COUPLED ELECTROMECHANICAL DYNAMICS FOR THE SYSTEM OF WORKING PLATFORM SUPPORTED WITH AMG IN APPLICATION OF MACHINE TOOL

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ABSTRACT

A coupled electromechanical dynamic model for the system of working platform supported with active magnetic guide (AMG) is set up according to special stiffness and damp of magnetic suspension system and vibration specialty of the working platform in machine tool. Analysis of coupled electromechanical dynamic performance of the system of working platform supported with AMG is carried out with different groups of control parameters. The outcomes show that the coupled electromechanical dynamic performance of the system• such as critical vibration frequency, vibration mode and so on can be improved by adjusting the control parameters. This research offers a theoretical support for ultra-high-speed and ultra-precise machine working platform design including the structure of platform, the design of electromagnet, and the establishment of the control parameters.

INTRODUCTION

Machining at super high speed and super precision is an advanced making technology of international modern manufacturing industry in recent years [1]. The super precision products must be manufactured on working platform where is supported with ultra-precise bearing and driven by ultra-high-speed driving system.

The technology of magnetic levitation is applied into working platform of machine tool as magnetically levitated train within sight with the development of it. Firstly the platform is lifted by the magnetic force from the active magnetic guide (AMG), then to be pushed to move by the linear motor. And because the magnetic levitation is such a kind of non-touched levitation that it has many advantages as following: no friction, never lubricated, long life, impossibly to make the precision decrease for no wear, and having little requirement about the environmental conditions. So, compared with the traditional mechanical guide, its working temperature range is very broad, at the same time it is very easy to maintain. Moreover it almost entirely eliminates the creep phenomenon lying in the mechanical guide and never occurs these problems including little life and precision owing to the wear and touching fatigue. In addition to such a truth that the reliability of electronic element is higher than that of mechanical part, obviously the reliability of AMG is over that of mechanical guide. For omitting the auxiliary oil pipe equipment, which is necessary for static pressure guide, the AMG does not pollute environment. And because its requirement of manufacturing precision is lower than that of air static pressure guide, it reduces the dustproof requirements of super clean. Because of active control, it also need not any extra investment to enhance the machine tool ability of information processing such as inspecting working conditions, forecasting, and making a diagnosis of trouble.

However, that what is harder than magnetically levitated train is in that the working platform supported with AMG in machine tool not only demands a quick speed driver but also needs a high precision bearing and location. Thus, so far no one can read any paper about working platform supported with AMG in industrial application of machine tool, now it is on theory research in laboratory. Until the recent ISMB-8, what are connected with AMG have included: In the pioneering research of Karl-Dieter Tieste *et al*. in 1994 [2], the dynamic behavior of a linear maglev supporting unit for fast tooling machines was studied about the frequency-response with first simulation. In 2000 [3], on the basis of the prototype guide, Martin Ruskowski built nonlinear modeling of a magnetically guided machine dynamic tool axis and in 2002 [4] an accuracy of about 5μm can be achieved over the full stroke since optical laser based sensors are applicable to this problem, a five-degree-of- freedom sensor has been built for the prototype guide and used as an absolute position reference. A magnetically linear carrier system with self-sensing magnetic suspension tracks is developed by Takeshi Mizuno *et al*. in 1998 [5]. But if we want to make it come into true, some deeper tests must be done. Gang Zhang *et al*. studied the coupled electromechanical dynamics of a rotor-active magnetic bearings system since 1999

[6,7,8]. The result is applied into coupled electromechanical dynamics for the system of working platform supported with AMG in application of machine tool now.

The AMG is a complex coupled electro-mechanical system. The magnets afford the levitation force. It takes the active control so that its character is completely different from traditional mechanical guide [6,7,8]. Its rigidity and damp, which are the function of vibration frequency, have a close relation with all kinds of circuit parameters [9].

A coupled electromechanical dynamic model of the system of working platform supported with AMG is set up in this paper according to special stiffness and damp of magnetic suspension system and vibration specialty of working platform in machine tool. The calculation and analysis of coupled electromechanical dynamic performance of the system of working platform supported with AMG are carried out with different groups of control parameters.

COUPLED ELECTROMECHANICAL DYNAMIC PARAMETERS

Figure 1 is a structure sketch map of one degree of freedom (DOF) magnetic suspension system. The working platform is supported with a pair of magnets up and down. The size of magnetic force is regulated through the sensor and control system to maintain the balance of platform.

