Life Analysis for the Main bearing of Aircraft Engines

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ABSTRACT: Life of main bearings in a aircraft engine directly affects the reliability, safety and feasibility of aircraft engines. In order to optimize and improve the performance of the existing aircraft engines, as well meet the needs of a new generation of aircraft engines, this paper analyses and estimate of main bearing's life of aircraft engines by taking the deep groove ball bearing as an example. Firstly, the 3D model of deep groove ball bearings is established by using Pro-E software and then converted into a finite element model. Secondly, such features as stiffness, strength and fatigue life of the deep groove ball bearing are investigated by ANSYS software, which result in some theoretical are discussion and some relevant measures that enable improving the service life of main bearings of the aircraft engine.

Keywords: aircraft engines, deep groove ball bearing, service life, reliability, finite element analysis

1. INTRODUCTION

The bearing of main shaft is a major component of aircraft engine. With the development of aircraft engines, the demand of working condition has been increasing rapidly. Since life and reliability of main bearing directly affect the life and reliability of the engine, failures of main bearing would lead to disastrous consequences.

As the loading support and movement connecting component, bearing can transform the sliding friction into the rolling friction. So main bearing plays a vital role in the aircraft engine, it's operating temperature is $300^{\circ}C$, the value of *DN* (bearing diameter (mm) × speed (r/min)) is larger than 2.3×10^{6}. Meanwhile, as the engine bearing unit, aircraft engine main bearing also bear the static load, the axial concentricity caused by static and dynamic load, maneuvering load, steady vibration load, rotor hot bending, the compressor dynamic load, temperature load and so on.

As the key component of the aircraft engine, its advanced research and development has been concerned in home and abroad. Such as NASA of America, it began a "speed bearing research and development program'' in the 1959[2]. But we have a large gap in the research of main bearing of aircraft engine, especially in the design. In order to meet the needs of aircraft development, we should develop and apply advanced design and analysis techniques of bearing.

According to study of Chen[2],Tang[3] and Zhao[4], we take a in-depth research about the recent progress of the current finite element analysis method for fatigue life prediction in this paper.

Lot of study has been done about the life prediction of aircraft engine in the world. In the present paper, we take the deep groove ball bearings as an example, which mainly composed of outer ring, cage, rolling elements and inner ring. Since fatigue failure generally does not occur on the retainer, the bearing life is mainly decided by the life of the outer ring, inner ring and rolling elements. According to the Lundberg-Palmgren bearing life theory, the life of the bearing outer ring, inner ring and rolling elements are a series of random numbers subjected to Weibull distribution. So the life of bearing can be estimated through the life of the bearing outer ring, inner ring and a rolling element .The main work of the present paper shown as follows:

(1) In section 2, we introduced the basic theory and models for fatigue life prediction of the deep groove ball bearing, and analyzed the basic procedure of fatigue life prediction.

(2) Established parametric model and loaded the appropriate boundary conditions by using the static analysis module of ANSYS, then obtained the bearing stress and deformation under static load.

(3) Got the fatigue life of the deep groove ball bearing with fatigue module of ANSYS, and compared it with the theory value in the end, and proposed a series of measures to improve the bearing's life.

2. COMPUTATIONAL BASIS OF MAIN BEARING MODELING

Fatigue life of rolling bearings is when the rear bearing starts running, including any part on the inner and outer ring or rolling, before material fatigue caused by the damage, the total number of bearing rotation or a few hours. Contact fatigue is the major failure mode of rolling bearing, so fatigue life is an important indicator of bearing's design and application. The experience shows that the L-P model is simple and has a sufficient accuracy for the bearing life assessment. Thus we use it to calculate the fatigue life in this paper. For the particular bearing materials, the greater space of stress and more cycles can make the material fatigue failure probability increasing, therefore gives the following empirical formula:

$$\ln\frac{1}{S} \propto \frac{\tau_0^c N^e V}{Z_0^h} \tag{1}$$

Where τ_0 is the maximum alternating shear stress;

Z_0 is the maximum depth of alternating shear stress is located;

c, e, h are determined by the index data bearing test

V is the volume by stress.

