The Performance Study on Two-Stage Vibration Isolation System with Active Magnetic Suspension Control

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Abstract
Two-stage vibration isolation system is normal equipment on naval vessel, which has good function of vibration isolation. Active control on vibration isolation system is better than passive control. Active magnetic suspension control has many virtues like alterable control methods and variable stiffness and damping. So that it is applicable to complex multi-interferes and multi-coupling floating raft. This paper set up mathematic models of two-stage vibration isolation with active magnetic suspension control system and solves the transfer functions. After simulating in MATLAB it analysis the effectiveness of K_p ,τ_d,τ_i of PID controller to vibration isolation and the parameters which influence the vibration isolation performance including up and down layer stiffness, damping and mass ratio. The simulation results indicate that active magnetic suspension has good works to vibration isolation. At the same time, on the basis of these results optimize the construction.
Keywords Two-stage vibration isolation system, Active control of magnetic suspension

1 Introduction

Two-stage vibration isolation systems is one kind of device of power equipment for damping vibration, which has better isolation vibration performance compared with one layer device. Internal and external research focus on dynamic performance of two-stage vibration isolate system affected by design parameters[1], effects of nonlinear spring and damping on two stage vibration isolate system[2], effect of pedestal stiffness on two stage vibration isolate system[3] and study of flexible mass on two stage vibration isolate system[4], and so on. Compared with passive control of isolation vibration system, active control can get better effectiveness by adjusting stiffness and damping. On this aspect, there are many researchs about methods[5], the fix position of active vibration isolator[6] and coupling vibration control[7]. Active magnetic suspension control system has many virtues including stiffness and damping controllable and tunable. Furthermore, it can apply classical or modern control theories according to different simulation force value and place to get the best effect, which make it become the best control system for multi-turbulent and multi-coupling floatin raft.This paper establish two-stage vibration isolation system with active magnetic suspension control equipment and its mathematic models. Then simulate in MATLAB and research how the three parapmeters of PID controller affect the peroformance of the system. Furthermore, study the influence of parameters such as up and down layer stiffness, damping and mass ratio to vibration isolation. Simulation results indicate that active magnetic suspension does good works to vibration isolation. In the end, on the basis of these results optimize the construction.

2 Active Magnetic Suspension Vibration Isolation control system

2.1 Theory of Active Magnetic Suspension

Put active magnetic suspension vibration isolation system into the layer between middle mass $m_2$and pedestal m_3 .Construction and theory is showed in Fig.1 and Fig.2.there are three masses —m_1 is up layer mass,m_2 is middle layer mass,$m_3$is pedestal mass.$k_1$and c_1 is stiffness and daming of the first stage.k_2 and c_2 is stiffness and daming of the second stage.U is the system of active magnetic suspension.U is the system of active magnetic suspension.$f = f_0 \sin \varpi t$ is outer simulate force act on up layer mass m_1.x_1 is displacement of m_1.x_2 is placement of m_2.m_3 is fixed to foundation firmly. So ignore its displacement. Active

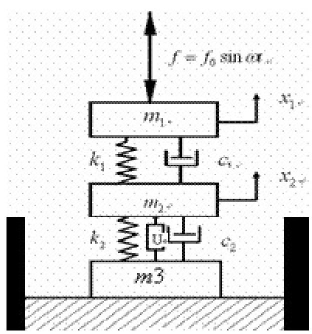


Fig.1 Model of active magnetic suspension in two-stage vibration isolation system

magnetic suspension feed back system can provide damping force direct proportion to absolute speed of isolated object. When vibration happened and isolated vibration object elasticity mass increase or actual elasticity factor decrease, isolation vibration spring damping static deflection still keep constant. Therefore active magnetic suspension feed back isolate vibration system is better than passive isolated vibration system, especially in low frequency area.

Middle mass m_2 displacement is control variable quantity which is recorded by displacement sensor. After comparing with setting amount, PID controller calculate discrepant value and change electric current to produce magnetic force which help middle mass keep balance position to get the purpose about decrease the force from up layer to base m_3.

