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Xiaocong Zhu, Xingjian Jing and Li Cheng

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# Magnetorheological fluid dampers: A review on structure design and analysis

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Xiacong Zhu, Xingjian Jing and Li Cheng

## 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., 2008; Lau and Liao, 2005), landing gear systems (Batterbee et al., 2007a, 2007b; Lee et al., 2009), ship isolator systems (Deng et al., 2009), seismic protection and cable-stayed bridges (Hietanen, 2002; Jung et al., 2004; Muhammad et al., 2005), and motion and advanced prosthetic systems (Carlson et al., 2001; Hietanen, 2002; Jolly, 2001). Moreover, some commercial MRF dampers such as Carrera MagneShocks™ and Delphi MagneRide™ have already been developed for vehicle suspensions since 2002 (Carlson, 2005; Craft, 2003; Jolly et al., 1999).

To accommodate the fast growing demand in MRF dampers, different issues, including design, analysis, modeling, and control of MRF damper systems have been extensively studied. Some excellent literature

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.

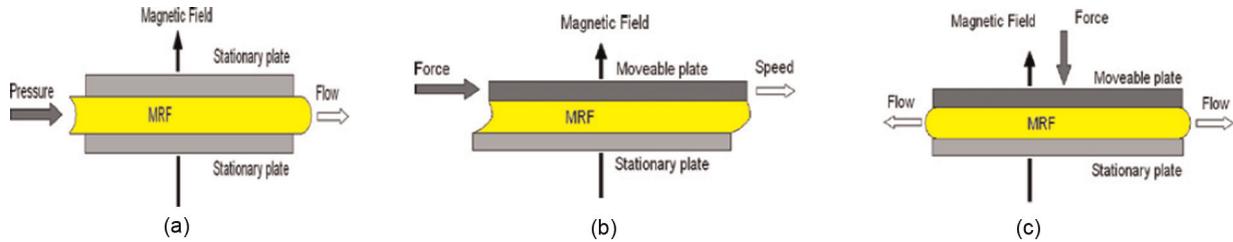
The organization of this article is as follows. The basic structure designs of MRF damper are addressed in section “Basic structure design issues of MRF dampers.” Section “Reinforced structure designs for performance improvement” presents some reinforced structure design methods for better damping performance. 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.”

## 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.,



**Figure 1.** Three basic operational modes of MRFs: (a) flow model, (b) direct shear mode, and (c) squeeze mode.

Source: Olabi and Grunwald (2007).

MRF: magnetorheological fluid.

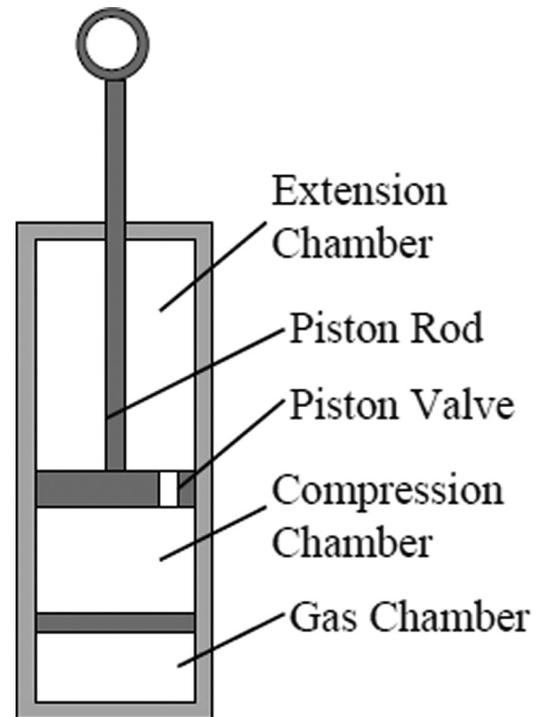
2008). Besides, multiple fluid modes or mixed-mode MRF devices that simultaneously employ at least two basic operational modes have alternatively been developed to enlarge available damping force and simple manufacture requirements of MRF damper (Brigley et al., 2007; Choi et al., 2008; Hong, Wereley, 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.

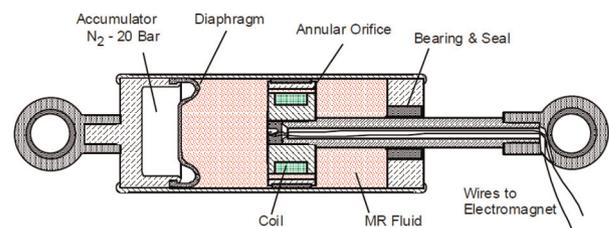
### MRF cylinders

**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



**Figure 2.** Principle of monotube passive damper.

Source: Ebrahimi (2009).



**Figure 3.** Monotube MRF damper.

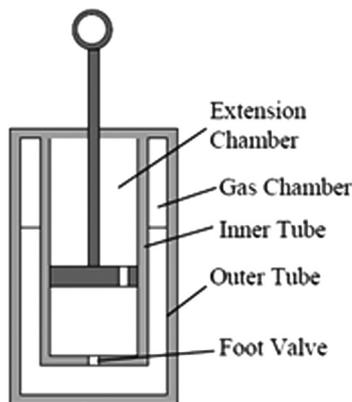
Source: Jolly et al. (1999).

MRF: magnetorheological fluid.

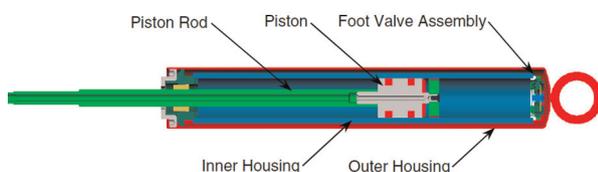
gas chamber adds a spring effect to the force generated by the damper simultaneously and maintains the damper at its extended length when no force is applied (Ebrahimi, 2009; Reichert, 1997). In addition, the bearing at the end of the extension chamber is utilized to guide movements and the corresponding 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



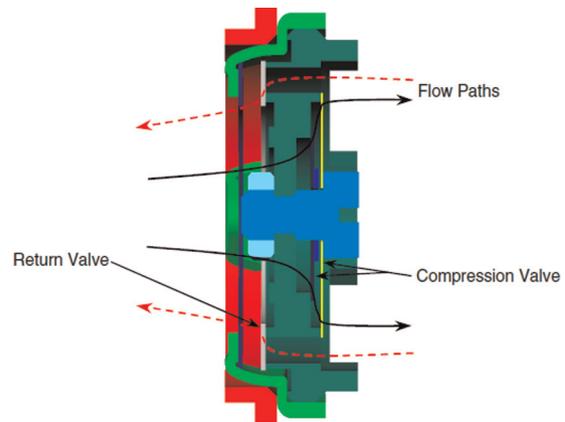
**Figure 4.** Principle design of twin-tube passive damper. Source: Ebrahimi (2009).



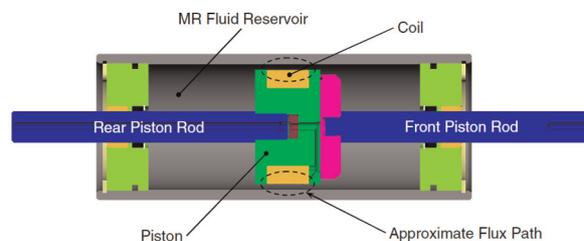
**Figure 5.** Section view of twin-tube MRF damper. Source: Poynor (2001). MRF: magnetorheological fluid.

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; Eslaminasab, 2008; Poynor, 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).

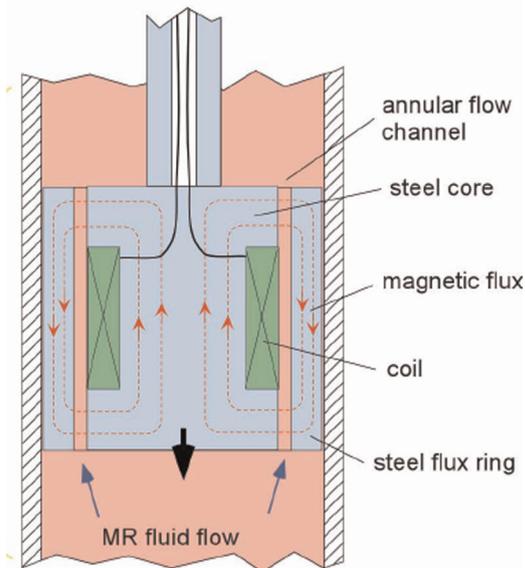
**Double-ended structure.** A double-ended structure is a special configuration from the monotube structure, which is shown in Figure 7. As seen, the piston rod has the same diameter and protrudes through both ends of the double-ended MRF damper. This arrangement does not need a rod-volume compensator 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 (Poynor, 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).



**Figure 6.** Details of foot valve. Source: Poynor (2001).



**Figure 7.** Double-ended MR fluid damper. Source: Poynor (2001). MR: magnetorheological.



**Figure 8.** Magnet flux in a monotube MR fluid damper.

Source: Carlson (2009).

MR: magnetorheological.

### MRF control valves

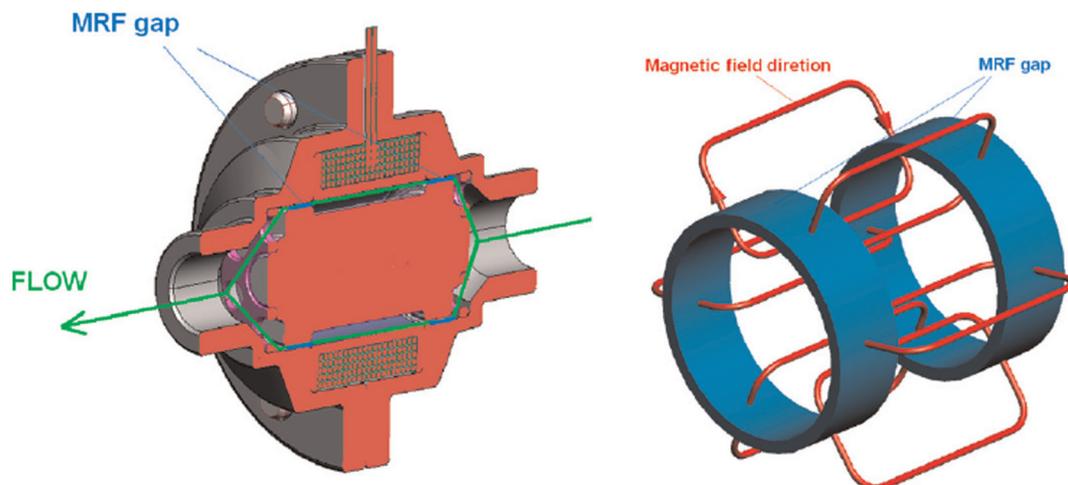
An MRF control valve is used for generating magnetic circuits, which achieve electric control of mechanical damping forces without mechanical moving parts. Figure 8 shows a typical MRF control valve incorporated in a moving piston of an MRF damper. As seen, the MRF control valve comprises a magnetic 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

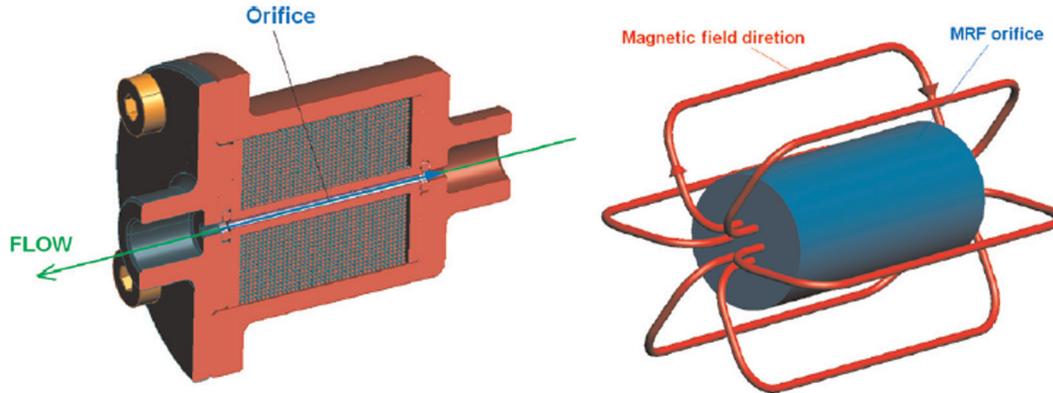
According to different arrangements of basic functioning components (i.e. MRF cylinder and MRF control valve), several typical MRF dampers are illustrated in this section.



**Figure 9.** Fluid flow and magnetic field directions in the MRF valve.

Source: Grunwald and Olabi (2008).

MRF: magnetorheological fluid.

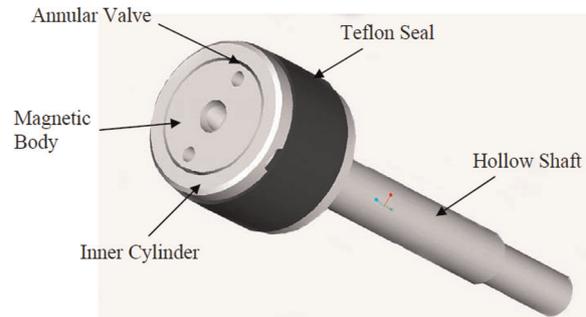


**Figure 10.** Fluid flow and magnetic field directions in the MRF orifice.  
Source: Grunwald and Olabi (2008).  
MRF: magnetorheological fluid.

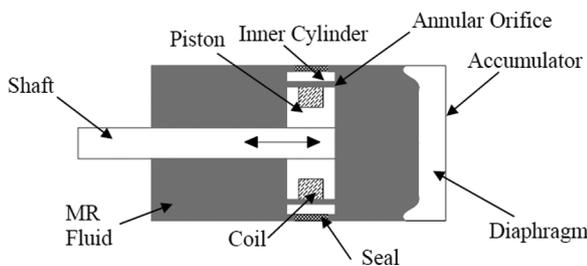
*MRF dampers with integrated valves.* The MRF control valve is integrated with the MRF cylinder as a whole, and the resistant flow occurs inside the MRF cylinder. For monotube MRF damper, it can be further divided into two categories according to the layout of coils in MRF control valve. For twin-tube 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

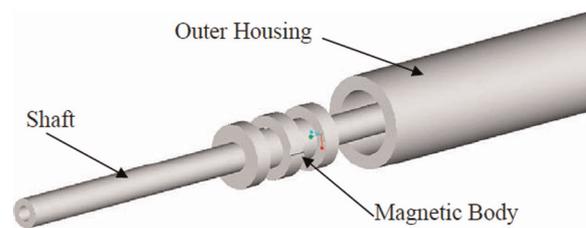
by a Teflon seal in the piston. This design has the flux line pass through the inner cylinder in their return path and is characterized by high seal requirement but low weight due to aluminum outer housing. Figure 13 shows the structure of a mixed-mode-integrated MRF damper, which employs an annular 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).