Assume that the width of contact ellipse stress value 2a, depth Z_0, the raceway length is l, then we obtain

$$V \propto a Z_0 \tag{2}$$

Meanwhile, the number of stress cycles is proportional to the fatigue life, i.e. $N \propto L$

In the case of point contact, with the Hertz theory of elastic rolling contact, there exists

$$a \propto Q^{\frac{1}{3}} \quad (3) \qquad \tau_0 \propto Q^{\frac{1}{3}} \quad (4) \qquad Z_0 \propto Q^{\frac{1}{3}} \quad (5)$$

Take Eq. (3), (4), (5) into Eq.

$$\ln\frac{1}{S}\propto Q^{\frac{c-h+2}{3}}L^{e} \qquad (6)$$

Where Q is rolling load

When a certain bearing is given, S is a constant probability of survival, we can get:

$$Q^{\frac{c-h+2}{3}}L^{e}=\text{constant} \qquad (7)$$

If Q_c described as the rolling loads (i.e., rolling dynamic load rating) when L = 1 (i.e., running one million revolutions), then get:

$$Q^{\frac{c-h+2}{3}}L^{e}=Q_c^{\frac{c-h+2}{3}} \qquad (8) \qquad\qquad L=\left(\frac{Q_c}{Q}\right)^{\frac{c-h+2}{3e}} \qquad (9)$$

In case of line contact, we get

$$\tau_0\propto Q^{\frac{1}{2}} \qquad (10) \qquad\qquad Z_0\propto Q^{\frac{1}{2}} \qquad (11)$$

Considering that the roller's length is equal to a, we get

$$L=\left(\frac{Q_c}{Q}\right)^{\frac{c-h+2}{2e}} \qquad (12)$$

Take Eq. (10), (11) into the Eq. (12)

$$L=\left(\frac{Q_c}{Q}\right)^{P} \qquad (13)$$

Where for the point contact $P=(c-h+2)/3e$

for the line contact $P=(c-h+2)/2e$

Where P is determined by the experimental data. and P = 3, so

$$L=\left(\frac{Q_c}{Q}\right)^{3} \qquad (14)$$

Eq. (14) is the basic formula for fatigue life calculation of the deep groove ball bearing

The fatigue life of inner and outer rings can be described as

$$L_i=\left(\frac{Q_{ci}}{Q_i}\right)^{3} \qquad (15) \qquad\qquad L_e=\left(\frac{Q_{ce}}{Q_e}\right)^{3} \qquad (16)$$

Where Q_{ci} Q_{ce} are inner and outer rings of the dynamic load rating

Q_i Q_e are inner and outer rings of the contact load rating

The fatigue life L of the bearing is the intersection of the fatigue life of inner and outer ring.

$$L=\left[\sum_{j=1}^{N_b}\left(L_i^{-e_w}+L_e^{-e_w}\right)_j\right]^{-1/e_w},\left(j=1,2,\cdots,N_b\right) \qquad (17)$$

Where e_w is the Weibull slope and $e_w = 10/9$

In the Eq. (17), we only consider the fatigue damage of the inner and outer rings. In the actual project, rolling fatigue spall also accounted for a considerable proportion. Therefore, it is necessary to consider the impact of fatigue life for rolling element bearing.
Rolling elements with the inner and outer contact fatigue life can be described as

$$L_{bi} = \left(\frac{Q_{bi}}{Q_i}\right)^3 \quad (18) \qquad L_{be} = \left(\frac{Q_{be}}{Q_e}\right)^3 \quad (19)$$

When considering the impact of rolling elements, bearing fatigue life can be described as L

$$L = \left[\sum_{j=1}^{N_b}\left(L_i^{-e_w} + L_e^{-e_w} + L_{bi}^{-e_w} + L_{be}^{-e_w}\right)_j\right]^{-1/e_w}, (j = 1, 2, \cdots, N_{\mathrm{b}}) \quad (20)$$

Where L_i, L_e are the fatigue life of the inner and outer ring.

L_{bi}, L_{be} are the fatigue life of the rolling elements contact with the inner and outer ring.
The calculation process for fatigue life, is shown as Figure 1.

