deformation. (a) $\lambda = 0$; (b) $\lambda = 0.00005$; (c) $\lambda = 0.00010$; (d) $\lambda = 0.00015$; (e) $\lambda = 0.00020$.

Figure 10 reflects the influence of the groove curvature radius coefficient on the maximum contact loads and fatigue life of the bearing under different degrees of outer ring elliptical deformation. As can be seen from Figure 10a, under varying degrees of elliptical

deformation, $Q_{1\max}$ exhibits a monotonically decreasing trend as the groove curvature radius coefficient increases. In Figure 10b, the variation in $Q_{2\max}$ shows differences: when λ is less than 0.0001, $Q_{2\max}$ increases monotonically with the groove curvature radius coefficient; when λ is 0.00015, $Q_{2\max}$ demonstrates a non-monotonic characteristic, first decreasing and then increasing; and when λ is 0.0002, $Q_{2\max}$ switches to a monotonically decreasing trend. When λ is 0, the initial loads for both $Q_{1\max}$ and $Q_{2\max}$ are minimal, consistently occupying the bottom of the load curves. As λ increases, the regulatory effect of the groove curvature radius coefficient on both $Q_{1\max}$ and $Q_{2\max}$ gradually becomes prominent. When the groove curvature radius coefficient is 0.505, the variation amplitude of $Q_{1\max}$ and $Q_{2\max}$ with changing λ is the greatest; whereas when this coefficient gradually increases to 0.53, their variation amplitude reaches its minimum.

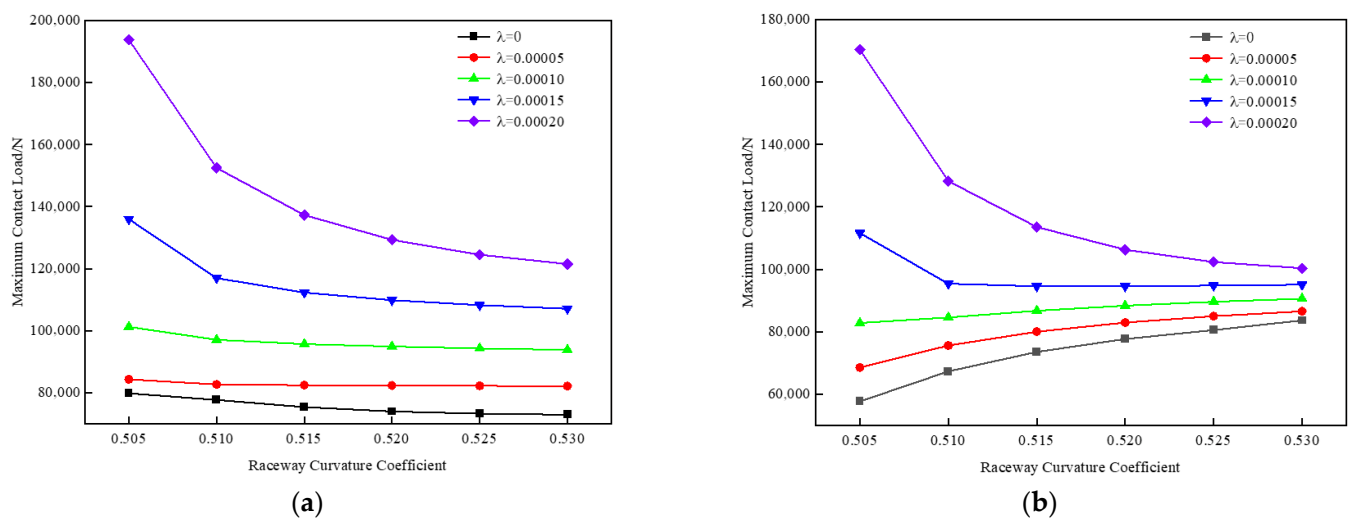


Figure 10. Influence of the groove curvature radius coefficient on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. (a) Variation in maximum contact load for Q_1 ; (b) Variation in maximum contact load for Q_2 .

Figure 11 illustrates the occurrence of the second peak in Q_1 and Q_2 , induced by elliptical deformation of the bearing's outer ring, as referenced from Figure 9. From Figure 11a, it can be observed that when λ is less than 0.00005, no second peak in contact load appears as the groove curvature radius coefficient increases. However, when λ is greater than or equal to 0.0001, a second peak emerges in Q_1 at the 180° azimuth angle. Furthermore, the contact load at this second peak gradually decreases as the groove curvature radius coefficient increases. As shown in Figure 11b, when λ is less than 0.0001, a second peak in contact load is still not generated with an increasing groove curvature radius coefficient. Once λ is greater than or equal to 0.00015, a second peak appears in Q_2 at the 0° azimuth angle. Similarly, the contact load at this peak also continuously decreases as the groove curvature radius coefficient increases.

As shown in Figure 12, under different values of λ , the bearing's fatigue life exhibits a monotonically decreasing trend with increasing groove curvature radius coefficient. When λ is 0, the bearing's fatigue life reaches its maximum value on the life curve. As λ increases, the fatigue life values corresponding to each groove curvature radius coefficient decrease significantly. When the groove curvature radius coefficient is 0.505, the corresponding bearing fatigue life is relatively the highest; however, the degradation rate of fatigue life at this coefficient with increasing λ is the most severe. Conversely, when the groove curvature radius coefficient is 0.53, the degradation rate of fatigue life with changing λ is the smallest.

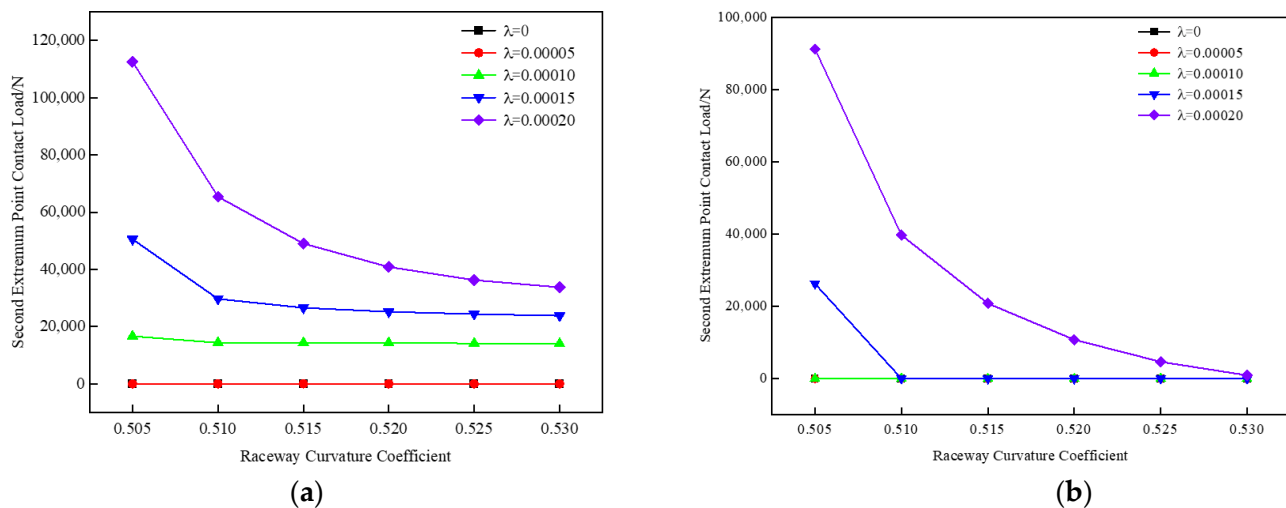


Figure 11. Influence of the groove curvature radius coefficient on the second peak contact loads of the bearing under different degrees of outer ring elliptical deformation. (a) Variation in second peak contact load for Q_1 ; (b) variation in second peak contact load for Q_2 .

