RESIDUAL LIFE PREDICTION FOR THE SPINDLE OF RETIRED MACHINE BASED ON FATIGUE DAMAGE MECHANISM

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This paper studies the fatigue damage mechanism of the spindle of the retired machine. Firstly, the mathematical model of the spindle-bearing of the retired machine is established, and the corresponding stress, displacement and velocity are calculated. Secondly, Roman is used to establish the model on the spindle of the retired machine tool; the static and dynamic analysis is carried out and the corresponding simulation results with stress-strain are obtained. Thirdly, the analysis data is introduced to the fatigue analysis software for the residual life prediction. By loading the spindle of the retired machine, the cloud picture of its life and the corresponding fatigue is achieved. Finally, the life computation formula and data are used to calculate the residual life of the spindle.

Keywords: Fatigue Damage Mechanism; Spindle; Residual Life; Retired Machine;

1. Introduction

Formation of the fatigue life prediction theory can reduce the waste of the residual value in the process of old product management. It can also prolong the life cycle of products and save the costs of the production industry. Thereby, life prediction has an important significance on the national economy and defense construction. Wang Yanrong developed a notched life prediction method considering the influence of the gradient, which depended on the stress distribution of the sample notch root area and considered the influence of the average stress, stress gradient and the size effect [1]. Liu Jinna et al. analyzed quantitatively the fatigue performance of film to nanoscale dynamic load method, including continuous stiffness method and nano impact test method [2]. Wen-bin Shangguan established three life prediction models of the rubber materials taking strain energy density, maximum principal Green-Lagrange strain and the effective stress as the fatigue damage parameter respectively[3,4]. Yang Maosheng et al.

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through micro fatigue damage mechanism analysis and from micro fatigue contact stress calculation established the finite-element global model and sub-model with a common cylinder/plane micro contact fatigue structure in the key parts of aviation equipment [5]. Dr. Zhang Guoqing studied the mechanical properties and fatigue behavior of the machine tool remanufacturing and estimated the life of the typical parts before and after the remanufacturing [6].

From the above, there are many methods for fatigue life prediction; however, there are few life predictions on the key parts of the retired machine. For the machine remanufacturing, they are only limited to the analysis of the whole machine, and most of them are based on probability statistics without certainty theory. These researches cannot reflect the essence of the fatigue failure process and even helpless due to little or a lack of experimental data. The paper study from the fatigue damage mechanism and takes a retired machine as an example to establish a new set of spindle residual life prediction method for the retired machine. A train of thought is provided for the prediction of the residual life prediction of other machine parts.

2. The calculation model on the finite-element analysis of the bearing fatigue damage

Fatigue life prediction needs to know the change of load, physical dimension of the structure, and the involved material performance parameter or curve. Finite element is used for fatigue life prediction; one is according to the load stress and physical dimension to calculate stress-strain, the dynamic stress strain response is obtained, and then combine with the material performance parameter, different life prediction of the fatigue damage model can be applied, calculation model is shown in Fig.1.

3. Solution model on the stiffness of the angular contact ball bearing

The angular contact ball bearing as the critical support part in the spindle system of the retired milling machine bears and delivers the axial force and radial force at the same time, and its rotation accuracy, stiffness and pre-compact directly restricts the service performance of the spindle.