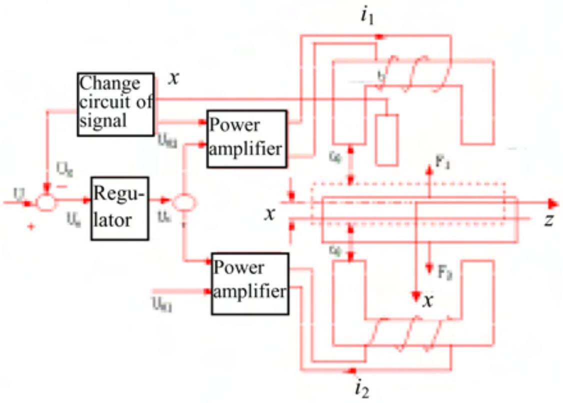

FIGURE 1: Structure sketch map of one DOF magnetic suspension system

The shutting loop rigidity and damping of magnetic suspension system [8,9] are:

$$k_{xx} = k_{xx0} + k_{xi0} R_e[G(j\omega)] \quad \textbf{(1)}$$

$$c_{xx} = k_{xi0} I_m[G(j\omega)]/\omega \quad \textbf{(2)}$$

where $G(j\omega)$ is transfer function of control system, ω is the vibration angular frequency.

The k_{xx0} is the open loop force-displacement rigidity:

$$k_{xx0} = -\frac{\mu_0 S_0 N^2 (I_0^2 + i_{x0}^2)}{c_0^3} \quad \textbf{(3)}$$

The k_{xi0} is the open loop force-current rigidity:

$$k_{xi0} = \frac{\mu_0 S_0 N^2 I_0}{c_0^2} \quad \textbf{(4)}$$

where μ_0 is the magnetic induction capacity of air ($4\pi\times10^{-7}$H/m); S_0 is the air gap area of magnet(m^2); c_0 is the thickness of single air gap(m^2); N is the circle numbers of coil; Because of differential structure, $i_1=I_0+i_{x0}+i_x$, $i_2=I_0-i_{x0}-i_x$, I_0 is the bias current, i_x is the control current caused by displacement in x direction, i_{x0} is the current used for counteracting the dead load in x direction.

The transfer function G of control system is composed of the transfer function G_s of sensor, the transfer function G_c of regulator and the transfer function G_{mp} of power amplifier:

$$G = G_s G_c G_{mp} \quad \textbf{(5)}$$

In the PID control system, the transfer function G is adjusted by four control resistances (gain parameter W_s, proportional parameter W_p, integral parameter W_i and differential parameter W_d). Therefore the shutting loop rigidity and damping of magnetic suspension system are a set of coupled electromechanical dynamic parameters. They can be adjusted by this four resistance control parameters.

COUPLED ELECTROMECHANICAL DYNAMIC MODEL

Figure 2 is the experimental rig of working platform supported with AMG.

FIGURE 2 Experimental rig of working platform supported with AMG

To achieve stable suspension of working platform in space, its five freedoms are required to be controlled. It is simplified into mechanical model in Figure 3 according to the basic principle of mechanical vibration.

The full system is composed of guide, six pairs of magnets in working platform (four pairs of load-bearing magnets in vertical direction, two pairs in horizontal direction) and control system (including sensor, regulator and power amplifier). When the working platform is in normal suspension, the single air gap c_0 is 0.2 *mm*.

The formulas (1)~(4) show that the smaller the air gap is, the larger the rigidity and damping is.

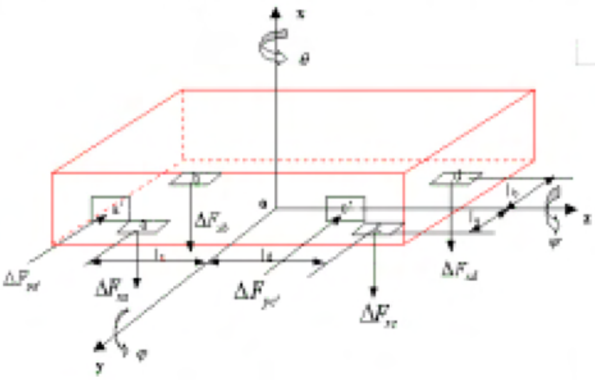

FIGURE 3 Mechanical model of platform supported with AMG

Given that the original point o of coordinate is in static mass center of platform, x and y are the dynamic mass center of platform, θ is swinging angle of platform around x axis, φ is swinging angle of platform around y axis, ψ is swinging angle of platform around z axis. The kinematic differential equation expressed by generalized coordinates $q=[x\ \varphi\ \psi\ y\ \theta]^T$ in mass center is:

$$M\ddot{q} + C\dot{q} + Kq = 0 \tag{6}$$

where M, C and K are mass matrix, coupled electro-mechanical damping and rigidity matrix respectively. The coupled electromechanical dynamic performance, such as critical vibration frequency, mode of platform, and so on, can be acquired through solving the equation (6).