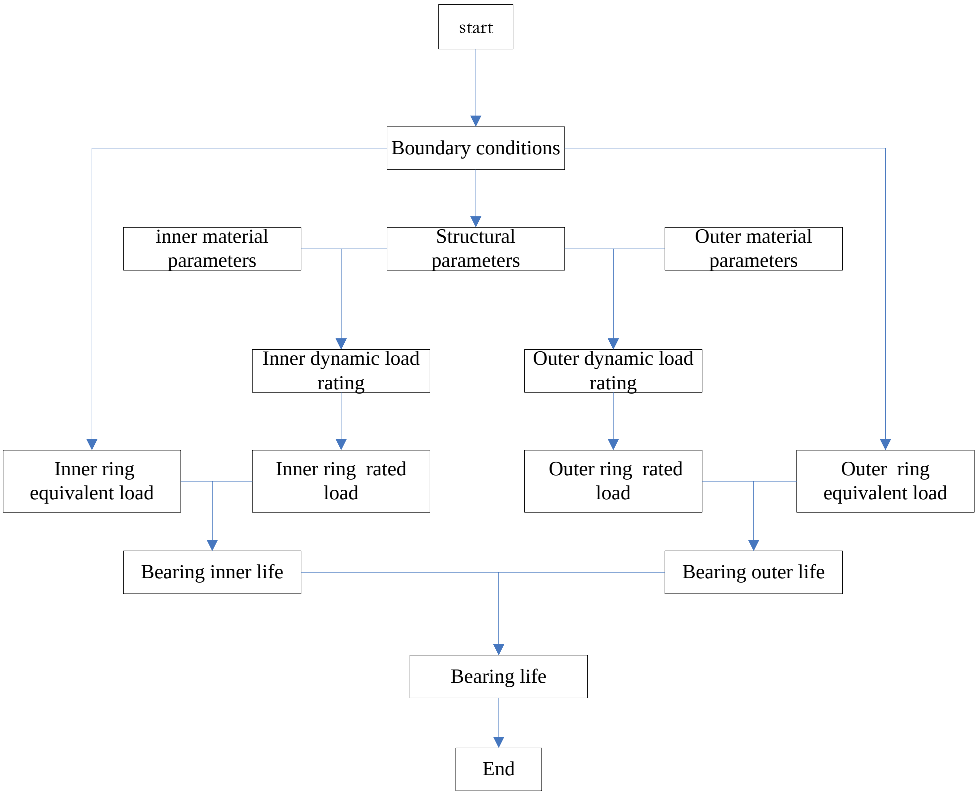

Figure 1 The calculation process of main bearing's fatigue life

3. CONSTRUCTION OF FINITE ELEMENT MODEL

Finite element method is a way that takes the continuous geometry into finite number units, which set a finite number of nodes in each unit. As the successive body is regarded as a set of units aggregates which connected only in node, we can convert continuous infinite degrees of freedom into a finite discrete-DOF domain, this way will greatly simplify the problem.

3.1. Descriptions of finite element models

1. A three-dimensional model

Using Pro/E establish the deep groove ball bearing 6217 3D model, the outer diameter D is 150mm, diameter d is 85mm, the thickness is 28mm. And import it into ANSYS workbench software modules. We can see Figure 2.

2. Definition of material properties

After importing the solid models of deep groove ball bearing successfully, we define material properties and choose the elastic modulus (2.07 × 1011Pa)and Poisson's ratio (0.3).

3. Meshing

Using free mesh, inner and outer rings of the grid cell size of 2mm, refining the grid of different levels to the contact area, the grid rolling contact area is more refined than the inner, outer ring raceway grid. We can see Figure 3.