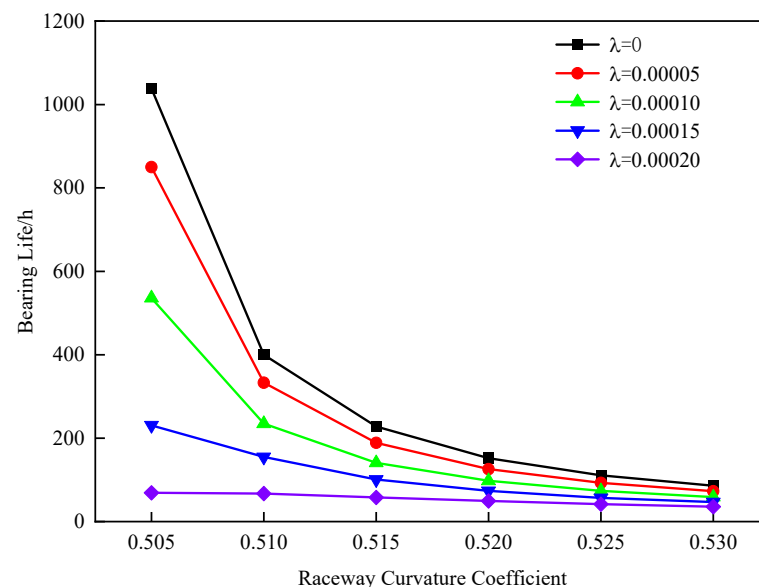


Figure 12. Effect of groove curvature radius coefficient on bearing fatigue life under different degrees of outer ring elliptical deformation.

A smaller groove curvature radius coefficient can provide a relatively higher initial fatigue life for a given λ ; however, its fatigue life also exhibits the greatest rate of decline as λ increases. Furthermore, when $\lambda > 0$, a second load peak is more likely to occur at and near the 0° and 180° azimuth angles, thereby affecting the bearing's load-carrying capacity and fatigue life. In contrast, a larger groove curvature radius coefficient, despite offering a lower initial fatigue life, possesses a stronger resistance to life degradation. Therefore, to mitigate the adverse effects of outer ring elliptical deformation on bearing performance and based on the above analysis, for this bearing model and while considering both load-carrying capacity and fatigue life, controlling the groove curvature radius coefficient within the range of 0.51 to 0.52 represents the optimal design choice.

3.3. Effect of Axial Clearance on Bearing Load Distribution and Fatigue Life Under Outer Ring Elliptical Deformation

Figure 13 illustrates the influence of axial clearance on the load distribution of balls under different degrees of outer ring elliptical deformation. Analysis indicates that under the influence of λ , the variation pattern of the load distribution peak with axial clearance evolves from initially decreasing and then increasing to a pattern of continuous decrease. When λ is small, an optimal clearance range exists, which can optimize the ball load distribution to a certain extent. However, as λ increases, the load concentration effect induced by geometric deformation becomes dominant, and its influence far exceeds the fine-tuning effect of clearance. Consequently, the load peak continuously worsens with increasing clearance. Simultaneously, axial clearance has a significant impact on the extent of the bearing's load-bearing zone, specifically manifested as a larger clearance leading to a smaller load-bearing zone. Furthermore, a smaller axial clearance is more sensitive to changes in λ , resulting in a greater amplitude of variation in the maximum contact load.