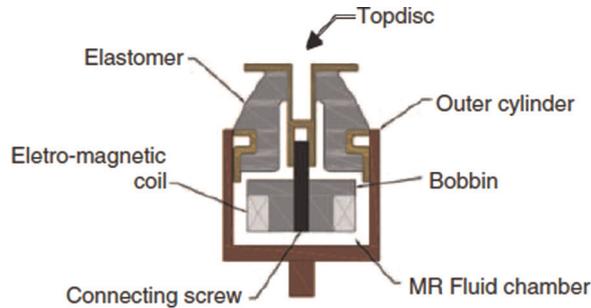
**Figure 12.** Schematic structure of valve mode MR fluid damper.  
Source: Unsal (2006).



**Figure 11.** Schematic structure of integrated MR fluid damper.  
Source: Unsal (2006).  
MR: magnetorheological.



**Figure 13.** Schematic structure of mixed-mode MR fluid damper.  
Source: Unsal (2006).  
MRF: magnetorheological.



**Figure 14.** Schematic of multimode MR isolator.

Source: Brigley et al. (2007).

MR: magnetorheological.

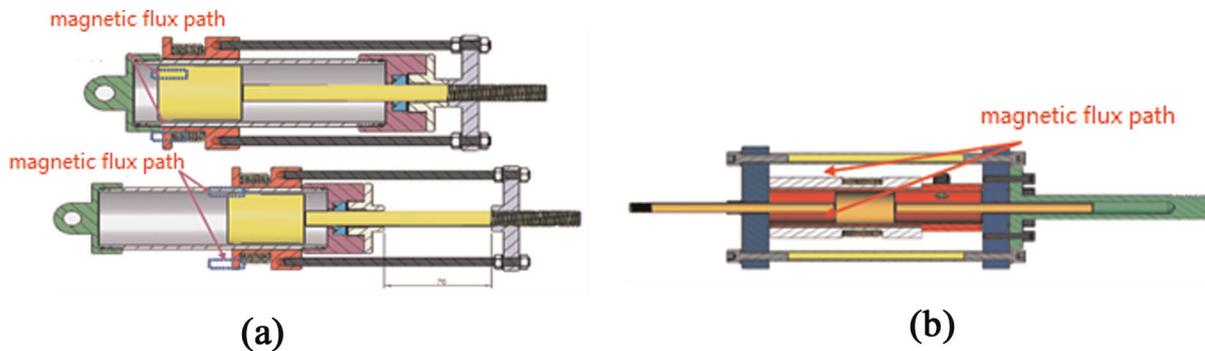
Besides, the design and fabrication of this mixed-mode-integrated MRF damper can be conducted based on commercially available air cylinder or hydraulic cylinder and an assembled custom-made low-carbon steel piston head sleeve with slots and cushion rings (Gavin et al., 2001; Lau and Liao, 2005). Figure 14 shows the structure of a multimode MRF isolator. The electro-magnetic 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).

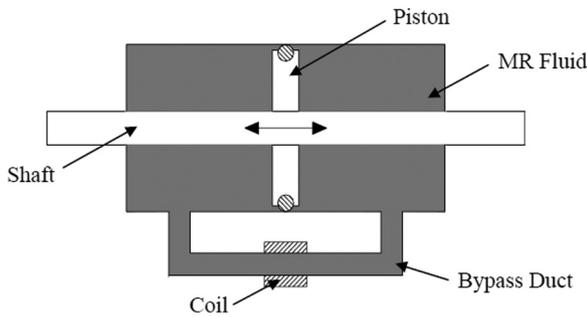
*Twin-tube-integrated MRF dampers.* Besides the mono-tube-integrated MRF dampers, several twin-tube-integrated MRF dampers have been developed in the literature, which incorporate the MRF control valve into the moving piston of a twin-tube MRF cylinder with the similar arrangement as that utilized in a monotube-integrated MRF damper (Chooi and



**Figure 15.** Schematic design of integrated MR fluid damper with external coils in MR fluid control valve: (a) damper with simultaneously moving piston and external coil and (b) damper with fixed external coil.

Source: Chen et al. (2010).

MR: magnetorheological.



**Figure 16.** Schematic of the bypass-type MR fluid damper.

Source: Unsal (2006).

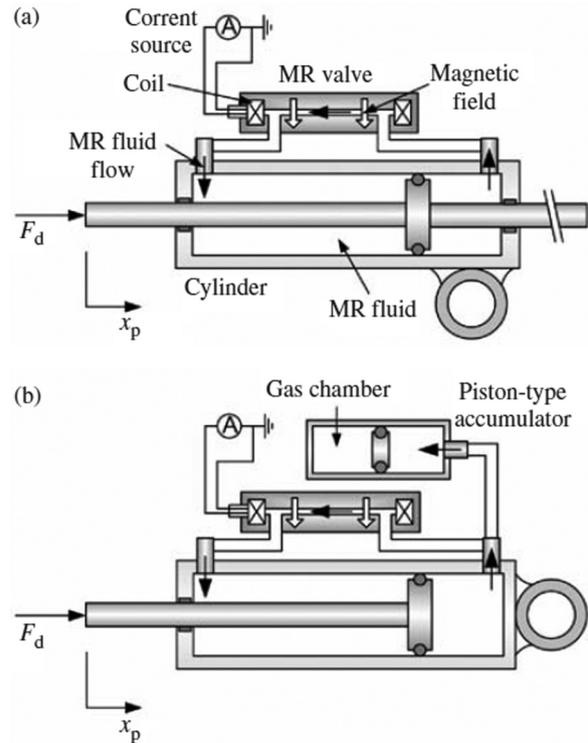
MR: magnetorheological.

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



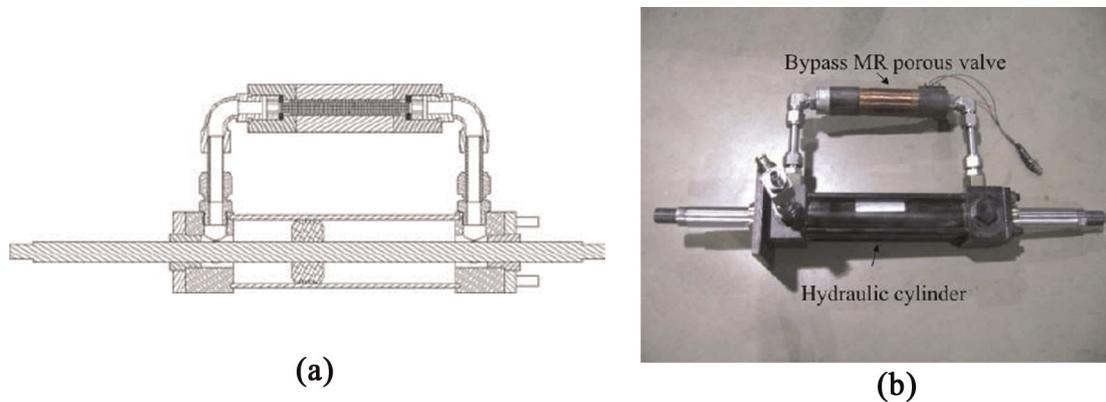
**Figure 17.** Schematic configurations of the bypass-type MR fluid shock dampers with internal coils: (a) double-ended MR shock damper and (b) single-ended MR shock damper.

Source: Nam and Park (2007).

MR: magnetorheological.

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).

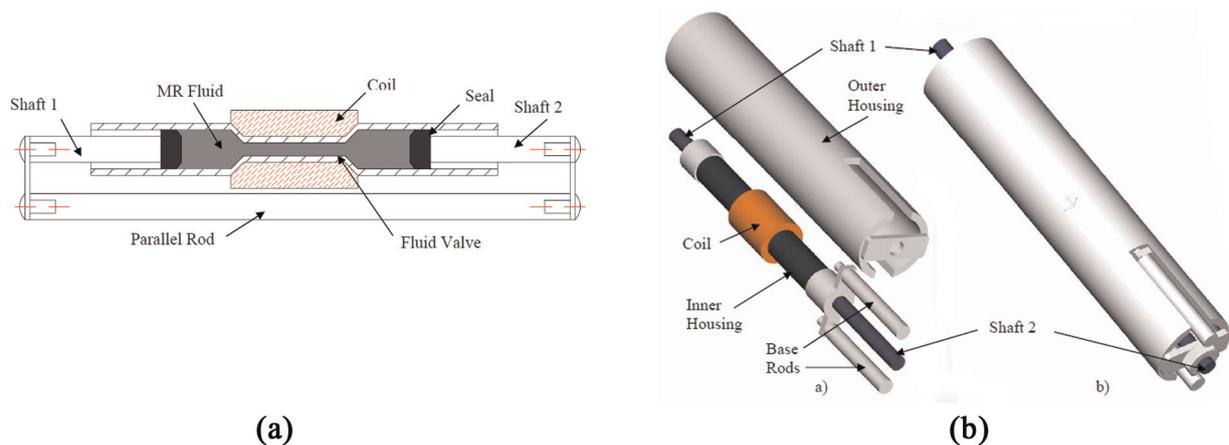
**Syringe-type MRF dampers.** A syringe-type MRF damper is shown in Figure 19. It consists of two syringes connected by a plastic hollow cylinder through which the fluid flows from one syringe to the other. Since no vacuum will be created when either of the shafts is pulled back, a parallel rod is connected 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



**Figure 18.** Schematic configuration and photo of the porous bypass-type MR fluid damper with external coils: (a) schematic structure and (b) photo.

Source: Hu et al. (2009).

MR: magnetorheological.



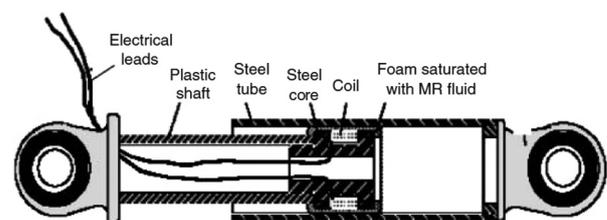
**Figure 19.** Syringe-type MR fluid damper: (a) syringe-type damper with parallel rod and (b) syringe-type damper with cylindrical outer housing.

Source: Unsal (2006).

MR: magnetorheological.

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



**Figure 20.** Schematic structure of the sponge-type MR fluid damper.

Source: Hoyle et al (2010).

MR: magnetorheological.

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 Carison, 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).

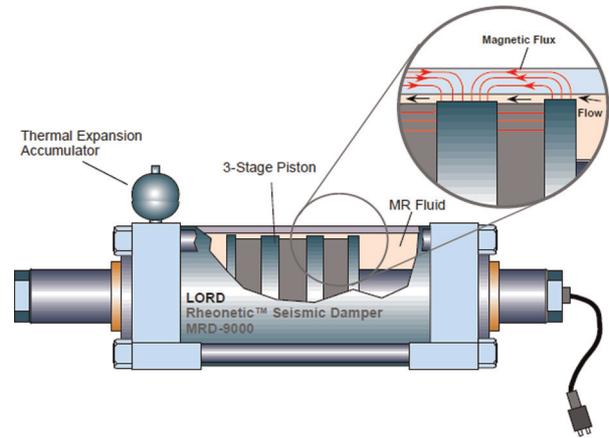
### Reinforced structure designs for performance improvement

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).

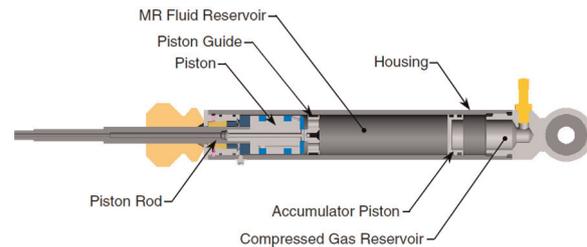
The multistage piston/coil MRF dampers are often favorable in enlarging damping force under a constraint



**Figure 21.** The prototype MR fluid damper with three-stage piston.

Source: Yang et al. (2002).

MR: magnetorheological.



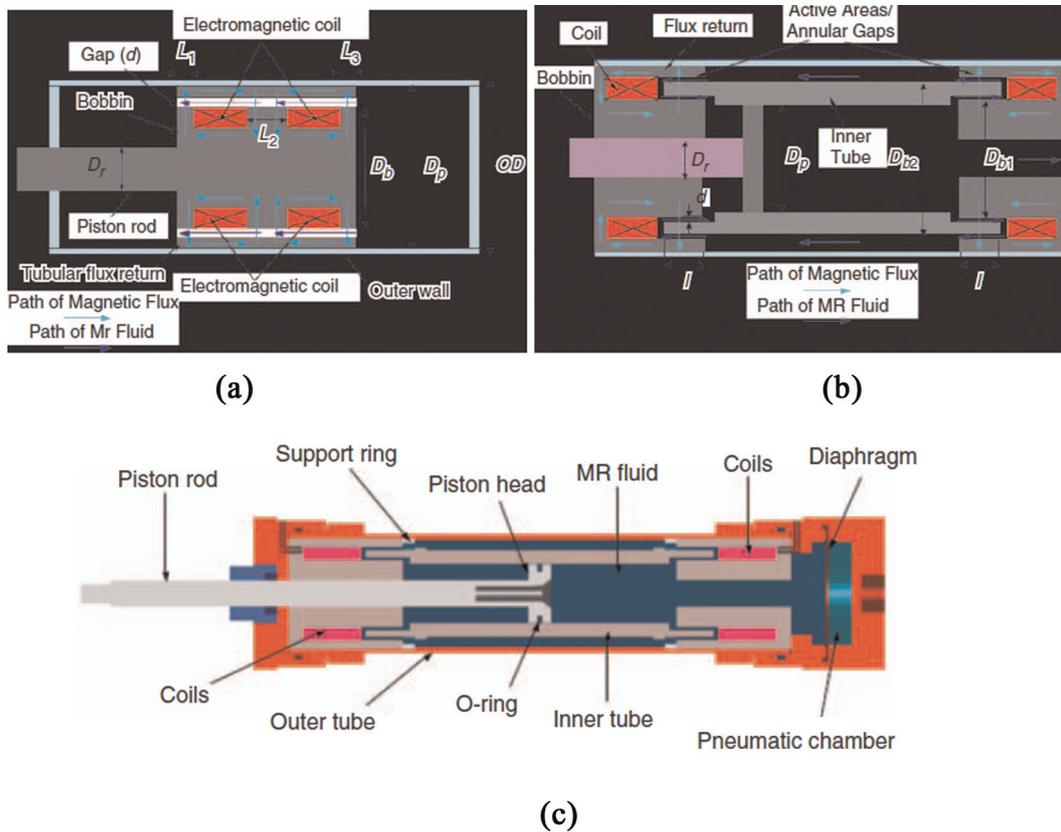
**Figure 22.** Expedition MR fluid damper with two-stage piston.

