Control of a uni-axial magnetorheological vibration isolator

Shuo Wang

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A Dissertation

entitled

Control of a Uni-Axial Magnetorheological Vibration Isolator

by

Shuo Wang

Submitted to the Graduate Faculty as partial fulfillment of the requirements for the Doctor of Philosophy Degree in Engineering

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May 2011
An Abstract of

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The technologies of hybridization of vehicles have been proven to significantly improve the fuel economy and reduce the environmental pollution. These technologies combine additional power sources with a traditional internal combustion engine. In some other modern vehicles, advanced cylinder management is the means to reduce fuel consumption and emissions. While these advanced technologies aim at energy savings and preserving the environment they create additional noise, vibration and harshness (NVH) problem.

The noise, vibration and harshness problem has been a major area of research in the automotive industry. Vibration is the main cause for noise. With the advent of alternative energy and hybrid vehicles come new vibration problems and challenges that require nontraditional solutions. Semi-active vibration isolation devices are preferred to address the problem due to their effectiveness and affordability. A magnetorheological (MR) fluid mount can provide effective vibration isolation for applications such as hybrid
vehicles. The MR fluid can produce different levels of damping when exposed to
different levels of magnetic field. The fluid can be working in three modes which are the
flow mode, shear mode and squeeze mode. A mixed mode MR fluid mount was designed
to operate in a combination of the flow mode and the squeeze mode. Each of the working
modes and the combined working mode has been studied. The mount’s performance has
been verified in simulation and experiments.

The focus of the current study is on the design of a control system for the mixed
mode MR fluid mount. Based on a model for the uni-axial MR mount a controller has
been designed to achieve the lowest possible vibration transmissibility. Furthermore, the
MR mount in two degree of freedom structure has been modeled. Displacement
transmissibility and force transmissibility are considered in this scenario. It is desirable to
minimize both transmissibilities. The controllers to achieve the lowest value for each type
of transmissibility were designed. Moreover, a hierarchical controller for realizing the
tradeoff between these two types of transmissibility was also constructed. At last, a fuzzy
logic controller is devised to closely reproduce the effect of the hierarchical controller.
The experiments are set up to realize the hardware-in-the-loop tests. Results from the
experiments show that the mixed mode MR fluid mount is able to achieve desired
dynamic stiffness which is directly related to vibration transmissibility.

This study provided a fundamental understanding on the behavior of MR fluid
mount in a single degree of freedom model and a two degree of freedom model. The
significantly reduced transmissibility demonstrates effectiveness of the designed control
system. The results of this research can shed some light on developing the control system
for other effective isolation devices.
Acknowledgements

I would like to express my great gratitude toward my advisor, Dr. Mohammad H. Elahinia. I thank him for taking me into his advisement so that I can work on this project. Due to his wonderful guidance, encouragement, and management, this dissertation can happen. I would send my thank you to my co-advisor Dr. Mansoor Alam for his considerate advice to make it smooth for each step of my research. I would like to show my deep appreciation to Dr. Mohamed Hefzy for supporting me to go through some changes. I would also like to thank Dr. Ezzatollah Salari for his generosity of his time and advices. Great appreciation is also sent to Dr. Mohsin Jamali for his teachings and accommodations. I show my thanks to all for taking their precious time serving as my committee members. Special thanks to the US EPA, US ARMY, and UTUTC who grant the financial support so that this research could be accomplished.

I am especially thankful to Dr. The Nguyen, Walter Anderson and Dr. Constantin Ciocanel as my research collaborators for their selfless assistance. I would also like to thank Tom Jacob for his generous and warm help. I thank all the members of Dynamic and Smart System Laboratory who make me laugh and feel encouraged.

I would like to show my love toward my family who are willing to sacrifice everything for me. Without them, I would never understand the substance of love and life.
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<tr>
<td>CAD</td>
<td>Computer Aided Design</td>
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<tr>
<td>DOF</td>
<td>Degree of Freedom</td>
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<tr>
<td>DS</td>
<td>Dynamic Stiffness</td>
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<tr>
<td>ECU</td>
<td>Electronic Control Unit</td>
</tr>
<tr>
<td>ER</td>
<td>Electrorheological</td>
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<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
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<tr>
<td>FLC</td>
<td>Fuzzy Logic Control</td>
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<tr>
<td>FIS</td>
<td>Fuzzy Inference System</td>
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<tr>
<td>HEV</td>
<td>Hybrid Electric Vehicle</td>
</tr>
<tr>
<td>HHV</td>
<td>Hydraulic Hybrid Vehicle</td>
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<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
</tr>
<tr>
<td>LQG</td>
<td>Linear Quadratic Gaussian</td>
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<tr>
<td>MR</td>
<td>Magnetorheological</td>
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<tr>
<td>NVH</td>
<td>Noise, Vibration and Harshness</td>
</tr>
<tr>
<td>P/M</td>
<td>Pump/motor</td>
</tr>
<tr>
<td>SDOF</td>
<td>Single Degree of Freedom</td>
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<tr>
<td>TDOF</td>
<td>Two Degree of Freedom</td>
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<tr>
<td>SMA</td>
<td>Shape Memory Alloy</td>
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<td>SME</td>
<td>Shape Memory Effect</td>
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$A_i$ .................. cross-sectional area of the flow path

$A_p$ .................. equivalent piston area of the top rubber

$b_r$ .................. upper rubber damping coefficient

$b_u$ .................. unsprung damping

$C_1$ .................. compliance of the top rubber

$C_2$ .................. compliance of the bottom rubber

$C_{sq}$ ................. damping coefficient induced from squeeze mode effect

$D_h$ .................. hydraulic diameter

$F_{exc}$ ................ excitation force

$F_{sq}$ ................ reaction force induced from squeeze mode effect

$F_{trans}$ .............. transmitted force

$H$ ..................... magnetic field strength

$h$ ..................... height of inertia track channel

$h_0$ ................. initial gap between the magnetic poles in squeeze mode

$I_i$ .................. fluid inertance

$k_r$ .................. upper rubber stiffness

$k_u$ .................. unsprung stiffness

$L$ ..................... length of the inertia track (flow passage)

$M$ ..................... mass of the engine

$M_c$ .................. mass of chassis

$m$ ..................... mass of the supported body

$P_1$ .................. pressure in the upper chamber

$P_2$ .................. pressure in the lower chamber

$Q_i$ .................. fluid flow rate

$R$ ..................... radius of the squeeze plate

$Ri$ .................. fluid resistance
\( x \) ................ displacement of the supported mass
\( \dot{x} \) ................ velocity of the supported mass
\( \ddot{x} \) ................ acceleration of the supported mass
\( y \) ................ displacement of the bottom
\( \dot{y} \) ................ velocity of the bottom
\( \ddot{y} \) ................ acceleration of the bottom
\( z \) ................ displacement of the chassis
\( \dot{z} \) ................ velocity of the chassis
\( \ddot{z} \) ................ acceleration of the chassis
\( x_i \) ................ fluid displacement through the inertia track
\( \dot{x}_i \) ................ fluid velocity through the inertia track
\( \ddot{x}_i \) ................ fluid acceleration through the inertia track
\( \Delta P_{MR} \) ................. pressure difference induced by the flow mode effect
\( \eta \) ................ viscosity of the fluid
\( \rho \) ................ density of the fluid
\( \tau_y \) ................ yield stress of the fluid induced by the magnetic field
Chapter 1

Introduction

Over a hundred thousand patents have been invented and many names have been remembered in the history of automobiles [1]. The evolution of the automotive technologies has never stopped to improve vehicle performance. Comfort of the ride has been one of most important aspects in developing the advanced technologies among power, fuel efficiency, emission, safety and cost.

Hybrid vehicles, including hybrid electrical vehicle (HEV) and hydraulic hybrid vehicle (HHV), gain their popularity in the modern vehicle development because of their significant improvement in fuel economy and less pollution to the environment. The majority of main stream vehicles are equipped with an internal combustion engine (ICE) which uses fossil fuels as the power source. When another power source is added, which could be electrical, hydraulic or any other form of power storage and reuse, a “hybrid vehicle” concept is formed. There are two types of hybrid vehicles: parallel or series. Take HHV as an example; there are parallel hydraulic hybrids and series hydraulic hybrids whose configurations are shown in Figure 1-1 and Figure 1-2.
For parallel hydraulic hybrids, the accumulators are the energy storage units and the hydraulic pump/motor (P/M) and engine are the drive units and the reservoir is for the fuel storage. Both the engine and pump/motor are connected to the same drive shaft. When the vehicle is braking, it drives the P/M (pump mode) to take the oil from reservoir to the accumulator which is called regenerative braking. When the vehicle is accelerating, the oil in high pressure accumulator pushes the P/M (motor mode) to drive the vehicle. The engine only works when the power of P/M is not sufficient to drive the vehicle.
The series hydraulic hybrids are more efficient because the ICE is decoupled. P/M 1 is attached to the ICE and P/M 2 is connected to the driveshift. Two accumulators are two passages between P/M 1 and P/M 2. When the vehicle is braking, the vehicle drives P/M 2 (pump mode) to take the oil from the lower pressure accumulator to the higher pressure accumulator. When the vehicle is accelerating, the high pressure accumulator pushes P/M 2 (motor mode) to drive the vehicle. The engine needs to operate only when all the energy stored from the regenerative braking process is used up and more energy is needed to propel the vehicle.

1.1 Noise and vibration and Harshness (NVH) problem in modern vehicles

In the modern vehicle industry, the NVH problem has always been the focus for the sake of a comfortable ride for the driver and the passengers. The NVH problem, also known as noise and vibration (NV), are caused by the vibration of the engine, driveline, road surface and brakes. Harshness reflects a subjective impression.

