Variable stiffness and damping MR isolator

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Abstract. This paper presents the development of a magnetorheological (MR) fluid-based variable stiffness and damping isolator for vibration suppressions. The MR fluid isolator used a sole MR control unit to achieve the variable stiffness and damping in stepless and relative large scope. A mathematical model of the isolator was derived, and a prototype of the MR fluid isolator was fabricated and its dynamic behavior was measured in vibration under various applied magnetic fields. The parameters of the model under various magnetic fields were identified and the dynamic performances of isolator were evaluated.

1. Introduction
Vibration isolation systems are used to separate the vibration excitation source and the object to be protected in order to reduce the negative effect from the unwanted vibration from the vibration excitation source. Conventional vibration systems are in passive mode, for the intrinsic limitation of passive suspension system, the improvement is effective only in a certain frequency range, because passive design for low frequency is difficult and often represents a compromise between isolation performance and the supported machinery alignment [1]. It is possible to improve the isolation performance over a wide range of frequency by active vibration control. However, active vibration isolation systems are less commonly used than passive systems due to their associated cost, power requirements, complexity and fail-safe problems [2, 3]. Semi-active suspensions can be nearly as effective as fully active suspensions in vibration suppression. They do not require either higher-power actuators or a large power supply. When the control system fails, the semi-active suspension can still work in a passive condition. In early semi-active suspension systems, the adjustable damping force can be achieved by using hydraulic semi-active dampers with electromagnetically controlled valves or Friction damper, of which the damping force is controlled by varying the force normal to a friction interface. More recently, with the introduction of Magnetorheological (MR) fluid, it is easy to achieve the variable damping with an MR damper in isolation system. The effort to improve the performance of the isolation systems by using a variable damping damper based on MR fluid has been done by many researchers [4-5]. However, variable damping and stiffness systems showed significant improvements in vibration reduction application, such as the seismic protection system [6] and the vehicle suspension [7]. However, all the methods to achieve variable stiffness in these devices are usually complicated and low reliability, and it is difficult to accomplish stepless control on the stiffness [8]. Kobori et al. developed a variable stiffness system for protecting the building from the earthquakes [9]. A valve in a cylinder device was controlled in open/close ways to lock and unlock the connection between the beam and the brace and two alternative stiffnesses were produced at each floor. Youn et al. developed a vehicle suspension using
air springs, which can be adjusted in three different states: soft, medium and firm, to achieve variable stiffness [10]. Liu et al. proposed a variable damping and stiffness system using two MR dampers and two springs [11]. By controlling the damping of the MR damper, the total stiffness of this system can be varied. However, all the methods to achieve variable stiffness in these devices are usually complicated and low reliability.

In this paper, we proposed a new vibration isolator, which uses a MR valve to connect with two stiffness units. The variable stiffness of the system is accomplished by controlling the damping of the MR valve. The prototype of the vibration isolator was built. The change on the damping and stiffness of system was demonstrated in theoretical and experimental methods.

2. MR fluid isolator

2.1. The Design of the MR Fluid Isolator

The schematic of the MRF isolator is shown in figure 1. The MR isolator consists of an air spring, two connectors, a MR valve and an accumulator. Two connectors link air spring, accumulator and MR valve together with plastic pipes. In each connector, there is a loose separate film, which can move freely, dividing the connector into two chambers. The chambers close to the MR valve are filled with MR fluid. The air spring supports the isolated object. Due to the effect of vibration excitation on the isolated object, the air spring is compressed or tensed by the isolated object in order to generate the deformation of the air spring, which is the cause of the pressure variation in the air spring. Since the change of the pressure in the air spring, the MR fluid is forced to flow from connector 1 to connector 2 through MR valve. When an input current is applied on the coil of the MR valve, a magnetic field is generated in the electromagnet. The shear stress of the MR fluid in MR valve is varied by controlling the magnetic field, which thereby varies the characteristics of the isolator, such as its stiffness and damping.

Figure 1 Schematic of the MRF isolator and Experiment Setup

2.2. Mathematical Model

Figure 2a shows the structure of the vibration isolation systems modeled by the linear springs $k_1$ and $k_2$, the damping factor of the system $c_0$, $c_1$, $k_1$ and $k_2$ are the stiffness of the accumulator and the air spring, respectively. The MR damping factor $c_1$ that can be changed with the magnetic field applied on the MR valve. $m_2$ is the sprung mass and $m_1$ is the mass of moving MR fluid. The displacement equation are presented by

$$m_2\ddot{x}_2 = k_2(x_2 - x_i) + c_0(x_2 - \dot{x}_0)$$
$$m_1\ddot{x}_1 = k_1(x_1 - x_i) - k_1(x_1 - x_0) - c_1(x_1 - \dot{x}_0)$$

where $x_0$ is the input, $x_i$, and $x_1$ are displacements of $m_1$ and $m_2$. In this design, $m_1$ is much less than $m_2$, so it can be assumed as 0. $c_0$ is the structure damping of the system.

Figure 2b shows the simplified physical model of MR Fluid isolator. If the damping of the damper is very little, two springs $k_1$, $k_2$ can be considered as being in series. The total stiffness of the system
is \( k_{\text{total}} = \frac{k_1 \cdot k_2}{k_1 + k_2} \). When the applied current on MR valve is large enough so that the damping force generated by MR valve stop the relative motion between the first layer of the system and foundation, namely \( x_0 = x_1 \), the mass \( m_2 \) is only isolated by \( k_2 \). The total stiffness of the isolation system is \( k_2 \). Therefore, when the input current on the MR valve is adjusted in the range between zero and maximum value, not only the damping of the system is controlled, but also the total stiffness of the system can be changed within the region between \( \frac{k_1 \cdot k_2}{k_1 + k_2} \) and \( k_2 \) as well.

