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Dynamics and Stability of Magnetic-air Hybrid Quasi-zero

Stiffness Vibration Isolation System

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Keywords:

Magnetic-air hybrid; Quasi-zero Stiffness; Vibration isolation; Stability; Dynamics

Abstract:

With the improvement of machining accuracy, external low frequency vibration has become one of the most important factors affecting the performance of equipment. The theory of quasi-zero stiffness vibration isolation shows favorable low frequency vibration isolation effect. Based on our previous research on the structure of magnetic-air hybrid quasi-zero stiffness vibration isolation system, the nonlinear mechanical expression of positive and negative stiffness structure has been analyzed in this paper, to improve application of the system and provide a theoretical basis for sequential studies of active control. To analyze the judgement criterion of the quasi-zero stiffness, an accurate mechanical model was first established. Then, the dynamical model based on external low frequency vibration was developed, to investigate the stability and natural frequency and deduce the amplitude frequency characteristics and displacement transfer rate. Finally, we carried out simulation and experimental analysis to verify the stiffness of high static and low dynamic and the low frequency vibration isolation effect of the vibration isolation system.

1. Introduction

With manufacturing equipment and processing technology advancing into nanometer, the low frequency vibration of the environment has gradually become a key factor affecting the performance and accuracy of precision instruments [1, 2]. After reviewing a large number of literature, we find that low frequency vibration mainly comes from the vibration and noise of seismic shudder, working environment and personnel flow. Besides, the frequency ranges from 0Hz to 10Hz [3-5]. Therefore, to

guarantee the precision performance of instrument, it is extremely necessary to suppress external low-frequency vibration interference. Because the nonlinear vibration isolation theory shows favorable low frequency vibration isolation effect, it has been researched and applied extensively in the field of low frequency vibration isolation. In particular, the theory of quasi-zero stiffness [6-10] is characterized by high static stiffness and low dynamic stiffness, which has attracted more and more attention in the field of nonlinear vibration isolation.

In 1957, Molyneux [11] first proposed the concept of quasi-zero stiffness, and classified and analyzed vibration isolation structures with different negative stiffness. R.A.Ibrahim [12] introduced the basic concepts and inherent nonlinear phenomenon of nonlinear vibration isolator, and summarized the specific types of nonlinear vibration isolator. To realize the stiffness characteristics of high static and low dynamic, Platus [13], Shaw [14], LAN [15] and Araki [16] have respectively applied Euler compression bar, bistable composite plate, planar spring and elastic Cu-Al-Mn bar of shape memory alloy to quasi zero stiffness vibration isolator, and analyzed the mechanical properties of various structural forms. Zhou [17] developed a semi-active quasi-zero stiffness vibration isolator by connecting mechanical spring and electromagnetic spring in parallel. Zhou pointed out that, when the current value of the vibration isolation system was 0, the system was a passive high-intensity solid-state laser vibration isolator. While the quasi-zero stiffness vibration isolation could be realized by adjusting the current. Robertson [18] established a 6-DOF quasi-zero stiffness structure with permanent magnet, proposed the design concept and relevant theory, and studied its mechanical structure and control system. To sum up, the theory and application of quasi-zero stiffness vibration isolation system have developed unprecedentedly through intensive analysis and research on the theory, structure and mechanical properties of the system.

Prior to this paper, we designed a magnetic-air hybrid quasi-zero stiffness vibration isolation structure, and preliminarily analyzed its mechanical properties. In this paper, based on the previous structure, the mathematical model of positive stiffness structure and negative stiffness structure will be further deduced to improve its applicability. A more accurate quasi-zero stiffness mechanical model is to be established, and the judgement criterion of quasi-zero stiffness is to be analyzed. The rest of this paper is organized as follows. In Section 3, the stability and natural frequency of the vibration isolation system are discussed. In Section 4, the dynamic model and displacement transfer rate of the vibration isolation system are analyzed, and the resistance to external low frequency interference is simulated and investigated. The experimental platform is built to test and verify the effect of the vibration isolation system in Section 5. Major conclusions and main contribution of this paper are presented in Section 6.