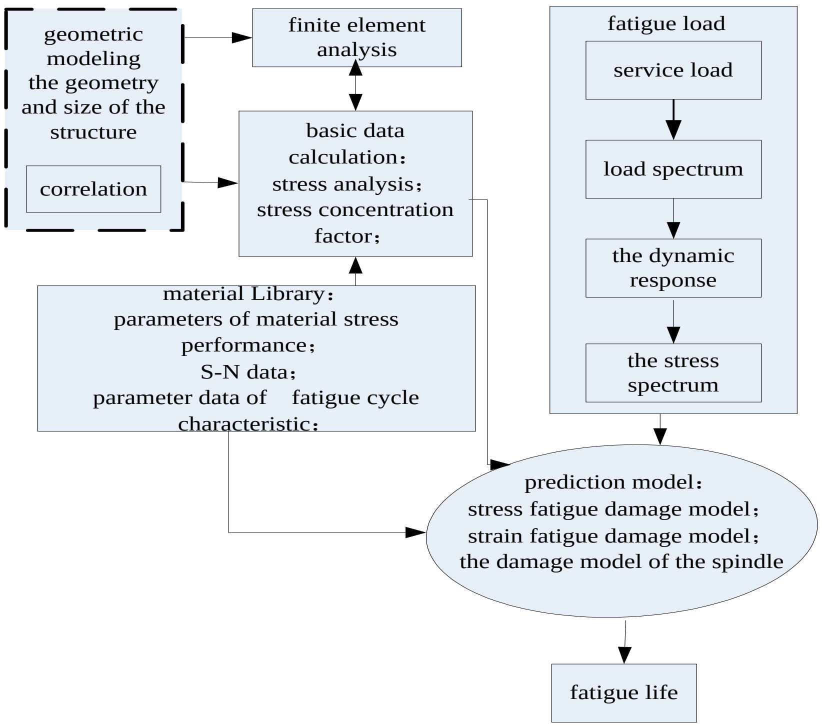


Fig.1 the model of fatigue damage mechanism based on finite element analysis

3.1 Geometric relationship analysis

The radial clearance P_d and the axial clearance P_e of the bearing under the initial status are respectively,

$$P_d = d_o - d_i - 2D_b; \qquad P_e = 2A\sin\alpha \tag{1}$$

Where, d_i is the circular diameter/mm of the gully bottom at the inner ring of the bearing; d_o is the circular diameter/mm of the gully bottom at the outer ring of the bearing; D_b is the diameter of the ball; P_d is the initial radial clearance/mm of the bearing.

Under the action of high-speed rotation and the external load of the bearing, angular contact ball bearings are looked as a unit consisted of an inner and outer ring node. The movement of each node contains five directions of freedom (three translational degrees of freedom δ_x, δ_y and δ_z; two rotational degrees of freedom θ_y and θ_z), assuming the corresponding displacement vector of the inner ring of the angular contact ball bearing is $\Delta\delta = \{\Delta\delta_x, \Delta\delta_y, \Delta\delta_z, \Delta\theta_y, \Delta\theta_z\}$. After the bearing load balance, the corresponding displacement variation of the groove curvature in the inside race is as follows,

$$u_j = \Delta\delta_x + \Delta\theta_z \cdot R_i \cdot \cos\psi_j + \Delta\theta_y \cdot R_i \cdot \sin\psi_j \qquad (2)$$
$$v_j = \Delta\delta_y \cdot \cos\psi_j + \Delta\delta_z \cdot \sin\psi_j$$

It can be known in the above, the distances of the groove curvature centre in the inner and outer ring at the axial direction and radial direction are respectively as follows,

$A_j = BD \cdot \sin\alpha^\circ + u_j$; $A_{2j} = BD \cdot \cos\alpha^\circ + v_j$

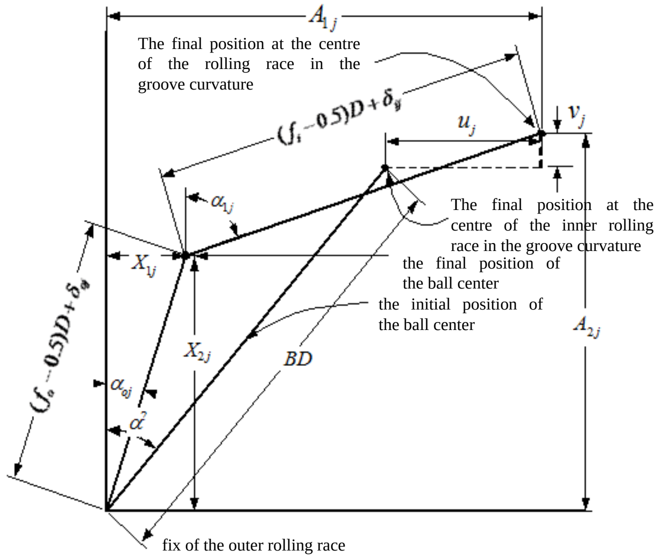


Fig.2 The geometrical relationship between groove curvature center and position change of the sphere centre

