Chapter 1. Introduction

Vibration isolation is the elimination or reduction of vibration of a mechanical system by inserting a resilient member or an isolator between the source of excitation and the said system which is also termed as receiver. The need for vibration isolator is felt for efficient and quiet operation of machines, measuring instruments, vehicles and other physical systems. As a result, enormous research endeavor has been spent in designing various kinds of isolators suitable for different applications. Depending upon the hardware requirements the isolators can be broadly classified as

- passive vibration isolator
- active vibration isolator
- semi-active vibration isolator

Passive isolators consist of resilient members together with different damping mechanisms. They have fixed design parameters and hence can be very robust and reliable. Since these devices do not require any external energy source the passive isolator systems are comparatively less expensive. However, the fixed design parameters of the isolator make them effective only in the narrow frequency ranges for which they are designed. The bandwidth of the isolators can be substantially improved by applying required forces for isolation by means of externally controlled force actuators. This improvement is achieved at high cost. The complexity of the isolating devices compared to the simple design of a passive isolator is very high. This is associated with the risk of malfunctioning of the actively controlled devices.

The novel concept of semi-active vibration isolator exploits the advantages of the two extremes. In semi-active isolator the design parameters of the passive devices are actively altered depending upon the requirements. External power is required only to activate certain elements in the otherwise passive devices. This device consumes less energy compared to the energy requirement of a fully active isolator. In addition to being less expensive the semi-active isolators are more reliable than its active counterpart. This is due to the fact than semi-active isolators are made of passive elements with adjustable parameters. Thus, in case of failure of the controller responsible for changing the parameters of the devices, the passivity of the isolator does not allow to fail completely.

In view of these alternative features of the semi-active isolators, they are increasingly becoming popular to the vibration engineers. The progress in designing very effective semi-active isolators has been accelerated by the rapid development in the materials science and engineering. New materials with easily adjustable properties are being fabricated at a rapid pace. The development of so called ‘smart’ or ‘intelligent’ materials has been key factor in designing new and effective vibration isolators.

Magneto-rheological (MR) fluid and electro-rheological (ER) fluids are two ‘smart’ fluids which are used for constructing semi-active vibration isolators. Because of some favorable properties like temperature stability, high yield stress, the former is preferred over the later. The MR-fluid is used as the fluid elements in dampers. Such dampers are commercially available in various sizes.

The main objective of the present work is to study the vibration isolation properties of an isolator involving MR-fluid based damper in a systematic manner. In the following pages the literature available on various types of isolators are briefly reviewed with special emphasis on those using MR fluid. In order not to be lost in the huge literature available on vibration isolation in general and vibration isolation using MR-fluid, in particular, the review is conducted in three sections,

1. Vibration isolators
2. Magneto-rheological fluid dampers
3. Vibration control using magneto-rheological fluid dampers

1.1 Vibration Isolation

The three types of isolators mentioned in the previous section are discussed separately below.

1.1.1 Passive Isolators

Passive vibration isolators use spring and dampers as the restoring and dissipative elements. These isolators for shock and vibration control have been extensively studied by Snowdon (1968), Piersol and Paez (2002), Crede (2002). A schematic diagram of a single degree of freedom base excited passive isolator is shown in Fig. 1.1(a).

Passive single and two stage isolators with linear spring and linear viscous damper has been studied by Snowdon (1968). The performance of the isolators are measured by force transmissibility and motion transmissibility for forced and base excitations respectively. Different measures are however, required for shock isolators. The performances of different shock isolators are studied considering shock response spectra.

Studies of passive isolators with different types of nonlinearities were also available in literatures. Ibrahim (2008) reviewed different types of nonlinear isolators. Different types of nonlinear isolators, such as isolators with Coulomb damper, cubic spring, velocity squared damper etc. and their response to the transient excitation were studied by Snowdon (1968). It was shown that the use of nonlinear elements can improve the response of the system. The performance of a single degree of freedom system with Coulomb friction for harmonically base excited system was studied by Hundal (1979). Kirk (1988) compared the performance of nonlinear isolators under random ground inputs. Three types of nonlinearities, namely, cubic hard spring, cubic soft spring and tangent spring were considered. A two stage isolator was considered and the optimum damping ratio for better performance of the system was obtained by Thompson (1981). The steady state response of a nonlinear isolator was obtained analytically by Ravindra and Mallik (1993). The nonlinear isolator consisted of a hard cubic spring connected parallel to a combined viscous and Coulomb damper. Ravindra and Mallik (1994) also considered a single stage isolator with nonlinear spring and nonlinear damping for both force and base excitation. In this paper it was found that for base excited systems asymmetric spring elements may not perform satisfactorily. Hamdan and Burton (1993) also used the method of harmonic balance to study the steady state response of a single stage isolator. The stability of the response was studied using Mathieu equation. Two major requirements of an isolator, namely, low dynamic stiffness and high static stiffness were achieved by configuring a nonlinear isolator by Carrella et al. (2012). The force and motion transmissibilities of such an isolator were calculated for harmonic excitation. The response of a passive isolator with cubic nonlinear damping was obtained by the method of harmonic balance by Peng et al. (2012). Tang and Brennan (2013) considered a single degree of freedom system in which the damper was oriented perpendicular to the linear spring. The performances of this isolator was compared with linear one and isolators with cubic nonlinearity in damping and shown that it performs better than the other two. Cubic nonlinearity was also studied by Xiao et al. (2013), Mokni et al. (2011), Ho et al. (2014) etc. A seat suspension system was considered by Le and Ahn (2011) to have two negative stiffness springs parallel to a positive linear spring to achieve the low dynamic stiffness and high static stiffness of the isolator. The performance of the isolator, measured in terms of r.m.s. values of the displacement, was found to be better in case of this type of nonlinear isolators. Negative stiffness springs were also used in many literatures [Huang et al. (2014), Wu et al. (2014)].