Source: Poynor (2001).

MR: magnetorheological.

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



**Figure 23.** Schematic diagrams of MR fluid damper with bifold configuration of electromagnet: (a) conventional design with MR fluid control valve inside the piston, (b) bifold design with MR fluid control valve at the end of the damper, and (c) section view. Source: Mao et al. (2007). MR: magnetorheological.

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.

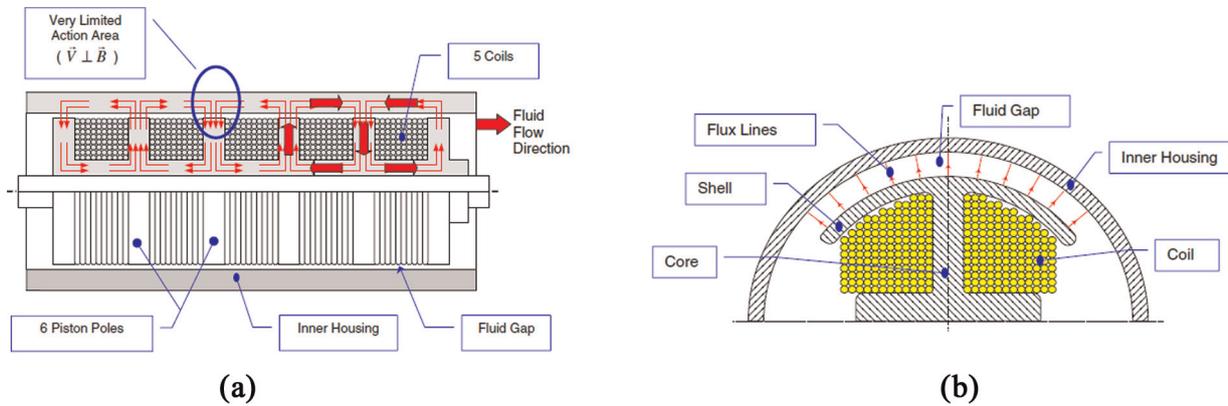
**Perpendicular coil axis configurations.** A novel MRF damper with perpendicular coil axis configuration (Figure 24) is proposed for providing large controllable damping force while only requiring a small amount of energy (Sassi et al., 2005). In contrast to conventional twin-tube MRF dampers, where 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.

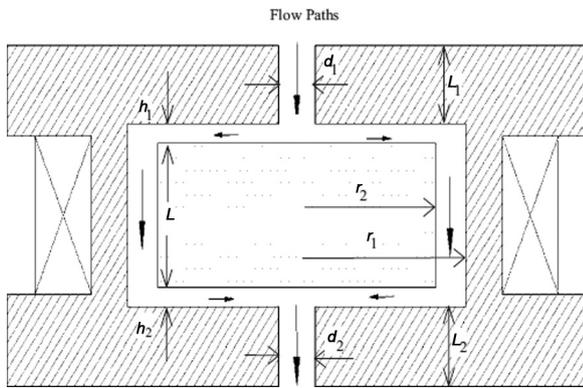
**Annular and radial fluid resistance gaps.** There could be some disadvantages in conventional MRF control valves that employs annular fluid resistance gap (i.e. the flow of the fluid is axial to the piston motion), such as small adjustable damping force, low utilization of magnetic field, and relatively 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



**Figure 24.** MR fluid damper with perpendicular coil axis configuration: (a) flux intensity and directions in a conventional MR fluid damper and (b) flux intensity and directions in the new MR fluid damper.

Source: Sassi et al. (2005).

MR: magnetorheological.



**Figure 25.** Schematic flow path of single-stage disk-type MR fluid control valve with radial fluid resistance gap.

Source: Gordaninejad et al. (2010).

MR: magnetorheological.

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).

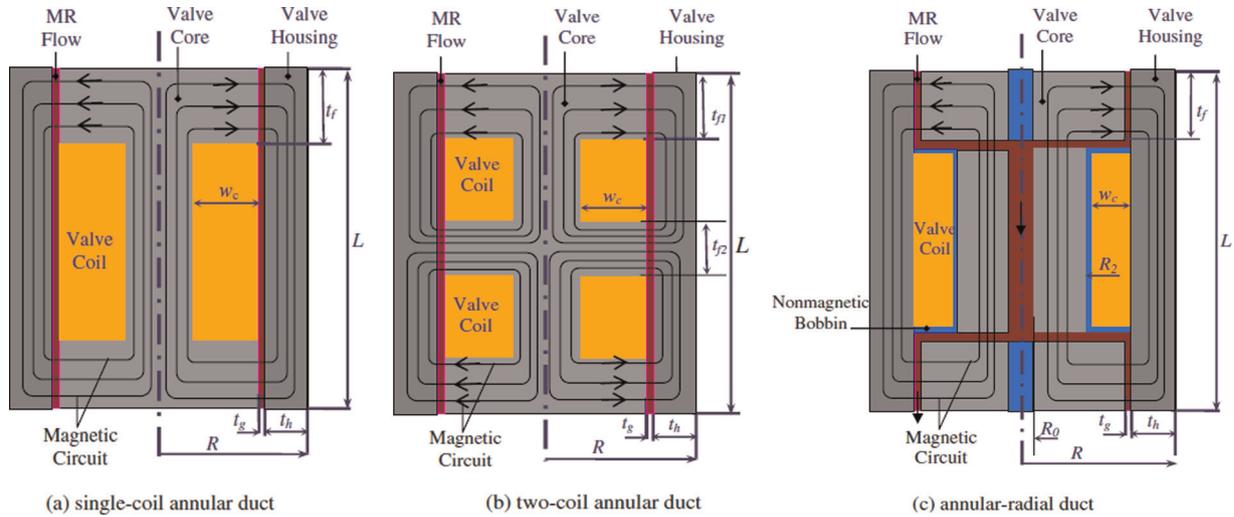
Since the efficiency of MRF control valves with circular disk-type fluid resistance gap is superior to that with annular fluid resistance gap under the same magnetic flux density and outer radius of the valve (Ai et al., 2006), an MRF control valve with both annular and radial fluid flow resistance gaps (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).

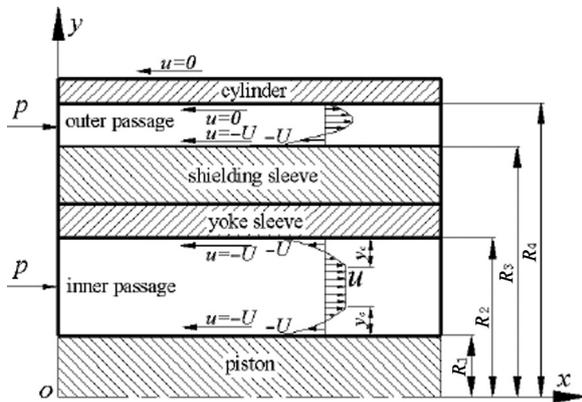
### Modification of electromagnetic designs

In order to improve the performance of MRF dampers, the magnetic field should be effectively supplied to the



**Figure 26.** Schematic flow path of MR fluid control valves with single-coil annular duct, two-coil annular duct, and annular-radius duct: (a) single-coil annular duct, (b) two-coil annular duct, and (c) annular-radius duct.

Source: Nguyen et al. (2008).  
MR: magnetorheological.



**Figure 27.** Schematic diagram and velocity profile of the two passages in an inner bypass MR fluid damper.

Source: Zhang et al. (2007).  
MR: magnetorheological.

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.

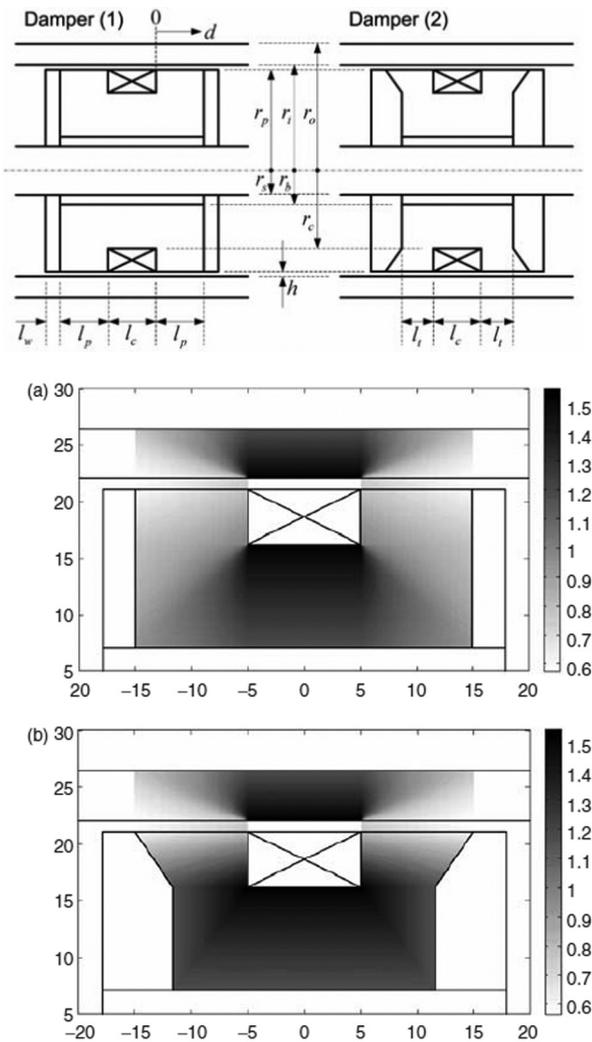
### Design of MRF dampers as a system

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)



**Figure 28.** Structural difference and magnetic flux density distributions between two types of damper design (Damper (1): conventional design; Damper (2): proposed electromagnetic design damper in the study by Nam and Park (2009)): (a) structure difference and (b) magnetic flux density distributions.

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 a 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

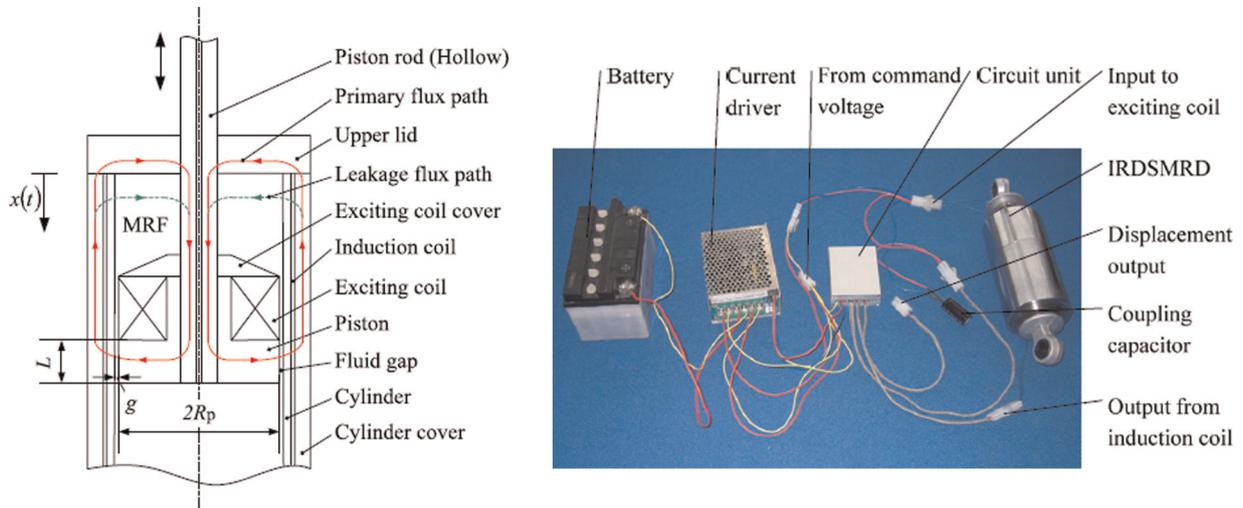
conventional actuation-only MRF damper (Figure 31) is developed in the study by Lam et al. (2010). Such systems have advantages in fulfilling real-time closed-loop feedback control applications for mitigating structural vibrations in a reliable and simple manner.