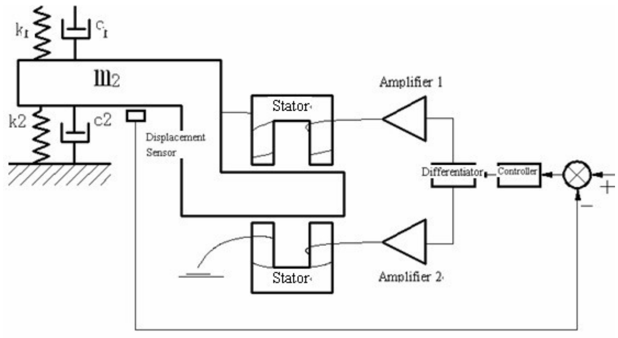


Fig.2 Theory of magnetic suspension system

2.2 Transfer Function

Establish system block plant (See Fig.3).$f(s)$ is outer simulation force,$f_1(s)$ is force from system to base. $H_k(s)$ is PID controller transfer function.$H_2(s)$ is power amplifier transfer function.$H_1(s)$ is the first stage transfer function.$H_3(s)$ is the second stage transfer function.

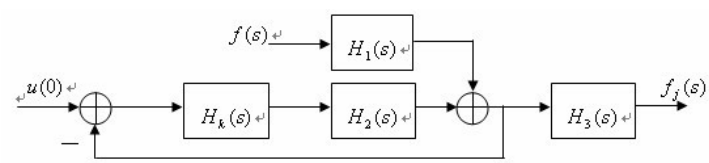


Fig.3 Active magnetic suspension vibration isolation system block plant

Without magnetic force, system force transfer function is

$$T(s) = \frac{f_1(s)}{f(s)} = H_1(s)H_3(s)$$

When apply feed back control, system force transfer function is

$$T_1(s) = \frac{f_1(s)}{f(s)} = \frac{H_1(s)H_3(s)}{1 + H_2(s)H_k(s)}$$

$$f_e = k\frac{i^2}{x^2}$$

Linearization force to

$$f_e = k_i(i - i_0) - k_x(x - x_0)$$

$$H_k(s) = K_p + \frac{\tau_i}{s} + \frac{\tau_d s}{1 + T_f s}$$

$$H_3(s) = k_2 + c_2 s, H_2(s) = k_s$$

Substitute all parameters into system differential equation.

$$\begin{pmatrix} m_1 & 0 \\ 0 & m_2 \end{pmatrix} \begin{pmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{pmatrix} + \begin{pmatrix} c_1 & -c_1 \\ -c_1 & c_1 + c_2 \end{pmatrix} \begin{pmatrix} \dot{x}_1 \\ \dot{x}_2 \end{pmatrix}$$

$$+ \begin{pmatrix} k_1 & -k_1 \\ -k_1 & k_1 + k_2 \end{pmatrix} \begin{pmatrix} x_1(t) \\ x_2(t) \end{pmatrix} = \begin{pmatrix} f \\ f_e \end{pmatrix}$$

Use Laplace transformation to equations and arrange to get

$$H_1(s) = \frac{x_2(s)}{F(s)} = \frac{s + 440}{(s + 3.62)^2 (s + 1.01)^2}$$

2.3 simulations

In Matlab Simulink build block plant of system transfer function (See Fig4.) System with and without active magnetic suspension control forcetime curves are list at Fig.5.

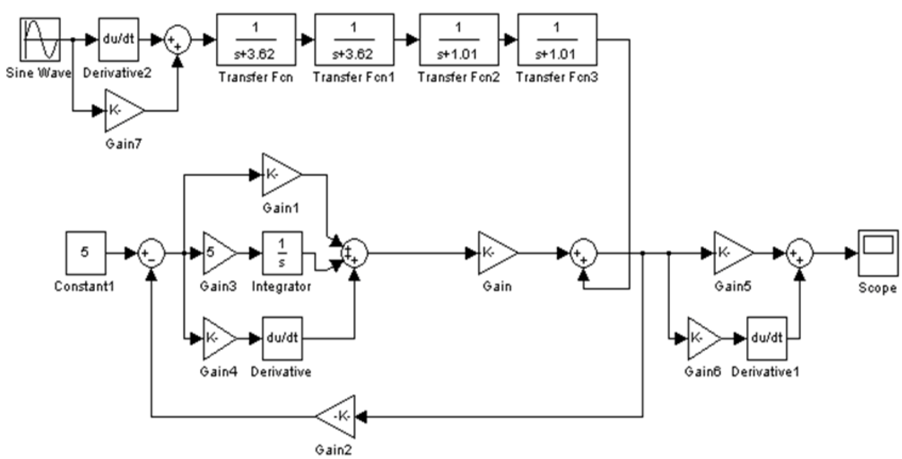
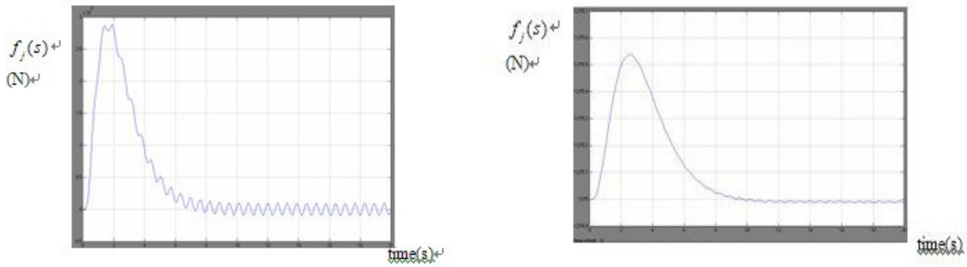


