Research Article

Design and Research of Semiactive Quasi-Zero Stiffness Vibration Isolation System for Vehicles

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1. Introduction

Vibration isolation is a common method to eliminate or weaken vibration [1, 2]. For vehicle vibration, the traditional linear vibration isolator lacks the ability to isolate low-frequency vibration, and the vibration isolation effect is not obvious for 0–20 Hz low-frequency vibration, even having an amplification effect [3]. In transportation, vehicle-mounted equipment is easily disturbed by multiple external vibration sources, which have adverse effects on the accuracy, performance, and the service life of the equipment. Thus, it is quite necessary to design the vibration isolation suitable for excitations with different frequencies to ensure the accuracy and reliability of the vehicle-mounted equipment.

In recent years, QZS vibration isolation has become a research hotspot because of its large bearing capacity and extremely low natural frequency, which can effectively isolate low frequencies. It has various forms, such as cam roller [4–6], oblique spring [7–9], disk spring combined with vertical linear springs [10], circular ring [11], magnet [12], inert elements [13], X-shaped structure [14], strut structure [15, 16], and other structural forms [17–20]. And some researchers studied the resonance response of nonlinear vibration [21] and the damping characteristics [22]. Most of the above literatures focus on the QZS vibration isolation characteristics, but the structure and parameters of the uncontrolled QZS vibration isolation system cannot be changed once determined, which is difficult to meet the complex and changeable working...
conditions and restricts the further improvement and universality of the vibration isolation system.

Due to low energy consumption, magnetorheological (MR) damper has been widely used to implement the semiactive control [23, 24]. The modelling of hysteretic characteristics of dampers can be divided into two types: modelling based on physical mechanism and modelling based on external characteristics of macro phenomena. The typical ones are Bouc–Wen model, Preisach model, Duham model, and neural network model, which can reflect the complex hysteretic characteristics of dampers with multiloops, multibranches, and nonsmoothness. The differential hysteretic model is widely used, such as Duham model [25].

At present, MR damper is widely used in vehicle suspension control [26–29] and gradually becomes a research hotspot in vehicle vibration isolation. Hu et al. [30] designed a kind of MR damper valve and verified the good damping performance under the sky-hook on-off semiactive control strategy. Dong Chae and Choi [31] proposed a vehicle-mounted vibration isolation system. Using the controllable damping force characteristics of MR damper, the semiactive control effect of the system was analysed and the vibration isolation performance was evaluated. Chae et al. designed a semiactive vibration reduction system for vehicle-mounted stretchers, which used MR dampers as the actuator and realized the vibration isolation control through sliding mode variable structure semiactive control. Gao et al. [32] improved a 4-PUU parallel mechanism as a vehicle stretcher, took MR damper as the output force device, and adopted the LQR method combined with Hrovat algorithm to study the semiactive control and vibration isolation performance of the system. Wang et al. [33] studied the ultralow-frequency vibration isolation in the process of neonatal transport through theory and experiment. However, the research studies considering semiactive control and QZS isolator simultaneously are still seldom found.

In this work, a semiactive QZS vibration isolation system with positive and negative stiffness parallel mechanism is proposed. MR damper is installed in parallel to the vertical spring. Through the analysis of the characteristics of MR damper, an improved Bingham model based on excitation current and response speed is established. The semiactive on-off control scheme is developed, and the control effect is simulated under harmonic, stochastic, and semisinusoidal shock excitations at different vehicle speeds. Finally, the mechanical device of the semiactive QZS isolator is designed and manufactured, and the isolation effect of the system is tested based on LabVIEW software and PXI embedded system to verify the effectiveness of the semiactive control scheme and the good vibration isolation performance of the vibration isolation system.

2. Design of Semiactive QZS Vibration Isolation System

2.1. Modelling of Semiactive QZS Vibration Isolation System

The QZS vibration isolation system controlled by MR semiactive control is shown in Figure 1, which is composed of a positive and negative stiffness spring parallel mechanism and MR damper. Here, \( m \) is the mass of the vibration isolated object, \( k_c \) and \( k_h \) are the vertical spring stiffness and horizontal spring stiffness, respectively, and \( L \) is the precompressed length of the horizontal spring. When the system reaches the static equilibrium position, the horizontal spring’s length is \( l \). \( y \) is the vertical displacement of the mass object; \( P \) is the excitation force of the system. The damping coefficient is set as \( c_h \), and \( F_c \) is the damping force provided by MR.