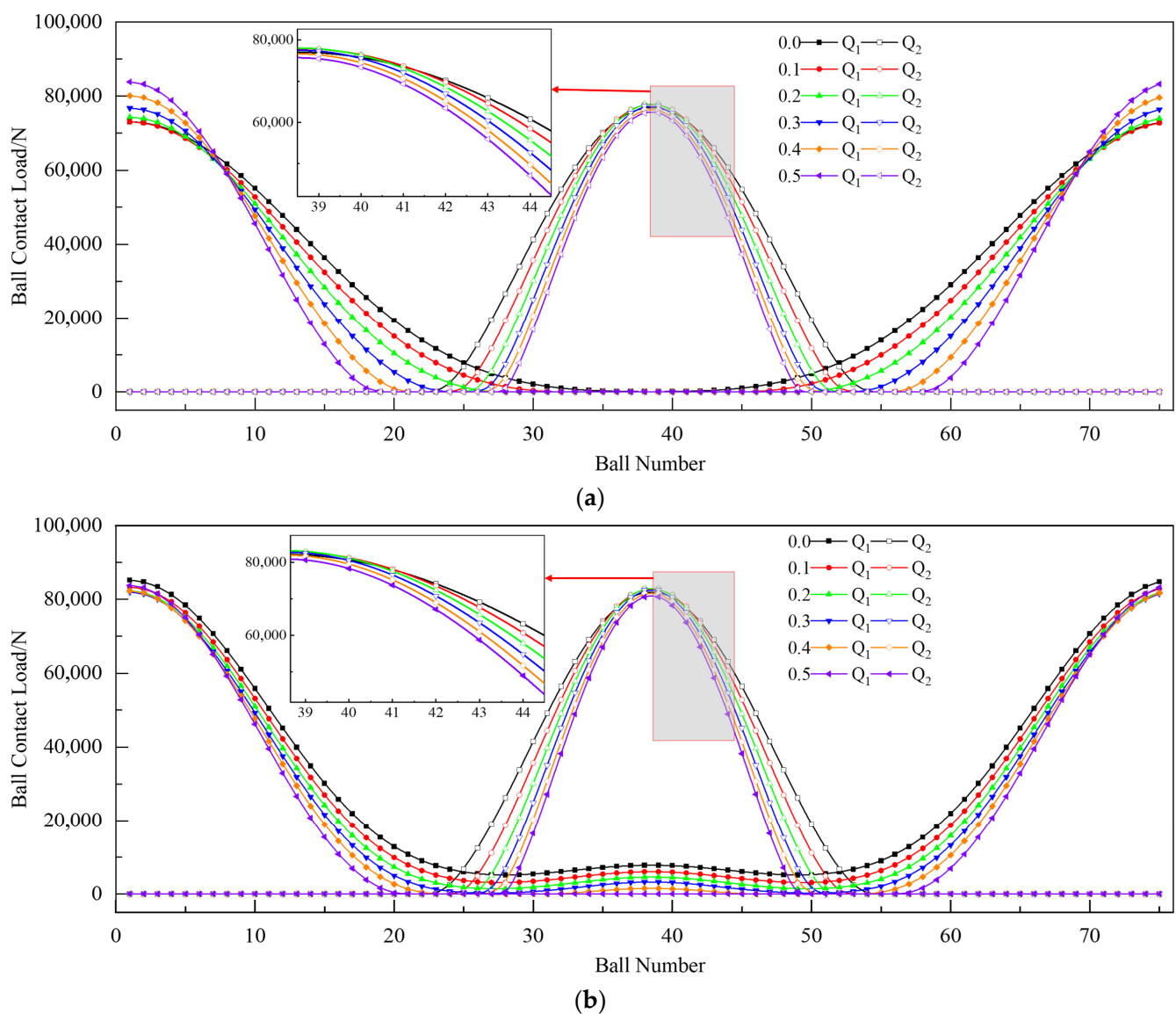


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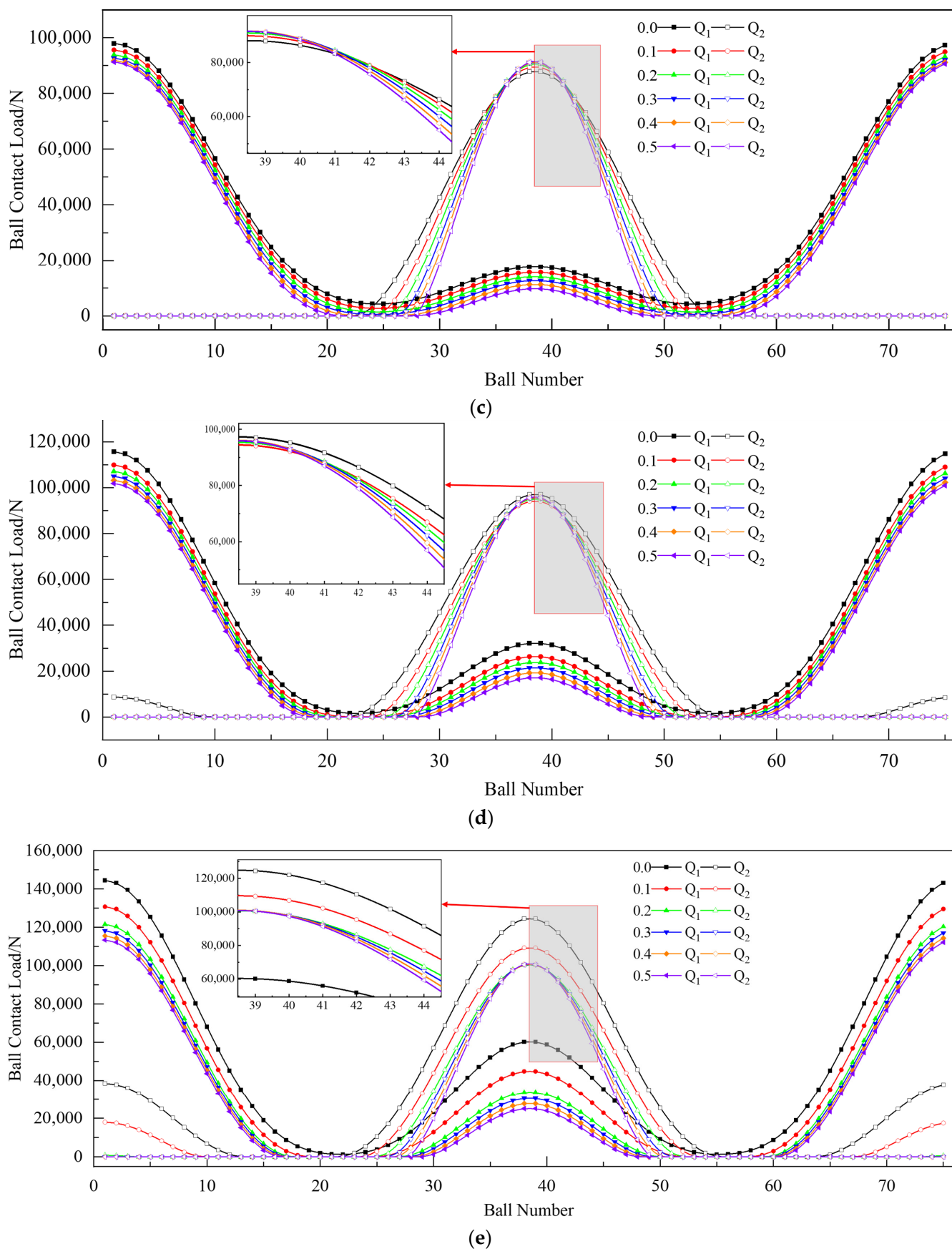


Figure 13. Effect of axial clearance on bearing load distribution under different degrees of outer ring elliptical deformation. (a) $\lambda = 0$; (b) $\lambda = 0.00005$; (c) $\lambda = 0.00010$; (d) $\lambda = 0.00015$; (e) $\lambda = 0.00020$.

Figure 14 illustrates the variation in the effect of axial clearance on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. As λ increases, both $Q_{1\max}$ and $Q_{2\max}$ under all clearance levels rise accordingly. When the axial clearance is small, the load is most sensitive to λ , exhibiting the largest amplitude of variation. As shown in Figure 14a, when $\lambda = 0$, $Q_{1\max}$ increases monotonically with increasing axial clearance. However, as λ continuously increases, $Q_{1\max}$ gradually transitions to a decreasing trend. As shown in Figure 14b, when λ is less than 0.0001, $Q_{2\max}$ first increases and then decreases with increasing axial clearance. At smaller clearances, due to the higher load on the bearing, the internal load distribution is not yet severely imbalanced, and Contact Pair Group 2 still participates in load-bearing. Once the clearance exceeds a critical value, the number of balls in contact continuously decreases. Contact Pair Group 1 begins to carry the majority of the load, while Contact Pair Group 2 slowly transitions from cooperative load-sharing to unloading, resulting in the observed trend of $Q_{2\max}$ first increasing and then decreasing. When λ is greater than or equal to 0.0001, $Q_{2\max}$ decreases with increasing axial clearance.

