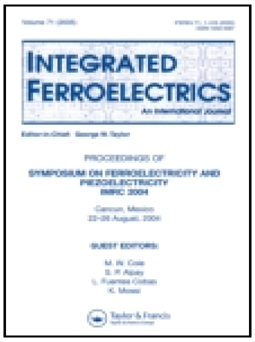

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Dynamic Characteristic Analysis of the Spindle for the Oriented Machine Remanufacturing System

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It was researched by a CNC spindle system after remanufacturing as the object, used the finite element analysis software ANSYS spindle system .Firstly, the urgency of the remanufacturing spindle and processing is analyzed; secondly it was established by the finite element model and being dynamic performance analysed, it was obtained by first five natural frequencies, vibration type and the largest integrated deformation on the specific conditions and different structural parameters of the spindle system; they are analyzed by the supporting stiffness, the supporting span and other structural parameters of the spindle system low natural vibration characteristics and its anti-vibration performance. For the spindle system dynamic design and similar components and residual stress VSR process provides a useful analysis of data and feasible method of analysis.

Keywords Spindle model; spindle stiffness; remanufacturing; damp; anti-vibration performance

1. Introduction

The machine tool spindle is the important components for the precision machining of the machine. Because the spindle system dynamic performance plays a key role in affecting the machining accuracy, it is particularly important to research the remanufactured spindle components and analyze the function of the repaired spindle.

With the rapid development of China's industries, the demand for CNC machine tools is increasing, the country has a large technical equipment control technology, design technology and assembly technology as a key development direction [1]. The nature of remanufacturing is the machine repair, is a high-tech restore, academician Binshi xu led research team is the pioneer remanufacturing technology, they has been successfully applied the surface of the repair technique to the machine remanufacturing recovery of mechanical precision [2]. The CNC machine tools is a key production line equipment, machines parts the specification, high precision, expensive features, does not allow waste, so large machine remanufacturing of machinery is high precision [3, 4]. Deformation caused by machine vibration source is complex, the processing errors are not easy to identify, the cause of the vibration generated by the machine movement has a significant impact on machining accuracy, especially straightness errors in the vertical plane, will directly affect the ultimate

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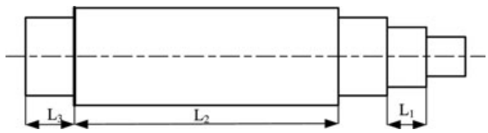

Figure 1. The repaired spindle structure diagram.

precision machine tools [5, 6, 7].The effect of vibration on the machine is one serious problem of large machine tools remanufacturing [8, 9].

This paper analyzes the spindle through the static and dynamic performance [10, 11], before the re-manufacturing to make reasonable arrangements for the parameters of the spindle, increase the accuracy level is too high to avoid causing parts remanufacturing costs [13].

2. Spindle Dynamics Model

After spindle disassembly, it is cleaned and has a damage detection techniques, then decided to re-manufacturing process based on test results. The project is a craft process when the spindle is slight wear: grinding- electroplating Cr- annealing-detecting. The spindle electroplate Cr, chrome plating thickness no more than 0.2mm, the binding force is not strong after more than. This term mainly researched the mechanical properties of the spindle after remanufacturing. After repairing a spindle structure diagram is as follows:

In order to facilitate research on the system to make the following assumptions: (1) The cutter system is rigid, the cutting process is not deformed. (2) In the process, the spindle does not move axially, and ignore the impact of the axial force. (3) The deputy campaign surface without gaps.

Let the cutter weight, moment of inertia and polar moment of inertia, respectively be m_1, J_{d1}, J_{p1}, let a large pulley weight, moment of inertia and polar moment of inertia respectively be m_2, J_{d2}, J_{p2}. The length of each shaft section is L_1, L_2, L_3; flexural rigidity is EI, displacement and angle of the spindle is z_1, z_2, z_3, z_4.