Fig.4 Block plant to simulate in Simulink of Matlab

Fig.5 showed that active magnetic suspension system indeed helps to improve the performance of two-stage isolate vibration. Force transfer from up layer outer simulate to base become quit so small apparently and vibration curve become smooth.



(a) Nno feedback time-force transfer (b) With feedback time-force transfer

Fig.5 Comparation with before and after using magnetic suspension feed back control

3 Influence of PID Controller Parameters to Isolate Vibration

3.1 Effective of Proportion Parameter K_p

Without change other parameters, $K_p = 100, 500, 1000$ Result curve showed at Fig.6 From curves we can see increase K_p will decrease force to base tremedously.

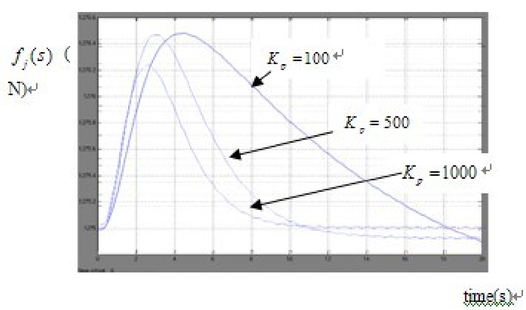


Fig.6 Different K_p time-force curves

3.2 Effective of Differential Constant τ_d

Without change other parameters, τ_d=100,500,1000. Result curve showedat Fig.7.

Form Fig.7, τ_d has a little influnce to force max value and the speed of isolation vibriation. However, increase τ_d can make smooth of force vibraition and improve the performance of the system.

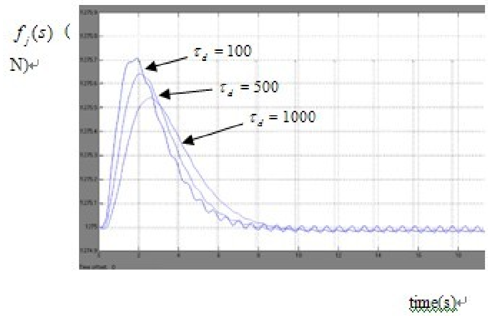


Fig.7 Different τ_d time-force curves

3.3 Effective of Integral Constant τ_i

Without change other parameters,τ_i=0.5,5,50. Result curve showed at Fig.8.

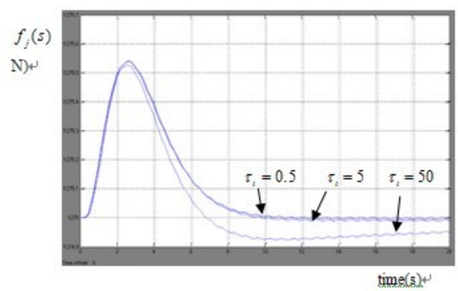


Fig.8 Integral constant τ_d time-force curves

Curve tell that τ_i has no big influence to max force value but only effect to min force value slightly.

