State-of-the-art recent developments of large magnetorheological (MR) dampers: a review

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Abstract
Large MR (MR) dampers are popular due to their higher damping force capabilities which makes them suitable in the field of civil engineering, structural engineering, suspension bridge structure, mining engineering, and agricultural engineering applications. This paper presents a comprehensive review of large MR dampers. The classifications and applications of large MR dampers, the principle of operation, different fluid models, their structural design and control systems are classified and reviewed in this paper. The large MR dampers have higher damping force controllability than conventional MR dampers. The review indicates that the large MR dampers have enough vibration mitigation ability and higher damping performances.

Keywords Magnetorheological (MR) damper · MR fluid · MR fluid model · Large MR damper · Control algorithm

List of symbols

| Symbol | Description |
|--------|-------------|
| a, b, c | Damper characterization parameters |
| $A_e$ | Effective area of the piston |
| $a$ | Velocity adjustment coefficient of coulomb damping force |
| $A_p$, $A_m$ | State matrices |
| $B_p$, $B_m$ | Input matrices |
| $C_i$ | Damping coefficient |
| $C_{post}$ | Post-yield damping |
| $c$ | Viscous coefficients |
| $c_1$ | Pre-yield viscous |
| $c_k$ | Viscous damping for force roll off |
| $c_1$ | Viscous damping at large velocities |
| $c_p(k)$ | Post yield plastic damping coefficient |
| $F$ | Damping force |
| $F_{pre}$ | Pre-yield damping force |
| $F_{post}$ | Post-yield damping force |
| $F_{y}$ | Yield force |
| $f_c$ | Frictional force |
| $f_{py}$ | Pre-yield damping force |
| $f_{py}$ | Yield force |
| $f_{0}$ | Damping force when the damper velocity is zero |
| $f_{m}$ | Damping force when the damper velocity is maximum |
| $f_{d}$ | Damp force offset |
| $f_{d}$ | Coulomb friction force |
| $F_{c}$ | Coulomb force |
| $F_{z}$ | Shear stress |
| $F_{k}$ | Damper force caused by seals and measurement bias |

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| Symbol | Definition | Symbol | Definition |
|--------|------------|--------|------------|
| $F_v$  | Viscous stress | $x_0$ | Spring initial displacement |
| $F_b$  | Control force | $\dot{x}_H$ | Hysteric velocity |
| $f$    | Measured force | $x_c$ | Characteristics displacement |
| $G$    | Influence vector | $x_v$ | Displacement value when damping force was zero |
| $g$    | Positive gain coefficient | $x_{s0}$ | Sprung mass displacement |
| $h$    | Heaviside step function | $\tilde{x}$ | Vertical displacement |
| $H_0$  | Strength of the magnetic field | $\tilde{x}^e$ | Tangential curve velocity |
| $I$    | Current | $g$ | Ground acceleration |
| $k$    | Stiffness coefficients | $y_p$ | Plant output |
| $k_1$  | Stiffness at large velocities | $y_m$ | Model output |
| $k_a$  | Accumulator stiffness | $y_j$ | Measured output vector |
| $k_s$  | Linear spring constant | $||z||_p$ | Norm of the state |
| $k_c$  | Consistency index | $\alpha, \beta, \gamma, \eta$ | Model parameters |
| $K_c$  | Linear optimal controller | $a_1, a_2, \mu$ | Positive constants |
| $k_1$  | Pre-yield stiffness | $\tau_y$ | Dynamical viscosity |
| $k_H$  | Shape coefficient | $\eta$ | Yield shear stress |
| $k_s, k_{wa}$ and $k_{wb}$ | Hysteresis loop controlling parameter | $\mu$ | Coefficient of friction |
| $l$    | Effective length of the piston | $\alpha, \beta$ | Angular position |
| $L$    | Laplace transform | $P_{ij}, A_j, B_j$ | System matrices |
| $M$    | Mass | $\gamma$ | Dynamic viscosity of MR fluid |
| $m_0$  | Mass acceleration | $\Gamma$ | Strain rate |
| $n$    | Power law index | $\lambda$ | Column vector |
| $N(t)$ | Controllable contact force | $\Phi$ | Vector |
| $q_0, q_1, q_2, q_3, r_0, r_1, r_2, r_3$ | Optimal co-efficient of the polynomial equation for control signal | $\Delta(t)$ | Convergence rate of sliding mode control |
| $R_p$  | State vector. | $R_{m}$ | N x 1 plant state vector $n_m$ x 1 model state vector |
| $\Delta(t)$ | Damping deformation | $u$ | Dimensionless displacement |
| $u_r$  | Relative velocity | $u_0$ | Harmonic motion with an amplitude |
| $u_{0}$ | Absolute displacement of the single degree of freedom | $u_a$ | M x 1 input control vector |
| $u_r$  | Harmonic motion with an amplitude | $u_m$ | M x 1 input command vector |
| $u_p$  | M x 1 input control vector | $V(z)$ | Lyapunov function |
| $V_{ref}$ | Reference velocity | $V^\alpha$ | Velocity of the piston |
| $v_0$  | Input voltage to the current driver | $V_i$ | Measured noise vector |
| $V_{max}$ | Maximum allowable voltage | $\omega_2$ | Angular speed of the upper lever |
| $\omega$ | Angular speed of the upper lever | $\omega$ | Frequency |
| $\chi$ | Frequency | $x$ | Displacement of MR damper |
| $x_0$  | Displacement initial mass | $\dot{x}$ | Velocity |
| $\dot{x}$ | Velocity | $\chi_0$ | Initial mass velocity |

1 Introduction

Magnetorheological (MR) fluid which is a smart material was first developed by Jacob Rainbow in the 1940s [1]. MR damper [2] is a vibration control device that uses MR fluid for its operating environment was first developed by Lord Corporation in the early 1940s [3, 4]. Since then MR fluid has become an important engineering field to develop. MR fluid contains suspended iron particles in oil or carrier fluid [5, 6]. In the presence of a magnetic field the rheological properties (yield stress) of MR fluid change within milliseconds [7]. The iron particles in MR fluid align along the direction of the magnetic field and form a chain structure thus transform it from viscous into a semi-solid state [8]. In an MR damper, the magnetic field is generated and controlled using an external power source that supplies current to the piston coil [9]. Thus controllable damping force can be achieved [10]. This controllable mechanical properties of MR fluid attracted many researchers to develop different MR devices [11–14] through the years. MR damper based semi-active control system came to the attention of many researchers and has been developing as a shock reducing device due to its controllable damping force [15], simple design [16], low power usage [17] and cost-effectiveness.
MR damper has been practically utilized in different engineering applications. It has been developed for automobile suspension [19, 20], railway vehicles [21, 22], helicopter landing gear systems [23, 24], civil infrastructure [25–27], cable bridge [28, 29], and vibration isolation system [30].

MR dampers were commercially applied on vehicle suspension system [31] as it reacts to vibration motion quickly and provides sufficient damping force. Thus, human comfort during riding is achieved. Desai et al. [32] examined the damping performance of the RD-8040-1 MR damper for seat suspension that ensured better damping range and ride comfort. Whereas, Du et al. [33] proposed an MR damper-based suspension system using an adaptive skyhook control that improved the vehicle ride performance further. Besides, MR damper is commercially implemented by many researchers on the washing machine [34, 35], prosthetic knee [36, 37] applications also. The required damping force is less in these cases.

Different damping devices are used against earthquake and wind-induced structural vibration. Passive control devices were incorporated inside the building structure to absorb energy from earthquake vibration. Among them, fluid viscous dampers [38, 39], viscoelastic dampers [40, 41], hysteretic dampers [42, 43], metallic and friction devices [44] are mostly used. But the use of these damping devices reduces due to higher cost, high nonlinear response, fluid leakage and less reliability issue [44]. MR damper is also proposed to use in different structural areas where higher damping force is required to isolate the large-frequency vibration. The first large MR damper with 300KN capacity was developed by Sanwa Tekki Corporation in 2001 and installed at Tokyo National Museum of Emerging Science and Innovation for protection against seismic excitation [45]. Later in 2003, a 400KN MR damper was used in a residential building at Keio University in Japan developed by Sanwa Tekki Corporation [46, 47]. As MR damper has better response control over passive dampers, researchers [48–53] have been developing large-scale MR damper for bridge, railway bridge, building structure through the years.