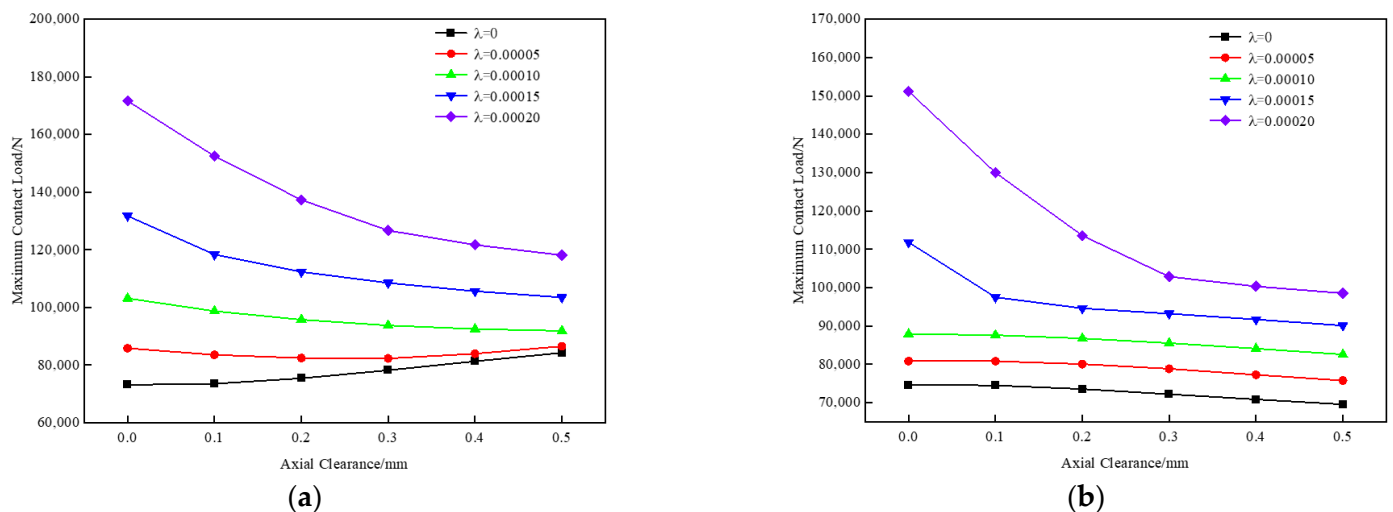


Figure 14. Effect of axial clearance on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. (a) Variation in maximum contact load for Q_1 ; (b) variation in maximum contact load for Q_2 .

Figure 15 illustrates the variation in the effect of axial clearance on bearing fatigue life under different degrees of outer ring elliptical deformation. As shown in Figure 15, for all values of λ , the bearing's fatigue life exhibits a trend of first increasing and then decreasing with increasing axial clearance (when λ is large, its inflection point shifts to a larger clearance value; in the figure, for λ greater than 0.0001, the maximum axial clearance shown is less than the clearance corresponding to this inflection point). However, the specific inflection point differs among bearings and under various loading conditions. When $\lambda = 0$, the bearing's fatigue life reaches its maximum value on the life curve. As λ increases, the fatigue life corresponding to each axial clearance decreases significantly, and the optimal clearance corresponding to the maximum fatigue life also continuously increases.

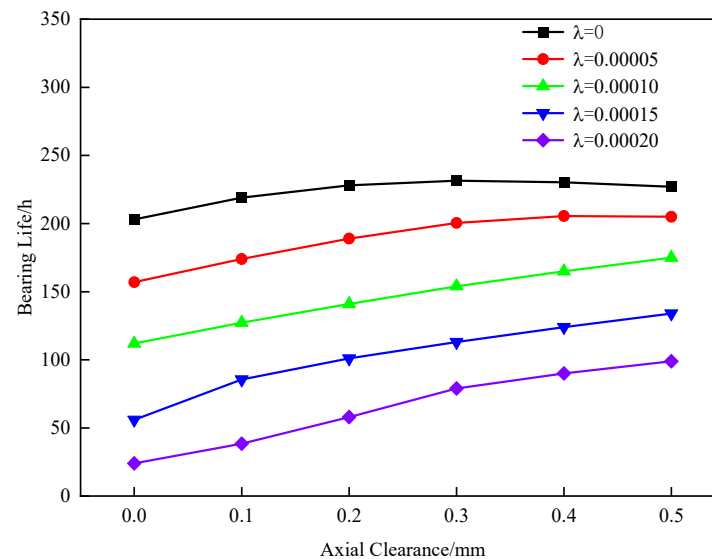


Figure 15. Effect of axial clearance on bearing fatigue life under different degrees of outer ring elliptical deformation.

In summary, elliptical deformation of the outer ring significantly increases the maximum contact loads and reduces the bearing fatigue life under all clearance conditions. The maximum contact load and fatigue life are most sensitive to deformation when the axial clearance is small. Conversely, a larger axial clearance results in a lower bearing fatigue life. The optimal clearance corresponding to the maximum bearing fatigue life continuously increases with the degree of elliptical deformation. Therefore, to achieve a longer bearing fatigue life and to mitigate the adverse effects of elliptical deformation on the bearing, the axial clearance selected should be the optimal clearance for this specific bearing model or slightly greater than this optimal value.

3.4. Effect of Design Contact Angle on Bearing Load Distribution and Fatigue Life Under Outer Ring Elliptical Deformation

Figure 16 illustrates the influence of the design contact angle on bearing load distribution under different degrees of outer ring elliptical deformation. The results show that as the design contact angle increases, the peak value of the load distribution gradually decreases. A larger design contact angle enhances the axial stiffness of the bearing, leading to a redistribution of load across a greater number of balls. This mitigates stress concentration in localized areas, effectively reducing the distribution peak. Although the variation in contact angle has a limited impact on the extent of the bearing's load-bearing zone, a smaller contact angle is more sensitive to changes in the degree of elliptical deformation. When $\lambda = 0.00005$, the bearing is prone to the four-point contact phenomenon at the 180° position when a smaller contact angle is employed. As the λ value increases, four-point contact also begins to appear at the 0° position when the contact angle is small, and the associated contact load value continuously rises. Therefore, a smaller design contact angle significantly lowers the system's stability threshold, inducing complex multi-point contact behavior at an earlier stage, thereby posing a potential risk to both load distribution and fatigue life.

