A novel scissor-type anti-shock seat employing rotary magnetorheological damper

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Abstract. In order to improve the cushioning performance of anti-shock seat in an impact environment and protect the lives of occupant to the greatest extent, the cushioning performance of a new type of anti-shock seat combining a rotary magnetorheological (MR) damper and a scissor seat was studied. Aiming at the “bottoming out” characteristic of scissor seats after the cushioning stroke is exhausted, and the dynamic model of scissor seats with MR damper is established. In order to test the cushioning performance of the new anti-shock seat in an impact environment, a prototype MR damper was processed and assembled, and the cushioning performance of the seat under different impact speeds was tested using a drop impact test platform. The test results show that the new anti-shock seat can adjust the cushion stroke and cushion force value to keep the acceleration limit of the occupant within the tolerance range, thereby effectively reducing the risk of injury.

1. Introduction

When ships encounter underwater explosions, helicopter crashes and other accidents, anti-shock seats play an important role in protecting the lives of occupants. At present, most seats use fixed load energy absorbers to limit the spine load of the occupants within the acceptable range, such as inverted tube type energy absorber and curved wire type energy absorber. These passive control systems have been widely used because of their simple structure and relatively reliable performance. However, due to their fixed damping and non-reusable, fixed stroke load will also increase the risk of injury for lighter or heavier occupants [1]. The active control system relies on external energy to provide force to the seat to achieve vibration control, eliminating the unadjustable damping defect of the passive control system. The control effect is excellent, but the cost is high, energy consumption is high, and failure safety is poor. Magnetorheological devices using magnetorheological materials as the medium have the advantages of adjustable damping, fast response time, wide operating temperature range and low energy consumption, so the semi-active control system based on MR damper has received extensive attention from researchers in the field of vibration control. Wereley et al. [2] modified the SH-60 helicopter seat and added a MR damper without affecting the work of the passive energy absorber to improve the comfort of the helicopter seat and achieved better results. Powell et al. [3] studied a MR landing gear shock absorber, which increased the adaptable range of impact speed from 3.7m/s to 7.9m/s, and improved the landing performance of light helicopters. Li et al. [4] applied the MR damper to the buffer system of the artillery, and obtained the optimal adjustable damping force through an open-loop control strategy.

For the design of shock absorbers in the field of buffering, the current research is more focused on linear MR dampers, the working principle of which is valve mode or shear valve mode [5]. As the piston speed increases during operation, the viscous damping force is greatly increased, so the controllability
of the MR damper is greatly reduced. Scholars have increased the multi-stage coil [6], set the internal bypass valve, and used the bi-fold MREA device [7] to improve the dynamic range of the damper, but will bring other disadvantages, such as increased structural volume. In this study, a rotary MR damper is introduced into the seat cushioning system. The working principle is shear mode, which has better controllability at high speeds, and achieves high torque density while maintaining a compact structure, which is beneficial to the limited space of the seat location of installation. The drop test results show that the new anti-shock seat can adjust the cushion stroke and cushion force value, and minimize the risk of occupant injury.

2. Dynamic model of a novel anti-shock seat

A new type of anti-shock seat cushioning system combines the rotary MR damper, the scissor seat and the braided belt. The seat structure adopts the X-shaped bracket scissor seat, and two MR dampers are arranged symmetrically behind the seat. The output shafts of the dampers are fixed on the movable rod of the scissor seat through a braided belt, as shown in figure 1.

The movement process of the scissor seat [8] can be shown in figure 2. When the seat is subjected to a vertical impact load, the X-shaped bracket structure will rotate relatively, driving the moving rod to move in the horizontal direction in the chute and tighten the braid to pull the output shaft of the MR damper to rotate. The impact energy is dissipated in this process.

To establish a coordinate system with point S as the origin, the kinetic energy of the scissor seat system can be expressed as:

$$T = \frac{1}{2} m_c^2 \dot{\theta}^2 = \frac{1}{2} m [\dot{q} - \dot{\phi} L \cos(\theta - \phi)]^2$$

(1)

The relative displacement of the movement of the roller of the scissor seat $\Delta x$ is:

$$\Delta x = L \cos(\theta - \phi) - L \cos \theta$$

(2)

The extended length of the braid on the output shaft of the damper $\Delta x_i$ is expressed as:

$$\Delta x_i = \sqrt{(L_d + L \cos(\theta - \phi))^2 + h_d^2} - \sqrt{(L_d + L \cos \theta)^2 + h_d^2}$$

(3)

$L_d$ represents the horizontal distance from the MR damper to the lower hinge point of the shear bar, and $h_d$ represents the vertical distance from the output shaft of the MR damper to the center of the shear bar roller. Because of $h_d \ll L_d + L \cos \theta$, we can draw that $\Delta x_i \approx \Delta x$. The damping dissipation energy of the scissor seat system [9] is expressed as:
The Lagrange equation of the scissor seat system can be expressed as:

\[
\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{\varphi}} \right) - \frac{\partial T}{\partial \varphi} + \frac{\partial D}{\partial \dot{\varphi}} = 0
\]  