In hybrid vehicles, the NVH problem is more prominent than the conventional vehicles. For conventional cars, the source of the vibration usually is the internal combustion engine. While in hybrid vehicles, the additional power sources make a great contribution to the problem. For either the electric generator/motor in hybrid electric vehicles or the hydraulic pump/motor(s) in the hydraulic hybrid vehicles, the switching of the operational modes aggravates the vibration problem.

It also should be noticed that advanced cylinder management has been adopted by some vehicle manufactures. For a regular V6 engine, the six cylinders are working in balance. By adopting the variable cylinder management, 6 cylinders are firing when the
vehicle is starting or accelerating, 4 cylinders are firing when the vehicle is at high-speed cruising and 3 cylinders are firing when the vehicle is at mid-speed cruising. The dynamic balancing has been greatly broken in such a strategy.

The noise, vibration and harshness problem has always been a big challenge in the design of vehicles. Variable solutions have been invented for the vibration isolation in the vehicle history. For modern vehicles, more advanced isolation techniques are required according to the more and more complicated development of the vehicles.

1.2 Solution of Magnetorheological fluid devices

Jacob Rabinow invented magnetorheological (MR) fluids in the 1940s but the technology of MR fluids was not practical until recently when other technologies began to be available for a real possibility of MR technology [3]. Typically, MR fluids contain 20–40 percent relatively pure 3-10 micron diameter iron particles by volume. These micron-sized iron particles are suspended in a carrier liquid, for instance, water, mineral oil, synthetic oil or glycol. The suspended iron particles align and produce yield strength under the effect of a magnetic field. The change from a free-flowing liquid to a semi-solid is rapid and reversible in the presence of a magnetic field. The maximum yield strengths of 50-100 kPa can be developed in the presence of the magnetic fields of 150-250 kA/m. MR fluids are not highly sensitive to moisture or other contaminants in the process of manufacture and usage. Also the magnetic polarization mechanism is not affected by temperature so that MR based devices could be used over a broad temperature range.

There are three working modes of MR fluids: flow (valve) mode, shear mode, and squeeze mode. These three working modes are shown in Figure 1-3. When MR fluid
operates in valve mode with fixed magnetic poles it can be used for hydraulic controls, servo valves, dampers, and shock absorbers. When MR fluid operates in direct shear mode with a moving pole it is possible for use in clutches and brakes, chucking/locking devices, dampers, breakaway devices and structural composites. When MR fluid operates in squeeze mode with two magnetic poles moving toward each other it is suitable for small amplitude vibration and impact dampers.

Figure 1-3: Three basic working modes of MR fluids [4].

MR fluid-based devices are able to work in a wide range of frequencies from zero to a few hundred hertz due to their millisecond respond time. Therefore, MR fluids provide great potentials for many vibration isolation applications in all three working modes over a wide frequency range [3].

According to LORD Corporation, the main manufacturers of MR fluids and related products, compared to conventional electro-mechanical solutions, MR vibration isolation solutions provide the following list of benefits: quick response time (less than 10 milliseconds), real time, continuously variable control of damping, simple design (few or no moving parts), consistent efficacy across extreme temperature variations, high dissipative force independent of velocity, greater energy density, minimal power usage (typically 12V, 1 Amp max current; fail-safe to battery backup), and inherent system
stability (no active forces generated) [3]. In contrast to electrorheological (ER) fluids, MR fluids have 20-50 times higher yield strength; also they are significantly less sensitive to contaminants and extremes in temperatures; furthermore, they can be operated directly from low-voltage power supplies. MR technology can offer flexible controllability which is less complicated and more reliable than those based on ER technology.

MR technology has been moved out of the laboratory and into viable commercial applications which include automotive primary suspensions, truck seat systems, cab suspensions, control-by-wire and tactile feedback devices, seismic mitigation and human prosthetics. MR technology can provide solutions for the following markets: automotive, military vehicle suspensions, off-highway & construction vehicles, agriculture vehicles and other applications.

A schematic of a MR seat suspension damper is shown in Figure 1-4. The SD-1000 linear MR fluid damper is a small monotube design, for use in a semi-active suspension system in large on/off highway vehicle seats. This damper is able to provide a wide range of force control for modest input power levels. The damper is 21.5 cm long in full extension and is 3.8 cm in diameter and has a stroke of 2.5 cm [5]. The piston, the magnetic circuit, an accumulator and 50 ml of MR fluid are housed in the cylinder. A linear current driver provides current for the electromagnet in the piston head and induces the magnetic field. The required peak power is less than 10 watts and the generated damping forces could be up to 3000 N. The stable working temperature ranges from -40 to 150 degrees Celsius. The rise time of the force generated by the damper is approximately 8 milliseconds when a step in the voltage is applied to the current driver.
The main reason of this short time delay is due to the time that the MR fluid takes to reach rheological equilibrium and the time associated with the dynamics of driving the electromagnet in the damper.

Figure 1-4: Schematic of the MR damper by Dyke et al [1].

A 20-ton large scale MR fluid mount is illustrated in Figure 1-5. When a current is introduced into the damper, a magnetic field is produced and the iron particles suspended in the fluid line up with the magnetic field and form chains, and therefore change the behavior of the controllable fluid. The magnetic field intensity influences the viscosity of the liquid and is proportional to the amount of current applied to the damper [6]. On a multi-story building, the placement of the MR damper has a significant impact on the equivalent damping ratio and also on the natural frequency of the building.
A schematic of a MR fluid brake (Lord MRB-2107-3) which is 9.2 cm in diameter and provides a controlled dissipative torque of 0-6 Nm for speeds up to 1000 RPM as shown in Figure 1-6 [7]. MR brakes can be used in cycling and stair climber type aerobic exercise machines and commonly used with velocity feedback so that the torque is controlled in real time and the user has to maintain a desired target speed. The MR fluid brakes require far less power than conventional brakes. They become very cost effective for a wide variety of applications such as controllable exercise equipment or precision active tension control due to their simplicity and ease of control.
MR fluid can operate in three modes: flow mode, shear mode and squeeze mode. The flow mode is the most widely used mode and motion of piston pushes the fluid through the valves located on the piston or between the inner and outer cylinder. It is also used in designing the mechanical mounting devices such as vehicle powertrain mounts [8]. In the MR mount, the flow takes place in the inertia track connecting the top and bottom chambers of the mount. Shear mode is used in a smaller number of vibration isolators. This mode can be used in the damper [9] or mount [10] and combined with the other modes. Using shear mode needs the small gap between two surfaces and large surface area. Also the shear mode is more commonly used for rotating applications [11].

MR fluid brakes can help reduce torsional vibration using the friction between the disks. One disk is stationary and the other is attached to the rotating body. Squeeze mode is the least understood and applied among the three working modes because of the complexity in both design and mathematical formulation. Squeeze mode devices usually require a
small gap between two parallel surfaces with large area. Additionally, promising vibration isolators can be made utilizing squeeze mode [12]-[14]. Research to better understand the properties of the squeeze working mode only has been carried out by [15], [16]. Working in combined modes has been investigated in some researches [9], [10], [17]. Separate coil sets are employed to generate different magnetic field for operating in different modes. The combined mode is effective when one of the modes loses their normal working conditions. Combining different modes of MR fluids is the most effective way to achieve vibration isolation.

Various levels of stiffness and damping are needed for MR devices based on the specific applications. To achieve different stiffness and damping, MR fluids change their rheologies by changing the magnetic field which is proportional to the voltage or current running through the electromagnetic coil. Therefore, the behaviors of the MR devices are ultimately controlled by altering the input voltage or current.

Ahn et al. presented a MR mount with a valve controlled by the presence and absence of the magnetic field. When the valve is closed, the MR mount displays high stiffness while when the valve is open, the mount shows low stiffness. The transmissibility shows that the switching frequencies $\omega_{s1}$ and $\omega_{s2}$ are where the valve changes its state from closed to open state and from open to closed state as illustrated in Figure 1-7. By opening and closing the valves at the switching frequencies, the lowest transmissibility can be secured. The numerical challenge is the main shortcoming which can influence the response time of MR mount.
Reichert [19] presented a study of a MR damper for vehicle seat suspension with different control methods including skyhook control and groundhook control and the combination of these two. In these control algorithms, the relative velocity is positive when the base and mass are separating, the absolute velocity of suspended mass is positive when move upwards. When semi-active damping force is applied in the same direction as the skyhook damping force, we applied this damping force; while semi-active damping force cannot be applied in the same direction as the skyhook damping force, the semi-active damper is desired to be set as zero ideally, though usually there is a small damping force. Also the paper discussed the alternative control strategies of groundhook control as well as the combination of skyhook and groundhook. The results show that the hybrid control achieves the best vibration isolation for their system.

Li et al. [20] presented a fuzzy control strategy for a quarter-car suspension system model with a MR fluid damper. Based on the modified Bouc-Wen model of MR fluid, the inherent nonlinear behavior of MR fluid was determined by 14 parameters.
Fuzzy logic control method is adopted and a micro genetic algorithm was used for the optimization of the rules. The simulation results show that the fuzzy control optimized by the micro genetic algorithm can effectively control the acceleration of the sprung mass, suspension travel and tire deflection which provides more comfortable ride and handling stability.

1.3 Problem statement and Motivation

MR fluid based devices have been proposed and verified by universities and companies. A few commercialized products, such as MR dampers and MR brakes/clutches are the most widely used. However, for MR mount products, there is not much available on the market. The most famous one is the MagneRide which developed by Delphi. The research and commercialization of MR mounts are still attractive to researchers and corporations. Most MR mount designs employ one of three working modes. However, each single working mode has its limitations. The analytical model and fabrication of a single-axis semi-active MR fluid mount has been completed in previous work [21]. The mount can be operated in two modes (or a combination thereof): flow mode and squeeze mode.