According to D’Alembert’s principle, the dynamic equation of the MR semiactive vibration isolation system is obtained.

\[
m\ddot{y} + c_h\dot{y}\left(1 - \frac{L}{\sqrt{l^2 + y^2}}\right) + k_h\dot{y}\left(1 - \frac{L}{\sqrt{l^2 + y^2}}\right) + k_vy + F_c = k_vq - mg.
\]

(1)

Compared with the semiactive control system shown in Figure 1, the dynamic equation of the passive QZS vibration isolation system is

\[
m\ddot{y} + c_v\dot{y}\left(1 - \frac{L}{\sqrt{l^2 + y^2}}\right) + k_h\dot{y}\left(1 - \frac{L}{\sqrt{l^2 + y^2}}\right) + k_vy + c_v\dot{y} = k_vq - mg,
\]

(2)

where \( c_v \) is the vertical damping coefficient. To the passive QZS vibration isolation system, it is assumed that the excitation displacement is given by

\[
\dot{q} = A \cos(\omega t),
\]

(3)

where \( A \) is the excitation amplitude, \( \omega \) is the excitation frequency, and \( t \) is the time.
The approximate analytical solution of equation (5) can be solved by the averaging method as follows:

\[
\tilde{x} = \tilde{a} \cos (\omega \tau + \theta) \\
\tilde{x}' = -\tilde{a}' \omega \sin (\omega \tau + \theta)
\]

(5)

where \(\tilde{a}\) and \(\theta\) are slowly varying functions of time \(\tau\).

According to the averaging method, the amplitude and phase of the first-order approximate solution can be obtained as follows:

\[
\tilde{a}' = -\frac{1}{\tilde{a} \omega} p(\tilde{a}, \theta) \sin (\omega \tau + \theta)
\]

\[
\theta' = -\frac{1}{\tilde{a} \omega} p(\tilde{a}, \theta) \cos (\omega \tau + \theta)
\]

(6)

where

\[
p(\tilde{a}, \theta) = \tilde{a} \tilde{a}^2 \cos (\omega \tau + \theta) - 2 \tilde{a} \omega (\tilde{\xi}_1 + \xi_2) \sin (\omega \tau + \theta)
\]

\[
+ \frac{2 \tilde{a} \omega \xi_1 \sin (\omega \tau + \theta)}{\sqrt{\tilde{a}^2 + \tilde{a}^2 \cos^2 (\omega \tau)}} + \tilde{a} (1 + r) \cos (\omega \tau + \theta)
\]

\[
- \frac{\tilde{a} r \cos (\omega \tau + \theta)}{\sqrt{\tilde{a}^2 + \tilde{a}^2 \cos^2 (\omega \tau + \theta)}} - \mu \omega^2 \cos (\omega \tau + \theta).
\]

In one period of \([0, 2\pi]\), the approximate values of amplitude and phase are obtained as follows:

\[
\tilde{a}' = -\frac{1}{2\pi \tilde{a} \omega} \int_0^{2\pi} p(\tilde{a}, \theta) \sin \phi \ d\phi
\]

(8)

\[
\theta' = -\frac{1}{2\pi \tilde{a} \omega} \int_0^{2\pi} p(\tilde{a}, \theta) \cos \phi \ d\phi
\]

Let \(\tilde{a}' = 0\) and \(\theta' = 0\); the analytical expression of steady-state amplitude frequency response of the passive QZS vibration isolation system is obtained as follows:

\[
\omega = \sqrt{-H(\tilde{a}) \pm \sqrt{[H(\tilde{a})]^2 + \frac{4 \tilde{a}^2 \omega^2 (\xi_1 + \xi_2)^2 + 4 \tilde{a}^2 \xi_2^2 [j(\tilde{a})]^2}{\mu^2 - \tilde{a}^2}}}
\]

(9)

\[
\theta = \arctan \left( \frac{2 \tilde{a} \omega (\xi_1 + \xi_2) + 2 \tilde{a} \omega \xi_2 j(\tilde{a})}{-\tilde{a} (1 + r) + \tilde{a} \omega^2 + \tilde{a} r l(\tilde{a})/\pi} \right)
\]
where

\[ I(\bar{a}) = \int_0^{2\pi} \frac{\cos^2 \varphi}{\sqrt{\alpha^2 + \bar{a}^2 \cos^2 \varphi}} \, d\varphi, \]
\[ J(\bar{a}) = \int_0^{2\pi} \frac{\sin^2 \varphi}{\sqrt{\alpha^2 + \bar{a}^2 \cos^2 \varphi}} \, d\varphi, \]
\[ H(\bar{a}) = \left[ 4\bar{a}^2 \omega^2 (\xi_1 + \xi_2)^2 + \frac{4\bar{a}^2 \xi_1^2 [J(\bar{a})]^2}{\pi^2} - 2\bar{a}^2 (1 + r) + \frac{2\bar{a}^2 r I(\bar{a})}{\pi} \right] \cdot \left[ 2(\mu^2 - \bar{a}^2) \right]^{-1}. \] (10)

When the passive QZS vibration isolation system is excited by a harmonic force, the steady-state amplitude frequency response is as follows:

\[ z = \tilde{a} \cos (\omega t + \theta) + \mu \cos (\omega t). \tag{11} \]

Then, \( z_{\text{max}} = \sqrt{(\tilde{a} \cos \theta + \mu)^2 + (\tilde{a} \sin \theta)^2} \) can be obtained, and the displacement transmissibility of the passive QZS system is obtained as follows:

\[ T_f = \frac{z_{\text{max}}}{\mu}. \tag{12} \]

2.2. Characteristics of MR Damper and Semiactive Control Scheme. The damping force of MR damper is expressed by S-shape function as follows [34]:

\[ F_c = \frac{2A_c}{1 + \exp(-0.0333y)} - A_c, \tag{13} \]

where \( A_c \) is the limit saturation value of the damping force and \( \alpha \) is the shape parameter; if the damper structure remains unchanged, \( A_c \) remains unchanged.