2. Structure of magnetic-air hybrid quasi zero stiffness vibration isolation system

2.1 System structure

The structure of the adjustable magnetic-air quasi-zero stiffness isolator in the early stage was shown in Fig. 1. An air spring installed vertically was used as the positive stiffness element while four electromagnetic springs installed horizontally as the negative stiffness element. The two were connected in parallel by universal joints. When the system was unloaded, the upper end of air spring was higher than the axis of electromagnetic springs. Meanwhile, the universal joint was in an inclined state, and the air spring and electromagnetic springs were in a natural state. When an equipment was installed on the quasi-zero stiffness isolator, the upper end of air spring was at the

same height as the axis of electromagnetic springs, which could be called as static balance. At the same time, the universal joint was in a horizontal state, and the air spring and electromagnetic springs were compressed.

When the system was in static balance, its bearing capacity was mainly provided by the stiffness of air spring. When the system was disturbed by dynamic load or external vibration, however, the air spring interacted with the electromagnetic springs to achieve quasi-zero stiffness, so as to fulfill the purpose of low frequency vibration isolation. The mechanical model was displayed in Fig. 2, and the corresponding basic parameters were shown in Table 1.

Figure 1 Structure of magnetic-air hybrid quasi zero stiffness vibration isolator

Figure 2 Mechanical model of magnetic air hybrid quasi zero stiffness vibration isolation system

Table 1 Basic parameters

Structural parameters					
L	h	a			
11.404 mm	8.5 mm	7.6026 mm			
Parameters of 086060H-1 air spring					
Minimum compression	Safe elongation	Stroke Length			
44 mm	90 mm	46 mm			
Electromagnetic parameters					
μ_0	μ_r	S	A	N	$I_{\max}$
$4\pi \times 10^{-7}$	1	10	1070.105	400	7
Vs/Am	(Vs/Am)	mm	mm ²	Turns	A

In the previous studies, it was assumed that when the pressure p was determined, the stiffness of air spring was constant and the static balance position was at $x = h$. As the corresponding static balance position of different equipment varied, the electromagnetic springs and universal joints need to be adjusted at the same time, thus leading to complex adjustment of the system. Therefore, this paper studied and analyzed the positive and negative stiffness element model to improve the structural mathematical model.

2.2 Mathematical model of positive stiffness element

Air spring is a vibration isolator which can achieve bearing and vibration isolation by compressing gas. It was assumed in this paper that there was no air leakage in the air spring, namely, the volume of air spring remained unchanged. When the volume kept constant, the greater the compression was, the larger the effective cross section would be. Therefore, it was assumed that there was a linear relationship between compression and effective cross section, as was depicted in Eq. (1).

$$\begin{cases} V_n = \text{constant} \\ s_e = \kappa x + s_0 \end{cases} \quad \text{when } p = \text{certain}, \quad (1)$$

where V_n was the volume of air spring at any time; s_e , x , s_0 and κ were effective cross section, compression, initial effective cross section and proportional coefficient of air spring, respectively.

According to ideal gas equation of state and mechanical equation [19-21], a new equation could be expressed as:

$$\begin{cases} (p_1 + p_a)V_1^\lambda = (p_0 + p_a)V_0^\lambda \\ F_a = p_1 s_e \end{cases}, \quad (2)$$

where p_a , p_0 and V_0 were atmospheric pressure, initial state air spring pressure and initial state volume, respectively; λ , p_1 and V_1 were polytrophic exponent, pressure and volume of air spring under current state, respectively; F_a was the air spring force when the pressure value was p_1 .

We imported Eq. (1) into Eq. (2) and calculated the partial derivative of compression x , then obtained the stiffness equation of air spring, as was shown in Eq. (3):

$$K_a = \frac{\partial F_a}{\partial x} = \frac{-\lambda \left(\frac{V_0}{V_1} \right)^\lambda (p_0 + p_a)}{V_1} \frac{\partial V_1}{\partial x} + \left[\left(\frac{V_0}{V_1} \right)^\lambda (p_0 + p_a) - p_a \right] \frac{\partial s_e}{\partial x} \quad (3)$$

$$\Downarrow$$

$$K_a \approx \lambda (p_0 + p_a) \left[\frac{\kappa^2}{V_1} x^2 + \frac{2\lambda (p_0 + p_a) \kappa s_0}{V_1} x + \frac{\lambda (p_0 + p_a) s_0^2}{V_1} \right] + p_0 \kappa$$

It could be figured out from Eq. (3) that the stiffness of air spring had a quadratic functional relationship with compression x and a proportional relationship with pressure p , namely, the force had a cubic relationship with compression x . Therefore, the stiffness of air spring could not be simply equivalent to a constant, which could affect the static balance of the magnetic-air hybrid quasi-zero stiffness vibration isolation system.

