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What is This?
Magnetorheological fluid dampers: A review on structure design and analysis

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Abstract
Magnetorheological fluid technology has gained significant development during the past decades. The application of magnetorheological fluids has grown rapidly in civil engineering, safety engineering, transportation, and life science with the development of magnetorheological fluid–based devices, especially magnetorheological fluid dampers. The magnetorheological fluid dampers could offer an outstanding capability in semiactive vibration control due to excellent dynamical features such as fast response, environmentally robust characteristics, large force capacity, low power consumption, and simple interfaces between electronic input and mechanical output. To address the fast growing demand on magnetorheological fluid damping technology in extensive engineering practices, the state-of-the-art development is presented in this article, which provides a comprehensive review on the structure design and its analysis of magnetorheological fluid dampers (or systems). This can be regarded as a useful complement to several existing reviews in the recent literature on magnetorheological fluids technology, magnetorheological fluid applications, modeling of magnetorheological fluids and dampers, control strategies of magnetorheological fluid systems, and so on. The review begins with an introduction of the basic features and relevant applications of magnetorheological fluids. Then several basic structure design issues of magnetorheological fluid dampers are introduced. Following this, typical magnetorheological dampers are discussed according to the arrangement configurations of magnetorheological fluid cylinders and magnetorheological fluid control valves. Furthermore, reinforced structure designs of magnetorheological fluid dampers are provided, which focus on coil configuration, fluid resistance channel design, and electromagnetic design. Thereafter, design issues of magnetorheological fluid damper systems are discussed, which involves sensor-based magnetorheological fluid damper systems, self-powered magnetorheological fluid damper systems, fail-safe magnetorheological fluid damper systems, and integrated spring magnetorheological fluid damper systems. Importantly, to have a systematic quantitative viewpoint of the analysis and design of magnetorheological fluid dampers, the review ends with a summary of performance analysis issues, including performance specification, analytical modeling, parameter optimization, and so on.

Keywords
magnetorheological fluids, structure design, dampers

Introduction
Magnetorheological fluids (MRFs) are suspensions of magnetically responsive particles in a liquid carrier, which is characterized as a class of “smart fluid” for its capability of producing MR effect widely applied in vibration control. The MR effect is primarily observed as a significant change of the yielding shear stress of MRFs continuously controlled by the intensity of applied magnetic fields (Carlson, 2009; Grunwald and Olabi, 2008). Concretely, MRFs have the ability of changing from a free-flowing liquid state into a semisolid condition with restricted fluid movement in fast response (several milliseconds) when exposed to an external magnetic field. In free-flowing liquid state (off-state), MRFs exhibit field-independent behavior with the plastic viscosity controlled only by the viscosity of the liquid vehicle and particle volume fraction. In the semisolid state (on-state), MRFs exhibit field-dependent behavior characterized by the variable yield stress dependent on the magnetic field strength. Only when the shear stress exceeds the yield shear stress,

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MRFs come back to free flow state with nearly unchanged plastic viscosity (Carlson, 2009; Goncalves and Carlson, 2007; Goncalves et al., 2006). In addition, the power consumption for on-state MRFs is not a concern since MRFs are able to reach high magnetic field–induced yield stress (50 kPa or more) with a magnetic field that can easily be generated by an electromagnet operated at low voltage and modest current (2–24 V and 1–2 A) (Carlson, 2009; Olabi and Grunwald, 2007).

During the past two decades, MRFs technology has gained significant developments in different aspects from the manufacture of MRF compositions, including metal particles, basic fluid or carrier fluid, and stabilizing additives (Vicente et al., 2011; Fuchs et al., 2010; Roszkowski et al., 2008) to performance evaluation and fast growing applications of innovative MRF-based devices in civil engineering, safety engineering, transportation, life science, and so on (Ahn et al., 2005; Chen, 2009; Chen and Liao, 2010b; Goncalves, 2005; Goncalves et al., 2006; Hietanen, 2002; John et al., 2008; Jolly et al., 1999; Jung et al., 2004; Kim et al., 2009; Liu et al., 2006; Nosse and Dapino, 2007; Rabinow, 1948; Song and Zeng, 2005; Thanh and Ahn, 2006). Noticeably, one of the most important applications of MRFs is development of MRF dampers or shock absorbers, in which MRFs are controlled by a magnetic field to allow the damping characteristics of the device to be continuously adjusted by varying the power of the electromagnet (Poynor, 2001). MRF dampers could offer an outstanding capability applicable in active/semiactive vibration control systems owing to their inherent advantages from MRF technology such as contactless and quiet fluid control without moving parts, large force capacity and large controllable range, high insensitivity to contaminants or impurities, simple interfaces between electronic controls and mechanical devices with rapid response, and low power requirements (Kciuk and Turczyn, 2006). Nowadays, more and more MRF dampers have been developed and applied successfully in different vibration control systems, such as vehicle suspension system (Hiemenz et al., 2006). Noticeably, the development of MRF damper systems with more complicated functions such as fail-safe and self-powered are presented. Section “Performance analysis and parameter optimization of MRF damper systems” summarized some important performance analysis and parameter design issues. A conclusion is given in the last section “Conclusion.”

**Basic structure design issues of MRF dampers**

Basically, for different application purposes, MRF dampers could be designed with different structural functions and configurations in the following fundamental aspects: (a) operational modes of MRFs, (b) MRF cylinder, and (c) MRF control valve.

**Operational modes of MRFs**

The MRFs have three basic operational modes: flow modes (i.e. valve mode), direct shear mode (i.e. clutch mode), and squeeze mode, as shown in Figure 1. The valve mode involves the fluid flowing as a result of a pressure gradient between two stationary plates, which is used in hydraulic controls, servo valves, dampers, shock absorbers, and actuators. The direct shear mode involves the fluid located between two plates that move relatively, which is used in clutches, brakes, chucking and locking devices, dampers, breakaway devices, and structural composites. The squeeze mode involves the fluid running between two plates moving perpendicular to their planes, which has less application and is only used in small-amplitude vibration and impact dampers (Carlson, 2009; Carlson and Jolly, 2000; Hoyle et al., 2010; Kim et al., 2008; Olabi and Grunwald, 2007; Sung et al., 2005; Tse and Chang, 2004; Wereley et al., reviews about MRF technology (Goncalves et al., 2006; Vicente et al., 2011), MRF applications (Muhammad et al., 2005; Wang and Meng, 2001), modeling of MRFs and dampers (Wang and Liao, 2011), and control strategies of MRF systems (Jung et al., 2004) have been presented recently. To have a clear insight into the structure design of MRF damper systems, this article presents a state-of-the-art review on structure design issues of MRF damper systems, including basic mechanical structure designs, reinforced structure design methods, structure design issues for supplemental functions, and several additionally important design considerations in performance analysis and parameter design.

The organization of this article is as follows. The basic structure designs of MRF damper are addressed in section “Basic structure design issues of MRF dampers.” Section “Reinforced structure designs for performance improvement” presents some reinforced structure design methods for better damping performance. In section “Design of MRF dampers as a system,” the development of MRF damper systems with more complicated functions such as fail-safe and self-powered are presented. Section “Performance analysis and parameter optimization of MRF damper systems” summarized some important performance analysis and parameter design issues. A conclusion is given in the last section “Conclusion.”
2008). Besides, multiple fluid modes or mixed-mode MRF devices that simultaneously employ at least two basic operational modes have alternatively been developed to enlarge available damping force and simple manufacture requirements of MRF damper (Brigley et al., 2007; Choi et al., 2008; Hong, Wereley, Choi, et al. 2008; Unsal, 2006; Yu et al., 2009). Recently, an alternative mode—magnetic gradient pinch—is developed for controlling MRFs (Goncalves and Carlson, 2009).

**Basic assembly components**

In view of mechanical structure and magnetic circuits, the basic assembly components in the design of an MRF damper involve the MRF cylinder and MRF control valve. The MRF cylinder is usually filled with MRFs and separated by a moveable piston for outputting mechanical motion. The MRF control valve is employed to produce the damping effect under the controllable magnetic fields using electromagnets.

**MRF cylinders**

**Monotube structure.** The fundamental and typical structure of a monotube passive damper is shown in Figures 2 and 3. The monotube MRF damper is basically based on a single-rod cylinder structure, which has only one reservoir for the MRF and the reservoir is divided into extension chamber and compression chamber by a moving piston (Poynor, 2001). During piston movement, MRFs in the cylinder pass through the MRF control valve that is assembled in the piston, which results in an apparent change in viscosity of MRFs, causing a pressure differential for the flow of fluids and consequently generating damping force proportional to the controllable magnetic field. An accumulator with the compressed gas (usually nitrogen) that is separated from MRFs by a floating piston or bladder has the following three functions. First, the accumulator is used to accommodate volume change of incompressible MRFs due to piston movement. Second, the accumulator can provide a pressure offset so that the low pressure side of the MRF control valve is not reduced to cause cavitations of MRFs. Third, the gas chamber adds a spring effect to the force generated by the damper simultaneously and maintains the damper at its extended length when no force is applied (Ebrahimi, 2009; Reichert, 1997). In addition, the bearing at the end of the extension chamber is utilized to guide movements and the corresponding seals for
preventing leakage of MRFs. Generally, the monotube damper has the advantages of simpler mechanical structure in terms of manufacturing and being lighter due to fewer parts and has the disadvantages of having higher gas pressure (normally more than 10 bar) and having a cylinder that is more susceptible to damage compared to that of the twin-tube damper (Ebrahimi, 2009).