Heo et al. [54] developed a sliding mode control with optimal polynomial control based MR damper (30KN) system with lumped mass to mitigate pounding between spans and abutment under seismic load. The experimental result showed it could mitigate the pounding of the bridge span effectively whereas the damage of bridge piers was experimentally reduced by Heo et al. [55] using a hybrid seismic response control based MR damper (1000KN) system. The active systems consume more energy in earthquake or wind vibration reduction [56]. To decrease the power consumption, semi-active or adaptive systems were developed for reducing the wind and earthquake induced structural vibration [57–59]. Yeganehfallah and Attari [60] proposed a robust controller and simulated the response phenomenon of the cable-stayed bridge structure with an MR damper (1000KN)-based semi-active control system. For the same control system, Bathaei et al. [61] proposed two different types of Fuzzy logic controller (FLC) where the type-2 FLC was proven more effective in reducing the response time of bridge structure, whereas six semi-active fuzzy controllers were devised by Hormozabad and Tanha [62]. A similar study was also examined using a building model by Bathaei et al. [63] with a tuned mass system with an MR damper(1000KN) where the type-2 FLC controller was also worthwhile in performance. As the fuzzy controllers have some lacking, Bozorgvar and Zahrai [64] designed an adaptive neuro-fuzzy interference system (ANFIS) for MR damper to reduce the response time of building structure. The system had better efficiency than other controllers. Bhaiya et al. [65] developed a control system for MR damper-based building structure and showed that it is less effective when subjected to near fi eld earthquake. Fu et al. [66] developed two control system and experimentally showed that a 20KN capacity MR damper-based isolation system in a concrete structure responds quickly against a different level of a large earthquake. Gong et al. [67] developed a 10 kN capacity MR damper with a pseudo-negative-stiffness (PSN) control system. Experimental results showed that under different level of earthquake it performs better than other control systems. Cruze et al. [68] proposed a multi coil large MR damper and experimentally validated that it can generate sufficient damping force of 5.83kN for seismic mitigation of building structure.

This paper aims to review a literature on large MR damper, their classification and application, their design strategy, implementation, and development over the years. This paper also presents the classification of large MR damper based on different mathematical models and control systems.

2 Applications of MR dampers

Both active and passive suspension systems can be summarized by MR dampers thus attracted the attention of many researchers to use MR damper in different applications. Besides, the high damping force and durability of MR damper replaced other vibration control devices in many engineering applications. Several MR damper systems and their applications are presented in Table 1.

Different MR fluid-based devices application are shown in Table 2.

2.1 Classifications

The optimization in design can enhance the performance by changing the number of the coil-like single-coil [108],
double coil [109], multi-coil [110]. The classifications of MR dampers depend on their design, coils turn number, piston coils, bypass valve, control valve, and power-producing capacity.

The main two basic types of MR dampers are monotube [111, 112] and twin-tube [112, 113] which are either can be double-ended [114, 115] or single-ended [116] MR dampers. Monotube MR damper contains one fluid reservoir while the twin-tube has two reservoirs [117]. MR damper with a single-ended structure has one piston rod while the double-ended structure has an extended piston rod from both ends of the cylinder. MR damper can either have inner or outer coils mechanism. In an inner coil mechanism, the coils are wounded inside the piston of the MR damper [118] while the external coils [119] are wound on the outer structure of the damper. The piston incorporates a different number of coils that can be a single coil, double coils or multi coils. Based on control valves, the flow mode in MR damper can be categorized as single flow mode [120] and mixed flow mode [121]. Single flow mode MR dampers can be characterized as flow mode [122], shear mode [123], and squeeze mode [124] MR dampers.

Based on different flow channel MR dampers can be classified as inner bypass or outer bypass which either can be single-ended [125], double-ended [126], or piston bypass [125, 127] type. The outer bypass MR dampers can be categorized as outer tube bypass [128, 129], double-ended bypass [129], bypass valve MR [130], meandering type valve [131] and bypass spool valve [132] MR dampers. According to the size, MR dampers can be classified into three types such as short stroke, long stroke and large MR dampers. In the short stroke and long stroke MR dampers, the stroke length varies from 55 to 74 mm [133] while for large stroke MR dampers, stroke length varies from 160 to 300 mm.

### 2.2 Working principle of MR damper

The working system of conventional MR damper is shown in Fig. 1. The MR damper device is installed with other sensor and power system that provides information about controlling damping force. An external power source is utilized to supply current to the piston coil while the piston reciprocates to and fro within the cylinder chamber [134]. This current induces a magnetic field around the fluid flow path and under the interaction with magnetic field the fluid changes its phase from liquid to solid state [135]. The system controller takes data from sensors that is connected to the system where damping force is required. Thus, the current driver delivers different level of current as per requirement and controllable damping force obtained. A typical MR damper is shown in Fig. 2.

### 2.3 Operation modes

The MR dampers are a type of MR dampers that utilize the larger stroke into a shear stress development in the MRFs region. The principles of operation of large MR dampers are based on shear mode operation, flow mode operation, squeeze mode and mixed-mode operation. MR damper

| Table 1 | MR dampers and their application |
|---------|---------------------------------|
| Field of application | References |
| Civil infrastructure | [69, 70] |
| Automobiles | [71, 72] |
| Battery recoil system | [73, 74] |
| Semi active vibration isolation systems | [75–77] |
| Railway vehicles | [78, 79] |
| Helicopters applications | [80, 81] |
| Gun recoil systems | [82, 83] |
| Military vehicles | [84] |
| Optical polishing | [85] |
| Fluid clutches | [86, 87] |
| MR powertrain mount | [88, 89] |
| Polishing industry | [90] |
| Engine mounts and clutch systems | [91, 92] |

| Table 2 | MR devices and their application |
|---------|---------------------------------|
| MR devices | Applications | References |
| MR damper | Shock absorber, heavy-duty, washing machine, robot, haptic devices, cable bridge | [34, 93] |
| MR-based elastomers | Engine mounting system, sensors | [94, 95] |
| MR valves | MR damper, actuator | [96, 97] |
| Seals | Rotating shaft of a machine | [98, 99] |
| Hydraulic valves | Actuator, converters | [100, 101] |
| Composite structure | Building structure, beams, panels, plate | [102–104] |
| Flexible structure | Turbine blades | [105] |
| Pneumatic motion control system | Linear resistance component | [30] |
| Braking system | Suspension device, aerospace applications | [106, 107] |
operations are divided into three parts namely single flow mode [137], mixed-mode [138], and multimode [139]. The combined operation of the valve and direct shear mode is called mixed mode. On the other hand, the squeeze mode [140], direct shear mode [141] and valve mode [142] are called multi-mode operations.

Particularly, when the fluid is translated parallel to the wall are called shear mode (Fig. 3a) [143]. In the flow mode MR damper, the bi-fold mode causes high-pressure differences to develop higher damping force in a small volume. This is also illustrated in (Fig. 3b) [37].

Figure 4a presents the mixed-mode MR damper as a combined working mode of shear and squeeze mode. In the mixed-mode operation, the MR damper generates a higher damping force in comparison with the MR damper [127, 144, 145]. Figure 4b presents the squeeze mode MR damper which transpires due to the wall sliding movement and squeezing out the fluid [146].

