Dynamic Characteristics of Magneto-Rheological Fluid Damper

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ABSTRACT

Two kinds of Magneto-rheological fluid damper (MRF damper) have been designed and manufactured. One has a nominal capacity of 2kN and the other 20kN. A bypass flow system is adopted for both dampers and each has the same capacity of electromagnet attached to the bypass portion. The effective fluid orifice is the rectangular space and the magnetic field is applied from the outside.

A test was performed by applying different magnetic fields to the orifice portion of the rectangular space. The damping force and the force-displacement loop were evaluated.

The test results yielded the following: 1) Two type’s of dampers functioned by using one unit of the electromagnet under an appropriate electrical current control. 2) The magnitude of the damping force depends on the input magnetic field, but it has an upper limit. 3) Without an applied magnetic field, the MRF damper exhibits viscous-like behavior, while with a magnetic field it shows friction-like behavior.

A mechanical model of the damper is estimated by taking account of the force-displacement loop.

It is clarified that MRF dampers provide a technology that enables effective semi-active control in real building structures.

Keywords: Magneto-rheological fluids, Magneto-rheological fluid dampers, Damping characteristics

1. INTRODUCTION

Vibration control systems are mainly divided into three types: passive, active, and semi-active. The semi-active type attracts a great deal of attention because of it’s the following superior properties: 1) It is basically stable in contrast to the active type. 2) Its control effect is more flexible than the passive type. 3) Its external energy requirements are orders of magnitude smaller than those of typical active types.

Until now, various kinds of semi-active control devices have been developed. One fundamental device for semi-active control systems is the capacity adjustable viscous damper. Its viscosity is adjustable in accordance with the opening rate of the flow control valve inside the damper. A series of studies with this type of damper has been carried out and they were found to have good characteristics and efficient control system properties \textsuperscript{[1,2]}. However, magneto-rheological (MR) and electro-rheological (ER) fluid dampers attracted significant attention. The damping properties of these types of dampers are changed by changing the magnetic/electrical field applied to each fluid \textsuperscript{[3-10]}

Research on semi-active control of building structures using controllable fluid dampers is being conducted in the U.S.-Japan cooperative research and development project of “Smart Materials and Structural Systems.” launched in 1998 by NSF and Building Research Institute, Japan Ministry of Construction \textsuperscript{[11]}. This paper outlines the results of experimental tests and simulations of magneto-rheological fluid dampers conducted as one of the research items of this cooperative research on the Japan side.

Two kinds of Magneto-rheological fluid damper (MRF damper) have been designed and manufactured. One has a nominal capacity of 2kN and the other 20kN. A test was performed by applying different magnetic fields to the orifice portion of the rectangular space. The damping force and the force-displacement loop were evaluated.

Furthermore, an analytical model is proposed to simulate the behavior of the MR dampers, and simulation and experimental results are compared.

2. DESIGN OF MRF DAMPER

The bypass flow type MRF damper has been developed in this study. Fig.1 shows its hydraulic circuit. The bypass flow portion is a passage for MR fluids connecting two pressure chambers. The bypass flow is at the outside of the cylinder. In
bypass type hydraulic dampers such as this, a cylinder is divided into two airtight pressure chambers by a piston with rubber O-rings. The MR fluid flow from a high-pressure chamber to a low-pressure chamber through the bypass flow portion. The bypass flow portion has an orifice and the MR fluid flow is narrowed rapidly at the orifice. Moreover, intense magnetic fields are applied to the MR fluid at the orifice by electromagnets. To simplify the analysis of the MR fluid flow state in the magnetic field, a rectangular cross section is selected as the orifice shape. The magnetic fields are applied perpendicularly to the MR fluid flow at the orifice. The electromagnet is formed by copper wire wound around a C-shaped iron rod. The air gap between the ends of the C-shaped iron rod forms the rectangular cross section of the orifice. Thus, the length of the air gap of the electromagnet is equivalent to the thickness of the orifice that is penetrated by the magnetic flux. This method of magnetizing MR fluids has the advantages that the MR fluid is rarely affected by the heat generated on the electromagnet.

Two different capacities of MRF dampers have been designed and built. Fig.2 and Fig.3 show photographs of them.

Figure 1  Hydraulic Circuit of Bypass Type MRF Damper

Figure 2  MRF Damper - 2kN

Figure 3  MRF Damper - 20kN
Fig. 2 shows the one whose the nominal maximum damping force is 2kN, and Fig. 3 shows the one whose the nominal maximum damping force is 20kN. Table 1 shows their design specifications. These two dampers have the same basic mechanism. In particular, their electromagnets are identical. The electromagnet’s design is based on the larger dampers. It is therefore used under the small input electric currents for the 2kN MRF damper. The basic specifications of the electromagnet are also shown in Table 1-1. Very low carbon steel is used for the C-shaped iron rod to obtain a high level of magnetic saturation and low residual magnetization. The 2kN MRF damper’s orifice is 0.6 x 16mm in cross section and 10mm long. The 20kN damper’s orifice is 2 x 20mm in cross section and 20mm long. When designing the orifice size, the Reynolds number of the fluid flow at the orifice was considered to be less than the critical Reynolds number.