The MR mount can achieve a large range of dynamics stiffness over a wide range of frequencies. It is preferred that the MR mount can achieve high dynamic stiffness for low frequency vibration and low dynamic stiffness for high frequency vibration and also be able to tune the dynamic stiffness at any working frequency. The control of the MR mount is crucial to obtain the desired dynamic stiffness over all working range. This dissertation is about the continuation of the research to develop a suitable controller for the mount. To this end a control system will be designed and tested. The control
algorithms will be considered so that the MR mount will be effective and able to respond to all possible shock and vibration conditions with minimal use of energy.

### 1.4 Objectives

The main objective of this study is to develop an effective controller for the single axis mixed mode MR fluid mount working in a combination of flow mode and squeeze mode. Specifically, the goals are:

1. to model and simulate the MR mount in a single degree of freedom (SDOF) structure and design the controller to achieve the lowest transmissibility,

2. to model and simulate the MR mount in a two degree of freedom (TDOF) structure and design the controller to achieve the lowest transmissibility,

3. to develop a hierarchical controller for two degree freedom to achieve a trade-off between the lowest force transmissibility and displacement transmissibility,

4. to test the controller for prototype of the MR mount to verify the design of the controller.

### 1.5 Outline

This chapter introduces the purpose of this dissertation and the rest is arranged as follows. Chapter two presents the background on the types of vibration isolators which have contributed to the development of MR mounts.
Chapter three describes the design, modeling and fabrication of proposed mixed mode MR fluid mount.

Chapter four presents modeling of the MR fluid mount in a single degree of freedom structure and in the two degree of freedom structure. The skyhook control has been used in each situation.

Chapter five displays the results from simulation in the SDOF and the TDOF system. A hierarchical controller is designed to manage the trade-off between force transmissibility and displacement transmissibility for the TDOF system based on the results. Experimental settings and results are shown also.

Chapter six concludes the dissertation and reveals certain future work needed for the advance of the MR mount research.
Chapter 2

Background

2.1 Types of Mounts

Mounts have been developed as effective vibration isolators to prevent vibration transmission between the two structures which they connect. Typically, these two parts involve a vibrating machine and a supporting structure in different systems such as vehicles, ships, airplanes, or buildings. Passive rubber mounts have been widely researched [22] and manufactured to support static loads and isolate vibration. For the nonresonant and high frequency domain, the rubber mounts exhibit favorable performance due to their low damping. While for resonance of low frequency domain, passive hydraulic mounts [23]-[27], which have high damping, are efficient choices to meet large stiffness requirements. However, the vibration isolation performance in the nonresonant domain is worse than that of rubber mount. Passive mounts cannot control the damping and stiffness simultaneously and therefore their performances are limited to meet the vibration isolation requirements.

Active control systems are also widely investigated [28]-[32]. External energy supplied by actuators to generate forces to counteract the vibration. These actuators
include an electromagnetic actuator, hydraulic servo actuator, and piezoelectric actuator. Although active mounts show good vibration efficiency compared to passive mounts, they require external energy to operate the actuators and supply the control forces. Such systems demand a relatively large amount of power in many applications. For instance, the civil engineers are reluctant to use active control devices for earthquake protection because these devices are vulnerable to power failure. Active control devices are not stable in terms of sensor failure especially for a centralized control system.

Semi-active control systems [33]-[43] can adjust the damping and the stiffness to isolate the vibration in the system without putting energy into the system. Semi-active devices are preferable due to their capability of varying their properties in real time with a minimal amount of power. They are stable in the sense that semi-active devices can only absorb or store vibratory energy in a system by reacting to its motion. MR fluid-based devices are regarded as semi-active devices. MR fluids exhibit very short response time with a substantial yield strength which makes them a desirable material for semi-active devices, especially for small and medium-size devices. The magnetic field changing the MR fluid’s properties is induced by the electromagnets installed in the mount. Typically when the magnetic field increases, the stiffness and damping will also increase. The geometry of the mount, the installation configuration of the coil and the properties of the MR fluid are critical for the behavior of the MR mount.

2.2 Conventional solutions to vibration isolation

The conventional solutions to vibration isolation are elastomeric mounts and hydraulic mounts. Elastomeric, or rubber, mounts are the original and simple design for isolating vibration. Besides providing a support to a payload, it also offers some isolation
for vibration. A typical elastomeric engine mount, seen as in Figure 2-1, is usually composed by a block of black rubber and two threaded rod extensions.

![Figure 2-1: Typical rubber engine mount [44].](image)

For a hydraulic mount, shown in Figure 2-2, hydraulic oil, glycol or other similar fluids are filled in the upper chamber and lower chamber. Both chambers are enclosed by rubber. The decoupler and inertia track are designed to provide the damping. When the displacement excitation is small, the decoupler is activated. It will vibrate up and down in its cage and the fluid can flow around it. Less energy is absorbed and lower damping is provided. The effective piston area, the compliance of top rubber and bottom rubber become the dominating parameters. When the displacement excitation is large, the decoupler will seat at the bottom of its cage and the fluid will be forced through the inertia track. More energy is absorbed through the inertia track and higher damping is provided.
Figure 2-2: Cross-section of a typical hydraulic engine mount [45].

The dynamic stiffness of a hydraulic mount and a rubber mount has been shown in Figure 2-3. For the rubber mount, the figure displays linearity between dynamic stiffness and the frequency. For the hydraulic mount, there exists a notch frequency and a peak frequency. It is clear that the dynamic stiffness curve is fixed, or at least without much change, as long as a rubber mount is designed. The same as the hydraulic mount, the mount is designed so that the notch could be placed at a certain frequency at which most vibration for the system happens. Neither rubber mount nor hydraulic mount is able to provide variable dynamic stiffness at a certain frequency.
Figure 2-3: Dynamic stiffness curves for a hydraulic mount and a rubber mount [46].

One design of hydraulic mount that draws special interest is shown in Figure 2-4. Fourmani et al. include shape memory alloy (SMA) wires into the structure of the top rubber. By controlling the current into the SMA wires, the top rubber compliance will be changed. Shape memory alloy (SMA) actuators have attracted significant attention due to their many potential applications in mechanical, electrical, civil, medical and aerospace systems. SMA actuation has many advantages over conventional actuation methods because of its significantly reduced weight, size, complexity and noiseless operation. The benefits over other smart materials such as piezoelectric materials, electrostrictive materials and magnetostrictive materials are the extremely high force to weight ratio, and large displacement. Among all the shape memory alloys such as Ag-Cd, Au-Cd, Cu-Sn, In-Ti, Ni-Al, Ni-Ti, Mn-Cu, Nickel-Titanium (Ni-Ti) has proven to be the most promising one in many kinds of applications. This is due to its greater ductility, more recoverable motion, excellent corrosion resistance, stable transformation temperatures,
high biocompatibility and convenience to be electrically heated. SMAs’ shape memory effect (SME) takes place due to transformation between the martensite phase (weak and stable at a lower temperature) and austenite phase (strong and stable at a higher temperature). When an SMA is in the 100% martensite phase it is heated from $A_s$ (austenite start temperature) to the phase transformation temperature $A_f$ (austenite finish temperature), the crystalline structure will transform to 100% austenite. When the SMA in the austenite phase is cooled down from $M_s$ (martensite start temperature) to the phase transformation temperature $M_f$ (martensite finish temperature) the crystalline structure will transform to martensite. An exertion of external stress causes the “twinned” martensite to change to “detwinned” martensite below $A_s$; a large strain will remain with the removal of the stress. When the “detwinned” martensite is heated above $A_f$ the remaining strain is completely recovered which realizes the “memory” effect.

Figure 2-4: Adaptive hydraulic engine mount by Foumani et al [47].
For Fourmani et al.’s design, the martensite and austenite phases of the SMA wires provided the maximum and minimum of top rubber compliance. On-off control method is adopted to switch between maximum and minimum values of the top rubber compliance. As a matter of fact, the hysteresis behavior can be observed as one of main disadvantages for control of SMA. Other nonlinear phenomena such as nonlinear heat transfer, nonlinear changes in temperature and stress also are undesired characters of SMA. Furthermore, SMA actuators have to operate at a low bandwidth, mainly due to the slowness of its cooling mechanism. All these factors make the control of SMA actuators a difficult topic and smart designs of control strategies are still required for these so called smart materials.

### 2.3 Mechanical Models for MR Devices

Models for MR fluid devices must be developed well enough to adequately characterize the intrinsic behavior to develop effective control algorithms that take maximum advantages of the unique features of the MR devices. Several mathematical models have been developed to feature the behaviors of MR fluid.

The Bingham viscoplastic model is the most popular model for describing behavior of MR fluids as shown in Figure 2-5. Based on model of the rheological behavior of ER fluids, Stanway et al. proposed and idealized mechanical model which comprises a Coulomb friction element placed parallel to a viscous damper. In this model, the plastic viscosity is defined as the slope of the measured shear stress versus shear strain rate data. Before the shear stress reaches the yield stress, the fluid is still. When it comes to post-yield state, i.e. the shear stress exceeds the yield stress, the fluid starts flowing like a highly viscous material. The shear stress is the sum of the yield stress and
the product of the viscosity and positive value of the shear rate. This model doesn’t exhibit nonlinear force-velocity response for some cases. This model is only good enough for response analysis but not adequate for the control analysis.

Figure 2-5: Schematic of Bingham plastic model for the MR damper [48].