**Twin-tube structure.** The basic structure of a twin-tube passive damper is shown in Figure 4, and an example is shown in Figure 5. The twin-tube MRF damper has an inner and an outer housing. The inner housing is filled with MRF and guides the piston/piston rod assembly just as the housing of a monotube damper does. The outer housing, which is partially filled with MRFs, accommodates volume changes due to piston movement fulfilling the same purpose as the pneumatic accumulator mechanism in monotube dampers. Moreover, the outer housing provides additional functions, including protection of the damper’s internal parts and transferring heat from the damper fluid to the surroundings. Note that a valve assembly called “foot valve” (Figure 6) attached to the bottom of the inner housing is employed to regulate the flow between the two reservoirs during extension and compression movement. As the piston rod enters the damper, the MRF flows from the inner housing into the outer housing through the compression valve in the “foot valve.” The amount of fluids that flow from the inner housing into the outer housing is equal to the volume displaced by the piston rod as it enters the inner housing. As the piston rod is withdrawn from the damper, the MRF flows into the inner housing through the return valve in the “foot valve” (Ebrahimi, 2009; Esfaminasab, 2008; Pynor, 2001; Sassi et al., 2005). Generally, the twin-tube dampers can work with lower gas pressure (less than 10 bar) but are more complex and have problems in dissipating the generated heat (Ebrahimi, 2009).

**Double-ended structure.** A double-ended structure is a special configuration from the monotube structure, which is shown in Figure 7. As seen, the piston rod has the same diameter and protrudes through both ends of the double-ended MRF damper. This arrangement does not need a rod-volume compensator to be incorporated into the damper, thus the gas chamber can be removed, and no spring effect could be generated by itself, although a small-pressurized accumulator may be provided to accommodate thermal expansion of fluids (Pynor, 2001; Yang et al., 2002). Double-ended MRF dampers have been used for impact and shock loading, gun recoil applications, and seismic protection in structures (Jolly et al., 1999; Norris, 2003; Yang et al., 2002).
MRF control valves

An MRF control valve is used for generating magnetic circuits, which achieve electric control of mechanical damping forces without mechanical moving parts. Figure 8 shows a typical MRF control valve incorporated in a moving piston of an MRF damper. As seen, the MRF control valve comprises a magnetic housing (flux ring), a magnetic core, a nonmagnetic bobbin that winds an induction coil, and a flow channel located between the outside of the magnetic core and the inside of the magnetic housing connected to a fluid inlet port and an outlet port (Carlson, 2009; Wang et al., 2009). The magnetic circuit guides the magnetic flux through the active part of the MRF control valve, and the dynamic yield stress of the MRF is changed with the magnetic field intensity generated by the electromagnetic coil. Consequently, the resultant damping force is activated against motion under the controlled magnetic field intensity. In the absence of a magnetic field, the damping force occurs only due to the viscosity of MRFs itself (Boese and Ehrlich, 2010; Lee et al., 2009). Note that the magnetic housing and magnetic core typically use low-carbon steel, which has a high magnetic permeability and saturation and can effectively guide magnetic flux into the fluid gap (Unsal, 2006). In addition, bushes and washers (monocast (MC) nylon) are employed for preventing leakage of magnetic fields (Lee et al., 2009).

The magnetic coil is laid inside the MRF cylinder above, which is referred to as an internal coil. Alternatively, the coils can be laid outside the MRF cylinder, which is referred to as an external coil. The internal coil layout employs a fixed-sized annular flow channel (or so-called rectangular duct), and the direction of the fluid flow is mainly perpendicular to the applied magnetic field, while the external coil employs a cylindrical duct and the direction of the fluid flow is parallel to the applied magnetic field. It is noted that higher pressure capacity with faster control response and less leakage could be achieved using internal coil (MRF valve layout in Figure 9) than using external coil (MRF orifice layout in Figure 10) (Grunwald and Olabi, 2008).

Typical MRF dampers

According to different arrangements of basic functioning components (i.e. MRF cylinder and MRF control valve), several typical MRF dampers are illustrated in this section.
**MRF dampers with integrated valves.** The MRF control valve is integrated with the MRF cylinder as a whole, and the resistant flow occurs inside the MRF cylinder. For monotube MRF damper, it can be further divided into two categories according to the layout of coils in MRF control valve. For twin-tube MRF damper, coils are usually placed inside the inner cylinder for generating magnetic field.

**Integrated MRF dampers with internal coils.** Figure 11 shows the schematic structure of an integrated MRF damper with internal coils in MRF control valve. The MR control valve is placed inside the moving piston, and the piston rod has a hollow structure to accommodate coil wires and separate them from MRFs. This type of MRF dampers could work in three complex fluid modes (i.e. single flow mode; mixed mode, using valve mode and direct shear mode; and multimode, using valve mode, direct shear mode, and squeeze mode) due to the available relative motion of magnetic core and outer housing. Figure 12 shows the structure of a single valve mode–integrated MRF damper, which employs a fixed annular orifice between the magnetic coil bobbin and an additional inner cylinder rounded by a Teflon seal in the piston. This design has the flux line pass through the inner cylinder in their return path and is characterized by high seal requirement but low weight due to aluminum outer housing. Figure 13 shows the structure of a mixed-mode–integrated MRF damper, which employs an annular gap between the magnetic outer housing and the core bobbin. This design has fewer requirements on sealing but increased weight due to low-carbon steel outer housing being employed as a part of magnetic circuit (Unsal, 2006).

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**Figure 10.** Fluid flow and magnetic field directions in the MRF orifice. Source: Grunwald and Olabi (2008). MRF: magnetorheological fluid.


**Figure 12.** Schematic structure of valve mode MR fluid damper. Source: Unsal (2006).

**Figure 13.** Schematic structure of mixed-mode MR fluid damper. Source: Unsal (2006). MRF: magnetorheological.
Besides, the design and fabrication of this mixed-mode–integrated MRF damper can be conducted based on commercially available air cylinder or hydraulic cylinder and an assembled custom-made low-carbon steel piston head sleeve with slots and cushion rings (Gavin et al., 2001; Lau and Liao, 2005). Figure 14 shows the structure of a multimode MRF isolator. The electromagnetic coil is wrapped around the bobbin attached to the top elastomer through the connecting screw. The outer cylinder serves as a magnetic flux return for creating valve and shear damping during the movement of the top elastomer. The squeeze mode damping is created by the axial motion of the elastomer and the bobbin that imposes radial flow of the fluid in the gap between the bottom of the bobbin and the outer cylinder. This structure can not only increase damping force but also lessen unwanted lockup state between the magnetic pole gap against relatively high-frequency and small-amplitude excitations, which often exists in a single-valve-mode MRF isolator (Brigley et al., 2007).

Generally speaking, integrated MRF dampers with internal coils are usually featured with a compact structure and good wire protection but suffer from harsh design requirement, increased weight of the piston head, hard heat dissipation due to the internal coil in MRFs, inconvenient assemble and maintenances, and inflexible replacement of MRF control valve. Nevertheless, it is noted that most MRF dampers utilize the integrated design of piston/valve monotube structure due to the efficiency in providing high on-state and low off-state force in a compact and robust package (Carlson, 2009).

**Integrated MRF dampers with external coils.** For the MRF damper with internal coils, an increase in the temperature of MRFs could cause the viscosity to drop and reduce the force capacity of the damper since MRF is directly affected by the heat buildup in the coil (Dogruoz et al., 2003). Alternatively, the external coil layout is utilized in some applications. Figure 15 shows a schematic design of an integrated MRF damper with external coils in MRF control valve. It is noted that the MRF damper with simultaneously moving piston and external coil (Figure 15(a)) could provide more stroke length than the MRF damper with fixed external coil (Figure 15(b)) (Chen et al., 2010). The MRF damper with external coils brings the advantage that all thermal energy generated through resistance of the electromagnet could be discharged to the atmosphere and could not contribute any heating to the damper’s hydraulic system since the coil is intact with MRF (Hitchcock et al., 2007). However, the available damping force is less than that of MRF damper with internal coils under the same volume constraint and number of coil turns, as analyzed in Grunwald and Olabi (2008).

**Twin-tube–integrated MRF dampers.** Besides the monotube-integrated MRF dampers, several twin-tube–integrated MRF dampers have been developed in the literature, which incorporate the MRF control valve into the moving piston of a twin-tube MRF cylinder with the similar arrangement as that utilized in a monotube-integrated MRF damper (Chooi and Zhu et al. 7
Oyadiji, 2009a; Poynor, 2001; Prabakar et al., 2009; Sassi et al., 2005; Yu et al., 2009; Zhou and Zhang, 2002). Note that the piston integrated with the MRF control valve can be placed with a considerable distance from the magnet poles in some mixed-mode twin-tube-integrated MRF damper, and the dynamic damping characteristics caused by the MR effect are a little different from that of pure mixed-mode since the magnetic field may affect the flow near the piston (Zhou and Zhang, 2002).

**MRF dampers with bypass valves.** The MRF control valve is incorporated into the MRF cylinder with a bypass duct, and the resistant flow occurs outside the MRF cylinder as shown in Figure 16. In practical design, the monotube or double-ended MRF cylinder is employed in convenient conjunction with a bypass duct. It is noted that the bypass-type MRF dampers are operated in single-valve mode since the flow channel and outer housing are always fixed.

Similar to the magnetic coil layout in integrated MRF dampers, there are bypass-type MRF dampers with internal and with external coils accordingly. Figure 17 shows two different bypass-type MR shock dampers with internal coils (single-ended damper and double-ended damper) for high impulsive force systems having mostly large stroke and large excitation amplitude such as gun recoils, aircraft landing gear units, and transportation system for dangerous objects (Nam and Park, 2007). Figure 18 shows a bypass-type MRF damper with external coils, which contains porous media and is externally wrapped by a stationary magnetic coil for energy absorber application with the capability of generating large damping force and a wide force range in a narrow and compact volume (Cook et al., 2007; Hu et al., 2009).