Among the three-damping operation, mixed-mode MR dampers are more controllable and generate a higher damping force.
Applications of large MR dampers

Table 3 shows the applications of different large MR damper.

| MR damper system                        | Applications                                                                 | References          |
|-----------------------------------------|-----------------------------------------------------------------------------|---------------------|
| MR damper                               | Heavy truck, knee prosthetics of limbs, a humanoid robot, haptic devices, motion master, shock absorber | [36, 147, 148]     |
| MR damper-based suspension system       | Railway suspension, Tracked vehicle suspension                              | [21, 149]           |
| MR damper-based structure               | Seismic and wind loading, earthquake mitigation, wind vibration reduction    | [10, 45, 150]       |
| Scissor jack braced MR damper           | Wind turbine seismic load control                                           | [151]               |
| MR valves                               | Actuator                                                                    | [152–154]          |
| Smart outrigger MR damper               | Reduction of the dynamic response of tall buildings                        | [155]               |
| MR damper-based bridge structure        | Pounding mitigation of multi-span bridge                                    | [156, 157]          |
| MR damper-based pipeline                | Reduction of low-frequency pipeline vibration                               | [158]               |
| Rotary MR damper                        | Vibration control of stay cables                                            | [159]               |
| Semi-active MR damper system            | The cable-stayed bridge, Building structure                                 | [62]                |

3 Applications of large MR dampers

Table 3 shows the applications of different large MR damper.

3.1 Large MR damper working principle

Large MR damper works similarly to conventional MR damper. Large MR damper consists of a piston, piston rod, cylinder, electromagnetic coil, seal, shaft bearing, MR fluid and accumulator [160]. An external power supply is used to supply sufficient current to the coil that produces a magnetic field. Figure 5 shows the schematic diagram of large MR damper. The sensors and controller are used to detect the displacement of the structure. During piston movement, the MR fluid in the cylinder flows through the orifice of the piston and the fluid transforms its phase from liquid to semi-solid due to the presence of the magnetic field [160]. Thus, required damping force obtained and vibration controlled. Figure 5 shows the large MR damper components.

4 MR Damper numerical models

To design an efficient semi-active control system for MR dampers several fluid models are required. Till to date, a
group of researchers developed different mathematical models for analyzing behavior and characteristics of MR dampers performances. To predict the response of physical MR damper, several techniques, parametric models, and reliable approach has designed already, and those models can predict non-linear response. Table 4 shows the different fluid models for MR dampers. Among these, the Bingham model, the Bouc-Wen and the Modified Bouc-Wen models are some of the most common models utilized to predict the characteristics of MR dampers [161]. In this regards, large MR dampers are one of the categories of MR dampers.

5 Large MR damper classification

5.1 Monotube Large MR damper (single-ended)

The first monotube large-scale MR damper was developed by Lord Corporation [193] in the 1990s. In 2005s the second generation of large MR damper was also developed by Lord Corporation [193].

5.1.1 Bingham model-based monotube large MR damper (single-ended)

Sodeyama et al. [194] developed the Bingham model-based three types of MR dampers having capacity of 2 kN, 20 kN, and 200 kN (Fig. 6a) which included two types of MR fluids, and two hysteretic models. The damping forces versus displacement showed a significant increment as frequency and trial product #104 by Bando Chemical than conventional MRF-132LD by Lord Corporation. A typical involution model was developed to characterize force–velocity as shown in Fig. 6b presents the involution model for large MR damper (Eq. 1).

In the Fig. 7a–7d presents several large MR damper applications and their real-world application. Figure 7a–7c shows that large MR damper connected to the structure as a

| Table 4 MR dampers fluid models | Modelling technique | MR damper Models |
|---------------------------------|---------------------|------------------|
| Bingham models                  | Original Bingham model [56] |
|                                 | Modified Bingham model [162] |
|                                 | Gamota and Filisko model [163] |
|                                 | Updated Bingham model [164] |
| Herschel-Bulkley model          | Nonlinear bi-viscous model [165] |
| Bi-viscous models               | Nonlinear hysteretic bi-viscous model [166] |
|                                 | Nonlinear hysteretic arctangent model [167] |
|                                 | Lumped parameter bi-viscous model [169, 170] |
| Visco-elastic–plastic models    | General visco-elastic–plastic models [171] |
|                                 | Visco-elastic–plastic model [172] |
| Stiffness-viscosity-elasto-slide (SVES) model | BingMax model [174] |
|                                 | Maxwell model [175] |
| Maxwell models                  | Simple Bouc-Wen model [56] |
|                                 | Modified Bouc-Wen model [56] |
| Bouc-Wen models                 | Bouc-Wen model for shear mode dampers [176, 177] |
|                                 | Bouc-Wen model for large-scale dampers [50] |
|                                 | Current dependent Bouc-Wen model [178] |
|                                 | Current-frequency-amplitude dependent [179, 180] |
|                                 | Non-symmetrical Bouc-Wen model [181] |
| Dahl models                     | Modified Dahl model [182] |
|                                 | Viscous Dahl model [182] |
| LuGre models                    | Modified LuGre model [183, 184] |
|                                 | Modified LuGre model [185, 186] |
| Hyperbolic tangent models       | [187] |
| Sigmoid models                  | [188] |
| Equivalent models               | [189] |
| Phase transition models         | [190–192] |
vibration support device for seismic vibration control. Figure 7d shows regenerative large MR damper used in structure support for vibration control.

\[ F = C_V V^n \]  

(1) where \( F \), \( C_v \) and \( V^n \) are the damping force, damping coefficient and velocity of the piston.

Stanway et al. [198] investigated the electrorheological (ER) damper and proposed a mechanical model. This model is known as the Bingham plastic model. This model
combines a viscous damper a dashpot and a coulomb friction element which are placed in parallel as shown in Fig. 8. The nonlinear Bingham plastic model (Eq. 2) usually used for characterizing MR dampers force from Fig. 8a

\[ F = C_i \dot{x} + f_c \text{sgn}(x) \]  

where, \( C_i \) and \( f_c \) represents the damping coefficient and frictional force connected to the fluid yield stress and \( \text{sgn} \) for signum function. \( x \) and \( F \) are the displacement of MR damper and damping force [28].

However, the Bingham behavior of an MR damper can also be derived from the Bingham plastic model for MR fluids given by Eq. (1) through the study of an axisymmetric model of the MR fluid flow [30]. Wereley et al. [93] investigate the Bingham model where the parallel plate geometry or axisymmetric model is used to develop an MR damper model using shear force mechanism and supports the several MR dampers models [93, 106, 108]. The following equations are as follows:

\[
F(t) = \begin{cases} 
C_{\text{post}} \dot{x} + F_y, & \dot{x} > 0 \\
-F_y < F(t) < F_y, & \dot{x} = 0 \\
C_{\text{post}} \dot{x} - F_y, & \dot{x} < 0
\end{cases}
\]  

where, \( C_{\text{post}} \) is the post-yield damping and \( F_y \) is the yield force and \( \dot{x} \) is velocity.

The model is given by Eq. (3) assumes that, in the pre-yield condition, the material is rigid and does not flow; hence, when \(|F(t)| < F_y\), the shaft velocity \( \dot{x} = 0 \). Once the force applied to the damper exceeds the yield force, then the fluid begins to flow and the material is essentially a Newtonian fluid with nonzero yield stress. In this constitutive model, the yield force is obtained from the post-yield force versus velocity asymptote intercept with the force axis. The Bingham model accounts for MR fluid behavior beyond the yield point, i.e., for fully developed fluid flow or sufficiently high shear rates. However, it assumes that the fluid remains rigid in the pre-yield region. Thus, the Bingham model does not describe the fluid elastic properties at small deformations and low shear rates, which are necessary for dynamic applications [113]. Considering that the width of the hysteresis loop with the Bingham model is relatively narrow, Weng et al. [127] constructed a more complicated model to represent the wider hysteretic loop and the updated model can be expressed by Eq. (4). The following equation can be written as:

\[
F(t) = C_0 \dot{a} + 2 \pi f_c \arctan \left( \frac{k_H}{\dot{a} - \dot{x}_H \text{sgn}(\ddot{x})} \right) + f_0
\]  

where \( k_H \), \( \dot{x}_H \) and \( \dot{a} \) are the represents the shape coefficient, hysteretic velocity and acceleration. \( k_H \) and \( \dot{x}_H \) are the functions of applied current \( I \).