Gamota et al. proposed an extension of the Bingham model which is illustrated by the viscoelastic-plastic model shown in Figure 2-6 predicting the behavior of ER materials. In this model, a standard linear solid model is in series with the Bingham model which is a frictional element in parallel with a dashpot. Its main shortcoming is the numerical challenges.

Figure 2-6: Schematic of model proposed by Gamota and Filisko [49].

The Bouc-Wen model is one which is numerically tractable and widely adopted in modeling hysteretic systems as shown in Figure 2-7. A Bouc-Wen model consists of a passive spring and a damper in parallel with a nonlinear element which is to predict the
hysteretic behavior. The Bouc-Wen model proved to be versatile in modeling the MR fluid damper. It can predict the force-displacement behavior of the damper well and possess force-velocity behavior more closely resembles to experimental data. However, the original Bouc-Wen model is not capable of fully capturing the nonlinear behavior of the ER/MR fluid. However, it has been modified to develop more sophisticated models.

![Schematic of Bouc-Wen model for the MR damper](image)

Figure 2-7: Schematic of Bouc-Wen model for the MR damper [50].

A schematic of modified Bouc-Wen model has been proposed by Spencer et al. shown in Figure 2-8. In this model, the original Bouc-Wen model in series with a dashpot is placed in parallel with a spring. The dashpot creates the nonlinear roll-off observed in the force as the velocity approaches zero; and the spring is employed to account for the stiffness of the accumulator present in the prototype MR damper. The modified model can accurately predict the response of the MR damper over a wide range of operating conditions and therefore can be effectively used for control design and analysis.
2.4 *MR mounts*

Ahn et al. designed one of the earliest magnetorheological mounts as illustrated in Figure 2-9. The mount uses inertia track in a conventional hydraulic mount as the valve that open or close the MR fluid flow from upper chamber to the lower chamber and it works as the flow mode.

Figure 2-9: Schematic diagram of a magneto-rheological mount by Ahn et al [52].
Ha et al. presents a squeeze mode type MR mount as shown in Figure 2-10. The MR fluid filled the gap between the two cores and was enclosed by a strip of natural rubber. When the upper electromagnetic core and lower electromagnetic core are close enough, squeeze reaction force is generated by the fluid.

![Diagram of a squeeze mode MR mount]

Figure 2-10: A typical squeeze mode MR mount [53].

Researchers are attracted to incorporate multiple working modes since each single mode has its own limitations. Choi et al. proposed a shear and flow mixed mode MR mount as shown in Figure 2-11. Brigley et al. presented an MR vibration isolator which is claimed to work in all three modes as shown in Figure 2-12. However, questions could be raised to these attempts. By the definition of the three working modes, it is challenging to explain the simultaneity of flow mode and shear mode. Also these designs were not able to identify the contribution of each mode.
2.5 Literature Review for Control of MR mount

Various control algorithms for MR fluid-based devices have been researched and proposed so that these semi-active devices can achieve satisfactory performance for vibration isolation [56]-[65]. In order to develop an effective controller for the MR mount, a literature review has been conducted to learn about the existing control schemes used for MR devices in vibration mitigation. The knowledge from this review will consequently be used in the controller design process.
The control algorithms investigated as shown below include LQG control, groundhook control, skyhook control, fuzzy logic control, neural networks control and inversion based control and integrator backstepping based control. These control methods are either simple in design such as skyhook control and fuzzy control or more advanced such as optimal control and neural networks control. These controllers are used in different systems to achieve effective vibration isolation.

1. LQG control

Hong et al. designed a linear quadratic Gaussian (LQG) controller for a mixed mode MR fluid to isolate the vibration of a structural system. MR mount is mixed mode type with flow and shear modes. Nondimensional formulation of the Bingham plastic flow model is adopted. The structural system consists of a vibrating mass, semi-active MR fluid mount, and passive rubber mounts. The MR mount is installed as a semi-active isolator between the vibrating mass and the beam structure which is supported by two passive rubber mounts as shown in Figure 2-13. The rubber element on the top of the MR mount supports the mass on the one hand and isolates the vibration transmission at the nonresonant frequency on the other hand. The governing equation is presented and rewritten in a state space model. The LQG controller is designed and experimentally verified. For obtaining feedback signals, two accelerometers are attached on the vibrating mass and the flexible beam. The control voltage calculated by LQG and applied to the MR mount via a digital to analog converter and a current supplier. The acceleration levels of structural system are attenuated effectively by controlling the damping of MR mount. The force transmission through the two rubber mounts is also suppressed by activating the MR mount. However, the results show that acceleration and force
transmission with the LQG control are higher than the one without control at higher frequencies.

Figure 2-13: Structure system with MR fluid mount by Hong et al [60].

2. On-off groundhook (on-off DBG) control

On-off groundhook (on-off DBG) control based on displacement was proposed by Koo et al to effectively reduce the vibrations of the primary structure for a MR tuned vibration absorber. A tuned vibration absorber (TVA) is a vibratory subsystem attached to a primary system which consists of a mass, a spring and a damper, as shown in Figure 2-14. TVAs are connected to the primary structure to offset its motion. A controllable MR damper changes a passive TVA into a semi-active TVA. The relative velocity is defined as the velocity of the structure mass minus the TVA mass. For other cases, positive means the object goes up and negative means the object goes down. The on-off DBG control provides two damping values which are identified as on state and off state damping. Which value is adopted is dependent on the product of the relative velocity across the damper and absolute displacement of the primary structure. When the relative
velocity and the absolute displacement of the primary structure have the same direction, the on state damping value is used and when they have the opposite direction, the off state damping value is chosen. The test results demonstrate that different transmissibility curves when the MR mount is activated with different levels of input. The primary system alone has a resonant peak with a maximum transmissibility. Two resonant peaks appear when a TVA is added to the primary structure without activating the MR mount simply because a degree of freedom was added to the system with the TVA. When the on state current is increased, it reduced the two resonant peaks while it increased the transmissibility between the two peaks. Continuing to increase the current makes the two peaks become one and increases; which shows that the primary mass and the TVA mass are linked together. The unique resonant response is observed to be lower than that of primary structure alone. The study showed that the controlled TVA outperformed a passive system, the resonant peaks reduced and good isolation around the structure’s natural frequency.

Figure 2-14: Passive TVA and Semi-active TVA by Koo et al [61].

3. Skyhook control
Unsal et al. developed a skyhook control system as the semi-active scheme with the advantages of selective energy dissipation for a model of a 6 DOF parallel platform. Each leg of the platform is modeled as a 2 DOF system and installs a MR damper as shown in Figure 2-15. The Bingham plastic model is used and the relationships between the yield force and control current input as well as the post-yield damping coefficient the control current input are derived. The semi-active damper is developed to dissipate energy and designed to apply an input current to the damper only under the condition that the relative velocity between the two bodies and the absolute velocity of the clean body \(m_2\) are in the same direction. With the skyhook control method, the lowest transmissibility can be obtained for both 1 DOF and 2 DOF as shown in Figure 2-16 and Figure 2-17. The MR damper is either activated by applying a 2A control input current or turned off by setting the current to zero. The control algorithm reduces the peak at the first resonant frequency and decreases the transmissibility at higher frequency at high frequencies for 2 DOF. This made a good foundation for further extending the MR damper controlled 2 DOF to a 6 DOF parallel platform.

Figure 2-15: Two DOF platform leg model by Unsal et al [62].
Figure 2-16: Transmissibility of single DOF model [62].

Figure 2-17: Transmissibility of 2 DOF model [62].
4. Fuzzy skyhook/groundhook control

Ahmadian et al. developed a fuzzy skyhook/groundhook control for the semi-active MR system to improve the overall performance for the vehicle heave and roll motion during vehicle maneuvers. The roll place vehicle model is as shown in Figure 2-18. There are three steps of fuzzy logic control: fuzzification, rule evaluation, and defuzzification. In fuzzification, the vertical velocity of the axle, the relative velocity across the suspension, the absolute velocity of the vehicle body, and the acceleration of the vehicle body are chosen as four inputs and each input has their own membership functions. The damping value is chosen to be output of the controller and has its own membership function. The rules of the system are developed for the inputs and the output. In defuzzification, the fuzzy values are converted into a crisp value by the weighted average defuzzification method. The output is limited by its high and low state by comparing the output value with the ones of high and low state. The results show that this fuzzy logic control for the semi-active suspensions could greatly reduce the vehicle heave and roll displacements while it increases the body acceleration peak value for road inputs at the tire which cause a less comfortable ride.
Wang et al. presented the modeling and control of MR fluid dampers using neural networks. Based on the modified Bouc-Wen model of MR fluid, the inherent nonlinear behavior of MR fluid was modeled in feedforward neural networks (FNN) and recurrent neural networks (RNN) in both direct identification as the schematic shown in Figure 2-19 and inverse dynamics as the schematic shown in Figure 2-20. The results indicate that the direct identification dynamic model using RNN can predict the damping force accurately and the inverse dynamic model using RNN can act as a damper controller to generate the command voltage for the MR fluid damper in semi-active mode. The simulation results are satisfactory which provide a new method for the damper controller of the MR fluid damper. However, it requires training. The training data needs to be able to cover most situations of practical applications and also needs to be simple in order to speed up the training process. Furthermore, the neural networks works like a “black box” as the process of generating the outcome is not explicitly stated.
Figure 2-19: The identification for MR fluid damper using neural network model [64].