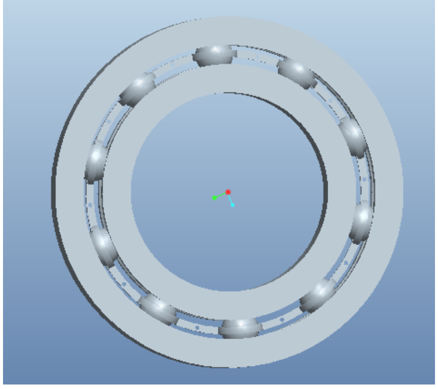

Figure 2 3D model

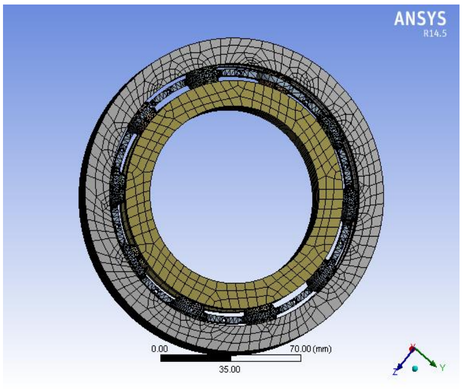

Figure 3 Finite element model

4. Contact pairs

To establish the contact between inner ring raceway, outer ring raceway and the rolling elements, determine the inner and outer ring raceway and rolling elements are flexible contact, the outer ring raceway surface and the inner ring raceway surface are the target surface, the rolling element surface as the contact surface. Contact element type as the node containing eight nodes quadrilateral element CONTA 174. Target surface contact element type as three-node which have no-middle-node unit TARGEl70, In each pick up on, set rolling with the inner and outer raceway friction coefficient MU = 0.003, normal contact stiffness factor FKN = 0.1, the initial near factor ICONT = 0.01. Each contact with the rolling element and inner and outer rings, create two contact pairs. There are 10 rolling elements, created 20 pairs of contacts right.

In order to ensure the accuracy of the calculation and reduce the amount of calculation, the occurrence of the contact area may use a denser meshing style; the model consists of 13,423 elements and 26,971 nodes, shown as Figure 3.

3.2. Determination of boundary condition

1. Impose boundary conditions

(1) Simulation bearing seat: Constraint all freedom degrees of the outer surface of the bearing outer

ring;
(2) Simulation the retainer: Constraint the circumferential (UY) and axial (UZ) displacement of all nodes in the neutral plane equator of rolling.
(3) Freedom of the inner surface of coupling load: The coupling of all nodes on the inner circle of the radial surface (UX) and circumferential (UY) degrees of freedom.
(4) Simulation flanges: Constraint the UZ displacement of the side of the bearing ring body.
(5) Apply radial load: Applying a radial node radial force is applied in the form of the average radial force to each node on a straight line, can be directly loaded in the direction UX.
(6) Applying a centrifugal force: Bearing rotary inertia force in ANSYS structural static analysis can be used to simulate the dynamic conditions imposed by way of inertial load
1) Add “Rotational Velocity”, the “Define By” option into “Component”.
2) The actual angular velocity is applied in the global Cartesian Z-axis.
2. Solving set
(1) Time-step control: “Analysis Settings” under “Static Structural”, the “Number of Steps” to step 8.
(2) Analysis of control: Contact analysis due to the large local strain, we choose “large displacement static”, and open the linear search.

4. APPLICATION OF LIFE PREDICTION FOR MAIN BEARINGS

4.1. Finite element analysis

Considering the impact of the number of rolling elements, the rolling element diameter and groove curvature coefficient, to get result of the contact stress before the finite element analysis of bearing life. Then analysis the various factors on the bearing contact stress and deformation, study for the next step various factors affecting bearing life and improve bearing life measures provide an important basis for the analysis and the analytical results more reasonable.
We get the conclusion as following：
(1) The maximum equivalent stress and equivalent strain occurred in the radial force at the bottom line is contact point, that is mean bearing the risk of fatigue damage part is the rolling element and raceway contact points. And the shape of the contact area is approximately ellipse.
(2) Maximum rolling contact stress occurs in the radial force beneath line where rolling contact with the raceway.
(3) Overall radial displacement occurs where the inner ring bear the radial load.