Then its equations of motion are in no vibration damping:

$[M]\left\{\ddot{z}\right\}+[K]\{z\}=\{P\}$, where is $[M]=\begin{bmatrix} m_1 & & & \\ & J_{p1} & & \\ & & m_2 & \\ & & & J_{p2} \end{bmatrix}$, $[K]=[a]^{-1}$; $[a]$ is compliance matrix

$$[a]=\frac{1}{3EI}\begin{bmatrix} l_1^2(l_1+l_2) & -\frac{l_1(3l_1+2l_2)}{2} & \frac{l_1l_2l_3}{2} & \frac{l_1l_2}{2} \\ -\frac{2l_2+3l_1}{2} & l_2+3l_1 & -\frac{l_2l_3}{2} & -\frac{l_2}{2} \\ \frac{l_1l_2l_3}{2} & -\frac{l_2l_3}{2} & l_3^2(l_2+l_3) & \frac{l_3(3l_3+2l_2)}{2} \\ \frac{l_1l_2}{2} & -\frac{l_2}{2} & \frac{l_3(3l_3+2l_2)}{2} & l_2+3l_3 \end{bmatrix};$$

$J_{p1}=m_1l_1^2$; $J_{p2}=m_2l_3^2$

3. Establish Objective Function

The objective function with stiffness: $\mathrm{F}(x) = \omega_1 f(x_1) + \omega_2 f(x_2)$, where is $f(x_1) \rightarrow$ stiffness function; $f(x_1) = \frac{a(a^2+al)}{3EI}$; $f(x_2) \rightarrow$ volume spindle; $f(x_2) = \frac{1}{4}\pi(D^2 - d^2)(1+a)$; ω_1, $\omega_2 \rightarrow$ weighting factor, indicates the importance of each part; $I \rightarrow$ C-section moment of inertia $I = \frac{1}{64}\pi(D^4 - d^4)$, so $F(\mathrm{x}) = \frac{64a\omega_1(a^2+al)}{3E\pi(D^4-d^4)} + \frac{\pi\omega_2(D^2-d^2)(1+a)}{4}$

From the above equation, that affect the objective function of the independent parameters is D, D, l, a. However, due to the spindle bore size is determined typically by the machine model or d/D value, it cannot as a design variable, the spindle structure design variables is:$X = (x_1, x_2, x_3, x_4)^T = (Ddla)^T$.

4. Constraints

4.1 Rigidity Constraint

The spindle stiffness is an important performance indicator, the outer extension end of the deflection y shall not exceed specified value y_0. According to these requirement, the rigidity constraints can be established:$g_1(x) = y - y_0 \leq 0$, Given force, y value is calculated as: $y = \frac{Fa^2(1+a)}{3EI}$, so $g(x_1) = \frac{64Fa^2(1+a)}{3\pi E(D^4-d^4)} - y_0 \leq 0$, Substituting values $g(x_1) = \frac{64\times2852\times397.5^2(1+397.5)}{3\times3.14\times2.1\times10^5(206^4-80^4)} = 329.7 \times 10^{-5}$, look-up table shows: $g(x_1) \leq y_0$.

4.2 Strength Constraints

Cutting stress intensity allowable limit, if $C_1 = \frac{9.55\times10^6 P/_n}{0.2[1-(d/_D)^3]}$, $[\tau_T]$, is a factor associated with the material. Substituting values: $C_1 = \frac{9.55\times10^6\times7.5/_{748.5}}{0.2\times[(80/_{206})^3]} = 8\times10^6$, so $g(x_2) = C_1/_{D^3} = 8\times10^6/_{206} - [\tau_T] \leq 0$

4.3 Restrictions Corner

The shaft deflection angle θ should be less than the allowable value $[\theta]$, $\theta = \frac{Fal}{3EI}$, substituting values:

$$\theta = \frac{2852 \times 397.5 \times 225}{3 \times 2.1 \times 10^5 \times 110.2 \times 10^6} = 3.67 \times 10^{-6}; \text{so } g(x_3) = \theta - [\theta] \leq 0$$

4.4 Torsional Deformation Limits

The shaft torsional deformation conditions is $\psi \leq [\psi]$, $T \rightarrow$ Suffered shaft torque $T = 9.55 \times 10^{6P}!/_n$

$G \rightarrow$ Shear modulus axis, $G = 8.1 \times 10^4 MPa$; $I_p \rightarrow$ Inertial axis from cross section, $I_P = (D^4 - d^4)/_{32}$

substituting values:$I_p = (206^4 - 80^4)/_{32} = 55 \times 10^6$, so $g(x_4) = \psi - [\psi] \leq 0$.

