Static and dynamic characteristics analysis of self-balancing motorized spindle

Z Wang1*, W Z He1, H M Zhou1 and S Y Du1

1School of Mechanical Engineering, Shenyang Jianzhu University, Shenyang 110168, China

*E-mail: juven1126@163.com

Abstract. In order to effectively suppress the vibration of motorized spindle caused by unbalance fault, realize online dynamic balance control of motorized spindle unit. A self-balancing motorized spindle with dynamic balancing device is designed and the finite element analysis of self-balancing motorized spindle is carried out. The static analysis is carried out, and the static deformation diagram of spindle is obtained after constraint and load are applied. Then modal analysis and harmonic response analysis are carried out, and the first six natural frequencies, mode shapes and displacement frequency curves of spindle are obtained. The results show that: The deformation of motorized spindle is small and rigidity of spindle meets design requirements: Within the working speed range, the spindle will not resonate and work stably and reliably. The rationality and reliability of structure design of self-balancing motorized spindle are verified.

1. Introduction
As an executive part, motorized spindle unite is a very important part. Its main function is to drive the cutter to rotate at high speed to complete the cutting task. In the cutting process, the spindle unit mainly bears the cutting force and the driving force from the machine tool. Modern CNC machine tools have high power, fast cutting speed, high-speed spindle rotation, and little human intervention in the process of work, only by the reliability of the machine itself to ensure the quality. So static and dynamic characteristics of the spindle must be good. It depends on the reasonable structure design, the application of high-performance materials and the continuous optimization and improvement of spindle. Cutter is installed on the spindle for cutting, so the static and dynamic characteristics of spindle affect processing quality through the cutter. In addition, the speed characteristics of spindle and the service life of the spindle unit parts are closely related to dynamic characteristics of spindle.

Therefore, in the past few years a large number of scholars have studied static and dynamic characteristics and dynamic balance of spindle, and achieved a series of achievements. Dongju Chen et al. have studied some properties of spindle supported by hydrostatic bearings, and analyzed the influence of parameter changes on transshaped of spindle and eccentric vibration response. The analysis results show that the position and rigidity of bearings have a high influence on the machining accuracy [1]. Jorgensen et al. built a model of spindle system based on Timoshenko beam theory, taking into account factors affecting the dynamic performance of the spindle such as high-speed centrifugal force, cutting force and non-linear bearing rigidity. Then they analyzed the dynamics performance of the motorized spindle system. The analysis results indicate that the cutting force parameters, speed, etc. all affect the natural frequency of the spindle unit [2]. Shin et al. considered the thermal deformation of
the bearing, established the bearing load-deformation model, and combined the model with the discrete dynamic model of spindle to obtain some more accurate dynamic parameters [3]. Alfares neglected the centrifugal force and damping of the system by adopting the parametric discussion method, studied the influence of axial pre-tension force of angular contact bearing of spindle, and established a five-degree-of-freedom grinder spindle mathematical model [4]. Rodrigues put forward a method of dynamic balance regulation and control based on sphere as dynamic balance actuator [5]. Chen et al. studied a real-time dynamic balancing regulation and control based on influence coefficient method and analyzed the correlation between power transfer function and influence coefficient [7]. Yue et al. studied the multi-order multi-plane transient balancing method for high-speed rotor system and completed the correction of the third order unbalance [8]. Sarhan A A D et al. established three-dimensional model of FUSM milling machine by using software ANSYS Workbench. Natural frequency and vibration mode were obtained by finite element analysis. The experimental modal analysis of milling machine tool was carried out by using vibration testing system. By comparing the two analysis results, the rationality of model establishment and analysis of machine tool was verified [9]. Matsubara A used spring damping in radial direction and moment direction to simulate rolling bearings and established the mathematical model of lathe spindle system. The experimental confirmed that damping coefficient of the system is determined by damping moment and the natural frequency is determined by radial stiffness value [10]. Although a series of research achievements have been made on dynamic characteristics and dynamic balance problems, few research have been done on the dynamic and static analysis of the special structure of self-balancing spindle. So the special spindle which can install the dynamic balance head is analyzed dynamically and statically in this paper.