The major advantages of bypass-type MRF dampers are their simplicity in structure design and assembly and convenience in maintenance (Unsal, 2006). This obviously facilitates realization of modular design according to different application requirements based on available hydraulic cylinder without special modification of piston/piston rod. Specifically, modular disk-type bypass MRF valves were designed for providing high force in large-scale seismic applications (Bangrakulur, 2004; Gordaninejad et al., 2010). A series of bypass-type MRF dampers with different force ability (2, 20, and 200 kN) were developed for full-scale civil engineering (Sodeyama et al., 2003, 2004). However, the disadvantage of bypass-type MRF dampers could be its less compact design due to the presence of a bypass duct and, additionally, requirements for wire protection in the bypass-type MRF dampers with internal coils, as compared to the integrated MRF damper with internal coils and wiring in the hollow piston (Lau and Liao, 2005).

**Syringe-type MRF dampers.** A syringe-type MRF damper is shown in Figure 19. It consists of two syringes connected by a plastic hollow cylinder through which the fluid flows from one syringe to the other. Since no vacuum will be created when either of the shafts is pulled back, a parallel rod is connected to both of the shafts or an additional aluminum hollow cylindrical outer housing is utilized. The main advantage of this design is that the coil is not inside the housing, which makes the wiring much simpler, and the MRF is not
affected much by the heat buildup in the coil (Unsal, 2006).

**Sponge-type MRF dampers.** A sponge-type MRF damper in which the MRF is contained in an absorbent matrix employs the shear mode operation with less complexity and seal problems. As shown in Figure 20, the outer cylinder is a hollow steel tube and the piston head is comprised of an open-celled polyurethane foam saturated with MRFs and an electromagnet that generates the yielding shear force resisting the motion of the piston within the cylinder. Sponge-type MRF dampers could be operated in a direct shear mode without seals, bearings, or precise mechanical tolerances and only requiring a minimum volume of MRFs, which facilitates the application of MRFs in cost-sensitive areas. In practice, a low-cost, sponge-based MRF damper has been utilized in high-performance washing machines (Carlson, 1999; Chrzan and Carlson, 2001; Hoyle et al., 2010). The feasibility of sponge-based force feedback MRF dampers for telerobotic systems was investigated in a study by Ahmadkhanlou (2008) recently.
**MR squeeze-film dampers.** Besides the multimode MRF dampers employing squeeze mode flow (Brigley et al., 2007), controllable MR squeeze-film dampers could be the most applicable devices employing pure squeeze mode of MRFs. This type of MR dampers possess high force and low stroke of simplicity and compactness, which has been applied in rotor or linear motion systems for suppressing unbalanced vibrations (Jolly and Carlson, 1996). Recently, the design problems of MR squeeze-film dampers have been studied by different authors relating to material selection, magnetic circuit analysis, sealing element design, modeling, identification, and so on using various methods (Carmignani et al., 2006; Jagadish and Ravikumar, 2010; Kim et al., 2008).

**Reinforced structure designs for performance improvement**

For an MRF damper, the damping performance such as achievable damping force, dynamic range, and response time is determined by some structure factors such as direction of the magnetic flux lines at fluid resistance gaps, geometry of flow channels, selection of the materials in magnetic circuits, and effective surface area of magnetic field (Wang et al., 2009). In order to improve damper performance, several methods are proposed in the literature.

**Modification of coil configurations**

**Multistage piston/coil configurations.** The damping force of an MRF damper is dependent on the activation region (i.e. the band-shaped volumes of MRF that are activated by a magnetic field). In order to increase damping force, larger activation region is required. However, for MRF dampers with a single-stage piston, the available damping force cannot be significantly increased due to strong sensitivity to the fluid gap in the activation region even though a larger size is utilized. Therefore, multistage pistons or so-called multicoils for MRF dampers are designed, which could be less sensitive to the fluid gap due to an increase in the area of activation regions that the additional coils produce with the same cross-sectional geometry (Poynor, 2001). Figure 21 shows a 20-ton large-scale MRF damper with a multistage piston for civil engineering applications, in which the electromagnetic coils with a total of about 1.5 km of copper wire are wired in three sections on the piston, resulting in four effective valve regions as the fluids flow past the piston (Yang et al., 2002). Figure 22 shows a monotube MR damper (for Ford Expedition in studying roll-over dynamics) with a two-stage piston, which has the capacity of 185-psi nitrogen accumulator and 4.1-in total stroke (Norris, 2003; Poynor, 2001).

The multistage piston/coil MRF dampers are often favorable in enlarging damping force under a constraint volume and compensating for larger gap size in applications (Goncalves et al., 2006). Importantly, some considerations should be noted in the design of multistage piston MRF dampers. First, the coils are alternately wound such that the magnetic fields are additive and do not cancel each other (Carlson, 2001; Poynor, 2001). Second, parallel connections of electromagnetic coils are preferred to achieve faster response due to smaller reluctance generated (Yang et al., 2002). Third, the increase of achievable damping force with respect to additional coil could reach a limitation due to magnetic saturation and volume requirements (Nguyen et al., 2008).

**Bifold configuration of electromagnets.** The bifold configuration that has MRF control valves at each end of the MRF cylinder (Figure 23(b)) rather than inside the piston head in the conventional MRF damper (Figure 23(a)) could achieve higher force, larger dynamic range with compact size, and larger shock velocity requirement through increasing the number of magnetically active volumes while minimizing damper length. As such, a flow-mode bifold MRF damper for shock and...
vibration mitigation application with compact high force and high piston velocity (6.75 m/s) is designed and fabricated in the studies by Mao et al. (2007) and Facey et al. (2005). A cylindrical MR bifold damper is presented in the study by Lee and Choi (2000) applicable to a middle-sized passenger car with the available damping force being about 2 kN under 2 A at a velocity of 0.45 m/s.

Perpendicular coil axis configurations. A novel MRF damper with perpendicular coil axis configuration (Figure 24) is proposed for providing large controllable damping force while only requiring a small amount of energy (Sassi et al., 2005). In contrast to conventional twin-tube MRF dampers, where the electromagnetic coil axis is usually superposed on the damper axis, this new design has the coils wound in a direction perpendicular to the damper axis for enlarging the active surface where the fluid flow is perpendicular to the magnetic field and thus providing more powerful and regularly distributed magnetic field for much lower overall inductance of the circuit and shorter time response. It is verified through experiments that a magnetizing system with eight independent coils may require a current of lower intensity and allow the damper to reach its optimum capacity over a wide range of frequencies (Sassi et al., 2005).

Modification of fluid resistance channels
In order to increase the pressure drop and consequently to enhance the fluid flow block forces without increasing its volume size and energy consumption, some modification methods of fluid resistance channels are developed.

Annular and radial fluid resistance gaps. There could be some disadvantages in conventional MRF control valves that employs annular fluid resistance gap (i.e. the flow of the fluid is axial to the piston motion), such as small adjustable damping force, low utilization of magnetic field, and relatively high-energy consumption. In order to enlarge the achievable pressure drop without obvious increase of volume size, a modular disk-type bypass MRF valve was developed that had fluid resistance channel mostly in radial flow configuration (i.e. the flow of the fluid is radial to the piston motion), while guaranteeing that the magnetic flux line is always perpendicular to the fluid flow and being flexible to
expand from single-stage to two-stage or multistage configuration (Figure 25) (Gordaninejad et al., 2010). Such MRF valves with radial fluid resistance gap are suitable for seismic applications with large force requirements since it is efficient to generate relatively large pressure drop and a considerable dynamic force range in a compact structure. For comparison, it is reported that a 700-psi pressure drop for a single-stage modular disk-type MRF control valve and 1400-psi pressure drop for a two-stage disk-type modular MRF control valve with radial fluid resistance gap could be generated, while an MRF control valve with pure annular fluid resistance gap could only achieve 450-psi pressure drop (Bangrakulur, 2004; Gordaninejad et al., 2010).

Since the efficiency of MRF control valves with circular disk-type fluid resistance gap is superior to that with annular fluid resistance gap under the same magnetic flux density and outer radius of the valve (Ai et al., 2006), an MRF control valve with both annular and radial fluid flow resistance gaps (Figure 26(c)) was developed (Wang et al., 2009). Through the analysis of the pressure drop for three MRF control valves (i.e. single-coil annular duct, two-coil annular duct, and annular-radial duct) with optimization design in a similar specific volume (Figure 26), it was shown that the annular-radial duct can achieve a higher pressure drop with minimum power consumption than the other two MRF control valves (Nguyen et al., 2008).

**Inner bypass flow channels.** An inner bypass MRF damper is proposed recently, which has two passages for MRF flow as shown in Figure 27: one is the outer passage between the cylinder and the shielding sleeve and the other is the inner passage between the yoke sleeve and the piston. The inner passage magnetic field is controlled by electromagnetic excitation since the piston and yoke sleeve are made of magnetic soft materials such as pure iron to form a magnetic circuit. The outer passage is in a magnetic-free state all the time for any applied current in order to avoid vacuum generation and flow block that occurs in the conventional single-resistance gap due to solidification of MRFs at high applied current since the shielding sleeve is made of nonmagnetic material such as aluminum to shield the outer passage from the magnetic field. In addition, the magnetic bias in the added gap is obtained via inclusion of a permanent magnet in the piston. Therefore, the inner bypass MRF damper could obtain large scalability up to 8, low base-damping force lower than 500 N with bidirectional applied current resulting from the inner passage, and avoidance of flow block resulting from the additional outer passage (Poynor, 2001; Zhang et al., 2007).