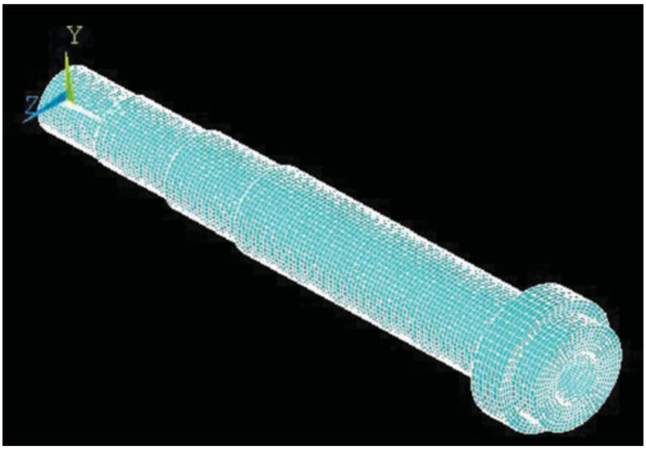

Figure 2. Mapping spindle model after gridding.

4.5 Limit Cutting Force

Cutting force $F_Z \leq {}^{P \cdot \eta}\!/_{V}$; $V = \pi Da$. $g(x_5) = F - {}^{P \cdot \eta}\!/_{V} \leq 0$, substituting values:

$$g(x_5) = 2852 - \frac{7.5}{3.14 \times 206 \times 397.5 \times 10^{-3}} \leq 0$$

5. The Spindle Frequency Domain Analysis Model

Where is $l_1 = 120\ mm$, $l_2 = 680\ mm$, $l_3 = 200\ mm$, $E = 2.1 \times 10^5 N/mm$, $I = \frac{\pi}{64}(D^4 - d^4) = 86.34 \times 10^6\, mm$, $D = 206\, mm$, $d = 80\, mm$, $m_1 = 60\, kg$, $m_2 = 6.4\, kg$. then: $J_{p1} = 8.64 \times 10^5 kg \cdot mm^2$; $J_{p2} = 2.59 \times 10^5\ kg \cdot mm^2$. Integrated into the data can be obtained predecessor z.

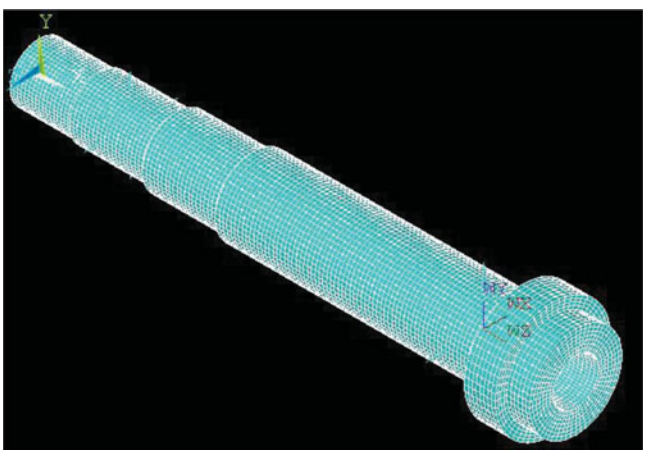

Figure 3. Spindle frequency domain model.

Table 1
Combin l4 Unit parameter input

Support position	Front support	After supporting
Stiffness (N/m)	10E8	10E8
Damp (N*s/m)	0	0

A three-dimensional system model was created by ANSYS. In order to analysis the frequency domain, the model is simplified, including threads and keyways by physical treatment; ignored undercut, chamfering and other local features. It can be improved computational efficiency after such a simplification and accuracy of the results has very small impact. The spindle solid model used Solid 45 unit.

