ANALYSIS ON DYNAMIC CHARACTERISTICS OF SLIDE SEAT IN PRECISION MACHINING CENTER

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ABSTRACT: In order to analyze the static and dynamic characteristics of the workbench slide of the horizontal precision machining center, this study first establishes the finite element model of the slide, obtains the natural frequencies of the first four orders and modality vibration modes of the slide through the static analysis and vibration modal analysis of the slide structure using ANSYS Workbench finite element analysis software, and points out the weak links of its structure stiffness, based on which the slide structure is optimized. The analysis shows that the maximum deformation of slide of the machine tool is reduced by 10.7%, with good stiffness and dynamic performance. Finally, through the comparison of the accuracy of test piece before and after the optimization of the slide structure, it is shown that the optimization scheme for the lathe bed is correct and can achieve the goal of optimization, thus providing a strong theoretical basis for the subsequent improvement of the machine tool.

KEY WORDS: slide, dynamic performance, structural optimization

1 INTRODUCTION

With the rapid development of modern manufacturing industry, higher and higher requirements are put forward for the dynamic performance of machine tools. In recent years, a great deal of research has been done on the dynamic design of machine tool structure at home and abroad. With the dynamics of the feed system and the workbench of the machining center as research object, Qiu Zheng, Zhang Song, et al., set up the finite element model of the feed system, conduct analysis using ANSYS Workbench software, and optimize the dimension of the workbench with improvement of its natural frequency as the goal (Liu et al., 2017). In the dynamic design of high-speed horizontal machining center, Wu Xiuhai et al. obtain the characteristic parameter of guide rail joint by dynamic testing method and apply it to digital simulation model, thus improving the precision of the model. In the structural optimization design of the machining center, the topology optimization of the main components improves the static and dynamic characteristics of the machining center (Li et al., 2011). Han Kun, Gao Dongqiang, et al. others take the high-speed vertical machining center slide as the research object, improve the original model based on the lightweight porous structure, perform the static analysis and modal analysis of the slide using finite element analysis software, and finally improve the dynamic performance of the slide by optimizing the slide of the lightweight porous structure (Han et al., 2014). This paper takes the slide of a large-scale horizontal machining center as the research object. On the one hand, slide is used as the support of the workbench and the moving part of the servo feed system, whose deformation directly affects the precision of the rotary feed and linear feed of the workbench (Xia et al., 2011; Zhang et al., 2008; Zhang et al., 2009), affecting the precision and stability of the machining tool accordingly. On the other hand, due to lack of support for the dynamic characteristic analysis in the traditional experience design of the workbench slide, the slide has such problems as vibration and unreasonable distribution of structures. Therefore, it is necessary to dynamically analyze the slide of the horizontal machining center and optimize its structure based on this. In this paper, the three-dimensional model of slide is established, the static and dynamic analysis of slide structure is carried out by finite element software ANSYS Workbench, and the slide structure is optimized according to the analysis results (Abd El Hady et al., 2016).

2 STATIC ANALYSIS

2.1 Foundation of static analysis

Linear static structural analysis is used to analyze the response of a structure under a given static load, focusing on the parameters such as displacement, restraint reaction, and stress and...
strain of the structure. In classical mechanics theory, the general equation for the dynamics of an object is:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}(t)$$  \hspace{1cm} (1)

Where, \(\mathbf{M}\)-mass matrix;
\(\mathbf{C}\)-damping matrix;
\(\mathbf{K}\)-stiffness coefficient matrix;
\(\mathbf{x}\)-displacement vector;
\(\mathbf{F}\)-force vector.

This study constructs the finite element model of the structure of the slide of precision machining center and conducts the linear static structure analysis to determine its stress distribution and static deformation, so as to find out the weak links of the machine tool, providing a theoretical basis for optimizing the structural design and improving the static stiffness of the machine tool.

