International Journal of Engineering & Technology
CENTRE OF PROFESSIONAL RESEARCH PUBLICATIONS
ISSN 2049-3444
International Journal of Engineering and Technology Volume 6 No.11, November, 2016

A New Model Based Machine Tool Spindle Bearing Preload Optimization Method

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ABSTRACT

Spindle is a core component of machine tool. Similarly, the bearings are the significant assembly of the spindle. Bearing preload optimization is beneficial to improve the dynamic characteristic of the spindle and the machine tool working accuracy. This paper presents the relationship between the bearing preload and bearing fatigue life, temperature rise, giving out a new method for bearing preload optimization.

Keywords*: Spindle; Bearing; Preload; Optimization; Method*.

1. INTRODUCTION

As the most important component of the machine tool, spindle has critical influence on the machine tool working accuracy [1]. Generally, the convectional spindle consists of shaft, bearing and housing. The bearings are the most complex thing in the system, the bearing properties, such as fatigue life and temperature rise, are closely related to the spindle dynamic characteristic. And bearing preload is the fatal factor of bearing properties. It is necessary to search bearing preload optimum value.

Bearing preload is to use appropriate methods to make a certain pre-deformation and ensure that the bearing inner and outer rings are in a compressed state, the bearing running with a negative clearance between the inner and outer ring. Bearing preload has a significant impact on bearing contact loads, gyroscopic torque, dynamic stiffness and bearing fatigue life, if get a proper preload will help to control bearing temperature rise, improve spindle rigidity, avoid gyro rotation and extend bearing fatigue life.

Bearing preload optimization has been studied for decades. Initially, the scholars measured the bearing temperature directly,such as Jiang[2], Tomas Holkupl[3] etc, and based on the data to adjust bearing preload. Afterwords the scholars used the spindle rigidity, such as S.A. Spiewak and T.Nickel. They proposed to complete measuring spindle stiffness with the exciter, acceleration sensors, and estimated the spindle preload [4]. And then ball skidding, such as Tao Xu, et al. They completed the analysis of the optimum preload by ball skidding condition and obtained limited bearing temperature rise when the spindle speed is less than 10000rpm[5]. In addition, Jenq-Shyong Chen and Kwan-Wen Chen did the experiment, when the spindle speed over 15000rpm, there is a serious skidding phenomenon with small bearing preload[6]. And also bearing fatigue life, such as Shuyun Jiang et al[4]. They completed the optimization of preload at low spindle speed by bearing fatigue life, calculated the bearing life with different preload under different spindle speed, found there is a hundred times gap, and the the bearing preload influence on the bearing is obvious. Young-Kug Hwang et al.[7] proposed bearing preload optimization methods according to the spindle stiffness and the bearing fatigue life . Some researchers used spindle speed to optimize the bearing preload, directly explored the relationship between the spindle speed and preload curves by measuring spindle performance [8-13]. As can be seen above, the bearing preload optimization ways have no unchangeable standard.

This paper presents the relationship between the bearing preload and bearing fatigue life, temperature rise, giving out a new method for bearing preload optimization. The new method can be used to set the preload value in the global speed range of the convectional spindle.

2. EXPERIMENT SETUP

For the bearing preload research, this paper used a designed traditional spindle. The specific structure is shown in Fig.1 and Fig.2.