Due to the axial stiffness of the spindle great, the damping have little effect on lateral vibration characteristics, affect of radial stiffness is considered only in establishing a frequency domain model, the use of the four-section with uniform circumferential spring-damping element simulation. Spindle force is treated as a portrait of a damping-spring only, the spindle tolerance axial tension and compression, bending and torsion is not considered. Spindle force is treated as a torsion spring -damping, subject to pure torsion, bending and axial load is not considered. Combin14 unit does not have the quality attributes, the concentrated quality unit can be used by Mass simulation.

When the model of spindle bearing being built, the circumferential section has evenly distributed four springs-damping. The length of the spring unit is determined by the bearing of the inner and outer radius. The outer nodes of the bearing used Key Points instruction and the inner nodes applied Hard PT when we established the frequency domain model, simultaneously ensuring that the spring unit is divided into a number of frequency-domain. The four nodes restrict freedom outside spring-damper unit, the front four-node of the inside bearings limit axial freedom. To limit the movement of the spindle axis X, UX constraints were added in a cross section M2 and four nodes of the spring. At the other end of the spring is completely fixed.

6. Spindle Model Analysis

The first five order natural frequencies and vibration of the spindle were calculated adopting to Block Lanczos mode extraction in ANSYS, obtained the first two natural frequencies of spindle: 874.74 Hz, 1019 Hz, as shown in Fig. 4. The two and three order natural frequencies are equal, and its performance is orthogonal modes, it can be regarded as a counterweight roots; similarly, the four and five is also true in Fig. 4. The natural frequency of the spindle can be seen high enough that the dynamic and static rigidity can meet the design requirements of high stiffness. That the first critical speed $n = 60 \times 874.74 = 52484$

Table 2
Solid 45 Unit material parameters

Parameter	Modulus of elasticity	Poisson's ratio	Density (kg/m^3)
input	2.06e11	0.28	7800

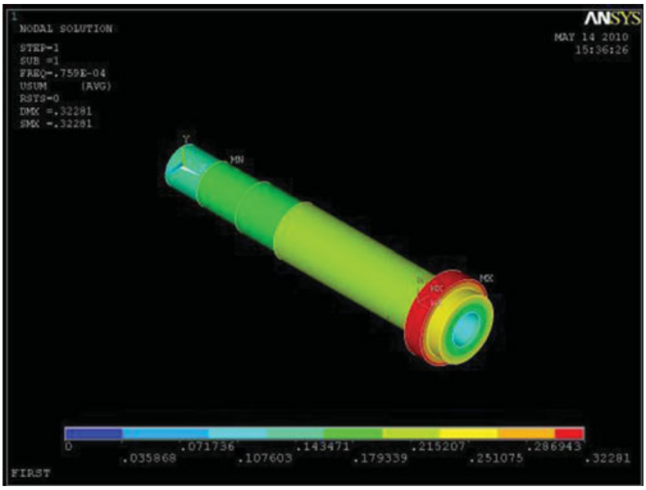

Figure 4. Five order natural vibration frequency of spindle.

r/min was calculated in accordance with the natural frequency of the spindle modal analysis is much larger than the work of the high-speed spindle turning speed (less than 5000 r/min), it showed that the work speed of the spindle can effectively avoid the resonance region and ensured precision spindle.

It can be seen from the front calculations, the machine tool spindle is analyzed with ANSYS, to calculate quickly, to get the modal analysis is intuitive and easy. It can be seen from the front graphics, the spindle bending deformation occurred in the second, third, fourth, fifth-order; the spindle axial deformation occurs during the first stage. The spindle is bending deformation mainly, at the same time occurs in the axial deformation. Thus, the spindle is the main bending deformation in operation.