Figure 2-20: Inverse modeling for the MR damper using neural network [64]: (a) force to command voltage model; (b) force to displacement model

6. Hybrid structural control

Ali et al. used RD-1005-3 MR damper by Lord Corporation for their study of a base isolation system for a three-storey building. A single MR damper is connected at the base of the building as shown in Figure 2-21. The Bouc-Wen model is used and dynamic inversion based control and integrator backstepping based control are stated and compared with clipped optimal control, optimal fuzzy logic control. Clipped optimal control makes the control force suboptimal when change the voltage of MR damper from zero to its maximum value. From the results, the dynamic inversion based control and integrator backstepping based control are better than clipped optimal control and genetic algorithm based fuzzy logic control because the latter two controllers decreased the
isolator displacement at the cost of an increase in superstructure acceleration while the former two controllers provide a trade-off between isolator displacement and superstructure acceleration. They are able to monitor the voltage needed to control the structural vibration taking into account the effect of the supplied and commanded voltage dynamics of the damper.

![Schematic of a hybrid base isolated building by Ali et al [65].](image)

Figure 2-21: Schematic of a hybrid base isolated building by Ali et al [65].

### 2.6 Modeling and Control of SDOF

The Control of MR mounts is crucial in achieving the satisfactory vibration isolation. There are various types of vibration such as shock or impulse, periodic, harmonic with fixed frequency or over a range of frequencies, or random vibration. The controller must be robust and adaptive enough to offset these vibration excitations and also achieve optimal performance for the MR mount which demonstrate hysteretic and nonlinear characteristics. This section presents the control of single degree of freedom
model which provide valuable insight for the control of MR mount. Vibration control algorithms such as the bang-bang control, the skyhook control as well as fuzzy logic control are investigated using the simulation model.

![Diagram](attachment:image.png)

(a)

Figure 2-22: Modeling of SDOF (a) physical, (b) Simulink

The study of a single degree of freedom model is fundamental in mechanical vibrations. To understand more advanced subjects, SDOF is a rewarding start. This section starts with modeling and control for SDOF mass-spring-damper structure as schematically shown in Figure 2-22. The excitation is chirp signal of a displacement...
excitation from bottom and displacement transmissibility is investigated in the section. This system, as shown in this section, provides the possibility of evaluating different control strategies for this vibration system without detailed modeling for the MR system. To this end the MR system is modeled as a variable and controllable damping element.

1. Bang-bang control

A bang-bang controller is also called on-off controller which abruptly switches between two states. It is often used to control the plant from one value to the other based on the occurrence of some event. For example, a furnace is turned on or off according to the environment temperature. The bang-bang control can be expressed in a Heaviside step function. For the MR mount, the bang-bang controller can simply designed to switch between two jerk values of damping ratio ($\zeta$) according to an input velocity condition ($v$).

$$\zeta_{MR} = \begin{cases} \zeta_{\text{min}} & \text{if } v < v_0 \\ \zeta_{\text{max}} & \text{if } v \geq v_0 \end{cases}$$

It is not hard to deduce that this simplest format can be extended from two jerk values to a multi-value algorithm since the damping ratios are related to various levels of magnetic field strength. The controller can be designed with multiple levels of damping ratios including the minimum and maximum jerk values with the change of values due to the input velocity.

$$\zeta_{MR} = \begin{cases} \zeta_{\text{min}} & \text{if } v < v_0 \\ \zeta_1 & \text{if } v_0 \leq v < v_1 \\ \zeta_2 & \text{if } v_1 \leq v < v_2 \\ \vdots & \text{.....} \\ \zeta_{\text{max}} & \text{if } v_{n-1} \leq v < v_n \end{cases} \quad (\zeta_{\text{min}} < \zeta_1 < \zeta_2 < \ldots < \zeta_{\text{max}})$$
The bang-bang controller designed for the SDOF is shown in Figure 2-23. The excitation profile is the input for the function blocks. When the simulation is run, the equations are solved numerically in time domain. Several solvers are available in Simulink® for selection. The values for the parameters are fetched from a MATLAB® .m file. Since the Simulink® program is run in time domain, the MATLAB® file is also used to convert the data into frequency domain. The MATLAB® file produces the result graphs for processing and report. Using Fast Fourier Transform (FFT) technique, the displacement/force excitation and the transmitted force can be transformed into frequency domain. The dynamic stiffness/transmissibility can be calculated by dividing the transmitted force in frequency domain by the displacement/force excitation in frequency domain.

The calculation of dynamic stiffness:

\[
\begin{align*}
\text{Displacement Excitation} & \quad x(t) \rightarrow X(\omega) = \text{FFT}(x) \\
\text{Transmitted Force} & \quad f(t) \rightarrow F(\omega) = \text{FFT}(f) \\
\text{Dynamic Stiffness} & \quad K_{\text{dyn}}(\omega) = \frac{F(\omega)}{X(\omega)} = A + iB \\
\text{Amplitude} = |K_{\text{dyn}}| = \sqrt{A^2 + B^2}; & \quad \text{Phase} = \tan^{-1}\left(\frac{B}{A}\right)
\end{align*}
\]

The calculation of transmissibility:
**Force Excitation** \[ f_{in}(t) \rightarrow F_{in}(\omega) = FFT(f_{in}) \]

**Transmitted Force** \[ f_{out}(t) \rightarrow F_{out}(\omega) = FFT(f_{out}) \]

**Transmissibility** \[ TR(\omega) = \frac{F_{out}(\omega)}{F_{in}(\omega)} = C + iD \]

\[ \text{Amplitude} = |TR| = \sqrt{C^2 + D^2}; \quad \text{Phase} = \tan^{-1}\left(\frac{D}{C}\right) \]

Figure 2-23: Modeling of SDOF with bang-bang control.

The result shows that the bang-bang controller provides lower transmissibility than the passive damping as illustrated in Figure 2-24. When the mass is not moving, the damping is off which is set to zero which the transmissibility is shown in dash red. When the mass starts to move, the damping value is set to a certain value which make the damper provide certain damping force. The transmissibility is shown in solid blue. However, for the high frequency range, the bang-bang controller shows a higher transmissibility.
Figure 2-24: Displacement transmissibility of SDOF: passive vs. bang-bang control.

Bang-bang control is simple control algorithm, but it requires full power at switching points which could lead the system to unstable states. Furthermore, the oscillation will happen when controller switches from the maximum jerk value to the minimum jerk value and vice versa. One more limitation is the bang-bang control requires the foreknowledge of the system behavior to choose the input condition values. The bang-bang control is not an ideal controller due to its energy consumption and damages to the systems and its inability to handle random systems.

2. Skyhook Control

To eliminate the tradeoff between resonance control and high frequency isolation, the skyhook control method is proposed [19]. The configuration changes the damper to connect from between the base and mass to an inertial reference in the sky which is a ceiling that remains vertically fixed relative to a ground reference as seen in Figure 2-25.
It is noticed that this configuration is a purely fictional. Practically, this effect can be realized by skyhook control algorithm:

\[
\begin{cases}
V_1 V_{12} \geq 0 & H = H_0 \\
V_1 V_{12} < 0 & H = 0
\end{cases}
\]

where:

\(V_1\) = absolute velocity of the suspended mass with respect to the ground

\(V_{12}\) = relative velocity of the suspended mass with respect to the base

\(H\) = magnetic field

\(H_0\) = value of the magnetic field

Figure 2-25: Passive based-excited system (left); Ideal skyhook configuration (right)[19].

The modeling of SDOF with the skyhook control strategy is illustrated in Figure 2-26. Both damping and stiffness have two jerk values and the timing of switch is needed to make the control effectively obtain the lowest transmissibility.
The skyhook control can achieve the lowest transmissibility as illustrated in Figure 2-27. Three cases are investigated: low stiffness with low damping, low stiffness with high damping, and high stiffness with high damping. It is easy to observe that the higher damping decreases the peak of transmissibility with the same stiffness and higher stiffness shift the peak to the right which is the higher frequency with the same damping. The desired profile for the vehicle on the road is that the mount has high stiffness for the low frequency for the shock and has low stiffness for the high frequency for the engine vibrations. The skyhook control algorithm is able to track the lowest transmissibility and therefore realize the desired profile.
3. Fuzzy logic control

Both bang-bang control and skyhook control involve the switching between values which may cause oscillation at the switching points. Fuzzy logic control has its popularity in control due to its ability of consulting several factors and continuous control surface. The modeling of SDOF with fuzzy logic control strategy is presented in Figure 2-28. The Fuzzy Logic Toolbox in Simulink is used to intelligently choose the value for damping and stiffness. The control algorithm is programmed in the Fuzzy Inference System (FIS) Editor. Membership Function Editor inside the FIS Editor is where the input and output membership functions are defined. The Rule Editor inside the FIS Editor is where the rules are added. The model adopted in the SDOF is Mamdani type. Three inputs for the FIS are relative displacement, relative velocity between the base and suspended mass, as well as absolute displacement. The output for FIS is damping or stiffness. The triangular membership functions are used for both inputs and outputs.

Figure 2-27: Realization of lowest displacement transmissibility using skyhook control.
The fuzzy logic control can effectively lower the peak of transmissibility and is also able to shift the peak to the left as shown in Figure 2-29. However, the understanding the system before your controller design is required to set the values for the membership functions such that the fuzzy logic controller can best achieve the vibration isolation.
Figure 2-29: Displacement transmissibility of SDOF: passive vs. fuzzy logic control.