### Self-powered MRF damper systems

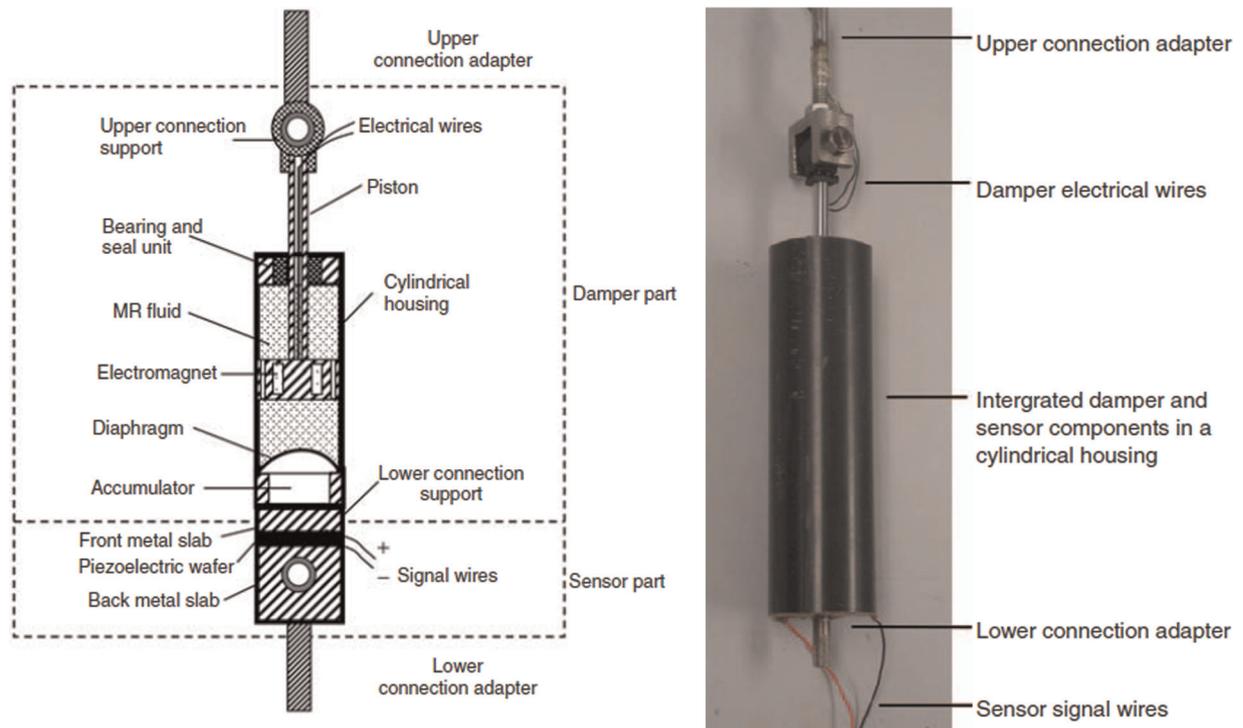
Usually, either a power supply or a current amplifier is required to activate the electromagnetic coils in MRF damper systems in practical applications. Since the mechanical energy resulting from vibration and shock motion in vibration systems could be employed in this power supply process, several methods 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



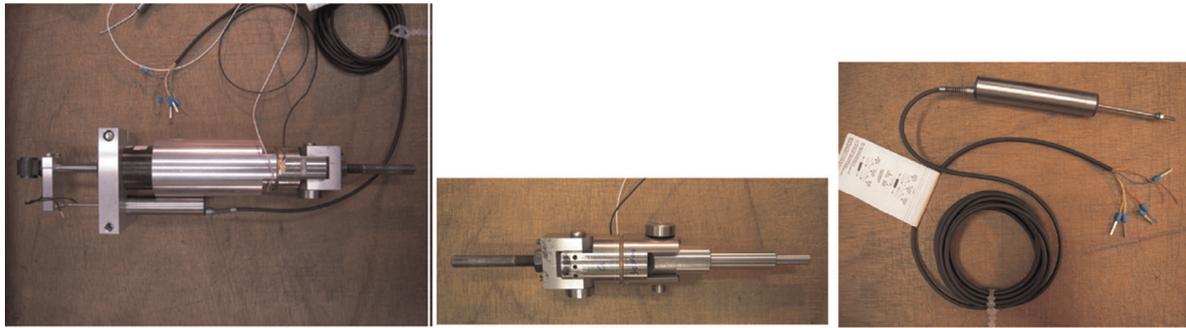
**Figure 29.** Schematic principle and photos of the integrated self-sensing MR damper.  
 Source: Wang and Wang (2009).  
 MR: magnetorheological.



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

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.,

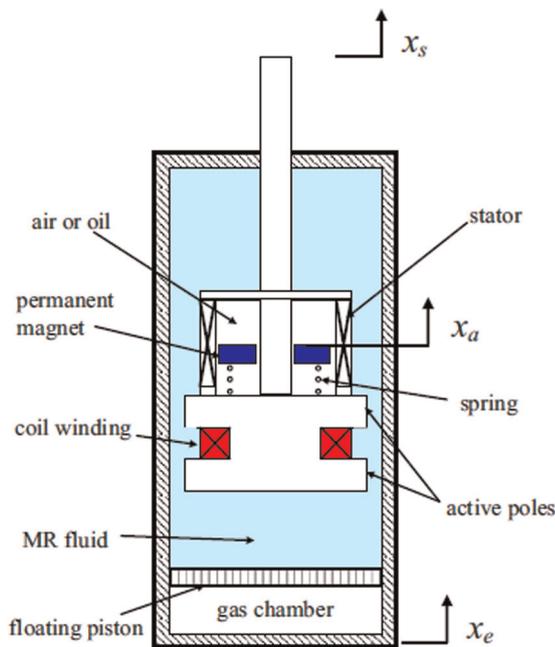
2005; Choi et al., 2007). In addition, a prototype of self-powered, self-sensing MR damper that composed of an MR damper part, a power generation part using stator and mover, a velocity sensor part using EMI, and an interaction and mounting part was designed recently (Figure 36) (Chen and Liao, 2010a).



**Figure 31.** Photograph of MR fluid damper with dual-sensing capability: (a) MR fluid damper with dual-sensing capability, (b) photograph of piezoelectric force sensor and (c) photograph of LVDT.

Source: Lam et al. (2010).

MR: magnetorheological; LVDT: linear variable differential transformer.



**Figure 32.** Schematic diagram of a self-powered MR fluid damper.

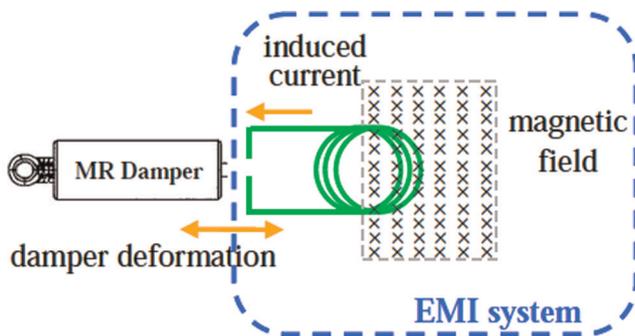
Source: Choi, Wereley (2009).

MR: magnetorheological.

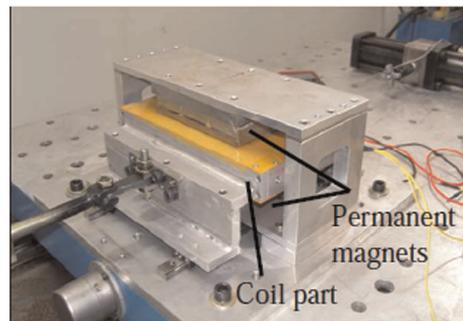
**Fail-safe MRF damper systems**

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



(a)

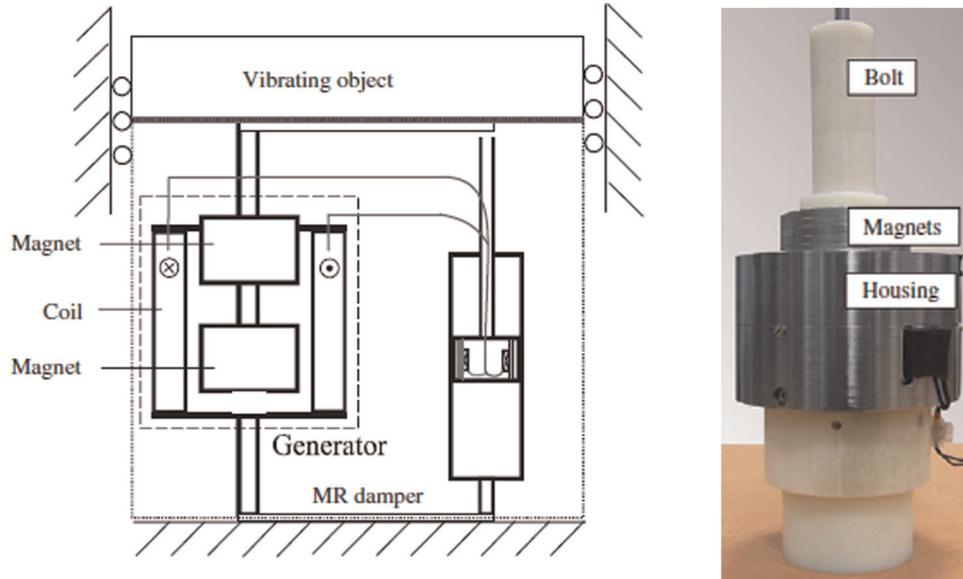


(b)

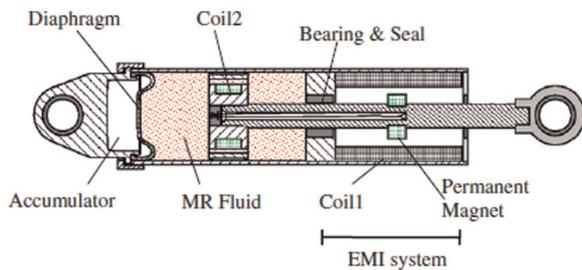
**Figure 33.** (a) Schematic of EMI system and (b) prototype of large-scale EMI system.

Source: Jung et al. (2009).

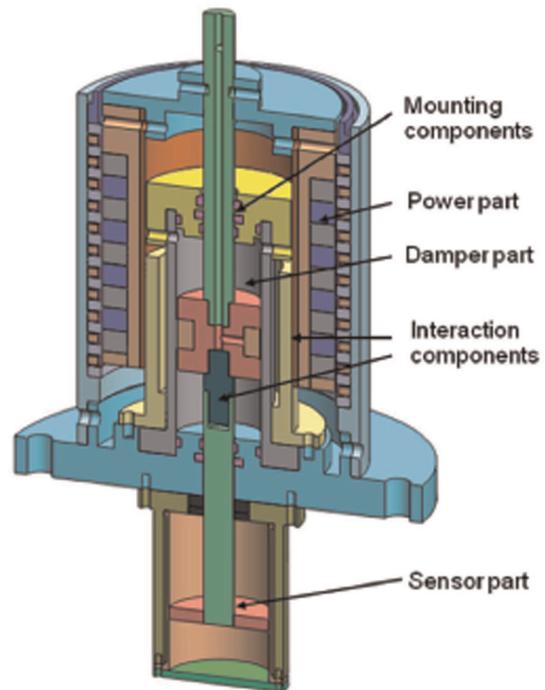
MR: magnetorheological; EMI: electromagnet induction.



**Figure 34.** Schematic diagram of the self-powered MR fluid damper–based vibration control system and the generator.  
 Source: Sapinski (2010).  
 MR: magnetorheological.



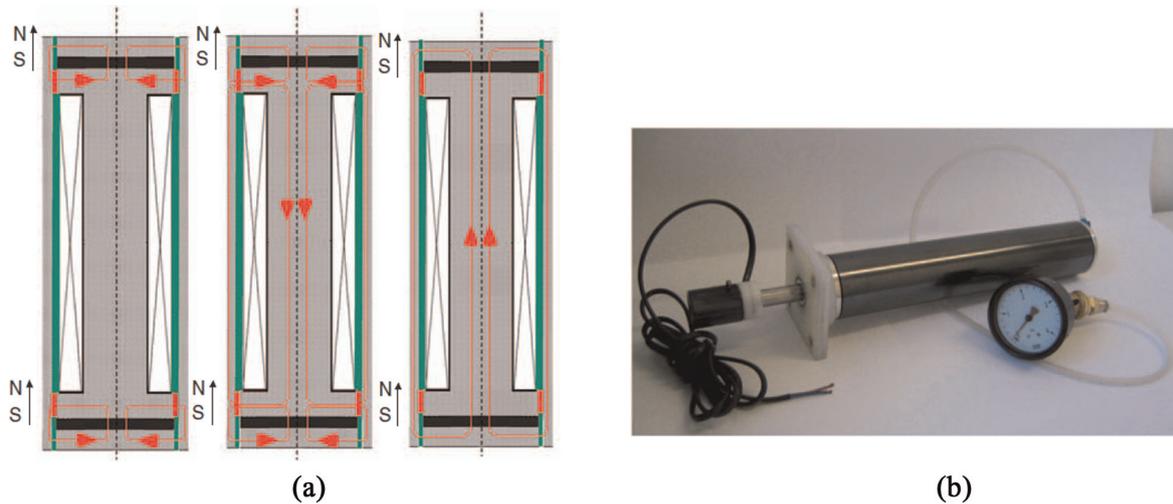
**Figure 35.** Schematic diagram of a smart passive system.  
 Source: Cho et al. (2005).  
 MR: magnetorheological; EMI: electromagnet induction.



**Figure 36.** Sectional view of self-powered, self-sensing MR damper.  
 Source: Chen and Liao (2010a).  
 MR: magnetorheological.

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 cancellation 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 semi-active and optimal passive dampers in the absence of control.

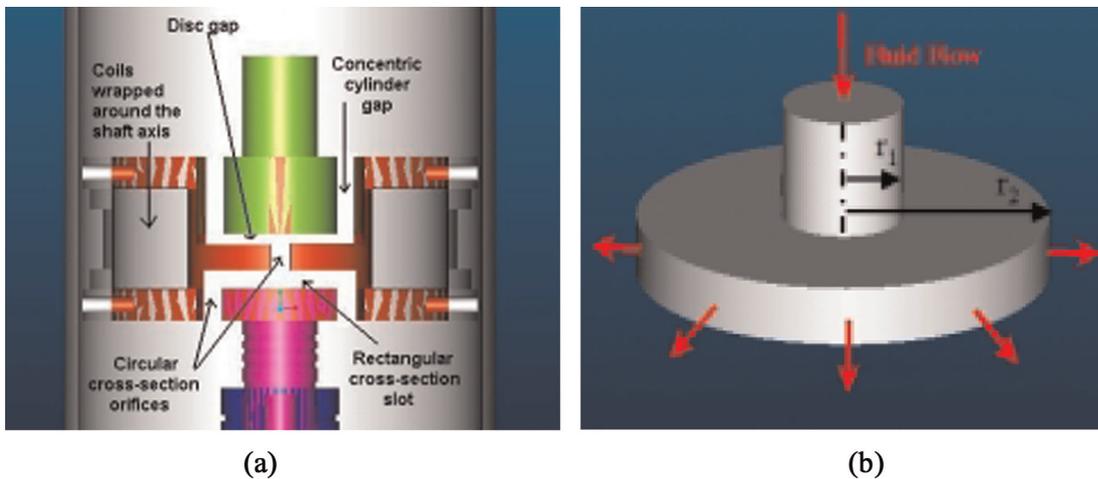
*Channel flow geometry design.* A bypass, fail-safe MRF damper with external electromagnet and several specially designed orifices in the piston (Figure 38(a)) is presented in the study by Gordaninejad et al. (2004). The disk-type orifice (Figure 38(b)) is regarded as a gap between two



**Figure 37.** Magnetic circuit in the piston and photographs of the fail-safe MR fluid damper with the permanent-electromagnet system. (a) Magnetic circuits in the piston in three different operation modes: without coil current (left), with positive coil current (middle), and with negative coil current (right). The annular MR fluid gap is green, the MR fluid control valves and magnetic flux lines are red, and the black disks are the permanent magnets and (b) photographs.

Source: Boese and Ehrlich (2010).

MR: magnetorheological.