ANALYSIS OF DYNAMIC PERFORMANCE

The tables 1 to 4 give four groups of control parameters W_s, W_p, W_i and W_d.

TABLE 1 The first group of control parameters

Passage	$W_s/k\Omega$	$W_p/k\Omega$	$W_i/k\Omega$	$W_d/k\Omega$
1	4.79	0.350	2.72	1.62
2	3.44	0.290	3.18	1.46
3	3.80	0.442	2.17	1.87
4	4.56	0.354	3.05	1.79
5	2.60	0.380	1.98	1.52
6	3.40	0.356	2.58	1.72

TABLE 2 The second group of control parameters

Passage	$W_s/k\Omega$	$W_p/k\Omega$	$W_i/k\Omega$	$W_d/k\Omega$
1	6.79	0.750	4.72	1.62
2	9.44	0.690	5.18	1.43
3	3.80	0.690	4.37	3.87
4	6.56	0.950	3.13	4.79
5	2.62	0.372	1.89	1.52
6	3.45	0.362	2.64	1.72

TABLE 3 The third group of control parameters

Passage	$W_s/k\Omega$	$W_p/k\Omega$	$W_i/k\Omega$	$W_d/k\Omega$
1	6.79	0.750	4.72	3.94
2	9.44	0.690	5.18	1.73
3	3.80	0.690	4.37	3.87
4	6.56	0.950	3.13	4.79
5	3.64	0.756	6.58	6.12
6	3.60	0.780	6.18	6.12

TABLE 4 The fourth group of control parameters

Passage	$W_s/k\Omega$	$W_p/k\Omega$	$W_i/k\Omega$	$W_d/k\Omega$
1	6.79	0.750	3.94	4.72
2	4.34	0.692	1.73	5.18
3	3.80	0.690	3.87	4.37
4	6.56	0.950	4.79	3.13
5	3.60	0.780	6.12	6.18
6	3.65	0.760	6.32	6.18

The tables 5 to 8 are all the critical vibration frequencies and relevant logarithmic attenuation rates acquired through calculation with the control parameters in tables 1 to 4.

TABLE 5 All critical vibration frequencies and relevant logarithmic attenuation rates calculated with the control parameters in table 1

Order of critical frequency	Critical frequencies (Hz)	Logarithmic attenuation rates
1^{st} order	40.18	3.1842
2^{nd} order	44.75	3.4661
3^{rd} order	64.96	0.9101
4^{th} order	74.74	0.9224
5^{th} order	83.35	1.1401
Resonant lost steady Frequency (Hz)	1662	

TABLE 6 All critical vibration frequencies and relevant logarithmic attenuation rates calculated with the control parameters in table 2

Order of critical frequency	Critical frequencies (Hz)	Logarithmic attenuation rates
1^{st} order	41.15	3.2018
2^{nd} order	65.41	0.9136
3^{rd} order	82.62	1.0911
Resonant lost steady Frequency (Hz)	1550	

TABLE 7 All critical vibration frequencies and relevant logarithmic attenuation rates calculated with the control parameters in table 3

Order of critical frequency	Critical frequencies (Hz)	Logarithmic attenuation rates
1^{st} order	46.51	2.4605
Resonant lost steady Frequency (Hz)	1406	

TABLE 8 All critical vibration frequencies and relevant logarithmic attenuation rates calculated with the control parameters in table 4

No critical frequencies (Hz)	
Resonant lost steady Frequency (Hz)	1400

From the tables 5 to 8 we can see, the critical vibration frequencies and their numbers of order change along with the variety of control parameter. When the first group of control parameters is used, the system has five orders of critical vibration frequencies, and working platform operated under the high frequency vibration needs to stride over five orders of critical frequencies, so working is not very steady. Because the logarithmic attenuation rate is over 0.9, stable abundant is enough and big. For the reason of the function of the damping, the amplitude of vibration is not too big. When the second group of control parameters is used, the system has three orders of critical frequencies, and working platform operated under the high frequency vibration correspondingly needs to stride over three orders of critical frequencies, so working is also not very steady. Because the logarithmic attenuation rate is over 0.9, stable abundant is big enough and big. For the reason of damping function, the amplitude of vibration is not also too big. When the third group of control parameters are used, the system has only one order of critical frequency, so working platform operated under the high frequency vibration need only to stride over one orders of critical frequency. Because the logarithmic attenuation rate is over 2.4, stable abundant is very big. For the reason of the function of the damping, the amplitude of vibration is hardly seen. Working is steady enough.