4.1.1 Results and Its Analysis

Figure 4 shows the effect of the number of rolling element bearings for bearing contact stress and deformation curves. The diameter of the rolling element bearings can be seen D_w = 16.5, the inner and outer groove curvature coefficients were $f_i = 0.505$ and $f_e = 0.51$.
Figure 5 shows the effect of the diameter of rolling element bearings for bearing contact stress and deformation curves. As the number of its rolling is 10, the inner and outer groove the curvature coefficients were $f_i = 0.505$ and $f_e = 0.51$.
Figure 6 shows the effect of the inner and outer groove curvature coefficient of rolling bearings for

bearing contact stress and deformation curves. The number of rolling bearings n = 10, the diameter of the rolling element bearing D_w = 16.5mm, difference in coefficient of curvature of the inner and outer groove $f_i - f_e = 0.005$.

According to the figures and data we obtained above, we can know the results as follows:

1. After getting a deep groove ball bearings, we know that the outer contact area is approximately elliptical, the contact area of the contact stress distribution; the inner and outer race on contact stress will decrease when the number of rolling increase, the inner and outer race on contact stress will decrease when the rolling diameter increase, the inner and outer race on contact stress decrease when inner and outer curvature coefficient is $f_i \le 0.51$, $f_e \le 0.515$;and its will increase when $f_i \ge 0.51$, $f_e \ge 0.515$.

2. The radial deformation will decrease when the number of rolling increase and the rolling diameter increase, but it will increase when the inner and outer curvature coefficient increase.