A 400KN large MR damper (Fig. 9) was developed by Fujitani et al. [47] using Bingham visco-plastic model for civil structural vibration control. Several research groups developed large MR damper using the Bingham model for time delay reduction [164, 198, 199] and they investigated time delays in the case of control systems, electrical parts, and mechanical parts of the dampers.
5.1.2 Maxwell Nonlinear Slider (MNS) model-based monotube large MR damper (single-ended)

Chen et al. [200] developed a monotube large MR damper (Fig. 10) based on Maxwell Nonlinear Slider (MNS) model for real-time hybrid simulation. The MR fluid behavior (pre-yield and post-yield region) was characterized by the MNS model by utilizing Herschel–Bulkley fluid model. Bouc-Wen model, hyperbolic tangent model and MNS model were compared with experimental results as shown in Fig. 11. Maximum damping forces were found for the MNS model and this damper was specially developed for seismic vibration control three-storied building structure.

To overcome the dynamics of a large MR damper, a variable current controller was then developed for the similar MNS model. The response time using the variable current MNS model showed an improved accuracy using RTHS [200]. The MNS model [175] has pre-yield and post-yield regions. The pre-yield and post-yield regions can be separated independently according to their behavior. The details of the MNS model can be found in Fig. 12 and Fig. 13 where \(x, y\) and \(z\) presents the degree of freedom responsible for damper deformation, pre-yield and post-yield region variables.

MR damper model [175].

The pre-yield region damping force behavior can be solved using Eq. (5) which is known as the Maxwell element model differential equation

\[
F = k(y - z) = cz
\]

(5)

where \(c\) and \(k\) are viscous and stiffness co-efficient.

When the damper is in pre-yield mode, \(y\) is equal to the damper velocity \(\dot{x}\). The initial value of \(y\) is set to be equal to \(x\); thus Eq. (5) can be solved in terms of \(z\) for a given \(x\) and the damper force is then determined. The values of \(c\) and \(k\) for the Maxwell element are obtained from the force–velocity relationship observed in damper characterization tests, selecting two appropriate points on the hysteretic response curve, and then applying visco-elasticity theory. Assuming the Maxwell element is subjected to a harmonic motion with an amplitude of \(u_0\) and circular excitation frequency of \(\omega\), the coefficients \(c\) and \(k\) are calculated from Eqs. (6) and (7) which are as follows:

\[
c = \frac{1}{u_0\omega} \left( \frac{f_0^2 + f_m^2}{f_0} \right)
\]

(6)
where \(f_0\) and \(f_m\) are the damper force when the damper velocity is zero and a maximum value, respectively. In the post-yield mode, \(\dot{x}\) defined as velocity. Post-yield curves are defined as the Herschel-Bulkley model \([201]\) and tangential curve velocity is \(\dot{x}^+\) or \(\dot{x}^-\). The mathematical model (Eqs. 8, 9, 10) can be written as follows:

\[
F_{\text{post}} = \begin{cases} 
    a_t(\dot{x} - \dot{x}^+) + f^+_t & \text{if } \dot{x} > \dot{x}^+ \\
    f^+_t & \text{if } \dot{x} < \dot{x}^+
\end{cases}
\]

where \(a, b, c\) are damper characterization parameters and \(a_t = b\dot{x}^+|x|^n\) and \(f^+_t = a + b|\dot{x}|^n\). However, in Eq. (9) the post-yield damping force \(F_{\text{post}}\) can be written as

\[
F_{\text{post}} = \left\{ \begin{array}{ll}
f_{py}(\dot{x}) & \text{if } \dot{x} \geq \dot{x}^+ \\
 f_{py}(\dot{x}) + m_0\ddot{x} & \text{otherwise}
\end{array} \right.
\]

where \(f_{py}\) and \(m_0\) are positive or negative force and mass acceleration which is predicted force by the MNS model. If the mode is changed from post-yield to pre-yield in the MNS model, then the equation can be written as

\[
\left| F_{\text{pre}} \right| = f_{py}(\dot{x})
\]

where \(F_{\text{pre}}\) is the pre-yield force.

### 5.1.3 Hyperbolic tangent function-based mono tube single-ended large MR damper

Based on hysteresis and linear function, Kwok et al. \([187]\) proposed the force of hyperbolic tangent function model where they analyzed viscous and stiffness of the MR damper (Fig. 14). To define the MR damper hysteretic force–velocity behavior, a strategy was deployed where a simple model is proposed here to model the hysteretic viscous damping (dashpot), spring stiffness and a hysteretic component as shown in Fig. 15. Equation (11, 12) shows the mathematical expression of the hyperbolic tangent function model \([187]\). The following equations are as follows:

\[
F = c\dot{x} + kx + az + f_d
\]

\[
\dot{z} = \delta\dot{x} - \beta\dot{x}|z|^n - \gamma z|\dot{x}|z|z|^{n-1}
\]

where \(a, \beta, \delta, \gamma, n\) are model parameters, \(c\) and \(k\) are the viscous and stiffness coefficients, \(z\) the hysteretic variable given by the hyperbolic tangent function and \(f_d\) is the damper force offset. This model is applicable for parameter identification and subsequent inclusion in controller design and implementation.

Figure 15b presents the component building hysteresis which describes force–velocity response using the effects of the parameter. The components building up the hysteresis are depicted in Fig. 15b which illustrates the effects of the parameters on the damper force–velocity response. The basic hysteretic loop, which is the smaller one is shown in Fig. 15b, which is determined by \(\beta\). This coefficient is the scale factor of the damper velocity defining the hysteretic slope. Thus, a steep slope results from a large value of \(\beta\). The scale factor \(\delta\) and the sign of the displacement determine the width of the hysteresis through the term \(\delta \text{ sign}(z)\), a wide hysteresis corresponds to a large value of \(\delta\). The overall hysteresis (the larger hysteretic loop shown in Fig. 15b is scaled...
by the factor $\alpha$ determining the height of the hysteresis. The overall hysteretic loop is finally shifted by the offset $f_d$.

After hyperbolic tangent function development, Gamota and Filisko [163] developed viscous and coulomb-based damping mechanisms and later Gavin [202] proposed a hyperbolic tangent model-based electro-rheological fluid damper. Bass and Christenson [203] developed a hyperbolic tangent model-based 200KN MR damper for structural vibration control where over-driven clipped optimal control (ODCOC) was used. Two simplified elements (spring-dashpot elements) constitute the hyperbolic tangent model as illustrated in Fig. 16. The following equations are as follows:

$$
\begin{align*}
\dot{x}_0 &= \left[\begin{array}{c}
0 \\
-k_0-k_1/m_i \\
-c_0-c_1/m_i
\end{array}\right] x_0 + \left[\begin{array}{c}
0 \\
k_1/m_i \\
c_1/m_i
\end{array}\right] \dot{x}_0 + \left[\begin{array}{c}
0 \\
1 \\
-1/m_i
\end{array}\right] F + F_y \tanh \left(\frac{\dot{x}_0}{V_{ref}}\right)
\end{align*}
$$

The inertial mass element resists motion employing a Coulomb friction element. The displacement and velocity of the inertial mass relative to a fixed base $x_0$, and $\dot{x}_0$, and displacement and velocity of damper piston end relative to the inertial mass, $x_1$ and $\dot{x}_1$, are summed together resulting in the displacement and velocity across the damper, $x$ and $\dot{x}$.

The pre-yield visco-elastic behavior is modelled by $k_1$ and $c_1$. The post-yield visco-elastic behavior is modelled by $k_0$.

Fig. 15 a Hysteresis model—component-wise additive approach and b hysteresis parameters [187]

Fig. 16 Hyperbolic tangent function-based dynamic model for a MR dampers [203] and b MR dampers [187]
and \( k_1 \). The term \( m_i \) represents the inertia of both the fluid and the moving piston. The parameter \( F_y \) is the yield force and \( V_{ref} \) is a reference velocity, which affects the shape of the transition from the elastic to the plastic region of the function. Figure 17 show the large MR damper fast hybrid test setup in three different floors.

