Experimental and analytical methods for predicting mechanical properties of MRF damper

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ABSTRACT

First part of this paper covers experimental studies on mechanical properties of two types of magneto-rheological fluid (MRF) dampers. One is a commercial built-in-pass type damper and the other an original by-pass type damper. In the test, they are subject to cyclic sinusoidal displacements with different amplitudes, velocities and intensities of magnetic field. Not only their hysteretic properties but also their quickness to respond to the applied magnetic field are examined. In the second part, two analytical methods to represent the mechanical properties of the dampers are presented. One is a semi-empirical method making use of a Bingham Model to simulate the hysteretic properties of the damper. The other one, an analytical method based on the theory of non-Newtonian fluid. A design formula to predict the resistance of the damper is so obtained as to take into consideration the damper’s dimensions, the properties of the fluid and the intensity of the magnet field applied.

Keywords: Magneto-rheological fluid, Damper, Magnetic Field, Hysteretic loops, Bingham model, non-Newtonian fluid, Semi-active control, Built-in-pass, By-pass

1. INTRODUCTION

After the 1995 Kobe Earthquake, Japanese Building Standard Acts had been extensively reviewed and new one was issued in June 2000, which encourages structural engineers to turn to a performance based structural design, in which damping capacity of buildings plays more important role than their strength or stiffness. Therefore, to supplement the damping capacity of most ordinary buildings, many buildings have come to be equipped with a variety of energy absorbing devices, usually called passive dampers. However, although it may sound like we are expecting too much, there is still a problem with those passive dampers. That is, their damping capacities are fixed. If the capacity of the damper could be adjusted to what is desirable to minimize the response to any unpredictable excitations, seismic performance of the building would be much better than that achieved by the usual passive dampers. This sort of structural control is called a semi-active control of buildings. Among many devices to be used in this control system, the one making use of a magneto-rheological fluid damper (MRFD) seems quite promising, since the capacity of the damper is large enough to use for even large scale building structures and is readily controlled with less electric power. However to put it into practical use, most structural engineers have to be supplied with as much information on mechanical properties of the damper as possible, hopefully in the form of design formulas.

First part of this paper deals with experimental studies on mechanical properties of two types of MRFD. One is a...
commercial built-in-pass type damper and the other an original by-pass type damper. There are three points of interest in the test. First one is what kind of hysteretic properties they have. Next one is how the temperature growth affects the hysteretic properties. And the last one is how fast the damper can change its properties. In the second part, two simple methods to represent the mechanical properties of the damper are presented. One is a semi-empirical method making use of a Bingham Model to simulate the hysteretic properties of the damper. The other one, an analytical method based on the theory of non-Newtonian fluid, yielding a design formula to predict the resistance of the damper, taking into consideration the damper’s dimensions, the properties of the fluid and the intensity of the electric current applied.

2 EXPERIMENTAL STUDY ON DYNAMIC PROPERTIES OF MRFD

2.1 Test on a built-in-pass type MRFD

2.1.1 Outline of the test

This section summarizes the preliminary test conducted to confirm the availability of the MRFD in a semi-active control of building structures. The damper tested, shown in Picture 1, is a commercial one named a “Motion Master” from Lord Corporation. Fig. 1 shows the basic mechanism of the damper. Magneto-rheological fluid (MRF-132LD) is filled in the cylinder. Unique features of this damper are that its piston rod is single ended and that the magnetic coil is placed along the flow channel within the piston. In accordance with the intensity of the magnet field in the flow channel, the viscosity of the passing fluid, hence the capacity of the damper changes. An accumulator inside the cylinder is to account for the effective volume change due to the movement of the piston rod. In the test, harmonic displacements with different amplitudes, frequencies were applied. Corresponding load and the temperature of the cylinder are measured.

![Picture 1](Motion Master and its controller)

![Fig. 1 Basic mechanism of Motion Master](Flow channel with magnetic choke)

2.1.2 Important findings in the test

The first point of interest in the test was how the loading frequency, amplitude and intensity of magnetic field affect the hysteretic properties of the damper. Hysteretic loops are shown in Fig. 2 for three cases of combinations of frequency and amplitude, each for different constant voltage levels. In the absence of electricity, that’s there is no magnet field, the loops are similar to ellipsoids like those of viscous dampers. But as the applied voltage increases, the hysteretic loops
change to rigido-plastic ones. And for greater voltage than 4.0 to 5.0 volts, the resistance of the damper seems to saturate. In general, the damper exhibits rigido-plastic load-deflection relationship with some viscous resistance as in Fig. 2a and 2b. But in the case that displacement amplitude is as small as of those in Fig. 2c, hysteric loops become a bit unstable.