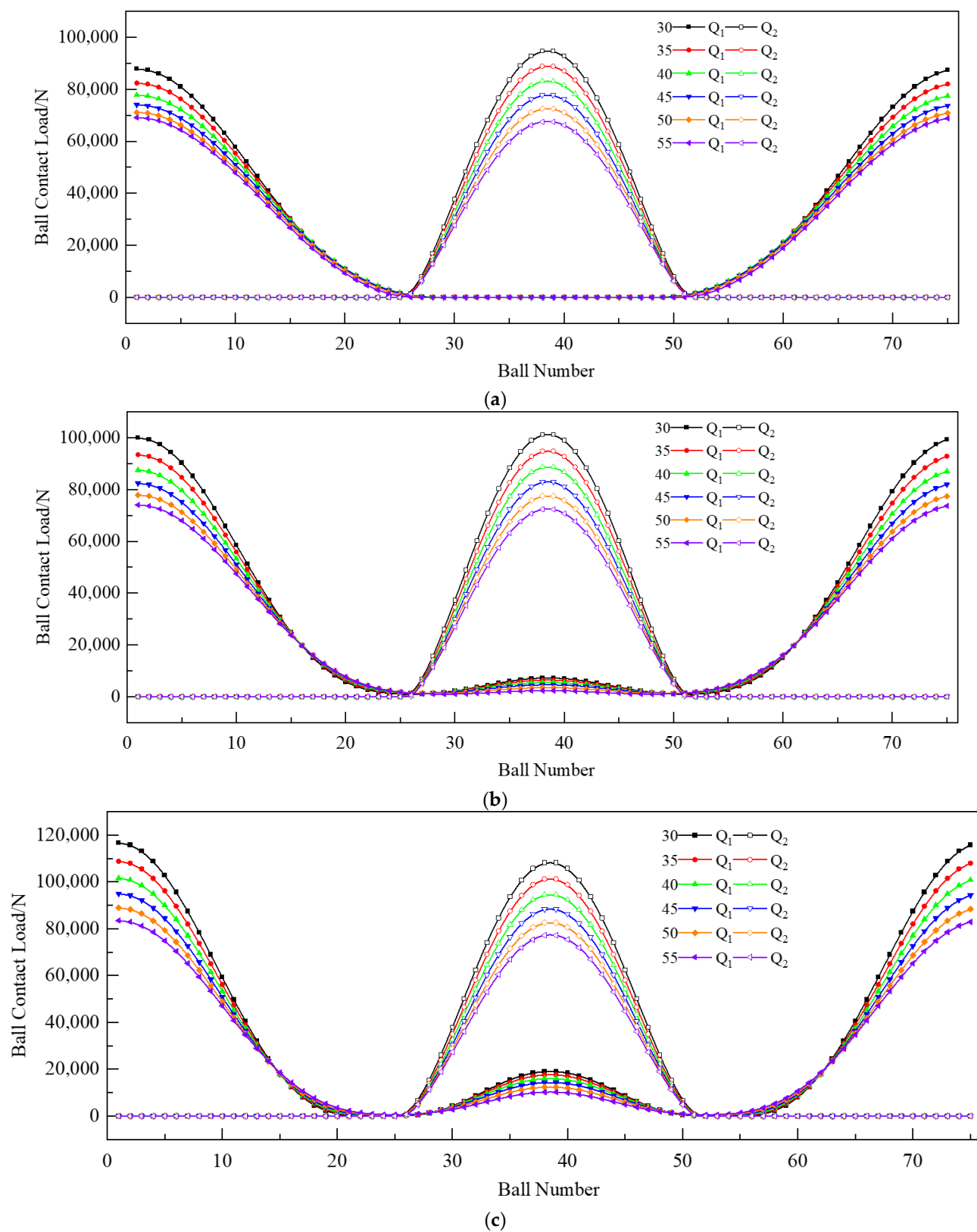


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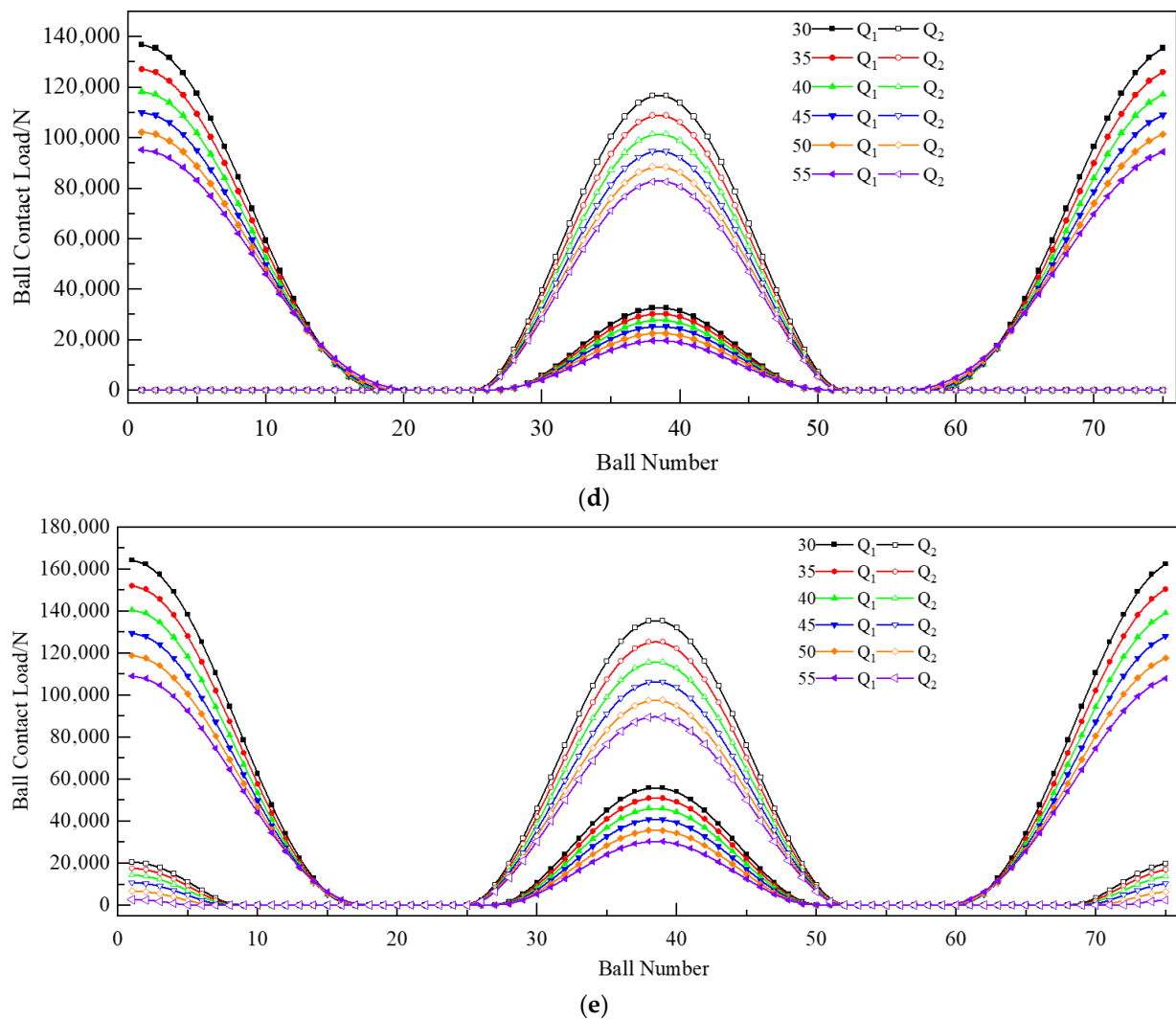


Figure 16. Effect of contact angle on bearing load distribution under different degrees of outer ring elliptical deformation. (a) $\lambda = 0$; (b) $\lambda = 0.00005$; (c) $\lambda = 0.00010$; (d) $\lambda = 0.00015$; (e) $\lambda = 0.00020$.

Figure 17 illustrates the variation in the effect of design contact angle on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. As shown in Figure 17, when $\lambda = 0$, both $Q_{1\max}$ and $Q_{2\max}$ exhibit a monotonically decreasing trend with increasing design contact angle, and their initial values are the smallest. As λ increases, the maximum contact loads corresponding to each design contact angle increase monotonically, and the magnitude of this increase grows with larger λ . However, when the contact angle increases, the amplitude of variation in the maximum contact load with changing λ shows a monotonically decreasing trend. Therefore, a larger contact angle has a significant regulatory effect on the variation in maximum contact loads induced by outer ring elliptical deformation.