As shown in Fig. 1, these MRF dampers have two check valves on the passages to the reservoir. These check valves are open under the non-vibration condition. Therefore, the thermal expansion due to the temperature rise of the MR fluid is absorbed by the reservoir. However, the check valves are shut during vibrations, and the flow of MR fluid occurs only through the bypass portion.

Two different kinds of MR fluids were tested on the 2kN MRF damper, in order to find the most suitable commercial MR fluid for the bypass type MRF damper. One was LORD MRF-132LD and the other was LORD MRF-128NB. Based on the experimental results of these 2kN dampers, MRF-132LD was selected as the fluid for the 20kN MRF damper.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Design Specifications of MRF Dampers</th>
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<tbody>
<tr>
<td>Max. Force (nominal)</td>
<td>MRF damper - 2kN</td>
</tr>
<tr>
<td>Stroke</td>
<td>□ 10 mm</td>
</tr>
<tr>
<td>Cylinder bore</td>
<td>35 mm</td>
</tr>
<tr>
<td>Orifice size</td>
<td>0.6 mm □ 16 mm</td>
</tr>
<tr>
<td>Orifice length</td>
<td>10 mm</td>
</tr>
<tr>
<td>MR fluid</td>
<td>□ MRF-132LD</td>
</tr>
<tr>
<td>Coil</td>
<td>□ 0.5 mm</td>
</tr>
<tr>
<td>Inductance</td>
<td>1.5 henries</td>
</tr>
<tr>
<td>Coil resistance</td>
<td>60 ohms</td>
</tr>
<tr>
<td>Max. current</td>
<td>0.08 A</td>
</tr>
</tbody>
</table>

3. EXPERIMENT

Various tests have been carried out using of the vibration-testing machine to verify the damping characteristics of the developed MRF dampers. Fig 4 shows a schematic diagram of the experimental setup for the MRF dampers. The actuator of the vibration-testing machine has a maximum dynamic load of 2000kN and a maximum velocity of 40cm/s. Various sinusoidal displacements are applied to the dampers and the generated damping forces are measured by a load-cell on the opposite side of the actuator. The input electric current to the electromagnet is selected as the one of the test parameters and is maintained to a constant value during the dynamic loading test.

Fig 5-1, Fig 5-2 and Fig 5-3 show the measured force-displacement loops for the 2kN MRF dampers under maximum input velocities of 5cm/s, 9cm/s and 20cm/s sinusoidal waveform movement, respectively. The right hand side loops of every figure show the test results for the MRF damper using the MR fluid, MRF-132LD, and the left hand side loops show the results for the MRF-128 damper. The dynamic loading tests were performed under input electric currents of 0A, 0.016A, 0.032A, 0.048A, 0.064A and 0.08A.
It was verified that the maximum damping force was controllable by adjusting the magnetic field. Furthermore, an upper limit of the controllable damping force was observed. The magnetic field of the electromagnet can be increased linearly to approximately 0.16A. Therefore, it is understood that this phenomenon is due to the magnetic saturation of the MR fluid. The force-displacement loop under no magnetic field exhibits the behavior of a viscous damper, especially in the high velocity range of around 20cm/s. The friction force of the sealing or the damping force due to the residual magnetization is dominant for the total damping force in the low velocity range.

Fig.6(a) and Fig.6(b) show the force-velocity relationships. Although the absolute values of the maximum damping forces of the two dampers are similar, the increase rate of the damping force of the MRF damper using the MRF-128NB is higher than that of the damper using the MRF-132LD. The dynamic range of the damper using the MRF-128NB is smaller in the high velocity range. These results mean that the MRF-132LD is more suitable for the developed bypass type MRF dampers. The MRF-132LD was used for the 20kN MRF damper.

Fig.7 shows the force-displacement loops for the 20kN MRF damper under 0.5Hz sinusoidal waveform. Fig.8 shows the relationship between the maximum input velocity and the measured maximum damping force. It was verified that the electrical currents applied to the electromagnet controlled the damping force of the 20kN MRF damper. The characteristics of the damping forces at the low velocity level, less than 5cm/s, were also investigated through these tests. It was confirmed from the Fig.7(a) and Fig.8 that the MRF damper functioned under a comparatively low velocity and over a small displacement range.