Parameter X_1 and X_2 are introduced, from the geometrical relationship in Fig.2, it can be obtained at j ,

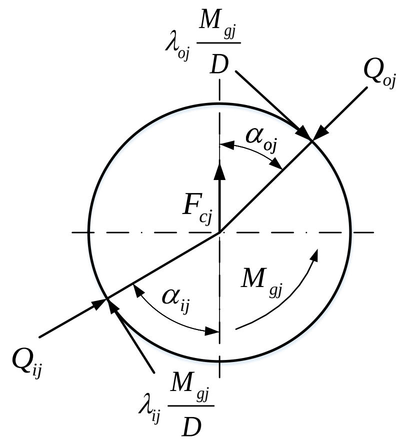
$$\cos\alpha_{ij} = \frac{A_{2j} - X_{2j}}{(f_i - 0.5)D + \delta_{ij}}; \quad \sin\alpha_{ij} = \frac{A_{1j} - X_{1j}}{(f_i - 0.5)D + \delta_{ij}} \qquad (3)$$
$$\cos\alpha_{oj} = \frac{X_{2j}}{(f_o - 0.5)D + \delta_{oj}}; \quad \sin\alpha_{oj} = \frac{X_{1j}}{(f_o - 0.5)D + \delta_{oj}}$$

The geometric consistency equation of bearing deformation is as follows according to the Pythagorean Theorem,

$$\begin{cases}(A_{1j}-X_{1j})^2+(A_{2j}-X_{2j})^2-L_{ij}^2=0\\ X_{ij}^2+X_{2j}^2-L_{oj}^2=0\end{cases} \tag{4}$$

3.2 Force equilibrium equation

The stress situation of the ball j in the plane of its centre and the bearing axis is shown in Fig.3.



α_{ij} is the contact angel /°between the ball and the inner ring of the bearing;

α_{oj} is the contact angel /°between the ball and the outer ring of the bearing.

Q_{ij} is the contact force/N between the ball and the inner ring of the bearing;

Q_{oj} /N is the contact force between the ball and the outer ring of the bearing.

F_{cj} is the centrifugal force/N of the ball and

M_{gj} is the gyroscopic couple/N·m of the ball.

Fig.3 Load of the ball j

According to Hertz contact theory, contact load and contact deformation have the following relationship [7, 8]

$$\begin{cases}Q_{ij}=K_i\delta_{ij}^{3/2}\\ Q_{oj}=K_o\delta_{oj}^{3/2}\end{cases} \tag{5}$$

Where, K_i is the contact force constant between the ball and the inner ring; K_o is the contact force constant between the ball and the outer ring.

From Fig3, the force equilibrium equation at the horizontal and vertical direction of the ball j is,

$$\begin{cases} Q_{ij}\sin\alpha_{ij} - Q_{oj}\sin\alpha_{oj} - \dfrac{M_{gj}}{D_b}(\lambda_{ij}\cos\alpha_{ij} - \lambda_{oj}\cos\alpha_{oj}) = 0 \\ Q_{ij}\cos\alpha_{ij} - Q_{oj}\cos\alpha_{oj} + \dfrac{M_{gj}}{D_b}(\lambda_{ij}\sin\alpha_{ij} - \lambda_{oj}\sin\alpha_{oj}) + F_{cj} = 0 \end{cases} \tag{6}$$

Where, the centrifugal force and gyroscopic couple of the inertial load supported by the angular contact ball bearing is respectively,

$$F_{cj} = \frac{1}{2} m d_m \omega^2 (\frac{\omega_m}{\omega})^2 \tag{7}$$

$$M_{gj} = J\omega^2 (\frac{\omega_R}{\omega})_j (\frac{\omega_m}{\omega})_j \sin\beta_j \tag{8}$$

Where, J is the rotational inertia /kg·m^2 of the ball; ω is the rotational angel velocity/rad·s^{-1}of the inner ring or rotor; ω_m is the revolving angle velocity /rad·s^{-1} of the ball; β_j is the attitude angle/° of the ball; m is the quality/kg of the ball; ω_R is the spin velocity of the ball.

