The dynamic characteristic analysis of spindle based on ANSYS

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Abstract

The basic concept of dynamic characteristics of spindle is introduced. The first ten order vibration mode and natural frequency of shaft are obtained by modal analysis and harmonic response analysis using software ANSYS. The deformation trend of spindle under resonance state is analysed and the contrast between theory and simulation critical speed is done. The frequency response curve of spindle is given. The first order, second order dynamic stiffness and dynamic stiffness under general conditions are calculated. The influence of bearing damping to spindle dynamics characteristics is discussed. In the end, the measures of improving spindle performance are summarized.

Keywords: Shaft, Dynamic characteristic, ANSYS, modal analysis, harmonic response analysis

1 Introduction

In actual manufacturing, cutting force and other exciting forces are applied in the form of dynamic load onto the structure. In order to ensure the manufacturing accuracy and efficiency of work piece, not only the spindle system of machine tool should have some static rigidity, but also the dynamic property of spindle structure should be considered. The dynamic property of spindle structure refers to its resistance to forced and self-excited vibration. Notably in high-speed precise machine tool, the dynamic property of spindle usually has great influence on the quality of manufactured goods [1]. For a long time, researches on the dynamic property of spindle have mainly adopted analogies of experience or other methods, which brings about some negative effects, such as prolonged product development cycle, relatively higher cost and blindness. In addition, its theoretic calculation method is found to be extremely difficult for relatively complex structures. Virtual electrodynamics research is therefore proposed. It mainly constructs the model of finite element analysis of digital mock-up through some large-size finite element analysis software to make dynamics analysis accordingly, which can be used to know the dynamics property of digital mock-up. In this way, product design cycle and cost incurred would be reduced with specific target. This paper focuses on the spindle of HTC3250µn and HTC2550hs CNC turning centre, and uses ANSYS software to research the dynamic properties of this machine tool spindle. This greatly simplifies the performance analysis process of spindle unit, lessens the intervention of manual operation and avoids repetitive work.

At present, the researches on dynamic property can be mainly classified into two respects: natural vibration characteristics and response characteristics. Wherein, the research of natural vibration characteristics focuses on undamped free vibration, which can be used to get the natural property of vibration system, namely natural frequency and vibration mode; while the research of response characteristics focuses on the steady-state response of structure that is sinusoidal-varying with time, which can be used to calculate the dynamic response of structure and get the displacement response and stress response [2].

2 Spindle modal analysis

2.1 SUMMARY OF MODAL ANALYSIS

Modal analysis is a kind of technology used for determining the natural frequency and vibration mode of the structure (Natural frequency and vibration mode of the structure are two extremely important parameters in designing the structure on dynamic loads) [3]. The natural frequency reflects the degree of structure rigidity, namely bigger natural frequency means bigger rigidity. Through modal analysis, the structure can be designed to avoid resonance or vibration at specific frequency, designers would know how the structure responds to the dynamic loads of different types, and control parameter will be provided for other dynamic analysis as well. As the structure vibration characteristics determines its response status at different dynamic loads, modal analysis is the basis for further analysing structure dynamics, such as transient dynamics, harmonic response and spectral analysis.

Machine tool vibration forces the change on relative position between cutting tool and work piece as well as relative speed, which changes the cutting process and limits the product manufacturing precision and efficiency. Most vibrations of machines tool come from the spindle.

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Therefore, the natural frequency and vibration mode of spindle are important metrics for analysing and evaluating the dynamic performance of machine tool. Although the spindle of machine tool is an infinite multi-degree of freedom system with infinite multiple natural frequencies and their corresponding vibration modes, the low-order mode of the spindle is actually enough to express its dynamic property. Therefore, a few of low-order modes starting from the first-order mode are enough to make a finite element modal analysis of the machine tool.

The steps of making a modal analysis by ANSYS software are more or less the same with other analyses, but it should be emphasized that the application of constraint and boundary condition on the modal analysis will affect the result to a larger degree. In this concern, a proper application of boundary condition is necessary to accurately get the natural frequency of structure. Normally the mode extraction methods are described as below:

1) Block Lanczos Method: it is applicable to the solution of the eigenvalue of large-size symmetric matrix, which has the characteristics of lower quality requirement of model unit and faster solution availability;

2) Power Dynamics Method: it is applicable to multi-degree of freedom problems, especially for only solving the first several orders of structure;

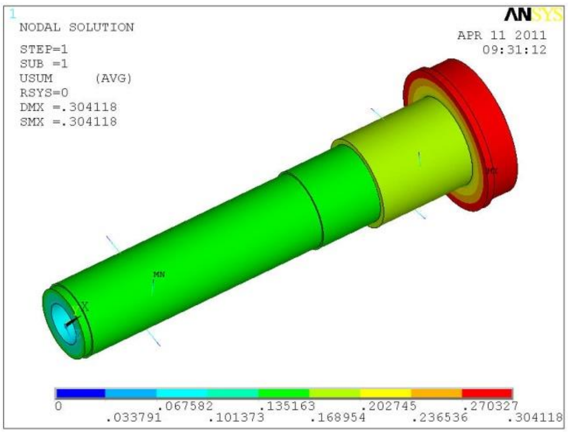
3) Unsymmetric Method: it is used for the problems when the system matrix is unsymmetric;

4) Damped Method: it is used for problems considering damp effect; this paper uses this method as the bearing damp effect cannot be ignored.