The parameters of air spring mathematical model could be obtained via the combination of experiment and fitting, which could be equivalently expressed as Eq.

(4). The comparison between experiment and fitting of air spring in Table 1 was displayed in Fig. 3.

$$\begin{cases} F_a = (aP + b)(cx^3 + dx^2 + ex + g) \\ K_a = (aP + b)(3cx^2 + 2dx + e) \end{cases} \quad (4)$$

Figure 3 Comparison of experiment and fitting

2.3 Mathematical model of negative stiffness element

To meet different state of static balance, the mathematical expression of electromagnetic spring was simplified to make it applicable in a certain range. At this time, the expression did not vary with the state of static balance. The state change of electromagnetic spring was shown in Fig. 4.

Figure 4 State change of electromagnetic spring

According to the electromagnetic theory and the structure of electromagnetic spring [22], an equation could be obtained as follows:

$$\begin{cases} F_z = 2k_s z - \frac{\mu_0 \mu_r AN^2 I^2}{4} \left(\frac{1}{(S-z)^2} - \frac{1}{(S+z)^2} \right) \\ K_z = 2k_s - \mu_0 \mu_r AN^2 I^2 S \cdot \frac{S^2 + 3z^2}{(S^2 - z^2)^3} \end{cases} \quad (5)$$

When $z = S$, from Eq. (6), the stiffness K_z approximately had a quadratic function relationship with I and z . If $z = nS$ ($0 \leq n \leq 1$), Eq. (5) could be expressed as:

$$\begin{cases} F_z = \alpha I^2 z^3 + \beta I^2 z^2 + (\gamma I^2 + \xi) z + \delta I^2 \\ K_z = 3\alpha I^2 z^2 + 2\beta I^2 z + \gamma I^2 + \xi \end{cases} \quad (6)$$

Taking the theoretical Eq. (5) as the data sample, this paper analyzed the reliability of Eq. (6) at the 95% confidence level. We imported the parameters in Table 1 into Eq. (5) to obtain the residual analysis results, as was showed in Fig. 5. Fig. 5 indicated that when $n \rightarrow 0$, the change of initial magnetic field B was excessively small, and when $n \rightarrow 0.7$, the nonlinearity of electromagnetic force was greatly strengthened. It followed that Eq. (6) was not credible because it was not within the confidence interval. Therefore, when $n \in (0.05, 0.6)$, the expression of electromagnetic force could be simplified to Eq. (6). In other words, the equation remained constant within this range.

Figure 5 Residual analysis

According to Eq. (6), when the current was constant ($I = I_b$), the vibration isolation system was a passive quasi-zero stiffness isolator with variable parameters; when the current $I = I_b + \Delta i$, the system became an active-passive isolator. Since the dynamics and control strategy of active-passive isolator were under research, they may not be described in depth in this paper.

3. Mechanical model and stability analysis of quasi-zero stiffness vibration isolation system

3.1 Mechanical model and judgement criterion

According to Fig. 2 and Fig. 4, if $y = x - h$, the mechanical model of magnetic-air hybrid quasi-zero stiffness vibration isolation system could be deduced as:

$$\begin{aligned}
 F &= F_a + 4F_z \cdot \tan\theta \\
 &= (aP + b) \left[c(y + h)^3 + d(y + h)^2 + e(y + h) + g \right] + \\
 &4 \left[\left(\alpha I^2 \left(\sqrt{L^2 - y^2} - a \right)^3 + \beta I^2 \left(\sqrt{L^2 - y^2} - a \right)^2 + (\gamma I^2 + \xi) \left(\sqrt{L^2 - y^2} - a \right) + \delta I^2 \right) \right] \frac{-y}{\sqrt{L^2 - y^2}}
 \end{aligned} \tag{7}$$

We input the parameters in Table 1 into Eq. (7), then the force and the stiffness of the system were displayed in Fig. 6.