All the contact force between the ball and the inner ring is for superposition, the resultant force on the inner ring can be obtained [9, 10]

$$\begin{cases} F_{xi} = \sum_{k=1}^{N} (Q_{ik}\sin\theta_{ik} + \dfrac{M_{gk}}{D}\cos\theta_{ik}) \\ F_{yi} = \sum_{k=1}^{N} (Q_{ik}\cos\theta_{ik} - \dfrac{M_{gk}}{D}\sin\theta_{ik})\cos\varphi_k \\ F_{zi} = \sum_{k=1}^{N} (Q_{ik}\cos\theta_{ik} - \dfrac{M_{gk}}{D}\sin\theta_{ik})\sin\varphi_k \\ M_{yi} = \sum_{k=1}^{N} \{R_i(Q_{ik}\sin\theta_{ik} + \dfrac{M_{gk}}{D}\cos\theta_{ik}) - f_i M_{gk}\}\sin\varphi_k \\ M_{zi} = -\sum_{k=1}^{N} \{R_i(Q_{ik}\sin\theta_{ik} + \dfrac{M_{gk}}{D}\cos\theta_{ik}) - f_i M_{gk}\}\cos\varphi_k \end{cases} \tag{9}$$

In a similar way, the resultant force bearing on the outer ring of the bearing is as follows,

$$\begin{cases} F_{xo} = \sum_{k=1}^{N}(Q_{ok}\sin\theta_{ok} + \frac{M_{gk}}{D}\cos\theta_{ok}) \\ F_{yo} = \sum_{k=1}^{N}(-Q_{ok}\cos\theta_{ok} + \frac{M_{gk}}{D}\sin\theta_{ok})\cos\varphi_k \\ F_{zi} = \sum_{k=1}^{N}(-Q_{ok}\cos\theta_{ok} + \frac{M_{gk}}{D}\sin\theta_{ok})\sin\varphi_k \\ M_{yo} = -\sum_{k=1}^{N}\{R_o(Q_{ok}\sin\theta_{ik} + \frac{M_{gk}}{D}\cos\theta_{ok}) + f_o M_{gk}\}\sin\varphi_k \\ M_{zo} = -\sum_{k=1}^{N}\{R_o(Q_{ok}\sin\theta_{ok} + \frac{M_{gk}}{D}\cos\theta_{ok}) + f_o M_{gk}\}\cos\varphi_k \end{cases} \quad (10)$$

The stiffness matrix of the bearing can be obtained through the partial derivative of the displacement.

$$K_B = \frac{\partial F_i}{\partial q_i} = \frac{\partial F_o}{\partial q_o} = \begin{bmatrix} k_{xx} & k_{xy} & k_{xz} & k_{x\theta_y} & k_{x\theta_z} \\ k_{yx} & k_{yy} & k_{yz} & k_{y\theta_y} & k_{y\theta_z} \\ k_{zx} & k_{zy} & k_{zz} & k_{z\theta_y} & k_{z\theta_z} \\ k_{\theta_y x} & k_{\theta_y y} & k_{\theta_y z} & k_{\theta_y\theta_y} & k_{\theta_y\theta_z} \\ k_{\theta_z x} & k_{\theta_z y} & k_{\theta_z z} & k_{\theta_z\theta_y} & k_{\theta_z\theta_z} \end{bmatrix} \quad (11)$$

4. System integration model of the spindle-bearing

Based on the model of spindle, dial and bearing, the motion equation of the whole spindle system can be achieved,

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\} \quad (12)$$

Where, system quality matrix $[M]=[M^b]+[M^d]$, system damping matrix $[C]=[C^s]-\Omega[G^b]-\Omega[G^d]$. And where, $[C^s]$ is the structural damping, which can be obtained by experimental modal analysis; $\{F\}$ is the external force vector of the system and $\{F\}=\{F^b\}+\{F^d\}$; For the nonlinear feature of the bearing stiffness, its size is affected by the system displacement $\{x\}$; on the contrary, the system displacement $\{x\}$ is influenced by the system stiffness and the external force. The interdependent relationship of the system stiffness and displacement is the source of the nonlinear feature of the spindle-bearing structure [11, 12].