As shown in Figure 18, under different degrees of elliptical deformation, the bearing fatigue life exhibits a monotonically increasing trend with an increasing design contact angle. When $\lambda = 0$, the bearing fatigue life reaches its maximum value on the life curve. As λ increases, the fatigue life values corresponding to each design contact angle significantly decrease.

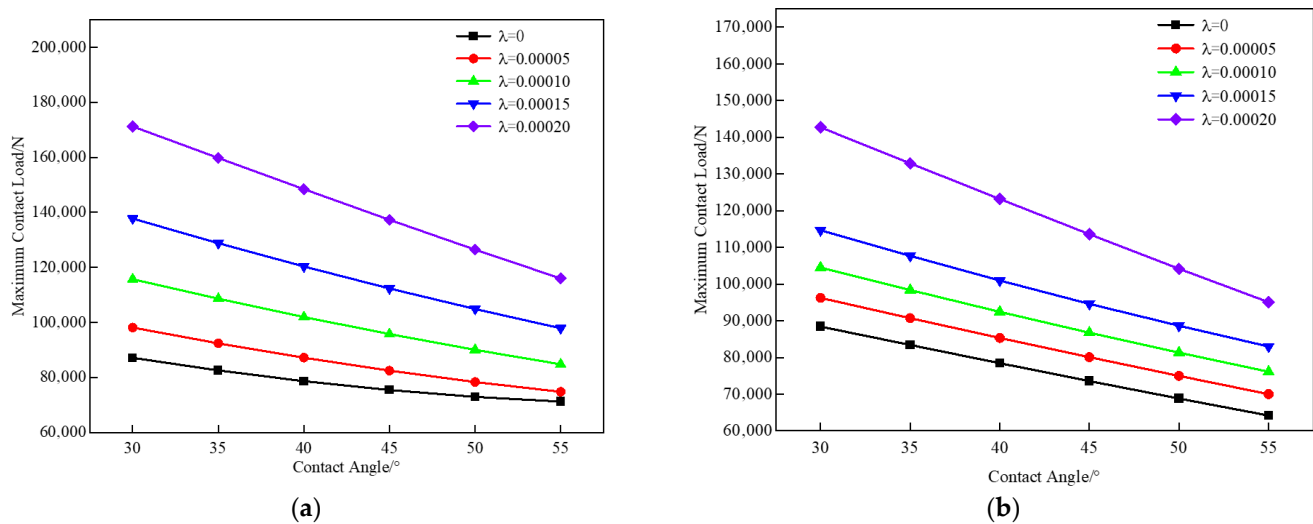


Figure 17. Effect of design contact angle on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. (a) Variation in maximum contact load for Q_1 ; (b) variation in maximum contact load for Q_2 .

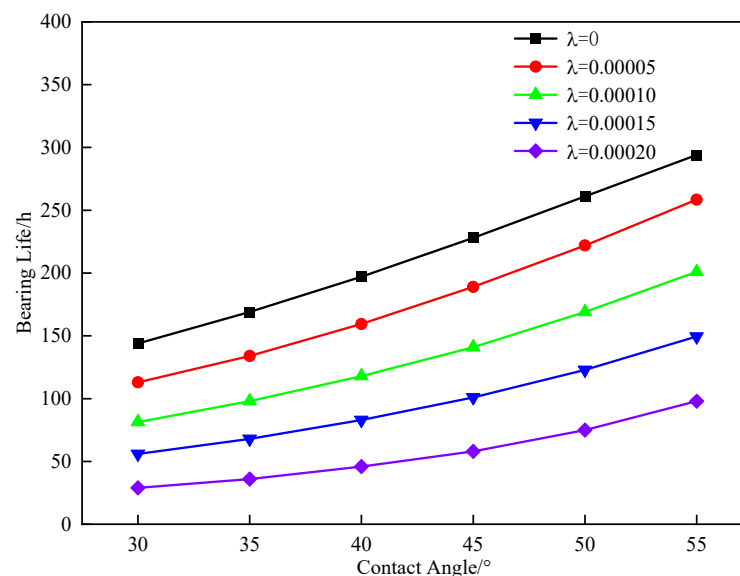


Figure 18. Effect of design contact angle on bearing fatigue life under different degrees of outer ring elliptical deformation.

With an increasing degree of outer ring deformation, the maximum contact load of the bearing rises significantly. Increasing the design contact angle can effectively reduce the value of this maximum contact load. Therefore, selecting a larger contact angle during the design phase can help mitigate the degradation of bearing fatigue life caused by elliptical deformation. However, an excessively large contact angle can lead to the phenomenon of “ellipse truncation.” Specifically, in rolling bearings, the theoretical contact area between a ball and the raceway is an ellipse. When the contact angle is too large, this ideal contact ellipse can experience geometric interference with the bearing’s rib, causing a portion of the contact area to be “cut off” or prevented from forming normally. Consequently, the design contact angle should be determined based on a comprehensive consideration of the actual dimensions and operating conditions.

3.5. Effect of Ball Parameters on Bearing Load Distribution and Fatigue Life Under Outer Ring Elliptical Deformation

Figure 19 illustrates the influence of the number of balls on the bearing load distribution under different degrees of outer ring elliptical deformation. Due to the variation in the number of balls, the azimuth angle is adopted as the horizontal coordinate in this analysis to better visualize the effect on load distribution. The results show that as the number of balls increases, the peak of the load distribution gradually decreases. This is because, under a constant total load, the load is distributed across a greater number of load-bearing elements. When λ is less than 0.0001, the four-point contact phenomenon occurs near the 180° position, and the contact range is essentially unaffected by the number of balls, indicating that outer ring elliptical deformation primarily affects a specific localized region. When λ is greater than 0.0001, four-point contact also begins to appear at the 0° position, and the associated contact load value increases with λ . However, under these conditions, the load value at the second peak increases with a greater number of balls.

