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A NOVEL LIGHTWEIGHT OPTIMIZATION DESIGN OF GEAR SHAPER

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The machine produces vibrations under the excitation of the cutting load and the non-equilibrium inertia of the actuator and actuator, which affects the machining accuracy and reduces the processing efficiency. A dynamic design method of machine tool structure is put forward and studied to ensure the dynamic properties of machine tool structure and realize the machining precision which improved machine tool weight is reduced by 18.3%, the machine natural frequency is increased by 2.1%, which improves the machine Performance, and access to the final design.

Keywords: Machining accuracy, Finite element analysis method, dynamic characteristics, dynamic characteristics analysis, optimization design

1. Introduction

At present, the research on dynamic performance of machine tool mainly focuses on dynamic analysis and optimization design. The dynamic models are the basis of dynamic analysis, and the traditional design methods of machine tools are mostly experience design or imitation design, focusing on and meeting the static performance requirements of machine tools. However, the increasingly diversified demands of mechanical products have increasingly higher requirements on the flexibility, precision and other performance of machine tools. The natural dynamic characteristics of machine tools mainly include the natural frequency, which has a significant impact on the seismic performance of machine tools, processing efficiency. Among the many factors affecting the design of large foundation parts, the machine tool weight is the key factor affecting the performance of the machine tool, especially related to the natural frequency of the machine tool. In addition, the unreasonable distribution of machine tool weight will lead to uneven distribution of machine tool stiffness, and greatly affect the machining performance of the machine tool.

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Considering that the traditional machine tool design method usually endows the machine tool structure with a large safety factor, which leads to excessive machine tool structure size and mass, affecting the dynamic performance of the machine tool. Therefore, Scholars and research institutions around the world have carried out a lot of research on the dynamic properties of machine tools. Rizy [1] used the distributed mass model to reveal the dynamic characteristics of the equipment, but it lost the simulation accuracy for its machine structure is too complex to simplify the components into beam unit. Jiang and Chiredast [2] of the University of Michigan in the United States on the basis of the application of dynamic analysis and finite element method, used mathematical models to simulate the connection form of the machine tool structure, they established the machine tool dynamics model, and created the topological optimization design of the number and location of machine tool joint surface connection. The finite element model still has superiority in today's machine dynamics analysis and can be a good response to the actual situation, but it cannot be used in machine tool dynamics design; Schmitz [3] and Davies [4] from American National Institute of Standards and Technology with analytical and experimental combination of parameter identification technology, established high-speed machine tool fixture - tool - spindle system model, which can predict the dynamic response of the system. There are a number of universities and research institutes which have conducted a study on machine dynamics modeling in China. Southeast University, Chen Xin and Sun Qinghong [5] used the ANSYS software to establish machine structural dynamics model of the high-precision internal grinding machine M2120A and made dynamic analysis.

A dynamic design method of machine tool structure based on process system was presented. Firstly, a dynamic physical model was established based on the process system to reveal the mapping relation of dynamic characteristics of machine tools under forced vibration. Then taking the main structure of the machine tool as an example, the variable density method was introduced to discretize the structure, and the dynamic model of the machine tool structure related to the density distribution and the prototype design of the corresponding structure were established. At last, the key design variables were extracted by sensitivity analysis. Based on the prototype of the machine tool structure, the dynamic analysis model and the optimization model of the machine tool structure were established to solve the problem that the machining precision of the machine tool can be optimized, and the lightweight design of machine tool was realized.

2. Modeling of Machine Tool Process System Model

2. 1 Establishment of physical model

Machine tools are composed of multiple parts of the complex mechanical system, in order to improve the dynamic properties of the machine, one or more parts cannot be considered in a single way. In order to obtain the exact data of the relative displacement between the tool and the workpiece, Figure 1 shows a physical model of the "Machine-Tool-Workpiece" system. Moreover, Figure 2 is a dynamic model for revealing the interaction of the machine in the free vibration, forced vibration and self-excited vibration of the "Machine-Tool-Workpiece".