![Hysteric Loops to sinusoidal displacement](image)

Another point of interest was the effect of temperature on the hysteric properties of the damper. The test consists of three cases. In case 1, a sinusoidal displacement with amplitude of 10mm, frequency of 1.0 Hz alone was continued for 60 minutes. In case 2, the same loading as in case 1 and an electricity of 4.0 volts are applied at the same time. And in case 3, an electricity of 4.0 volts alone was applied. Figure 3 shows the test results of the case 2. There is a considerable amount of temperature growth, from 30 to 155 degrees Celsius, which causes a slight deterioration in the resistance. Since it is confirmed that there is almost the same amount of temperature growth in case 3, and it is not so great in case 1, the temperature growth can be attributed to the heat from the electric magnet coil inside the cylinder.

![Temperature Growth during Loading](image)

Now, the third point of interest was how fast the damper can respond to the change of intensity in magnet field and attain the resistance having been signaled to be. In this test, a triangle displacement wave with 0.25 Hz frequencies and 20mm amplitude is used to perform the test under constant velocity. On the top of Fig. 4, shown is the time history of the resistance in response to the step-on voltage, from 0 to 4 volts. On the bottom is the case of step-off voltage. These figures show that it takes 0.03 to 0.1 seconds for the damper to attain the signaled resistance. This time lag may be short enough to
perform an on-off semi-active control on buildings with relatively long natural period, but a bit too long to realize a fully adaptive control on buildings with relatively short natural period.

Fig. 4 Time Lag of resistance to electric current

2.2 Test on a built-in-pass type MRFD

2.2.1 Outline of the test

Since the results of the preliminary test suggested that a slight modification in mechanism of the MRFD is needed, we developed a new type MRFD. Test results on this new damper will be shown in this section.

Mechanism of the new damper is shown in Fig. 5. It is different from that of the previous one in two ways. For one thing, the piston rod is double ended and so an accumulator is not provided. And another thing, the MRF passes through a by-pass to which magnet field is applied from outside. The same fluid as used in the previous one is filled in the cylinder. With the movement of the piston rod, the fluid flows in the by-pass. Nominal capacity of this damper is 20 kN. Picture 2 shows a test setup. In the test, harmonic displacements of different amplitudes and frequencies are applied. Corresponding...
load and the temperature of the cylinder are measured

2.2.2 Important findings in the test

Some hysteretic loops obtained in the test are shown in Fig. 6. In these figures, the loops are shown for the same combinations as in the case of the Motion Master, each for nine different levels of electric current. It is confirmed that hysteretic loops of this damper are more stable than those of Motion Master, especially when the amplitude is very small.

![Fig. 6 Hysteretic loops of a Built-in-pass type MRFD to sinusoidal displacement](image)

![Fig. 7 Temperature change in case 1](image)

![Fig. 8 Temperature change in case 3](image)

![Fig. 9 Hysteretic loops of case 1 at different times](image)

Figure 7 shows the time history of the temperature of the damper exposed to a cyclic displacement with frequency of 0.5 Hz and amplitude of 10 mm for 60 minutes without electric current. In this case, there is a little temperature growth but no deterioration is observed in hysteretic loops as can be seen in Fig.9. Figure 8 shows the temperature change of the damper that is subject to just a constant electric current of 0.1A. No change is observed in this case, which is quite different from the results observed in the case of the built-in-pass type MRFD in the previous section. It can be said that installment
of an electric magnet coil outside the cylinder really works to prevent extreme temperature growth. Figure 10 corresponds to the case that both electric current of 0.1 A and harmonic displacement are applied at the same time. Since the resistance of the damper and hence the energy absorption are great in this case, the temperature increases more than 30 degrees Celsius. However, hysteretic property of the damper shows no difference at all. This result seems to have come from not only that the magnet coil is placed outside the cylinder but also that the piston rod is double ended and an accumulator is not provided.