Different types of passive shock isolation systems are studied in literatures. Hundal studied different types of nonlinear shock absorbers, like absorbers with linear spring and quadratic law damper [Hundal (1976, 1981)], pneumatic shock absorber [Hundal (1982)]. The absorber with a quadratic damper designed with optimal damping ratio was compared with the absorber with linear damper. The two isolators were subjected to rectangular shaped base acceleration pulse and the performances are studied in terms of maximum acceleration of the mass [Hundal (1981)]. Hundal (1982) studied the performance of a pneumatic shock absorber which is subjected to the same kind of shock as discussed. The optimal design criteria were found and the effect of these parameters on the performance was studied. Different nonlinear shock absorbers were studied by Surace et al. (1992). Nelson (1996) reviewed the shock isolation properties for different types of mechanical shock inputs. Su et al. (1992) considered a half car model with nonlinear suspension system. The nonlinear restoring and damping forces were linearized by using harmonic linearization technique and it was shown that the behavior of the suspension system can be improved by using this type of nonlinearity. Chandrashekhar et al. (1998) studied the response of shock isolator having nonlinear elements for three types of base excitations. It was shown that a symmetric quadratic damping with negative coefficient improves the overall shock isolation performance. Chandrashekhar et al. (1999) also considered four different absorbers and the responses of these for different types of shock inputs were studied. From the performance indices, namely Shock Acceleration Ratio (SAR), Relative Displacement Ratio (RDR) and Shock Displacement Ratio (SDR), it was concluded that the two stage absorbers performed better than the other types. An asymmetric shock absorber was used in a vehicle system by Silveira et al. (2014). The responses of this system for different shock inputs were compared with the symmetrical one and shown that asymmetrical absorbers had smoother and better performance. Another type of nonlinear viscous isolator for shock isolation was studied by Narkhede and Sinha (2014).

1.1.2 Active Isolators

Vibration isolators whose parameter values can be altered depending on the response and excitation, perform better than the passive isolators. Active isolators are largely used in vehicle suspensions. These provide a good ride quality and road handling. A schematic diagram of a single degree of freedom base excited active isolator system is shown in Fig. 1.1(b).

Collette et al. (2011) reviewed the active control strategies of vibration isolation. Hullender et al. (1972) studied vehicle model with active pneumatic suspension. It was shown that the heave acceleration can be improved by a significant amount by the use of active

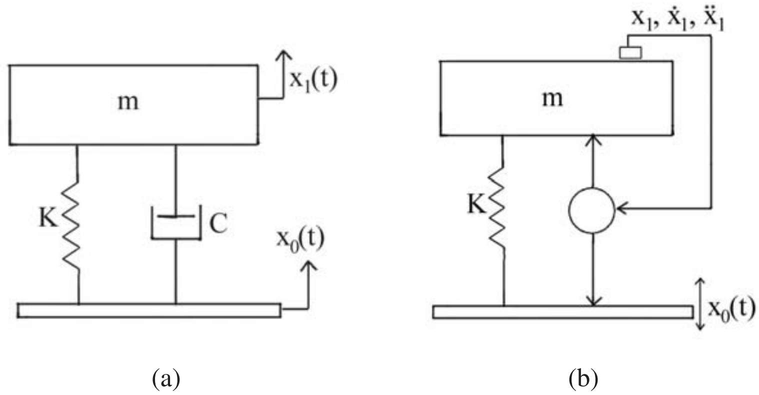


Fig. 1.1: Schematic diagram of the single stage isolator: a) Passive system, b)Active system

control elements. Sievers and von Flotow (1988) considered three control strategies on an engine mounted flexible structure. Variations of Linear Quadratic Gaussian (LQG) control schemes were used and the stability and performances of the isolators are studied. An active railroad vehicle suspension was studied by Hirata et al. (1995) to reduce the yaw, lateral motion and roll of that vehicle. The force was controlled using H^{∞} control strategy. Multistage active vibration isolator systems were used by Richman et al. (1998). An active vibration isolator, which was used to protect precision measuring instruments from the base vibration, was proposed by Hensley et al. (1999) using a combination of a linear spring and an optical table floating on compressed air. Many other types of active isolators were found in literatures [Coppola and Liu (2010); Wang et al. (2012); Lin and Lian (2013); Sun et al. (2014)]

The active system uses elements which are costly, sophisticated and require large source of energy, which limits its application. Semi-active vibration isolators posses the advantages of both the isolators and require energy only when needed. Thus the energy requirement for semi-active systems are much lesser compared to the active systems. Segla and Reich (2007) compared the performances of passive, active and semi-active vehicle suspension systems. It was shown that the semi-active systems could achieve performance close to that of active systems.

1.1.3 Semi-Active Isolator

Karnopp et al. (1974) first introduced the concept of semi-active systems. A sky-hook damper was attached to a single stage isolator to give better performance than a passive system. After the invention of semi-active systems many other types of semi-active control elements were studied in the literatures.

A new semi-active control strategy based on relative displacement feedback and velocity feedback was proposed by Alanoly and Sankar (1987). The performances of the proposed control scheme was compared with that of sky-hook control scheme. Alanoly and Sankar (1988) considered a single stage quarter car model with a semi-active isolator which is subjected to base excitations. Rounded step and rounded pulse types of excitations were given to the system. Two types of control strategies , sky-hook and the one proposed by Alanoly and Sankar (1987), were studied. The performance of this semi-active control schemes, in terms of acceleration of the sprung mass and relative displacement, were compared with the passive and active isolator and shown that this semi-active system can produce significant improvement in the performance.

Liu et al. (2008) configured a semi-active system with two controllable dampers and two springs. The equivalent stiffness and damping of the system were controlled using eight different semi-active control schemes. The system with damping and stiffness on-off control provided effective performance improvement. One of the mostly accepted sources of semi-active damping element were variable damping valve elements [Oh et al. (2007); Spelta et al. (2010); Oh and Choi (2013)].

The mostly used controllable semi-active devices are electro-ehaological (ER) and magneto-rheological (MR) fluid dampers. These fluids can be changed from viscous fluid to semi-solid non-Newtonian fluid within a very small time after application of electric and magnetic field respectively. In the starting ER fluids were used in most of the semi-active vibration control applications. Stanway et al. (1996) did a brief review on the applications of ER dampers. ER dampers were widely used in vehicle suspension systems [Peel et al. (1996); Sims et al. (2000); Sung et al. (2007)], structural vibration isolation [McClamroch and Gavin (1995); Kamath et al. (1996); Kugi et al. (2005)] etc. In recent years MR fluid dampers were found to be more efficient than ER fluid dampers. Different important semi-active control strategies are now briefly discussed.