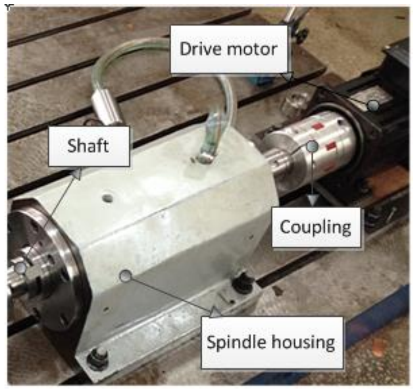

Fig.1 The specific structure of the spindle

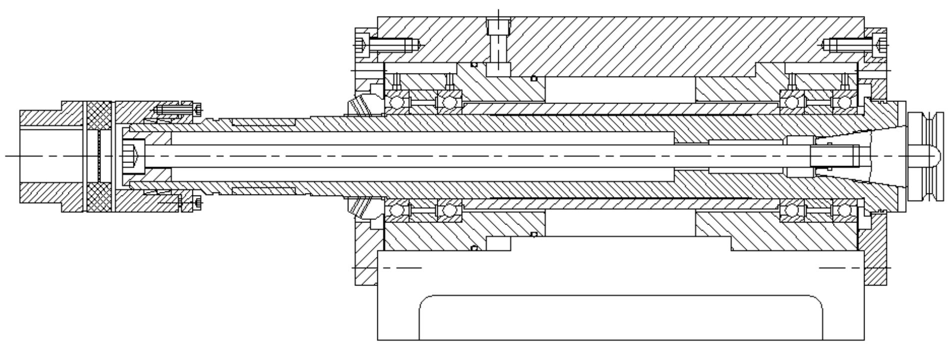

Fig.2 Section view of the spindle

From the figures, there are four same bearing in the spindle system, the shaft has approximately diameter, which is beneficial to simply the structure for simulation.

3. BERING PRELOAD AND BEARING FATIGUE LIFE, TEMPERATURE RISE RELATIONSHIP ANALYSIS

3.1. Bearing preload and bearing fatigue life analysis

For 7012AC bearing, the specific parameters are shown in Table 1.

Table 1 The bearing specific parameters values.

Parameter name	value	Parameter name	value	Parameter name	value
d (mm)	60	D (mm)	95	Bearing width (mm)	18
Z	19	D_b (mm)	10.7	d_m (mm)	77.5
α^0	25	f_e	0.525	f_i	0.515

According to T. A. Harris's research, the stiffness of angular contact bearing is linked to the contact angle, so the first step to calculate the bearing stiffness is numerical solution of the contact angle. The following equation could be used to obtain the contact angle at different bearing preload solving be Newton-Raphson method.

$$\frac{F_a}{ZD^2K} = \sin\alpha(\frac{\cos\alpha^0}{\cos\alpha} - 1)^{1.5}$$

Where, K=890Mpa, the iterative equation is as following:

$$\alpha' = \alpha + \frac{\frac{F_a}{ZD^2K} - \sin\alpha(\frac{\cos\alpha^0}{\cos\alpha} - 1)^{1.5}}{\cos\alpha(\frac{\cos\alpha^0}{\cos\alpha} - 1)^{1.5} + 1.5\tan^2\alpha(\frac{\cos\alpha^0}{\cos\alpha} - 1)^{0.5}\cos\alpha^0}$$

The specific results is shown in Table 2.

Table 2 The bearing preload and contact angle relationship

Bearing preload(N)	Contact angle(rad)	Bearing preload(N)	Contact angle(rad)	Bearing preload(N)	Contact angle(rad)
500	0.4546	3000	0.4843	5500	0.5030
1000	0.4626	3500	0.4885	6000	0.5061
1500	0.4691	4000	0.4924	6500	0.5092
2000	0.4747	4500	0.4961	7000	0.5121
2500	0.4797	5000	0.4996	7500	0.5149

Then the calculation this paper did by the equation of $L = (L_i^{-\frac{10}{9}} + L_o^{-\frac{10}{9}})^{-\frac{9}{10}} / (60n)$, so this paper described the relation between the preload, spindle speed and the bearing fatigue life. The final result is shown in Fig.3. The desired service life is presented in the figure. The gap between the data is more than a hundred times.