![Graph showing temperature change over time with cyclic loading and electric current.](image)

Fig. 10 Temperature change to Cyclic loading with electric current of 0.1A

![Graph showing time lag of resistance against electric current.](image)

Fig. 11 Time lag of resistance against increasing/decreasing electric current

Although the signal to change electric current is instantly applied, an
actual electric current changes very slowly because of the power supplier’s capacity. It may be seen from Fig. 11 that change of electric current and that of resistance corresponds fairly well. However, if you take a closer look at the initial points of the change, there is a little difference between them. The delay of the resistance against electric current is found to range from 0.02 seconds to 0.04 seconds.

2.3 Semi-empirical method to construct mechanical model of MRFD

Figure 12 shows the relationship between the velocity and corresponding resistance for different values of applied electric current to the 20 kN by-pass type damper. Although the test results show some dispersion, the relation suggests that the mechanical model of the by-pass type MR damper can be represented by a simple Bingham model in Fig. 13, that is a combination of a Coulomb slider and a dashpot. The coefficients of a Coulomb slider and a dashpot are determined based on the test results.

Figure 14 is prepared to confirm the accuracy of the proposed Bingham model to represent dynamic behaviors of the MRFD. On the top to the left is the input random displacement and to the right is an input random electric current. On the
bottom to the right, you can see that the time history of the resistance for experimental result and analytical result coincide fairly well. And on the bottom to the left, two hysteretic curves are compared. Upper one is obtained in the test and the lower one is obtained by the analysis. From these results, it is confirmed that the Bingham model can be used to simulate the real behaviors of the by-pass type MRFD.

**3. DESIGN FORMULA FOR DAMPING FORCE OF MRFD**

**3.1 Basic theory**

In this section, a simple formulation to predict the damping force of the MRFD is introduced. Since the MRF, when exposed to external magnetic field, exhibits non-Newtonian behaviors, a Herschel-Bulkeley model\(^1\) is used in this study. Let the fluid in the by-pass be flowing in x-direction as shown in Fig.15 at the speed of \(u\), then the relationship between the shear strain rate \(\frac{du}{dy}\) and the shear stress \(\tau_y\) will be expressed as

\[
\begin{align*}
\tau_y &= \tau_0 + k \left| \frac{du}{dy} \right|^n \quad \text{for} \quad \tau_y \geq \tau_0 \\
\frac{du}{dy} &= 0 \quad \text{for} \quad \tau_y \leq \tau_0
\end{align*}
\]

(1)

where, \(\tau_0\) is the yield stress of the fluid. Symbols \(k\) and \(n\) are constants depending on the fluids' own mechanical properties. In Fig. 15, \(d\) denotes the bypass height and \(\delta\) is the plug thickness defined as \(\delta = 2\tau_0 \frac{dx}{dp}\) which is obtained by considering the boundary conditions of the shear stress and the momentum equation of laminar flow described as

\[
\frac{dp}{dx} = \frac{d}{dy} \tau
\]

(2)

where \(\frac{dp}{dx}\) is the pressure gradient.

Substituting Eq. (1) into Eq. (2), distribution of the velocity \(u\) along the height is obtained as

\[
u = \frac{n}{n+1} \left( -\frac{1}{k} \frac{dp}{dx} \right)^\frac{1}{n} \left( \frac{d - \delta}{2} \right)^\frac{n+1}{n}
\]

(3)
for plug flow region, and
\[
u(y) = \frac{n}{n+1} \left( -\frac{1}{k} \frac{dp}{dx} \right)^{\frac{1}{n}} \left[ \left( \frac{d - \delta}{2} \right)^{\frac{n+1}{n}} - \left( \frac{d - \delta}{2} - y \right)^{\frac{n+1}{n}} \right]
\]  
(4)

for post yield region(1).