From (1), obtain the following:

\[
\frac{\partial T}{\partial \varphi} = m[\ddot{\varphi} \cdot \dot{L} \cos(\theta - \varphi)] \left[-\dot{\varphi} \cdot L \sin(\theta - \varphi)\right]
\]

\[
\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{\varphi}} \right) = m[\ddot{\varphi} \cdot \dot{L} \cos(\theta - \varphi)] \left[-\dot{\varphi} \cdot L \sin(\theta - \varphi)\right]
\]

\[+ m[\ddot{\varphi} \cdot \dot{L} \cos(\theta - \varphi) - \dot{\varphi}^2 \cdot L \sin(\theta - \varphi)] \left[-L \cos(\theta - \varphi)\right] \]  

Let \( f_i = m[\ddot{\varphi} \cdot \dot{L} \cos(\theta - \varphi)] \left[-\dot{\varphi} \cdot L \sin(\theta - \varphi)\right]\) + \( m[\ddot{\varphi} \cdot \dot{L} \cos(\theta - \varphi) - \dot{\varphi}^2 \cdot L \sin(\theta - \varphi)] \left[-L \cos(\theta - \varphi)\right] \), at the same time, let \( f_0 = mL^2 \cos^2(\theta - \varphi) \). From (7), obtain the following:

\[
\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{\varphi}} \right) = f_i + f_0 \dot{\varphi}
\]

From (4), obtain the following:

\[
\frac{\partial D}{\partial \dot{\varphi}} = CL^2 \sin^2(\theta - \varphi) \dot{\varphi}
\]

For the convenience of calculation, let \( h_1 = \frac{\partial T}{\partial \varphi}, \ h_2 = \frac{\partial D}{\partial \dot{\varphi}} \). From (5) to (9), obtain the following:

\[
\dot{\varphi} = \frac{h_1 - h_2 - f_i}{f_0}
\]

Since \( h_1, h_2, f_i \) and \( f_0 \) are all related to \( \varphi \) and \( \dot{\varphi} \), formula (12) can be written as \( \dot{\varphi} = \rho(\varphi, \dot{\varphi}) \).

This article chooses a typical pulse excitation signal to simulate the impact of a scissor seat. The selected pulse excitation can be expressed as:

\[
q(t) = q_{\text{max}} \cdot 0.25e^{2\left(\gamma \omega_0 t\right)^2}e^{-2\omega_0 t}
\]

Among them \( \omega_0 = 2\pi f_0 \). \( f_0 \) is the natural frequency of the seat system; \( q_{\text{max}} \) is the excitation amplitude of the shock pulse; \( \gamma \) represents the severity level of the impact signal, and its value affects the acceleration amplitude of the impact excitation.

When the scissor seat runs out of cushioning stroke, that is \( \Delta x = s_{\text{max}}, \) the impact energy of the system is still not completely dissipated by the damper, at this time the upper seat plate will have a rigid impact with the lower plate. \( s_{\text{max}} \) is the maximum cushion stroke of the seat. In order to characterize the impact of the secondary shock, introduce impact \( F \), and its expression is:

\[
F = \begin{cases} 
\frac{m(\ddot{\varphi} - L \cos(\theta - \varphi))}{\Delta t} & L \cos(\theta - \varphi) - L \cos \theta > s_{\text{max}} \\
0 & L \cos(\theta - \varphi) - L \cos \theta \leq s_{\text{max}}
\end{cases}
\]
From (10) to (12), The vibration response of the seat and the occupant can be solved, and the acceleration of the occupant in the vertical direction can be expressed as:

\[
a_x = 2 \cos \phi \sin \phi / L - 2 \phi L \sin (\theta - \phi) + F / m
\]

The vibration responses of anti-shock seat and occupant were simulated in Matlab using the ode45 algorithm. The simulation parameters are as follows: The occupant quality \( m = 65 \) kg; the currents of the MR damper of the seat system are 0.5, 1.0, 1.5, 2.0, and 2.5A; the natural frequency of the seat system \( f_0 \) is 1.6Hz; the maximum cushion stroke of the seat \( s_{\text{max}} \) is 100mm. The simulation results are shown in the figure 3.