The performance of semi-active controller of vibration depend on the choice of the semi-active control strategy. Important semi-active control strategies are presented in the following.

Sky-Hook Control Strategy:

In the first semi-active controller, Karnopp et al. (1974) proposed a sky-hook damper connected to a single degree of freedom isolator as shown in Fig. 1.2. The controlling force depends on the velocity of mass and relative velocity. The control logic is given as,

$$\begin{aligned} F_d &= C\,\dot{x}, \qquad \dot{x}(\dot{x}-\dot{x}_0) > 0 \\ &= 0, \qquad \dot{x}(\dot{x}-\dot{x}_0) < 0 \end{aligned}$$

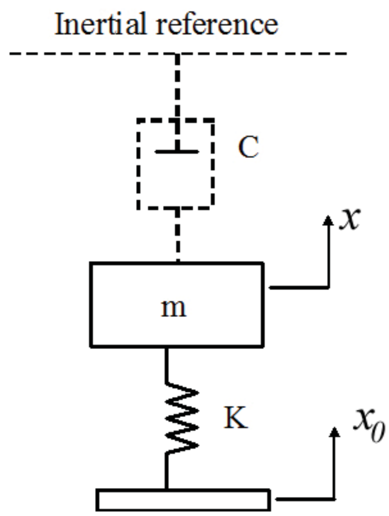


Fig. 1.2: Schematic diagram o Sky-Hook damper, proposed by Karnopp et al. (1974)

The sky-hook control logic is largely accepted because of its simplicity. Several modifications of this logic are also presented in literatures.

A modification of the sky-hook damping logic is proposed by Krasnicki (1980), where a simpler on-off scheme is proposed as follows,

$$\begin{aligned} F_d &= C\,(\dot{x}-\dot{x_0}), \qquad && \dot{x}(\dot{x}-\dot{x_0}) > 0 \\ &= 0 && \dot{x}(\dot{x}-\dot{x_0}) < 0 \end{aligned}$$

Here the force is directly proportional to the relative velocity which makes the system easy to operate.

Liu et al. (2005) reviewed both the on-off and continuous sky-hook control strategies for a single stage isolator. They showed that both the control strategies perform better than the passive systems. A new adaptive sky-hook control strategy along with preview control was proposed by Yi and Song (1999). They compared the performance of a quarter car suspension system using this control scheme with that of sky-hook controlled semi-active suspension. The proposed control scheme was shown to perform better than sky-hook system. Ahmadian (1999) studied continuous sky-hook and on-off sky-hook control schemes and the isolator performances of these two schemes were compared to the that of the passive systems. Xubin (2009) proposed a cost effective sky-hook control strategy. Goncalves (2001) studied different modified sky-hook control strategies for a quarter car system. It was shown that the conventional sky-hook control strategy performed better than its modifications for vehicle suspension systems.

Balance Control Strategy:

The sky-hook control strategy, both the on-off and continuous, require the measurement of absolute and relative velocities. In applications like automobile suspension systems, no inertial reference frame is available. In these cases the absolute velocity is calculated from the measured data. Thus, Rakheja and Sankar (1985) proposed a control strategy based on the relative velocity and relative displacement data. This control system is referred as "balance control" because during the on state of the damper, the spring force and damper forces oppose each other or balance each other. The on-off balance control scheme is given as

$$\begin{aligned} F_d &= C\,(\dot{x}-\dot{x_0}), && (x-x_0)(\dot{x}-\dot{x_0}) \leq 0 \\ &= 0 && (x-x_0)(\dot{x}-\dot{x_0}) > 0 \end{aligned}$$

The modification of the balance control as continuous balance control is proposed by Alanoly and Sankar (1987), where the damping coefficient is controlled as,

$$\begin{aligned} C &= -K(x-x_0)/(\dot{x}-\dot{x_0}), && (x-x_0)(\dot{x}-\dot{x_0}) \leq 0 \\ &= C_{off} && (x-x_0)(\dot{x}-\dot{x_0}) > 0 \end{aligned}$$

Wu et al. (1994) suggested a modification of the continuous balance control because following that logic, a large value of C may come which is practically impossible to achieve. They included another control statement with the previous logic as

$$\begin{aligned} C &= C_{on}, && -K(x-x_0)/(\dot{x}-\dot{x_0}) > C_{th} \\ &= C_{off} && -K(x-x_0)/(\dot{x}-\dot{x_0}) < C_{th} \end{aligned}$$

where, C_{th} is refereed to as threshold damping coefficient. They suggested to take a value which is 30% of the critical damping coefficient value. A detailed study on different types of balance control are done by Wu and Griffin (1997).

Ground-Hook Control Strategy

The ground-hook control strategy is applicable for two stage isolators where the semi-active element is connected to the unsprung mass. The schematic diagram is shown in Fig. 1.3

and the control logic is given as,

$$\begin{aligned} F_d = C\,\dot{x}_2, &\qquad \dot{x}_2(\dot{x}_2 - \dot{x}_0) < 0 \\ = 0, &\qquad \dot{x}_2(\dot{x}_2 - \dot{x}_0) > 0 \end{aligned}$$

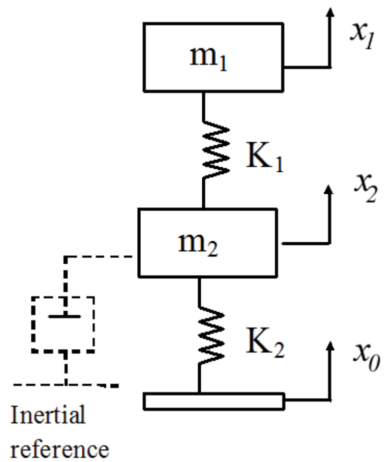


Fig. 1.3: Schematic diagram o ground-hook damper

A combination of sky-hook control with ground-hook control strategy, known as hybrid control strategy for truck suspensions are studied by Valaswek and Novak (1996); Valaswek et al. (1997). Ahmadian and Vahdati (2006) studied the transient responses of a two stage quarter car model using hybrid control logic. The schematic diagram for hybrid system is shown in Fig. 1.4 and the damping force is given as

$$F_d = G(\alpha_h\, \sigma_{sky} - (1-\alpha)\sigma_g)$$

where,

$$\begin{aligned} \sigma_{sky} = \dot{x}_1, &\qquad \dot{x}_1(\dot{x}_1 - \dot{x}_2) > 0 \\ = 0, &\qquad \dot{x}_1(\dot{x}_1 - \dot{x}_2) < 0 \end{aligned}$$

$$\begin{aligned} \sigma_g = \dot{x}_2, &\qquad \dot{x}_2(\dot{x}_2 - \dot{x}_0) < 0 \\ = 0, &\qquad \dot{x}_2(\dot{x}_2 - \dot{x}_0) > 0 \end{aligned}$$

where, α_h is the weighting factor and σ_{sky} and σ_g are sky-hook switching variable and ground-hook switching variables respectively. A comparison of sky-hook, ground-hook and hybrid control has also been given by Sankaranarayanan et al. (2008).