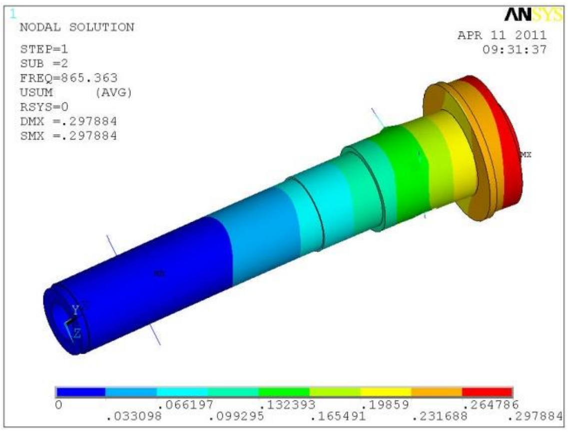
In typical modal analysis, the only effective load is zero-displacement constraint. If one non-zero displacement constraint is appointed at some point of DOF, the program will replace the position of this DOF with zero-displacement constraint. Although other types of loads can be appointed in modal analysis, such as force, stress, temperature and acceleration, they are invariably ignored in modal extraction. All loads vector will be calculated by the program and saved in vibration mode documents, which can be used in harmonic response analysis or transient analysis by the mode superposition method.

2.2 SPINDLE MODAL ANALYSIS BASED ON ANSYS

In order to get an accurate result of the spindle modal analysis, this paper properly modifies the constructed three-dimensional static analysis model, which would be used as the finite element model for spindle modal analysis. The researcher alters the analysis type to modal analysis in the solver, and sets the frequency range of extracted mode from the minimum "0" to the maximum "3000". Spring damped unit is used to simulate the boundary condition and supporting form, the pitch point of inner ring is noted to restrict the axial degree of freedom , partial useless natural frequency and vibration mode are filtered out, and the spindle modal analysis and modal extension are finally made. Calculated by ANSYS software, the first 10-order modes of spindle are extracted to get their vibration characteristics. Vibration modes diagram are shown as Fig.1. From Fig.1, the vibration modes of spindle are mainly torsion, oscillation and bend. First-order displays the vibration mode of rigid body; the natural frequencies for two-order and three-order, four-order and five-order, six-order and seven-order are the same and vibration modes are in orthogonal, which can be mathematically understood as different characteristic vectors for one same characteristic value or multiple root, but the deformation trends are differently from one another.



a)