According to the parameters of RD-1097-01 MR damper manufactured by Lord company, the resistance of the excitation coil is 20 Ω at 25°C, the maximum damping force is 135 N, the maximum continuous working current is 0.5 A, and the current tends to be saturated when it is greater than 1 A. Then, the damping force of the MR damper can be expressed as follows:

\[ F_c = \begin{cases} 
2A_c, & y \geq 0; \quad I = I_{\text{on}}, \\
A_c, & (y - \dot{q})y < 0; \quad I = 0,
\end{cases} \tag{14} \]

where \( \dot{q} \) is the relative velocity of the mass; \( \dot{y} \) is the velocity of the mass; \( f_d \) is the semiactive control damping force, which is related to the input current, and \( f_d \leq 135 \) N; \( f_{\text{min}} \) is the minimum value of the output force of the MR damper when the current is the minimum; and \( f_{\text{min}} = 0 \) when the current is zero.

By judging the direction of relative velocity \( \dot{y} - \dot{q} \) and absolute velocity \( \dot{y} \) of the controlled mass and controlling the switching of the excitation voltage or current between finite discrete values, the damping force is adjusted. Therefore, when the relative motion of the mass is consistent with the
direction of the damping force of the MR damper and the excitation current is on, the damper provides the maximum damping force to restrain the mass movement. On the contrary, the damper provides the minimum damping force to reduce the obstruction to the mass movement.

3. Simulation Analysis of Vibration Isolation Effect under Different Excitations

Figure 5 shows the cosimulation scheme of the semiactive MR vibration isolation system based on TruckSim and Matlab/Simulink. The vehicle model is an 8 × 8 vehicle established in TruckSim. The vertical displacement $x_c$, vertical velocity $v_c$, and vertical acceleration $a_c$ of the body centre are taken as the outputs, and $x_c$ is the input.

The parameters of the semiactive QZS vibration isolation system are shown in Table 1.

| Parameters | Values |
|------------|--------|
| $M$        | 9.2 kg |
| $k_v$      | 1 017 N/m |
| $k_h$      | 1 010 N/m |
| $c_h$      | 100 N·s/m |
| $L$        | 0.121 m |
| $L$        | 0.081 m |
| $H$        | 0.090 m |

3.1. Harmonic Road Excitation. A harmonic road excitation with an amplitude of 5 mm and length of 500 m is used, and the vehicle speed is 30 km/h, 40 km/h, and 50 km/h, respectively. The vibration isolation characteristics of the passive QZS and semiactive QZS isolators are obtained, as shown in Figures 6 and 7. The root mean square and relative differences of the vertical displacement and acceleration of the body centre and the two isolation systems are shown in Table 2 and Figure 8.

It can be seen from Figures 6–8 and Table 2 that

(a) From the vertical displacement response, RMS of the passive QZS vibration isolation system is reduced by 80.6% and the maximum RMS of the semiactive QZS vibration isolation system is reduced by 95.49%, compared with that of the body centre. The maximum vertical displacement reduction of semiactive QZS isolator is 78.1% better than that of the passive QZS isolator.
Compared with the passive vibration isolation system, the maximum vertical acceleration reduction of the semiactive QZS isolator is 74.2%.

Under 5 mm harmonic excitation, the two vibration isolation systems show good vibration isolation performance at different vehicle speeds, but the semiactive QZS vibration isolation system has better vibration isolation performance.

The change trend of body response of the passive and semiactive QZS is the same, but the maximum amplitude and RMS of the semiactive QZS are obviously decreased.

### 3.2. Stochastic Road Excitation.

In practical engineering, the external excitation of a vehicle is mostly random or has strong randomness. The road is built adopting the three-dimensional stochastic road based on fractal theory [35], and the irregularity of the stochastic road surface is given by

\[
q(t) = -2\pi f_0 v q(t) + 2\pi n_0 \sqrt{G_0} \omega(t),
\]

where \( q \) is the stochastic road, \( v \) is the speed, \( G_0 \) is the road irregularity coefficient, \( \omega(t) \) is the Gaussian white noise, \( f_0 \) is the cut-off space frequency, generally taken as 0.011 m\(^{-1}\), and \( n_0 \) is the reference space frequency, taken as 0.1 m\(^{-1}\).

The three-dimensional road spectrum can better reflect the three-dimensional texture characteristics, which not only reflect the longitudinal irregularity excitation of the road but also meet the requirements of the simulation test for the transverse elevation change, as shown in Figure 9.

Figures 10 and 11 show the vertical time domain response curves of the body centre and two kinds of vibration isolation systems under the condition of C-level stochastic road at different vehicle speeds. See Table 3 and Figure 12 for RMS and relative difference of the two vibration isolation systems and body centre response.