Figure 4 to Figure 8 show vibration modes of all critical frequencies in table 5.

Figure 9 to Figure 11 show vibration modes of all critical frequencies in table 6.

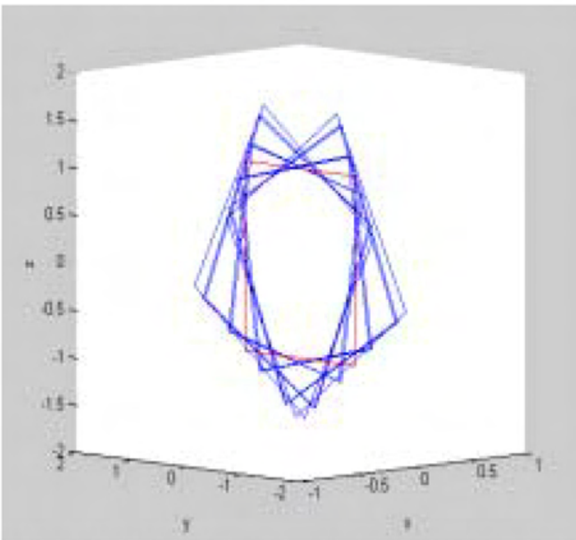

FIGURE 4 1st order main vibration modes under the first group of control parameters

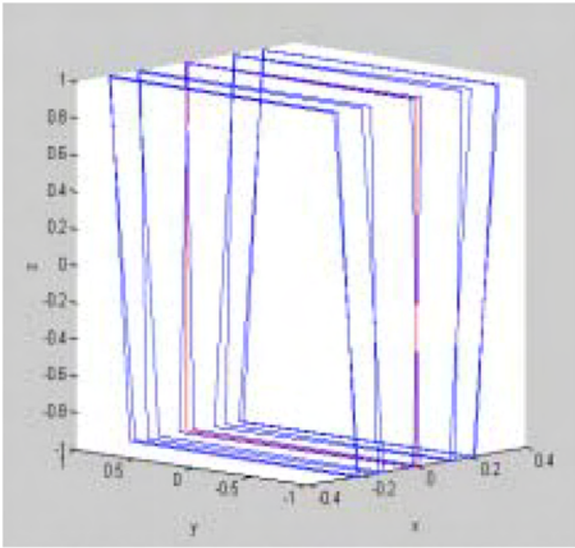

FIGURE 5 2nd order main vibration modes under the first group of control parameters

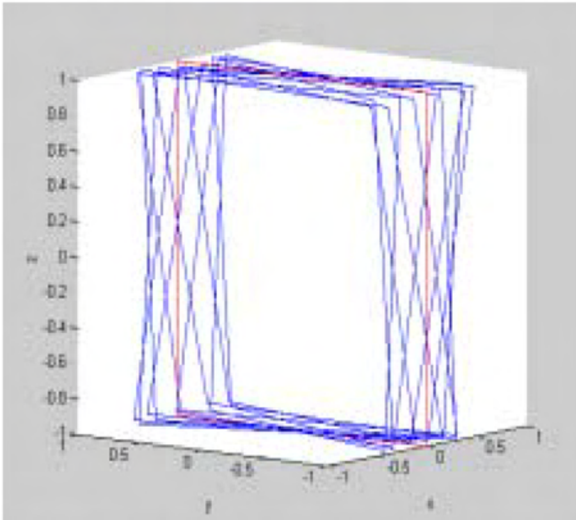

FIGURE 6 3rd order main vibration modes under the first group of control parameters

