SHOCK VIBRATION CONTROL USING “SMART” DAMPING DEVICES

Guangqiang Yang*, Bill F. Spencer Jr.*, and Frank Leban**

*Department of Civil and Environmental Engineering, University of Illinois at Urbana-Champaign,
205 North Matthews Ave., Urbana, IL 61801, U.S.A.

**Naval Surface Warfare Center, Carderock Division, Mobile Support Systems Programs,
Code 2930, 9500 Macarthur Blvd., West Bethesda, MD 20817-5700

ABSTRACT

Both passive and active shock isolation systems have been previously investigated. However, passive systems are limited in that they cannot adapt to different load conditions; active systems are not practical in many applications, as they may require large power supplies. Another solution to the problem is to implement a “smart” damping system, which typically requires little power, is controllable in real-time, and achieves nearly the same performance as an active system. This paper focuses on shock isolation using these “smart” damping devices. Following the basic problem formulation, the optimal design of passive and active shock isolators is given. These results are then extended to the “smart” (semi-active) shock isolation case. Theoretical analysis shows that if the linear spring stiffness is properly chosen and the control force is appropriately regulated, the shock isolator using “smart” damping can achieve optimal performance and has a better robustness than the passive shock isolator.

1. INTRODUCTION

The concept of shock isolation is frequently encountered. For example, driving vehicle on a bumpy road causes shock vibration which may severely impair vehicle handling performance; paradropping deliveries in a military expeditionary logistics environment introduces unfavorable impact disturbances to the sensitive equipment packed inside at the time of landing. The impact loads are frequently quite large when compared with the low- or moderate-level disturbances encountered during normal operating conditions; consequently, the equipment may be severely damaged or even destroyed. To mitigate these undesirable effects, shock isolators which are devices that react to impact disturbances can be utilized to regulate force transmission to the protected equipment [1].

There exist a wide variety of shock isolation systems, ranging from relatively simple passive shock isolators (e.g., viscous shock absorbers) to complicated active shock isolation systems with actuators, sensors, and digital controllers. Research on passive shock isolation systems has a long history, having been thoroughly investigated. Until now, passive isolation systems have been widely used in civil and military applications. A thorough review of vibration isolation with various damping components can be found in the book written by Rusiecka and Derby [2].

Although many shock vibration problems can be solved in an inexpensive, reliable, and satisfactory way with passive devices, active systems are limited in that these systems cannot adapt to the change of either external loading or usage pattern. Alternatively, active vibration isolation systems were developed in 1960’s. However, active isolation systems are more complex and expensive, and are therefore less reliable than passive systems, and may require relatively large external power supplies [3, 4]. These factors limit their usage in many applications.

Shock isolation systems using “smart” damping devices may be used to address these aforementioned issues. “Smart” (semi-active) damping devices share many similarities with conventional passive isolators, but are controllable—that is, the characteristics of the damper may be changed in real-time to alter the forces generated by the damper. This feature allows the damper to respond differently to varying load characteristics and changing control objectives [5]. Karnopp et al. [3] and Alanoly and Sankar [6] proposed an “on-off” control algorithm for variable orifice dampers. Alanoly and Sankar [6] also utilized a similar algorithm for variable stiffness devices. Bobrow et al. [7] utilized a resetable actuator for shock isolation appli-
cation. Note that both variable orifice and stiffness dampers are still very similar to conventional passive devices, in which the force generated by the damper is also highly dependent on the isolator velocity and deformation although these devices feature variable damping and stiffness properties. Therefore, the use of these devices may not achieve optimal isolation performance.

One particularly promising "smart" damping device for shock isolation is found in the magnetorheological (MR) fluid damper [4, 8]. The force generated by the MR damper is mainly related to the input current to the electromagnet. A force-feedback approach recently proposed by Yang et al. [8] can be used to compensate the damper force under various velocities by regulating the input current through a feedback loop. Therefore, the MR damper can generate an arbitrary force within its force envelope controlled only by an input current. Because of these unique features, as well as their low power requirements, mechanical simplicity, and large controllable force and dynamic range, MR dampers mesh well with the demands and constraints of shock isolation applications, and they appear to offer the greatest likelihood to achieve optimal isolation performance.