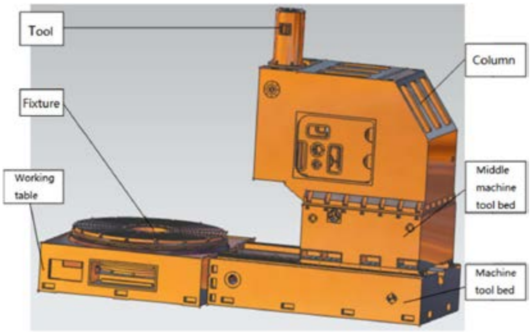

Fig. 1. Physical model of “Machine-Tool-Workpiece” system of gear shaping

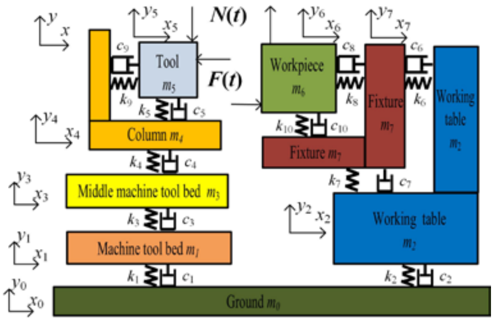

Fig. 2. Dynamic physical model of “Machine-Tool-Workpiece” system of gear shaping

m is mass matrix, *c* is damping matrix, *k* is stiffness matrix, *y(t)* is displacement vector, *F(t)* is external excitation vector.

2. 2 Establishment and solution of mathematical model

The mathematical model, obtained based on the mechanical vibration theory according to the physical model of the process system, is given by

$$M\ddot{y}(t) + C\dot{y}(t) + Ky(t) = f(t) \tag{1}$$

M is the mass matrix, *C* is the damping matrix and *K* is the stiffness matrix; $\ddot{y}(t)$, $\dot{y}(t)$, $y(t)$ and $f(t)$ are respectively for the acceleration vector, the velocity vector, the displacement vector and the external excitation vector.

According to basic data of machine parameter, multi-group machine parameters are obtained by orthogonal experiment method. And the Newmark-β method [6] can be used to solve the vibration differential equation response of machine tool system model. The Newmark-β method is an unconditionally stable implicit integration format, which calculates the displacement response, and then obtains the relative displacement response of the workpiece and the tool at the tool point.

Combined with model parameters (the parameters are shown in Figure 2):

$$\begin{cases} m_1\ddot{y}_1 = k_1(y_0 - y_1) + c_1(\dot{y}_0 - \dot{y}_1) - k_3(y_1 - y_3) - c_3(\dot{y}_1 - \dot{y}_3) \\ m_2\ddot{y}_2 = k_2(y_0 - y_2) + c_2(\dot{y}_0 - \dot{y}_2) - k_7(y_2 - y_7) - c_7(\dot{y}_2 - \dot{y}_7) \\ m_3\ddot{y}_3 = k_3(y_1 - y_3) + c_3(\dot{y}_1 - \dot{y}_3) - k_4(y_3 - y_4) - c_4(\dot{y}_3 - \dot{y}_4) \\ m_4\ddot{y}_4 = k_4(y_3 - y_4) + c_4(\dot{y}_3 - \dot{y}_4) - k_5(y_4 - y_5) - c_5(\dot{y}_4 - \dot{y}_5) \\ m_5\ddot{y}_5 = k_5(y_4 - y_5) + c_5(\dot{y}_4 - \dot{y}_5) - F(t) \\ m_6\ddot{y}_6 = k_{10}(y_7 - y_6) + c_{10}(\dot{y}_7 - \dot{y}_6) + F(t) \\ m_7\ddot{y}_7 = k_7(y_2 - y_7) + c_7(\dot{y}_2 - \dot{y}_7) - k_{10}(y_7 - y_6) - c_{10}(\dot{y}_7 - \dot{y}_6) \\ m_5\ddot{y}_5 = k_9(x_0 - x_5) + c_9(\dot{x}_0 - \dot{x}_5) - N(t) \\ m_6\ddot{y}_6 = -k_8(x_6 - x_7) - c_8(\dot{x}_6 - \dot{x}_7) + N(t) \\ m_7\ddot{y}_7 = k_8(x_6 - x_7) + c_8(\dot{x}_6 - \dot{x}_7) - k_6(x_5 - x_6) - c_6(\dot{x}_5 - \dot{x}_6) \end{cases} \tag{2}$$

2.3 Establishment of BP neural network model

When the neural network model shown in Figure 3 is used to predict the relative displacement response of the tool point. In order to improve the dynamic properties of machine tools and quickly find the dynamic parameters of machine tools with less response to the tip displacement, it is necessary to carry out forward prediction and reverse multi-objective optimization for machine tools.