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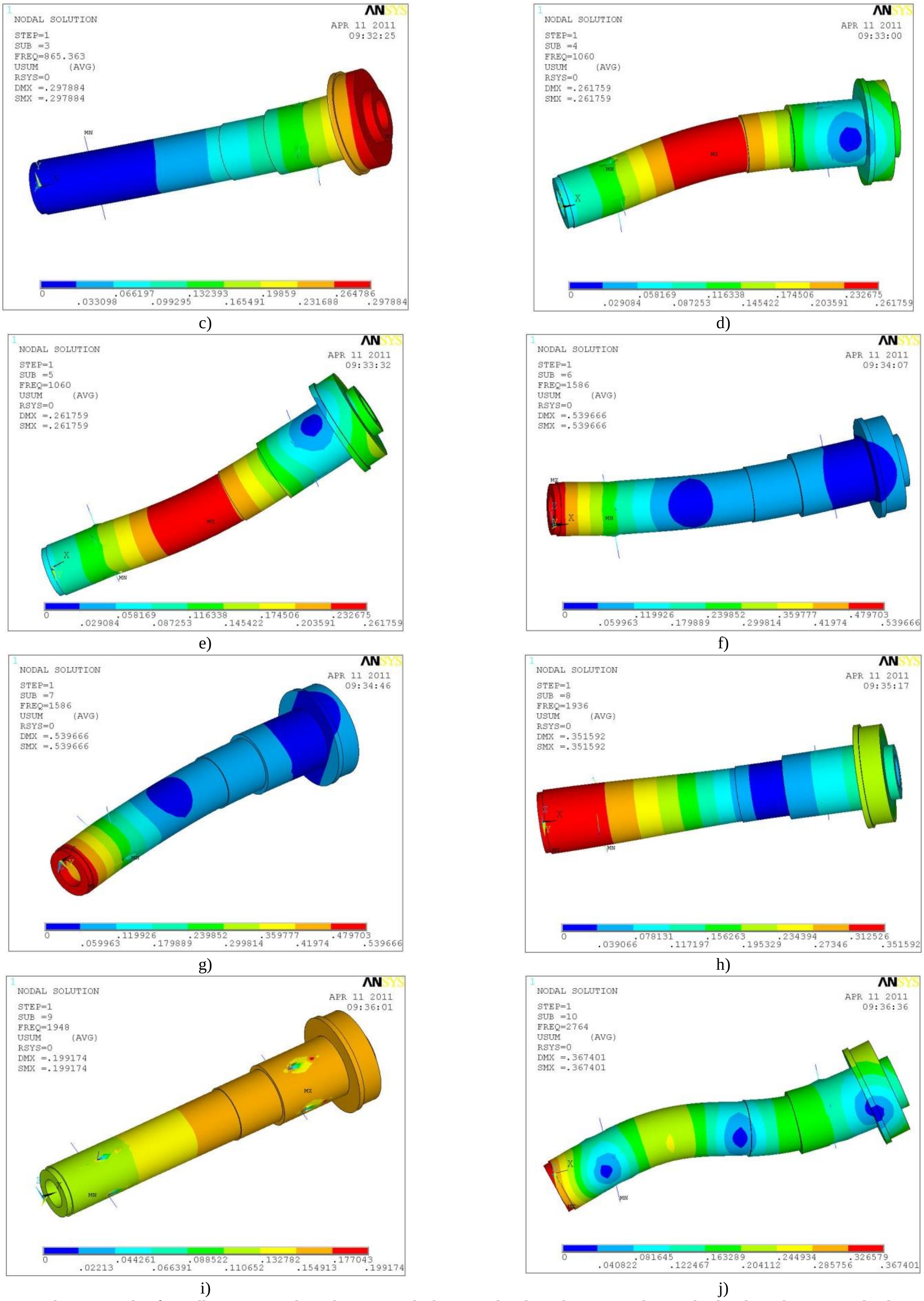

FIGURE 1 Vibration mode of spindle: a) First-order Vibration Mode, b) Second-order Vibration Mode, c) Third-order Vibration Mode, d)Four-order Vibration Mode, e) Five-order Vibration Mode, f) Six-order Vibration Mode, g) Seven-order Vibration Mode, h) Eight-order Vibration Mode, i) Nine-order Vibration Mode, j) Ten-order Vibration Mode

Through a separate analysis of all acquired vibration diagrams, combined with the natural frequencies for all orders calculated by ANSYS software, the vibration characteristics of spindle can be summarized as shown in Table 1.

TABLE 1 Natural vibration frequency and mode of spindle

Degree	Frequency(Hz)	Vibration Mode
1	0	Translational
2	865.36	First-order Z oscillation
3	865.36	First-order Y oscillation
4	1060.4	Two-order Y oscillation
5	1060.4	Two-order Z oscillation
6	1585.8	Three-order Y oscillation
7	1585.8	Three-order Z oscillation
8	1936.1	Torsion

In actual, the modal analysis of spindle can be made by use of the two-dimensional finite element analysis model as well. Although the natural frequency solved by the two-dimensional finite element analysis model is basically the same with that of three-dimensional finite element analysis model, the solved vibration mode isn't as resourceful as that of three-dimensional finite element analysis model, which is mainly caused by huge difference of nodes quantity among different unit types as well as the consideration of different treatment methods for boundary conditions. Therefore, the calculation simulation effect of three-dimensional finite element model proves to be more accurate.

2.3 ANALYSIS OF SPINDLE CRITICAL SPEED

Traditionally, critical speed of spindle mainly refers to the spindle natural frequencies at all orders. As the common spindle has the characteristic of relatively lower working speed, the influence of spindle speed on the dynamic characteristics can be ignored. Therefore, the traditionally defined critical speed is reasonable.

With the growing speed of machine tool, the traditional definition of critical speed cannot meet the requirement of high-speed spindle. In this concern, the influence of speed cannot be ignored in the dynamic analysis of high-speed spindle. The critical speed of high-speed spindle is defined as the corresponding speed at one natural frequency. In the analysis of high-speed spindle critical speed, the spindle will resonate when the rotation frequency of spindle equals its natural frequency. However, if this case should occur in actual manufacturing, it not only impairs the manufacturing quality, but also irreversibly damages the spindle, reduces its service lifetime and increases the maintenance cost.