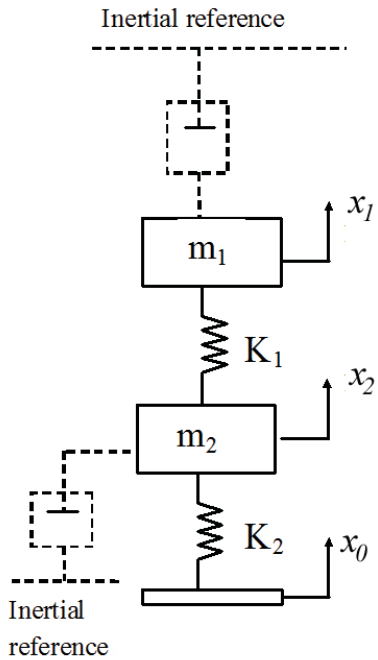


Fig. 1.4: Schematic diagram of hybrid damper system

The sky-hook control improves the ride comfort by reducing the sprung mass acceleration. The ground-hook control improves the road handling of the suspension system. The hybrid system is a good compromise between both the logics.

Other Semi-Active Control Strategies:

A detailed study of semi-active suspension systems are presented by Guglielmino et al. (2008). Several other types of semi-active control strategies are used in isolator systems like optimal control of quarter car [Gordon (1995)], a new adaptive sky-hook control [Yi and Song (1999)], LPV control technique [Poussot-Vassal et al. (2008)], linear quadratic based controller [Unger et al. (2013)] etc.

Fuzzy-logic control is one of the growing research which is reported in many literatures [Sung et al. (2007); Oh and Choi (2011); Li and Zhao (2010); Eltantawie (2012)]. Fuzzy sliding mode control algorithm is used in a ER suspension system by Sung et al. (2007). This real-time control algorithm is formulated in a discrete-time manner. A quarter car ER model is considered and it is shown that acceleration properties are improved by this control. Oh and Choi (2011) studied a fuzzy sky-hook control logic on a quarter car model. This control logic is shown to reduce jerk and control effort, which is the most important in vehicle suspension system. The performance of this control logic is comparable with that of a conventional sky-hook control scheme. Another new fuzzy logic is applied on a quarter car model by Li and Zhao (2010). A half car model with MR damper is controlled using a decentralized neuro-fuzzy controller by Eltantawie (2012). This proposed control logic improved the ride comfort and stability of a vehicle by a considerable amount.

The application of different semi-active control strategies in civil structures are studied by many structural researchers [Hrovat et al. (1983); Symans and Constantinou (1996); Soong and Spencer (2000)].

1.2 Magneto-rheological Fluid Based Dampers

1.2.1 Magneo-rheological Fluid

Magneto-rheological fluid is one of the controllable fluids which has the ability to change its viscosity when the applied magnetic field changes. MR fluids are widely used over other controllable fluids like ER fluids because of its advantages in attaining high yield stress, high operating temperature etc [Carlson et al. (1996)]. Because of these properties MR fluids are widely used as semi-active control element for vibration isolation.

MR fluids consist of micron-sized, magnetically polarized particles dispersed in a non-magnetic medium. When a magnetic field is applied to the fluid, it polarizes the particles. The interaction between these induced polarized particles causes the particles to form chain like structures, which are parallel to the applied magnetic field. These chain-like structures restrict the flow of fluid. Thus the fluid becomes semi-solid and exhibits viscoplastic behavior. The transition to rheological equilibrium can be achieved in a few milliseconds. When the magnetic field is removed, the fluid particles return to its original positions and the fluid behaves like Newtonian fluid.

There are 3 components of MR fluid, namely, base fluid, metal particles and stabilizing particles. Base fluid functions as carrier and naturally combines lubrication and damping features. These fluids are generally chosen on the basis of tribological and rheological properties as well as temperature stability. There are different types of liquids which can be used as base fluid or carrier fluid i.e., Hydrocarbon oils, mineral oils, silicon oils. The base fluid should have high viscosity so that even in off-state its viscosity is quite high. Usually the dynamic viscosity in ambient temperature will be around 100 MPa. When a magnetic field is applied the metal particles are guided by the field to form a chain like structure. This restricts the motion of the fluid and thereby changes the rheological properties. The metal particles are usually carbonyl iron or powder iron or iron/cobalt alloys. Stabilizing additives are used in the MR fluids to increase its stability.

Apart from the temperature and applied magnetic field, the rheological properties of MR fluid also depend on various parameters like volume fraction of particle, particle size,

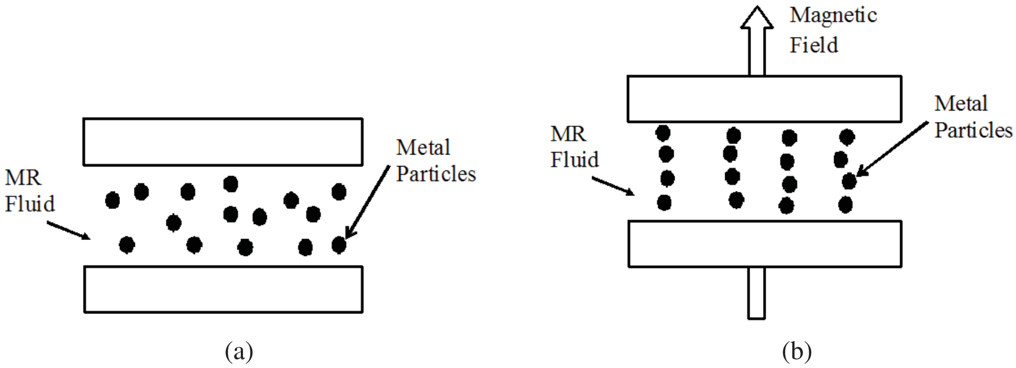


Fig. 1.5: MR fluid configuration: a) without applied magnetic field and b) after application of magnetic field.

properties of the base fluid, particle magnetization etc. Jolly et al. (1996) derived a relationship of MR effect as a function of particle magnetization. The constitutive relation of yield stress of the MR fluid is modeled by Bajkowski et al. (2008) considering the friction and viscosity of the fluid which are functions of applied magnetic field. Bossis et al. (2002) modeled the yield behavior of the MR fluid by formulating a chainlike structure of spheres.

