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Dynamic Analysis and Optimization of WEDM Based on AWE and LMS

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Abstract

In the operation process,the Wire Electrical Discharge Machine (WEDM) has certain imperfections such as vibration and the descent of machine precision which vibration produces.This paper studies the dynamic parameter of the machine tool and optimizes the natural frequency and vibration. Taking the DK7725 taper machine tool as an example, the paper establishes a 3Dmodel with ProEngineer 5.0. According to the Masataks Yoshimura method, the authors could ascertain the stiffness and damping of joint surfaces among machine main parts and ascertain the equivalent dynamic model. In order to have a modal analysis about the machine tool structure, the virtual dynamic analysis module of ANSYS Workbench Environment (AWE) is used. Through the study of dynamic parameter, the authors optimize and improve the natural frequency and vibration of machine tools, compared with the finite element analysis results and the no-optimization data.And the final results show that the change rates of each order natural frequencies optimized ranges from 0%to18.9%,and the whole mechine's optimization achieves satisfied effect.

Keywords: *joint surface, equivalent model, ansys workbench environment (AWE), modal analysis, LMS test, optimization*

1. Introduction

With the rapid development of industrial technology,the important role of wire electrical discharge machine is more and more prominent. High precision,high efficiency and high quality processing performance become the target of wire-cutting technology. Because of the close relationship between machine's processing performance and the machine's dynamic characteristics, the authors need to have a thorough dynamics analysis of machine structure. Dynamics also have a close relation with the surface, and research shows that over 60% dynamic flexibility of machine tools is from the joint surface [1]. The stiffness and damping of the joint surface has a great effect on the dynamic performance of machine tool, so joint surface can be found in many mechanical structures.

There still have some problems of DK7725 Taper machine when it processes taper parts. For example, the machine tool vibration leads to obvious machining taper error. In order to make the machine have a good dynamic and static performance under the condition of low cost and high working efficiency, we had a dynamics analysis of the main components and the overall structure of the machine tool. Taking the influence of the joint surface parameters into consideration, the authors set up an equivalent model about the joint surface, and analyze and test the machine's dynamic model by theoretical calculation and modal simulation. Moreover, the authors analyze the weak parts of the machine tool structure and improve them correspondingly to lay a solid foundation for the future improvement of the machine tool.

The authors establish a dynamic equation and mathematical model which can reflect the structural characteristics of machine tool and can influence the performance of the whole analysis process. According to the principle of dynamics, the autors can know that the general motion differential equation of a system with N degree of freedom under incentive is [2-3]:

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=f(t).$$

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Among them, $[M]$ is the mass matrix for the machine tool, $[C]$ is the damping matrix, $[K]$ is the stiffness matrix, $f(t)$ is the external incentives the machine tool received. The $\ddot{x}, \dot{x}, x$ is the variation of the machine tools vibration acceleration,respectively (acceleration), velocity and displacement variation respectively.If the machine vibration is free vibration and the damping and excitation can be ignored, then the theory equation for modal analysis is:

$$[M]\{\ddot{x}\}+[K]\{x\}=f(t)=0.$$

This equation is homogeneous differential equation.The result will reflect the automatic vibration characteristics of the machine tool,and the results of the characteristic equation will reflect the natural characteristics of the machine tool.

2. The Finite Element Model of the Whole Machine Tool

The main parts of DK7725 taper machine includes the base,post,the upper and lower arm and slider. According to Saint Venant's Principle [4-5], the authors eliminate or simplify the small chamfer, fillet, small craft hole, the edge of small protrusion and so on. ProE5.0 and ANSYS Workbench are used as tools for modeling and anslysis. After finishing the model ,the authors import it into the AWE platform. And element type will be chosen and mesh level will be graded. The grid division on the whole will be carried on before the local fine division, because this can reduce computation. In all,184707 nodes and113059 solid elements are divided, as showed in the Figure1.