**Modification of electromagnetic designs**

In order to improve the performance of MRF dampers, the magnetic field should be effectively supplied to the
MRF. Two approaches for performance improvement are proposed in Nam and Park (2009): one is to shorten the magnetic flux path by removing the unnecessary bulk of the yoke in order to improve the static characteristic of the MRF damper and the other is to increase the magnetic reluctance of the magnetic circuit by minimizing the cross-sectional area of the yoke through which the magnetic flux passes in order to improve the dynamic and hysteretic characteristics. The proposed geometry design for electromagnet is shown in Figure 28. It is concluded from the magnetic field analysis and a series of basic experiments that the static damping characteristics of the proposed and conventional MRF dampers are basically identical to each other and that the dynamic and hysteretic characteristics of the proposed MRF damper can be significantly improved in the response time and the nonlinearity of the damping force can be reduced at the excitation velocity of 8 mm/s irrespective of applied currents.

**Design of MRF dampers as a system**

MRF dampers are widely used in vibration control systems, including seismic protection and cable-stayed bridge systems in civil engineering, shock absorption and vibration isolation systems in mechanical engineering, and advanced prosthetics in biomechanical engineering. Generally, for a vibration control system based on MRF dampers, some functions are very important or practically preferred such as force and displacement sensors for control, stiffness elements for softening vibrations and pushes, fail-safe functions for protected usage in case of power failure, and self-power for energy saving. Therefore, the design of MRF damper systems that considers all these fundamental elements together and can fulfill more practical requirements have achieved significant progress in recent years. Instead of the design of a single MRF damper, MRF dampers are designed as a complicated system, which are summarized in this section.

**MRF damper systems with embedded sensors**

In order to use MRF dampers without installing extra sensors for industrial and civil applications, some MRF dampers with embedded sensors are developed. A relative displacement self-sensing MR damper is proposed in the study by Wang and Wang (2009), which can achieve self-sensing of relative displacement and controllability of damping. The system (Figure 29)
mainly comprises an exciting coil wound on the piston and an induction coil wound on the nonmagnetic cylinder. The coil wound on the piston simultaneously acts as the exciting coils of MRFs and the embedded integrated relative displacement sensor (IRDS), while the coil wound on the cylinder acts as the induction coil of the IRDS. The MRF and the IRDS are simultaneously energized through frequency division multiplexing of the exciting coil (Wang et al., 2010; Wang and Wang, 2009).

A prestress-type piezoelectric force sensor-based smart MRF damper is developed in the study by Or et al. (2008) by integrating an actuation-only MRF damper with prestress-type piezoelectric force sensor (Figure 30), and an MRF damper with dual-sensing capability by integrating piezoelectric force and a linear variable differential transformer (LVDT) with a conventional actuation-only MRF damper (Figure 31) is developed in the study by Lam et al. (2010). Such systems have advantages in fulfilling real-time closed-loop feedback control applications for mitigating structural vibrations in a reliable and simple manner.

Self-powered MRF damper systems

Usually, either a power supply or a current amplifier is required to activate the electromagnetic coils in MRF damper systems in practical applications. Since the mechanical energy resulting from vibration and shock motion in vibration systems could be employed in this power supply process, several methods that translate the mechanical energy into electric energy have therefore been utilized in the development of self-powered MRF damper systems. This will be particularly important to large-scale civil constructions where the power supply is impractical.

Addition of energy-harvesting devices. An energy-harvesting device is added into an MRF damper to achieve self-powered effect in Choi and Wereley (2009). This device consists of a stator, a permanent magnet, and a spring, which operates as an energy-harvesting dynamic vibration absorber (DVA) (Figure 32) for not only providing power to the MRF dampers without external energy through converting mechanical energy from shock and vibration into electrical energy but also realizing substantial reductions in system volume, weight, cost, control complexity reduction, and maintenance. It is observed through simulation that this self-powered MRF damper in a single-degree-of-freedom engine mount system can provide good vibration isolation performance using neither a sensor nor a control processor/algorithm and, therefore, can be employed at remote locations where supplying power is impractical.

Addition of electromagnet induction systems. The self-powered MRF damper can also be realized by incorporating electromagnet induction (EMI) part into an MRF damper system. The EMI device (Figure 33) consists of a permanent magnet and a solenoid coil and can function as a power source or a velocity sensor in an MRF damper system since the kinetic energy of reciprocal relative motion is converted into the electric energy and the damping characteristics of the MRF damper is therefore changed by itself (Choi et al., 2007; Jung et al., 2009). Specifically, an adaptive passive control system consisting of an MRF damper and EMI part for suppressing vibration of building structures subjected to ground accelerations was investigated (Jung et al., 2010). A vibration power generator according to EMI principle was developed and integrated with a linear MRF damper in parallel in vibration control...
systems (Figure 34) (Sapinski, 2010). A smart passive system for large-scale civil constructions using MRF damper and an EMI system (Figure 35) was presented, which has adaptability by itself without any controller or sensors since the output of electric energy is proportional to input loads such as earthquakes (Cho et al., 2005; Choi et al., 2007). In addition, a prototype of self-powered, self-sensing MR damper that composed of an MR damper part, a power generation part using stator and mover, a velocity sensor part using EMI, and an interaction and mounting part was designed recently (Figure 36) (Chen and Liao, 2010a).


Figure 30. Schematic diagram and photograph of prestress-type piezoelectric force sensor-based smart MR fluid damper. Source: Or et al. (2008). MR: magnetorheological.
Fail-safe MRF damper systems

A conventional MRF damper usually has poor fail-safe function since it cannot retain a minimum required damping capacity in case of power failure due to its inherent small off-state damping force (Liu et al., 2005). Therefore, the magnetic bias variation technology and channel flow geometry design are presented to realize not only small minimum damping force and large damping range for excellent control performance of an MRF damper system but also appropriate required damping capacity for a good fail-safe behavior.

Magnetic bias variation technology. Magnetic bias variation technology is employed in an MRF damper system to achieve bias of damping force and therefore to increase off-state damping force for a fail-safe function. Figure 37 shows a fail-safe MRF damper with the permanent-electromagnet system, in which two additional permanent magnets are added into the electromagnetic circuit of an MRF damper. The permanent magnets generate a magnetic base field that remarkably solidifies the MRFs without any electric power supply. The electromagnet coil generates a magnetic field that strengthens or weakens the magnetic field of the permanent magnets.
magnets in the active MR gap as requested, depending on the polarity of the current in the coil. In addition, low power consumption could be achieved due to the inherent energy from permanent magnet besides electric power required by the electromagnetic coil (Aydar et al., 2010; Boese and Ehrlich, 2010). A novel two-way controllable MRF damper is presented in the studies by Aydar et al. (2010) and Aydar (2006), which can achieve a softer or harder damping force by cancelation or enhancement of the effect of permanent magnet through bidirectional current for application in washing machines with a fail-safe function. A fail-safe and low-power-consumption MRF damper for suppressing the vibration of a space flexible structure is presented in the studies by Oh (2004) and Oh and Onoda (2002), which achieved much better isolation performance than the conventional ones owing to employing both semiaactive and optimal passive dampers in the absence of control.

Channel flow geometry design. A bypass, fail-safe MRF damper with external electromagnet and several specially designed orifices in the piston (Figure 38(a)) is presented in the study by Gordaninejad et al. (2004). The disk-type orifice (Figure 38(b)) is regarded as a gap between two
parallel fixed disks with radial flow, which introduces uncontrollable viscous pressure drop in addition to other common orifices to create the minimum damping force in the inactivated state. The damper is simple and compact while being capable of generating a considerable dynamic force range, and can be sized for specific vibration-control applications (Dogruer et al., 2008; Gordaninejad et al., 2004; Hitchcock et al., 2002, 2007).

MRF damper systems with embedded spring elements

In vibration isolation systems, MRF dampers are more and more widely utilized owing to their excellent performances such as continuously controllable damping force, large dynamic range, and fast response. Several MRF damper systems with embedded spring functions are developed, which utilize available embedded spring elements such as coil spring, liquid like spring, rubber spring, and air spring.

**MRF dampers integrated with coil springs.** An MRF damper that replaced the original equipment manufacturer (OEM) dampers around by a coil spring is naturally designed in application. Specifically, MRF dampers around by coil springs for shock absorber application (Figure 39) are presented in the study by Breese and
Gordaninejad (2003). A novel MR shock and impact damper with high damping force (nominally 5 kN) at high velocities (nominally 1 m/s) was developed in the studies by Mao et al. (2007) and Facey et al. (2005), which had compact size owing to the bifold MRF control valve arrangement and an integrated linear spring used for absorbing a portion of the impact forces and preventing damage to MR components from excessive damper travel (Figure 40). A small-sized MR isolator that has an internally installed coil spring for preventing static deflection of the mass and aluminum outer hosing except for the magnetic activation areas in manufacture (Figure 41) is developed in the study by Choi et al. (2005) to cope with the vibration isolation problem of avionic packages.

MRF dampers integrated with liquid springs. In view of large bulk modulus of hydraulic system that can produce considerable spring effect, a bypass liquid spring MR shock absorber (Figure 42) that can generate large spring rate and controllable damping through designing different diameter of rods on either side of the piston was proposed in the study by Hong et al. (2006) for use as a compact automotive strut without integrating a steel spring. Specially, some compressible MRFs have been synthesized recently, which are very appropriate for use in such liquid spring damper with benefits of lightweight and high-energy absorption associated with the device (Fuchs et al., 2010).

MRF damper integrated with elastomer/rubber springs. Hybrid dampers that employ fluid damper and elastomeric damper are recently developed for providing adaptive or programmable lag damping augmentation with little stiffness variation in helicopters. There into, a novel hybrid MRF-elastomeric (MRFE) lag damper (Figure 43), which consists of a double lap shear elastomeric damper in parallel with two MR flow-mode dampers, is investigated for helicopter stability augmentation control (Hu and Wereley, 2008). A snubber-type
MRFE lag damper, which consists of a flow valve, a flexible snubber body, and a flexible center wall separating the body into two fluid chambers, is developed for a hingeless helicopter rotor (Ngatu et al., 2010).