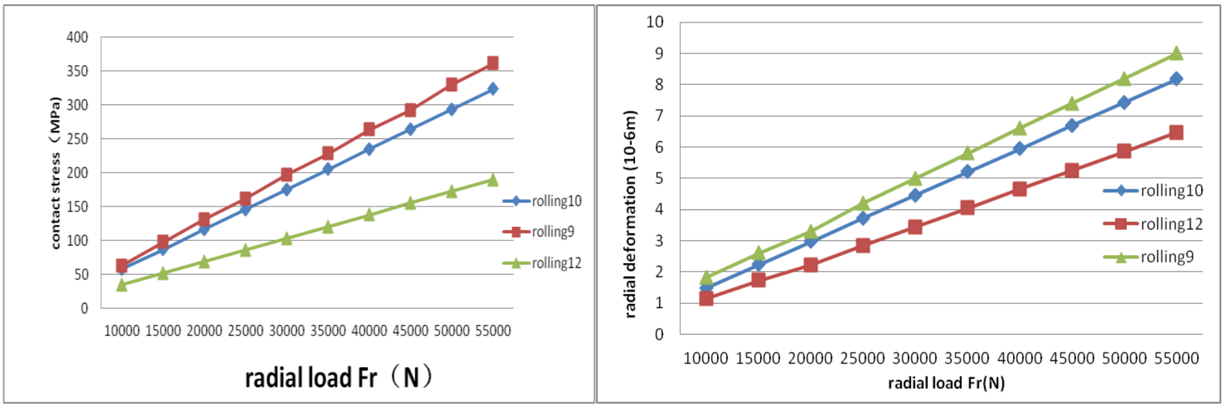

Figure 4 The impact under the change of the number of rolling elements

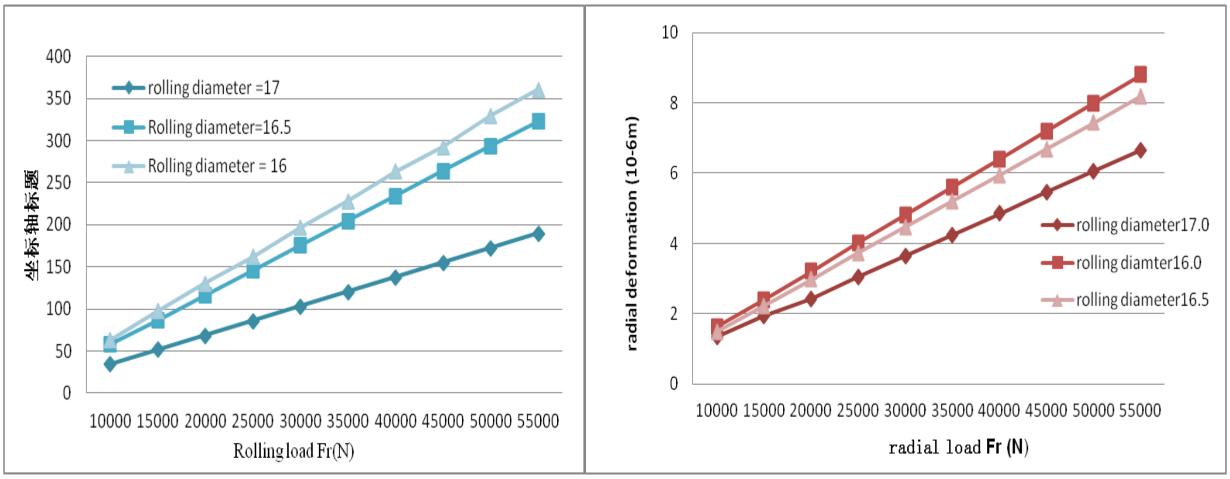

Figure 5 The impact of the chang of the rolling diameter

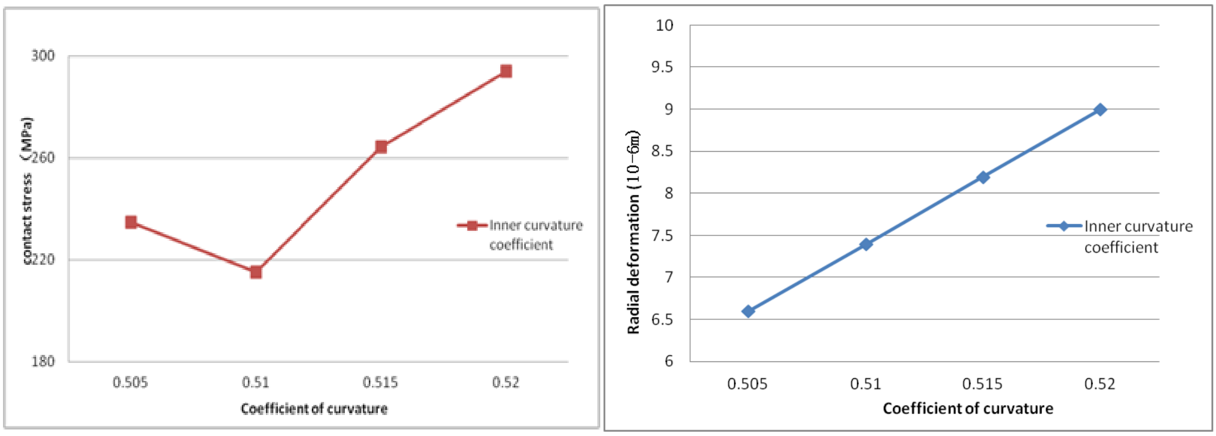

Figure 6 The change of the curvature coefficient of inner and outer race

4.2. Fatigue life analysis

In terms of numerical simulation of the fatigue life, the article will consider the effect of the number of rolling of bearings, rolling diameter and curvature coefficient of inner and outer race and other factors, and use the nominal stress method for fatigue life of deep groove ball beating through numerical simulation. With the help of ANSYS software fatigue analysis module, the fatigue life of bearing is simulated, and the influence discipline of various factors on the beating fatigue life is obtained, the results with the theoretical fatigue life of L-P results are compared, and discuss difference reason between the two results; finally it propose the technical measures to improve bearing life. We also get the diagram which describes the changes of the fatigue life of the bearing.

The results show that the fatigue dangerous parts of deep groove ball bearings appears at the contact point of between rolling units and raceway; bearings fatigue life will increase when the number of rolling increase, fatigue life will increase when rolling diameter increase, fatigue life will increase when inner and outer curvature coefficient is $f_i \le 0.51$, $f_e \le 0.515$, and it will decrease when $f_i \ge 0.51$, $f_e \ge 0.515$. Simulation method of fatigue life for deep groove ball bearing can calculate the complex working conditions of bearing and the fatigue life prediction has a high reliability.