In this paper, the shock isolation problem without rebound is considered. Following the basic problem formulation, the optimal design of passive and active shock isolators is given. The results are then extended to "smart" (semi-active) shock isolation systems. Theoretical analysis shows that if the linear spring stiffness is properly chosen and the control force is appropriately regulated, the shock isolator using a "smart" damping device, such as the MR damper, can achieve optimal performance and has a better robustness as compared with that of the linear shock isolator.

2. PROBLEM FORMULATION

To study the shock isolation problem, consider the rigid body $M$, shown in Fig. 1, which is to be protected by the shock isolation system attached at the base. This base, for example, might be a cargo container, and the protected body $M$ may be sensitive equipment loaded in the container. For this preliminary study, the base is assumed to be rigid with no deformation during the impact, and the rigid body $M$ displaces without rebounding.

The equation of motion after impact for this problem is

$$M\ddot{z} = f = f_0(z, z, t) + f_I(t)$$

(1)

with initial conditions $z(0) = z_0$ and $\dot{z}(0) = v_0$, where $v_0$ = impact velocity and $f$ = isolator force acting on mass $M$, which includes the state-dependent force $f_0(z, z, t)$ and the state-independent force $f_I(t)$. Without loss of generality, one can assume that the initial displacement $z_0 < 0$ and initial velocity $v_0 > 0$ for this specific type of problem.

To evaluate the performance of the shock isolator considered herein under impact excitation, the peak values of isolator deformation and acceleration which characterize the transient process are usually the most important measures [9]. The performance indices normally considered have the form

$$J_1 = \max_t |z(t) - z_0| \quad \text{and} \quad J_2 = \max_t |\dot{z}(t)|$$

(2)

where $J_1$ = peak isolator deformation (assuming that the isolator deformation is zero at time zero); and $J_2$ = peak acceleration. Referring to Eq. (1), one can see that because the acceleration is proportional to the isolator force $u = f/M$, the performance index $J_2$ is equivalent to

$$J_2 = \max_t |u(t)|$$

(3)

Note that these indices represent competing demands. By minimizing one performance index and constraining the other, two types of optimization problems can be defined as follows:

**Problem 1: Minimization of Peak Isolator Deformation**

$$\min_u \{ \max_t |z(t) - z_0| \} \quad \text{while} \quad \max_t |\dot{z}| \leq U_0$$

(4)

**Problem 2: Minimization of Peak Acceleration**

$$\min_u \{ \max_t |\dot{z}| \} \quad \text{while} \quad \max_t |z(t) - z_0| \leq D_0$$

(5)

Problems 1 and 2 are closely related dual problems.

3. LIMITING PERFORMANCE ANALYSIS

Limiting performance analysis is used to determine the
4. LINEAR SHOCK ISOLATION SYSTEM

A linear isolator consisting of a linear viscous damper and a linear spring connected in parallel generates an isolation force given by

\[ f = -kz - cz \]  

(6)

and the equation of motion after impact is

\[ z = u - 2\xi\omega_n z - \omega_n^2 z \]  

(7)

and \( \omega_n = \sqrt{k/M} \), \( \xi = c/(2\sqrt{MK}) \). The initial conditions are \( z(t_0) = -g/\omega_n^2 \) and \( z(t_0) = v_0 \), where \( g = \) constant of gravity. The maximum deformation \( D \) and acceleration \( U \) are given by

\[ D = \frac{1}{\omega_n^2} \left( \frac{1}{\sqrt{1 - \xi^2}} \sqrt{g^2 + \omega_n^2 v_0^2 - 2\xi g \omega_n v_0 + g} \right) \]  

(8)

where

\[ \beta = \arctan \left( \frac{\sqrt{1 - \xi^2} \omega_n v_0}{-\xi \omega_n v_0 + g} \right) - n\pi \]  

(9)

and \( n = \min(0, 1) \) such that \( \beta < 0 \).

\[ U = \max \left( \sqrt{\frac{g^2}{\omega_n^4 v_0^2 + g^2 - 2\xi^2 g \omega_n v_0 + g^2}}, \sqrt{\frac{g^2}{\omega_n^2 v_0^2 + g^2 + \xi^2 g \omega_n v_0 - g^2}} \right) \]  

(10)

where

\[ \gamma = \arctan \left( \frac{\sqrt{1 - \xi^2} (-4\xi^2 \omega_n v_0 + 2\xi g + \omega_n v_0)}{-4\xi^2 \omega_n v_0 + 2\xi^2 g + 3\xi^2 \omega_n v_0 - g^2} \right) + \pi \]  

(11)

and \( n = \min(0, 1) \) such that \( \gamma > 0 \). As shown in Eqs. (8)–(11), the maximum deformation and acceleration for the linear isolator depend largely on the initial velocity and isolator parameters. If the isolator parameters are optimized for a specific design velocity, it may not perform well at other impact velocities.