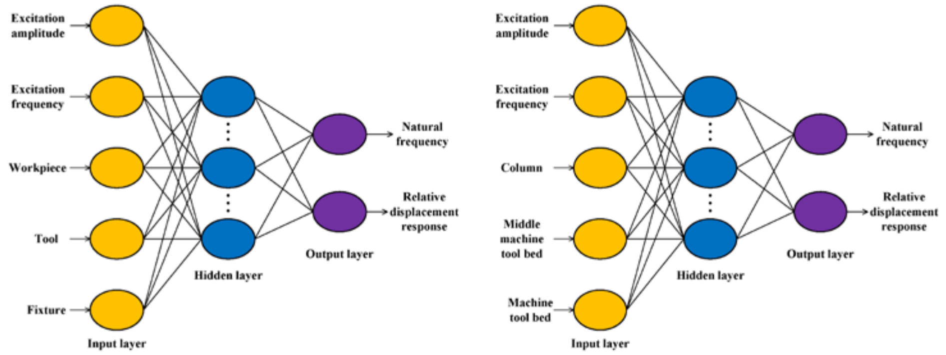

Fig. 3. BP neural network model of X and Y direction of machine tool system

The strong self-learning and self-adaptive ability of artificial neural network is suitable for solving the problem of dynamic parameters of machine tool according to the relative displacement response of tool tip. By using the artificial neural network model, the mapping relationship between the relative displacement response of the tool tip and the dynamic parameters of the machine tool can be established. The data of multi-group machine tool parameters and corresponding displacement response and natural frequency results are taken as samples of the neural network.

3. Machine tool original structure modeling

Topology optimization is a kind of optimal topology that satisfies design constraints in a given structural design region, based on known constraints, loads, and boundary lines. Through scientific optimization calculations, seeking to meet the optimal topology design constraints, it is an effective method to guide the structural design of the concept. At present, the method of homogenization of continuum topology optimization is as follows: homogenization method [7-8], variable density method [9], independent continuous mapping (ICM) [10] and progressive structure optimization method [11]. Based on the conceptual configuration of the gear shaping machine, the distribution density is introduced into the structure, the structure is discretized, and the machine structure dynamics model related to the distribution density is established. The modal analysis is carried out, that is, the fundamental frequency of the workbench structure. The maximum value is used as the objective function, and the volume is used as the constraint. Under the condition that the performance requirement of the bed is satisfied, the optimal mass distribution of the workbench that satisfies the dynamic characteristics can be solved. Then get the prototype design of machine tool structure; Based on the concept design prototype, the original structure of gear shaping machine is designed.

3. 1 Topological optimization of structural dynamics for workbench

The structural performance of the workbench directly affects the performance of the machine. This section will use the relative density method of CNC gear shaper workbench for dynamic topology optimization to solve the maximum mass distribution satisfying the dynamic characteristics of the workbench, and the design flow is as follows in Fig. 4.

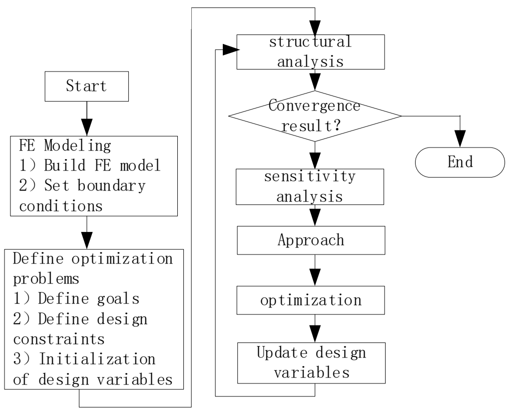

Fig. 4. Design process of structural dynamic topology optimization

3. 2 Variable Density Method (VDM)

In this section, the VDM is used to optimize the dynamic topology of the structure model of the workbench, and the optimal weight distribution of the workbench satisfying the dynamic characteristics is solved under the condition that the constraints and the influence of the bed and other parts installation are considered. Primarily, refine the original model and divide the workbench into devisable areas and undesignable areas which are the main part of the component and should not be optimized. The segmentation result is shown in Fig. 5.