4 Effectiveness of Two-Stage Vibration Isolation System Ather Parameters

4.1 Natural Frequency and Force Coefficient of Transmission Calculation

Without thinking of damping, system natural frequency will be calculated by listed formula(1)

$$\omega_n{}^2 = \frac{k_1 + k_2}{2m_1} + \frac{k_2}{2m_2} \pm \sqrt{(\frac{k_1 + k_2}{2m_1} + \frac{k_2}{2m_2})^2 - \frac{k_1 k_2}{m_1 m_2}}$$

Substitute parameters into the formula:

$$\xi_2 = 0.06N/(m/s), \xi_1 = 0.04N/(m/s)$$

The natural frequencies are:

$$\omega_{n1} = 78HZ, \omega_{n2} = 2HZ$$

When force excited, the force coefficient of transmission is formula (2)

$$T_f = \sqrt{\frac{(\alpha^2 - 4\xi_1\xi_2\alpha\varpi_1^2)^2 + \varpi_1^2(2\xi_1\alpha^2 + 2\xi_2\alpha)^2}{A^2 + B^2}}$$

In the formula

$$A = {\varpi_1}^4 - {\varpi_1}^2(\alpha^2 + 4\xi_1\xi_2\alpha + \mu + 1) + \alpha^2$$

$$B = {\varpi_1}^3(2\xi_2\alpha + 2\xi_1\mu + 2\xi_1) - \varpi_1(2\xi_1\alpha^2 + 2\xi_2\alpha)$$

$$\mu = m_1/m_2, \varpi_1 = \omega/\omega_1, \varpi_2 = \omega_2/\omega_1$$

$${\omega_1}^2 = k_1/m_1, {\omega_2}^2 = k_2/m_2$$

$$\xi_1 = c_1/2\sqrt{k_1 m_1}, \xi_2 = c_2/2\sqrt{k_2 m_2}$$

When substitute the parameters into the formula, we get the curve of force-coefficient transmission—simulation frequency (See Fig.9).

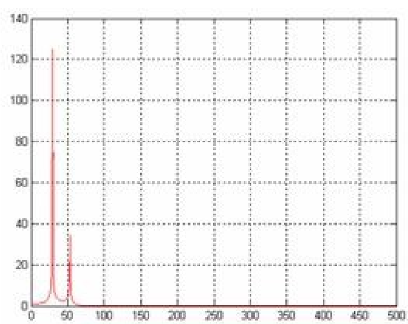


Fig.9 Curve of force-coefficient transmission-simulation frequency

From the curve we can see the influence of simulation frequency to force-coefficient transmission:

- After 53HZ, force- coefficient of transmission is near zero that indicates the effectiveness of isolation vibration is good;
- Before 53HZ, there are two peak values on resonance vibration that indicates this frequency region is the target we will research. to improve the effectiveness.

4.2 *Influence of Stiffness k_2 to Natural Frequency*

According to formula (1), we can get the influence of stiffness to natural frequency (See Fig.10).

From the curve, we can see stiffness is almost linear direct proportion to natural frequency. So, the more stiffness is the more natural frequency is. On different working condition, we can change stiffness aim at natural frequency to avoid resonance vibration.

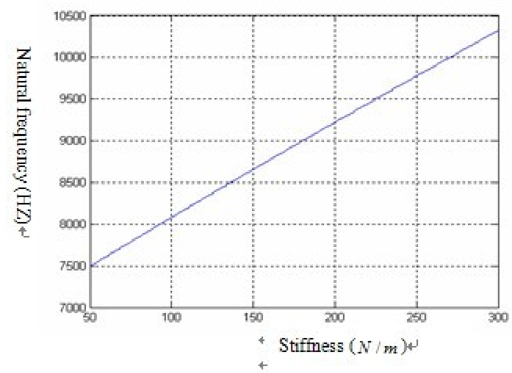


Fig.10 Curve of stiffness to natural frequency

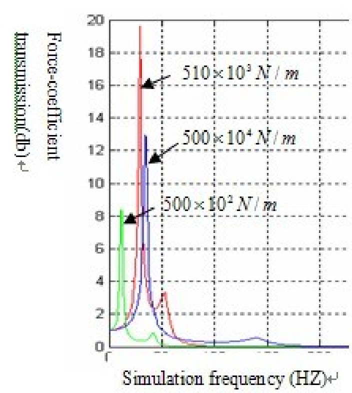


Fig.11 Influence of stiffness to force-coefficient of

4.3 *Influence of Stiffness k_2 to Force-coefficient Transmission*

According to formula (2), choose three different value of stiffness $k_2 500\times10^2 N/m, 500\times 10^4 N/m, 510\times10^3 N/m$.We get three curves about k_2 — force-coefficient of transmission (See Fig.11).