5. ACTIVE SHOCK ISOLATION SYSTEM

When performance obtained from the passive isolator is inadequate, the use of an active isolation system can be considered. Eq. (1) can be rewritten as

\[ z = u \]  

(12)

where \( u = f/M \). The initial conditions are \( z(t_0) = z_0 \) and \( z(t_0) = v_0 \).

Because the maximum displacement occurs at zero velocity, performance index \( J_1 \) can also be written as [10]

\[ J_1 = z^2(t_f) \]  

(13)

with an inner point constraint \( z(t_p) = 0 \), where \( t_p = \) unspecified terminal time; and \( 0 < t_p < t_f \). This problem can be solved using optimal control theory.

If \( z(t_p) > 0 \), a possible control law for Problem 1 is the minimum time control law [10, 11]

\[ u^* = -U_0 \operatorname{sgn}(z + \frac{1}{2U_0} |z|) \]  

(14)

with maximum deformation \( v_0/(2U_0) \) and peak acceleration \( U_0 \). In this case, there obviously exists one force switching to bring the states back to the origin. If
\[ z(t_i) \leq 0 \], the isolator force is given by
\[ u^* = \frac{v_0^2}{2|z|} \text{sgn}(z) \]  
(15)
with maximum deformation \( |z_0| \). The peak acceleration is \( v_0^2/(2|z_0|) \). There is no force switching.

Fig. 3 shows the trajectory of the transient response using an active shock isolator for Problem 1. If there exists force switching or \( z(t_i) > 0 \), the trajectory can basically be divided into three regions. Region I is the region before the maximum displacement is achieved; region II is the region after the maximum displacement has been achieved and before the force switching; and region III is the region after the force switching. In regions I and II, the isolator force is \(-U_0\), and the isolator force is \( U_0 \) in region III. In regions I and III, the isolator dissipates the energy. However, note that in region II, this control adds energy to the system because isolator force and velocity are in the opposite direction.

### 6. "SMART" SHOCK ISOLATION SYSTEM

As proposed herein, a "smart" shock isolation system consists of a linear spring and a "smart" damper connected in parallel. To achieve limiting performance, the isolation system needs to add energy into the system in region II, as shown in Fig. 3. However, a smart damper can only dissipate energy. Therefore, in this region, the damper resisting force is set to zero, and energy stored in the spring is released back to the system [11].

The equation of motion for a shock isolator with a "smart" damper after impact is
\[ \ddot{z} = u = u_0 - \omega_n^2 z \]  
(16)
If the origin is set at the equilibrium point, the initial conditions are \( z(0) = -g/\omega_n^2 < 0 \) and \( \dot{z}(0) = v_0 > 0 \).

For Problem 1, the optimal isolator force \( u = -U_0 \) in region I, and the damper force is
\[ u_d = -U_0 + \omega_n^2 z \]  
(17)
The maximum displacement occurs at time \( t_i \), given by
\[ z(t_i) = \frac{v_0^2}{2U_0} \frac{g}{\omega_n^2} \]  
(18)
If \( z(t_i) > 0 \), the damper force \( u_d \) is set to zero in region II, and the isolator force is given by \( u = -\omega_n^2 z \). The maximum acceleration in this region occurs at time \( t_i \). The acceleration constraint must not be violated, yielding
\[ \frac{v_0^2}{2U_0} \frac{g}{\omega_n^2} < \frac{U_0}{\omega_n^2} \]  
(19)
Eq. (19) also implies that the damper force \( u_d < 0 \) in region I. Because \( z(t) > 0 \), the damper can always dissipate energy in this region. Moreover, the isolator force does not change the direction at time \( t_i \), which means \( z(t_i) \) is the maximum displacement. From Eq. (19), the spring stiffness needs to satisfy
\[ \omega_n^2 < \frac{2U_0(U_0 + g)}{v_0^2} \]  
(20)
In region III, the optimal isolator force is \( U_0 \), and the damper force is
\[ u_d = U_0 + \omega_n^2 z \]  
(21)
Fig. 4 shows the relationship of the damper and isolator forces versus displacement, respectively. The damper force is bounded by \( [-U_0, 2U_0] \). Fig. 3 shows the trajectory comparison for Problem 1 between the active shock isolator and shock isolator with the "smart" damper.