At a specific speed, the deflection of spindle can be arbitrary value, wherein the deflection at each point can be arrayed in an elastic straight line. The specific speed is namely the critical speed as output in Equation (1).

$$w_c = l^2 \sqrt{\frac{EJ}{m}} = \frac{(la)^2}{a^2} \sqrt{\frac{EJg}{A\gamma}} , \quad (1)$$

Wherein, $la = \pi, 2\pi, 3\pi$, A is the cross-sectional area of spindle segment, and γ is the material density.

Actual and ideal structure of spindle differs in the uniform section of shaft end, and the critical speed of actual structure has slightly higher critical speed as output from Equation (1). In most cases, the critical speed of spindle can be directly output through Equation (2) with the unit of speed converted to r/min.

$$n = 60f , \quad (2)$$

Wherein, n is the speed, f is the natural frequency. The critical speeds of spindle at all orders can be calculated by the natural frequency through ANSYS, but it starts from the two-order frequency because the first-order is the vibration mode of rigid body. Among all critical speeds of spindle at all orders as listed in Table 2, the low-order speed of spindle is of most referential value. However, the low-order speed value output from theoretical equation is in discrepancy with the actual value unless the design speed is within the absolute safety range; or otherwise it may easily result in irrecoverable result caused by misjudgement. In order to accurately get the critical speed of spindle, it can be calculated much closer to the actual value through ANSYS, which would avoid the misjudgement.

TABLE 2 Critical speed of spindle (Theoretical value)

Degree	Frequency(Hz)	Speed(r/min)
2	865.36	51921.6
3	865.36	51921.6
4	1060.4	63624
5	1060.4	63624
6	1585.8	95148
7	1585.8	95148
8	1936.1	116116

The speed of spindle is set as 0rad/s, 5000rad/s, 10,000rad/s and 15,000rad/s respectively for modal analysis, and Campbell Diagram [4] is eventually drawn up as shown in Fig.2.

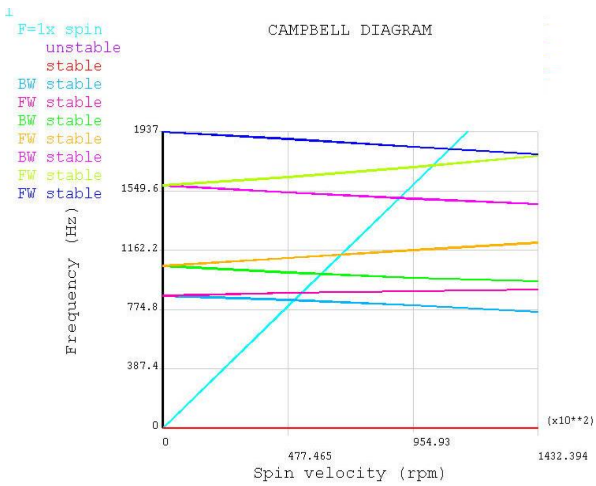

FIGURE 2 Campbell diagram for critical speed

Though Campbell Diagram we can find there are two curves diverging from each natural frequency, wherein one curve increases with the speed, while the other one decreases with the speed. These are called as forward precession frequency and backward precession frequency respectively in rotor dynamics. According to the definition of critical speed, the speed at the cross point between the 45°straight line as shown in Campbell diagram and forward precession frequency curve is the critical speed [5] at this very moment. The critical speed of spindle (Simulation value) is shown in Table 3. In comparison between the values in Table 2 and Table 3, the critical speed of spindle calculated by drawn-up Campbell diagram through ANSYS is approximately 1% higher than the critical speed by theoretical equation. The equation for critical speed of forward precession is described as Equation (3):

TABLE 3 The critical speed of spindle (Simulation value)

Degree	Frequency(Hz)	Speed(r/min)
2	865.36	52372.3
3	865.36	52372.3
4	1060.4	64707.8
5	1060.4	64707.8
6	1585.8	97252
7	1585.8	97252
8	1936.1	118458

$$\left|(-[M_1]+[J_1])\omega^2+\left[K_{g1}\right]=0\right|. \tag{3}$$

Through it we can find the critical speed would invariably increase if there is any forward precession, which is found to be in conformity with the reality; the highest design speed of HTC3250μn and HTC2550hs CNC turning centre is 6000r/min, which is lower than 2/3 of the first-order critical speed. Consequently, the spindle would not have natural vibration in actual use, which meets the running requirement.

3 Harmonic response analysis of spindle

3.1 SUMMARY OF HARMONIC RESPONSE ANALYSIS

Harmonic response analysis is a technology used to solve the structure response under the simple harmonic load when the frequency is known. The input value is the harmonic load (force and forced displacement) that its size and frequency is known, or under multiple loads at the same frequency, the output value is the harmonic displacement [3] at each degree of freedom. In actual manufacturing, cyclic exciting force will apply on the spindle when the machine tool runs. When the natural frequency of spindle equals the frequency of cyclic exciting force, resonance effect will take place. During actual manufacturing, resonance will not only reduce the manufacturing accuracy that result in rejected parts, but also impair the machine tool and cutting tool. As a result, it is necessary to make a harmonic analysis of spindle in the phase of designing the spindle.