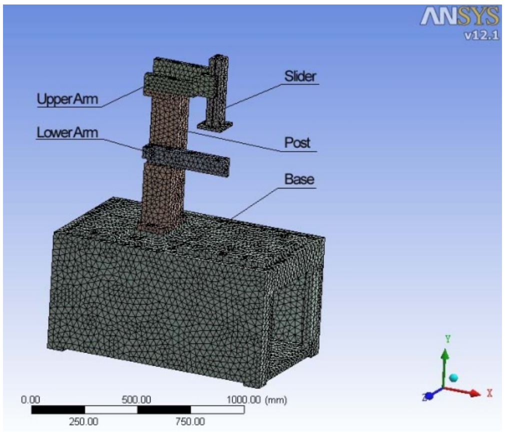

Figure 1. The Finite Element Model of Machine

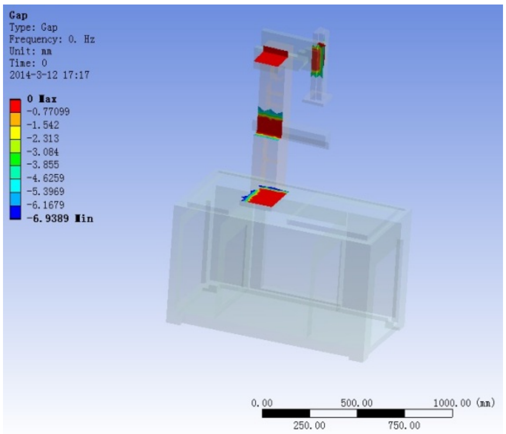

Figure 2. The Sketch of Joint Surface

3. The Equivalent Model and Parameter Identification of Joint Surfaces

The dynamics analysis depend on the accuracy of the overall structure model of the machine, which should take the joint surface characteristics into consideration. The fixed joint surfaces which is used widely in this paper includes the connection of the base with the post, the connection of the post with the upper and lower arm and the connection of the upper arm with the slider. And the slider and upper arm connection belongs to the unfixed joint surface connection. Parameter identification refers to a process which determines another system to make sure that system is nearly equivalent with the real system. The another system is determined by the given amount of input and output in the known structural vibration system and the objective function of minimum principle [6]. One of the difficulties of machine tool dynamics research at home and abroad is how to accurately identify and optimize the dynamic parameters of machine joint surface [7].

By researching on the joint surface, Masataks Yoshimura claims that the stiffness and damping of the joint surface can be acquired through integral on the situation that the positive pressure of units and equivalent stiffness and damping figures under different conditions are known [6]. From the dynamic perspective, the joint surface must have normal static force, which can produce normal stress which in turn produces damping and stiffness. Because the joint surface has such characteristics as elasticity, damping, the storage and consuming of energy,so the authors construct the dynamic model by spring and damper [8]. According to Masataks Yoshimura method, one joint surface has six dynamic forces with different forms, including the shearing force f_x in x, z direction, the normal force f_y in the y direction, and the bending moment $M_\theta x, M_\theta y, M_\theta z$ around the x, y, z directions. The actual surface can be a combination of one or more [9]. The calculation formula of the equivalent stiffness and damping is:

$$K_i = \iint k_i(P_n)dxdz \quad (1)$$

$$C_i = \iint c_i(P_n)dxdz \quad (2)$$

Among them, the $K_i(P_n)$ is the stiffness on per unit area, the $C_i(P_n)$ is the damping on per unit area, the K_i is the equivalent stiffness of the joint surface, the C_i is the equivalent damping of the joint surface. Assuming that other constants will not influence the $k(p)$、 $c(p)$, the authors will get the stiffness and damping of the joint surface by simplifying the equation (1) and (2):

$$K = k_i \times A, C = c_i \times A \quad (3)$$

The joint surface of the machine tool mainly has four parts: the combination of the base and the post, the post and the upper and lower arms,the upper arm and the slider. The former three are fixed joint surface connection, and the last one is the sliding joint connection. The joint surface of DK7725 shows in Figure 2.