**Figure 38.** Cross-sectional view of the piston of a fail-safe MR fluid damper and schematic drawing of a disk-type orifice: (a) cross-sectional view of the piston and (b) schematic drawing of disk-type orifice.

Source: Gordaninejad et al. (2004).

MR: magnetorheological.

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.

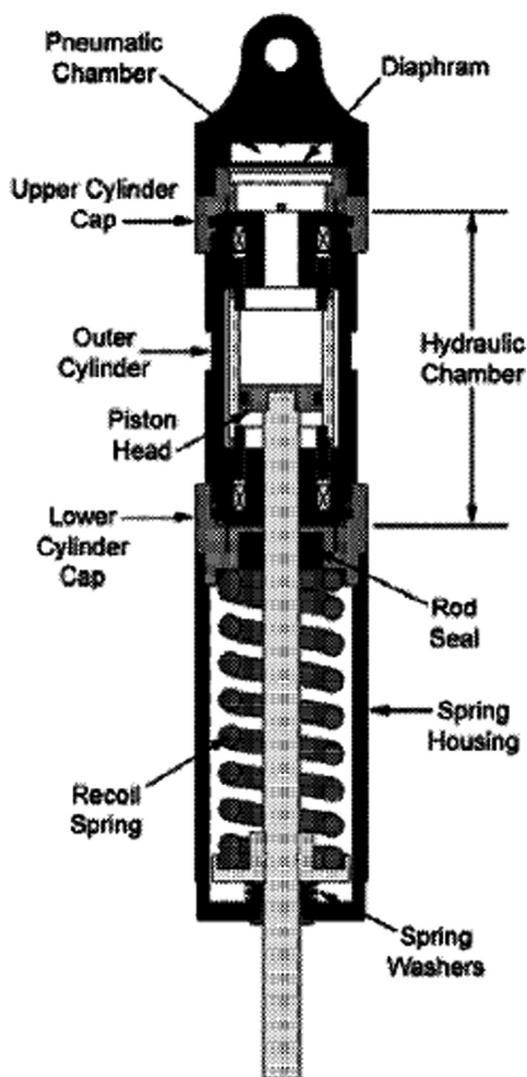
*MRF dampers integrated with coil springs.* An MRF damper that replaced the original equipment manufacturer (OEM) dampers around by a coil spring is naturally designed in application. Specifically, MRF dampers around by coil springs for shock absorber application (Figure 39) are presented in the study by Breese and



**Figure 39.** UNR-MRD-B2 rear damper around by coil spring.  
Source: Breese and Gordaninejad (2003).

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 housing 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.

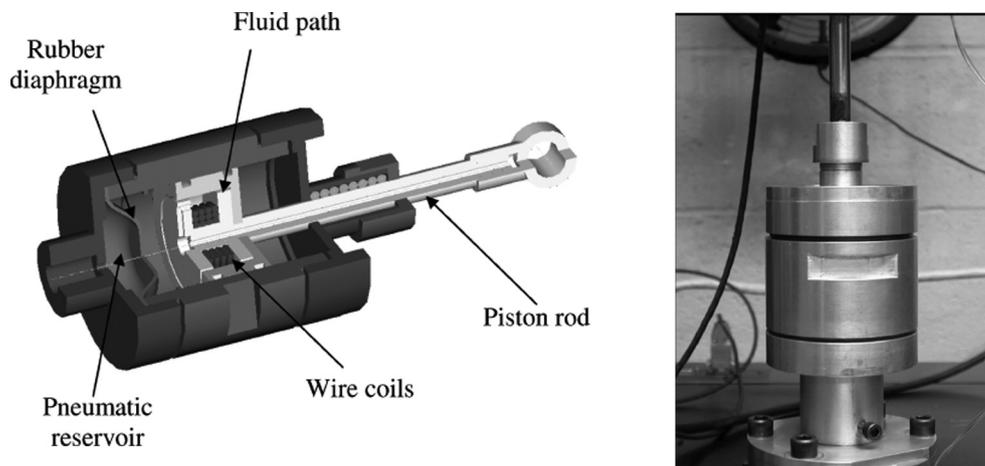
*MRF dampers integrated with liquid springs.* In view of large bulk modulus of hydraulic system that can produce considerable spring effect, a bypass liquid spring MR shock absorber (Figure 42) that can generate large spring rate and controllable damping through designing different diameter of rods on either side of the piston was proposed in the study by Hong et al. (2006) for use as a compact automotive



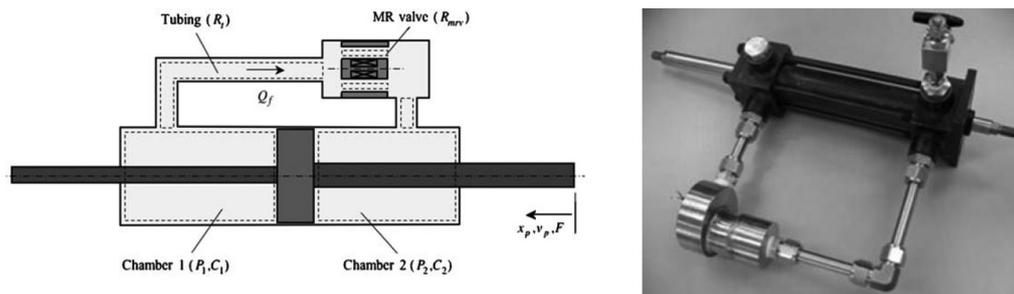
**Figure 40.** MR fluid damper with spring mechanisms in place.  
Source: Facey et al. (2005).  
MR: magnetorheological.

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



**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.



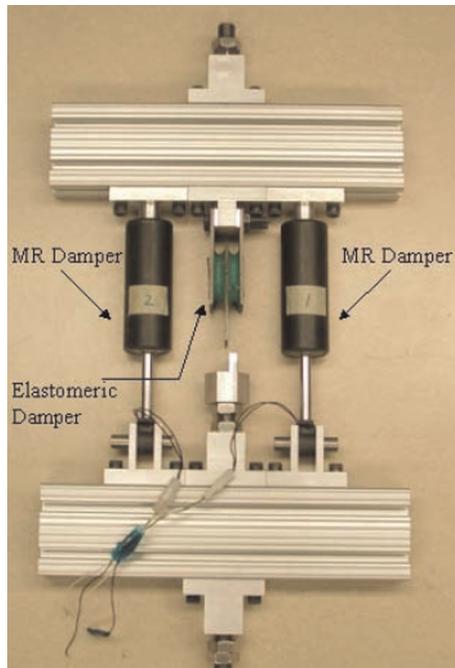
**Figure 42.** Hydromechanical model of the compressible magnetorheological strut and fabricated compressible MR fluid strut.  
Source: Hong et al. (2006).  
MR: magnetorheological.

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).

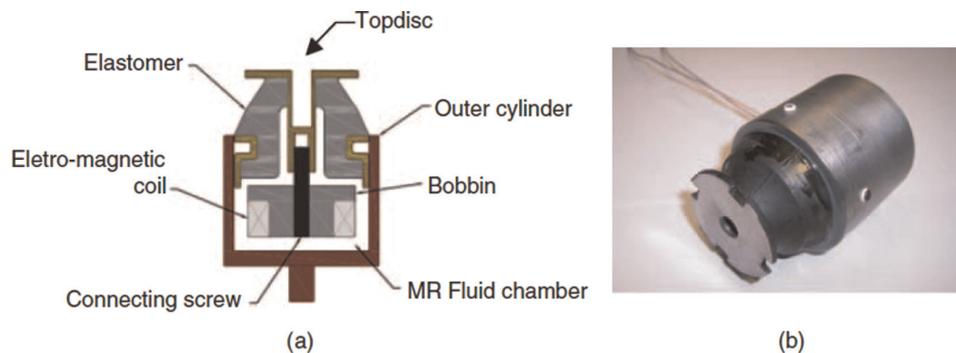
MRF dampers based on the passive mount utilizing rubber elements have also been adopted to support a static load and to isolate a dynamic load over high- and postresonance frequency range. Specifically, a mixed-mode MR mount that can be operated under both flow and shear modes was constructed through 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.,



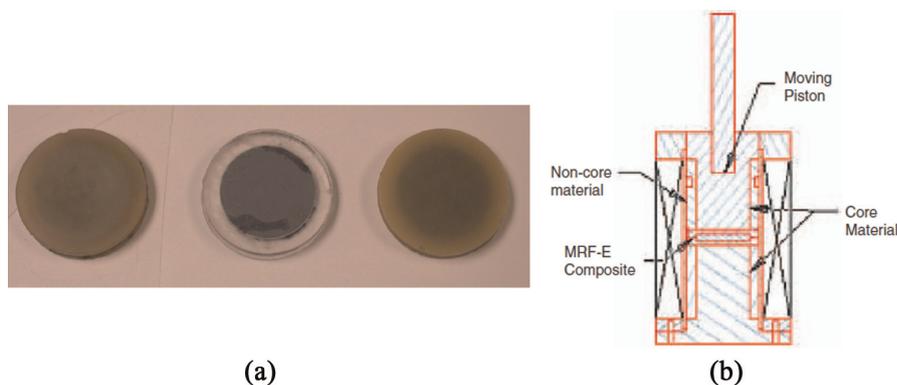
**Figure 43.** Test fixture for a hybrid damper.  
Source: Hu and Wereley (2008).

2005). 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).

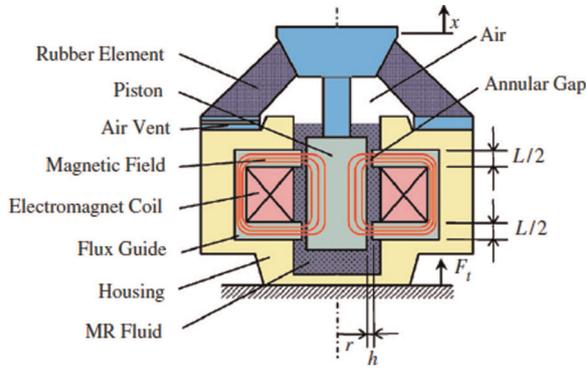
*MR dampers integrated with air springs.* Considering low natural frequency and high load capacity inherited from air spring as well as simplicity of MR technology, vibration isolators consisting of air springs and MRF dampers have been presented for application in vehicle suspensions and civil constructions (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 44.** The multimode MR isolator: (a) schematic diagram and (b) fabricated isolator.  
Source: Brigley et al. (2007).  
MR: magnetorheological.



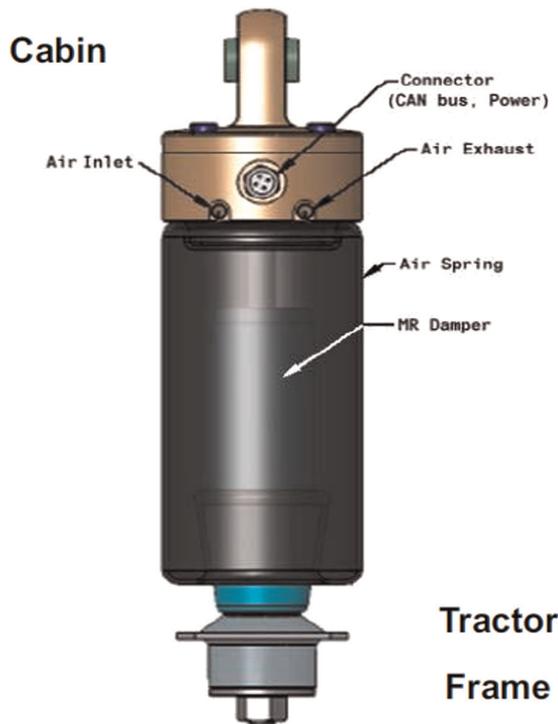
**Figure 45.** MR fluid–elastomer composite samples and schematic of MR fluid–elastomer vibration isolator design: (a) MR fluid–elastomer composite samples and (b) schematic diagram of MR fluid–elastomer vibration isolator design.  
Source: Wan and Gordaninejad (2009).  
MR: magnetorheological.



**Figure 46.** Schematic configuration and photos of the MR mixed-mode mount.

Source: Choi et al. (2008).

MR: magnetorheological.



**Figure 47.** Integrated MR suspension prototype.

Source: Achen et al. (2008).

MR: magnetorheological.

### Performance analysis and parameter optimization of MRF damper systems

Obviously, 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.

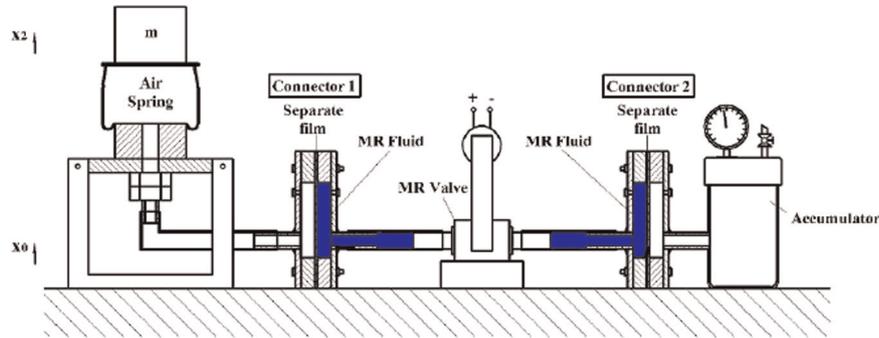
In practice, the complex stiffness ( $K^*$ ) is utilized to quantitatively characterize the viscoelastic behavior of the MRF damper as (Pang et al., 1998)

$$K^* = K' + jK'' = K'(1 + j\eta_{lost}) \tag{1}$$

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

$$K' = \frac{F_c X_c + F_s X_s}{X_c^2 + X_s^2} \tag{2a}$$

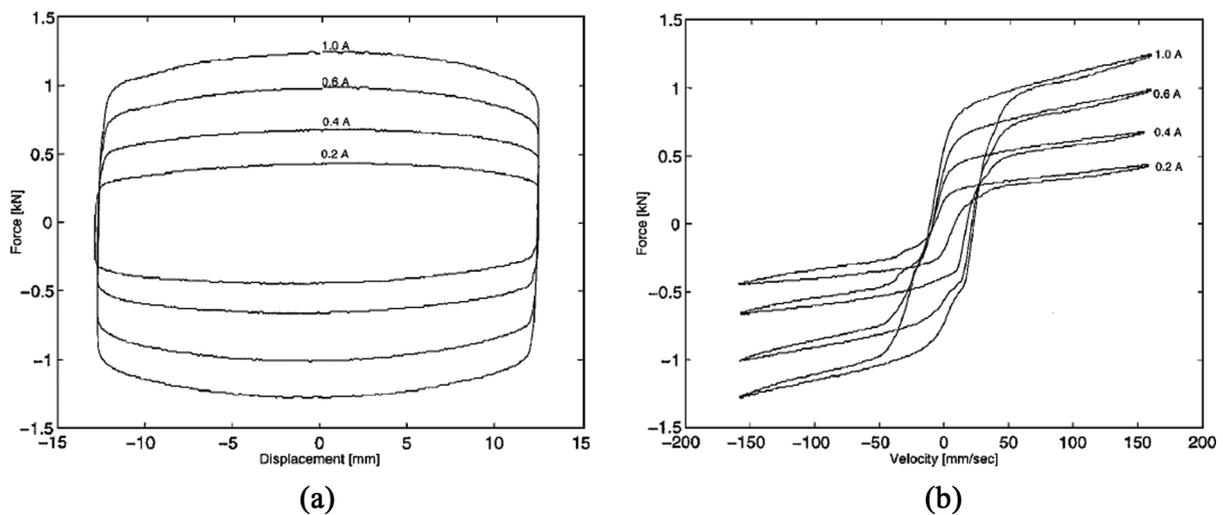
$$K'' = \frac{F_c X_s - F_s X_c}{X_c^2 + X_s^2} \tag{2b}$$