The volumetric flow rate is written as
\[
|Q| = A \cdot \nu = w \int_0^d u(y) \, dy
\]
and substituting Eqs. (3) and (4) into Eq. (5), the total damping force \(F\) is obtained as
\[
F = A \cdot L \frac{dp}{dx} = \frac{(n+1)^n (2n+1) \, L \cdot k \cdot A'}{n^n} \frac{1}{w^a} \frac{1}{[n(d + \delta) + d]^{\frac{n+1}{n}}} \left( \frac{d}{d - \delta} \right)^{\frac{n+1}{n}} v^n
\]
where \(L\) is the by-pass length, \(w\) is the by-pass width, \(A\) is the piston head area, and \(\nu\) is the piston velocity.

Since the closed form solution for \(F\) can’t be obtained, several approximate formulas have been proposed(1-3).

For example, supposing \(n=1\), Eq. (6) becomes
\[
d^3 \frac{dp}{dx}^3 - \left( 3d^2 \tau_0 + \frac{12kAv}{w} \right) \frac{dp}{dx}^2 + 4\tau_0^3 = 0
\]
(7)

and this equation can be solved analytically\(^3\) as
\[
P = \frac{2}{3} (1 + 3T) \left[ \cos \left( \frac{1}{3} \arccos \left( 1 - 54 \left( \frac{T}{1 + 3T} \right)^3 \right) \right) + \frac{1}{2} \right]
\]
(8)

where, \(P = \frac{wd^3}{12kA} \frac{dp}{dx}\) and \(T = \frac{wd^2 \tau_0}{12kA} \).

In this study, an exact relationship between \(F\) and \(\nu\) will be given by substituting \(\frac{dp}{dx}\) into Eq. (6) and by solving it for the velocity \(\nu\). For the two types of MR dampers dealt in this paper, the solution is obtained in more concise form as
\[
F = \left( \frac{dp}{dx} \right) A \cdot L = \left( \frac{C_1}{d^3 w} + \frac{3 \cdot 2^7 C_2}{d^3 w C_3^3} + \frac{C_3}{3 \cdot 2^3 d^3 w} \right) A \cdot L
\]
(9)

where \(C_1 = 4akv + d^2 \tau_0 w, C_2 = 54C_1^3 - 108d^6 \tau_0^3 w^3\) and \(C_3 = C_2 + \sqrt{C_2^2 - 2916C_1^6}\).

For \(\nu \neq 0\), the damping force, \(F\), collapses to:
The yield stress $\tau_0$ is assumed here to be expressed as the quadratic function of the electric current (Amp).

$$\tau_0 = E_1 \cdot Amp + E_2 \cdot Amp^2$$ (11)

in which, $E_1$ and $E_2$ are constants which are identified based on the experimental results. In order to get more accurate damping force, the force caused by the pressure loss given by Eq. (12), should be taken into consideration.

$$F_1 = \frac{\rho A^2 v^2}{2w^2h^2}$$ (12)

where, $\rho$ is the density of the fluid. However, since it is readily predicted from Eq. (6) that this force is orders of magnitude smaller than that given by Eq. (6), this force is not taken in this study.

### 3.2 Analytical results

Availability of the formulation above will be confirmed here by comparing the analytical results with the experimental results of two types of MRF dampers described in sections 2. Through a series of numerical studies, $n$ was set to 0.85 for the built-in-pass type 2KN damper and 1.1 for the bypass type 20KN damper respectively. Comparisons between experimental and analytical results is shown in Fig. 16. It is found that analytical results and experimental ones coincide fairly well for different levels of electric current.

![Comparison of the damping force between experimental and analytical results](image)

**a. 2KN Motion Master (n=0.85)**

**b. 20KN Bypass type (n=1.1)**

Fig. 16 Comparison of the damping force between experimental and analytical results

### 4. CONCLUSIONS

It has been confirmed that the MRFD is quite attractive to be used as a device to perform a semi-active structural control of building structures. For one thing, the resistance of the single MRFD is large enough to accommodate shear force in building structures induced by intense seismic force. And another thing, the range in which the damper changes its
capacity is so wide as to be effectively made use of in the semi active control of buildings. Rise time against sudden change of magnet field seems short enough to apply even an adaptive control of buildings, but full-scale shaking table test seems necessary to confirm the real availability of the MRFD to structural control of buildings against strong ground motion.

ACKNOWLEDGMENT

The authors wish to thank the members of ER/MR working group in the effectors technology section of the Japan side of the US-Japan cooperative research project, “Smart Materials and Structural Systems” for their fruitful discussions.

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