It can be seen from Figures 10–12 and Table 3 that for the road condition of C-level stochastic road excitation, the RMS.
of vertical displacement and acceleration of the passive and the semiactive isolators are significantly reduced, with the maximum reduction over 60%. The semiactive vibration isolator shows much better isolation performance than the passive one. At the same time, the passive QZS vibrator is greatly affected by the vehicle speed, the stability is poor, and the vibration isolation efficiency is reduced while the semiactive QZS vibration isolator is less affected by the excitation and vehicle speed, and the vibration isolation efficiency is over 88%.

3.3. Semisinusoidal Shock Road. To verify the shock performance of the isolators, a semisinusoidal shock road with an amplitude of 0.1 m and frequency of 1.4 Hz is established in TruckSim, as shown in Figure 13.

The vertical displacement and acceleration response curves of the body centre and the two vibration isolators are shown in Figures 14 and 15. The RMS and relative differences of the vertical displacement and acceleration response are shown in Table 4 and Figure 16.

As can be seen from Figures 14–16 and Table 4,

(a) The isolation effect of the semiactive QZS isolator is clearly better than that of the passive one under semisinusoidal shock road excitation.

(b) With the increase in vehicle speed, the isolation efficiency of the passive QZS decreases, the response peak value increases, and the stability is poor. While the vibration isolation efficiency of the semiactive control system is about 90%, its stability is better than that of the passive QZS system.

To evaluate the shock resistance performance, the ratio of the peak acceleration response of the isolated object to that of the vehicle body centre is defined as the maximum acceleration ratio, as shown in Table 5.
Table 2: RMS and relative differences of displacement and acceleration under harmonic excitation.

| Vehicle speed (km·h⁻¹) | Body centre | Passive QZS system | Semiactive QZS system |
|------------------------|-------------|---------------------|-----------------------|
|                        | Displacement (m) | Acceleration (m·s⁻²) | Displacement difference (%) | Acceleration (m·s⁻²) | Displacement (%) | Acceleration (m·s⁻²) | Acceleration (%) |
| 30                     | 1.16 × 10⁻³ | 0.322               | 2.25 × 10⁻⁴          | −80.60              | 0.066           | −79.50              | 7.68 × 10⁻⁵      | −93.38          | 0.017           | −94.72          |
| 40                     | 1.64 × 10⁻³ | 0.374               | 3.42 × 10⁻⁴          | −79.15              | 0.083           | −77.81              | 8.93 × 10⁻⁵      | −94.55          | 0.022           | −94.12          |
| 50                     | 2.17 × 10⁻³ | 0.396               | 4.48 × 10⁻⁴          | −79.35              | 0.093           | −76.52              | 9.79 × 10⁻⁵      | −95.49          | 0.027           | −93.18          |
Figure 8: Relative difference between the vibration isolation system and vehicle body under harmonic excitation: (a) displacement; (b) acceleration.

Figure 9: C-level stochastic road excitation: (a) cross section elevation setting; (b) road model.

Figure 10: Continued.
Figure 10: Time domain response of displacement under stochastic road: (a) 30 km/h; (b) 40 km/h; (c) 50 km/h.

Figure 11: Time domain response of acceleration under stochastic road: (a) 30 km/h; (b) 40 km/h; (c) 50 km/h.
Table 3: RMS and relative differences of displacement and acceleration response under stochastic road.

| Vehicle speed (km·h⁻¹) | Body centre Displacement (m) | Body centre Acceleration (m·s⁻²) | Displacement difference (%) | Passive QZS system Displacement (m) | Passive QZS system Acceleration (m·s⁻²) | Displacement difference (%) | Semiactive QZS system Displacement (m) | Semiactive QZS system Acceleration (m·s⁻²) | Displacement difference (%) |
|------------------------|-----------------------------|----------------------------------|-----------------------------|-------------------------------------|----------------------------------------|-----------------------------|----------------------------------------|----------------------------------------|-----------------------------|
| 30                     | 0.011                       | 0.742                            | 2.37 × 10⁻³                  | 0.242                               | −78.45                                 | 1.67 × 10⁻³                  | −84.82                                 | 0.074                                  | −90.03                       |
| 40                     | 0.013                       | 1.163                            | 3.54 × 10⁻³                  | 0.441                               | −72.77                                 | 2.04 × 10⁻³                  | −84.31                                 | 0.126                                  | −89.17                       |
| 50                     | 0.021                       | 1.654                            | 5.89 × 10⁻³                  | 0.647                               | −71.95                                 | 3.35 × 10⁻³                  | −84.05                                 | 0.187                                  | −88.69                       |
Figure 12: Relative difference of RMS of (a) displacement and (b) acceleration under C-level random pavement.

Figure 13: Semisinusoidal shock road: (a) cross section elevation setting; (b) road model.

Figure 14: Continued.
Figure 14: Time domain response curve of displacement under semisinusoidal shock road: (a) 30 km/h; (b) 40 km/h; (c) 50 km/h.