In ANSYS, the finite model of harmonic response analysis is more or less the same with the finite model of modal analysis. Full method requires a direct integral computation, which would invariable consume a great amount of time; comparatively, the method of mode superposition directly multiplies and sums the obtained modal type through modal analysis, which has the characteristics of quickness, high efficiency and well-intended accuracy. To that regard, this paper adopts the method of mode superposition to make a harmonic response analysis.

3.2 HARMONIC RESPONSE ANALYSIS OF SPINDLE BASED ON ANSYS

Before making a harmonic response analysis of the spindle, the load changing in line with time shall be determined, or the exciting force shall be determined. The equation for typical exciting force can be expressed as follows:

$$F(t)=F_1\sin\left(\omega t+\phi\right), \tag{4}$$

wherein, F_1 stands for the amplitude of exciting force; ω stands for the range of forced vibration frequency; and ϕ stands for the phase angle. The amplitude of exciting force is actually the maximum value of external load; the range of forced vibration frequency refers to the frequency range of simple harmonic load; and the phase angle refers to the metric for load lagging or preceding the reference time [6]. In this paper, the amplitude of exciting force is turning force, which is to be put into the equation. Equation (4) in the manner of Fourier series is extended:

$$\begin{aligned}F(t)=F_0+F_1\sin\left(\omega t+\phi_1\right)\\+F_2\sin\left(\omega t+\phi_2\right)+\cdots\end{aligned}. \tag{5}$$

If the higher order term is not considered, ϕ_1 is approximated to be “0”, therefore the exciting force equation is further simplified as below:

$$F(t)=F_1\sin\left(\omega t\right). \tag{6}$$

According to general manufacturing, the vibration range is between 0 to 600Hz. In this way, the exciting force of harmonic response analysis is finally determined.

In the above part, the modal analysis is made on the spindle, which is followed by the harmonic response analysis after saving and analysing the result. In view of large time consumption in an accurate harmonic response analysis, this paper firstly analyses the entire range of vibration frequency, and gets the spindle radial response displacement curve within the range of vibration frequency by reducing the sub-steps and analysing time. Due to limited sub-steps, this curve only shows the approximate position of change tendency and vibration points. In order to obtain the spindle radial response displacement in a more accurate way, accurate analysis could be made within one frequency range respectively. By adding more sub-steps within this frequency range, more accurate analysis result can be obtained to evaluate the response characteristics of spindle.

In time-history postprocessor of ANSYS, the response relationship between variable and frequency cannot be observed until the observable variables are defined. In reality, the definition of variable directly determines the result of spindle response analysis. Generally, dangerous points in the spindle shall be contained in those observable variables. In case some dangerous points are ignored, the evaluation of spindle response characteristics would possibly be erroneous, the dynamic rigidity of spindle would deviate, which would lead to the manufactured machine tool lower than the required precision of actual manufacturing.

In this concern, this paper intends to analyse the five dangerous points of spindle, combined with the analysis of spindle front-end, front prop, rear prop, middle point and rear-end, so that the spindle response characteristics can be obtained. In order to compare the influence of the damping of bearing on the dynamic characteristics of spindle, respective discussions under damping and absence of damping must be made. Set 0-1200Hz as the excitation frequency, through harmonic response analysis, the radial amplitude-frequency curves for spindle front-end, front prop, rear prop, middle point and rear-end can be shown in Figure 3.

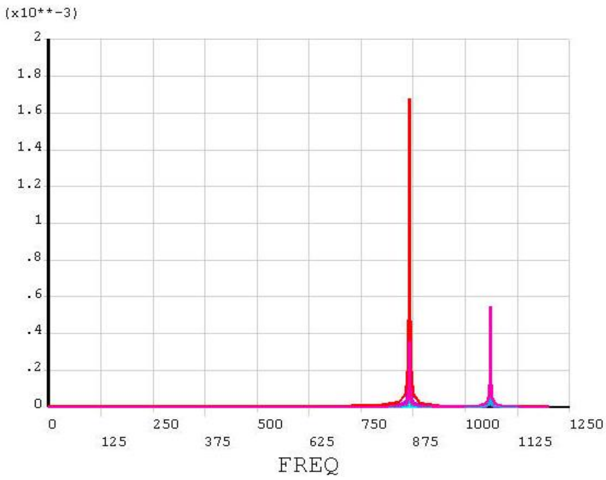

FIGURE 3 Displacement-frequency curve of spindle(0～1200Hz)