![Figure 2 (a) schematic of the MR isolator model; (b) model simplification.](image)

3. Experimental procedures

3.1. Experimental Setup

The schematic of the experimental setup is shown in figure 1. The isolator is connected with a vibration exciter (Type: JZK-5, SINOCERA PIEZOTRONICS, INC. China) which provides the vibration source for experiments. The vibration signal is generated by LabVIEW software in computer and is transmitted to a power amplifier (YE5871-100W) to amplify through the Data Acquisition (DAQ) board (Type: LabVIEW PCI-6221, National Instruments Corporation. U.S.A). After amplifying, the vibration signal is transmitted to vibration exciter. Force sensor (Type: CL-YD-302, SINOCERA PIEZOTRONICS, INC. China) monitors the force \( F \) generated by exciter. The displacement sensor (ILD1700-10) measures the displacement \( x_2 \) of the compressed surface of the air spring. The signals from force sensor and displacement sensor are amplified by Charge amplifier (YE5851) and processed and transferred by DAQ to the computer. A GW laboratory DC power supply (Type: GPR-3030D, TECPEL CO., LTD. Taiwan) can adjust the input current on the MR valve in order to control the magnetic field intensity of the isolator and change its dynamic performance.

3.2. Testing

The maximum force generated by vibration exciter is 50 N. The initial stiffness of system in absence of any current on the MR valve is 5500N/m, when air spring is in open status. Due to the high initial stiffness of the system, it is difficult to carry out the test with large displacement. The maximum displacement in our tests is 1 mm. For each test, the vibration exciter is programmed to move forward and backward in a sinusoidal wave at certain displacement amplitude and frequency. The resulting force can be measured with force sensor. The MR isolator was initially tested without any applied current. Current was then applied to the coils and the MR valve tested again. The currents varied from 0 to 0.5A, and the experimental data including time, displacement, force, were recorded. All the experiments were carried out at the room temperature of 23°.

4. Experimental and theoretical analysis result

4.1. Experimental Results

The relationships of the MR isolator response damping force versus displacement at various magnetic fields are shown in figures 3a. It can also be seen that the response force of isolator increases with the
increasing magnetic field. The maximum force increases from 2.82 N (input current 0A) to 12.37 N (input current 0.4A). The increase on the response force is more the 4 times. Under the uniform displacement input, the rise of the response force indicates that the total stiffness of the isolation system increases, and the increasing extend of the total system stiffness is more 4 times as well. The hysteresis on the response force is usually caused by the increase of damping. Therefore, the experimental result indicates that the damping of system is increasing as the rise of the magnetic field. Furthermore, the response force varies very little when the input current is greater than 0.4A due to the saturating magnetization of MR fluid. Therefore, our experiment was stop when input current was increased to 0.5A. Based on experimental results, the equivalent stiffness and damping capacities were calculate and shown in figure 3b. These results demonstrate that the developed MR isolator have the capabilities that both its stiffness and damping capacities can be varied by an external field. For the developed MR valve, the flux density is proportional to the coil current, which is given by

$$B_g = 0.4I \quad (I \leq 0.5A).$$

Figure 3. The relationship of (a) the damping force versus displacement (b) Stiffness and Dispersed Energy at various field strengths (coil currents): $X = 0.5$ mm, $f = 0.5$ Hz, $I = 0, 0.1, 0.2, 0.3$ and 0.5A.

Neglecting the mass of moving MR fluid $m_1$, as shown in equation (1), the parameters of the system were identified and listed in table 1.

| $k_1$ (kN/m) | 5.5 | $c_1$ I=0.1A (kNs/m) | 1.9 |
| $k_2$ (kN/m) | 25 | $c_1$ I=0.2A (kNs/m) | 4.9 |
| $c_0$ (kNs/m) | 1 | $c_1$ I=0.3A (kNs/m) | 8.9 |
| $c_1$ I=0.0A (kNs/m) | 2E-6 | $c_1$ I=0.4A (kNs/m) | 14 |

With these parameters, a MATLAB Simulink model was built for reconstruction of the data. Fig. 4 shows the reconstructed force versus displacement at various magnetic fields. Obviously, the simulation can present the experimental results well.

Figure 4. Reconstructed force versus displacement using the identified parameters.
4.2. Discussion

Conventional air spring often needs to control complex and nonlinear air pressure. The control strategy is difficult to be effectively implemented. The developed MRF isolator is easier to be controlled and cheaper, because it does not need air compressor and maintenance system. However, it is relatively expensive than a passive isolator. So it would be more effective to be used for some special applications, which need high suspension effect and low level of maintenance. The applications of MRF damper in buildings have been intensively investigated. Comparing with MRF damper systems, our developed system is more cost-effective because the controlled valve can be separated as a modular and connects with air spring and accumulator via pipes.

5. Conclusion

In this paper, a new semi-active vibration isolation system, which consists of an air spring, a MR valve and an accumulator, is designed and manufactured. A mathematical model of the MR isolator is built for latter theoretical analysis. To describe the practical damping force generated by the MR valve in the isolator, a numerical model for the MR valve is proposed. Then, using experimental method, the parameters of MR isolator are identified. Meanwhile, the dynamic behavior of MR isolator with different current input is measured and compared with that resulting from theoretical analysis. It was shown that the model calculation result was in sufficiently good agreement with the dynamic properties of the isolator measured from the experiment. From the experiment results, it is demonstrated that the stiffness of the MR isolator can be adjusted in a relative large scope, more then 4 times change in the stiffness, using only one MR damping controllable unit.

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