Figure 6 Force and stiffness of the system

Fig. 6 showed that the current I can change the force and stiffness of the system. When the structural parameters of vibration isolation system were determined, the quasi-zero stiffness could be adjusted by changing the current I . Due to the nonlinear characteristic of air spring and electromagnetic spring, the static balance was not at the state of $y=0$ ($x=h$). Therefore, $K|_{y=0}=0$ could not be simply used as the judgement criterion of quasi-zero stiffness.

On the basis of the stability of the system and the theory of quasi-zero stiffness, the paper put forward the following judgement criterion of quasi-zero stiffness of the magnetic-air hybrid quasi-zero stiffness vibration isolation system:

$$\text{When } p = \text{certain}, \exists I \text{ make } \begin{cases} \exists y, K \rightarrow 0 \text{ and } \frac{\partial K}{\partial y} = 0 \\ \forall y, K \geq 0 \end{cases} (y_{\min} \leq y \leq y_{\max}, I_{\min} \leq I \leq I_{\max}) \quad (8)$$

According to the judgement criterion of Eq. (8) and the structure corresponding to Table 1, current I and static balance position y of quasi-zero stiffness were shown in Fig. 7.

(a) Stiffness and derivative of stiffness

(b) Plan of (a)

Figure 7 Current I and static balance position y

3.2 Stability analysis

The analysis methods of kinematics of nonlinear system can be divided into qualitative analysis and quantitative analysis [23]. Among them, quantitative analysis can directly reflect the main characteristics of nonlinear system. Phase plane method, one of the most intuitive qualitative analysis methods, describes the kinematics of the system through phase trajectory [24, 25].

In this paper, the vibration isolation system mainly isolated vibration from the ground and reduced the external interference to the equipment installed on vibration isolation system. The principle was shown in Fig. 8. x_j and x were the displacement signal of ground vibration and the response displacement signal of equipment, respectively.

Figure 8 Principle of vibration isolation

The dynamics equation of the system is as follows:

$$c(\dot{x} - \dot{x}_j) + F - mg = m\ddot{x}. \quad (9)$$

We assumed that the ground vibration interference was $x_j = A\cos(\omega t)$ and brought it into Eq. (9). The phase trajectory in the coordinate plane of $x - \dot{x}$ was then obtained. Based on the parameters in Table 1, the phase trajectory at $\omega = (1\text{Hz}, 3\text{Hz}, 5\text{Hz}, 7\text{Hz}, 10\text{Hz}, 15\text{Hz})$ was displayed in Fig. 9.

Figure 9 Phase trajectory

From the phase trajectory under different frequency displacement interference, it could be figured out that the vibration isolation system showed good stability. The comparative analysis between interference signal and response signal indicated that when the frequency of external interference signal was 5Hz ($\omega = 5\text{Hz}$), the system had favorable vibration isolation effect in terms of the displacement signal; while no vibration isolation effect could be found in terms of the speed signal. This suggested that the natural frequency of the magnetic-air hybrid quasi-zero stiffness vibration isolation system was about 3 Hz or less. The phase trajectory not only verified the stability of the system, but also indirectly demonstrated the low dynamic stiffness of the vibration isolation system.

4. Dynamics and simulation

4.1 Dynamics analysis

Based on Fig. 8 and Eq. (9), the dynamics equation from the ground to the equipment was established to analyze the favorable effect of the vibration isolation system on external interference. According to the above equations, the pressure p and current I didn't change with time, and the force of the system was a cubic function of the compression y . At the same time, the mass supported by the system had nothing to do with dynamic stiffness, but was related to the initial height of air spring. Therefore, we assumed that, when the system approached static balance, the force expression of

the system was approximately defined as $F=k\Delta x^3 + mg$.