Figure 15: Time domain response curve of acceleration under semisinusoidal shock road: (a) 30 km/h; (b) 40 km/h; (c) 50 km/h.
Table 4: RMS and relative differences of vertical displacement and acceleration under semisinusoidal shock road.

| Vehicle speed (km·h⁻¹) | Displacement (m) | Body centre Acceleration (m·s⁻²) | Displacement (m) | Displacement difference (%) | Passive QZS system Acceleration (m·s⁻²) | Displacement difference (%) | Semiactive QZS system Acceleration (m·s⁻²) | Acceleration difference (%) |
|------------------------|------------------|----------------------------------|------------------|----------------------------|---------------------------------|----------------------------|---------------------------------|----------------------------|
| 30                     | 0.032            | 4.278                            | 0.012            | -62.50                     | 1.267                           | -70.38                     | 2.59 × 10⁻³                     | -91.91                     |
| 40                     | 0.057            | 6.096                            | 0.023            | -59.65                     | 2.118                           | -65.26                     | 5.65 × 10⁻³                     | -90.09                     |
| 50                     | 0.064            | 7.257                            | 0.029            | -54.69                     | 2.626                           | -63.81                     | 9.37 × 10⁻³                     | -85.36                     |
It can be seen from Table 5 that the maximum acceleration ratio of the two isolators at different vehicle speeds increases with the increase in vehicle speed. The maximum acceleration ratio of the passive QZS vibration isolation system is larger than that of the semiactive one, which indicates that the semiactive QZS vibration isolator has better shock resistance performance.

4. Experimental Study on Vibration Isolation System

4.1. Experiment Scheme Design. The semiactive QZS vibration isolation system is composed of a mechanical structure and MR semiactive control system. The mechanical structure is shown in Figure 17. Here, the negative stiffness mechanism mainly includes a spring, an inner and outer sleeve, a screw, and an adjusting nut. The screw is connected to the inner sleeve in a spiral manner. The precompression of the spring is adjusted by the adjusting nut to realize the negative stiffness of the horizontal spring.

The MR semiactive control system is mainly composed of a motion state sensor, controllable constant current power supply, signal conditioning converter, control arithmetic unit, input/output board, and shielded junction box, as shown in Figure 18.

Using LabVIEW RT as the real-time control module can improve the reliability and time certainty of program operation. The program is written and debugged in the upper computer, and the running state of the system is monitored. The lower computer is connected to the upper computer through the network cable to ensure the real-time performance of the system and realizes the functions of data transmission and human-computer interaction.

The hardware of the semiactive control system is listed in Table 6, and the experiment site is shown in Figure 19.

The excitation system is a six-degree-of-freedom vibration test bench jointly developed by the University of Wollongong and Hefei University of Technology, which is mainly composed of NI control system, PC computer controller, DMKE electric cylinder, and so on. The data acquisition system includes keyence LK-G500 laser displacement sensors and data collectors (model: INV306U), and the real-time waveforms are captured through DASP software on PC computed.

4.2. Analysis of Test Results. The design parameter of the spring is shown in Table 7.

When the QZS system is in a static balance, the following relationship should be satisfied:

$$L^2 = l^2 + h^2,$$

where $L$ is the precompressed length of the horizontal spring, which can be adjusted by the adjusting nut. When the system reaches the static equilibrium position, the horizontal spring’s length is $l$. $h$ is the compression deformation of the

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**Table 5: Maximum acceleration ratio at different vehicle speeds under semisinusoidal impact pavement.**

| Vehicle speed (km·h$^{-1}$) | Body centre Peak acceleration (m·s$^{-2}$) | Passive QZS system Peak acceleration (m·s$^{-2}$) | Ratio | Semiactive QZS system Peak acceleration (m·s$^{-2}$) | Ratio |
|-----------------------------|-------------------------------------------|-----------------------------------------------|-------|-----------------------------------------------|-------|
| 30                          | 9.762                                     | 3.356                                         | 0.34  | 0.834                                         | 0.086 |
| 40                          | 17.187                                    | 6.195                                         | 0.36  | 1.787                                         | 0.103 |
| 50                          | 21.158                                    | 8.791                                         | 0.42  | 2.489                                         | 0.118 |

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**Figure 16**: Relative difference of RMS of (a) vertical displacement and (b) acceleration under semisinusoidal shock road.
vertical spring under static load, which can be obtained by the relationship $k_h = mg$. The meaning and value of $L$, $l$, and $h$ are shown in Figure 1 and Table 1.

The test conditions include harmonic excitation with different amplitudes and frequencies:

(a) Test Condition 1. Harmonic excitation with different amplitudes: The initial vibration isolation frequency of the QZS vibration isolation system is about 1.4 Hz, so the resonance region frequency (1.2 Hz) and the vibration isolation region frequency (1.4 times of the resonance frequency, 2.5 Hz) are selected as the excitation frequency. The vertical amplitudes are chosen as 3 mm, 5 mm, and 7 mm, respectively.

(b) Test Condition 2. Harmonic excitation test with different frequencies. The amplitude is 5 mm, and the excitation frequencies are 1.0 Hz, 1.2 Hz, 1.4 Hz, 1.6 Hz, and 1.8 Hz, respectively.