Curves indicates that when stiffness is small the peak value move to left. If change up-layer stiffness, result is on the contrary that means when becomes larger the peak value move to left. So, increase up-layer or decrease down-layer stiffness all is good for isolation vibration.

4.4 Influence of Damping k_1 Ratio k_1 to Force-coefficient Transmission

Without changing other parameters, damping ratio
$\xi_1 = 0 \quad \xi_2 = 0$ become $\xi_1 = 0.04 \quad \xi_2 = 0.06$

We get the curve of simulation frequency—force e-coefficient of transmission (See Fig.12).Fig.12 indicates that damping ratio influence peak value directly but it doesn't influence the position and tendency of peak value. The smaller damping ratio is the bigger peak value is.

When $\xi = 0.04, \xi_2 = (0.01, 0.06, 0.2, 2)$ respectively, we get the curves (See

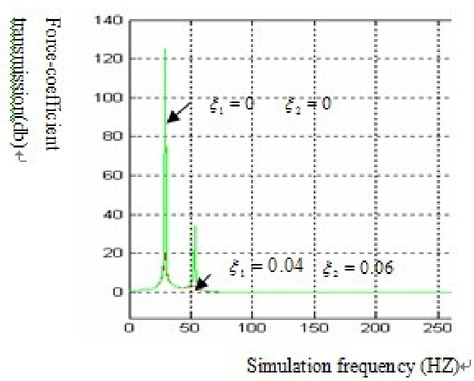


Fig.12 $\xi_1 = 0, \xi_2 = 0; \xi_1 = 0.04, \xi_2 = 0.06$ Simulation frequency—force e-coefficient of transmission

Fig.13).When damping ratio is less than 1, it can't influence peak value position and tendency, but influence peak value. The bigger damping ratio is the smaller peak value is, that is good for isolation vibration. While when damping ratio is more than 1, one hand it can decrease the peak value, on the other hand, peak value position move from left to right.

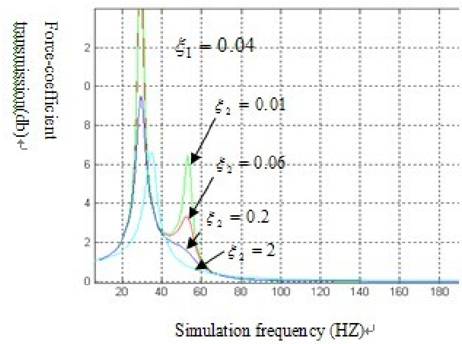


Fig.13 $\xi_1 = 0.04$,$\xi_2 = (0.01, 0.06, 0.2, 2)$ Simulation frequency—force e-coefficient of transmission

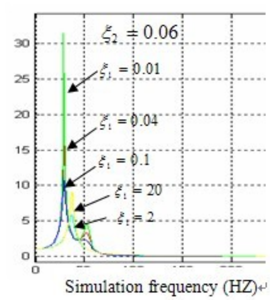


Fig.14 $\xi_2 = 0.06$,$\xi_2 = (0.01, 0.04, 0.1, 2, 20)$ Simulation frequency—force e-coefficient of transmission

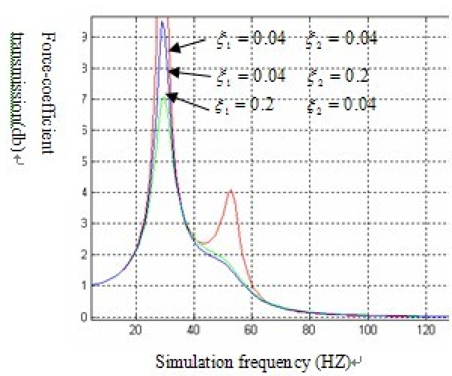


Fig.15 Change ξ_2 and ξ_1 Simulation frequency—force e-coefficient of transmission

When $\xi_2 = 0.06$,$\xi_1 = (0.01, 0.04, 0.1, 2, 20)$ respectively, we get the curves (See Fig.14).When damping ratio is less than 1, it can't influence peak value position and tendency, but influence peak value. The bigger damping ratio is the smaller peak value is, that is good for isolation vibration. While when damping ration is more than 1, peak value position move from left to right and peak value become bigger when damping ratio is bigger. So, damping ratio than close to 1 is the best value to isolation vibration.