4. Summary

Different control strategies: bang-bang, skyhook, fuzzy logic, are investigated for the SDOF. Bang-Bang control is simple and easy to implement. Skyhook control can track the lowest transmissibility. Fuzzy logic can provide adaptive control result while it requires expertise for designing the controller parameters. Each control strategy has its own advantages and disadvantages. They all contribute to the control aspect of the vibration isolation. All these simulations provide a solid foundation for the controller design of a mixed mode MR fluid mount. Comparatively speaking, skyhook control and fuzzy logic control are more advanced ones and therefore are used in controller design of MR fluid mount in this research.
Chapter 3

MR Fluid Mount

The design and modeling of ER and MR mounts have been investigated by many researchers. A unique MR mount design is proposed by Dynamic and Smart Systems Laboratory [21]. The proposed MR mount have been designed based on the design of hydraulic mounts. A MR fluid mount consists of several parts: a rubber top, two hollow chambers, an inertia track, a center rod and squeezing plates. Each part contributes to the overall behavior of the mount which is specified by mathematical formulas. In order to capture the behavior of the MR each of these parts should be modeled. In this chapter, the proposed mount design with advanced features in [21] is recapitulated. The characteristics of the mount are identified in order to develop a control algorithm that can fully accommodate to the unique design.

3.1 Design and modeling of hydraulic mount

Since the design of the MR mount is based on a similar hydraulic mount, in this section the later are briefly discussed. A simple hydraulic mount design has two chambers and a flow path as illustrated in Figure 3-1. The parameter and variable of the top chamber are the compliance $C_1$ and the pressure $P_1$. The parameter and variable of the bottom chamber are the compliance $C_2$ and the pressure $P_2$. The parameters of the flow
path are its fluid inertia $I_i$ and flow resistance $R_i$ and they are both related to the geometry of the flow path. The flow path can be a short orifice or a long inertia track. The top rubber property is expressed by the stiffness $k_r$ and the damping coefficient $b_r$.

Figure 3-1: Configuration of a hydraulic mount: (a) schematic, (b) physical model [21].

The pressure difference between the top and bottom chambers is calculated based on the linear moment equation:

$$P_1 - P_2 = I_i \dot{Q}_i + R_i Q_i$$

(3.1)

where $Q_i$ is the fluid flow rate through the passage, $I_i$ and $R_i$ are the fluid inertance and resistance respectively, $P_1$ and $P_2$ are pressures in the top and bottom chambers respectively. In addition, the generalized flow continuity equations for the systems yields based on [66]:

$$\dot{P}_1 = \frac{A_p}{C_1} \dot{x} - \frac{Q}{C_1}$$

(3.2)

$$\dot{P}_2 = \frac{Q_i}{C_2}$$

(3.3)
where $x$ is the displacement at the mount top, $A_p$ is the equivalent piston area, $C_1$ and $C_2$ are the top and bottom chamber compliances respectively. The overall equation of motion of the mount can be written as

$$M\ddot{x} + b_r\dot{x} + k_r x + A_p P_1 = F_{exc}$$

(3.4)

where $M$ is the loading mass, $F_{exc}$ is the excitation force, $b_r$ and $k_r$ are the rubber damping coefficient and stiffness respectively. The hydraulic parameters can be defined as below [67]:

$$I_i = \frac{\rho L}{A_i}$$

(3.5)

$$R_i = \frac{128\eta L}{\pi D_h^4}$$

(3.6)

where $\rho$ is the density of the fluid, $L$ is the length of the flow path, $A_i$ is the cross-sectional area of the flow path, $\eta$ is the viscosity of the fluid, $D_h$ is the equivalent “hydraulic diameter” of the flow path. Hydraulic diameter is generally used for non-circular cross-sections.

3.2 Design and modeling of mixed mode MR mount

The previous researches of MR mounts mostly operate in a single working mode. Even for the combined mode designs, the effects of each mode are not able to be separately identified. This makes it difficult to analyze and further control the behavior of the MR mount in different situations. The proposed novel MR mount incorporates the
flow working mode and squeeze working mode induced by separate electromagnetic coils. Therefore the effects of each working mode and combined working modes can be investigated. The schematic of the mixed mode MR fluid mount is shown in Figure 3-2. When the MR fluid travels from one chamber to the other through the inertia track, the outer coil is activated and produces the magnetic field, the flow mode is on. The activation of the inner coil can prevent the motion of the rod which is the squeeze mode. The magnetic field at the inertia track and plates can be activated individually or at the same time according to the magnitude of the input excitation. The stiffness and damping are therefore adjusted by the activation of the coils to achieve the low transmissibility.

![Schematic design of the mixed mode MR mount](image)

Figure 3-2: Schematic design of the mixed mode MR mount [21].

The design concept of the mixed mode MR mount is illustrated in Figure 3-3. In the design of the mixed mode MR mount, flow mode and squeeze mode are working independently since the coils are placed separated and therefore the control of each working mode of the MR fluid can be individually or simultaneously. Besides the hydraulic structure in the design, the MR fluid element is added which makes a MR fluid mount.
Figure 3-3: Configuration of the MR mount: (a) schematic, (b) physical model [21].

The physical laws for hydraulic mount are still applied for MR mount since the authors consider that the MR fluid is Newtonian when the field is not activated. This assumption is applied to equations (3.1) and (3.4) to get

\[ P_1 - P_2 = I_i \dot{Q}_i + R_i Q_i + \Delta P_{MR} \tag{3.7} \]

\[ M \ddot{x} + b_x \dot{x} + k_x x + C_{sq} \dot{x} + F_{sq} + A_p P_1 = F_{exc} \tag{3.8} \]

where \( C_{sq} = \frac{3\eta \pi R^4}{2(h_0 + x)^3} \) \( \tag{3.9} \) and \( F_{sq} = \frac{4\pi R^3 \tau_y (H)}{3(h_0 + x)} \text{sign}(\dot{x}) \) \( \tag{3.10} \)

The convention adopted here for the effect of the squeeze mode is the subscript “sq” [68].

\( R \) is the radius of the squeeze plate, \( h_0 \) is the initial gap between the magnetic poles in squeeze mode after the preload is applied. \( C_{sq} \) and \( F_{sq} \) are the damping coefficient and reaction force introduced by the squeezing working mode. Moreover, according to [69], the pressure difference induced by the MR effect can be expressed as

\[ \Delta P_{MR} = C \frac{L}{h} \tau_y (H) \text{sign}(\dot{x}) \tag{3.11} \]
where $\Delta P_{MR}$ is the pressure drop due to the yield stress $\tau_y$ of the MR fluid which depends on the magnetic field $H$; $h$ is the distance between the magnetic poles, which is equal to the height of a rectangular flow passage channel, or equal to the difference between the outer radius and inner radius of an annular flow path; $C$ is a constant in the range of 2 to 3 depending on the steady-state flow conditions. $C$ is chosen to be 2 in this work representing the low-flow conditions.

Integrating equations (3.2') and (3.3') and substituting the expressions for $P_1$ and $P_2$ in equations (3.7, 3.8) yields

$$\frac{A_p}{C_1}x - \frac{A_i x_i}{C_1} - \frac{A_i x_i}{C_2} = I_i A_i \ddot{x}_i + R_i A_i \dot{x}_i + \Delta P_{MR}$$  \hspace{1cm} (3.12)$$

$$M \ddot{x} + b \ddot{x} + k x + C_{sq} \dot{x} + F_{sq} + \frac{A_p^2}{C_1} x - \frac{A_p A_i x_i}{C_1} = F_{exc}$$  \hspace{1cm} (3.13)$$

where $A_i \ddot{x}_i$ and $A_i \dot{x}_i$ were substituted for $Q_i$ and $\dot{Q}_i$, respectively; $x_i$ is the fluid controlled volume displacement.

Rearranging equation (3.12, 3.13) arrives at the following coupled equations of motion for the MR mixed mode mount:

$$I_i A_i \ddot{x}_i + R_i A_i \dot{x}_i + A_i \left( \frac{1}{C_1} + \frac{1}{C_2} \right) x_i - 2 \frac{L}{h} \tau_y (H) \text{sign}(\dot{x}_i) = \frac{A_p}{C_1} x$$  \hspace{1cm} (3.14)$$

$$M \ddot{x} + b \ddot{x} + \left( k + \frac{A_p^2}{C_1} \right) x + C_{sq} \dot{x} + F_{sq} = \frac{A_i A_p}{C_1} x_i + F_{exc}$$  \hspace{1cm} (3.15)$$
The transmitted force is calculated in equation (3.16) and it is further used to calculate for the transmissibility and the dynamic stiffness. The transmissibility is the ratio between the transmitted force and the excitation force. The dynamic stiffness is the ratio between the transmitted force and the excitation displacement.

\[ F_T(t) = b_r \ddot{x} + \left( k_r + \frac{A_p^2}{C_1} \right) x + C_{sq} \dot{x} + F_{sq} - \frac{A_c A_p}{C_1} x_i \]  
(3.16)

### 3.3 Fabrication of mixed mode MR mount

The mixed mode MR mount prototype was designed in Solidworks®, a computer aided design (CAD), and then was manufactured as a physical mount.

A cut-out view of the designed mixed mode MR mount is showed in Figure 3-4. The main components of the mount are numbered in this figure as follows: ① - upper rubber part, ② – bottom rubber part, ③ – inner coil, ④ – inner coil housing, ⑤ - outer coil, ⑥ - outer coil housing, ⑦ – flow passage, ⑧ - mount housing, ⑨ – closing collar, ⑩ – upper mounting screw, ⑪ – upper squeeze plate, ⑫ – lower mounting screw.
Figure 3-4: Section view of the mixed mode MR mount in Solidworks® [21].