Figure 3 shows there are significant resonances at two natural frequencies: 865Hz and 1060Hz, which also shows bigger fluctuation of deformation amplitude. As a result of no damping, the vibration amplitude has the infinite tendency to increase and dynamic rigidity significantly declines, when the spindle does not meet the manufacturing requirement. In order to better reduce the influence of spindle vibration on the manufacturing precision, damping is usually introduced to reduce the vibration response amplitude of spindle. The bearing damping in this paper is set as 5000 $N \cdot S / \mu m$, and the excitation frequency is set as 0～1200Hz. Through the harmonic response analysis, the amplitude-frequency curves for spindle front-end, front prop, rear prop, and middle-point, rear-end can be shown as Figure 4. The Figure 4 shows there are significant resonances at two natural frequencies: 865Hz and 1060Hz, while in consideration of damping effect, the vibration amplitude would significantly decline. The discussion is respectively made for two resonance points in the following part.

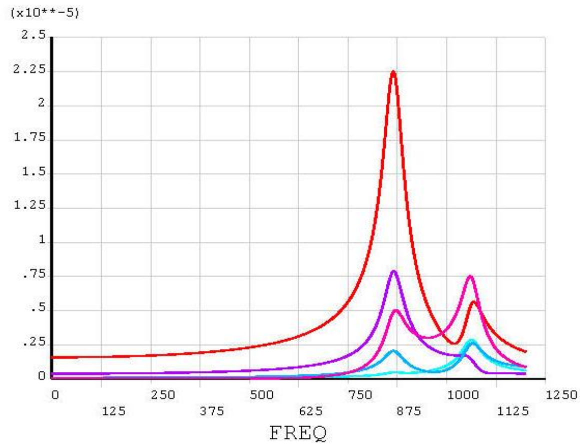

FIGURE 4 Displacement-frequency curve of spindle (0～1200Hz)

Set 400～900Hz as the range of excitation frequency, control the sub-steps as 5,000, and reanalyse the harmonic response, which is shown in Figure 5 and Figure 6.

Figure 5 and Figure 6 show that the dynamic stiffness of spindle remains almost unchanged before the excitation frequency could approximately reach the first-order natural frequency. When it closes to the natural frequency, the response displacement begins to upward while the spindle stiffness begins to downward. When it reaches the resonance point, the response displacement is the highest and the spindle dynamic stiffness is the lowest. After that, the response displacement begins to downward, while the spindle dynamic stiffness gradually upwards.

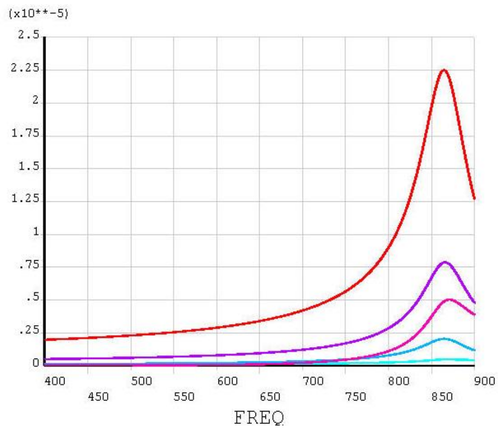

FIGURE 5 Displacement-frequency curve of spindle (400～900Hz)