2. The establishment of the finite element model of self-balance motorized spindle

The self-balancing motorized spindle unit model has been established in SolidWorks. Considering the practical operability, after simplifying it, the three-dimensional model to be imported into ANSYS Workbench for finite element analysis is shown in figure 1.

![Figure 1. Three-dimensional spindle model.](image)

The material properties of the spindle need to be set in finite element analysis of the spindle. According to experience, the material properties of the spindle can be selected as 40Cr. Poisson's ratio, elastic modulus and density need to be set. Material properties are shown in table 1.

| Material Properties | Density($kg/m^3$) | Young’s Modulus(Pa) | Poisson’s Ratio |
|---------------------|-------------------|---------------------|----------------|
| 40Cr                | 7820              | 215000              | 0.3            |

For finite element analysis of spindle, grid partition is the key step. Grid partition includes selection of grid type, size of grid unit and number of node units. The quality of grid partition directly determines the reliability of calculation accuracy. Sparse partition easily leads to the calculation accuracy cannot meet the requirements, and too close will affect the calculation speed. In this paper, the method of combining automatic partition and size control is used for the mesh partition of the
spindle. Most of the meshes are hexahedron units and element size is controlled at 6mm. Figure 2 shows the finite element mesh model of spindle.

![Finite element meshing of spindle model](image)

**Figure 2.** Finite element meshing of spindle model.

3. **Static characteristic analysis of spindle**

3.1. **Apply constraints and loads**

In practical engineering, the bearing system will have inelastic deformation, which will affect the machining quality of the machine tool. In this paper, the elastic constraint in Ansys Workbench is equivalent to the elastic support, and the radial stiffness of the bearing is taken as the constraint stiffness value.

First, cylindrical constraint is added to the front and rear bearing segment surfaces of the motorized spindle. Cylindrical surface is restrained and two cylindrical surfaces are set free in radial and axial directions, with tangential restraint. Fixing constraints are added to the front bearing set of the spindle and the rear end face of the cylindrical surface of the spindle to which the front bearing set is installed is selected. Finally, elastic constraints are added to the simplified bearing set.

To apply a load is to apply a concentrated force to the front load acting surface of the spindle. Specific operation, select Loads->Force, then select the milling force action surface at the front of the shaft, set the load type to Components, Z-axis negative direction, the size of 200N. The applied restraints and loads are shown in figure 3.

![Restraints and loads](image)

**Figure 3.** Restraints and loads.

3.2. **Solution of Static Analysis**

The main manifestation of static analysis is the static deformation of the model under external load. The main shaft is solved under the static analysis module. Figure 4 and figure 5 show the static total deformation displacement diagram and the displacement nebulae in three directions respectively.
Figure 4. Static deformations diagram of the spindle: (a) X-axis to the static deformation maps; (b) Y-axis to the static deformation maps; (c) Z-axis to the static deformation maps.

Figure 5. Stress change diagram of the high speed shop spindle.
From the static total deformation diagram of the spindle, it can be seen that the deformation mainly is in the first hard of the spindle, while the deformation in other parts is small. Since the radial force 500N is applied in the Z-axis direction, the deformation of X-axis and Y-axis is small and that of Z-axis is large. The maximum value at 500N is expressed as $\delta=2.8 \times 10^{-7}$ m = 0.28 $\mu$m. Other deformations are smaller than this value and can be ignored. From the static stiffness calculation formula of the motorized spindle, it is clear that the stiffness value of the motorized spindle is reasonable under this small amount of deformation.