The top rubber ① provides a support for the static load applied to the mount which has a very low compliance value, and bottom rubber ② is designed to contain the MR fluid which has a very high compliance value. They enclose the upper chamber and bottom chamber and both are from an original equipment manufacturer (OEM) hydraulic mount. The inner electromagnetic coil ③ contributes to activate the squeeze working mode. The inner coil housing ④ for directing the magnetic field was made by highly magnetically permeable material. The outer electromagnetic coil ⑤ is responsible for the activation of the flow working mode. The same magnetically permeable material was used for the outer coil housing ⑥ to help direct the magnetic flux inducted by the outer
coil. The flow passage⑦, the annular gap, is where the flow working mode happens when the MR fluid is passing through. The mount housing ⑧ provides the protection for the bottom rubber and the necessary space for the bulging of the rubber. The closing collar ⑨ can be tightened against the mount housing by using eight screws, which completes the assembly of the mount. The upper mounting screw ⑩ makes a connection between the supported mass (the engine) and the upper squeeze plate⑪. The lower mounting screw ⑫ connects the mount housing and the chassis.

The upper rubber is made of hard rubber and the bottom rubber is made of soft rubber. They are shown in green. The inner coil and outer coil are made of copper wire and they are shown in red. The inner coil housing, outer coil housing and upper squeeze plate are made of 1018 Steel as shown in pink. The mount housing and closing collar are made of aluminum as shown in black or gray.

The inner coil assembly and outer coil assembly are shown in Figure 3-5 and Figure 3-6. The inner coil is placed in the inner coil housing and enclosed by the non-magnetic ring. The outer coil is enclosed by the coil housing and top disk and bottom disk.
The middle assembly in exploded view can be seen in Figure 3-7. The sizes for each part are designed to fit each other so that the whole middle assembly can be put together into a cylinder.
Figure 3-7: The entire Middle Assembly in exploded view [21].
Each part of the mixed mode MR fluid mount has been manufactured. The fabricated inner coil assembly, outer coil assembly and upper portion of the mount and lower portion of the mount are shown in Figure 3-8 to Figure 3-11.

Figure 3-8: Fabricated inner coil assembly [21].

Figure 3-9: Fabricated outer coil assembly [21].
3.4 Experimental verification of mixed mode MR mount

The MR fluid mount has been tested in ElectroForce® 3330 test instrument by BOSE Corporation as shown in Figure 3-12. The machine provides a displacement excitation from the top, a load cell shown as the black cylinder is used to measure the
force transmitted through the MR fluid mount. Based on the displacement and measured force, the dynamic stiffness could be calculated and displayed by the WinTest® DMA software.

![Experimental setup for the mixed mode MR mount](image)

**Figure 3-12:** Experimental setup for the mixed mode MR mount [21].

The flow mode effect can be shown in Figure 3-13.

![Flow mode only, displacement excitation of 1.0 mm](image)

**Figure 3-13:** Flow mode only, displacement excitation of 1.0 mm [21].
The squeeze mode effect is shown in Figure 3-14.

![Figure 3-14: Squeeze mode only, displacement excitation of 1.0 mm [21].](image)

The individual and combined mode effect is shown in Figure 3-15.

![Figure 3-15: Comparison in performance of individual modes and combined mode [21].](image)

The flow mode works to increase dynamics stiffness at lower frequencies and decrease dynamics stiffness at higher frequencies. The flow mode shows the profile similar to an elastomeric mount at high current input. Also when the current used is high enough, the MR fluid is solidified and the dynamic stiffness doesn’t change any more. When this state is reached the flow of the fluid is totally blocked. The increase of the
current doesn’t cause any increase of the dynamic stiffness. The squeeze mode works as a spring in the mount, the dynamic stiffness increase when we increase the current used for the coil for squeeze working mode. Furthermore, the two modes can be worked in combined mode which shows the “superposition” principle, a simple addition of the effects of the two modes.

These experimental results show that the mixed mode MR fluid mount greatly enlarge the dynamic stiffness area with combined working mode. The mount is able to adjust to provide the appropriate dynamic stiffness for the change of the vibration conditions so that the lowest transmissibility is able to be realized.

The comparison between the simulation and experiment for flow mode, squeeze mode and combined mode are shown in Figure 3-16 to Figure 3-18.

![Graph showing the comparison between theoretical prediction and experimental result for flow mode only.](image)

Figure 3-16: Theoretical prediction vs. Experimental result for flow mode only [21].
As shown in these figures, the simulation results match the experimental results closely. A small discrepancy happens at the very small portion frequency range. These
results are due to the fact of incomplete degassing. A setup of a vacuum filling station is shown in Figure 3-19. It helps to eliminate the trapped air in the mount.

![Figure 3-19: A setup of vacuum filling station.](image)

The MR fluid reservoir is sufficiently large to prevent the fluid being sucked into the vacuum line when it is degassed. The air trap is used to protect the vacuum pump if the MR fluid is pulled into the vacuum line. 6 valves are used to form the vacuum lines. The vacuum procedure is elaborated as following steps:

1. Close valve 1, 4 and 5, and open 2 and 3 to vacuum the MR fluid mount by the energizing the vacuum pump. The vacuum should hold at least one hour indicating the mount is without leak.
2. Close valve 1, 3, 5 and 6, and open 2 and 4 to check if the MR fluid reservoir is air tight as step 1.
3. Close valve 4 and 5 and open 6 the let air into MR reservoir.
4. Add MR fluid in to the reservoir through the lid on the top of the reservoir (which is not seen from the perspective in Figure 3-19). Make sure the added MR fluid is 20% more than the required to fill the mount.

5. Tighten the lid on the top of the reservoir and seal it.

6. Close valve 1, 3, 5 and 6 and open 2 and 4. Energize the pump to degas the MR fluid in the reservoir. Eventually the MR fluid will become quiescent and no more foaming on the surface of the fluid.

7. Close valve 4 and open 6 to let the air pressure in the reservoir equal to the one outside.

8. Slowly open valve 5 to let the MR fluid into the mount. Watch and avoid the whirlpool and sucking air into the fluid.

9. When there is no more drop of the fluid in the reservoir, close valve 5.

10. Slowly shake the mount and reopen valve 5 to see if the mount can accept more fluid.

11. Repeat steps 9-10.

12. Remove the MR fluid mount from the vacuum filling apparatus.
Chapter 4

Control of MR Fluid Mount

In order to understand how the MR fluid mount works, the modeling of the MR mount in single degree of freedom system and two degree of freedom system have been built. Also the skyhook control methodology for both situations is also investigated. In two degree of freedom system, force transmissibility and displacement transmissibility are both investigated.

4.1 Modeling of MR mount in Single DOF

The design concept of the mixed mode MR fluid mount with a supported mass is illustrated in Figure 4-1. Among the most important functions of engine mount, two of them are to accommodate engine block misalignment from the residual stresses due to chassis or body frame distortion and protect the engine from excessive movement from rough roads [71]. Therefore, response for the displacement excitation from bottom is inspected. And the displacement transmissibility is defined as the ratio of the movement of the supported mass over the movement of the excitation. Furthermore, for a single degree of freedom model, the displacement transmissibility for displacement excitation from bottom shares the same expression as the force transmissibility for force excitation.
Investigating the displacement transmissibility with displacement excitation from bottom is equivalent, at least show the same trend, to do the force transmissibility with force excitation from top in the sense that MR fluid mount is a more advanced spring & damper device with variable stiffness and damping.

Figure 4-1: Configuration of excitation from the bottom of the mixed mode MR mount with a supported mass: (a) schematic, (b) physical, model modified based on [21].

The trend of the equations of this model follows those in section 3.1 and 3.2 except the following ones due to the structure of single degree of freedom:

\[ \dot{P}_1 = A_y (\dot{x} - \dot{y}) / C_1 - Q_1 / C_1 \]  

(4.1)

\[ C_{sq} = 3\eta \pi R^4 / 2(h_0 + (x - y))^3 \]  

(4.2)

\[ F_{sq} = 4\pi R^3 \tau_y (H) \text{sign}(\dot{x}) / 3(h_0 + (x - y)) \]  

(4.3)

The overall equation of motion for the MR fluid mount can be written as:

\[ m \ddot{x} + b_r (\dot{x} - \dot{y}) + k_r (x - y) + C_{sq} (\dot{x} - \dot{y}) + F_{sq} + A_y P_1 = 0 \]  

(4.4)
Rearranging those equations arrives at the following coupled equations of motion for the MR mixed mode mount:

\[
I_i A_i \ddot{x}_i + R_i A_i \dot{x}_i + A_i \left( \frac{1}{C_1} + \frac{1}{C_2} \right) x_i + 2 \frac{L}{h} \tau_y (H) \text{sign}(\dot{x}_i) = \frac{A_p}{C_1} (x - y) \quad (4.5)
\]

\[
m \ddot{x} + (b_r + C_{sq})(\ddot{x} - \dot{y}) + \left( k_r + \frac{A_p^2}{C_1} \right) (x - y) + F_{sq} = \frac{A_i A_p}{C_1} x_i \quad (4.6)
\]

The system of equations (4.5) and (4.6) are constructed in MATLAB/Simulink® to obtain the response of MR fluid mount. Also the skyhook controller is designed based on the modeling of the MR fluid mount.

**4.2 Skyhook control of MR mount in Single DOF**

Skyhook control has been applied to the control of the MR damper in previous researches [19, 33]. The equations governing the control algorithms are:

\[
\begin{cases}
\dot{x}(\dot{x} - \dot{y}) > 0 & \text{MagneticField ON} \\
\dot{x}(\dot{x} - \dot{y}) < 0 & \text{MagneticField OFF}
\end{cases}
\]

Where

\[\dot{x} = \text{absolute velocity of the supported mass}\]

\[\dot{x} - \dot{y} = \text{relative velocity between the supported mass and the base}\]

When the absolute velocity of the supported mass and the relative velocity between the supported mass and the base have the same direction, the magnetic field is turned on. When they have the opposite direction, the magnetic field is turned off. The skyhook control is fulfilled by turning on different levels of the magnetic field which is realized by controlling the input current to the coils.
This dissertation investigates the effect of skyhook control of flow mode and squeeze mode separately and simultaneously. For the skyhook control on flow mode only, the control law is

\[
\begin{align*}
    \dot{x}(\dot{x} - \dot{y}) > 0 & \quad H_F = H_F^{\text{max}} \\
    \dot{x}(\dot{x} - \dot{y}) < 0 & \quad H_F = H_F^{\text{min}}
\end{align*}
\]

where \( H_F \) is the field for flow mode, \( H_F^{\text{min}} \) and \( H_F^{\text{max}} \) are the minimum and maximum values used for the flow mode.