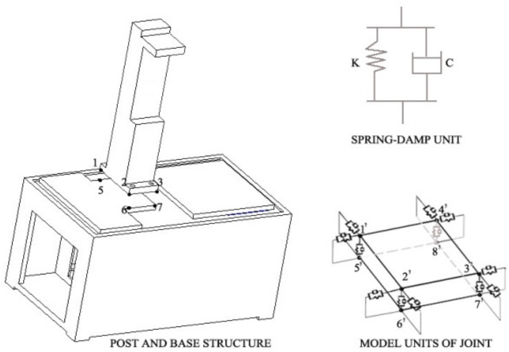

Figure 3. The Equivalent Mode of Base and Post

The base is connected with the post by four bolts, which belongs to fixed joint surface. As for HT200 base material, the elastic modulus is 1.48×10^{11}N/m^2, and Poisson's ratio is 0.31, and density is 7.2×10^3kg/m^3. The four bolts have certain influence on the joint stiffness. The radial stiffness of bolt is equivalent to spring-damping unit, while the axial stiffness of bolts is equivalent to the normal stiffness of joint. The pretension of bolt is about 2.8×10^4N measured by torque wrenchesthe. According to the Masataks Yoshimura method and the damping of unit joint surface and stiffness number table [10], the ratio of per unit joint surface stiffness to damping is $c_1/k_1=0.55\times10^{-14}$,$c_2/k_2=0.5\times10^{-3}$. From the formula (3), the authors can know the stiffness of the base and post and the damping in vertical. The stiffness of joint surface between the base and post is largely affected by their own gravity and the contact stress is even-distributed. So equivalent dynamic characteristics of the joint surface can be simulated by the four sets damping units. The equivalent model is showed as Figure 3. Model units of joint set in the connection position between post and base. And points 1,2....8 correspond to points 1',2'....8', respectively.

The post and the upper and the lower arms are connected with four bolts. All of them belong to the fixed joints.All together, 8 spring-damp units are used. The connection between the upper arm and the slider are slip connection. The motion direction of the interface is easy to be recognized, so the authors could set up spring-damp units by characteristic parameter of the single degree of freedom system. The equivalent dynamic model of the column, the upper and lower arms and the slides are showed as follows:

Table 1. The Stiffness and Damping Values of Joints

Joints Names	Bonded Area cm^2	Shear Stiffness N/m^3	Shear Damping Ns/m^3	Normal Stiffness N/m^3	Normal Damping Ns/m^3	Average Stress pa
Joint surface between base and post	338.8	3.39E16	1.86E2	3.39E6	1.69E3	3.34E6
Joint surface between post and upper arm	183.4	2.25E16	124	2.25E6	1.13E3	3.2E6
Joint surface between post and lower arm	152.6	8.4E10	8.4E-4	1.53E6	1.53E2	2.8E6
Joint surface between upper and slider	139.1	5.5E10	5.5E-2	5.5E3	2	9.9E5

4. The Dynamic Analysis and LMS Test of the Whole Machine Tool

4.1. Modal Analysis of the Whole Machine Tool

ANSYS, which can optimize the analysis for the structure indirectly, is also the basis of dynamic analysis such as harmonic analysis, transient dynamic analysis and spectrum analysis [11]. The basic idea of modal analysis is to analyze the dynamic performance of mechanism of the matrix equation, which can show the dynamic characteristics of muti-degree of freedom with single degree of freedom system and identify the natural frequency and vibration mode of structural dynamic characteristics [12].

Natural frequency and vibration mode,whose stability and reliability can improve the structure's performance and avoid unnecessary losses,are important parameters of dynamic properties.Introducing the 3Dmodel into the ANSYS Workbench,the authors apply the finite element constraints on the model to limit the degree of freedom in X, Y and Z direction of the base so as to identifying the stiffness and damping of the interface. After computing, the authors extract the natural frequency and vibration type of the first 6 steps modes. The modal changing is as follows:

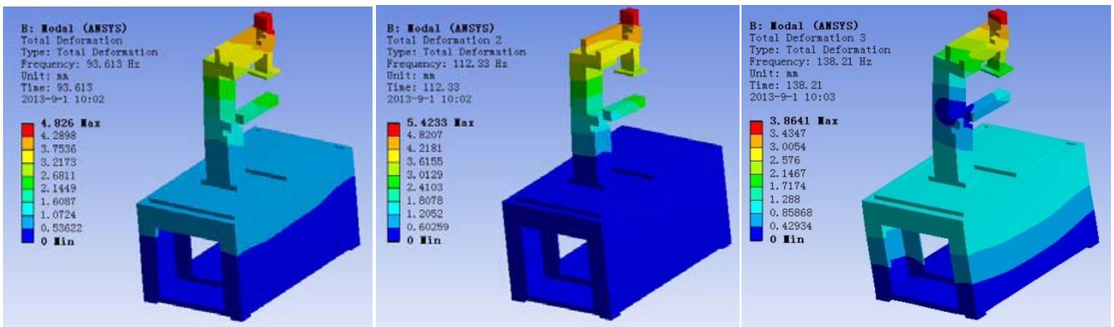

Figure 4. The First Three Modes of the Whole Machine Tool

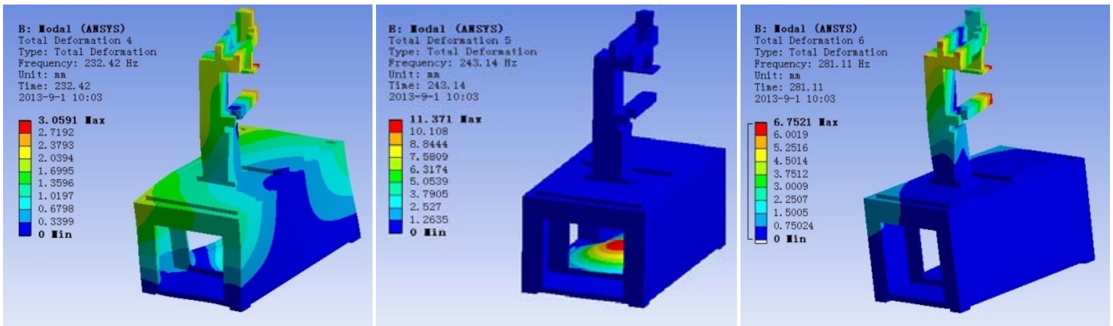

Figure 5. The Second Three Modes of the Whole Machine Tool

In summary, the fluctuating range of each frequency order the machine tool is not wide. The deformation of the first-order is not obvious while the second has obvious bending deformation. Both the third-order and fourth-order have serious bending and distortion. The vibration mode of the fifth-order shows that the base vibration mode change is bigger. The bottom of the base occurrs the bulging deformation in +y direction. If we improve the base structure, the deformation may decrease. Seen from the vibration mode of the sixth-order, the upper machine occurrs bending deformation, seriously. And distortion is accompanied. The deformation of the post is a bit serious. Obviously, it affects the performance of the machine tool.

4.2. LMS Test.Lab Experiments and Optimization

This experiment used the LMS TEST.Lab noise and vibration equiment of Gelgium LMS Inc. The functions of data processing and acquisition of this device is very strong. When the machine is in non working state, four sensors are respectively fixed on the base, post, lower arm and slider. The authors use a hammer with a plastic section to hammer the top of post.The trigger level is 0.025V, and pretrigger is zero second. When the authors set the actual sensitivity of channel setup, the input of force is set as 2.25mV/N. The FRF estimator is H1. Different knocking position, different knocking force, sensors placement and others factors will influence the results directly of indirectly. After several experiments,the authors got different results, and there some deviations among them. The following data is extracted from them, as shown in the Table 2.

Table 2. Comparison of LMS Experiment Data and the Optimization of Data

Names	First-order	Second-order	Third-order	Forth-order	Fifth-order	Sixth-order
LMS experimental data	69.7 Hz	102.49Hz	130.7Hz	175.37Hz	236.59Hz	285.59Hz
Relative errors	34.3%	9.6%	5.75%	32.5%	2.8%	1.6%

According to changing of the natural frequency and vibration mode, the upper part and the base of the machine tool have major damage. On the condition of appropriate stiffness and strength, the authors could optimize the machine structure to minimize the changing rate of natural frequency and vibration mode. The optimization aim is to decrease natural frequency and the bending as well as deformation. Optimization variation is the structural change of the machine tool and the density of each unit. The authors gradually optimizate the structure by improving key parts size, increasing the stiffener and adjusting the areas of the joint surfaces.