The integrated vibration isolators with MRF damper and elastomer as a whole have been developed. Namely, a multimode MR isolator having an MRF chamber attached to the top elastomer element through the connecting screw (Figure 44) was developed, which demonstrated strong nonlinearity of equivalent damping, displacement amplitude, and excitation frequency (Brigley et al., 2007, 2008). An MRF–elastomer vibration isolator/mount (Figure 45) using MRF–elastomer composite samples that is obtained by an MRF encapsulated into an elastomer was designed for small-amplitude and high-frequency vibration control. Both the stiffness and the damping capability of the system is a function of the displacement amplitude and magnetic field strength, and only weakly dependent upon the frequency of excitation (York et al., 2007; Wan and Gordaninejad, 2009).

MRF dampers based on the passive mount utilizing rubber elements have also been adopted to support a static load and to isolate a dynamic load over high- and postresonance frequency range. Specifically, a mixed-mode MR mount that can be operated under both flow and shear modes was constructed through incorporating both a rubber element and an MRF damper with the top end of the rubber element being connected to the piston of the MRF damper and the bottom plate of the rubber element being attached to the MRF damper housing (Figure 46). This type of MR mounts have a compact structure without friction components for realizing adjustable damping and certain support capability without high stiffness like hydraulic mounts, which are proposed to reduce vibration of a structural system consisting of vibrating mass and flexible beams (Choi et al., 2008; Hong et al.,

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**Figure 41.** Schematic diagram and photograph of a small-sized MR isolator with an internally installed coil spring. Source: Choi et al. (2005).

**MR:** magnetorheological.

**Figure 42.** Hydromechanical model of the compressible magnetorheological strut and fabricated compressible MR fluid strut. Source: Hong et al. (2006).

**MR:** magnetorheological.
In addition, a small-sized variable-damping mount filled with MRFs and combined with a magnetic coil at the bottom is presented for precision equipment of automobiles (Ahn et al., 2005).

**MRF dampers integrated with air springs.** Considering low natural frequency and high load capacity inherited from air spring as well as simplicity of MR technology, vibration isolators consisting of air springs and MRF dampers have been presented for application in vehicle suspensions and civil constructions (Maciejewski, 2010). Specifically, an integrated vehicle car suspension that contains an MRF damper, air spring, automatic electronic leveling, sensors, controller, and CANbus data communication (Figure 47) was developed in the study by Achen et al. (2008). A combined shock and vibration isolation using a passive air spring in parallel with a controlled MRF was presented in the study by Tanner (2003) for shipboard application. An MRF isolator consisting of an air spring, an MRF control valve, and an accumulator is proposed in Li et al. (2009) to achieve variable stiffness and damping in structural vibration control (Figure 48).

**Figure 43.** Test fixture for a hybrid damper.

**Figure 44.** The multimode MR isolator: (a) schematic diagram and (b) fabricated isolator.
Source: Brigley et al. (2007).
MR: magnetorheological.

**Figure 45.** MR fluid–elastomer composite samples and schematic of MR fluid–elastomer vibration isolator design: (a) MR fluid–elastomer composite samples and (b) schematic diagram of MR fluid–elastomer vibration isolator design.
MR: magnetorheological.
Performance analysis and parameter optimization play an important role in the analysis and design of an MRF damper system. To this end, three aspects are addressed in this section, including performance specification of MRF dampers, system modeling for quantitative analysis, and parameter optimization.

**Performance specification**

The performance of an MR damper is evaluated by its damping property, dynamic response time, dimension size, control effort, power consumption, and so on (Ebrahimi, 2009).

**Damping property.** Damping property of an MRF damper could be evaluated by force-velocity and force-position characteristics empirically obtained through continuous velocity plot (CVP) method or peak velocity pick-off (PVP) method as shown in Figure 49 (Ebrahimi, 2009; Snyder et al., 2001). It is noted that the damping force is relatively independent of velocity, and large forces could be achieved even at low or zero velocity. Importantly, there is hysteretic behavior under low velocity, which is influenced by mechanical and electrical dynamic parameters such as dynamic response time.

In practice, the complex stiffness ($K'$) is utilized to quantitatively characterize the viscoelastic behavior of the MRF damper as (Pang et al., 1998)

$$K' = K' + jK'' = K'(1 + j\eta_{lost})$$  \hspace{1cm} (1)

where $K'$ is the in-phase or equivalent spring stiffness, $K''$ is the quadrature or loss stiffness, and $\eta_{lost} = K''/K'$ is the loss factor. $K'$ and $K''$ could be calculated by equation (2) from experimental tests, supposing the MRF damper system subjects to sufficient harmonic displacement excitation $x(t)$ and the viscoelastic force $f(t)$ can be measured as

$$K' = \frac{F_x X_e + F_s X_s}{X_e^2 + X_s^2}$$  \hspace{1cm} (2a)

$$K'' = \frac{F_x X_e - F_s X_s}{X_e^2 + X_s^2}$$  \hspace{1cm} (2b)
where \( x(t) = X_c \cos \omega t + X_s \sin \omega t \), \( f(t) = F_c \cos \omega t + F_s \sin \omega t = K''x(t) + (K''/\omega)\dot{x}(t) \). \( X_c \) and \( X_s \) are the cosine and sine Fourier coefficients of \( x(t) \) at frequency \( \omega \) and \( F_c \) and \( F_s \) are the cosine and sine Fourier coefficients of \( f(t) \) at frequency \( \omega \).

In fact, \( K''/\omega \) is the equivalent viscous damping of the damper, which could also be given by

\[
C_{eq} = \frac{E}{\pi \omega V_{p0}^2} \quad (3)
\]

where \( E = \int_{cycle} f(t) dX_p \), \( f = \int_{0}^{2\pi/\omega} f(t) V_{p0} \) is the dissipated energy in a motion cycle, \( V_{p0} \) is the sinusoidal displacement input amplitude, and \( V_{p} \) is the velocity.

Both equivalent stiffness and equivalent damping of MRF dampers are dependent on excitation frequency, amplitude, motion velocity, and applied current due to their inherent nonlinear characteristics (Liao and Lai, 2002; Snyder et al., 2001). In practical design, the damping property could be usually evaluated by maximum damping force and dynamic range (i.e. the ratio of on-state damping force vs. off-state damping force) or equivalent viscous damping with an appropriate equivalent stiffness for static support.

**Dynamic response time.** The dynamic response time has a significant influence on hysteresis behavior and subsequent control performance of an MRF damper, which is affected partially by the inductance of a system that is evaluated by an electrical time constant or so-called the inductive time constant given by (Eslaminasab, 2008; Nguyen et al., 2008, 2009)

\[
T = \frac{L_{in}}{R_w} \quad (4)
\]

where \( L_{in} \) is the inductance of the coil in the driving electronic system given by \( L_{in} = N\Phi/I \). \( N \) is the number of coil turns, \( I \) is the applied coil current, \( \Phi \) is the magnetic flux, and \( R_w \) is the resistance of the coil wire given by

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**Figure 48.** Schematic of MR fluid isolator.  
Source: Li et al. (2009).  
MR: magnetorheological.

**Figure 49.** MR damping properties: (a) force versus displacement and (b) force versus velocity.  
Source: Snyder et al. (2001).  
MR: magnetorheological.
The overall response time is defined as the required time for the force signal transited from initial state to 95% of the final-state/command signal under constant velocities using the excitation of a triangle wave (Koo et al., 2006) The response time of an MRF damper is not only influenced by internal system compliance and external excitation such as operating current, piston velocity, and working temperature (Ebrahimi, 2009; Koo et al., 2006; Yu et al., 2009; Zhang et al., 2011) but is also dependent on the design of the valving geometries (e.g. large response time for annular flow geometry) and small response time for radial flow geometry).

**Geometry dimension size.** The geometry of a conventional MR valve is shown in Figure 50, which consists of a valve core, an outer housing, electromagnetic coils, magnetic poles (active length), and annular flow channels. Note that the generic MR valve can either be placed outside of the cylinder in a bypass configuration or integrated in a moving piston inside the cylinder and that the geometry of MR damper/valve with multiple coils could be the combination of each MR damper/valve with one coil. As shown, the geometry dimension of an MRF damper includes the size of the MRF cylinder such as body diameter, shaft diameter, stroke length/range, damper weight, and the size of MRF control valve such as annular gap size \( t_{a} \), active length of electromagnet \( L_{a} \), and so on, which will have great influences upon achievable damper performance as commented in the following sections.

**Power consumption.** The power consumption of a semiactive MRF damper is given by (Nguyen et al., 2008, 2009)

\[
E = IR^2_w
\]  

The above performance should be specified to fulfill different application requirements such as maximum velocity, stroke length, excitation frequency, power flow rate, and so on (Ahmadian and Norris, 2008; Brigley et al., 2007; Hong et al., 2005; Norris, 2003; Yang et al., 2002). Generally speaking, small off-state damping and high on-state damping are beneficial for vibration control. Small response time and power consumption are required to enhance the dynamic control performance and energy saving effect. Suitable geometry size and force level are specified for different assemble configurations. Other requirements such as durability, cost, little fluid leakage, and appropriate fail-safe function should also be considered.

**System modeling**

A reliable mathematical model is very important to system performance assessment (such as damping characteristics) and parameter optimization. Several mathematical models have been proposed for MRF damper systems in the literature (Wang and Liao, 2011). Physical models are discussed only analytically here for the system design purpose. The physical model is usually derived based on quasi-static analysis of MRFs flowing through an annular duct under a constant velocity and fully developed fluid flow (Yang et al., 2002). A complex quasi-static axisymmetric model based on constitutive models is introduced first to have a comprehensive view on field-dependent MRF behaviors. Approximate models such as parallel-plate model and nondimensional model are then discussed for simplification of analysis and design in practice.