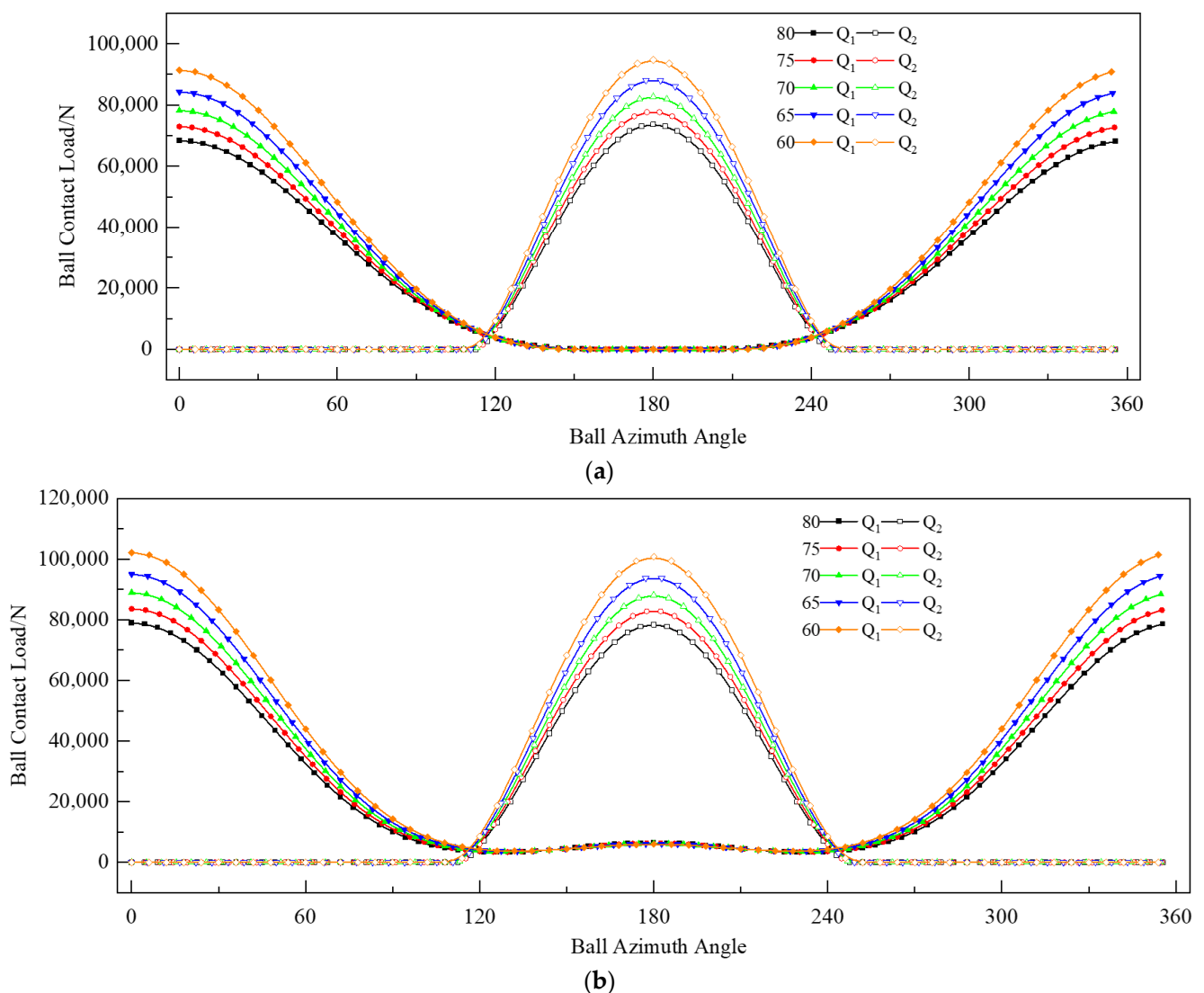


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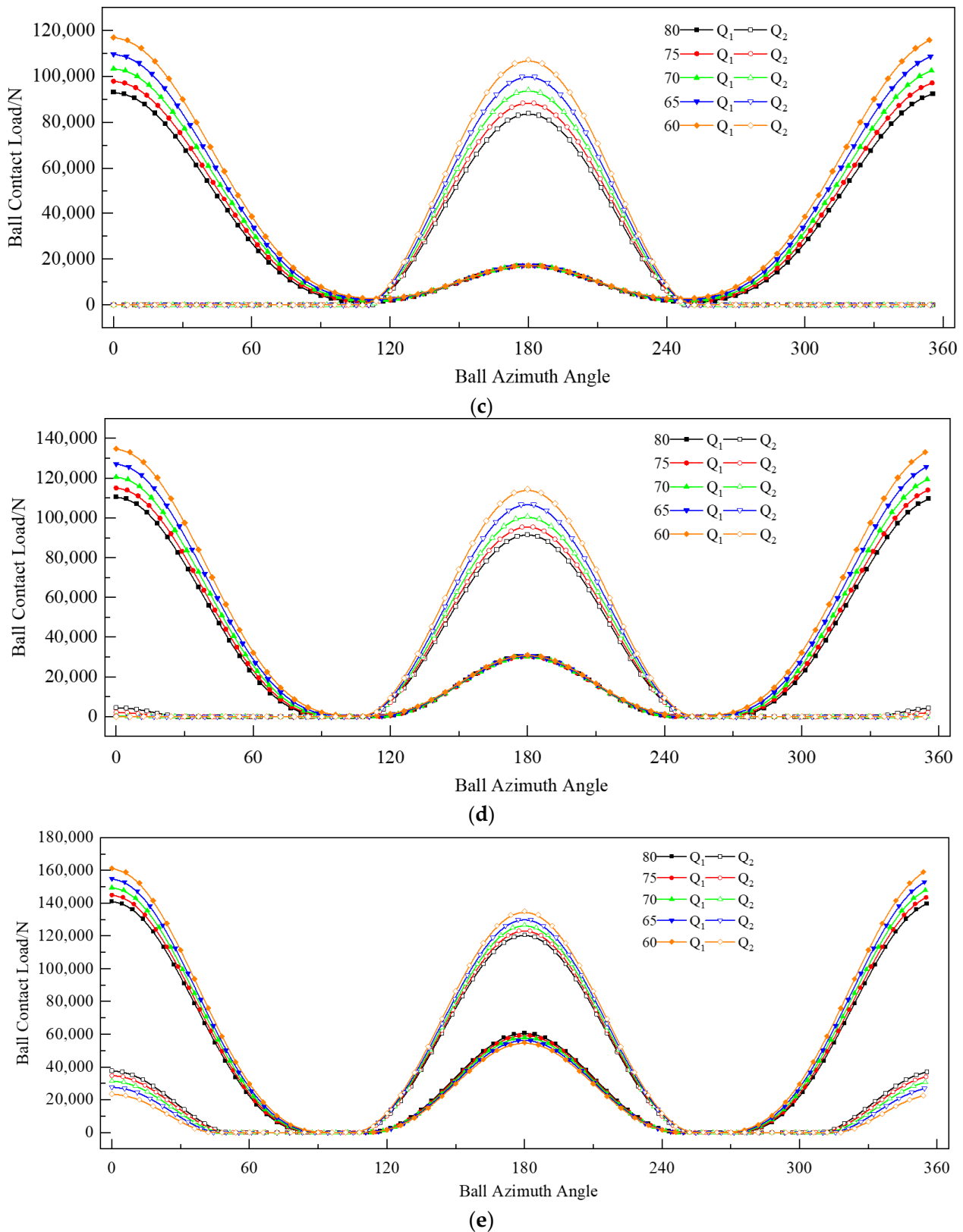


Figure 19. Effect of number of balls on bearing load distribution under different degrees of outer ring elliptical deformation. (a) $\lambda = 0$; (b) $\lambda = 0.00005$; (c) $\lambda = 0.00010$; (d) $\lambda = 0.00015$; (e) $\lambda = 0.00020$.