In general, the load applied by the finite element analysis is mainly the external force, regardless of the effects of inertia and damping. However, for inertial loads (such as gravity and centrifugal forces) and time-varying loads that can be approximated as equivalent static forces, loading is performed only when these loads need to be taken into account. At the same time, all the time-dependent variables are ignored in the linear static structure analysis. Thus, the following equation is obtained from equation (1):

$$\mathbf{K}\mathbf{x} = \mathbf{F}$$  \hspace{1cm} (2)

### 2.2 Establishment of the finite element model

The workbench slide of the horizontal machining center is a large piece supporting the workbench, which is equipped with a transmission device, a clamping device, a bearing, and the other structures. The slide studied in this paper and workbench is connected by a conical pin cylinder clamping device, which can be considered as a rigid connection. The workbench slide is cast from high strength grey cast iron HT250, with the dead weight of the part of 753kg and a ring-shaped reinforcement. There is a radial transverse reinforcement in the middle, with the thickness of gusset of 25mm, and the thickness of the guide rail support surface of 50mm. Its structure is shown in Figure 1. Taking the elastic modulus of 1.30E+11Pa and Poisson's ratio of 0.25, the mesh is divided freely. After division, there are 162,321 nodes and 86,001 cells in total, and the mesh generation is shown in Figure 2.

![Figure 1. The structure of the slide](image)

![Figure 2. Simplified finite element model of slide](image)

### 2.3 Static analysis results

According to the research content of the subject and the actual working conditions of the machine tool (the upright column is at the middle position of X-axis travel, slide is near the limit position at the rear end of Z-axis travel, and the spindle box is near the limit of 150mm at the lower end of Y-axis), the horizontal machining center is mainly subject to its own gravity, maximum bearing load of the workbench, and cutting force, while neglecting the influence of other attachment forces. The undertaken force is shown in Table 1.
Table 1. Dead weight and load of components of the horizontal machining center

<table>
<thead>
<tr>
<th>Project</th>
<th>Unit</th>
<th>Force</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bed component gravity $G_1$</td>
<td>N</td>
<td>90 000</td>
<td>Chip conveyor</td>
</tr>
<tr>
<td>Column assembly gravity $G_2$</td>
<td>N</td>
<td>25 000</td>
<td></td>
</tr>
<tr>
<td>Spindle assembly gravity $G_4$</td>
<td>N</td>
<td>4 500</td>
<td></td>
</tr>
<tr>
<td>Workbench slide and table assembly gravity $G_5$</td>
<td>N</td>
<td>9 500</td>
<td>Workbench center</td>
</tr>
<tr>
<td>Maximum load-bearing of worktable $G_6$</td>
<td>N</td>
<td>12 000</td>
<td>Maxload of worktable</td>
</tr>
<tr>
<td>Exchange desk component gravity $G_7$</td>
<td>N</td>
<td>26 000</td>
<td></td>
</tr>
<tr>
<td>cutting force $F_C$</td>
<td>N</td>
<td>10 000</td>
<td>Max cutting resistance</td>
</tr>
</tbody>
</table>

Figure 3. Dead weight and load of the horizontal machining center

When a load is applied to the workbench slide, the slide is fixed first and a fixed support is applied to the inner cylindrical surface of the 24 bolt holes connected to the slide on the workbench. In accordance with the dead weight and load conditions of the horizontal machining center component in Table 1 and the load conditions shown in Figure 3, the dead weight is applied to the slide model of the workbench by the Standard Earth Gravity command, the workbench’s gravity is applied with mounting surface of the gyrator of the workbench slide as the application surface, and the workpiece weight $G_6$ is applied on the same application surface. The condition after application of the load is shown in Figure 4. According to the same analysis steps, the final comprehensive deformation of the working slide obtained is shown in Figure 5.

Figure 4. Slide load

Figure 5. Comprehensive deformation of slide

Through the analysis of the comprehensive deformation, it can be seen that the maximum deformation point of workbench slide is in the mounting seat of ball screw nut, and the maximum deformation is 0.0303mm. The deformation is due to the maximum cutting resistance of the spindle and is also related to the screw drive system. This deformation will affect the geometric precision of the machine tool and the machining precision of the parts directly, so when designing the ball screw nut joint surface, it shall be strengthened. In addition, it can be seen that the number of studs in stud cavity in the middle layer of the workbench slide is not enough, resulting in larger deformation of the bearing or chain wheel mounting surface. In the subsequent structural optimization design, these problems shall be improved in order to obtain better performance of the structure.
3 MODAL ANALYSIS

3.1 Theories and methods of modal analysis

Modality is the natural vibration characteristic of mechanical structure, and each modality has its corresponding natural frequency, damping ratio and modality vibration mode. Modal analysis refers to the analysis and calculation process, in which by converting the physical coordinates of the linear time-invariant system to the oscillatory differential equations into modal coordinates, the equations are decoupled into a set of equations that are independent of each other and are expressed with modal parameters and modal coordinates so as to obtain the modal parameters of the system. Its essence is to solve the modal vector of the motion equation with a finite degree of freedom under the conditions of no damping and no external load.