The structure can be improved by the following steps: firstly, set a quadrate reinforcement rib15mm in width and 5mm in height under the base. Secondly, change the shape and size of the base and thicken the supporting plate in the base. Thirdly, adjust the shape and area of the interface and strengthen the stiffness as well as increase the damping parameter. The authors set the variation range of optimization parameters in the ANSYS Workbench environment platform. P1-DS_d143 is the thickness of stiffener, and its range is [9, 11]. P2-DS_d6 is the post height, and its range is [648, 792]. P3-DS_d5 is the width of the bottom of the post, and its range is [142.2, 173.8]. The initial sample value is100. By calculation, the stress cloud of the structure size and the natural frequency of the whole machine is shown as Figure 6.

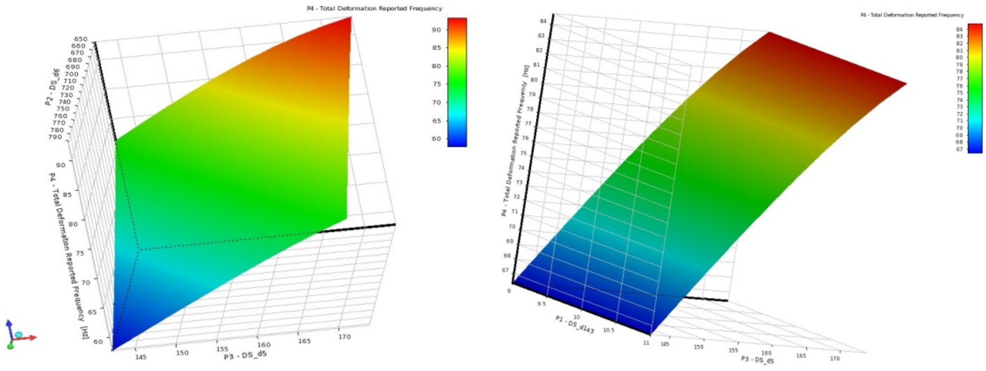

Figure 6. The Stress Cloud of Three Variables (P2,P3 and Natural Frequency)

Figure 6 shows the relatationship among the thickness of stiffener, the width of the bottom of the post and the post height, the natural frequeny of every steps. The optimization results shows that there are some difference between optimization and no-optimization. When the thickness of the stiffener is 9mm, the height of the post is 648.72mm and the width of the post bottom is142.36mm, the optimization effect is better, while size partly becomes small and the structure is simplified. The optimized data is shown as Table 3.

Table 3. The Optimized Data of Natural Frequency

Names	First-order	Second-order	Third-order	Forth-order	Fifth-order	Sixth-order
The optimized data	77.86Hz	91.12Hz	128.62Hz	243.37Hz	243.37Hz	267.3Hz
Relative errors	16.8%	18.9%	6.9%	4.7%	1%	4.9%

In conclusion, the optimized data is more close to the LMS Test experimental data. Compared with the natural frequency that is not optimized machine, the decreased range of optimized natural frequencies is from 0%to20%, and the variance ratio of the first-order and the second-order is bigger. Most of the natural frequencies got some declines, and the vibration modes also have some changes. However, there are still some gaps among the optimized data of ANSYS, the result of LMS experiment and the no-optimization. The gaps which is probably caused by the interference frequency produced during the test process, and some components such as base, post affect the measurement of the dynamic performance, or inaccurate measurement modeling, improper operation and other reasons. If the authors want to get better dynamic performance, they should make further research on the structure of the whole machine, and improve the understanding of the details including the joint pararmeter selection, calculation of stiffness and damping and so on.

5. Conclusion

Through the finite element analysis of DK7725 machine based on AWE, the authors learn the effects on the machine performance caused by the deformation and take measures of stiffener or changed structure to reduce the deformation. According to the results of the Modal analysis,the joint surfaces between the slider and upper arm,and the stiffness of post have the greatest influence on the whole machine performance, which are also the weakest part of the machine. The optimized natural frequency is decreased by 0% to 17% by DOE module. The authors should control the deformation of the main components so that the natural frequency is more close to the result of LMS experiment. The LMS experiment can reflect the machine performance in the actual work condition. The contrastive analysis of experimental data and the optimized data lays the foundation for further study on the effects of interference frequency on the whole machine.

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