1.2.2 Applications of Magneto-rheological Fluids

Because of various advantages stated above the MR fluids are used in a number of applications depending on their modes of operation. There are mainly four modes of operation, namely, valve mode, shear mode, squeeze mode and mixed mode.

Valve Mode:

The schematic diagram of the MR fluid operating in a valve mode is shown in Fig. 1.6. The applied magnetic field is perpendicular to the direction of MR fluid flow. The flow of MR fluid is caused due to the pressure drop and fluid flows through annular gap or orifice or between two plates. If the magnetic field is changed, the volume of fluid flowing through the gap and velocity profile of the flow also change.

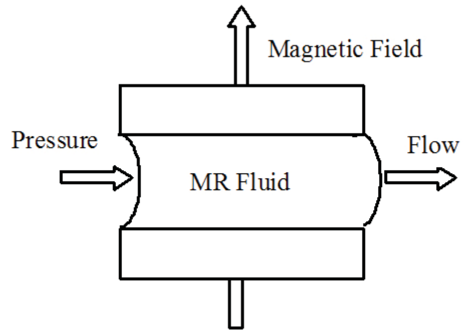


Fig. 1.6: Schematic diagram for Valve Mode of operation of MR fluid

MR fluids are used in valve mode in many applications like damper, valve, actuator etc. MR fluid dampers are used in various applications like knee prosthesis [Herr and Wilkenfeld (2003)], aircraft landing gear [Mikuowski and Holnicki-Szulc (2007); Shixing et al. (2011); Berasategui et al. (2014)], seismic applications [Dyke et al. (1996); Symans and Constantinou (1999); Yang et al. (2002); Azar et al. (2011)] etc. The main application of MR dampers are included in automobile industry in shock absorbers, vibration isolators, engine mounts, seat suspension etc. which will be discussed in next chapters.

Shear Mode:

While operating in shear mode, the fluid is kept between two surfaces and one of the surfaces slides or rotates over the other. The magnetic field is applied perpendicular to the direction of motion. The schematic diagram of MR fluid operating in shear mode is shown in Fig. 1.7.

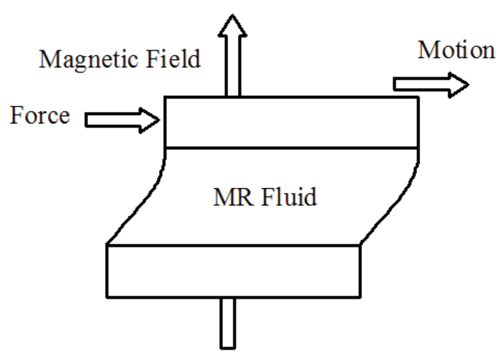


Fig. 1.7: Schematic diagram for shear Mode of operation of MR fluid

The applications of MR fluid in shear mode includes MR brakes [Karakoc et al. (2008); Nguyen and Choi (2010)], MR clutch [Kavlicoglu et al. (2007); Kikuchi et al. (2009)], rotary bearings [Wang and Meng (2003); Forte et al. (2004)] etc.

Squeeze Mode:

In squeeze mode the magnetic field is applied in the same direction of the applied force which causes the squeeze flow of the fluid as shown in Fig. 1.8. The displacement in squeeze mode is relatively small as compared to the previous two modes.

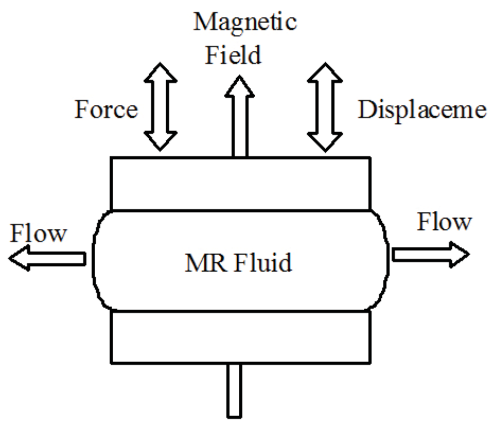


Fig. 1.8: Schematic diagram for squeeze Mode of operation of MR fluid

Squeeze mode was first studied by Sproston et al. (1994). They studied the squeeze flow performance of ER fluids. Squeeze mode dampers are widely used in vibration isolation systems. MR fluids used in squeeze mode are also used in rotor applications [Wang et al. (2005); Carmignani et al. (2006)], polishing [Singh et al. (2004); Kim et al. (2009)] etc.

Mixed Mode:

A combination of modes are made for better performance. Wereley and Pang (1998) considered the combination of shear mode and valve mode. Kamath et al. (1996) developed a mixed mode dashpot damper which is a combination of valve mode and shear mode. Both the papers considered the MR flow to be Bingham Plastic flow and a quasi-static analysis is done. Kulkarni et al. (2003) compared the behavior of linear damper with that of a mixed mode damper, which is a combination of shear mode and squeeze mode.