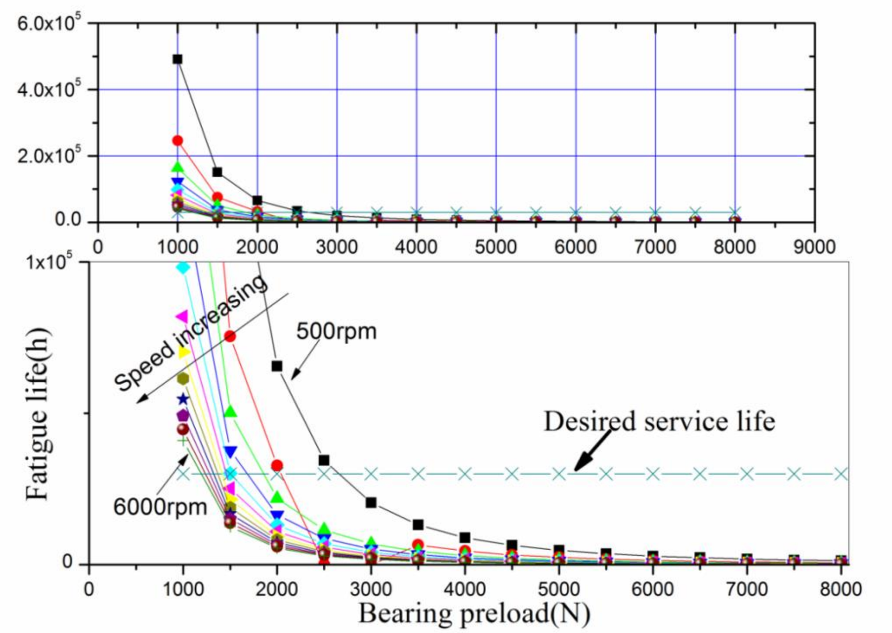

Fig.3 Bearing fatigue life vs bearing preload and spindle speed

3.2. Spindle thermal analysis

3.2.1. Bearing heat generation

From Fig.1 and Fig.2, the heat source of the spindle is the bearing heat generation. And bearing heat is mainly generated at the bearing raceways and balls, due to the friction influenced by speed, preload and lubricant. Due to the heat generated by friction losses and rolling resistance, bearing temperature would rise continuous. The heat generated by a bearing can be computed by the following equation:

$$H_f = 1.047 \times 10^{-4} nM$$

Where, n is the rotating speed of the bearing; M is the total frictional torque of the bearing and H_f is the generated heat. The frictional torque M is a sum of three torques: (1) the torque due to applied load, M_1; (2) the torque due to viscous friction, M_0; (3)the torque due to spinning motion at contact area, $M_{si(e)}$.

The torque due to the applied load can be empirically approximated by the following equation:

$$M_1 = f_1 P_1 d_m$$

Where, f_1 is a factor depending upon bearing design and relative load, for Single-row angular contact ball bearings $f_1 = 0.0013(p_0 / c_0)^{0.33}$; P_1 is dynamical equivalent load, for angular contact ball bearings, $P_1 = F_a - 0.1F_r$; d_m is Pitch circle diameter, $d_m = 0.5(D+d)$.

The torque due to viscous friction is related to bearing type, rotating speed and lubricant type. For bearings that operate at moderate speeds and under non-excessive load, the viscous friction torque can be empirically expressed as follows:

$$vn \geq 2000,\ M_0 = 10^{-7} f_0 (vn)^{2/3} d_m^3$$

$$vn \leq 2000,\ M_0 = 160 \times 10^{-7} f_0 d_m^3$$

Where, f_0 is connected with bearing type and lubrication mode, for single-row angular contact and grease lubrication ball bearings f_0 =2; v is the movement viscosity of the lubricant under operating temperature, unit:m^2/s, the movement viscosity of grease lubrication is 30.5cst at 20℃, 18cst at 40℃, 10cst at 60 ℃, 1cst=10^{-6} m^2/s. And according to Jiang et al. research[2], the $M_{si(e)}$ only 2.1% of the total friction torque under spindle rotational speed, 7000rpm; axial preload, 1000N with a considerable friction coefficient μ, so this paper ignored this part.

3.2.2. Heat sinks in spindle

The spindle heat sinks mainly include the convection between the shaft and air, the housing inner and air, the ambient air; the thermal resistance between bearing outer ring and housing, bearing inner ring and shaft; and the thermal conduction between the components.

The convection η between the shaft and air:

$$\eta = \frac{Nu \cdot k_{air}}{d}$$

Where, $Nu = 0.133 \mathrm{Re}^{2/3} \mathrm{Pr}^{1/3}$, d is the shaft diameter, $\mathrm{Re} = V * d / v_{air}$, Pr is the Prandtl number of air, V is the speed of air on the shaft surface, v_{air} is the viscosity coefficient of air, k_{air} is the air conduction coefficient. The specific results is shown in Fig.4