4.2.1. Time Domain Response Analysis. When the excitation frequency is 1.2 Hz and 2.5 Hz, the test and simulation results of the displacement response of the isolated object are shown in Figures 20 and 21. The differences between the test and the theoretical results are shown in Table 8.

From Figures 20 and 21 and Table 8, it can be seen that the test results are in good agreement with the theoretical results. The peak value of the test is slightly larger than the simulation.
Figure 19: Experiment site: ① control computer; ② control program interface; ③ PXI embedded system; ④ data acquisition computer; ⑤ acquisition instrument; ⑥ regulated power supply; ⑦ MR damper; ⑧ QZS vibration isolator; ⑨ laser displacement meter.

Table 7: The parameter of the spring.

| Medium diameter (mm) | Material diameter (mm) | Effective laps | Measured stiffness (N/m) |
|----------------------|------------------------|----------------|-------------------------|
| Vertical spring      | 65                     | 4              | 8                       | 1017                     |
| Oblique spring       | 42                     | 2.8            | 7.5                     | 1085                     |

Figure 20: Time domain response of displacement under 1.2 Hz harmonic excitation with different amplitudes: (a) 3 mm; (b) 5 mm; (c) 7 mm.
results, and the difference increases with the excitation amplitude. The slight difference is inevitable due to the installation error and the friction between the parts. So, the test results verify the correctness of the established model and simulation results.

The vibration experiment of the semiactive QZS system with different harmonic excitation frequencies is carried out. The amplitude is 5 mm, and the frequencies are 1.0 Hz, 1.2 Hz, 1.4 Hz, 1.6 Hz, and 1.8 Hz, respectively. The tested and simulated displacement responses of the mass block under different excitation frequencies are shown in Figure 22. The relative differences of the peak values are listed in Table 9.

From Figure 22 and Table 9, it can be seen that the change trend of the experiment and the simulation results are the same. With the increase in the excitation frequency, the relative difference between the experiment and the theoretical results increases, and the maximum difference is 11.27%, which further verifies the correctness of the theoretical model and the feasibility of the control scheme.

![Figure 21](image1.png)

**Figure 21:** Time domain response of displacement under 2.5 Hz harmonic excitation with different amplitudes: (a) 3 mm; (b) 5 mm; (c) 7 mm.

| Excitation amplitude (mm) | Excitation frequency 1.2 Hz | Excitation frequency 2.5 Hz |
|---------------------------|-----------------------------|-----------------------------|
|                           | Simulation | Experiment | Difference (%) | Simulation | Experiment | Difference (%) |
| 3                         | 3.32       | 3.451      | 5.12           | 0.96       | 1.047      | 1.047          |
| 5                         | 6.05       | 6.456      | 6.71           | 1.05       | 1.261      | 10.54          |
| 7                         | 9.17       | 9.718      | 5.97           | 1.47       | 1.613      | 9.76           |

**Table 8:** Relative differences of displacement peak between experiment results and simulation results.
Figure 22: Time domain response of displacement under different excitation frequencies: (a) 1 Hz, (b) 1.2 Hz, (c) 1.4 Hz, (d) 1.6 Hz, and (e) 1.8 Hz.
4.2.2. Analysis of Vibration Isolation Effect. Let $a_{\text{max}}$ and $a_{\text{min}}$ be the peaks and troughs of the time domain response waveforms, respectively, and $b$ be the excitation amplitude; the displacement transmissibility of the vibration isolation system is

$$T = \frac{|a_{\text{max}} - a_{\text{min}}|}{2b}$$

(18)

The tested and simulated displacement transmissibility of the passive and semiactive QZS isolators under different excitation amplitudes is shown in Figure 23.

As can be seen from Figure 23,

(a) The resonance peak of the system increases with the rise of the excitation amplitude. The initial vibration isolation frequencies of the two kinds of isolators are

Table 9: Displacement peaks from experiment results and simulation results under different excitation frequencies.

| Excitation frequency (Hz) | Simulation (mm) | Experiment (mm) | Difference (%) |
|--------------------------|----------------|----------------|----------------|
| 1.0                      | 6.01           | 6.377          | 5.93           |
| 1.2                      | 6.05           | 6.456          | 6.71           |
| 1.4                      | 4.04           | 4.369          | 8.14           |
| 1.6                      | 2.76           | 3.071          | 11.27          |
| 1.8                      | 2.02           | 2.234          | 10.56          |

Figure 23: Displacement transmissibility under different harmonic excitation: (a) amplitude is 3 mm; (b) amplitude is 5 mm; (c) amplitude is 7 mm.
A semiactive QZS vibration isolator is proposed and designed based on MR damper. The simulation analysis is carried out under different road conditions and different vehicle speeds. The test device and semiactive on-off control system are developed and manufactured, and the correctness of the theoretical derivation and simulation method is verified by experimental results. It can be concluded that

(a) For the condition of harmonic, stochastic, and shock road excitations, the semiactive QZS isolator is always superior to the passive QZS in different working conditions, with more obvious control effects.