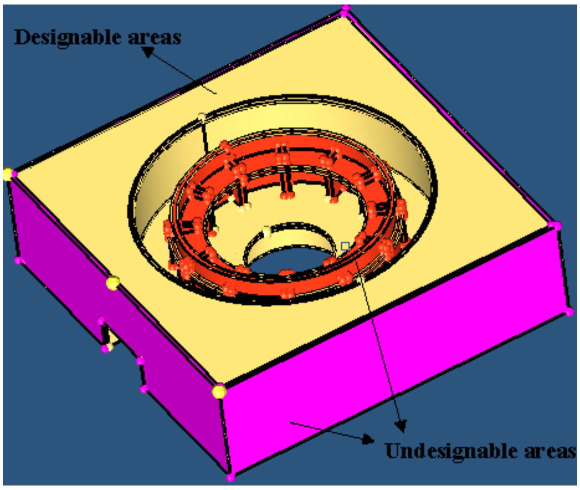

Fig. 5. Divided topological optimization model

After optimization of the design variables, objective functions, and constraints by defining the machine bed, the HyperWorks/Optistruct modal analysis module is automatically optimized. As can be seen from Fig. 6, the blue part belongs to the area of low-density unit, which is mainly concentrated around the work table and is the part that is recommended to be removed.

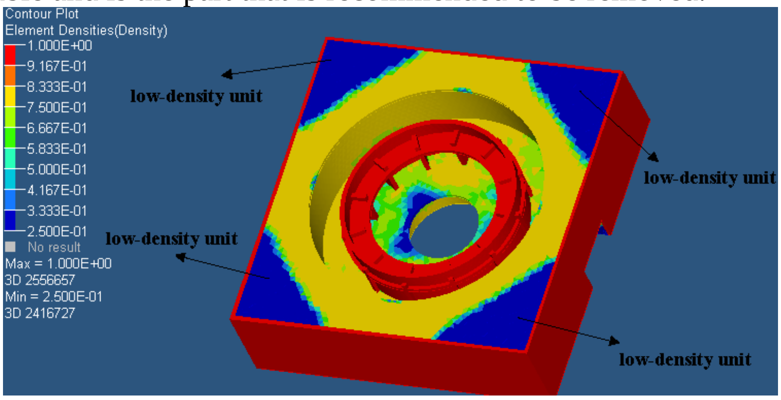

Fig. 6. Dynamic topology optimization density contours of workbench model

As shown in Fig. 7, when the unit density is set to 0.3, the low-density units of the workbench are mostly hollowed out.

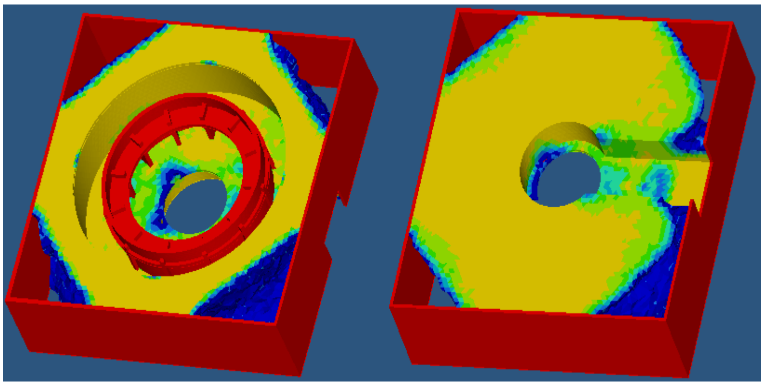

Fig. 7. Topology optimization result at 0.3 relative density

From the optimization results, in order to ensure the optimization of the results of process ability, it needs optimal design of the secondary design. In the second optimization design of the table structure, the weight is concentrated in the middle part while lowering the weight of the surrounding part. Therefore, the improvement of the central and peripheral portions of the structure is able to be used for improving the dynamic performance of the whole structure of the machine bed. Based on the analysis of the optimization results, the HyperGesh post-processing OSSmoth module is used to derive the. iges format file. Considering the assembly relationship between the constraint and the machine bed and the ease of processing of the machine bed, finally, we get the modified bed optimization model through the three-dimensional modeling software UG secondary design. The modified topology optimization model of workbench is shown in Fig. 8.