**Axisymmetric models.** The essential field-dependent MRF characteristics are described by rheological models or so-called constitutive models (i.e. the shear stress vs. shear rate of MRFs). Basically, a simple nonlinear Bingham plastic constitutive model (equation (6)) is employed, which assumes that the fluid does not move until the shear stress has exceeded the controlled yield stress (Yang et al., 2002).

\[
\{ \tau = \tau_0 (H) \sgn(\dot{\gamma}) + \eta \dot{\gamma} \} \text{ for } \dot{\gamma} > 0 \quad \eta \dot{\gamma} = 0 \quad \tau \leq \tau_0 \quad (6)
\]

where \( \tau \) is the shear stress; \( \tau_0 \) is the dynamic yield stress of MRFs, which is assumed to be a monotonic function of radius due to the radial distribution of the magnetic field in the gap; \( H \) is the field intensity; \( \eta \) is field-independent postyield plastic viscosity; \( \dot{\gamma} \) is the shear rate; \( \dot{\gamma} = du_s(r)/dr; u_s(r) \) is flow velocity with the boundary condition \( u_s(R_i) = -v_0; v_0 \) is the relative piston velocity; and \( r \) is radial coordinate.

The typical shear stress diagram and velocity profile of the Bingham plastic shear flow in an annular gap is

\[
R_w = N/I^2 = L_wr_w = N\pi d_c(r_{coil}/A_w) \quad \text{in which} \quad L_w \quad \text{is the length of the coil wire,} \quad r_w \quad \text{is the resistance per unit length of the coil wire,} \quad d_c \quad \text{is the average diameter of the coil,} \quad A_w \quad \text{is the cross-sectional area of the coil wire, and} \quad r_{coil} \quad \text{is the resistivity of the coil wire. Note that the electrical time constant could be reduced by using the parallel coil configuration and increasing the maximum achievable voltage through current-driven power supply instead of voltage-driven power supply (Yang et al., 2002).}
\]
shown in Figure 51. As seen, there are three different flow regions whose boundary is dependent on a plug thickness \( \delta = r_2 - r_1 = -2r_0/(dp/dx) \) due to the controlled yield stress. The fluid flows in regions I and II (postyield regions) since the shear stress has exceeded the yield stress, while no flow occurs in region C (preyield region) since the shear stress is lower than the yield stress. The pressure gradient/drop \( (\Delta p = p_L - p_0 = -L(-dp/dx) \) with \( x \) being longitudinal coordinate) is solved numerically by combining the equations of velocity profile, volume flow flux, and boundary conditions in three different flow regions. And then the damper output force is calculated from the effective cross-sectional area, active magnetic length, and pressure drop of fluid flow motion (Yang et al., 2002). In addition, a nonuniform gap allowing for control of postyield damping in addition to the uniform gap adjusting the yield force above is proposed (Lindler et al., 2000).

The assumption of constant plastic viscosity in Bingham plastic model may not be valid for the case where the MRF experiences postyield shear thinning or shear thickening. Thus, improved rheological models are utilized to describe more nonlinear and complex behavior of MRFs. Specifically, a nonlinear viscoelastic model expanded the Bingham model by incorporating a preyield linear damping (Stanway et al., 1996). A Buckingham model (Bingham modified) was applied through supposing a nonlinear magnetic flux distribution across the small gap (Leicht et al., 2009). A Herschel–Bulkey model that introduces a flow behavior index into Bingham plastic model for postyield shear thickening and shear thinning of MRFs was presented, which well described the behavior of the MRF dampers under high-velocity and high-field input (Goncalves et al., 2006; Lee et al., 2002; Wereley, 2008).

**Parallel-plate models.** A parallel-plate model is developed based on the fact that the axisymmetric flow of MRFs can be approximated as a flow through a parallel duct in case of a small ratio between the flow gap and the diameter of piston. The governing equation for pressure gradient in the flow of a Bingham fluid through a parallel duct (Figure 52) is given by (Hong et al., 2005; Yang et al., 2002)

\[
3(P - 2\Gamma)^2 [P^3 - (1 + 3\Gamma - V)^2 + 4\Gamma^3] + \Gamma V^2 P^2 = 0, \\
|V| < 3(P - 2\Gamma)^2 / P
\]  

where \( P \) is the dimensionless pressure gradient that is the ratio of the pressure gradient of Bingham fluid, \( p' \), to the pressure gradient of the Newtonian fluid, \( p_n' \); \( \Gamma \) is the dimensionless yield stress that is the ratio of the smallest pressure gradient, \( p_n' \), when flow occurs between two plates to the pressure gradient of Newtonian fluid, \( p_n' \); and \( V \) is the dimensionless velocity of piston that is the ratio of the volumetric flow rate of pure shear, \( Q_s \), to the volumetric flow rate of Bingham flow \( Q \). Namely, \( p = p'/p_n' \), \( \Gamma = p_n'/2p_n' \), \( V = Q_s/Q \), \( p_n' = 12\Omega_h/wh^3 \), \( p_n' = -dp/dx \), \( p_n' = 2r_0/h \), and \( Q_s = -(whv_0/2) \), where \( L \), \( w \), and \( h \) are geometric length, width, and gap size of the parallel duct, respectively. When \( |V| > 3(P - 2\Gamma)^2 / P \), the pressure gradient \( P \) is governed by equations independent of the dimensionless yield stress \( \Gamma \). Note that equation (7) is applicable for multiple flow modes of MRFs. For example, solely flow-mode operation is modeled when the nondimensional velocity \( V \) approaches zero (Hong et al., 2005).

In practical design, an approximate numeric solution of equation (7) can be given with the assumption of a constant yield stress under the condition of \( 0 < \Gamma < 1000 \) and \( -0.5 < V < 0 \). And it is indicated that the maximum error between parallel-plate model and axisymmetric model is small (not less than 2%) (Yang et al., 2002).

\[
P(\Gamma, V) = 1 + 2.07\Gamma - V + \frac{\Gamma}{1 + 0.4\Gamma}
\]  

The following equations are given through further manipulating equation (8)

\[
F_\eta = \left(1 + \frac{whv_0}{2Q}\right)\frac{12\Omega_hLAp}{wh^3}
\]  

**Figure 51.** A typical stress and velocity profile in an annular gap. 
Source: Yang et al. (2002).

**Figure 52.** MR fluid flow through a parallel duct. 
Source: Yang et al. (2002). 
MR: magnetorheological.
\[ F_{\tau} = c \frac{\tau_0 L A_p}{h} \text{sgn}(v_0) \]  

(9b)

where \( F_\eta \) is the passive damping force due to field-independent viscosity of the MRF and \( F_{\tau} \) is the active damping force due to controllable yield stress. \( A_p \) is the effective cross-sectional area of the piston head. 

c = 2.07 + 1/(1 + 0.4f) = 2.07 + (12Q \eta/12Q \eta + 0.4wh^2 \tau_0)

is a empirical coefficient that depends on the flow velocity profile and is bounded to the interval \([2.07,3.07]\) with lower value relating to more pure rheological part and higher value relating to more significant MR part (Grunwald and Olabi, 2008).

Based on aforementioned analytical models, practical formulations of damping force under different operational modes and flow resistance channels are given in Table 1 (Ai et al, 2006; Brigley et al., 2007, 2008; Grunwald and Olabi, 2008; Wang et al., 2009).

### Nondimensional quasi-static model

The quasi-static nondimensional models are derived to simplify the mechanical analytical models and to provide favorable insights to the compact of various design parameters. Hereinto, hydraulic amplification ratio, Bingham number, and nondimensional plug thickness are presented as nondimensional design parameters, and equivalent damping coefficient and damping capacity or nondimensional damping force and dynamic range are utilized to characterize the nondimensional damper performance.

Based on aforementioned analytical models, practical formulations of damping force under different operational modes and flow resistance channels are given in Table 1 (Ai et al, 2006; Brigley et al., 2007, 2008; Grunwald and Olabi, 2008; Wang et al., 2009).

### Table 1. Practical passive and active damping force under different operational modes and flow resistance channels.

<table>
<thead>
<tr>
<th>Configurations</th>
<th>Passive and active damping force</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valve mode</td>
<td></td>
</tr>
<tr>
<td>Rectangular annular channel</td>
<td>[ F_\eta = \frac{12Q \eta L A_p}{wh} \text{sgn}(v_0) ] ( (10a) ) |</td>
</tr>
<tr>
<td>[ F_{\tau} = \frac{6\mu_0 h}{h} \text{sgn}(v_0) A_p ] ( (10b) ) |</td>
<td></td>
</tr>
</tbody>
</table>
| Cylindrical-shaped annular channel | \[ F_\eta = -\frac{8\mu_0 Q_p}{\mu_0 (d_0/2)} A_p \] \( (11a) \)
| \[ F_{\tau} = \frac{6\mu_0 h}{d_0} \text{sgn}(v_0) A_p \] \( (11b) \) \|
| Radial flow resistance channel | \[ F_\eta = \frac{5Q_p}{2\pi h} \ln \frac{R_0}{r_0} A_p \] \( (12a) \)
| \[ F_{\tau} = \frac{5Q_p}{\pi h} (R_0 - r_0) \text{sgn}(v_0) A_p \] \( (12b) \) \|
| Direct shear mode              |                                                                                                 |
| \[ F_{sh, \eta} = \frac{\tau_0 L}{v_0} \] \( (13a) \)
| \[ F_{sh, \tau} = wL_0 \tau_0 \text{sgn}(v_0) \] \( (13b) \)
| Squeeze mode                   |                                                                                                 |
| \[ F_{sq, \eta} = \frac{3\pi \mu_0 h^2}{2(x_0 + x)} v_0 \] \( (14a) \)
| \[ F_{sq, \tau} = \frac{4\pi \mu_0 h^2}{3(x_0 + x)} \text{sgn}(v_0) \] \( (14b) \)

Note: Rectangular annular channel is a resistance channel where the magnetic field is perpendicular to the direction of fluid flow and cylindrical-shaped annular channel is a resistance channel where the magnetic field is parallel to the direction of fluid flow; \( d_0 \) is the diameter of cylindrical shape flow resistance channel; \( R_0 \) and \( r_0 \) are the radii of the radial flow resistance channel and the central hole of the valve core, respectively; \( r_a \) is the active radius to activate the magnetorheological fluid in squeeze mode; \( x_0 \) is the initial gap between bottom of the bobbin and the bottom of the outer cylinder.
\( \phi_r \), proportional to \( \bar{A} \), can be used alternatively for the integrated MRF damper with MRF control valve in the moving piston due to \( r_p = r_d \). The hydraulic amplification ratio could not only influence the damping capacity of MRF dampers but also significantly characterize the dynamic behavior since the characteristics of flow and mixed-mode MRF dampers are similar when \( \bar{A} \) is large.