The rheological properties have been modeled in various ways by various researchers. The non-Newtonian nature of the fluid has been clearly demonstrated. A common constitutive relationship between shear stress and shear strain rate, that has been used by a number of researchers to model a MR fluid, is the one applied to HerschelBulkley flow given by

$$\tau = \tau_y + \eta_d \dot{\gamma}^n, \qquad (\tau > \tau_y)$$

where, τ_y is the yield shear stress and η_d is the dynamic viscosity and n is the fluid index

[Wang and Gordaninejad (1999); Lee and Wereley (2000); Yang et al. (2002); Chooi and Oyadiji (2008)]. When $n = 1$, the fluid is known as Bingham plastic fluid. The Bingham plastic flow has also been used for modeling MR fluid [Stanway et al. (1996); Kamath et al. (1996); Wereley et al. (1998); Cesmeci and Engin (2010)]. More detailed models are also discussed from continuum theoretic approach. Brigadnov and Dorfmann (2005) considered a full constitutive relation of an isotropic MR fluid damper. The MR fluid, considered here is shear dependent. Three-dimensional nonlinear consecutive law was given by Dorfmann et al. (2007). A continuum electromechanical model of MR damper is proposed by Costa and Branco (2009). Sternberg et al. (2011, 2014) constructed a 3D finite element model of MR fluid flow.

1.2.3 Modeling of MR Dampers

A complete analysis of fluid flow of MR fluid in a damper where it is used in different modes becomes very complicated to predict the behavior of the damper. This difficulty prompted researchers to construct various parametric models for the prediction of force-displacement or force-velocity relationship of the MR-fluid based dampers. Wang and Liao (2011) gave a detailed review of the parametric modeling of MR-fluid based dampers. The hysteresis models of MR dampers were discussed. These include simple Bouc-Wen model[Spencer et al. (1997); Ismail et al. (2009)], modified Bouc-Wen model[Spencer et al. (1997)], current dependent Bouc-Wen model [Yang et al. (2002, 2004)] etc. The different variations of Bouc-Wen model, like the one for large scale MR damper, current dependent Bouc-Wen model, Current-Amplitude-frequency depended Bouc-Wen model, non-symmetric Bouc-Wen model etc. were discussed. Sahin et al. (2010) also reviewed different parametric models. This paper proposed a simple algebraic parameter model that predicts the force-displacement or force-velocity relationship with accurately comparable with the models having complex mathematical models. Hassani et al. (2014) reviewed different types of hysteresis models which are used to model MR fluid dampers.

The simplest parametric model to predict the dynamic behavior of the damper is Bingham model is proposed by Stanway et al. (1996). The model consisted of a Coulomb friction element connected parallel to a viscous damper (see Fig. 1.9(a)). The force-velocity relationship was given by the expression,

$$F = f_c \operatorname{sgn}(\dot{x}) + C_0\dot{x} + f_0$$

where, C_0 is viscous damping coefficient, f_c is the frictional force and f_0 is the force from accumulator of the damper. A modification of Bingham model, piecewise Bingham model was proposed in literatures [Kamath et al. (1996); Wereley et al. (1998); Snyder et al. (2001)]. Occhiuzzi et al. (2003) proposed a modified Bingham model which incorporate the changes of parameters with current.

Another parametric model was Gamota-Filisko model, proposed by Gamota and E. (1991), where the Bingham model was connected in series with a solid model as shown in Fig. 1.9(b). The force velocity relationship is given by

$$F_{GF}(x,\dot{x}) = \begin{cases} K_1(x_2-x_1)+C_0(\dot{x_2}-\dot{x_1})+f_0 & \\ f_c\ \mathrm{sgn}(\dot{x_1})+C_0\dot{x}+f_0 & \text{when } |F|>f_c \text{ and } \dot{x}_1 \neq 0 \\ K_2(x_3-x_2)+f_0, & \end{cases}$$

$$F_{GF}(x,\dot{x}) = \begin{cases} K_1(x_2-x_1)+C_1\dot{x_2}+f_0 & \\ K_2(x_3-x_2)+f_0. & \text{otherwise}s \end{cases}$$

where x_1 and x_2 are the displacements of the points indicated in Fig. 1.9(b). These two models were very stiff which made their use limited.

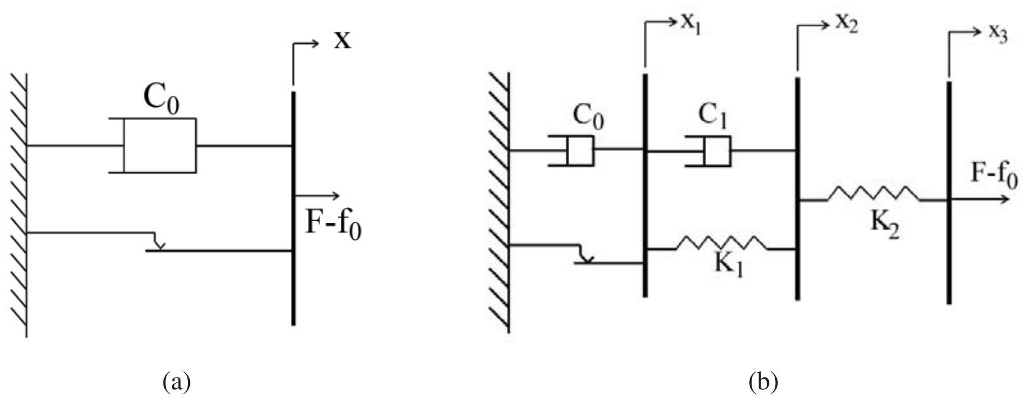


Fig. 1.9: Schematic diagram of a) Bingham Model and b) Gamota-Filisko model.

Bouc-Wen model is a well accepted model for constructing any hysteresis system. Spencer et al. (1997); Yang et al. (2002) and many more researchers used Bouc-Wen model to model the MR dampers. A simple Bouc-Wen model consists of a viscous damper and a linear spring connected parallel to a hysteresis element, called Bouc-Wen element. A configuration of a simple Bouc-Wen model is shown in Fig. 1.10(a). The force-velocity relationship of the MR damper is expressed as,

$$F_{MR} = C_0\dot{x}+K_0(x-x_0)+\alpha r$$

where, r is an evolutionary variable which is governed by,

$$\dot{r} = A\dot{x} - \gamma\,|\dot{x}|\;r\;|r|^{n-1} - \beta(\dot{x})\;|r|^{n}.$$