The natural frequency and vibration mode are important parameters in the design of dynamic load carrying structures, while modal analysis can determine the natural frequency and vibration mode of a structure. As for the precision horizontal machining center studied in this paper, through the modal analysis of the machine tool slide, the natural frequency and vibration mode of the structure are determined, so that the vibration characteristics of the system and the response of the structure to various dynamic loads are clearly defined, providing a theoretical basis for the study of the vibrations of the horizontal machining center at work and the optimized design of the dynamic characteristics of the structure.

For modal analysis, free vibration is assumed and damping is ignored (ignore $c$, and $\{F(t)\}=0$). According to the general equation of dynamics of an object, the following equation is obtained from equation (1):

$$[M]\ddot{u} + [K]u = 0$$

(3)

When the harmonic vibration occurs, that is, when $u = U \sin(\omega t)$, the equation is:

$$([K] - \omega^2 [M])\phi_i = 0$$

(4)

Where, $\omega_i$-inherent circumferential frequency;

$\phi_i$-inherent vibration mode, which is a relative value, not an absolute value.

3.2 Slide modal analysis

The dynamic performance of the machining center slide reflects the vibration resistance of the structure when subject to dynamic loads, which has an important influence on the accuracy of the machine tool. Higher-order modality has a higher damping value, and their role in the vibration mode analysis is relatively small. Therefore, the general modal analysis mainly focuses on the lower-order modality with a relatively large impact on the vibration mode (Wu et al., 2010). This paper applies ANSYS Workbench to the modal analysis of slide, gives the first four-order modal analysis results under full constraint, obtains the natural frequencies of the first four orders of lathe bed and the corresponding vibration modes, as shown in Figure 6.

![Figure 6. Vibration modes of the first four order modality of slide](image)
Table 2. Natural frequencies and vibration modes of the first four order modality of slide

<table>
<thead>
<tr>
<th>Modal order</th>
<th>Natural frequency (Hz)</th>
<th>Mode shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>First order</td>
<td>595.27</td>
<td>The left and right ends are tilted around the X axis</td>
</tr>
<tr>
<td>Second order</td>
<td>661.98</td>
<td>The left and right ends are twisted around the X axis</td>
</tr>
<tr>
<td>Third order</td>
<td>716.82</td>
<td>The mounting end of the nut is twisted around the X axis</td>
</tr>
<tr>
<td>Fourth order</td>
<td>800.44</td>
<td>The front and rear ends are twisted around the Z axis</td>
</tr>
</tbody>
</table>

The following results can be obtained from the first four order vibration modes of the workbench slide. Maximum vibration appears at the top end of the workbench slide and the mounting seat of ball screw nut. The machine tool guide rail slider, the mounting surface of the workbench slider and the adjacent stud cavity are the weak links, which is reflected as the twisting and tilting of the top end and the mounting seat of ball screw nut in the vibration mode. The main reason for this is the insufficient stiffness of the mounting seat of ball screw nut, and the unreasonable setting of the stud cavity or study at the top end of slide, which needs to be reinforced.

4 STRUCTURAL OPTIMIZATION

The mounting seat of ball screw nut and the mounting face of chain wheel of workbench slide of the trial machine are all weak links. The optimized design of the weak links of the trial machine is carried out without making major changes to its structure. Through repeated design, modeling and analysis of the slide, the optimal scheme is drawn up as follows:

Under the permit of the lathe bed space, the thickness of the mounting seat of ball screw nut and the gusset on both sides is increased, and a 25mm thick stud is added to the weak link at the bottom of the chain wheel mounting surface. The specific changed structure is as shown in Figure 7.