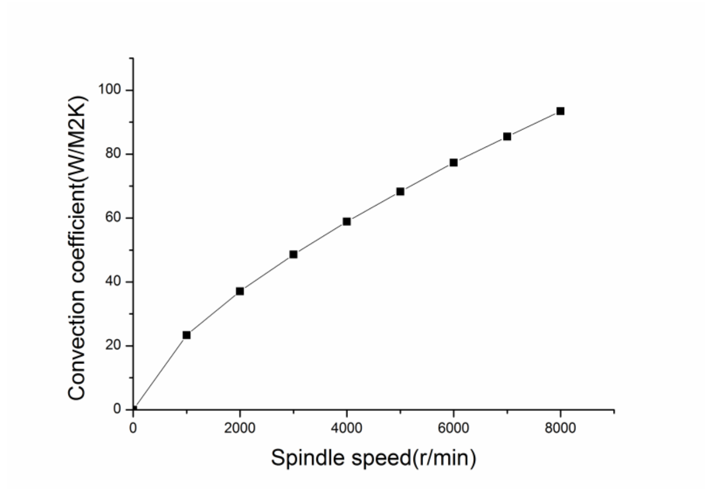

Fig.4 The convection between shaft and air under different spindle speed

The thermal resistance between bearing outer ring and housing is the reciprocal number of the thermal conduction coefficient number $\prod$ between the clearance. And the thermal conduction coefficient number $\prod$ is:

$$\Pi = \frac{1}{\dfrac{h_{ring}}{\lambda_{ring}} + \dfrac{h_{gap}}{\lambda_{air}}} A$$

Where, h_{gap} is the average of the clearance, h_{ring} is the thickness of the ring, λ_{ring} is the thermal conduction coefficient of the bearing ring, λ_{air} is the thermal conduction coefficient of air, A is the contact area. According to the specific values, the thermal in this paper is 1439.5W/m^2K of shaft and bearing outer ring, 908.75W/m^2K of the housing and bearing inner ring. All the needed parameters for bearing temperature rise analysis are conformed so far.

3.3 Bearing temperature rise simulation

The popular way for spindle thermal analysis is using Commercial FEM software, such as ANSYS, and this paper also choose ANSYS 14.0 for bearing temperature rise solving. The model is reconstructed by CAD. Specially, the mesh size is refined at the bearings ensuring the results accuracy. After loading all the boundary conditions, the bearing rise is obtained, the specific results are shown in Fig.5 and Table 3.

According to the bearing temperature rise standard, the grease lubricated bearing could not withstand the temperature above the ambient temperature 40℃. So the colored text in Table 3 represent the spindle is in appropriate working area.

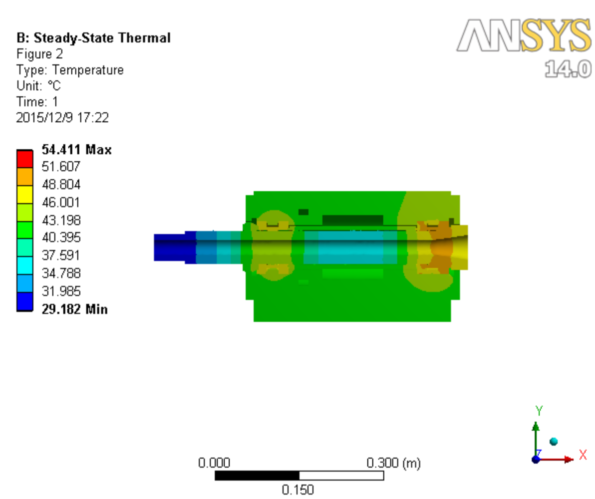

Fig.5 The temperature field of speed 5000 rpm and preload 1000N

Table 3 The bearing temperature rise vs spindle speed and bearing preload

Preload(N)	Spindle speed(rpm)							
	1000	2000	3000	4000	5000	6000	7000	8000
500	30.756	36.108	42.248	47.130	51.635	58.962	66.693	74.761
1000	31.411	37.525	44.166	49.494	54.411	62.121	69.112	78.625
1500	32.550	39.207	46.443	52.301	57.704	65.869	74.397	83.208
2000	33.639	41.040	48.989	55.44	61.387	70.024	79.069	88.39
2500	34.822	43.131	51.756	58.981	65.390	74.616	84.147	93.902
3000	36.087	45.315	54.713	62.497	69.668	79.484	89.573	99.854
3500	37.422	47.624	57.837	66.438	74.188	84.628	95.306	106.15
4000	38.824	50.043	61.027	70.387	78.92	90.022	101.32	112.74

4. Bearing preload determination and model validation

4.1 Bearing preload determination

Fig.3 and Table 3 are the data needed for baring preload optimization. On the one hand, it is necessary to extend the bearing fatigue life or spindle service life; on the other hand, controlling the bearing temperature rise is inevitable. Synthesizing the two desired goals, this paper gives out the appropriate bearing preload values. The specific results are shown in Fig.6.