**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.

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

$$C_{eq} = \frac{E}{\pi \omega X_p^2} \tag{3}$$

where  $E = \int_{\text{cycle}} f(t) dX_p = \int_0^{2\pi/\omega} f(t) V_p dt$  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)

$$T = \frac{L_{in}}{R_w} \tag{4}$$

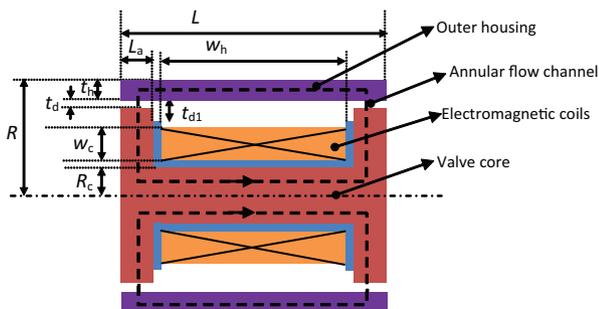
where  $L_{in}$  is the inductance of the coil in the driving electronic system given by  $L_{in} = N\Phi/I$ ,  $N$  is the number of coil turns,  $I$  is the applied coil current,  $\Phi$  is the magnetic flux, and  $R_w$  is the resistance of the coil wire given by

$R_w = N/I^2 = L_w r_w = N \pi \bar{d}_c (r_{coil}/A_w)$  in which  $L_w$  is the length of the coil wire,  $r_w$  is the resistance per unit length of the coil wire,  $\bar{d}_c$  is the average diameter of the coil,  $A_w$  is the cross-sectional area of the coil wire, and  $r_{coil}$  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).

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  $t_d$ , 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)



**Figure 50.** Geometry of a conventional MR valve with one coil. MR: magnetorheological.

$$E = IR_w^2 \quad (5)$$

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

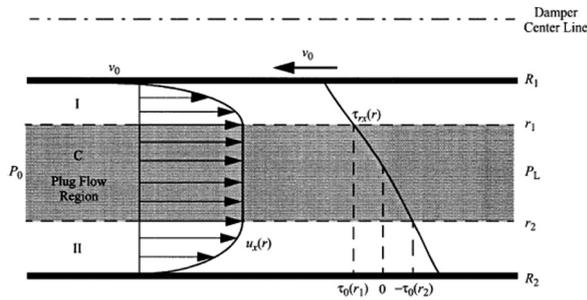
A reliable mathematical model is very important to system performance assessment (such as damping characteristics) and parameter optimization. Several mathematical models have been proposed for MRF damper systems in the literature (Wang and Liao, 2011). Physical models are discussed only analytically 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).

$$\begin{cases} \tau = \tau_0(H) \operatorname{sgn}(\dot{\gamma}) + \eta \dot{\gamma} & \tau > \tau_0 \\ \dot{\gamma} = 0 & \tau \leq \tau_0 \end{cases} \quad (6)$$

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;  $\dot{\gamma} = du_x(r)/dr$ ;  $u_x(r)$  is flow velocity with the boundary condition  $u_x(R_1) = -v_0$ ;  $v_0$  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

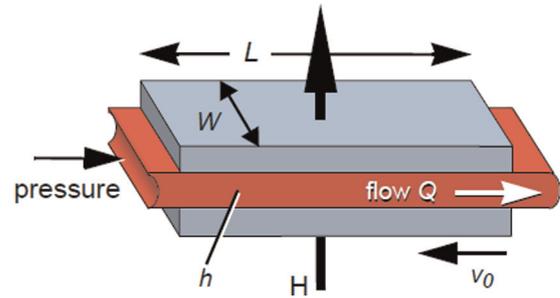


**Figure 51.** A typical stress and velocity profile in an annular gap.  
Source: Yang et al. (2002).

shown in Figure 51. As seen, there are three different flow regions whose boundary is dependent on a plug thickness ( $\delta = r_2 - r_1 \approx -2\tau_0 / (dp(x)/dx)$ ) 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 combing 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 biviscous 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–Bulkley 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



**Figure 52.** MR fluid flow through a parallel duct.  
Source: Yang et al. (2002).  
MR: magnetorheological.

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)

$$3(P - 2\Gamma)^2 [P^3 - (1 + 3\Gamma - V)P^2 + 4\Gamma^3] + \Gamma V^2 P^2 = 0, \quad |V| < 3(P - 2\Gamma)^2 / P \quad (7)$$

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,  $p_c'$ , 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,  $p = p' / p_n'$ ,  $\Gamma = p_c' / 2p_n'$ ,  $V = Q_s / Q$ ,  $p_n' = 12Q\eta / wh^3$ ,  $p' = -dp/dx$ ,  $p_c' = 2\tau_0 / h$ , 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).

$$P(\Gamma, V) = 1 + 2.07\Gamma - V + \frac{\Gamma}{1 + 0.4\Gamma} \quad (8)$$

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

$$F_\eta = \left(1 + \frac{whv_0}{2Q}\right) \frac{12\eta QLA_p}{wh^3} \quad (9a)$$

$$F_{\tau} = c \frac{\tau_0 L A_p}{h} \operatorname{sgn}(v_0) \quad (9b)$$

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.  $c = 2.07 + 1/(1 + 0.4\Gamma) = 2.07 + (12Q\eta/12Q\eta + 0.4wh^2\tau_0)$  is an 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 compacts 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. The definitions and formulations of these nondimensional variables based on Bingham plastic constitutive model are listed in Tables 2 and 3 (Hong et al., 2005; Hong, Wereley, Choi, et al. 2008; Wereley and Pang, 1998; Yoo and Wereley, 2005).

Some practical implications for these nondimensional variables should be noted (Hong, Wereley, Choi, et al. 2008):

1. Hydraulic amplification ratio  $\bar{A}$  is used to connect the piston velocity  $v_0$  and the average duct velocity of the fluid inside the gap  $V_d$  due to different geometry configurations. Geometric ratio

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

Configurations	Passive and active damping force	
Valve mode	Rectangular annular channel	$F_{\eta} = \frac{12\eta L Q}{wh^3} A_p \quad (10a)$
		$F_{\tau} = \frac{c\tau_0}{h} \operatorname{sgn}(v_0) A_p \quad (10b)$
	Cylindrical-shaped annular channel	$F_{\eta} = \frac{8\eta L Q}{\pi(d_0/2)^4} A_p \quad (11a)$
		$F_{\tau} = \frac{c\tau_0}{d_0} \operatorname{sgn}(v_0) A_p \quad (11b)$
	Radial flow resistance channel	$F_{\eta} = \frac{6\eta Q}{\pi h^3} \ln \frac{R_0}{r_0} A_p \quad (12a)$
		$F_{\tau} = \frac{c\tau_0}{h} (R_0 - r_0) \operatorname{sgn}(v_0) A_p \quad (12b)$
Direct shear mode	$F_{sh, \eta} = \frac{\eta w L}{h} v_0 \quad (13a)$	
	$F_{sh, \tau} = w L_a \tau_0 \operatorname{sgn}(v_0) \quad (13b)$	
Squeeze mode	$F_{sq, \eta} = \frac{3\pi\mu r_p^4}{2(x_0 + x)^3} v_0 \quad (14a)$	
	$F_{sq, \tau} = \frac{4\pi r_a^3 \tau_{0, sq}}{3(x_0 + x)} \operatorname{sgn}(v_0) \quad (14b)$	

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.

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

Name of variables	Definition	Interactive relationship
Hydraulic amplification/geometry ratio	$\bar{A} = \frac{A_p}{A_d}, \phi_r = \frac{r_p}{h}$	$A_p = \pi r_p^2, A_d = wh = 2\pi r_d h \quad (15)$ $Q = Q_p = A_p v_0 = Q_d = A_d V_d \quad (16)$
Bingham number	$Bi_{vp} = \frac{\tau_0 h}{\eta v_0}, Bi_{vd} = \frac{\tau_0 h}{\eta V_d}$	$Bi_{vp, mix} = 6\bar{\delta} \left(\frac{1}{2} + \bar{A}\right) \frac{1 + \sqrt{1 - \bar{\delta}/6(1 + \bar{\delta}/2)(\bar{A} + 1/2)^{-2}}}{2(1 - \bar{\delta})^2(1 + \bar{\delta}/2)} \quad (17)$
Nondimensional plug thickness	$\bar{\delta} = \frac{\delta}{h}$	$Bi_{vp, flow} = 6\bar{\delta}\bar{A} \frac{1}{(1 - \bar{\delta})^2(1 + \bar{\delta}/2)} \quad (18a)$ $Bi_{vp} = \bar{A}Bi_{vd} \quad (18b)$

MRF: magnetorheological fluid.

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

Name of variables	Formulations for damper performance
Equivalent damping coefficient	$C_{eq} = \frac{F}{V_p} = \frac{A_p \Delta P}{V_p} = \frac{2\eta L A_p}{d^2} \frac{Bi_{vp}}{\bar{\delta}} \quad (19)$
Newtonian viscous damping coefficient	$C_{0, mix} = \frac{12\eta A_p L}{h^2} \left(\frac{1}{2} + \bar{A}\right) \quad (20a)$ $C_{0, flow} = \frac{12\eta A_p L}{h^2} \bar{A} \quad (20b)$
Damping capacity	$\frac{C_{eq}}{C_{0, mix}} = \frac{1 + \sqrt{1 - \bar{\delta}/6(1 + \bar{\delta}/2)(\bar{A} + 1/2)^{-2}}}{2(1 - \bar{\delta})^2(1 + \bar{\delta}/2)} \quad (21a)$ $\frac{C_{eq}}{C_{0, flow}} = \frac{1}{(1 - \bar{\delta})^2(1 + \bar{\delta}/2)} \quad (21b)$
Nondimensional damping force and dynamic range	$\phi_D = \frac{F_{mr}}{F_r} = \frac{a_1 + a_2 \Gamma - a_3 V}{a_1 - a_3 V} = 1 + \frac{a_2}{6} \frac{\phi_c}{a_1 \phi_r + a_3} \quad (22a)$ $\phi_F = a_1 \phi_r^3 + \left(\frac{a_2}{6} \phi_c + a_3\right) \phi_r^2 \quad (22b)$

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).

$\phi_r$ , proportional to  $\bar{A}$ , can be used alternatively for the integrated MRF damper with MRF control valve in the moving piston due to  $r_p \approx r_d$ . 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.

- Two Bingham numbers in terms of different velocities have different applications. It is preferable to use  $Bi_{vp}$  for analyzing damper

performances with consideration of external excitations due to its incorporated explicit measurable piston velocity and, on the other hand, to use  $Bi_{vd}$  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 non-dimensional 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 loading, the piston velocity for the Bingham number is given by a root mean square (RMS) value of the velocity ( $V_{p,rms} = \left(\frac{1}{2\pi/\omega} \int_0^{2\pi/\omega} v_0^2 dt\right)^{1/2} = \frac{V_a}{\sqrt{2}}$ , where  $V_a$  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 ( $\bar{\tau}$ ) 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)

$$NI = \int_c H dl = H_f l_f + H_s l_s \quad (23)$$

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

$$\Phi = \int_s B ndS = B_f S_f = B_s S_s \quad (24)$$

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,  $B_s = \mu_0 \mu_s H_s$  and  $B_f = \mu_0 \mu_f H_f$ , in which  $\mu_0$  is the constant magnetic permeability of free space, and  $\mu_s$  and  $\mu_f$  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)

$$B_{s,sat} \geq \frac{\mu_0 NI}{S_s l_f / (S_f \mu_f) + l_s / \mu_s} \quad (25a)$$

$$H_f = \frac{NI}{l_f + \mu_f S_f l_s / (\mu_s S_s)} \quad (25b)$$

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)

$$wh^2 = \frac{12}{c} \left( \frac{\eta}{\tau_0} \right) \lambda Q \quad (26a)$$

$$h = \frac{\eta}{\tau_0} \lambda v_0 \quad (26b)$$

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)

$$V_{\min} = Lwh = k \left( \frac{\eta}{\tau_0^2} \right) \lambda W_m \quad (27)$$

where  $k = 12/c^2$ ,  $\lambda = \Delta P_\tau / \Delta P_\eta$ , and  $W_m = Q\Delta P_\tau$  for flow mode and  $k = 1$ ,  $\lambda = F_\tau / F_\eta$ , and  $W_m = v_0 F_\tau$  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,  $V_{\text{valve}} \approx (25 \sim 50) V_{\min}$  (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 a 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 antiferce 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 conversational single-MRF damper system, involve the sensor-based MRF damper systems that integrate displacement sensors or force sensors into the conversational 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 antiferce 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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### References