![Figure 7. 3D model of the optimized slide structure](image)

Figure 8. Comprehensive deformation of setting machine slide

As can be seen from the Figure 8, the maximum deformation position of the workbench slide in the setting machine scheme is consistent with that of the trial machine, and the maximum deformation is reduced by 10.7%, thus the goal of optimizing the workbench slide to reduce the maximum deformation is achieved. The reduction of stress and the natural frequency of the first four orders (the natural frequency of the fourth order is increased) is within the acceptable range, so the direction of optimization is accurate and the optimization scheme is feasible. The vibration modes of the first four orders of the setting machine slide of horizontal machining center are shown in Figure 9 and its performance before and after optimization are shown in Table 3.

![Figure 8(a). First order](image)

![Figure 8(b). Second order](image)
Figure 9. Modality vibration modes of the first four orders of the setting machine slide

5 PRECISION TEST OF PRECISION MACHINING TEST PIECES

In order to verify the results of the above-mentioned theoretical analysis, the precision test of the test pieces of the same specification is performed using the trial machine and the setting machine respectively. The test object is the precision machining of end milling. There are 3 end milling test pieces for the trial machine and 3 for the setting machine, which meet the machining and testing modes. The precision test value takes the arithmetic mean of the test results.

Table 3. Comparison of slide performances before and after structural optimization

<table>
<thead>
<tr>
<th>Project</th>
<th>Trial machine scheme</th>
<th>Setting machine scheme</th>
<th>change rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max deformation (mm)</td>
<td>0.010356</td>
<td>0.0089</td>
<td>-10.7%</td>
</tr>
<tr>
<td>First order (Hz)</td>
<td>595.27</td>
<td>574.21</td>
<td>-3.5%</td>
</tr>
<tr>
<td>Second order (Hz)</td>
<td>661.98</td>
<td>637.15</td>
<td>-3.7%</td>
</tr>
<tr>
<td>Third order (Hz)</td>
<td>716.82</td>
<td>708.36</td>
<td>-1.1%</td>
</tr>
<tr>
<td>Fourth order (Hz)</td>
<td>800.44</td>
<td>824.58</td>
<td>+3.0%</td>
</tr>
</tbody>
</table>

The material of the end milling test piece is 45 steel, with a face width of 100. The test piece and its machining schematic diagram are shown in Figure 10.
Tool: Indexable sleeve type φ120 face milling cutter with 8 teeth.

End milling parameters are as shown in Table 4.

<table>
<thead>
<tr>
<th>Project</th>
<th>Trial machine scheme</th>
<th>Setting machine scheme</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cutting speed (mm/min)</td>
<td>300</td>
<td>300</td>
<td></td>
</tr>
<tr>
<td>Cutting depth (mm)</td>
<td>0.2</td>
<td>0.2</td>
<td></td>
</tr>
<tr>
<td>Cutting width (mm)</td>
<td>80</td>
<td>80</td>
<td></td>
</tr>
</tbody>
</table>

Test results are shown in Table 5.

<table>
<thead>
<tr>
<th>Project</th>
<th>Permissible error</th>
<th>Trial machine scheme</th>
<th>Setting machine scheme</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flatness (mm)</td>
<td>0.02mm</td>
<td>0.015</td>
<td>0.010</td>
</tr>
</tbody>
</table>

It can be seen from Table 5 that after the slide structure is optimized, the precision of the end milling test piece of the machining center of the setting machine is higher by 0.005 mm than that of the trial machine. It is shown that the optimization scheme for the lathe bed is correct and can achieve the goal of optimization, thus providing a strong theoretical basis for the subsequent improvement of the machine tool.

6 CONCLUSIONS

Through the analysis of the static and dynamic characteristics of slide, this study obtains the slide deformation under different frequencies of vibration mode, points out the weak links of structural stiffness, and analyzes the reasons for those weak links. On this basis, the structure of the slide is optimized for improving the stiffness of the slide, and the reasonable structure is determined by comparing the mechanical properties of slide structure before and after optimization. The analysis shows that the maximum deformation of slide is reduced by 10.7%, which achieves the optimization goal. Finally, the accuracy of structural optimization design is verified by comparing the precision of precision machining test pieces of the setting machine with that of the trial machine.

7 ACKNOWLEDGEMENTS

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8 REFERENCE