(b) The proposed semiactive QZS isolator shows better universality at different frequencies and amplitudes of excitations in the test, and the control algorithms are feasible for the mechanical devices of the isolator and the hardware system.

### 5. Conclusions

A semiactive QZS vibration isolator is proposed and designed based on MR damper. The simulation analysis is carried out under different road conditions and different vehicle speeds. The test device and semiactive on-off control system are developed and manufactured, and the correctness of the theoretical derivation and simulation method is verified by experimental results. It can be concluded that

(a) For the condition of harmonic, stochastic, and shock road excitations, the semiactive QZS isolator is always superior to the passive QZS in different working conditions, with more obvious control effects.

(b) The proposed semiactive QZS isolator shows better universality at different frequencies and amplitudes of excitations in the test, and the control algorithms are feasible for the mechanical devices of the isolator and the hardware system.

### Data Availability

The data used to support the findings of this study are included within the article.

### Conflicts of Interest

The authors declare that they have no conflicts of interest regarding the publication of this paper.

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### Table 10: RMS and differences of displacement transmissibility under different harmonic excitations.

| Excitation amplitudes (mm) | RMS of passive QZS (mm) | RMS of semiactive QZS (mm) | Differences (%) |
|----------------------------|-------------------------|----------------------------|-----------------|
| 3                          | 0.907                   | 0.672                      | -25.9           |
| 5                          | 1.166                   | 0.684                      | -41.3           |
| 7                          | 1.474                   | 0.698                      | -52.6           |

### References

[1] Z. Lu, Z. Wang, Y. Zhou, and X. Lu, “Nonlinear dissipative devices in structural vibration control: a review,” *Journal of Sound and Vibration*, vol. 423, pp. 18–49, 2018.

[2] Z. Lu, D. Wu, H. Ding, and L. Chen, “Vibration isolation and energy harvesting integrated in a Stewart platform with high static and low dynamic stiffness,” *Applied Mathematical Modelling*, vol. 89, no. 1, pp. 249–267, 2020.

[3] N. Du, M. Hu, Y. Bi, and Q. Zhu, “A low frequency horizontal vibration reduction method for vehicle mounted equipment,” *Journal of Vibration and Shock*, vol. 36, no. 7, pp. 184–190, 2017.

[4] Q. Zhang, S. Xia, D. Xu, and Z. Peng, “A torsion-translational vibration isolator with quasi-zero stiffness,” *Nonlinear Dynamics*, vol. 99, no. 2, pp. 1467–1488, 2019.

[5] Y. Kan, J. C. Ji, and B. Terry, “Design of a quasi-zero stiffness isolation system for supporting different loads,” *Journal of Sound And Vibration*, vol. 471, Article ID 115198, 2020.

[6] J. Han, J. Sun and L. Meng, “Design and characteristics analysis of a nonlinear vibration isolator using a curved surface-spring-roller mechanism as negative stiffness element,” *Journal of Vibration and Shock*, vol. 38, no. 3, pp. 170–178, 2019.

[7] K. Wang, J. Zhou, D. Xu, and H. Ouyang, “Lower band gaps of longitudinal wave in a one-dimensional periodic rod by exploiting geometrical nonlinearity,” *Mechanical Systems and Signal Processing*, vol. 124, pp. 664–678, 2019.

[8] Q. Meng, X. Yang, W. Li, E. Lu, and L. Sheng, “Research and analysis of quasi-zero-stiffness isolator with geometric nonlinear damping,” *Shock and Vibration*, vol. 2017, Article ID 6719054, 9 pages, 2017.

[9] D. Liu, Y. Liu, D. Sheng, and W. Liao, “Seismic response analysis of an isolated structure with QZS under near-fault vertical earthquakes,” *Shock and Vibration*, vol. 2018, Article ID 9149721, 12 pages, 2018.

[10] L. Meng, J. Sun, W. Wu, and K. V. Singh, “Theoretical design and characteristics analysis of a quasi-zero-stiffness isolator using a disk spring as negative stiffness element,” *Shock and Vibration*, vol. 2015, Article ID 813763, 19 pages, 2015.

[11] Z. Q. Lu, D. H. Gu, H. Ding, W. Lacarbonara, and L. Q. Chen, “Nonlinear vibration isolation via a circular ring,” *Mechanical Systems and Signal Processing*, vol. 136, Article ID 106490, 2020.

[12] B. Yan, H. Ma, B. Jian, K. Wang, and C. Wu, “Nonlinear dynamics analysis of a bi-state nonlinear vibration isolator with symmetric permanent magnets,” *Nonlinear Dynamics*, vol. 97, no. 4, pp. 2499–2519, 2019.

[13] N. Alujević, D. Čakmak, H. Wolf et al., “Passive and active vibration isolation systems using inerter,” *Journal of Sound and Vibration*, vol. 418, pp. 163–183, 2018.