We consider the external force of the spindle system is static; that is, when $\{F\}=\{F\}_0$, the bearing stiffness is solved. And then, Newton - Ralph iterative equation is,

$$[M]\left\{\ddot{x}\right\}^i + [C]\left\{\dot{x}\right\}^i + [K]^i\{x\}^i = \{F\}_0 \tag{13}$$

$$[K]^i = \left[K^b\right] + \left[K^b\right]_P + [K]_B^i - \Omega^2\left[M^b\right]_C \tag{14}$$

$$\{x\}^i = \{x\}^{i-1} + \{\Delta x\}^i \tag{15}$$

Where, i is the iterations.

We define the initial stiffness of the bearing is $[K]_B^0$, and the initial stiffness of the spindle system is,

$$[K]^0 = \left[K^b\right] + \left[K^b\right]_P + [K]_B^0 - \Omega^2\left[M^b\right]_C \tag{16}$$

Under the static external force, the system velocity vector $\left\{\dot{x}\right\}^i$ and acceleration vector $\left\{\ddot{x}\right\}^i$ are both zero, and the initial displacement vector of the system is,

$$\{x\}^0 = \left([K]^0\right)^{-1}\{F\}^0 \tag{17}$$

When the iteration is calculated to the i, from the previous step $\{x\}^{i-1}$ to call the quasi statics force model of the bearing, and the bearing stiffness $[K]_B^i$ can be calculated; and then it is introduced into Eq.(13) for system stiffness matrix $[K]^i$ updating; thereby, the residual unbalanced force is as follows.

$$\{R\}^i = \{F\}_0 - [K]^i\{x\}^{i-1} \tag{18}$$

From oHooke's law, the displacement increment caused by the residual unbalanced force is as follows,

$$\{\Delta x\}^i = ([K]^i)^{-1}\{R\}^i \tag{19}$$

In the following, call the equation to achieve the displacement vector of the step i, and then it is used for the calculation on the bearing stiffness in the step $i+1$.

We definite the imbalance energy in step i as,

$$\Delta E^i = \left(\{R\}^i\right)^T\{\Delta x\}^i \tag{20}$$

Eq. (17) is taken as the termination criterion for iteration; when ΔE^i is less than the set threshold value ε, it can be thought for iterative convergence. Under the action of external force $\{F(t)\}$, the system response and the bearing stiffness make dynamic changes over time. $\{F(t)\}$ is divided in a time interval, for the time node $t+\Delta t$, the motion equation of the system can be wrote as,

$$[M]\{\ddot{x}\}_{t+\Delta t}+[C]\{\dot{x}\}_{t+\Delta t}+[K]_{t+\Delta t}\{x\}_{t+\Delta t}=\{F\}_{t+\Delta t} \tag{21}$$

When $t=0$, the displacement, velocity and acceleration response of the system are all zero, and taking this as the initial condition of iteration. The dynamic response of the system can be obtained approximately when $t+\Delta t$ through the integral method, that is, the displacement $\{x\}_{t+\Delta t}$, velocity $\{\dot{x}\}_{t+\Delta t}$ and the accelerated velocity $\{\ddot{x}\}_{t+\Delta t}$. At each time node, $\{F\}_{t+\Delta t}$ can be regarded as a static force and the iterative equation in the above is used for solution [13, 14].

5. Numeral calculation of the machine spindle

Romax Designer is used for establishment and model analysis to the spindle system, calculating the critical velocity spectrum, the natural frequency spectrum of the damping and the unbalanced response analysis.

5.1 static analysis of the spindle

The external load conditions should be determined first when the static analysis of the spindle is made. Using Romax software to obtain the x after the spindle receiving the external load, y direction displacement, radial displacement, and the axial displacement; results of the internal tensile stress and bending stress of the spindle is as follows,

Table.1

The external load parameters of the milling machine

Rotation speed (rmp)	Power(Kw)	torque (Nm)
5500	17.56	305
Fx(N)	Fy(N)	Fz(N)
900	2200	2400

According to the external load of the spindle system under the working conditions of Table.1, making the static analysis for the spindle system, the deformation of the spindle system obtained is shown in Fig.4. It can be seen through the analysis on the radial displacement diagram of the spindle, due to the large load of the spindle end, and the displacement at the spindle end is larger. In the scope of 400 mm to 650 mm, due to weak support, the radial displacement is relatively large.