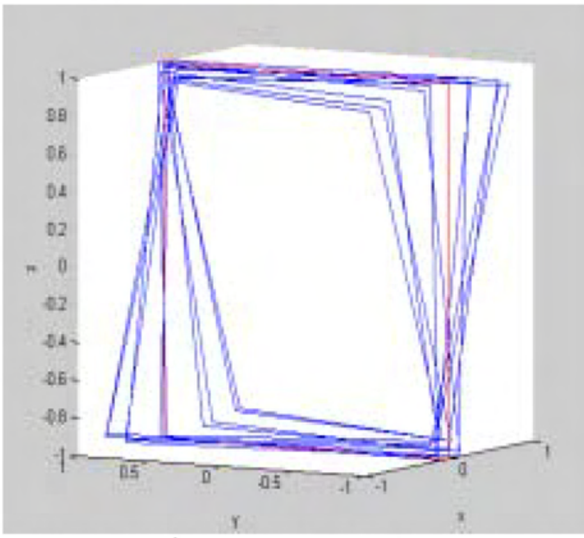

FIGURE 7 4th order main vibration modes under the first group of control parameters

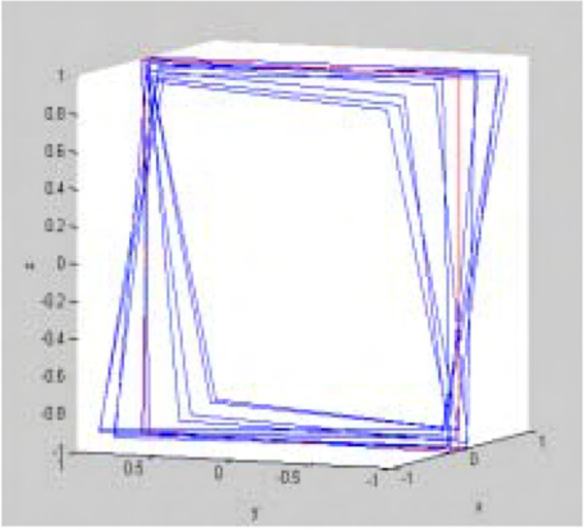

FIGURE 8 5th order main vibration modes under the first group of control parameters

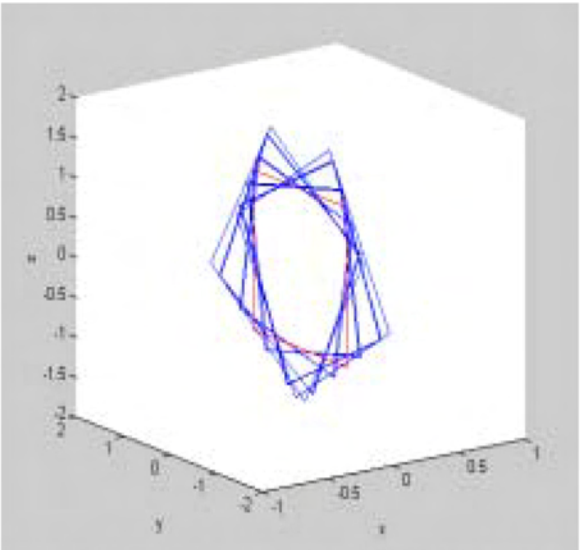

FIGURE 9 1st order main vibration modes under the second group of control parameters

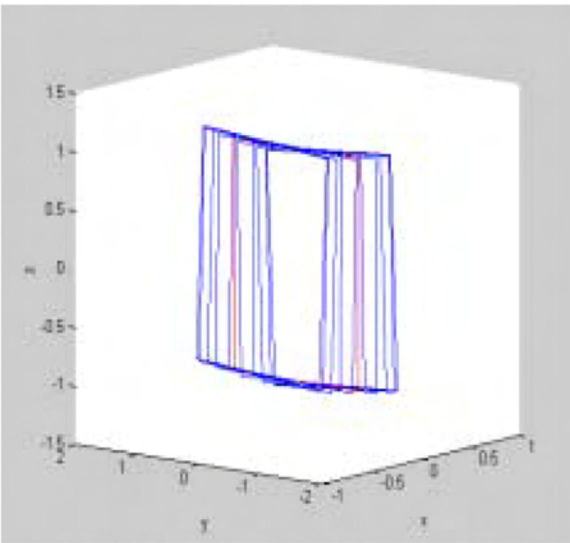

FIGURE 10 2nd order main vibration modes under the second group of control parameters