Figure 20 illustrates the variation in the effect of the number of balls on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. As shown in Figure 20, when $\lambda = 0$, both $Q_{1\max}$ and $Q_{2\max}$ exhibit a monotonically decreasing

trend with an increasing number of balls. As λ increases, the maximum contact loads corresponding to different numbers of balls all show a monotonically increasing trend, and the magnitude of this increase grows with larger λ .

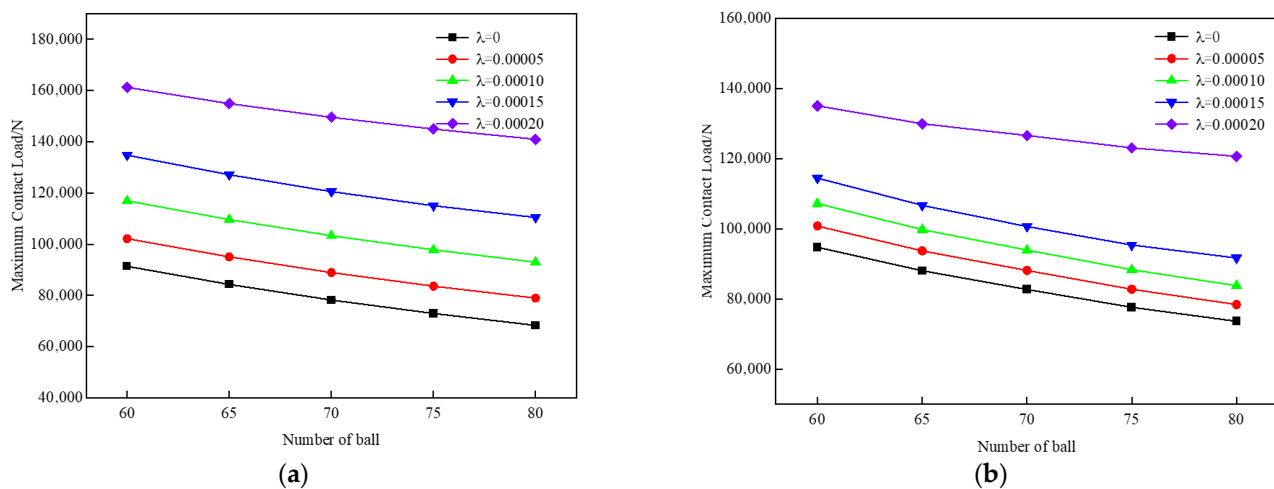


Figure 20. Effect of number of balls on the maximum contact loads of the bearing under different degrees of outer ring elliptical deformation. (a) Variation in maximum contact load for Q1; (b) variation in maximum contact load for Q2.

As shown in Figure 21, under different degrees of elliptical deformation (λ), the bearing's fatigue life exhibits a monotonically increasing trend with an increasing number of balls. As λ increases, the fatigue life of the bearing continuously decreases. Notably, bearings with a greater number of balls typically experience a more pronounced magnitude of life degradation. Despite this, for any given λ value, bearings with more balls consistently maintain the highest fatigue life.

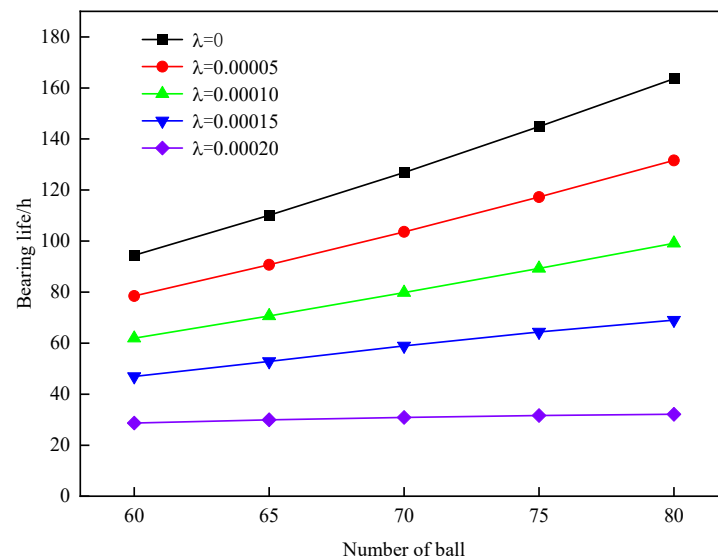


Figure 21. Effect of number of balls on bearing fatigue life under different degrees of outer ring elliptical deformation.

With an increasing degree of outer ring deformation, the maximum contact load of the bearing rises significantly. Conversely, increasing the number of balls can effectively reduce this maximum contact load. Therefore, during the bearing design stage, the number

of balls should be maximized as much as possible, provided that other design parameters meet their requirements, to enhance the overall service life of the bearing.

4. Conclusions

This paper focuses on the variation in the load-carrying capacity of four-point contact slewing bearings for excavators before and after elliptical deformation of the outer ring, analyzing four key structural parameters: the groove curvature radius coefficient, axial clearance, design contact angle, and number of balls. By comparing the maximum contact loads of the two contact pair groups and the bearing's fatigue life, the influence patterns of these parameters are revealed, leading to the following conclusions:

1. The presence of elliptical deformation in the bearing outer ring has a significant impact on its fatigue life. Therefore, during installation and design, measures should be taken to minimize the occurrence of such deformation.
2. When the groove curvature radius coefficient is relatively small, the influence of the degree of elliptical deformation is more pronounced. The extreme values (peaks) in the load distribution for both contact pair groups increase, and the bearing becomes more susceptible to the four-point contact issue, consequently leading to a reduction in its fatigue life. Conversely, when the groove curvature radius coefficient is relatively large, the resulting increase in contact stress causes the bearing fatigue life to be lower. Therefore, for the selection of the groove curvature in four-point contact slewing bearings, a moderate groove curvature radius coefficient should be adopted. This choice must balance considerations of service life with maintaining the intended four-point contact geometry. For this specific bearing type, a groove curvature radius coefficient in the range of 0.51 to 0.52 is recommended.
3. For given parameters and operating conditions of a four-point contact slewing bearing, there exists an optimal axial clearance. This optimal axial clearance increases with the degree of elliptical deformation of the outer ring. Therefore, during bearing design, the axial clearance should be set equal to or slightly greater than this optimal value.
4. A larger design contact angle results in a stronger bearing load-carrying capacity and reduces the influence of elliptical deformation variation on bearing fatigue life. Therefore, employing a larger design contact angle during design can be considered to enhance both the bearing's load capacity and fatigue life. However, the use of a larger contact angle simultaneously leads to "ellipse truncation". Consequently, the size of the design contact angle should be determined based on a comprehensive consideration of the actual bearing structural parameters and applied loads.
5. Increasing the number of balls mitigates the adverse effects of elliptical deformation on load distribution and improves the bearing's fatigue life. Therefore, during the bearing design stage, the number of balls should be maximized as much as possible, provided that other design parameters meet their requirements, to enhance the overall service life of the bearing.

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