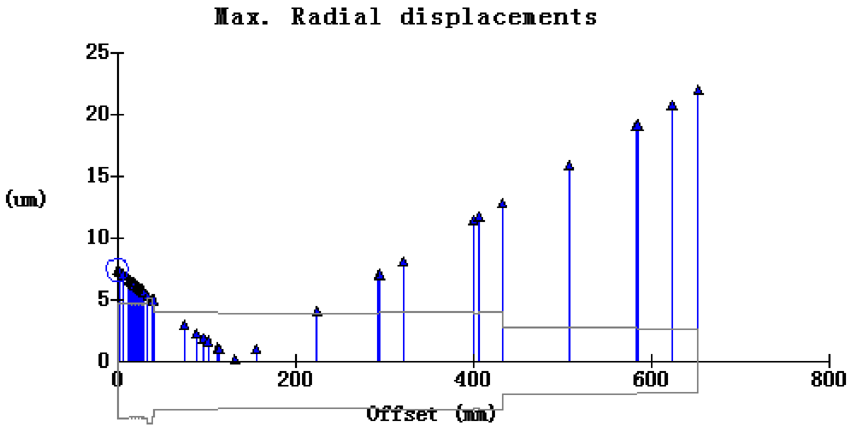


Fig.4 The radial deformation displacement of the spindle with milling

In the process of spindle revolving, the torque and longitudinal stress of the spindle become little and the effect on the performance is also little, thus, we mainly consider the bending stress in here. It can be known from Fig.5, the bending stress at the spindle end is the largest in the front bearing. The internal stress of the entire spindle is under 2Mpa basically; therefore, the fatigue life of the spindle is the longest.

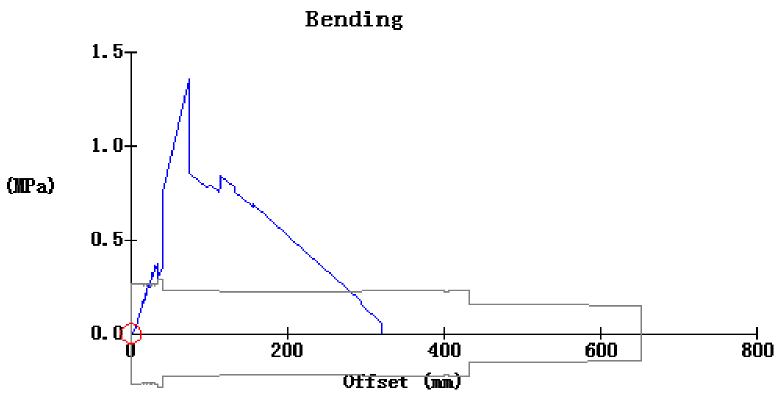


Fig.5 Bending stress distribution of the spindle with milling

5.2 The dynamic analysis of the spindle

Romax is used for the modal analysis to the spindle system; the modal vibration of the spindle system is obtained as shown in Fig.6 and Fig.7. Fig.8 is the cloud picture of the absolute maximum stress method of the spindle analyzed by ABAQUS.