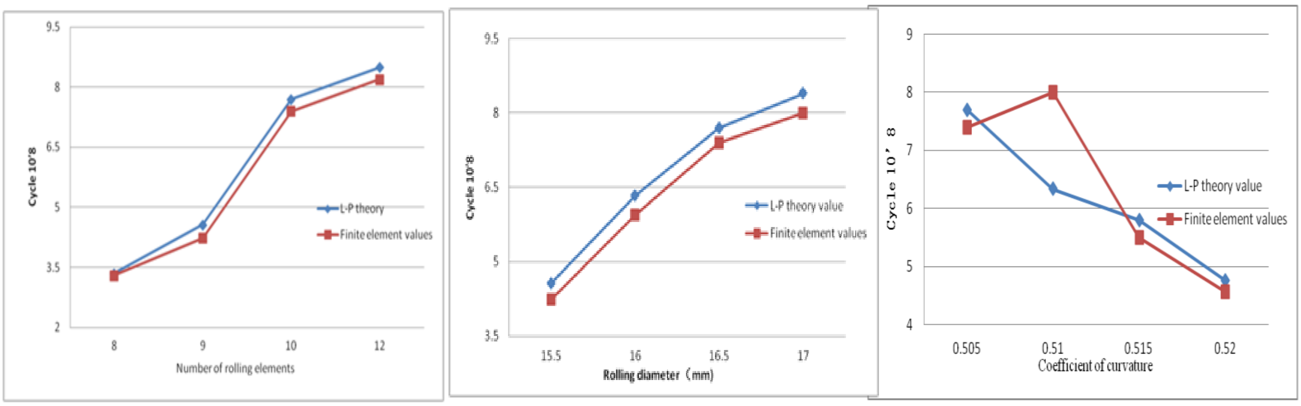

Figure 7 the change of the fatigue life

4.3. Measures to improve the fatigue life for bearings

Based on the analysis above and references [8]-[12], in order to improve the fatigue life of deep groove ball bearings, the following steps can be used in the design.

1. An increase of the number of rolling elements

When increase the number of rolling elements, each rolling element exposed the radial load is decrease, the contact load for each rolling element is also smaller, so fatigue life of improve.

2. An increase of the diameter of the rolling

Rolling diameter increases, the equivalent bearing structure increases bearing capacity increases, so the rolling element bearing life increases with the diameter increased.

3. Reasonable assignment of rolling element and raceway contact parameters.

For deep groove ball bearings, inner groove curvature coefficient $f_i \le 0.52$, outer groove curvature coefficient of the bearing $f_e \le 0.53$. It should also be noted between f_i and f_e matches.

In addition, you can also reduce the pitch diameter of the bearing, the bearing does not slip under the premise of guaranteed minimize preload, at high speed, rolling ceramic material instead of steel materials.

5. CONCLUSION

1. The results indicate that the outer contact area of the deep groove ball bearings is approximately elliptical, also the contact stress distribution of the contact area is obtained. Besides, we also get the conclusion that the inner and outer race on contact stress will decrease when the number of rolling increase, the inner and outer race on contact stress will decrease when the rolling diameter increase, the inner and outer race on contact stress decrease when inner and outer curvature coefficient is $f_i \le 0.51$, $f_e \le 0.515$, and it will increase when $f_i \ge 0.51$, $f_e \ge 0.515$.

2. The radial deformation will decrease when the rolling number and the rolling diameter increase, but it will increase when the inner and outer curvature coefficient increase. The results of the simulation of the fatigue life indicate that the fatigue dangerous parts of deep groove ball bearings appears at the contact point of between rolling units and raceway, bearings fatigue life will increase when the rolling number increase, fatigue life will increase when rolling diameter increase, fatigue life will increase when inner and outer curvature coefficient is $f_i \le 0.51$, $f_e \le 0.515$, and it will decrease when $f_i \ge 0.51$, $f_e \ge 0.515$.The simulation method of fatigue life for deep groove ball bearing can simulate the complex working conditions of bearing and has a high reliability for fatigue life prediction.

In order to shorten the test cycle, reduce cost and improve the accuracy of life prediction, the numerical simulation method used to simulate the fatigue life of the bearing calculation in present paper is very effective.

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