Assuming $\mu = x - x_j$ and $x_j = X_j \cos(\omega t)$, the dynamic differential equation was expressed as:

$$m\ddot{\mu} + \frac{c}{m}\dot{\mu} - \frac{k}{m}(-\mu)^3 = \omega^2 X_j \cos(\omega t). \quad (10)$$

According to the theory of harmonic balance method, if $\mu = A \sin(\omega t + \varphi)$, Eq. (10) could be equivalent to:

$$-\omega^2 A \sin(\omega t + \varphi) + \frac{c}{m} \omega A \cos(\omega t + \varphi) - \frac{k}{m} (-A \sin(\omega t + \varphi))^3 = \omega^2 X_j \cos(\omega t). \quad (11)$$

By solving dynamic equations, the amplitude frequency characteristic and phase frequency characteristic could be obtained:

$$\begin{cases} (\omega^2 X_j)^2 = \left(\omega^2 A - \frac{3kA^3}{4m} \right)^2 + \left(\frac{c\omega A}{m} \right)^2 \\ \tan \varphi = \frac{-\omega m}{c} + \frac{3kA^2}{4c\omega} \end{cases} \quad (12)$$

On the basis of mathematical theory, when the two roots were equal, the quadratic function took the maximum value. Thus, when $\omega_1 = \omega_2$, the displacement amplitude A reached the maximum value A_{max} , and the expression was shown in Eq. (13):

$$A_{max} = \sqrt{\frac{c^4}{3kmc^2 - \frac{9}{4}k^2m^2X_j^2}}. \quad (13)$$

Eq. (13) indicated that the denominator approached zero, namely, $3kmc^2 - \frac{9}{4}k^2m^2X_j^2 \rightarrow 0$. If the denominator approached zero, the maximum displacement amplitude A_{max} approached infinity. In other words, in the resonance region, the displacement response of the system was an infinity. Therefore, to achieve better performance, the following requirements shall be met:

$$3kmc^2 - \frac{9}{4}k^2m^2X_j^2 > 0 \Rightarrow 4c^2 > 3kmX_j^2. \quad (14)$$

Displacement transfer rate [26], an important index of vibration isolator, reflect the ability of vibration isolator to isolate external interference. Based on the previous dynamical model, the displacement transfer rate of the system against ground interference was expressed as:

$$\begin{aligned} \eta &= 20 \cdot \log \left(\frac{|x|}{|x_j|} \right) = 20 \cdot \log \left(\frac{|x_j - y|}{|x_j|} \right) = 20 \cdot \log \left(\frac{\sqrt{(x_j - y)^2}}{|x_j|} \right) \\ &= 20 \cdot \log \left(\frac{\sqrt{X_j^2 - A^2 + 2X_j A \cos(\varphi)}}{X_j} \right) \end{aligned} \quad (15)$$

4.2 Simulation analysis

According to the dynamical model of the system, the simulation model was developed. We chose sinusoidal displacement excitation applied on the base as the

simulation interference. The amplitude of sinusoidal displacement excitation in this paper was 5 mm and the simulation step was 0.001s. The simulation diagram was displayed in Fig. 10.

Figure 10 Simulation diagram

The displacement transfer rate between the response signal of the equipment and the excitation signal of the base was analyzed when $w = (1Hz, 3Hz, 5Hz, 7Hz, 10Hz, 15Hz)$. The simulation results were as follows.

Figure 11 Simulation results

The simulation results revealed that the magnetic-air hybrid quasi-zero stiffness vibration isolation system had good effect of vibration isolation from $w = 5Hz$. Therefore, we verified the natural frequency of the vibration isolation system was about 3Hz or less. The displacement transfer rates η between 5Hz and 10Hz (interval 1Hz) were respectively -16.07dB, - 21.8dB, - 25.06dB, - 27.26dB, - 30.33dB and -33.37dB,

which suggested that the system had good effect of low frequency vibration isolation and could effectively resist external low frequency interference.

5. Vibration isolation experiment

To verify the performance of the system against external low frequency interference, we built a magnetic-air hybrid quasi-zero stiffness vibration isolation platform, as was shown in Fig. 12. Due to the limitation of experimental conditions, the acceleration signal was collected to analyze the effect of vibration isolation.