- Achen A, Toscano J, Marjoram R, et al. (2008) Semi-active vehicle cab using magnetorheological (MR) technology. In: *Proceedings of the 7th JFPS international symposium on fluid power*, Toyama, Japan, 15–18 September 2008.
- Ahmadian M and Norris JA (2008) Experimental analysis of magnetorheological dampers when subjected to impact and shock loading. *Communications in Nonlinear Science and Numerical Simulation* 13: 1978–1985.
- Ahmadkhanlou F (2008) *Design, modeling and control of magnetorheological fluid-based force feedback dampers for tele-robotic systems*. Doctor Thesis, The Ohio State University, Columbus, OH.
- Ahn YK, Yang BS, Ahmadian M, et al. (2005) A small-sized variable-damping mount using magnetorheological fluid. *Journal of Intelligent Material Systems and Structures* 16: 127–133.
- Ai HX, Wang DH and Liao WH (2006) Design and modeling of a magnetorheological valve with both annular and radial flow paths. *Journal of Intelligent Material Systems and Structures* 17: 327–334.
- Aydar G (2006) *A new magneto-rheological fluid (MRF) washing machine damper and a novel two-way-controllable MRF valve*. Master Thesis, University of Nevada, Reno, NV.
- Aydar G, Wang X and Gordaninejad F (2010) A novel two-way-controllable magneto-rheological fluid damper. *Smart Materials and Structures* 19: 065024 (7pp.).
- Bangrakulur KK (2004) *A seismic magneto-rheological (MR) fluid by-pass damper*. Master Thesis, University of Nevada, Reno, NV.
- Batterbee DC, Sims ND, Stanway R, et al. (2007a) Magnetorheological landing gear 1: a design methodology. *Smart Materials and Structures* 16: 2429–2440.
- Batterbee DC, Sims ND, Stanway R, et al. (2007b) Magnetorheological landing gear 2: validation using experimental data. *Smart Materials and Structures* 16: 2441–2452.
- Boese H and Ehrlich J (2010) Performance of magnetorheological fluids in a novel damper with excellent fail-safe behavior. *Journal of Intelligent Material Systems and Structures* 21: 1537–1542.
- Breese DG and Gordaninejad F (2003) Semi-active, fail-safe magneto-rheological fluid dampers for mountain bicycles. *International Journal of Vehicle Design* 33: 1–18.
- Brigley M, Choi YT and Wereley NM (2008) Experimental and theoretical development of multiple fluid mode magnetorheological isolators. *Journal of Guidance, Control and Dynamics* 31(3): 449–459.
- Brigley M, Choi YT, Wereley NM, et al. (2007) Magnetorheological isolators using multiple fluid modes. *Journal of Intelligent Material Systems and Structures* 18: 1143–1148.
- Carlson JD (1999) Low-cost MR fluid sponge devices. *Journal of Intelligent Material Systems and Structures* 10: 589–594.
- Carlson JD (2001) Magnetorheological fluid seismic damper. US patent 6(296): 088 B1.
- Carlson JD (2005) MR fluids and devices in the real world. *International Journal of Modern Physics B: Condensed Matter Physics, Statistical Physics, Applied Physics* 19(7–9): 1463–1470.
- Carlson JD (2009) *Magnetorheological fluids*, in *Smart Materials*, CRC Press 2009, Taylor & Francis Group, LLC. eBook ISBN: 978-1-4200-4373-0
- Carlson JD and Jolly MR (2000) MR fluid, foam and elastomer devices. *Mechatronics* 10: 555–569.
- Carlson JD and Spencer BF Jr (1996) Magneto-rheological fluid dampers: scalability and design issues for application to dynamic hazard mitigation. In: *Proceedings of the 2nd international workshop on structural control*, Hong Kong, University of Science and Technology Research Centre, December 1996, pp.99–109.
- Carlson JD, Toscano JR and Matthis W (2001) Smart prosthetics based on magnetorheological fluids. *Proceedings of SPIE* 4332: 308.
- Carmignani C, Forte P and Rustighi E (2006) Design of a novel magneto-rheological squeeze-film damper. *Smart Materials and Structures* 15: 164–1701.
- Chen C and Liao WH (2010a) A self-powered, self-sensing magnetorheological damper. In: *Proceedings of the 2010 IEEE international conference on mechatronics and automation*, Xi'an, China, 4–7 August 2010, pp.1364–1369.
- Chen JZ and Liao WH (2010b) Design, testing and control of a magnetorheological actuator for assistive knee braces. *Smart Materials and Structures* 19: 035029 (10pp.).
- Chen L (2009) Using magnetorheological (MR) fluid as distributed actuators for smart structures. In: *IEEE, ICIEA 2009*, 25–27 May, Adelaide University, Australia, pp.1203–1208.
- Chen YC, Lin YH and Pu JP (2010) Study on the performance of magnetorheological damping performance. *Journal of Engineering* 21(1): 11–25 (in Chinese).
- Cho SW, Jung HJ and Lee IW (2005) Smart passive system based on magnetorheological damper. *Smart Materials and Structures* 14: 707–714.
- Choi KM, Jung HJ, Lee HJ, et al. (2007) Feasibility study of an MR damper-based smart passive control system employing an electromagnetic induction device. *Smart Materials and Structures* 16: 2323–2329.
- Choi SB, Hong SR, Sung KG, et al. (2008) Optimal control of structural vibrations using a mixed-mode magnetorheological fluid mount. *International Journal of Mechanical Sciences* 50: 559–568.

- Choi YT and Wereley NM (2009) Self-powered magnetorheological dampers. *Journal of Vibration and Acoustics* 131: 044501–044501.
- Choi YT, Wereley NM and Jeon YS (2005) Semi-active vibration isolation using magnetorheological isolators. *AIAA Journal of Aircraft* 42(5): 1244–1251.
- Chooi WW and Oyadiji SO (2008) Design modelling and testing of magnetorheological (MR) dampers using analytical flow solutions. *Computers and Structures* 86: 473–482.
- Chooi WW and Oyadiji SO (2009a) Experimental testing and validation of a magnetorheological (MR) damper model. *Journal of Vibration and Acoustics* 131: 061003–061001.
- Chooi WW and Oyadiji SO (2009b) Mathematical modeling analysis and design of magnetorheological(MR) dampers. *Journal of Vibration and Acoustics* 131: 061002–061001.
- Chrzan MJ and Carlson JD (2001) MR fluid sponge devices and their use in vibration control of washing machines. *Smart Structures and Materials: Damping and Isolation* 4331: 370.
- Cook E, Hu W and Wereley NM (2007) Magnetorheological bypass damper exploiting flow through a porous channel. *Journal of Intelligent Material Systems and Structures* 18: 1197–1203.
- Craft MJ (2003) *Design optimization of MagneShock™ magnetorheological shock absorbers and development of fuzzy logic control algorithms*. Master Thesis, North Carolina State University, Raleigh, NC.
- Deng ZC, Yao XL and Zhang DG (2009) Research on the dynamic performance of ship isolator systems that use magnetorheological dampers. *Journal of Marine Science and Application* 8: 291–297.
- Dogruer U, Gordaninejad F and Evrensel CA (2008) A new magneto-rheological fluid damper for high-mobility multi-purpose wheeled vehicle (HMMWV). *Journal of Intelligent Material Systems and Structures* 19: 641–650.
- Dogruoz MB, Wang EL, Gordaninejad F, et al. (2003) Augmenting heat transfer from fail-safe magneto-rheological fluid dampers using fins. *Journal of Intelligent Material Systems and Structures* 14: 79–86.
- Ebrahimi B (2009) *Development of hybrid electromagnetic dampers for vehicle suspension systems*. PhD Thesis, University of Waterloo, Canada.
- Eslaminasab N (2008) *Development of a semi-active intelligent suspension system for heavy vehicles*. PhD Thesis, University of Waterloo, Canada.
- Facey WB, Rosenfeld NC and Wereley NM (2003) Design and characterization of a compact damper for high impulsive loads. In: *44th AIAA/ASME/ASCE/AHS/ASC structures, structural dynamics, and materials conference*, 7–10 April, 2003 Norfolk, VA.
- Facey WB, Rosenfeld NC, Choi YT, et al. (2005) Design and testing of a compact magnetorheological damper for high impulsive loads. *International Journal of Modern Physics B: Condensed Matter Physics, Statistical Physics, Applied Physics* 19(7–9): 1549–1555.
- Fuchs A, Rashid A, Liu Y, et al. (2010) Compressible magnetorheological fluids. *Journal of Applied Polymer Science* 115: 3348–3356.
- Gao JX and Cheng L (2004) Modelling of a high performance piezoelectric actuator assembly for active and passive vibration control. *Smart Materials and Structures* 13: 384–392.
- Gao JX and Cheng L (2005) Modeling and control of power flow in a double-beam system. *Thin Walled Structures* 43: 751–771.
- Gavin H, Hoagg J and Dobossy M (2001) Optimal design of MR dampers. In: *Proceedings of U.S.–Japan workshop on smart structures for improved seismic performance in urban regions*, Seattle WA, 14 August 2001, pp.225–236.
- Goncalves FD (2005) *Characterizing the behavior of magnetorheological fluids at high velocities and high shear rates*. PhD Thesis, Virginia Polytechnic Institute and State University, Blacksburg, VA.
- Goncalves FD, Ahmadian M and Carlson JD (2006) Investigating the magnetorheological effect at high flow velocities. *Smart Materials and Structures* 15: 75–85.
- Goncalves FD and Carlson JD (2007) Investigating the time dependence of the MR effect. *International Journal of Modern Physics B: Condensed Matter Physics, Statistical Physics, Applied Physics* 21(28 and 29): 4832–4840.
- Goncalves FD and Carlson JD (2009) An alternate operation mode for MR fluids—magnetic gradient pinch. *11th Conference on Electrorheological Fluids and Magnetorheological Suspensions, Journal of Physics: Conference Series* 149: 012050.
- Goncalves FD, Koo JH and Ahmadian M (2006) A review of the state of the art in magnetorheological fluid technologies—part I: MR fluid and MR fluid models. *The Shock and Vibration Digest* 38(3): 203–219.
- Gordaninejad F, Fuchs A, Dogrou U, et al. (2004) *A new generation of magneto-rheological dampers*. Final Progress Report, Army Research Office, 27 August 2004.
- Gordaninejad F, Wang X, Hitchcock G, et al. (2010). Modular high-force seismic magneto-rheological fluid damper. *Journal of Structural Engineering* 136(2): 135.
- Grunwald A and Olabi AG (2008) Design of magneto-rheological (MR) valve. *Sensors and Actuators A: Physical* 148: 211–223.
- Hiemenz GJ, Hu W and Wereley NM (2008) Semi-active magnetorheological helicopter crew seat suspension for vibration isolation. *Journal of Aircraft* 45(3): 945–953.
- Hietanen S (2002) ER fluids and MR materials—basic properties and some application developments. In: *Smart materials and structures VTT industrial systems 2000–2002*, Espoo, Finland, 4 December 2002 (Book section).
- Hitchcock GH, Gordaninejad F and Wang X (2002) A new by-pass, fail-safe, magneto-rheological fluid damper. In: *Proceedings of SPIE conference on smart materials and structures*, San Diego, CA, March 2002, pp.1–7.
- Hitchcock GH, Wang X and Gordaninejad F (2007) A new bypass magnetorheological fluid damper. *Journal of Vibration and Acoustics* 129(5): 641–647.
- Hong SR, Choi SB, Choi YT, et al. (2005) Non-dimensional analysis and design of a magnetorheological damper. *Journal of Sound and Vibration* 288: 847–863.
- Hong SR, John S, Wereley NM, et al. (2008) A unifying perspective on the quasi-steady analysis of magnetorheological dampers. *Journal of Intelligent Material Systems and Structures* 19(8): 959–976.
- Hong SR, Wang G, Hu W, et al. (2006) Liquid spring shock absorber with controllable magnetorheological damping. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering* 220(8): 1019–1029.

- Hong SR, Wereley NM, Choi YT, et al. (2008) Analytical and experimental validation of a nondimensional Bingham model for mixed-mode magnetorheological dampers. *Journal of Sound and Vibration* 312: 399–417.
- Hoyle A, Arzanpour S and Shen Y (2010) A novel magnetorheological damper based parallel planar manipulator design. *Smart Materials and Structures* 19: 055028 (9pp.).
- Hu W and Wereley NM (2008) Hybrid magnetorheological fluid–elastomeric lag dampers for helicopter stability augmentation. *Smart Materials and Structures* 17: 045021 (16pp.).
- Hu W, Robinson R and Wereley NM (2009) A design strategy for magnetorheological dampers using porous valves. *11th Conference on Electrorheological Fluids and Magnetorheological Suspensions, Journal of Physics: Conference Series* 149: 012056.
- Ibrahim RA (2008) Recent advances in nonlinear passive vibration isolators. *Journal of Sound and Vibration* 314: 371–452.
- Jagadish HP and Ravikumar L (2010) Calibration of the stiffness and damping characteristics of a magnetorheological fluid long squeeze film damper. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science* 224: 2121–2128.
- Jing XJ and Lang ZQ (2009) Frequency domain analysis of a dimensionless cubic nonlinear damping system subject to harmonic input. *Nonlinear Dynamics* 58: 469–485.
- Jing XJ, Lang ZQ and Billings SA (2008a) Frequency domain analysis for suppression of output vibration from periodic disturbance using nonlinearities. *Journal of Sound and Vibration* 314: 536–557.
- Jing XJ, Lang ZQ and Billings SA (2008b) Output frequency response function based analysis for nonlinear Volterra systems. *Mechanical Systems and Signal Processing* 22: 102–120.
- Jing XJ, Lang ZQ and Billings SA (2011a) Nonlinear influence in the frequency domain: alternating series. *Systems & Control Letters* 60(5): 295–309.
- Jing XJ, Lang ZQ and Billings SA (2011b) Nonlinear influence in the frequency domain: Alternating series. *Systems & Control Letters* 60(5): 295–309.
- John S, Chaudhuri A and Wereley NM (2008) A magnetorheological actuation system: test and model. *Smart Materials and Structures* 17: 025023 (15pp.).
- Jolly MR (2001) Pneumatic motion control using magnetorheological technology. In: *Proceedings of smart structures and materials 2001: industrial and commercial applications of smart structures technologies* (ed. A-MR McGowan), 14 June 2001, vol. 4332, pp.300–307, <http://dx.doi.org/10.1117/12.429669>.
- Jolly MR and Carison JD (1996) Controllable squeeze film damping using magnetorheological fluid. In: *International conference on new actuators*, Bremen, Germany, 26–28 June 1996, pp.333–336.
- Jolly MR, Bender JW and Carlson JD (1999) Properties and applications of commercial magnetorheological fluids. *Journal of Intelligent Material Systems and Structures* 10(1): 5–13.
- Jung HJ, Jang DD, Cho SW, et al. (2009) Experimental verification of sensing capability of an electromagnetic induction system for an MR fluid damper based control system. *11th Conference on Electrorheological Fluids and Magnetorheological Suspensions, Journal of Physics: Conference Series* 149: 012058.
- Jung HJ, Jang DD, Lee HJ, et al. (2010) Feasibility test of adaptive passive control system using MR fluid damper with electromagnetic induction part. *Journal of Engineering Mechanics* 136(2): 254–259.
- Jung HJ, Spencer BF, Ni YQ, et al. (2004) State-of-the-art of semiactive control systems using MR fluid dampers in civil engineering applications. *Structural Engineering and Mechanics* 17(3–4): 493–526.
- Kciuk M and Turczyn R (2006) Properties and application of magnetorheological fluids. *Journal of Achievements in Materials and Manufacturing Engineering* 18(1–2): 127–130.
- Kim KH, Nam YJ, Yamane R, et al. (2009) Smart mouse 5-DOF haptic hand master using magneto-rheological fluid actuators. *11th Conference on Electrorheological Fluids and Magnetorheological Suspensions, Journal of Physics: Conference Series* 149: 012062.
- Kim KJ, Lee CW and Koo JH (2008) Design and modeling of semi-active squeeze film dampers using magnetorheological fluids. *Smart Materials and Structures* 17: 035006 (12pp.).
- Koo JH, Goncalves FD and Ahmadian M (2006) A comprehensive analysis of the response time of MR dampers. *Smart Materials and Structures* 15: 351–358.
- Lam KH, Chen ZH, Ni YQ, et al. (2010) A magnetorheological damper capable of force and displacement sensing. *Sensors and Actuators A: Physical* 158: 51–59.
- Lang ZQ, Jing XJ, Billings SA, et al. (2009) Theoretical study of the effects of nonlinear viscous damping on vibration isolation of SDOF systems. *Journal of Sound and Vibration* 323: 352–365.
- Lau YK and Liao WH (2005) Design and analysis of magnetorheological dampers for train suspension. *Proceedings of IMechE Part F: Journal of Rail and Rapid Transit* 219(4): 261–276.
- Lee DY, Choi YT and Wereley NM (2002) Performance analysis of ER–MR impact damper systems using Herschel–Bulkley model. *Journal of Intelligent Material Systems and Structures* 13: 525–531.
- Lee DY, Nam YJ, Yamane R, et al. (2009) Performance evaluation on vibration control of MR landing gear. *11th Conference on Electrorheological Fluids and Magnetorheological Suspensions, Journal of Physics: Conference Series* 149: 012068.
- Lee HS and Choi SB (2000) Control and response characteristics of a magneto-rheological fluid damper for passenger vehicles. *Journal of Intelligent Material Systems and Structures* 11: 80–87.
- Leicht Z, Urreta H, Sanchez A, et al. (2009) Theoretical and experimental analysis of MR valve. *11th Conference on Electrorheological Fluids and Magnetorheological Suspensions, Journal of Physics: Conference Series* 149: 012070.
- Li WH, Du H and Guo NQ (2003) Finite element analysis and simulation evaluation of a magnetorheological valve. *International Journal of Advanced Manufacturing Technology* 21(6): 438–445.
- Li WH, Wang XY, Zhang XZ, et al. (2009) Development and analysis of a variable stiffness damper using an MR bladder. *Smart Materials and Structures* 18: 074007 (8pp.).