[14] Y. Wang and X. Jing, “Nonlinear stiffness and dynamical response characteristics of an asymmetric X-shaped structure,” *Mechanical Systems and Signal Processing*, vol. 125, pp. 142–169, 2019.
[15] X. Liu, X. Huang, H. Hua et al., “Influence of excitation amplitude and load on the characteristics of quasi-zero stiffness isolator,” Journal of Mechanical Engineering, vol. 49, no. 6, pp. 89–94, 2013.

[16] X. Huang, Y. Chen, H. Hua et al., “Shock isolation performance of a nonlinear isolator using Euler buckled beam as negative stiffness corrector: theoretical and experimental study,” Journal of Sound and Vibration, vol. 345, pp. 178–196, 2015.

[17] Y. Zhou, P. Chen, and G. Mosqueda, “Analytical and numerical investigation of quasi-zero stiffness vertical isolation system,” Journal of Engineering Mechanics, vol. 145, no. 6, Article ID 04019035, 2019.

[18] H. Yasuda, Y. Miyazawa, C. Efstathios et al., “Origami-based impact mitigation via rarefaction solitary wave creation,” Science Advances, vol. 5, no. 5, Article ID aau2835, 2019.

[19] X. Sun, J. Xu, F. Wang, and S. Zhang, “A novel isolation structure with flexible joints for impact and ultralow-frequency excitations,” International Journal of Mechanical Sciences, vol. 146, pp. 366–376, 2018.

[20] X. Liu, S. Chen, J. Wang, and J. Shen, “Analysis of the dynamic behavior and performance of a vibration isolation system with geometric nonlinear friction damping,” Chinese Journal of Theoretical and Applied Mechanics, vol. 51, no. 2, pp. 371–379, 2019.

[21] Z. Q. Lu, H. Ding, and L. Q. Chen, “Resonance response interaction without internal resonance in vibratory energy harvesting,” Mechanical Systems and Signal Processing, vol. 121, pp. 767–776, 2019.

[22] C. Liu, K. Yu, and J. Tang, “New insights into the damping characteristics of a typical quasi-zero-stiffness vibration isolator,” International Journal of Non-linear Mechanics, vol. 124, Article ID 10351, 2020.

[23] Z. Lu and L. Chen, “Some recent progresses in nonlinear passive isolations of vibrations,” Chinese Journal of Theoretical and Applied Mechanics, vol. 49, no. 3, pp. 550–564, 2017.

[24] X. Li, J. Zhang, and J. Yao, “Effect of the time-varying damping on the vibration isolation of a quasi-zero-stiffness vibration isolator,” Shock and Vibration, vol. 2020, Article ID 4373828, 10 pages, 2020.

[25] S. Yang and Y. Shen, Bifurcation and Singularity of Delayed Nonlinear Systems, Science Press, Beijing, China, 2003.

[26] Q. Pang, L. Zhang, Y. He, Z. Gong, and Z. Feng, “Verification platform for magnetorheological semi-active suspension control algorithm,” Journal of Tsinghua University (Science & Technology), vol. 59, no. 7, pp. 567–574, 2019.

[27] Z. Peng, J. Zhang, and X. Fu, “Experimental study on a semi-active magnetorheological suspension,” Automotive Engineering, vol. 40, no. 5, pp. 561–567, 2018.

[28] K. Hemanth, H. Kumar, and K. V. Gangadharan, “Vertical dynamic analysis of a quarter car suspension system with MR damper,” Journal of the Brazilian Society of Mechanical Sciences and Engineering, vol. 39, pp. 41–51, 2017.

[29] S. Chen, G. Zu, M. Yao, and X. Zhang, “Taylor series-LQG control for time delay compensation of magneto-rheological semi-active suspension,” Journal of Vibration and Shock, vol. 36, no. 8, pp. 190–196, 2017.

[30] P. Hu, J. Zhang, Z. Peng, and Y. Zhang, “Verification of magnetorheological damper mechanical model and skyhook on-off control,” Journal of Academy of Armored Force Engineering, vol. 32, no. 1, pp. pp42–49, 2018.

[31] H. D. Chae and S. B. Choi, “A new vibration isolation bed stage with magnetorheological dampers for ambulance vehicles,” Smart Materials and Structures, vol. 24, no. 1, Article ID 017001, 2015.

[32] X. Gao, J. Niu, Z. Liu, and J. Tian, “Semi-active control of ambulance stretcher system based on parallel mechanism with MR dampers and perturbation analysis,” International Journal of Mechanics and Materials in Design, vol. 15, no. 4, pp. 817–831, 2019.

[33] Q. Wang, J. Zhou, D. Xu, and H. Ouyang, “Design and experimental investigation of ultra-low frequency vibration isolation during neonatal transport,” Mechanical Systems and Signal Processing, vol. 139, Article ID 106633, 2020.

[34] C. Pan, Research on Vibration Isolation System with Hysteric Nonlinear Actuator, Beijing Jiaotong University, Beijing, China, 2009.

[35] S. Li, Y. Lu, and J. Ren, Study on Dynamics of Three-Dimensional Interaction between Heavy Vehicle and Road, Science Press, Beijing, China, 2020.