The error for frequency-amplitude combination and error for larger current across the large MR damper is 17% and 5%, respectively. The hyperbolic tangent model was implemented to capture the silent behavior of a large MR damper. In another research studied hyperbolic tangent function-based large MR damper. In terms of convergence and stability, the hyperbolic tangent function model can run up to \( 12/1024 \) (0.012) s while the RMS error at \( 1/2048 \) (0.0005) s for the hyperbolic tangent model converges. The hyperbolic tangent model is much slower than other models except the Bouc-Wen model. The hyperbolic tangent shows better accuracy where RMS error at numerical time step equal to \( 1/1024 \) (0.001) \[204\]. The schematic of a large-scale semi-active damper shows in Fig. 18.

Phillips et al. \[51\] developed a large 596KN MR damper for building structure control using the hyperbolic tangent function model and four control strategies. The RTHS predicts the performance of large MR damper and force tracking controller found to be higher in performance. Equation (15) expresses the structural behavior of building which is as follows:

\[
m\ddot{x} + c\dot{x} + kx = GF - mL\ddot{x}_g
\]  

(15)

where, \( m, c, k, GL, F, x \) and \( \ddot{x}_g \) are the mass, damping, stiffness, influence vectors, force, displacement vector and ground acceleration.

### 5.2 Monotube large MR damper (double-ended)

#### 5.2.1 Modified Bouc-wen model-based Monotube Large MR damper (double-ended)

Yang and Cai \[205\] developed a mixed-mode control system using a 20KN capacity MRD 9000 (Fig. 19) \[206\] to attenuate the vibration of the suspension bridge generated from vehicle braking force and earthquake. A total of seven control strategies were investigated to get the maximum efficiency. A combination of semi-passive on control and fuzzy control strategies was analyzed that showed better performance on vibration reduction. Figure 19 shows the MR damper installed on bridge.

The damping force can be expressed in Eqs. (16, 17, 18) using the modified Bouc-wen model which can be written as

\[
F = c_\lambda \dot{y} + k_u (x - x_i) = az + c_\lambda (\dot{x} - \dot{y}) + k_u (x - y) + k_a (x - x_i)
\]  

(16)

where \( z \) and \( y \) are expressed by

\[
\dot{z} = -\gamma |z - \dot{y}| z|z|^{n-1} - \beta (\dot{x} - \dot{y})|z|^n + A(x - \dot{y})
\]  

(17)

\[
\dot{y} = \frac{1}{C_i + C_k} [az + c_\lambda \dot{x} + k_f (x - y)]
\]  

(18)

where \( c_\lambda \) is the viscous damping at large velocities, \( c_k \) is the viscous damping for force roll off at low velocities, \( k_u \) is the accumulator stiffness, \( k_f \) stiffness at large velocities, \( x_i \) spring
initial displacement and $A, \beta, \gamma,$ and $n$ are constant. But the application of this control system was limited to low vibration. During the excessive earthquake, this control system fails to protect the pier and bearing damage to the bridge.

To save the bridge members under excessive earthquakes a real-time semi-active control algorithm based on the damage of bridge members (RTSD) was proposed using a similar 20KN large MR damper [207] by Li et al.[208]. Figure 20 shows the MRF-04 K damper. This proposed model ensured that it can reduce the chance of damage to the bearing and pier more effectively and can set the damping force in a different range. The nonlinearity of the damper was measured using the modified Bouc-wen model.

5.2.2 Phenomenological Bouc–Wen model-based monotube large MR damper (Double-ended)

Yang et al. [209] developed a 200 KN large MR damper (Fig. 21) for structural vibration control using the Bouc-Wen model. They found higher damping force using a small amount of energy and quicker response time (damper coils) using parallel coil connection.

Sanwa Tekki Cooperation (Japan) [210] and Lord Corporation [133] jointly developed a 300 KN MR damper for seismic vibration control application while in 2003 they also developed a 400 KN MR damper for residential building applications [46, 47]. In 2003, 312 SD-1005 MR dampers were installed at Dongting Lake Bridge in Hunan and Ou et al. [48] developed several large MR dampers for Binzhou Yellow River Bridge China. Meanwhile, Binzhou Yellow River Bridge used several MR dampers for world longest cable-stayed bridge [211]. They used 6000 KN, 12 large MR dampers for vibration control.

In the Fig. 22a–d presents several large MR damper applications in suspension Fig. 22a–d also shows that large MR damper with control system and sensor network.

In 2011, Tu et al. [52] developed a sedimentation proof 500 KN large MR damper (Fig. 23) using a modified

![Fig. 20 Cross section of MRF-04 K damper [207]](image)

![Fig. 21 Schematic of the large-scale 200KN MR damper [209]](image)

![Fig. 22 a Dongting Lake Bridge-China (Large MR damper) [48], b, c Cable-stayed-Eiland Bridge (Netherlands) equipped with large MR damper [212] and d Earthquake-proof large MR damper [213]](image)
Bingham plastic model which was used for parameter identification.

Yang et al. [50] proposed a Bouc-Wen model-based double ended large-scale MR damper for structural vibration mitigation. Fluid inertial and shear-thinning effects were also analyzed using the Bouc–Wen model and it is found from the experiment that the current driven power supply is suitable for quicker response time. Cha et al. [214, 215] investigated the time delay of large MR damper using semi-active algorithms for 200KN MR damper to address robustness. Four types of control algorithms were used for semi-active control using the Bouc–Wen model where decentralized output feedback passive controller were more robust for time delay calculation than the clipped-optimal controller.

Bahar et al. [216] proposed a Bouc–Wen hysteresis model based large MR damper for real-time hybrid simulation using a parameter identification algorithm. They also studied large MR damper for benchmark building using parameter identification algorithm [217].

A similar large scale practical MR damper (Fig. 24) developed by Dyke et al. [218] and Rodríguez et al. [219] developed seismic vibration control MR damper using Bouc–Wen model and clipped-optimal control algorithm for real-time hybrid testing (RTHT) utilized for time delays and MR damper dynamics control [220].

Bouc–Wen model-based shear mode large MR damper developed for seismic vibration of a five-storied building where they used Bang-Bang, the Lyapunov and Clipped-Optimal controllers [221]. Other research groups developed MR damper integrated with base isolation system for large structure vibration control where they used Lyapunov controller [222].

Chen et al. [223] minimized actuator time delay using the CR algorithm and demonstrated the RTHS technique for experimental validation. Other research group used El-Centro, Kobe and Northridge seismic protection large MR damper numerical and experimental analysis investigation done by Bouc–Wen model and Clipped Optimal Control strategies. They found that the property of the damper can cope with normal natural frequencies and placement of the MR dampers were sensitive cases which include floor optimum location also [224]. The long term reliability of large scales MR dampers such as response time, dissipative capacity, control technique and force response are the critical point of MR damper applications in seismic vibration control [225].

A large MR damper RTHS was done for seismic vibration protection using the Bouc–Wen model where they used a semi-active neuro controller (SA-NC) and found that SA-NC is capable of reducing acceleration and displacement [226]. A similar SA-NC based study was proposed by Chae et al. [227] and Moon et al. [228].

The Bouc–Wen model-based MR damper was proposed by Spencer et al. [56], which is known as modified Bouc–Wen model. The Bouc–Wen model proposed by Bouc [229, 230] and later generalized by Wen [230] for MR dampers numerical investigations such as hysteresis behaviour. The damper force is given by Eq. (19) which can be written as:

\[ F - f_c = m\ddot{x} + c_o(\dot{x})\dot{x} + k_o x + az \]  

where the evolutionary variable \( z \) is governed by Eq. (20) which is as follows:

\[ \dot{z} = -\gamma|\dot{z}|^{n-1} - \beta|z|^n + A\dot{x} \]
In this model, \( m \) = equivalent mass which represents the MR fluid stiction phenomenon and inertial effect; \( k_a \) = accumulator stiffness and MR fluid compressibility; \( f_c \) = damper friction force due to seals and measurement bias; and \( c_p(x) \) = post-yield plastic damping coefficient.

To describe the MR fluid shear thinning effect which results in the force roll-off of the damper resisting force in the low-velocity region, the damping coefficient \( c(x) \) is defined as a mono decreasing function with respect to absolute velocity \(|x|\). The post-yield damping coefficient is expressed in Eq. (21). The post-yield damping coefficient can be written as:

\[
c(x) = a_1 e^{-a_2 |x|^p}
\]

where \( a_1, a_2, \) and \( p \) = positive constants.