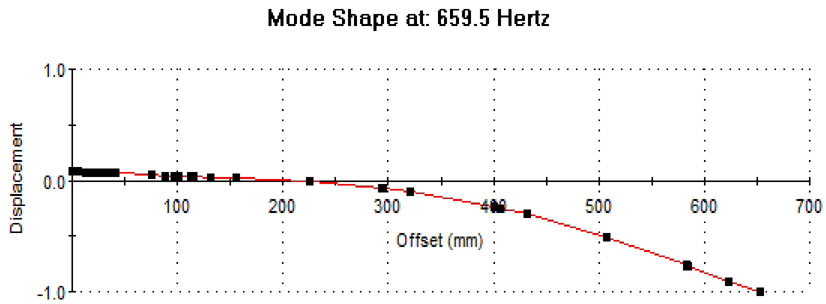


Fig.6 vibration mode of the first-order

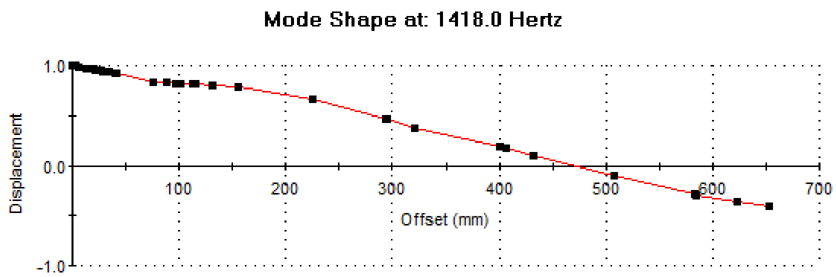


Fig.7vibration mode of the second-order

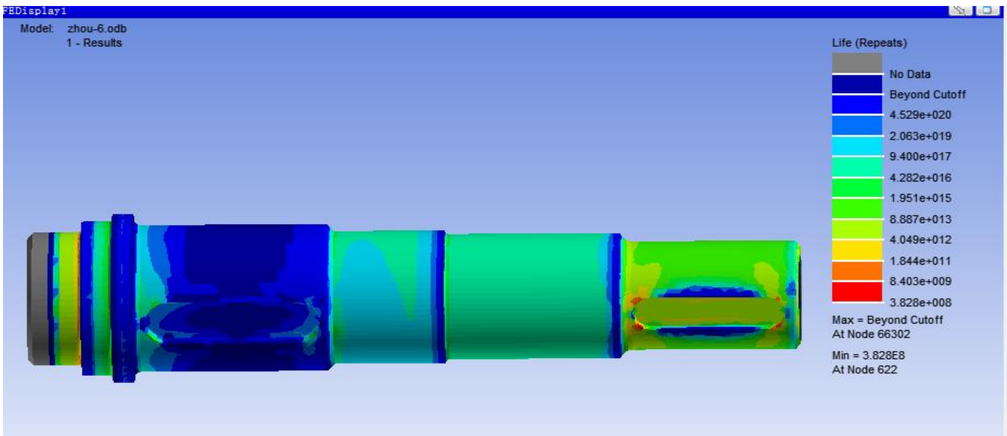


Fig.8 Cloud picture of Life (the absolute maximum principal stress analysis method)

It can be seen from the Fig. 8 that the higher is the frequency, the greater is the offset, and the deformation of the spindle is also larger. The maximum principal stress AbsMaxPrincipal is analyzed, the analysis result is taken as the fatigue life of the spindle, which are 8.403×10^{9} times. Spindle speed by 8500r / min, working 12 hours a day, working 300 days a year, the remaining life is 4.58 years.

6. Experimental results

The spindle is tested by the shaft parts fatigue tester; the experimental results are as follows table 2.

Table 2

Spindle residual life detection

sample	Residual life（yearh）	Loading force（N）
1	4.03	1563
2	5.61	1563
3	5.89	1563
4	4.70	1563
5	4.45	1563

7. Conclusion

The paper studies the spindle of a retired milling machine through finite element and fatigue analysis. The following conclusions are achieved:

(1) The paper proposes a mathematical modeling theory method analyzing the spindle-bearing of the spindle. According to the mathematical modeling on the spindle-bearing, the corresponding stress, displacement and speed are calculated to provide corresponding parameters for the prediction of the maximum stress life on the spindle.

(2) Romax and ABAQUS are used to establish finite element model on the milling machine spindle, and the static analysis is carried out on it to achieve the results of the corresponding stress strain simulation.

(3) A residual life prediction method is proposed on a retired milling machine. The date in the above is introduced into the fatigue analysis software. For the load on the spindle of the retired milling machine, the corresponding cloud pictures and damage picture of the life are obtained. Thereby, the residual life of the spindle is achieved.

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