In summary, the control laws for shock isolator with "smart" damping device for Problems 1 and 2 are:

**Problem 1**
If \( v_0^2/(2U_0) \leq g/\omega_n^2 \), then the damper force is given by
\[ u_d = -\frac{\omega_n^2 v_0^2}{2g} \text{sgn}(z) + \omega_n^2 z \]  
(22)
with maximum displacement \( g/\omega_n^2 \) and peak acceleration \( \omega_n^2 v_0^2/(2g) \), which is less than \( U_0 \). The linear spring stiffness needs to satisfy
The damper force is given by

\[
\begin{cases} 
  - \frac{v_0^2}{2D_0} + \omega_n^2 z & \text{if } z(t) \geq 0 \\
  0 & \text{if } -\frac{D_0}{v_0^2} \lvert z \rvert > 0, z(t) < 0 \\
  \frac{v_0^2}{2D_0} + \omega_n^2 z & \text{if } -\frac{D_0}{v_0^2} \lvert z \rvert \leq 0, z(t) < 0
\end{cases}
\]  

with maximum displacement \( \frac{v_0^2}{2U_0} \) and peak acceleration \( v_0^2 \).

7. NUMERICAL EXAMPLE

To illustrate the results described above for design of a "smart" shock isolator, consider the following example problem:

Equipment with nominal weight of 20 kg is to be protected from a shock with a maximum impact velocity of 2 m/s. The maximum allowable acceleration is 20g and the maximum deformation of the shock isolator is 0.02 m. The design impact velocity is chosen to be 2 m/s. The equipment is sufficiently stiff and can be considered rigid.

7.1 Problem 1: Minimization of Peak Isolator Deformation

For the linear isolator, the optimal spring stiffness is \( 2 \times 10^3 \) N/m, and the damping coefficient is 2060 Ns/m. The isolator has a peak displacement of 0.0102 m, peak deformation of 0.0112 m, and peak acceleration of 20g. For the active shock isolator, the optimal force is 3924 N. The peak displacement is 0.0092 m, peak deformation is 0.0102 m, and peak acceleration is 20g. For the shock isolator with the "smart" damper, the spring stiffness is chosen to be \( 2 \times 10^3 \) N/m. The optimal force is 3924 N. The results for peak values are the same as obtained for the active shock isolator. Fig. 5 shows the trajectory comparison of three types of designs for Problem 1.

7.2 Problem 2: Minimization of Peak Acceleration

For the linear isolator, the optimal stiffness is \( 7.94 \times 10^5 \) N/m, and the damping coefficient is 1065 Ns/m. The isolator has a peak displacement of 0.0175 m, peak deformation of 0.02 m, and peak acceleration of 10.66g. For the active shock isolator, the optimal force is 2000 N. The
peak displacement is 0.0175 m, peak deformation is 0.02 m, and peak acceleration is 10.19g. For the shock isolator with a "smart" damper, the spring stiffness is chosen to be \(7.94 \times 10^4\) N/m. The optimal force is 2000 N. The results for peak values are the same as obtained for the active shock isolator. Fig. 6 shows the trajectory comparison of the three types of designs for Problem 2.

8. CONCLUSIONS

Limiting performance is defined as the best performance a shock isolator can achieve. Any design requirements beyond limiting performance are not physically realizable. The criteria for choosing the stiffness of the linear spring and associated control law to regulate the "smart" damper force are derived in this paper. In comparison with linear isolators, whose performance highly depend on the impact velocity and payload mass, shock isolators using "smart" dampers can achieve optimal performance if the stiffness of the linear spring is properly chosen and the damper resisting force is appropriately regulated. Experimental verification of this algorithm using MR dampers is currently underway.

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