Besides the proposed phenomenological model (Fig. 26), two other types of dynamic models (Fig. 27) based on the Bouc–Wen model are also investigated. One is the simple Bouc–Wen model with the mass element (Fig. 27a). Note that the damping coefficient is set to be a constant in this model. The other one is the phenomenological model [56] with the mass element (Fig. 27b). To assess their ability to estimate the MR damper behaviour, these three dynamic models are employed to fit the damper response under a 1 in., 0.5 Hz sinusoidal displacement excitation at an input current of 2 A. As can be seen, all models can describe the damper force–displacement behaviour very well. However, the simple Bouc–Wen model fails to capture the force roll-off in the low-velocity region. The damping force is shown in Eq. (22) and the damper force is as follows:

\[
F(t) = m\ddot{x} + c\dot{x} + k_s(x - x_0) + az
\]

where \( \dot{x} \) is the velocity of the piston, \( c \) is the damping coefficient and \( k_s \) is the linear spring constant.

### 5.2.3 Phenomenological Dhel friction model based

monotube large MR damper (Double-ended)

Dhel friction model [231] was developed by Dahl [231] to characterize the frictional behaviour and a differential equation was used for stress–strain curve modeling. Let \( x \) be the displacement, \( f_c \) the friction force and \( F_c \) the Coulomb friction force. Figure 28 presents the typical solid friction force function.

Solid friction mathematical model (Eq. 23), in terms of time rate of change of solid friction can be written as

\[
\frac{dF(x)}{dt} = \frac{dF(x)}{dx} \frac{dx}{dt}
\]

where \( F(x) \) is a solid friction force (function of displacement \( x \)). When \( x \) is positive then friction force will be \( +F_c \) and in case of reverse force, \( x < 0 \) and \( F(x) \) will be negative that is \(-F_c\). Though \( x \) changes then the friction function slope \( \frac{dF(x)}{dx} \), remains positive. The friction slope functions can be expressed from Eqs. (24, 25, 26, 27) and will be simulated.

---

**Fig. 26** The proposed phenomenological model of MR dampers [50]

**Fig. 27** Two other types of phenomenological models of MR dampers based on Bouc–Wen hysteresis model [56]
with hysteresis behavior. The following equations are as follows:

\[
\frac{dF(x)}{dx} = \sigma |1 - \frac{F}{F_c}| \text{sgn}(1 - \frac{F}{F_c}) \text{sgn} \dot{x}
\]  

(24)

For positive velocities \( \text{sgn} \dot{x} = +1 \), then the dimensionless ratio \( r = \frac{F}{F_c} \)

\[
r = 1 - [1 - (1 - i)u]^{1/1 - i}
\]

(25)

With

\[
u < \left( \frac{1}{1 - i} \right) \text{ for } i > 1
\]

(26)

where \( u = \frac{x}{x_c} \) is a dimensionless displacement variable, and \( x_c \) is a characteristics displacement which can be written as

\[
x_c = \frac{F_c}{\sigma}
\]

(27)

The modified Dahl model proposed by Zhou and Qu [162] is shown in Fig. 29. This model is used to simulate Coulomb force and avoid too many parameters. The damping force can be written as:

\[
F = kx + C_i \dot{x} + F_d Z - f_k
\]

(28)

where \( k, C_i, F_d, x, f_k \) and \( Z \) are stiffness, damping coefficient, Coulomb force modulated by the applied magnetic field, displacement of MR damper, damper force caused by seals and measurement bias and nondimensional hysteretic variable governed by [231] the following equation:

\[
\dot{z} = \sigma \dot{x}(1 - Z \text{sgn}(\dot{x}))
\]

(29)

where \( \text{sgn} \) determines hysteretic loop shape and \( \sigma \) is the rest stiffness or slope of the displacement curve.

After modified Dahl model, Ikhouane and Dyke [182] developed a viscous Dahl model for the shear mode MR damper (Figs. 30, 31).

The viscous dry friction model for MR dampers can be written as

\[
F(t) = cx + \alpha z
\]

(30)

Here \( z \) is a no dimensional hysteretic variable and the constants \( \alpha \) and \( c \) depend on voltage.

---

**Fig. 28** Typical solid friction force function [231]

**Fig. 29** Modified Dahl model of MR damper [162]

**Fig. 30** Shear mode MR damper [182]

**Fig. 31** Viscous + Dahl model for the MR damper [182]
Using the viscous Dahl model, Rodriguez et al. [232] proposed a large MR damper for vibration mitigation using Bouc-Wen and Dahl frictional model [233]. The proposed model verified the viscous term which was smaller than hysteresis one and modified identification technique. They found Dahl friction model [233] generated higher error than Bouc–Wen model. Bouc–Wen model was more suitable for large MR fluid damper modeling. Dahl friction model was used for three-storied building vibration reduction while for larger storied was not considered. The issue of high payload, particle sedimentation and magnetic flux distribution was not considered also. Jiang and Christenson [204] investigated the Dahl friction model using Aguirre et al. [234] viscous plus Dahl model. It was found from the RTHS that the Dahl friction model is more sensitive during the change of numerical integration time step than algebraic model and viscous plus Dahl models follows the simpler equations modeling the force behavior.

5.2.4 Bingham model-based Monotube Large MR damper (double-ended)

Kui et al. [235] proposed a large 1400 N MR damper to mitigate the unwanted pipeline vibration using Bingham plastic non-linear fluid model and linear quadratic regulator (LQR) control algorithm. The use of an LQR control system and magnetism insulator ensures the high magnetic flux density that results in high damping performance. Figure 32 shows the 3D model of the MR damper. The damping force is given in Eq. (32). The damping force using the Bingham plastic non-linear model can be expressed as

$$ F = F_\eta + F_\tau + F_f = \frac{12\eta l A_e^2}{\pi D^3 h^3} \dot{u}_r + \left( \frac{3l \tau_y A_e}{h} + F_f \right) \text{sgn}(\dot{u}_r) $$

(31)

where, $F_\eta$ is the shear stress, $F_\tau$ is the viscous stress, $\eta$ is the dynamic viscosity of MR fluid, $h$ is the width of damping gap, $A_e$ is the effective area of the piston, $l$ is the effective length of the piston, $\dot{u}_r$ is the relative velocity of the piston and cylinder and $D$ is the mean diameter of the damping gap. The dynamic range of MRF damper can be written as

$$ \beta = \frac{F_\eta + F_\tau + F_f}{F_\eta + F_f} = 1 + \frac{F_\tau}{F_\eta + F_f} $$

(32)

5.2.5 Herschel-Bulky model-based Monotube Large MR damper (double-ended)

A semi-active control system incorporating a large 200KN MR damper was proposed by Peng and Zhang [236] to understand the full operating environment of the system for control structure. The Herschel-Bulky model was employed to understand the MR fluid characteristic. The simulation results match well with the experimental data. Figure 33 shows the MR damper.

The features of Herschel-Bulky model (Eqs. (33, 34) defines both Bingham plastic model and power law model. The rheological behavior of MR fluids using Herschel-Bulkey model can be written as

$$ \eta = \frac{\tau_y}{\dot{\gamma}} + k_c \left( \frac{\tau_y}{\dot{\gamma}} \right)^{n-1} $$

(33)

where $\eta, k_c, \tau_y, n$ and $\dot{\gamma}$ are the dynamical viscosity, consistency index, yield shear stress, power law index and strain rate.