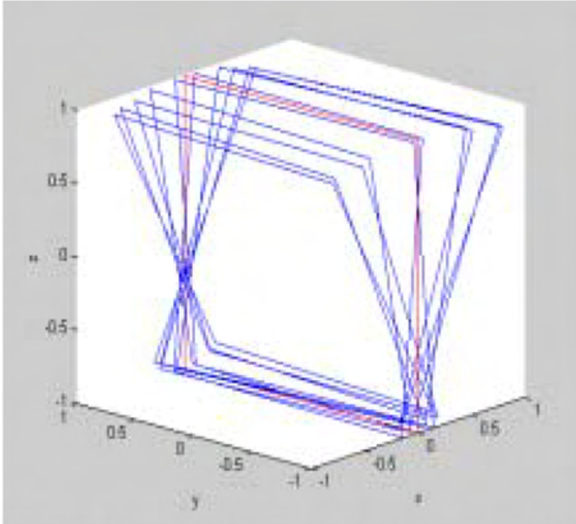

FIGURE 11 3rd order main vibration modes under the second group of control parameters

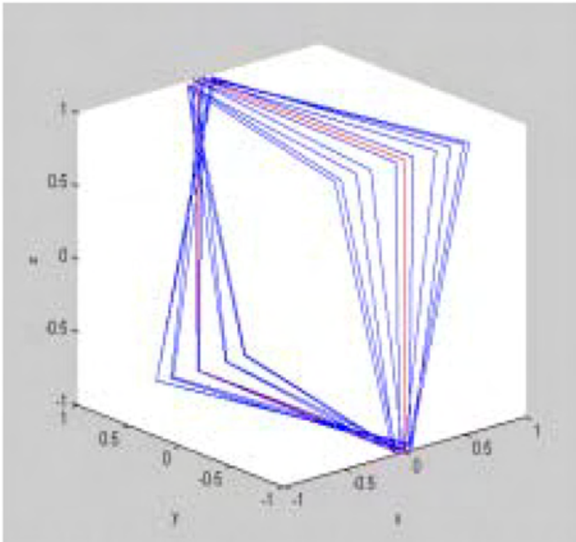

FIGURE 12 1st order main vibration modes under the third group of control parameters

Figure 12 shows vibration modes of critical frequency in table 7.

From the figures 4 to 12 we can see, the vibration modes and its number of order also change along with the variety of the control parameter. When the first group of resistance control parameters is used, the system has five orders of main vibration modes. Among of them the first order is a swing around the some axis paralleling the direction of z, and the axis swings in taper. The second order not only has translational vibration along the x direction but also swings around some axis paralleling the direction of y. The third order has translational vibration along the x direction, swings around some axis paralleling the direction of y, and also swings around the some axis paralleling the direction of z. The fourth and fifth orders are all the swing around some diagonal line axis in platform. When the second group of resistance control parameters is used, the system has three orders of main vibration modes. Among of them the first order is also a swing around the some axis paralleling the direction of z, and the axis swings in taper. The second order is only a swing around some axis paralleling the direction of z. The third order is only a swing around some axis paralleling the direction of y. When the third group of resistance control parameters is used, the system has only the first order swing mode around some diagonal line axis in platform.

When the fourth group of resistance control parameters is used, there is no critical vibration frequency and mode. The platform can't produce resonance entirely in working process. The ultra-precise bearing requirement of platform in machine tool is satisfied because the working process is very steady. The resonant lost steady frequency under above-mentioned four groups of resistance control parameters is greater than 1400 Hz. Which shows that the working platform can operate steadily under disturbance frequency less than 1400 Hz. It satisfies the machine requirement of ultra-high-speed and ultra-precise working platform. The static equilibrium amplitude of Experimental rig in figure 2 is less than 2μm and the dynamic equilibrium amplitude is less than 10μm through debugging.

Above-mentioned the improvement of working platform dynamic performance is because the rigidity and damping change along with the variety of the resistance control parameters. The rigidity and damping of 4th group of resistance control parameters are better than those of the other three groups of resistance control parameters. This shows that regulation of the resistance control parameters in system can improve the dynamic performance of the working platform.

CONCLUSIONS

The working platform supported with AMG can satisfy the machine requirement of ultra-high-speed and ultra-precise machine tool. It is a development trend of machine tool in the future. The design of working platform supported with AMG in machine tool must accord with the demands of system dynamics design ideals, and the loading capacity in statics is a foundation, at the same time the rigidity and damping in dynamics are extremely important. The rigidity and damping of AMG system can be improved through regulating the resistance parameters in control system. So the active control can be realized, thus the dynamic performance of working platform in machine tool can be improved.

Further research will be directed to the coupled electromechanical dynamics for the system of working platform supported with AMG and driven by linear motor in application of machine tool at the same time.

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