2. Two Bingham numbers in terms of different velocities have different applications. It is preferable to use \( B_{ivp} \) for analyzing damper

### Table 2. Nondimensional design parameters in quasi-static design of an MRF damper.

<table>
<thead>
<tr>
<th>Name of variables</th>
<th>Definition</th>
<th>Interactive relationship</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulic amplification/geometry ratio</td>
<td>( \bar{A} = \frac{A_p}{A_d} \phi_v = \frac{L}{h} )</td>
<td>( A_p = \pi r_p^2, A_d = wh = 2 \pi r_d h ) (15)</td>
</tr>
<tr>
<td></td>
<td>( Q = Q_p = A_p V_0 = Q_d = A_d V_d )</td>
<td>(16)</td>
</tr>
<tr>
<td>Bingham number</td>
<td>( B_{ivp} = \frac{\varrho h v_0}{\eta}, B_{ivd} = \frac{\varrho h u_0}{\eta} )</td>
<td>( B_{ivp, mix} = 6 \bar{\delta} \left( \frac{1}{2} + \bar{A} \right) \frac{1 + \sqrt{1 - \bar{\delta} / 6 (1 + \bar{\delta} / 2) (\bar{A} + 1 / 2)^2}}{2 (1 - \bar{\delta}) (1 + \bar{\delta} / 2)} ) (17)</td>
</tr>
<tr>
<td>Nondimensional plug thickness</td>
<td>( \bar{\delta} = \frac{\delta}{h} )</td>
<td>( B_{ivp, flow} = 6 \bar{\delta} \bar{A} \frac{1}{(1 - \bar{\delta}) (1 + \bar{\delta} / 2)} ) (18a)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>( B_{ivp} = \bar{A} B_{ivd} ) (18b)</td>
</tr>
</tbody>
</table>

MRF: magnetorheological fluid.

### Table 3. Nondimensional formulations of damper performance in quasi-static design of an MRF damper.

<table>
<thead>
<tr>
<th>Name of variables</th>
<th>Formulations for damper performance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equivalent damping coefficient</td>
<td>( C_{eq} = \frac{F}{V_p} = \frac{A_p \Delta P}{V_p} = \frac{2 \pi A_p B_{ivp}}{\eta} ) (19)</td>
</tr>
<tr>
<td>Newtonian viscous damping coefficient</td>
<td>( C_{0, mix} = \frac{12 \varrho h L}{\eta} \left( \frac{1}{2} + \bar{A} \right) ) (20a)</td>
</tr>
<tr>
<td></td>
<td>( C_{0, flow} = \frac{12 \varrho h L \bar{A}}{\eta} ) (20b)</td>
</tr>
<tr>
<td>Damping capacity</td>
<td>( \frac{C_{eq}}{C_{0, mix}} = \frac{1 + \sqrt{1 - \bar{\delta} / 6 (1 + \bar{\delta} / 2) (\bar{A} + 1 / 2)^2}}{2 (1 - \bar{\delta}) (1 + \bar{\delta} / 2)} ) (21a)</td>
</tr>
<tr>
<td></td>
<td>( \frac{C_{eq}}{C_{0, flow}} = \frac{1}{(1 - \bar{\delta}) (1 + \bar{\delta} / 2)} ) (21b)</td>
</tr>
<tr>
<td>Nondimensional damping force and dynamic range</td>
<td>( \phi_D = \frac{F}{\varrho} = \frac{a_1 + a_2 \pi - \phi_v}{a_1 - a_2 V} = 1 + \frac{\phi_v}{\phi_1} \frac{\phi_v}{\phi_1 + \phi_2} ) (22a)</td>
</tr>
<tr>
<td></td>
<td>( \phi_f = a_1 \phi_r \frac{3}{2} + \left( \frac{\phi_v}{\phi_1} \phi_v + a_3 \right) \phi_r \frac{2}{2} ) (22b)</td>
</tr>
</tbody>
</table>

MRF: magnetorheological fluid.

Note: The subscript “mix” represents mixed-mode characteristics and “flow” represents flow-mode characteristics. \( a_1, a_2, \) and \( a_3 \) are coefficients from equation (7).
performances with consideration of external excitations due to its incorporated explicit measurable piston velocity and, on the other hand, to use $B_{rim}$ for analyzing fluid flow of MRFs inside the gap due to its direct indication of Bingham plastic behavior and unified inspection of flow characteristics with other well-known nondimensional variables such as Reynolds number.

3. The nondimensional plug thickness is dependent on Bingham numbers that are valuable for evaluating the strength of field-controlled flow status. Concretely, large Bingham numbers imply that the dampers operate at low speed or in a weakly postyield condition, while small Bingham numbers imply that the damper operates at high speed or in a strongly postyield condition. And zero Bingham numbers indicate zero dynamic yield shear stress due to the absence of field.

Note that Bingham number is usually used under a constant piston velocity. If the damper is operated under dynamic piston loading, the piston velocity for the Bingham number is given by a root mean square (RMS) value of the velocity ($V_{p,\text{rms}} = \left(\frac{1}{2}\pi^2/\omega \int_0^\infty v_0^2\,dt\right)^{1/2} = \frac{v_0}{\sqrt{2}}$, where $v_0$ is the amplitude of piston velocity $v_0$). Nondimensional quasi-static analysis and design of the MRF damper based on Herschel–Bulkley constitutive model rather than the aforementioned Bingham plastic constitutive model were investigated in the study by Wereley (2008). A unified expression of nondimensional damping capacity, Bingham number, plug thickness, and velocity gradient based on a unified constitutive model containing Bingham plastic, biviscous, and Herschel–Bulkley models by introducing flow index ($n$) and stress coefficient ($\tilde{\tau}$) were proposed in the study by Hong, John, Wereley, et al. (2008).

**Dynamic fluid models.** Some dynamic fluid models that could better address nonlinear, asymmetric, and hysteretic behaviors of MRFs have been presented using the actual physical process analysis of MRFs through an annular gap. For example, a dynamic mathematical model based on the general fluid mechanics and the compressibility of MRFs inside the chambers was developed, which confirmed the influence of annular gap in the general analytical quasi-static expressions of parallel-plate approximation (Chooi and Oyadiji, 2008, 2009b; Wang and Gordaninejad, 2001); a dynamic model of an MR shock strut that accounts for the effects of fluid compressibility was developed, and primary optimization design was conducted based on the equivalent MR model (Batterbee et al., 2007a, 2007b). However, these dynamic fluid models are usually too complex to apply in an MRF damper design.

**Models considering magnetic circuits.** The dynamic yield stress $\tau_0$ in the mechanical design model above is controlled by the magnetic field intensity $H_f$, which is characterized by a nonlinear magnetization curve in the magnetic circuit. The magnetomotive force that generated in the magnetic circuit resulting in the corresponding magnetic field intensity is given by Ampere’s law (Grunwald and Olabi, 2008; Nam and Park, 2009)

$$NI = \int_C Hdl = H_f l_f + H_s l_s$$  \hspace{1cm} (23)

where $H_f$ and $H_s$ are the magnetic field intensity applied to MRFs and the yoke, respectively. $l_f$ and $l_s$ are the length of the magnetic flux path related to MRFs and the yoke that includes the valve core and housing for flux return, respectively.

On the other hand, the magnetic flux and the cross-sectional area of the MRF control valve are constrained by Gauss’ law

$$\Phi = \int_S BndS = B_f S_f = B_s S_s$$  \hspace{1cm} (24)

where $B_f$ and $B_s$ represent the magnetic flux densities of the MRFs and yoke, respectively, and $S_f$ and $S_s$ represent the cross-sectional areas of the MRFs and yoke, respectively. Note $B_s$ is upper-bounded by magnetic saturation of the used ferromagnetic material. At low magnetic fields, flux density is proportional to the induction of a material, that is, $B_f = \mu_0 \mu_s H_s$ and $B_s = \mu_0 \mu_y H_f$, in which $\mu_0$ is the constant magnetic permeability of free space, and $\mu_s$ and $\mu_y$ is the relative permeability for each material (i.e. MRFs and yoke).