Finally, the Herschel-Bulkey fluid model [237] can be written as

$$ \tau = \left( \tau_y(H_0) + K|\dot{\gamma}|^{\frac{1}{n}} \right) \text{sgn}(\dot{\gamma}) $$

(34)

where $H_0$ is the strength of the magnetic field.
5.2.6 Double sigmoid model-based Monotube MR damper (double-ended)

A semi-active control system base MR damper is proposed by Ji et al. [158] to reduce the low-frequency vibration in the pipeline by introducing three different control modes. Results showed that the sliding mode variable structure control mode had better vibration reducing proficiencies than PID control at high frequency level. A double-sigmoid model was developed to express the damping force of the MR damper shown in Eq. (35) which can be written as

\[ F = F_c \frac{1 - e^{-a(x-x_v,\text{sign}(\dot{x}))}}{1 + e^{-a(x-x_v,\text{sign}(\dot{x}))}} + C_i \dot{x} \]  

(35)

where, \( F_c \) is the adjustable coulomb damping force, \( x_v \) is the displacement value when damping force was zero, \( \dot{x} \) and \( \ddot{x} \) are relative displacement and velocity of MR damper piston and cylinder, \( C_i \) is the viscous damping coefficient and \( a \) is the velocity adjustment co-efficient of coulomb damping force.

5.3 Twin tube large MR damper (single-ended)

5.3.1 Bingham model-based twin tube large MR damper

Zolfagharian et al. [238] developed an unsteady analytical model combined with quasi-static analysis and experimentally investigated the MR fluid flow behavior through the piston annular channel of a twin-tube MR damper. Figure 34 presents single ended twin tube large MR damper structure. The result showed that the new unsteady analytical model can measure the phase difference more effectively than other models which also results in higher damping force. The non-Newtonian fluid characteristic was described through the Bingham plastic model, where the developed shear stress (Eq. 36) can be written as

\[ F_\tau = \tau_y (H) \text{sign}(\dot{\gamma}) + \eta \dot{\gamma} \]  

(36)

where, \( F_\tau \) is the shear stress, \( \tau_y \) is the yield stress, \( H \) magnetic field amplitude, \( \dot{\gamma} \) shear strain rate and \( \eta \) is the viscosity of MRFs.

5.4 Twin tube large MR damper (double-ended)

5.4.1 Phenomenological Bouc-wen model-based twin tube large MR damper (double-ended)

A new phenomenological model was proposed by Spencer et al. [56] and applied by Wang et al. [239] to improve the long-term operation capability of the MR damper by analyzing the mechanical characteristic of the dampers which were in operation for a long time in cable bridge. Figure 35 presents schematic of twin-tube large MR damper. The modified model is shown in Fig. 36. The final damping force (Eq. 37) of the model can be written as
where \( c_b \) is used to model the roll off phenomenon of MR damper at low motion velocities, \( k \) is the stiffness of the accumulator, \( x \) is the displacement of the piston, \( x_i \) is the initial displacement of the spring and \( A_1 \) and \( A_2 \) are the modified co-efficient for the bottom right part and top left part of the displacement damping force loop.

The experimental results showed that the used dampers were a lack in efficiency due to the leakage problem of MR fluid and the new proposed model had a better effect on the mechanical properties of the dampers.

6 Control algorithm strategies of large MR dampers

Several control strategies were taken last decades to minimize response time, time delays, dissipative energy capacity, force responses, robustness and excessive cost etc. The control techniques of large MR dampers are passive, active, and semi-active [46]. Passive control techniques are used in base-isolators, elastomeric and frictional dampers, and tuned-mass dampers while active control systems are used in active bracing/tendon systems, active-mass drivers, and active variable-stiffness devices [240]. A semi-active control system combines both passive and active control strategies which is especially used in large force requirements using lesser power [241]. Semi-active device used in variable friction/stiffness dampers and controllable-fluid dampers (electrorheological (ER) and MR (MR) fluid dampers) [242]. The semi-active control methods are model-based control and soft computing-based control. Model-based control techniques are bang–bang control, back-stepping control, sliding mode control, \( H_2 \) and \( H_\infty \) control, adaptive/non-linear control, and bilinear control while soft computing-based control are neural network-based control, fuzzy logic control, and genetic algorithm-based control [53, 214, 243].

6.1 Skyhook control algorithm

Karnopp et al. [244] proposed a ‘skyhook’ damper control algorithm (Fig. 37a) for a vehicle suspension system [135,
245]. An MR damper [246] with skyhook control system for vehicle suspension system is shown in Fig. 37b.

The skyhook control law can be written as.

\[ F_b = +c(\dot{x} - \dot{x}_0) + +k(\dot{x} - \dot{x}_0) \]  
\[ F_b = c\dot{x} + k(x - x_0) \]

Where \( F_b \) is the control force, \( c \) and \( k \) viscous and stiffness co-efficient.

### 6.2 Decentralized bang-bang control

Other research group such as McClamroch and Gavin [247] proposed decentralized bang-bang control law using the Lyapunov control algorithm. They reported that this control system is accurately working for ER dampers application with maximum and minimum dissipation rate. The control law can be represented as

\[ v_o = V_{max}H(\dot{X}^T \lambda f) \]

where \( v_o \) is the input voltage to the current driver, \( V_{max} \) is the maximum allowable voltage and \( H \) is the Heaviside step function.

### 6.3 Clipped-optimal control (COC)

Acceleration feedback-based Clipped-optimal control (COC) (Fig. 38) was proposed by Dyke et al. [248] to overcome the full-state feedback or on velocity feedback control system. Accelerometers based COC can provide a reliable and inexpensive solutions. COC algorithm needed to design a linear optimal controller \( K_c \), which will provide control force \( F_b \) based on measured response \( y \) i.e.:

\[ F_b = L^{-1}\left( -K_c(s) \dot{y} \left( \begin{array}{c} y \cr f \end{array} \right) \right) \]

where \( L \) is Laplace transform, \( f \) is the measured force, \( \dot{y} \) is measured output vector, \( v_i \) is the measured noise vector and the control law can be written as

\[ v_i = V_{max}H\left( [F_b - f]f \right) \]

where \( V_{max}, H \), are the voltage to the current driver related to the saturation of the magnetic field in the MR damper and Heaviside step function.

Heo et al. [249] proposed an MR damper (Fig. 39) using clipped optimal control system for a cable stayed bridge to control seismic vibration.

### 6.4 Homogeneous friction controller

Inaudi [251] proposed a Homogeneous friction controller for semi-active control of structures. This controller system is also known as modulated homogeneous friction (MWF) controller. This proposed controller shows quadratic dissipation of energy per cycle in the deformation amplitude, maximum dissipation efficiency for resistance-force level proportional to deformation, and simple and accurate linearization. In addition, a modified type of modulated homogeneous friction controller proposed by He et al. [252] that is capable of increasing the performance of MR dampers. The proposed control law is shown in Eqs. (43, 45, 46) which can be written as

\[ N(t) = g|P[\Delta(t)]| \]
where, $N(t)$ is controllable contact force, $\Delta(t)$ is damping deformation, $\mu$ is coefficient of friction, $g$ is the positive gain coefficient and $\Delta(t - s)$ is local peak of deformation signal.

### 6.5 Semi-active control algorithms

Xu et al. [199] proposed semi-active control algorithms which is based on neural networks applied for MR dampers structures. The control algorithm can be written as [253]

$$M_s \ddot{x} + C_s \dot{x} + K_s x = \lambda F_b - M_s \Gamma \ddot{x}_g$$

where, $x$ is the vector of relative displacement of the floors of the structure, $\dot{x}_g$ is one-dimensional ground acceleration, $F_b$ is measured control force, $\Gamma$ is column vector of ones, $\lambda$ is the vector determined by the position of MR damper. An MR damper with semi active control system [199] based on neural network is shown in Figs. 40 and 41.

### 6.6 Quasi-bang-bang control algorithm

The quasi-bang-bang control algorithm (Eq. 48) proposed by Barroso et al. [254] for MR dampers structures and proposed controllers considered static equilibrium conditions. The equation can be written as follows [255]

$$V_i = \begin{cases} V_{\text{max}} & \text{if moving away from the center} \\ 0 & \text{if moving towards the center} \end{cases}$$

where $V_{\text{max}}$ is the maximum voltage.
6.7 Lyapunov control theory

To provide higher performance Spencer and Nagarajaiah [46] proposed Lyapunov control theory-based damping control system for MR dampers. The control law (Eq. 49) for Lyapunov control theory can be written as

\[ c(t) = \begin{cases} c_{\text{max}}(\omega_n u_0 + \dot{u}_r) \dot{u}_r > 0 \\ c_{\text{min}}(\omega_n u_0 + \dot{u}_r) \dot{u}_r < 0 \end{cases} \]  

(49)

where \( c_{\text{max}} \) and \( c_{\text{min}} \) are maximum and minimum damping coefficients, respectively, and \( \omega_n = \sqrt{k/m} \) and \( u_0(t) \) is the absolute displacement of the single degree of freedom (DOF) and \( \dot{u}_r \) is the relative velocity.