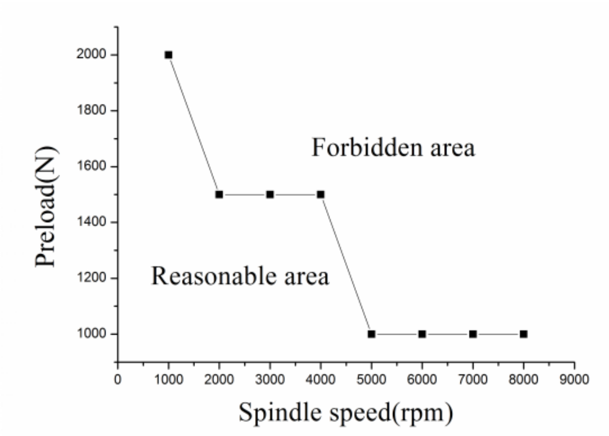

Fig.6 The appropriate bearing preload curves

4.2 Model validation

Cause barely bearing preload control system has been used on the spindle which was in practical application, this paper just measured the temperature based on FBG sensors which were distributed on the bearing outer ring, as shown in Fig.7. The specific measurement results are shown in Fig.8.

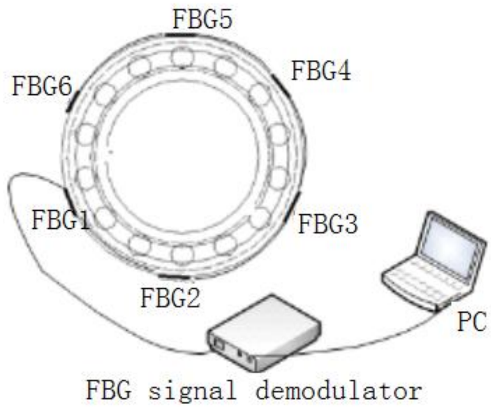

Fig.7 The FBG sensors distribution positions

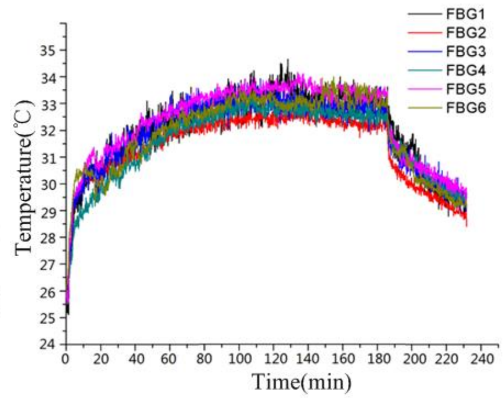

Fig.8 The specific measurement results

From the simulation results, the bearing which has the highest is at the position of far away from the drive motor. And the simulation result has been extracted. The results are perfectly consistent with the simulation results by 1.5℃, showing the thermal model built in this paper is applicable.

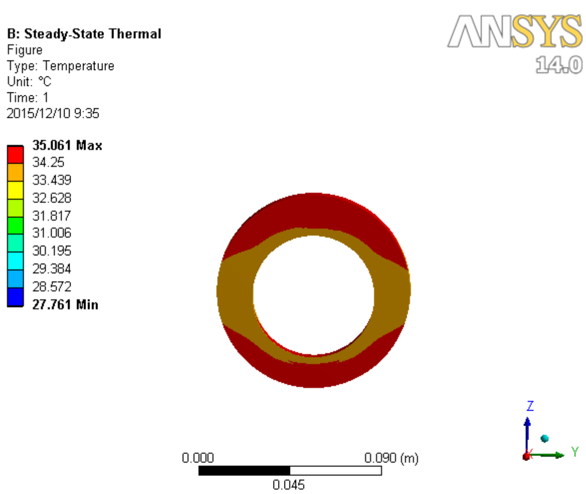

Fig.9 The simulation result of the highest temperature bearing

5. CONCLUSION

The purpose of this paper is giving out a bearing preload value for the global speed range of a traditional spindle. And the relationships between bearing preload and bearing fatigue life, bearing temperature rise are thoroughly analyzed, the reasonable preload applied area and the forbidden are given. Also the thermal model validation has been done through using FBG temperature sensors. The methodology presented in this paper can be applied to increase the accuracy of the spindles, and extend the spindle working life.

Acknowledgement

This experimental study is supported by National Natural Science Foundation (NO: 51475343).

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