![Figure 3](image_url)

(a) The buffer stroke \( \Delta x \)  
(b) The occupant acceleration response \( a_x \)

It can be seen from figure 3(a) that the smaller the current, the greater the cushion stroke of the seat. Figure 3(b) shows that the smaller the current, the smaller the acceleration of the occupant on the seat for the working condition where the current is 1.0-2.5A. When the current is set to 0.5A, the seat cushioning stroke reaches the maximum stroke near 0.52s, so the seat upper plate and the lower plate collide rigidly at this time. The acceleration of the occupant is caused by the rigid impact, and its value exceeds the working condition of current 2.5A. Therefore, the simulation results show that, in order to ensure the life safety of the occupant to the greatest extent, the current should be adjusted to make the seat use up the cushion stroke as much as possible. Within the maximum cushioning stroke of the seat, a smaller current is more beneficial to ensure that the acceleration of the occupant does not exceed the human tolerance limit.

3. Drop test of the anti-shock seat

In order to verify the cushioning performance of the anti-shock seat with MR dampers, a prototype MR damper was manufactured, and conducted a system test.

3.1. Principle and experimental arrangement of drop test system

The schematic diagram of the anti-shock seat cushioning performance test system is shown in figure 4. A triangle wave shock is applied to the seat, which changes the current of the MR damper to change the controllable damping force. Acquire the acceleration signal of the seat, the acceleration signal of the dummy after being cushioned by the seat, the output force of MR damper and the relative displacement signal base through dSPACE hardware to verify the cushioning performance of the anti-shock seat.
According to the principle of the above-mentioned system, the experimental platform of the anti-shock seat cushioning system is shown in figure 5. It mainly includes the frame of the drop test platform, electromagnet release mechanism, dummy, seat frame, dSPACE hardware, DC power, acceleration sensors, X-frame scissor seat with MR dampers, displacement sensor, force sensor and computer. The seat is fixedly connected to the inner frame. After the inner frame is raised to the specified height by the electromagnet, the current is disconnected, and the inner frame will make a free fall motion. The seat is lifted to a specified height through the inner frame and electromagnets and then released to study the cushioning performance of the anti-shock seat under different impact speeds.

The test conditions and parameters are as follows: the weight of the dummy is 65Kg. The drop height is 300, 400, 500 and 600mm, the corresponding speed is 2.42, 2.80, 3.13 and 3.43m/s respectively. The excitation current applied to the MR damper is 0.5, 1.0, 1.5, 2.0 and 2.5A.

3.2. Test results

Figure 6 is the time history curve of the output damping force of the MR damper when the drop height is 0.3m. It can be seen from the figure 6 that as the current increases, the output damping force of the MR damper continues to increase. When the current is 0.5A, the maximum damping force is only 1580N. If the current is increased to 2.5A, the maximum damping force is 3800N, an increase of 141%. At the same time, it can be seen from figure 6 that as the current increases, the buffer time becomes shorter and the damping force changes faster. On the contrary, the smaller the current, the more stable the damping force changes with time, which is more conducive to the dissipation of impact energy.
Figure 7 is the acceleration signal of the drop test with a drop height of 0.6m, including the acceleration of the seat frame and the occupant acceleration change curve. The acceleration of the seat frame reflects the acceleration generated by the seat frame without buffering. The acceleration of the occupant refers to the acceleration of the seat at the position of the dummy after part of the impact energy is dissipated by the MR damper, which also reflects the cushioning performance of the anti-shock seat. It can be seen from the figure 7 that the acceleration of the seat frame is much greater than the acceleration of the occupant, which also proves that the buffer solution of MR damper and braided belt is effective.

Figure 7 shows that when the current is 0.5A, there are two spikes in the acceleration curve of the occupant with a time interval of 80ms, and the amplitude of the secondary peak exceeds 2.5A. Due to the small damping force at this time, the MR damper cannot completely dissipate the energy of the system after the cushion stroke of the seat is exhausted, which causes the seat to "bottom out".

When the current is greater than or equal to 1.0A, the acceleration curve has only one peak, so there is no rigid impact between the upper and lower plates of the seat. As the current continues to increase, the occupant's acceleration also shows an increasing trend, where the occupant's acceleration amplitude is the largest when the current is 2.5A. The analysis shows that for the same drop height, the impact energy is the same. Therefore, the smaller the current applied to the MR damper, the smaller the output damping force, and the smoother the damping force change, so it is more conducive to energy dissipation. To sum up, in the case of high impact speed, the current is not as small as possible. In order to minimize the acceleration amplitude of the occupant and ensure life safety, appropriate current should be selected to ensure that the seat stroke is used up as much as possible without "bottoming".

4. Conclusions
In an impact environment, anti-shock seats play an extremely important role in protecting the lives of occupants. This article proposes a new anti-shock seat, introducing the rotary MR damper and the braided belt. The dynamic model is established, and the numerical simulation results show that the current should be adjusted to make the seat as full as possible of the cushion stroke.

The drop test results show that the new anti-shock seat can adjust the cushion stroke and cushion force value, and can significantly reduce the acceleration amplitude at the occupant. In order to ensure the life safety of the occupant to the greatest extent, the cushion stroke of the anti-shock seat should be used as much as possible without "bottoming", so as to minimize the acceleration level of the occupant.

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