A Lyapunov control system-based MR damper [256] and the control block diagram is shown in Figs. 42 and 43.

6.8 Decentralized Output feedback polynomial controller (DOFPC)

Cha and Agrawal [257] investigated decentralized output feedback polynomial controller (DOFPC) for both active and semi-active controls of the highway suspension bridge. The control strategy is expressed in terms of velocity and displacement across MR dampers using 3rd order polynomial equation. The equation can be written as

\[ v = \left( q_0 + q_1 x + q_2 x^2 + q_3 x^3 \right) + \left( r_o + r_1 \dot{x} + r_2 \dot{x}^2 + r_3 \dot{x}^3 \right) \]  

(50)

where \( v \) is the control signal, \( x, \dot{x} \) and \( \dot{\dot{x}} \) are the inter-story drift and inter-story velocity, respectively. \( q_0, q_1, q_2, q_3, r_o, r_1, r_2, r_3 \) are optimal co-efficient of the polynomial equation for control signal.

6.9 Maximum energy dissipation controller

Jansen and Dyke [176] proposed maximum energy dissipation controller for six story building using MR dampers and considered Lyapunov controller. Maximum Energy Dissipation Controller specialized for multi-input control system. The equation can be written as [255]

\[ V(z) = \frac{1}{2} ||z||_P^2 \]  

(51)

where, \( V(z) \) is Lyapunov function, \( ||z||_p \) = \( p \) norm of the state and \( P = \) real symmetric, positive define matrix.

6.9.1 Simple-passive control (SPC)

Zhang [258] proposed simple-passive control (SPC) system for seismic MR damper where zero-displacement positions are available. MR damper can cope with large control force with its zero-displacement position. The simple-passive controller formulation can be written as

\[ V_b = \begin{cases} v_1 & |x| < x_1 \\ v_2 & x_1 \leq |x| < x_1 + x_2 \\ v_3 & x_1 + x_2 \leq |x| < x_1 + x_2 + x_3 \\ 0 & x_1 + x_2 + x_3 \leq |x| \end{cases} \]  

(52)

where \( V_b \), \( x \) are the control voltage to the ith MR damper and inter-story displacement. \( x_1, x_2, x_3, v_1, v_2 \) and \( v_3 \) are the design parameter which can be determined by optimization process.

6.9.2 Back-stepping control

Back-stepping control provides higher performance and accuracy which was proposed by Zapateiro et al. [259] for the vehicle suspension system. Primarily Back stepping Controller used Dahl model and a proposed Back-stepping Controller for seismic protection and vehicle neural network for MR dampers. The neural network can achieve inverse
dynamics or reproduce using Back stepping Controller in the MR damper. A back-stepping technique-based MR damper is shown in Fig. 44 [260]. The control law is shown in Eqs. (53, 54, 55). The following equation can be written as

\[
F = - \frac{(a_1 - a_2)(1 + h_1 h_2) + (h_1 + h_2) \omega_2 + f}{g} 
\]

\[
F = \bar{k}_x \dot{x} + [k_{wa} + k_{wb} V_b] w 
\]

\[
V_b = - \frac{(a_1 - a_{eq})(1 + h_1 h_2) + (h_1 + h_2) \omega_2 + f - g(k_x x + \dot{k}_{wb} w)}{k_{wb} \omega_2} 
\]

where, \( a_1, a_2 \) are the angular position, \( F \) is the generated damping force, \( \omega_2 \) is the angular speed of the upper lever, \( V_b \) is the control voltage and \( \bar{k}_x, k_{wa} \) and \( k_{wb} \) are hysteresis loop controlling parameter.

6.9.3 Sliding mode controller

The sliding mode controller (Fig. 45) was used for driving the response trajectory along with a sliding surface [261] and particularly applied in nonlinear and hysteretic structures while several research group used for MR/ER dampers (Fig. 46) for seismic structures [262, 263]. The required equation can be written as

\[
s = \dot{e}_s(t) + \Phi e_s(t) 
\]

\[
e_s(t) = x_s(t) + x_{s0}(t) 
\]

\( x_{s0} \) is sprung mass displacement, \( e_s \) is sprung mass displacement error, \( x_s \) is the vertical displacement and \( \Phi \) is convergence rate of sliding mode control.
6.9.4 Non-linear closed-loop controller

Kane et al. [266] proposed non-linear closed-loop controller for MR damper structures. The advantages of nonlinear controller over linear controller are controlling the dynamic force saturation limit and agent-based control structure. The control equation can be written as

\[
\dot{R}_p(t) = A_p R_p(t) + B_p u_p(t) + d(t)
\]

(59)

\[
y_p(t) = C_p R_p(t) + d_q(t)
\]

(60)

\[
\dot{R}_m(t) = A_m R_m(t) + B_m u_m(t)
\]

(61)

\[
y_m(t) = C_m R_m(t)
\]

(62)

where, \(A_p, A_m\) are state matrices, \(B_p, B_m\) are input matrices, \(C_p, C_m\) are the output matrices, \(R_p, R_m\) are the n \times 1 plant state vector \(n_m \times 1\) model state vector, \(y_p\) is plant output, \(y_m\) is the model output, \(u_p\) is the \(m \times 1\) input control vector, \(u_m\) is the \(m \times 1\) input command vector and \(d_i\) and \(d_o\) are the input and output disturbances.

6.9.5 A non-linear/adaptive control

A non-linear/adaptive control (Fig. 47) was proposed by Bitaraf et al. [267] after combining the study of simple adaptive control method [268] and genetic-based fuzzy control method. An adaptive control-based MR damper [269] for seat suspension is shown in Fig. 48. A nonlinear or adaptive controller can control both displacement, acceleration and response time effectively. Fuzzy logic control method is a combination of several control methods such as sliding mode and genetic algorithm control or combined method of fuzzy logic and neural network where neural network models [270] and black box model [271] are used. Fuzzy logic algorithm was proposed in civil structure [272–275] and MR dampers modeling [276–279]. Equation (59–62) shows the control law of the system. The equation are as follows:

\[
\sum_{i=1}^{N_i} \mu_{P_i}(Z_i^{(m)}) \mu_{P_j}(Z_j^{(m)})
\]

(58)

where \(W_j(Z_j^{(m)}) = \mu_{P_j}(Z_j^{(m)})\) is the grade of membership of \(Z_j^{(m)}\) in \(P_{ij}\), \(A_j, B_j\) are system matrices and \(R\) is state vector.

7 Conclusion

Large MR dampers have been developing for large vibration control systems. The review presents different structural design, mathematical models, their applications, classifications and different control system used for large MR dampers. Large MR dampers are developed by modifying the conventional MR dampers both in internal and external design structure. The main feature of a large MR damper over normal MR damper is its higher damping force and large frequency vibration control capability. The mono tube large MR damper with single ended and double ended structure are mostly developed over the years due to its cost-effectiveness and availability while the twin-tube large MR damper are less developed.

Among different mathematical model, the phenomenological bouc-wen model-based mono tube MR damper was mostly developed due to its fast response time, resistance to particle sedimentation, lower energy consumption and effective in both low and high frequency vibration control system. Whereas the Phenomenological Dhel friction model-based monotube large MR damper had more error in efficiency and not suitable for large scale MR damper modeling due to its high sedimentation, low magnetic flux distribution problems.

It is essential to react fast when subjected to large frequency vibration in case of large vibration mitigation. Large MR damper requires faster response time and better reliability for long-term large vibration control system. Among different control system the non-linear or adaptive control system was proved to be the most effective in case of better response, displacement and acceleration control. It is a combination of adaptive control and fuzzy control method which
are used for many civil engineering applications where large damping force requires.

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