4.5 Influence of Mass Ratio to Force-coefficient Transmission

When $m_1 = 104kg, \xi_1 = 0.04, \xi_2 = 0.06, \mu = 0.04042, 2, 5, 10$ respectively, we get the curves showed in Fig.16. Isolation vibration system brings about two resonance points which position are determined by mass ratio.

When $\mu \geq 1$, the first peak value position move to right but not very clearly and the second peak value position move to right quiet obvious. There is small influence to curve tendency in lower frequency region ,such as 40HZ. But in low frequency region, the bigger mass ratio is the smaller peak value is. In middle frequency region, from 40HZ to 180HZ, curve change apparently, the second peak value decrease quickly. It is obviously that mass ratio influence middle frequency greatly. To make sure system work at frequency far away from resonance points we hope the distance between the first peak value and the second peak value is large, which means $\mu \geq 1$(mass $m_1 \geq m_1$) is good for isolation vibration.

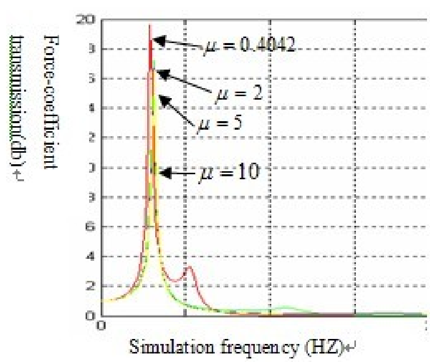


Fig.16 Different curve of simulation frequency—force e-coefficient of transmission

5 Improve Isolation Vibration System

According to above analysis, we take three proposals to improve the system performance. Because mass has been chosen, then

- Without changing other parameters, decreasing under layer stiffness by reducing spring from 6 to 4, then stiffness

$$k_2 = 430 \times 10^3 N/m$$

- Without changing other parameters, raise up and under layer damping

$$\xi_1 = 0.4N/(m/s), \xi_2 = 0.6N/(m/s)$$

- Decrease under layer stiffness and increase up layer stiffness at the same time,

$$k_2 = 430 \times 10^3 N/m, \xi_1 = 0.4N/(m/s), \xi_2 = 0.6N/(m/s)$$

Results showed in Fig.17, from which we can see that damping is the most important influence factor and only change stiffness is good for middle-high frequency isolation vibration. The best solution is decreasing under layer stiffness and increase up and under damping simultaneously.

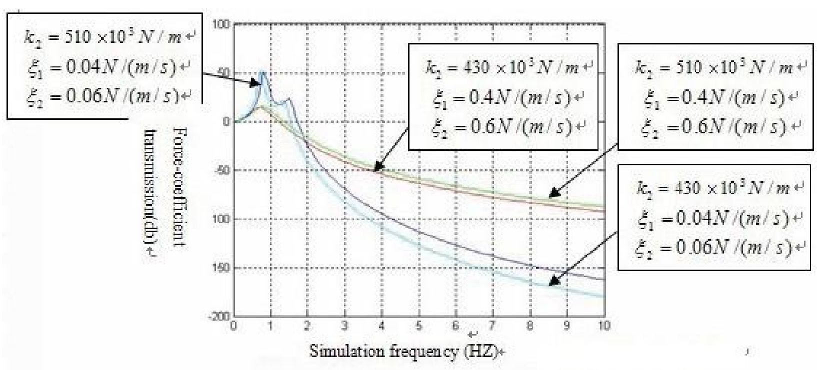


Fig.17 Change stiffness and dampin Simulation frequency—force e-coefficient of transmission

6 Conclusion

This paper research on the performance of two-stage vibration isolation system and the effectiveness of system parameters including three parameters of PID controller,stiffness, damping and mass ratio. Conlusion are

- Essence of active magnetic suspension control system act on two-stage vibration isolation system is change the second layer stiffness and damping to decrease vibration.
- Adjusting stiffness by active magnetic suspension to avoid system nature frequency is a good method to prevent resonance vibration.
- Decreasing the second stiffness by adjust active magnetic suspenion control has good effect.

- Adjusting damping by active magnetic suspension control to keep damping less than 1is good effect.
- Increasing proportion parameter K_p is good for vibration isolation speed.
- Increasing differential constant τ_d ralatively is good for decreasing vibration.
- Integral constant τ_i has no apparent influence to prevent vibration.

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