4. Modal analysis of spindle

The main requirement of high-speed machining center is high accuracy and speed. During the development process, it is inevitable to encounter deformation problems caused by vibration, which affects the machining quality and precision of high-speed machining center. Therefore, it is very important to comprehend the dynamic performance of the core components of the self-balancing spindle to improve the working ability of the whole motorized spindle. Since the dynamic performance of the mechanical model is mainly related to the natural frequency and the external excitation to the system. The dynamic performance analysis of self-balancing spindle starts with the modal analysis and the harmonic response analysis.

Modeling, material selection, meshing and constraints of modal analysis are the same as those of static analysis and are no longer analyzed. It is judged empirically that when studying the modal characteristics of the motorized spindle structure system, its low-order modal values have more influence on the vibration. Therefore, the first six modes are taken in this paper. The results of the modal analysis are shown in figure 6 and figure 7.

| Mode | Frequency [Hz] |
|------|----------------|
| 1    | 3884.6         |
| 2    | 3886.0         |
| 3    | 7773.9         |
| 4    | 8177.0         |
| 5    | 8177.9         |
| 6    | 8957.4         |

**Figure 6.** Natural frequencies of the spindle.

These results indicate that results that the overall vibration mode of spindle is relatively stable and reasonable, and the first-order frequency of spindle is 3884.6Hz, which is enough to meet the requirements of the dynamic and static stiffness of the spindle. Converting the natural frequency of the spindle to the first critical speed is: $n=60 \times 3884.6 = 2.3 \times 10^5$ r/min. Obviously, it is greater than the maximum speed. Therefore, the resonance will not occur within the normal working range of the spindle.

5. Harmonic response analysis of spindle

It is based on the modal analysis of spindle. When the spindle rotates to drive the cutting tool to process the workpiece, the workpiece will transfer a periodic excitation force to the spindle. When frequency of excitation force coincides with natural frequency of spindle, it will cause the resonance of the spindle and seriously affect the quality of the machining. Therefore, the resonance phenomenon must be avoided. Harmonic response analysis is very important part. The displacement and strain response curves of the important points on spindle at different frequencies can be obtained by it. As modeling process is consistent with modal analysis and the boundary constraints are also consistent with that of harmonic response analysis, the finite element model for modal analysis can be directly selected.
Figure 7. Sixth-order mode diagrams of the spindle: (a) first order; (b) second order; (c) 3\textsuperscript{rd} order; (d) 4\textsuperscript{th} order; (e) 5\textsuperscript{th} order; (f) 6\textsuperscript{th} order.
The magnitude of the excitation force loaded in this harmonic response analysis is 200N in Z direction. As a result of natural frequency of the sixth-order mode is 8957.4Hz in modal analysis. Therefore, harmonic response is studied by taking the excitation frequency of 0-90000Hz, as shown in figure 8.

![Figure 8. Front axle end face 0HZ-9000HZ frequency amplitude curve.](image)

It can be seen from the above diagram that the resonance frequency of the electro-spindle occurs near 9000Hz. So the image range is narrowed down to 8800-9100Hz, as shown in figure 9.

![Figure 9. Front axle end face 8800HZ-9100HZ frequency amplitude curve.](image)

The results show that the first peak value of the curve appears at the frequency 8956HZ of the external excitation load, i.e. the first resonance region, which is very close to the sixth-order natural frequency in modal analysis results of spindle. From the above analysis results of harmonic response, working speed of spindle is less than the speed at which resonance occurs. Spindle will not resonate and works stably and reliably.

6. Conclusion
Through static analysis of motorized spindle, static deformation diagram is obtained. The deformation of the motorized spindle is small under the load, and the structure meets the rigidity requirements. The modal analysis of the motorized spindle is done, and the first six order natural frequency and mode diagram are obtained. Finally, the harmonic response is analyzed and the displacement frequency curve is obtained. The results show that spindle will not have resonance, and the work is stable and reliable. Next, dynamic model of self-balancing motorized spindle can be established from the point of view of numerical analysis to study the influence of gyroscopic moment, bearing stiffness et al. on system dynamic performance.
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