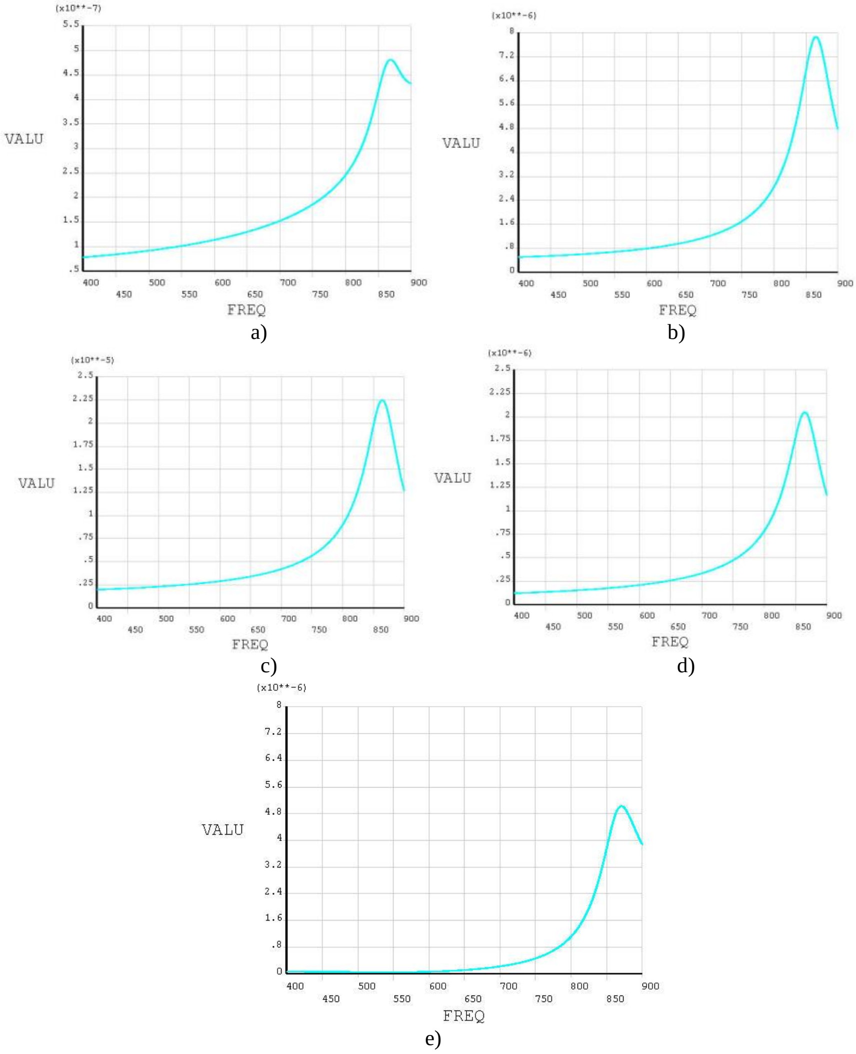

FIGURE 6 Displacement-frequency curve of spindle (400～900Hz): a) Displacement-frequency curve of front bearing, b) Displacement-frequency curve of rear bearing, c) Displacement-frequency curve of spindle front-end, d) Displacement-frequency curve of spindle rear-end, e) Displacement-frequency curve of spindle middle-point

Through the analysis of all spindle points, roughly around the first-order natural frequency, the displacement response of spindle front-end is more significant, and at 865Hz, the response displacement abruptly increases to 22.5 μm . The spindle dynamic stiffness significant downwards; after that the displacement response abruptly decreases while the spindle dynamic stiffness gradually upwards. During this process, the minimum dynamic stiffness of spindle is shown as below:

$$K = \frac{452.9}{22.5} = 20.13 N / \mu m . \quad (7)$$

In view of the analysis around spindle two-order resonance points, set 900～1400 Hz as the range of excitation frequency, control sub-steps as 5,000, select five key points on the spindle to draw up the amplitude-frequency characteristic curve and make harmonic response analysis, the result is shown as Figure 7. Figure 7 shows the displacement response under the influence of spindle exciting force is lower than the first-order resonance point, when the spindle is at the two-order resonance point. However, the spindle vibration at this order remains evident, and the spindle stiffness shows the tendency of upward, downward and gradually upward after leaving the resonance point. The displacement-frequency curves for all key observing points are shown in Figure 8.

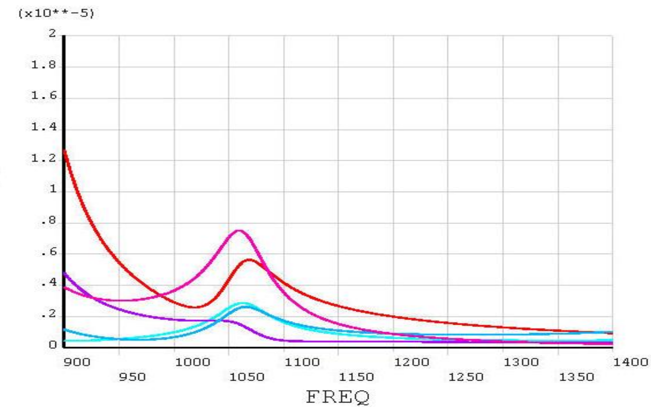

FIGURE 7 Displacement-frequency curve of spindle (900～1400Hz)