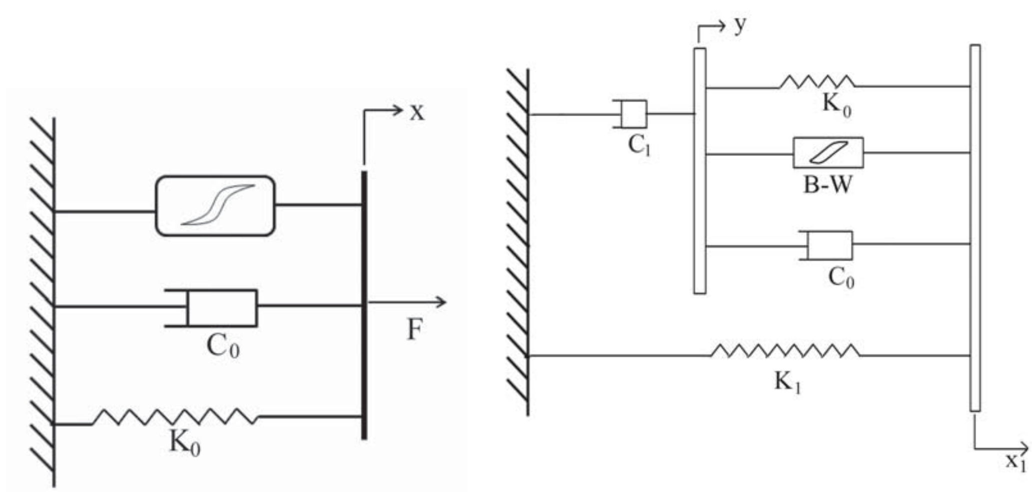
The parameters, A, γ and β are the parameters used in Bouc-Wen hysteresis element. The damper properties can be changed by varying these parameters. Bouc-Wen model is less stiff than the two previously stated models, but it fails to predict the roll-off behavior which is seen near low velocity and when the operating sign of the velocity is opposite to the acceleration. To predict this behavior Spencer et al. (1997) proposed a modified Bouc-Wen model in which the simple Bouc-Wen model was connected in parallel to a linear spring and and in series with a linear viscous damper as shown in Fig. 1.10(b). The linear spring provided the accumulator stiffness and viscous damper corresponds to the leak flow within the damper. The force-velocity relationship was governed by

$$f_{MR} = K_1 x_1 + K_0(x_1 - y) + C_0(\dot{x_1} - \dot{y}) + \alpha r,$$

or,

$$f_{MR} = K_1 x_1 + C_1 \dot{y}.$$

where r, the evolutionary variable as defined earlier.



(a) (b)

Fig. 1.10: Schematic diagram of a) Simple Bouc-Wen model and b) Modified Bouc-Wen model

The parameters, A, C_0 and C_1 are dependent on input current or voltage. Spencer et al.

(1997) proposed that a linear current relationship which is given as

$$\alpha = \alpha_a + \alpha_b u, \qquad C_0 = C_{0a} + C_{0b} u, \qquad C_1 = C_{1a} + C_{1b} u.$$

where, u was the voltage output of the first order filter which represents the dynamics involved in the MR fluid. The relation was given as

$$\dot{u} = -\eta (u - v)$$

where v is the commanded voltage sent to the current driver.

It was shown by Spencer et al. (1997) that the proposed modified Bouc-Wen model improves modeling accuracy. For large-scale MR dampers, which were used in structural applications, Yang et al. (2002) proposed that the parameters were dependent on input voltage by a third order polynomial relationship. For large-scale MR dampers Yang et al. (2004) considered the shear thinning effect of the fluid. They also proposed two new phenomenological models which are modifications of modifications of Bouc-Wen model and modified Bouc-Wen model with a mass element. Dominguez et al. (2004) proposed a model based on simple Bouc-Wen model where the coefficients, C_0, K_0, α, γ were directly depending on applied Current. Ali and Ramaswamy (2009) considered the parameters of modified Bouc-Wen model to be dependent on amplitude of excitation and current. Dominguez et al. (2006, 2008) also proposed a modification of Spencer model where the above stated parameters are functions of frequency, amplitude of excitation and current.

Many other parametric models have been studied by many researchers. The force-velocity hysteresis behavior of an MR damper was modeled by Guan et al. (2011) using a quasi-static element which was connected in series with an air spring. Dahl model is another type of hysteresis model where the Bouc-Wen hysteresis element was replaced by Dahl element [Dahl (1976); Zhou et al. (2006)]. Tsouroukdissian et al. (2008) compared simple Bouc-Wen model and Dahl model to predict the dynamic behavior of MR dampers for varying voltage experimental data. A trigonametric function based model for modeling was proposed by Guo et al. (2006).

The identification of the parameters of above discussed parametric models for MR fluid dampers is a challenging task. Many parameter identification techniques were used by the researchers. Chang and Roschke (1998) developed a multi-layer perception neural network model for a MR damper. The model was optimized using optimal neural network to represent the MR damper behavior properly. A feed-forward and recurrent neural network based model for MR damper was proposed by Metered et al. (2010). Genetic algorithm (GA) based parameter identification techniques were carried out by Giuclea et al. (2004a,b). A new parametric model for MR damper was proposed using hyperbolic tangent functions by Kwok et al. (2006). The model parameters are identified using particle swarm optimization technique. An improved particle swarm optimization algorithm was proposed by Ye and Wang (2009) to identify the parameters of hysteretic Bouc-Wen model. The proposed model was compared with Genetic Algorithm (GA) technique and it was shown that the model may provide better quality solution. A black-box model of MR damper has been proposed by Truong and Ahn (2010), where the parameters of MR damper were directly identified through a fuzzy mapping system.

1.2.4 Vibration Control Using MR Fluid Dampers

Various semi-active control strategies are used with the magneto-rheological damper in vibration isolator. A type of clipped optimal control based on acceleration feedback is proposed by Dyke et al. (1996). Lee and Choi (2000) considered a full vehicle model with magneto-rheologicl suspension system. The vehicle vibration is suppressed using conventional sky-hook control strategy. Jansen and Dyke (2000) compared various semi-active control algorithm asscociated with MR dampers. The control strategies compared are Lyapunov controller decentralized bang-bang controller, modulated homogeneous friction algorithm and clipped optimal controller. The strategies were compared for a simulated earthquake data. Yao et al. (2002) considered quarter model with MR damper using conventional sky-hook control strategy. Song et al. (2005) presented a nonlinear model based adaptive semi-active control strategy for MR system, which was subjected to random excitation. The proposed adaptive algorithm provided provision for on-line parameter identification. Parametric study of the proposed model was done by Song et al. (2007). Hudha et al. (2005) compared three control strategies, modified sky-hook control, modified ground-hook control and hybrid control strategies, applied on magneto-rheological damper. A cost-effective sky-hook control strategy was proposed by Xubin (2009). The proposed model required one sensor whereas the conventional sky-hook strategy requires two sensors which reduced the cost and complexity of an automobile suspension system. Zapateiro et al. (2012) studied different semi-active control strategies applied to a quarter car system with MR damper. The different control strategies considered were back-stepping technique with H_∞ constraints and quantitative feedback theory. Ali and Ramaswamy (2009) also used back-stepping control algorithm.