Based on the experimental platform, two groups of experiments were carried out. One group was aimed to analyze different vibration isolated objects. We chose 13.5kg, 16kg and 21kg respectively as the mass of equipment. In addition, the excitation signal was the sinusoidal signal with frequency of 5Hz generated by the actuator. The experimental results were shown in Figs 13-15. The other group was designed to investigate the effect of vibration isolation under different low frequency interference. The external interference was sinusoidal excitation with frequency of 3Hz, 5Hz and 7Hz. Additionally, the mass was a fixed value and the pressure of air spring was 0.1 MPa. Figs 16-18 showed the experimental results. The transfer rate of all experiments was displayed in Table 2.

Figure 12 Experimental platform

Figure 13 Experimental results of vibration isolation with mass of 13.5kg

Figure 14 Experimental results of vibration isolation with mass of 16kg

Figure 15 Experimental results of vibration isolation with mass of 21kg

Figure 16 Experimental results of vibration isolation when $w = 3\text{Hz}$

Figure 17 Experimental results of vibration isolation when $w = 5\text{Hz}$

Figure 18 Experimental results of vibration isolation when $w = 7\text{Hz}$

Table 2 Transfer rate

	13.5Kg	16Kg	21Kg	3Hz	5Hz	7Hz
η	-10.29dB	-6.41dB	-5.73dB	-8.80dB	-7.09dB	-9.29dB

According to Figs 13-15 and Table 2, when the mass was 13.5kg, 16kg and 21kg, respectively, the vibration isolation effects are, respectively, 10.29dB, 6.41dB and 5.72dB. At this time, the magnetic-air hybrid vibration isolation system showed high static stiffness and good effect of low frequency vibration isolation. In short, the magnetic-air hybrid quasi-zero stiffness system could adapt to equipment of different mass.

On the basis of Figs 16-18 and Table 2, when the frequency of external excitation was 3Hz, 5Hz and 7Hz, respectively, the vibration isolation effect could respectively reach 10.29 dB, 6.41 dB and 5.72 dB. Experiments indicated that the natural frequency of the vibration isolation system was lower than 3Hz. In other words, the magnetic-air hybrid vibration isolation system showed low dynamic stiffness. To sum up, in a certain low frequency vibration range, the magnetic-air hybrid quasi-zero stiffness system had a good low frequency vibration isolation effect.

6. Conclusion

Based on the unique characteristics of quasi-zero stiffness theory in the field of nonlinear vibration isolation and our previous research on the structure of magnetic-air hybrid quasi-zero stiffness vibration isolation system, the mathematical model expression of positive and negative stiffness elements has been further analyzed to improve the application of the mathematical model. The mathematical model of vibration isolation system has been established, and the judgement criterion of magnetic-air hybrid quasi-zero stiffness has been analyzed. According to the new judgement criterion, it has been figured out that the static balance is not at $y = 0$ ($x = h$) due to the nonlinearity of air spring and electromagnetic spring. In this paper, external interference is considered as excitation signal, while vibration of the equipment as response signal. Employing the phase trajectory method, we have validated the stability of the vibration isolation system and that the natural frequency of the system is within 3Hz.

Additionally, this paper has developed the dynamical model and the displacement

transfer rate. The low frequency vibration isolation effect of the vibration isolation system has been simulated and analyzed. The simulation results reveal that the magnetic-air hybrid quasi-zero stiffness vibration isolation system shows good performance of low frequency vibration isolation. Finally, magnetic-air hybrid quasi-zero stiffness vibration isolation platform has been built to verify the adaptability of the system and the vibration isolation effect of low frequency interference. The final experimental results indicate that the magnetic-air hybrid quasi-zero stiffness vibration isolation system can adapt to different equipment whose mass is in a certain range. Meanwhile, for different external low frequency interference, the system can achieve vibration isolation effect of more than 7dB. The vibration isolation experiment, therefore, indirectly validate the stiffness of high static and low dynamic.

Declaration of competing interest

The authors declare that there was no conflicts of interest regarding the publication of this paper.

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Data availability

The data used to support the findings of this study have been included within the article

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