- Liao WH and Lai CY (2002) Harmonic analysis of a magneto-rheological damper for vibration control. *Smart Materials and Structures* 11: 288–296.
- Lindler JE, Dimock GA and Wereley NM (2000) Design of a magneto-rheological automotive shock absorber. *Proceedings of SPIE* 3985: 426–437.
- Liu B, Li WH, Kosasih PB, et al. (2006) Development of an MR-brake-based haptic device. *Smart Materials and Structures* 15: 1960–1966.
- Liu Y, Gordaninejad F, Evrensel CA, et al. (2005) Comparative study on vibration control of a scaled bridge using fail-safe magneto-rheological fluid dampers. *Journal of Structural Engineering* 131(5): 743–751.
- Maciejewski I (2010) Modeling and control of semi-active seat suspension with magneto-rheological damper. In: *XXIV symposium vibrations in physical systems*, Poznan–Bedlewo, Poland, 12–15 May 2010.
- Mao M, Hu W, Choi YT, et al. (2007) A magneto-rheological damper with bifold valves for shock and vibration mitigation. *Journal of Intelligent Material Systems and Structures* 18: 1227–1232.
- Marouze JP and Cheng L (2002) A feasibility study of active vibration isolation using thunder actuators. *Smart Materials and Structures* 11: 854–862.
- Muhammad A, Yao XL and Deng ZC (2005) Review of magneto-rheological (MR) fluids and its applications in vibration control. *Journal of Marine Science and Application* 5(3): 17–29.
- Nam YJ and Park MK (2007) Performance evaluation of two different bypass-type MR shock dampers. *Journal of Intelligent Material Systems and Structures* 18: 707–717.
- Nam YJ and Park MK (2009) Electromagnetic design of a magneto-rheological damper. *Journal of Intelligent Material Systems and Structures* 20: 181–191.
- Ngatu GT, Hu W, Wereley NM, et al. (2010) Adaptive Snubber-type magneto-rheological fluid–elastomeric helicopter lag damper. *AIAA Journal* 48(3): 598–610.
- Nguyen QH and Choi SB (2009a) Optimal design of MR shock absorber and application to vehicle suspension. *Smart Materials and Structures* 18: 035012 (11pp.).
- Nguyen QH and Choi SB (2009b) Optimal design of a vehicle magneto-rheological damper considering the damping force and dynamic range. *Smart Materials and Structures* 18: 015013 (10pp.).
- Nguyen QH, Choi SB and Kim KS (2009a) Geometric optimal design of MR damper considering damping force control energy and time. *11th Conference on Electrorheological Fluids and Magneto-rheological Suspensions, Journal of Physics: Conference Series* 149: 012076.
- Nguyen QH, Choi SB, Lee YS, et al. (2009b) An analytical method for optimal design of MR valve structures. *Smart Materials and Structures* 18: 095032 (12pp.).
- Nguyen QH, Choi SB, Wereley N, et al. (2008) Optimal design of magneto-rheological valves via a finite element method considering control energy and a time constant. *Smart Materials and Structures* 17: 025024 (12pp.).
- Nguyen QH, Han YM, Choi SB, et al. (2007) Geometry optimization of MR valves constrained in a specific volume using the finite element method. *Smart Materials and Structures* 16: 2242–2252.
- Norris JA (2003) *Behavior of magneto-rheological fluids subject to impact and shock loading*. Master Thesis, Virginia Polytechnic Institute and State University, Blacksburg, VA.
- Nosse DT and Dapino MJ (2007) Magneto-rheological valve for hybrid electrohydrostatic actuation. *Journal of Intelligent Material Systems and Structures* 18(11): 1121–1136.
- Oh HU (2004) Experimental demonstration of an improved magneto-rheological fluid damper for suppression of vibration of a space flexible structure. *Smart Materials and Structures* 13: 1238–1244.
- Oh HU and Onoda J (2002) An experimental study of a semi-active magneto-rheological fluid variable damper for vibration suppression of truss structures. *Smart Materials and Structures* 11: 156–162.
- Olabi AG and Grunwald A (2007) Design and application of magneto-rheological fluid. *Materials and Design* 28: 2658–2664.
- Or SW, Duan YF, Ni YQ, et al. (2008) Development of magneto-rheological dampers with embedded piezoelectric force sensors for structural vibration control. *Journal of Intelligent Material Systems and Structures* 19: 1327–1338.
- Pang L, Kamath GM and Wereley NM (1998) Analysis and testing of a linear stroke magneto-rheological damper. In: *AIAA/ASME/AHS adaptive structures forum*, Long Beach, CA, April 1998, Paper no. AIAA 98-2040, vol. CP9803, Part 4, pp.2841–2856.
- Poyner JC (2001) *Innovative designs for magneto-rheological dampers*. MS Thesis, Virginia Polytechnic Institute and State University, Blacksburg, VA.
- Prabakar RS, Sujatha C and Narayanan S (2009) Optimal semi-active preview control response of a half car vehicle model with magneto-rheological damper. *Journal of Sound and Vibration* 326: 400–420.
- Rabinow J (1948) The magnetic fluid clutch. *AIEE Transactions* 67: 1308–1315.
- Reichert BA (1997) *Application of magneto-rheological dampers for vehicle seat*. Master Thesis, Virginia Polytechnic Institute and State University, Blacksburg, VA.
- Rosenfeld NC and Wereley NM (2004) Volume-constrained optimization of magneto-rheological and electrorheological. *Smart Materials and Structures* 13: 1303–1313.
- Sapinski B (2010) Vibration power generator for a linear MR damper. *Smart Materials and Structures* 19: 105012 (12pp.).
- Sassi S, Cherif K, Mezghani L, et al. (2005) An innovative magneto-rheological damper for automotive suspension from design to experimental characterization. *Smart Materials and Structures* 14: 811–822.
- Sathianarayanan D, Karunamoorthy L, Srinivasan J, et al. (2008) Chatter suppression in boring operation using magneto-rheological fluid damper. *Materials and Manufacturing Processes* 23: 329–335.
- Snyder RA, Kamath GM and Wereley NM (2001) Characterization and analysis of magneto-rheological damper behavior under sinusoidal loading. *AIAA Journal* 39(7): 1240–1253.
- Sodeyama H, Sunakoda K, Fujitani H, et al. (2003) Dynamic tests and simulation of magneto-rheological dampers. *Computer-Aided Civil and Infrastructure Engineering* 18: 45–57.
- Sodeyama H, Suzuki K and Sunakoda K (2004) Development of large capacity semi-active seismic damper using

- magneto-rheological fluid. *Journal of Pressure Vessel Technology* 126(1): 105–109.
- Song G and Zeng M (2005) A thin-film magnetorheological fluid damper/lock. *Smart Materials and Structures* 14: 369–375.
- Stanway R, Sprostonz JL and El-Wahed AK (1996) Applications of electro-rheological fluids in vibration control a survey. *Smart Materials and Structures* 5: 464–482.
- Sung KG, Choi SB, Lee HG, et al. (2005) Performance comparison of MR dampers with three different working modes. *International Journal of Modern Physics B: Condensed Matter Physics, Statistical Physics, Applied Physics* 19(7–9): 1556–1562.
- Tanner ET (2003) *Combined shock and vibration isolation through the self-powered, semi-active control of a magnetorheological damper in parallel with an air spring*. Doctor Thesis, Virginia Polytechnic Institute and State University, Blacksburg, VA.
- Thanh TDC and Ahn KK (2006) Intelligent phase plane switching control of pneumatic artificial muscle manipulators with magneto-rheological brake. *Mechatronics* 16(2): 85–95.
- Tse T and Chang CC (2004) Shear-mode rotary magnetorheological damper for small-scale structural control experiments. *Journal of Structural Engineering* 130: 904–911.
- Unsal M (2006) *Semi-active vibration control of a parallel platform mechanism using magnetorheological damping*. PhD Thesis, University of Florida, Gainesville, FL.
- Vicente J, Klingenberg DJ and Hidalgo-Alvarez R (2011) Magnetorheological fluids: a review. *Soft Matter* 7: 3701–3710.
- Wan X and Gordaninejad F (2009) A new magnetorheological fluid-elastic mount phenomenological modeling and experimental study. *Smart Materials and Structures* 18: 095045 (9pp.).
- Wang DH, Ai HX and Liao WH (2009) A magnetorheological valve with both annular and radial fluid flow resistance gaps. *Smart Materials and Structures* 18: 115001 (16pp.).
- Wang DH and Liao WH (2011) Magnetorheological fluid dampers a review of parametric modelling. *Smart Materials and Structures* 20: 023001 (34pp.).
- Wang DH and Wang T (2009) Principle, design and modeling of an integrated relative displacement self-sensing magnetorheological damper based on electromagnetic induction. *Smart Materials and Structures* 18: 095025.
- Wang DH, Bai XX and Liao WH (2010) An integrated relative displacement self-sensing magnetorheological damper prototyping and testing. *Smart Materials and Structures* 19: 105008 (19pp.).
- Wang J and Meng G (2001) Magnetorheological fluid devices: Principles, characteristics and applications in mechanical engineering. *Proceedings of the Institution of Mechanical Engineers, Part L: Journal of Materials: Design and Applications* 215: 165–174.
- Wereley NM (2008) Nondimensional Herschel-Bulkley analysis of magnetorheological and electrorheological dampers. *Journal of Intelligent Material Systems and Structures* 19: 257–268.
- Wereley NM and Pang L (1998) Nondimensional analysis of semi-active electrorheological and magnetorheological dampers using approximate parallel plate models. *Smart Materials and Structures* 7: 732–743.
- Wereley NM, Cho JU, Choi YT, et al. (2008) Magnetorheological dampers in shear mode. *Smart Materials and Structures* 17: 015022 (11pp.).
- Yang G, Spencer BF Jr, Carlson JD, et al. (2002) Large-scale MR fluid dampers modeling, and dynamic performance considerations. *Engineering Structures* 24: 309–323.
- Yoo JH and Wereley NM (2002) Design of a high-efficiency magnetorheological valve. *Journal of Intelligent Material Systems and Structures* 13(10): 679–685.
- Yoo JH and Wereley NM (2005) Nondimensional analysis of annular duct flow in magnetorheological–electrorheological dampers. *International Journal of Modern Physics B: Condensed Matter Physics, Statistical Physics, Applied Physics* 19(7–9): 1577–1583.
- York D, Wang X and Gordaninejad F (2007) A new MR fluid-elastic vibration isolator. *Journal of Intelligent Material Systems and Structures* 18: 1221–1225.
- Yu M, Dong XM, Choi SB, et al. (2009) Human simulated intelligent control of vehicle suspension system with MR dampers. *Journal of Sound and Vibration* 319: 753–767.
- Zhang HH, Liao CR, Yu M, et al. (2007) A study of an inner bypass magneto-rheological damper with magnetic bias. *Smart Materials and Structures* 16: N40–N46.
- Zhang HH, Xu HP, Liao CR, et al. (2011) Dynamic response of magnetorheological fluid damper for automotive suspension and the influence by long-time standing-still. *Applied Mechanics and Materials* 105–107: 1689–1692.
- Zhou GY and Zhang PQ (2002) Investigation of the dynamic mechanical behavior of the double-barreled configuration in a magnetorheological fluid damper. *Smart Materials and Structures* 11: 230–238.
- Zhu XC, Jing XJ and Cheng L (2011) A magnetorheological fluid embedded pneumatic vibration isolator allowing independently adjustable stiffness and damping. *Smart Materials and Structures* 20: 085025 (18pp.).