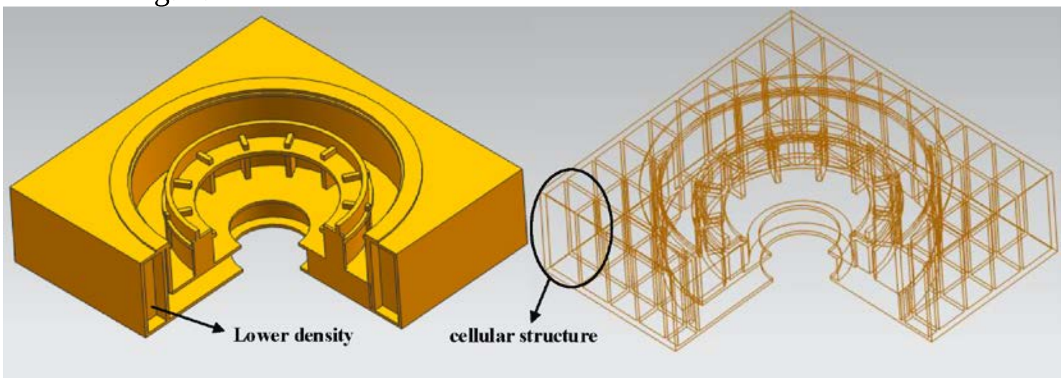

Fig. 8. Modified table topology optimization model

3. 3 Sensitivity analysis of dynamic performance

The sensitivity analysis of dynamic performance of machine tool is to select the key design parameters which have significant influence on the natural frequency through analysis. After establishing the parametric finite element analysis model of each part of the gear shaper, the sensitivity of the structural dynamic performance to the structural parameters is defined as the ratio of the structural dynamic performance increment, considering that the height of the workbench is not in the same order of magnitude as the thickness of the plate. For ease of comparison, the sensitivity is normalized. The proportional expression can be expressed as:

$$SI_{ij} = \left|\frac{\Delta f_i}{\Delta \xi_j}\right| \Bigg/ \sum_j \left|\frac{\Delta f_i}{\Delta \xi_j}\right| \times 100\% \tag{3}$$

$\Delta\xi_j$ is the relative increment of the structural parameters; Δf_i is the dynamic performance increment.

The ratio of the mass increment caused by the increment of configuration parameter to the relative change value of this variable is used as the sensitivity index of weight performance analysis. Then, according to the sensitivity of weight performance, the key structural parameters related to weight performance are selected. Considering the sensitivity of variables to the natural frequency and weight, the key structural parameters were selected as the design variables for the structural optimization of the gear shaping machine. The sensitivity analysis results of the workbench are shown in Fig. 9.

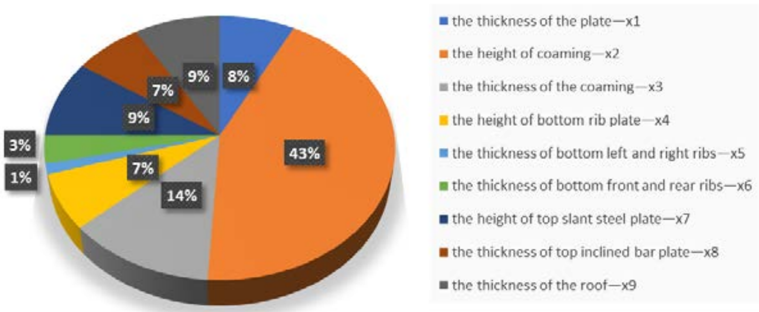

Fig. 9. Sensitivity of the natural frequency for table size parameters

4. Structure Optimization Design of Gear Shaper

4.1 Mathematical model of tablebench natural frequency and weight

According to the sensitivity analysis, the 3 parameters with higher sensitivity are determined. Moreover, PYTHON was used to establish the parameter table model, and input 20 tests of CCD to get the weight of each group of workstations.

Considering the nonlinear relation between the natural frequencies and the structural parameters of the stage, the second-order response surface model is obtained by using the response surface method. The prediction equations of the natural frequency, mass and work structure parameters are as follows:

$$y = 94.815 + 0.679x_1 + 0.529x_2 - 0.071x_3 - 0.00088x_1x_2 + 0.0001x_1x_3 + 0.0003x_2x_3 - 0.0056x_1^2 - 0.0038x_2^2 - 0.0003x_3^2 \quad (4)$$

$$y' = 8.392 + 0.048x_1 + 0.0296x_2 + 0.0137x_3 - 0.0000125x_1x_2 + 0.0000125x_1x_3 + 0.0000125x_2x_3 + 0.0000011x_1^2 - 0.000017x_2^2 + 0.0000011x_3^2 \quad (5)$$

Five groups were randomly selected from the 20 groups of experimental data whose experimental data arranged by CCD test design method; the regression equation was derived under the corresponding conditions. After the regression value is obtained, the theoretical mass value is compared, and the error analysis is shown in Table 1.