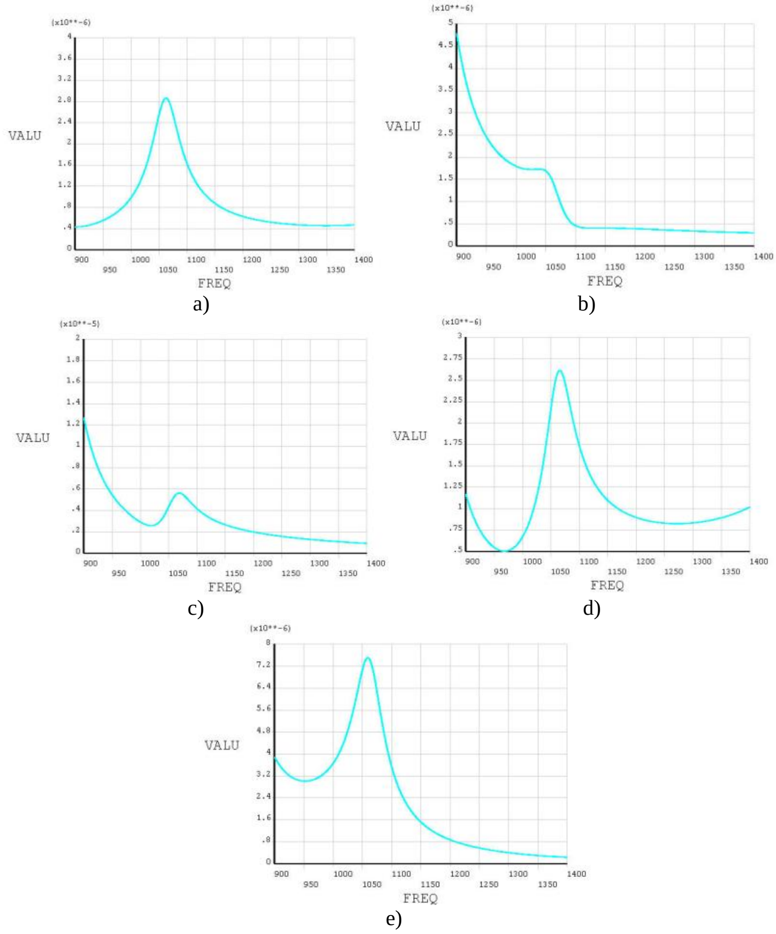

FIGURE 8 Displacement-frequency curve of spindle(900～1400Hz): a) Displacement-frequency curve of front bearing, b) Displacement-frequency curve of rear bearing, c) Displacement-frequency curve of spindle front-end, d) Displacement-frequency curve of spindle rear-end, e) Displacement-frequency curve of spindle middle-point

Through the analysis of each point on the spindle, it can be found that the displacement response of middle point is the most prominent around the second-order natural frequency. At 1060Hz, response displacement increases suddenly and the maximal displacement increases to 7.8 μm. The dynamic stiffness of spindle declines, thus it can be judged that spindle is prone to crack at the moment. Then displacement response suddenly declines and spindle dynamic stiffness gradually improves. During this process, the smallest dynamic stiffness of the spindle is as follows:

$$K = \frac{452.9}{7.8} = 58.06 N / \mu m . \tag{8}$$

By comparing the amplitude of the first-order resonance point and two-order resonance point, the spindle vibration tendency matches with the modal vibration as solved in the previous part. The spindle vibration tendency does not increase strictly with the increase of natural frequency, but its amplitude usually depends on the mutual effect of characteristics and damping.

The above analysis can be used to evaluate the vibration characteristics at the time of spindle resonance. However, when applied to the actual manufacturing, as the spindle keeps off the forced resonance area at the designing phase, the analysis of the resonance point cannot completely evaluate the spindle dynamic characteristics. In order to cover this point, according to the variation range of excitation frequency at the normal state of manufacturing, the analysis result of spindle harmonic response can be shown in Figure 9. Figure 9 shows the response displacement of all spindle points increase with the excitation frequency, notably on the spindle ends and front-end bearing. At 600Hz, the maximum response amplitude is 2.9 μm, when the spindle dynamic stiffness can be expressed as:

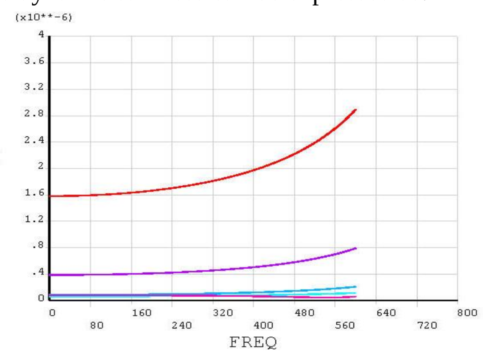

FIGURE 9 Displacement-frequency curve of spindle(0～600Hz)

$$K = \frac{452.9}{2.9} = 156.17 N / \mu m . \tag{9}$$

By analysing the dynamic stiffness, it can be concluded that the spindle adopted in this paper meets the design requirement; in actual manufacturing, it can meet the precision requirement.

4 Conclusions

The performance analysis and optimization of the spindle based on ANSYS can effectively shorten the design time and improve design efficiency. Through the dynamic characteristics analysis of spindle, it can be summarized that the following method can improve the dynamic characteristics of spindle:

(1) Improving the stiffness and damping of bearings. From the vibration diagram and harmony response analysis of spindle, it can be clearly found that the deformation around the front support of spindle is more obvious. Thus, improving the stiffness and damping of the front bearing can effectively reduce the vibration deformation in the area, improve the natural frequencies of spindle and avoid the happening of resonance phenomena.

(2) Improving the structure of spindle. The configuration way of bearing, spindle span and length of overhanging end has very important influence to the dynamic characteristics of spindle. On the premise of conditions permit, the structure optimization design can significantly improve the dynamic characteristics of spindle.

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