Then, the minimum saturation magnetic flux density of a yoke material and the applied magnetic field intensity to the MRFs could be reasonably determined from equations (23) and (24)

$$B_{n,sat} \geq \frac{\mu_0 NI}{S_f l_f / (S_f \mu_f) + l_s / \mu_s}$$  \hspace{1cm} (25a)

$$H_f = \frac{NI}{l_f + \mu_s l_s / (\mu_s S_s)}$$  \hspace{1cm} (25b)

Thus, the electromagnetic and mechanical design should be considered simultaneously to effectively supply the magnetic field to MRFs. That is, the maximized magnetic field energy in the MRF gap while minimized energy lost in steel flux conduit of yoke and regions of nonworking areas is preferred, which required that the total amount of steel in the magnetic circuit should be minimized while sufficient cross-section of steel must be maintained for avoiding magnetic saturation (Unsal, 2006). In addition, it is preferred to have a resistor in parallel with a Zener place in parallel with the coil for
safely dissipating the current in an MR device (Gavin et al., 2001). However, the nondimensional design of the MRF damper with both mechanical and electromagnetic variables is less reported in the literature due to the complex analytical solution of magnetic distribution that is usually obtained using a three-dimensional electromagnetic finite element analysis now and should be further investigated (Dogruer et al., 2008; Gavin et al., 2001).

**Parameter optimization and analysis**

The critical design parameters of MRF characteristics (yield stress and plastic viscosity) and geometric dimensions (cylinder diameter, gap size, active pole length and coil width) can be optimized in practice to accomplish the best performance of an MRF damper system.

**Initial parameter design.** From formulations of active damping force and dynamic range, geometry constraints for flow mode and direct shear mode are given, respectively, as (Carlson and Spencer, 1996; Jolly et al., 1999; Yang et al., 2002)

\[ \frac{wh^2}{c} = \frac{12}{12} \left( \frac{\eta}{\tau_0} \right) \lambda Q \]  
\[ h = \frac{\eta}{\tau_0} AV_0 \]  

(26a)

(26b)

The minimum active fluid volume is the volume evaluation of MRFs exposed to the magnetic field, which is obtained through further manipulation of equation (26)

\[ V_{min} = Lwh = k \left( \frac{\eta}{\tau_0} \right) \lambda W_m \]  

(27)

where \( k = 12/c^2 \), \( \lambda = \Delta P_L/\Delta P_m \), and \( W_m = Q \Delta P_L \) for flow mode and \( k = 1, \lambda = F_z/F_\eta \), and \( W_m = v_0 F_z \) for direct shear mode.

Therefore, the initial geometric parameters of the MRF damper \( (L, w, h) \) could be determined according to required damping performances, including the dynamic range \( \lambda \), the controllable mechanical power dissipation \( W_m \), and fluid material properties \( (\eta/\tau_0^2 \) and \( \eta/\tau_0). \) In addition, it is suggested that the overall size of a well-designed and magnetically efficient MRF valve is 25–50 times the minimum active fluid volume, that is, \( V_{valve} = (25–50) V_{min} \) (Carlson, 2005).

**Parameter optimization.** As listed in section “Performance specification,” there are quite a few indexes for quantifying the overall system performance. In a semiactive MRF damper control system, these indexes usually exhibit design conflicts according to mathematical models illustrated in section “System modeling.” For example, large damping force can be achieved with small gap size while large dynamic range is obtained with large square times of gap size. Therefore, the optimal design is necessary to achieve the best trade-off between these conflicts. Namely, an object function that consists of multiple performance indexes of the MRF damper is presented in terms of design parameters. Then typical optimal procedure involving interactions combining with mechanical geometric dimension design and electromagnetic circuit design is conducted through employing a sequential least squares method or ANSYS magnetic routines.

The general object function for parameter optimization usually involves maximizing the integrated damping force and dynamic range (Yang et al., 2002) and is consequently extended to minimize the time constant and power consumption for advantageous dynamic performances in the MRF damper control system under constrained damper size and electrical characteristics (Gavin et al., 2001). Especially, the volume-constrained optimization of MRF control valve/damper has been mostly investigated in recent years. The geometric dimensions such as the coil width, the flange thickness, and the housing thickness, which significantly affect the damper performance, are considered as design variables (DVs), and the valve ratios (dynamic range) are employed as the objective function. Then, the parameter optimization is conducted in a specified volume determined by outer radius and height through a golden-section algorithm and a local quadratic-fitting technique via commercial finite element method (Li et al., 2003; Nguyen et al., 2007; Yoo and Wereley, 2002). Moreover, the objective function could be extended to a weighted combination of control energy and inductive time constant (Nguyen et al., 2008) and a combination of damping force, dynamic range, and inductive time constant (Nguyen and Choi, 2009a, 2009b; Nguyen et al., 2009a) under specific operational requirements such as the pressure drop. Considering that FEM-based optimal design cannot clearly show the meaning of DVs, an alternative analytical method is developed for maximizing the yield stress pressure drop of an MRF control valve or the yield stress damping force of an MRF damper using constant magnetic flux density (Nguyen et al., 2009b). Besides, volume-constrained optimal design in terms of nondimensional plug thickness and damping coefficient is also primarily developed by geometric and magnetic optimization with ANSYS magnetic finite element routines (Rosenfeld and Wereley, 2004; Sathianarayanan et al., 2008).

It is noted that the parameter design and optimization of an MRF damper in the literature is usually based on quasi-static constitutive models of MRFs in addition to complex finite element analysis of magnetic flux density and is usually conducted with prespecified constraints such as valve flow rate and geometric size based on an unclear weighted multiobjective function without direct application considerations. Moreover, the performance variation with respect to multiple
design parameters is seldom commented due to hard analysis of complex fluid flow characteristics and magnetic flux distribution. Practically, optimization design of MRF damper could be further explored with the consideration of application matching with a systematic and unified process that has appropriate constraints directly from application requirements. To this end, the achievable performances could be evaluated based on an effective nondimensional dynamic model of an MRF damper system and then specified according to application requirements with dimensional realization. On the basis of such application-matching analysis and constraint specification, the unified object function with clear and appreciative weights in terms of effective nondimensional design parameters of MRF damper including both material and geometric characteristics should be formulated, and the corresponding parameter sensitivity analysis could be further discussed to determine manufacturing tolerance and assess vulnerable parts in application.

**Analysis of nonlinear damping characteristics.** The damping characteristics of MRFs obviously demonstrate nonlinearity due to complex fluid dynamics and magnetic field with saturation constraints. In practice, the strongly nonlinear behavior of MR dampers could bring unexpected difficulties in identification, analysis, and control of corresponding dynamic systems such as hysteresis behaviors, time-varying and discontinuous model parameters, difficulty in inverse dynamics, and so on (Chen et al., 2010; Chooi and Oyadiji, 2008; Jolly, 2001; Jolly et al., 1999; Jung et al., 2004; Kciuk and Turczyn, 2006; Maciejewski, 2010; Muhammad et al., 2005; Thanh and Ahn, 2006; Wang and Liao, 2011). The recent reviews on these topics can be referred from Wang and Liao (2011), Ahmadkhanlou (2008), Jung et al. (2004), and Muhammad et al. (2005). From a different perspective, recent advances in the analysis of nonlinear damping systems indicate that noticeable advantages could be produced by the nonlinearity in system vibration control compared with linear viscous damping (Gao and Cheng, 2004, 2005; Ibrahim, 2008; Jing et al., 2008a, 2008b, 2011a, 2011b; Jing and Lang, 2009; Lang et al., 2009; Marouze and Cheng, 2002).

For MRF dampers, it is shown that the equivalent damping of MRF damper is much greater at lower frequencies but tends to field-off equivalent viscous damping as frequency increases (Facey et al., 2003; Liao and Lai, 2002). It is further shown in the study by Zhu et al. (2011) that MR dampers is advantageous to achieve effective resonance mitigation while maintaining fast-decaying velocity of vibration attenuation for higher frequencies of interest as well as small force mobility at all frequencies for strong antiforce disturbances. Therefore, systematic exploration of the potential nonlinear benefits that could be brought by MRF dampers would be an interesting topic to be developed.

**Conclusion**

A systematic overview on structure design issues of MRF dampers is presented, including basic structure design, reinforced structure design, design of MRF damper systems, and performance analysis and optimization issues. The basic mechanical structure designs are categorized as several types of MRF dampers according to arrangement manners of MRF cylinders and MRF control valves. The improved methods for better performance of an MRF damper such as enlarging damping force and decreasing response time are presented with a focus on coil configuration design, fluid resistance channel design, and electromagnetic design in magnetic circuits. The design problems of MRF damper systems, which possess supplemental functions in addition to the conventional single-MRF damper system, involve the sensor-based MRF damper systems that integrate displacement sensors or force sensors into the conversional action-only MRF damper system, self-powered MRF damper systems that add an energy-harvesting device or EMI part into the MRF damper system, fail-safe MRF damper systems that increase the off-state damping force through magnetic bias technology and channel flow geometry design, and integrated spring MRF damper systems that employ spring elements such as coil spring, elastomer, and air spring to construct complete vibration control systems. The performance analysis issues for MRF damper systems are discussed by summarizing performance specifications, analytical design models in both mechanical and electromagnetic area, and optimization development of design parameters. For the structure design of MRF dampers, the following conclusions could be drawn:

1. The structure design of MRF damper systems is always aimed at damper performance enhancement, but more and more practical factors are considered such as the effective mechanical geometry and electromagnetic circuit characteristics, and innovative structure or intelligent materials are introduced to construct a considerably complex but more compact and versatile systems.

2. Optimization design of MRF dampers could be further systematically explored with the consideration of practical constraints in a systematic and unified manner. This could be realized through an effective nondimensional dynamic model of MRF dampers, a unified objective function for optimization in terms of effective design parameters from both material and geometric characteristics, and effective parameter sensitivity analysis for assessing manufacturing tolerance and vulnerable parts in application.

3. Nonlinear MRF dampers in vibration suppression exhibits much better performances with
effective resonance attenuation and better high-frequency isolation performance as well as low force transmissibility in a broad band of frequencies for strong antiforce disturbances compared with linear viscous damping systems. The potential benefits inherited from nonlinear damping characteristics of MRF dampers in vibration control could be further explored.

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