Figure 4  Experimental Setup for MRF Damper
Figure 5-1  Force-displacement Loops at 5cm/s

Figure 5-2  Force-displacement Loops at 9cm/s

Figure 5-3  Force-displacement Loops at 20cm/s
Figure 6  Force-velocity Relationship for 2 kN MRF Damper

Figure 7  Force-displacement Loops for 20kN MRF Damper
4. ANALYTICAL MODEL FORMULATION

This section proposes the analytical model for simulating the behavior of the MRF dampers, and compares the simulation and experimental results. Until now, some types of mechanical models for MR(ER) fluid dampers have been proposed. The simplest one is the Bingham viscoplastic model, in which a couple comprising a dashpot and a friction slider are connected in parallel. Gavin used the mechanical model proposed by Gamota to simulate the behavior of ER fluid dampers, in which the Zener element shows frequency dependent behavior over a wide range of the frequencies [Gavin et al. (1998)] \(^3\). The model proposed by Spencer consists of two springs, two dashpots and the Bou-Wen model, which can exactly simulate the behavior of both the displacement-force and velocity-force relationships of MRF dampers [Spencer et al. (1997)] \(^6\).

In this paper, the involution model is used to simulate the behavior of MRF dampers. In this model, the velocity-force relationship is expressed by:

\[
F = CV^\alpha 
\]

where \(F\) is the damping force, \(C\) is a constant independent of the frequency; \(V\) is the velocity of the piston and \(\alpha\) is an exponent such that \(0.0 < \alpha \leq 1.0\). This expression has often been used to simulate viscous fluid dampers, and is available to simulate the behavior of MRF(ERF) dampers because the damping force remains within the specified bound under the condition that \(\alpha\) is close to zero. In addition, since there are only two parameters in this expression(\(C\) and \(\alpha\)), they are easier to estimate than the elements in the mechanical models mentioned above.

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Fig.9 and Fig.10 shows the simulation results of the 2kN damper obtained from Expression (1). Both parameters(\(C\) and \(\alpha\)) are assumed to be independent of the amplitude and the frequency. Values of \(C\) and \(\alpha\), shown in Table2, were determined to minimize the square of the errors between the experimental force and the analytical one. Good agreement with experimental results is confirmed for each applied current.

The experimental results of the 20kN damper were simulated by following the same procedure as for the 2kN damper. The results are shown in Fig.11. Values of \(C\) and \(\alpha\) are identified as shown in Table3. As for the 2KN damper, it is found that the analytical model exactly predicts the behavior of the experimental results. In this case, though, some fluctuation is found in the simulation results. This behavior is due to noises in the observed displacement and can be removed by filtering.

Through a series of simulations, it is confirmed that the behavior of MRF dampers is exactly predicted by the velocity-force relationship expressed in (1) over a wide range of applied current, amplitude, and frequency. However, it should be noted that this analytical model can be applied only to simulate the dynamic behavior of MRF dampers, because the intrinsic property of the term “\(CV^\alpha\)” cannot simulate the static behavior.
Table 2  Values of Analytical Model \((F=CV^\alpha)\) for 2kN Dampers

<table>
<thead>
<tr>
<th>Current [A]</th>
<th>MRF-128NB</th>
<th>MRF-132LD</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(C[N(mm/s)^{\alpha}])</td>
<td>(\alpha)</td>
</tr>
<tr>
<td>0.000</td>
<td>97.4</td>
<td>0.47</td>
</tr>
<tr>
<td>0.016</td>
<td>349</td>
<td>0.31</td>
</tr>
<tr>
<td>0.032</td>
<td>608</td>
<td>0.23</td>
</tr>
<tr>
<td>0.048</td>
<td>788</td>
<td>0.20</td>
</tr>
<tr>
<td>0.064</td>
<td>940</td>
<td>0.18</td>
</tr>
<tr>
<td>0.080</td>
<td>920</td>
<td>0.19</td>
</tr>
</tbody>
</table>
Figure 11  Simulation Results (20kN Damper, Frequency=0.5[Hz])

Table 3  Values of Analytical Model ($F=CV^\alpha$) for 20kN Damper

<table>
<thead>
<tr>
<th>Current[A]</th>
<th>C[KN(mm/s)^\alpha]</th>
<th>\alpha</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>0.065</td>
<td>0.91</td>
</tr>
<tr>
<td>0.08</td>
<td>2.59</td>
<td>0.28</td>
</tr>
<tr>
<td>0.16</td>
<td>3.84</td>
<td>0.28</td>
</tr>
</tbody>
</table>
5. CONCLUSIONS

Various tests have been carried out using a vibration-testing machine to verify the damping characteristics of developed MRF dampers. The following test results were obtained: (1) Two types of dampers functioned by using one unit of the electromagnet under an appropriate electrical current control. (2) The magnitude of the damping force depends on the input magnetic field, but it has an upper limit. (3) In the absence of an applied magnetic field, an MRF damper exhibits viscous-like behavior, while it shows friction-like behavior in a magnetic field.

Through a series of simulations, it is confirmed that the behavior of the MRF dampers is fairly predicted by the velocity-force relationship expressed in equation (1) over a wide range of applied current, amplitude, and frequency.

It is clarified that the MRF dampers provide a technology that enables effective semi-active control in real building structures.

ACKNOWLEDGMENTS

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REFERENCES