Table 1

The comparison of the measured values and the return value

No.	x_1/mm	x_2/mm	x_3/mm	Regression value/10^3kg	Theoretical value/10^3kg	Relative error/%
1	23.18	46	48	11.520	11.520	0
2	40	46	64.82	12.5796	12.580	0.003
3	40	46	48	12.3290	12.330	0.008
4	40	29.18	48	11.8504	11.850	0.003
5	30	56	58	12.2782	12.280	0.015

4.2 Structural optimization design based on response surface model

The natural frequency and weight structure parameters of the workbench can be optimized by using the genetic algorithm of non-dominant sequencing [12]. The partial results of 200 sets of Pareto solutions were listed to maximize the natural frequency and minimize the weight. The results show that the maximum natural frequency is not the same as the minimum mass in Table 2, that is, the machine tool cannot achieve both the maximum natural frequency and minimum weight of the dual goals. The natural frequency and weight of the Pareto solution set are taken into account to find the optimum structural parameters.

Table 2

Extremum points of mass and natural frequency

Variable	x_1/mm	x_2/mm	x_3/mm	Natural frequency/Hz	Workbench weight/10^3kg
The minimum value of weight	30.06	36.42	38.08	121.69	11.430
The maximum value of natural frequency	49.82	55.97	38.04	129.90	12.930

In order to obtain a more feasible solution for the natural frequency of the machine, we also use the NSGA-Ⅱ algorithm to optimize the Pareto solution of the machine structure and performance parameters. See the brightest point shown in Fig. 10 for the Pareto optimization solution.

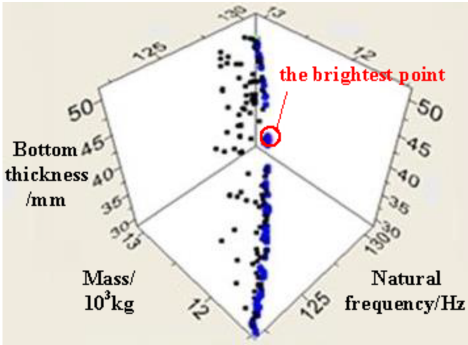

Fig. 10. Pareto3D solutions of the natural frequency and mass optimization design

4.3 Optimization of Design and Simulation of Natural Frequency and Weight of Prototype

In addition, data collection was conducted through lmstest.lab in this paper, and self-spectrum, transfer function and coherence analysis of excitation force signal and acceleration response signal were conducted to obtain the transfer function of each point. Finally, the test data of each measurement point are imported into Modal Analysis module, and the curve fitting of transfer function is carried out to obtain the natural frequency of the machine tool and its large parts.

Fig. 11. LMS Test.Lab10B modal analysis system & Impact hammer

To the structure of light weight and natural frequency to maximize the target, the design model of the multi - objective optimization design of the slotting machine is established. By solving the optimization model, we will obtain the design scheme of the main structure of the gear shaper with high cost performance. The comparison between the optimized results and the initial prototype in terms of dynamic performance and weight is shown in Table 3.

Table 3

Comparison of optimization design result and initial data

Variable	Machine bed	Middle machine Bed	Column	Workbench	Whole machine
Prototype weight/10^3kg	10	6	9	15	40
Optimized weight /10^3kg	8.67	5.12	7.47	11.43	32.69
Weight reduction/%	13.3	14.7	17	23.8	18.3
Prototype frequency/Hz	201	392	84	126	23. 1
Optimized frequency/Hz	204.17	397.71	89.42	129.90	27.86
Frequency increase/%	1.6	1.46	6.5	3.1	2.1

From the comparison results we can see that the overall weight of the machine after optimization reduced by 18.3%, the machine natural frequency increased by 2.1%.

5. Conclusions

Based on the "Machine-Tool -Workpiece" process system model, the dynamic method of machine tool structure has been put forward and constructed. The topology optimizes the weight distribution of the machine tool structure by introducing the distribution density and discretizing the structure. In addition, multi-objective genetic algorithm NSGA- Ⅱ has been used for multi-objective optimization between machine weight and first-order natural frequency, which gets optimal solution. Finally, experimental results show that the mass distribution obtained by topology optimization can greatly reduce the weight of machine tools, thus improve the natural frequency of machine tools, improve the life and stability of machine tools.

Acknowledgment

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R E F E R E N C E S

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