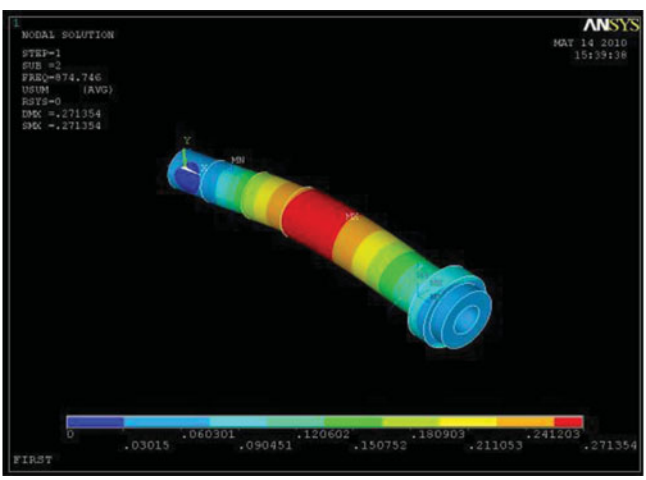

Figure 5. Second-order vibration mode.

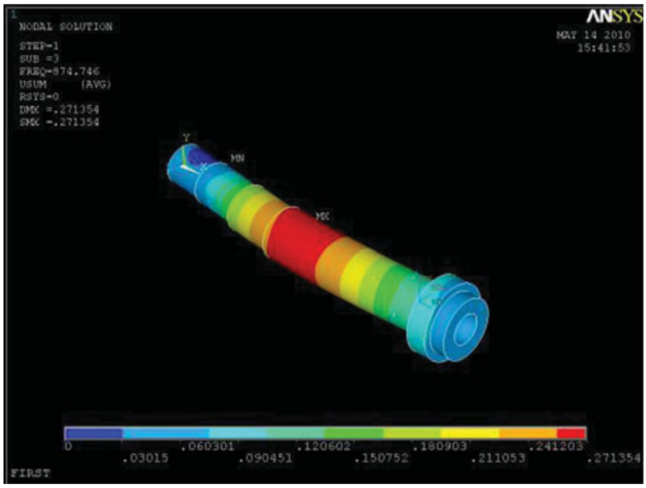

Figure 6. Three-order vibration mode.

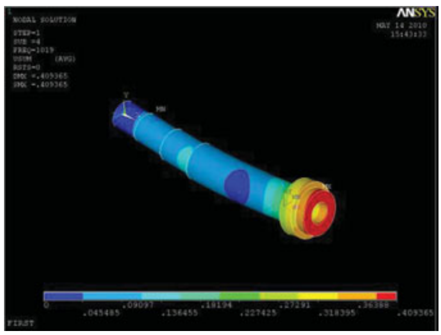

Figure 7. Four-order vibration mode.

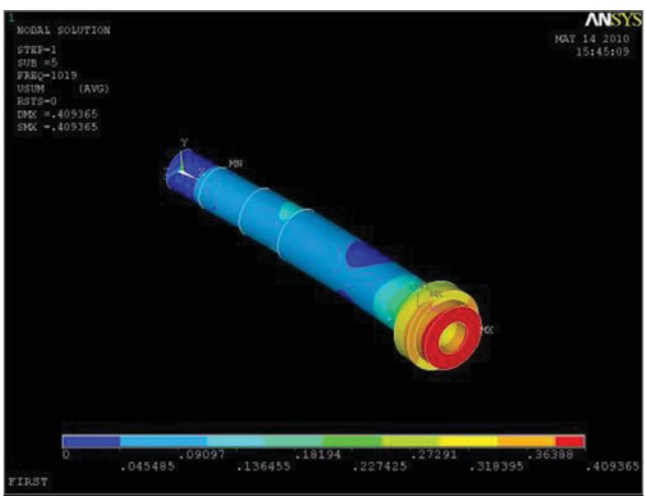

Figure 8. Five-order vibration mode.

The CNC machine tools develop in the high-speed and high stiffness currently. The machine tools must have good dynamic performance to make the machine safe and reliable work, to ensure the high precision machined parts. The spindle is analyzed and checked by ANSYS to find the problem from the results, to eliminate hidden dangers in time, not only to save investment, but also shorten the product development cycle.

7. Conclusion

According to the theory of vibration, the vibration energy are mainly concentrated in the first and second order, bending vibration is main in the process. Because of the approximate linear model (including linear characteristic of the material and linear frequency-domain model), the theoretical value of the spindle and the experimental measured value are closer in the case of the lower order, and the error is growing on the high-end part.

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