Liao and Lai (2002) considered a single order system with MR damper and the system is controlled using sliding mode controller. Lai and Liao (2002) considered a single degree of freedom system with MR damper and proposed a semi-active control strategy which has two controller. The outer controller was sliding mode controller and the inner controller controls the voltage input to the damper using continuous-state control. Lam and Liao (2003) considered the same two nested control strategy controlled quarter car model with MR fluid damper. The performances were evaluated using transmissibility plots for harmonic excitation and responses for bump and random inputs are compared with sky-hook system. Ribakov and Gluck (2002) proposed a selective control strategy for base isolated system with MR fluid damper. An autonomous control system for a 3 degree of freedom pitch-plane suspension system was proposed by Sapinski and Rosol (2008). Yu et al. (2009) considered a half car suspension system. A Human Simulated Intelligence Control (HSIC) scheme has been proposed to reduce the vibration level and the performances were compared with that of conventional Linear Quadratic Gaussian (LQG) control method. Potter et al. (2011) proposed a quasi-active control law for Magneto-rheologcial dampers. Prabakar et al. (2009) considered a optimal control strategy for a half car system with MR damper.

1.3 Objectives of The Thesis

The brief literature review presented in the previous sections clearly indicates the varied nature of research activities on the use of magneto-rheological fluid based dampers for vibration suppression. Although different control strategies have been proposed which are supposed to be effective for different kinds of excitation, a detailed analysis of the usefulness of MR damper is not available in open literature.

The main objective of the present study is to investigate the performance of a MR-fluid damper when used as a part of a passive and a semi-active vibration isolator. To this end, attention has been paid to the following topics.

- Modeling and characterization of a MR-fluid damper: The objective of this work is to construct the lumped parameter model of a commercially available MR-fluid damper. With the help of this physical model the input-output characteristics of the damper can be estimated. The input-output behavior is normally expressed by impedance values at different frequency and excitation levels.

- Performance analysis of passive isolator with MR-fluid damper: Here the objective is to analyze the performance of a passive isolator that uses a MR damper as dissipative element. The performance of the isolator can be measured by different indices, namely, force transmissibility, motion transmissibility, relative motion transmissibility when the excitation is harmonic. Although the excitation to which a mechanical system is subjected may contain many frequencies or may even be random, the analysis of behavior under harmonic excitation provides useful insights that help one to take decisions in a situation more complex than considered here.

- Performance analysis of semi-active isolator using MR-fluid damper for isolating harmonic excitation: It is already stated in the previous section that the chief use of MR damper is to modify its damping by external energy supply so that optimal vibration isolation is achieved. The first aim of this part is to study how a MR-fluid damper performs when its damping is semi-actively controlled according to a pre-fixed logic. The disturbance excitation is considered to be harmonic. The second objective is to study the chatter problem associated with this isolator. The other aim of this section is to provide feasible solutions for improving performance of the isolator.

- Study of isolator characteristics of semi-actively controlled MR-fluid damper under transient and shock excitation: The primary objective is to investigate the behavior of a semi-active isolator when the system is excited by a shock type excitation. The performance of the isolator, when used to mitigate the problem of transient excitation, can be measured by different shock spectra. The other objective of this part is to provide curtain guidelines for improving the performance of shock isolator.

- Performance analysis of semi-actively controlled MR-fluid damper under stochastic excitation: In most of the situations a system, which is required to be protected, is subjected to random or stochastic excitation whose nature may widely vary depending upon the application. The statistical properties of the excitation are specified by the power-spectral density functions. The effectiveness of the isolator can be quantified with the help of similar statistical measures. The aim of this part of the thesis is to study behavior of the damper under different kinds of stochastic excitation.

The above mentioned objectives are pursued in the thesis, both analytical and numerically. In the following pages the work carried out on each topic is briefly described.

1.4 Thesis Outline

The thesis consists of seven chapters of which this is the first. In the present chapter a brief literature review of the researches on different aspects of MR fluid dampers have been outlined.

In chapter 2, a mono-tube MR damper is modeled and input-output characteristic is analytically estimated. Equivalent stiffness and damping coefficients and mechanical impedance of the MR damper are analytically calculated. The response of the MR damper for varying magnetic field is shown in this chapter.

Harmonic response of a magneto-rheological isolator is studied in chapter 3. The magnetic field is kept constant for a system which is excited by either a harmonic force or base excitation. The solution of the system is analytically calculated using a modified technique which is a combination of averaging technique and harmonic balance. The performance characteristics for a nonlinear isolator are formulated as force and motion transmissibilities.

In chapter 4, a two stage isolator with a magneto-rheological damper is considered which is controlled using on-off sky-hook control strategy. The problem of jerk and chatter, introduced from high frequency excitation, are addressed and different anti-jerk solutions are presented. An optimization study has been made to find out suitable voltage range which produces better result for harmonically excited system.

The response of the same two stage isolator for transient excitation or shock inputs are studied in chapter 5. Responses for different types for shock inputs, namely pulse and step inputs, are studied. New performance indices for shock isolation, such as maximax displacement, distortion amplitude, residual amplitude etc are formulated. An optimization study is carried out to find the optimal range of voltages for which the isolator performs better.

Chapter 6 discusses the response of the isolator when subjected to random excitation. The time signal of a random excitation whose power spectral density (PSD) is given is calculated numerically using Shinozuka method. The PSD of the responses of the system is studied. Attempt has been made to find an optimum value of the voltage which leads to the best performance of the isolator.

The main contributions of thesis and the future work are discussed in the last chapter.