For the skyhook control on squeeze mode only, the control law is

\[
\begin{align*}
    \dot{x}(\dot{x} - \dot{y}) > 0 & \quad H_S = H_S^{\text{max}} \\
    \dot{x}(\dot{x} - \dot{y}) < 0 & \quad H_S = H_S^{\text{min}}
\end{align*}
\]

where \( H_S \) is the field for squeeze mode, \( H_S^{\text{min}} \) and \( H_S^{\text{max}} \) are the minimum and maximum values used for the squeeze mode.

For the skyhook control on both flow mode and squeeze mode, the control law is

\[
\begin{align*}
    \dot{x}(\dot{x} - \dot{y}) > 0 & \quad H_F = H_F^{\text{max}}, \ H_S = H_S^{\text{max}} \\
    \dot{x}(\dot{x} - \dot{y}) < 0 & \quad H_F = H_F^{\text{min}}, \ H_S = H_S^{\text{min}}
\end{align*}
\]

4.3 Modeling of MR mount in Two DOF

Solutions to the noise, vibration and harshness (NVH) issue have always been interested in many fields especially in the automotive industry for designing vehicles with improved ride and handling characteristics. The frame is subject to both external excitation source as road condition and internal excitation as engine or pump/motor vibration when a vehicle is moving on the road. The suspension is designed to eliminate the vibration transmitted from the road to chassis. Meanwhile, the chassis is at the mercy
of the vibration transmitted from the engine or the pump/motor of hydraulic hybrid vehicles (HHV) and engine mounts are required.

Displacement transmissibility [61], [62], and [73] and force transmissibility [74] are the most popular index to display the performance of the vibration isolation. The control algorithms are designed to target realizing the lowest values. Most papers focus on the analysis of one type transmissibility: either displacement transmissibility or force transmissibility. Research on both types of transmissibility is seldom seen and how to minimize both requires more attention. This section studied the behavior of a mixed mode MR fluid mount used in a two DOF model based on quarter car concept. Both displacement transmissibility and force transmissibility are examined for the effect of flow mode and squeeze mode separately and simultaneously and skyhook control on flow mode and squeeze mode separately and simultaneously.

A two degree of freedom (TDOF) model based on quarter car concept is built to investigate the effect of the MR fluid mount for preventing the vibration transmitted from engine to the chassis. Because the vibration isolation ability of the MR fluid mount is the focus, force excitation from the engine is adopted and the unsprung components are simplified to a damper and a spring as shown in Figure 4-2.
The trend of the equations of this model follows those in section 3.1 and 3.2 except the following ones due to the structure of two degree of freedom:

\[ \dot{P}_1 = A_p (\ddot{x} - \ddot{z}) / C_1 - Q_i / C_1 \]  
\[ C_{sq} = 3\eta \pi R^4 / 2(h_0 + (x - z))^3 \]  
\[ F_{sq} = 4\pi R^3 \tau_y (H) \text{sign}(\ddot{x}) / 3(h_0 + (x - z)) \]

The governing equations characterizing the system were derived as:

\[ I, A, \ddot{x}_i + R_i, \dddot{x}_i + A_i \left( \frac{1}{C_1} + \frac{1}{C_2} \right) x_i + 2L \tau_y (H) \text{sign}(\ddot{x}_i) = A_p (x - z) \]  
\[ M\dddot{x} + (b_x + C_{sq})(\dddot{x} - \dddot{z}) + \left( k_r + \frac{A_p^2}{C_1} \right) (x - z) + F_{sq} - \frac{A_i A_p}{C_1} x_i = F_{exc} \]
\[(b_r + C_{sq})(\ddot{x} - \dot{\dot{z}}) + \left( k_r + \frac{A_p^2}{C_1} \right) (x - z) + F_{sq} - \frac{A_i A_p}{C_1} x_i - k_u z - b_u \dot{z} = M_c \ddot{z} \quad (4.12)\]

Equation (4.12) represents the movement of chassis. The previously unspecified parameters appeared in the equations of motion are: \(x\): displacement of the engine, \(z\): displacement of the chassis, \(M\): mass of engine, \(k_u\): unsprung stiffness, \(b_u\): unsprung damping, \(M_c\): mass of chassis.

The transmitted force can be represented by:

\[F_{\text{trans}} = (b_r + C_{sq})(\ddot{x} - \dot{\dot{z}}) + \left( k_r + \frac{A_p^2}{C_1} \right) (x - z) + F_{sq} - \frac{A_i A_p}{C_1} x_i \quad (4.13)\]

Both the force transmissibility and displacement transmissibility can be investigated with this model as the excitation force, transmitted force, and engine displacement, chassis displacement are either known or calculable. The force transmissibility is defined as the ratio of the transmitted force divided by the excitation force. The displacement transmissibility is defined as the ratio of the displacement of the chassis divided by the displacement of the engine. They are shown in the following equations:

Transmissibility _ Force \[= \frac{F_{\text{trans}}}{F_{\text{exc}}} \quad (4.14)\]

Transmissibility _ Displacement \[= \frac{z}{x} \quad (4.15)\]

The system of equations (4.10) (4.11) and (4.12) are constructed in MATLAB/Simulink® to obtain the response of MR fluid mount. Furthermore, the skyhook controller is designed on the basis of the modeling of the MR fluid mount.
4.4 Skyhook control of MR mount in Two DOF

When the skyhook control for the MR mount is used in two degree of freedom model shown in Figure 4-2 the governing equations can be represented as:

\[
\begin{aligned}
\dot{x}(\dot{x} - \dot{z}) > 0 & \quad \text{Field On} \\
\dot{x}(\dot{x} - \dot{z}) < 0 & \quad \text{Field Off}
\end{aligned}
\]

Where

\[\dot{x} = \text{absolute velocity of the engine}\]
\[\dot{x} - \dot{z} = \text{relative velocity between the engine and the chassis}\]

In this two degree of freedom mode, when the absolute velocity of the engine and the relative velocity between the engine and the chassis have the same direction, the magnetic field is turn on. When they have the opposite direction, the magnetic field is turn off.

This dissertation investigates the effect of skyhook control of flow mode and squeeze mode separately and simultaneously. For the skyhook control on flow mode only, the control law is

\[
\begin{aligned}
\dot{x}(\dot{x} - \dot{z}) > 0 & \quad H_F = H_{F\ max} \\
\dot{x}(\dot{x} - \dot{z}) < 0 & \quad H_F = H_{F\ min}
\end{aligned}
\]

For the skyhook control on squeeze mode only, the control law is

\[
\begin{aligned}
\dot{x}(\dot{x} - \dot{z}) > 0 & \quad H_S = H_{S\ max} \\
\dot{x}(\dot{x} - \dot{z}) < 0 & \quad H_S = H_{S\ min}
\end{aligned}
\]

For the skyhook control on both flow mode and squeeze mode, the control law is

\[
\begin{aligned}
\dot{x}(\dot{x} - \dot{z}) > 0 & \quad H_F = H_{F\ max}, H_S = H_{S\ max} \\
\dot{x}(\dot{x} - \dot{z}) < 0 & \quad H_F = H_{F\ min}, H_S = H_{S\ min}
\end{aligned}
\]
Chapter 5

Results

After the models of the MR fluid mount in single degree of freedom and two degree of freedom and the skyhook control algorithm had been constructed, the equations were built in MATLAB/Simulink® environment. From the simulation, the control method was successfully verified to achieve the lowest possible transmissibility. Experiments are also conducted with the hardware in the loop. The experimental results can verify that the MR fluid mount can provide the desired dynamic stiffness for the working frequency.

5.1 Simulation results of modeling and control in SDOF

As presented in this section, simulations have been conducted to investigate the effect of the flow mode and the squeeze mode. Based on the results, simulations for the skyhook control were also performed to observe the effectiveness of the skyhook control laws for the flow mode and the squeeze mode separately and simultaneously. The simulation is on the displacement transmissibility for SDOF in the frequency range from 0 Hz to 100 Hz because of the working range of the engine. The noisy data in some
Simulation results are due to numerical solver limitation. The displacement excitation is chosen to be a chirp signal with constant amplitude and the range of frequencies of interest.

Figure 5-1 demonstrates the effect of flow working mode. The major changes of transmissibility take place in 0-20 Hz which is shown for flow mode. When magnetic field for the flow mode is turned on from 0kA/m, the extra damping is provided which leads to the decrease of peak resonance value. Also the stiffness of the mount is increased due to the field which results in higher resonance frequency. When the magnetic field for the flow mode is increased to 35kA/m, the fluid starts to solidify; as a result the damping is reduced, therefore the peak value increases. Higher stiffness at higher field strength results in higher resonance frequencies. When the magnetic field is increased to 100kA/m, the fluid flow path is totally blocked. This is due to the fact that when the magnetic field reaches a certain value, the yield stress of the MR fluid is so high that the pressure difference cannot drive the fluid through the flow path. At this point, there is no more damping by the fluid provided, so the peak value goes up back to the value when there is no field. The stiffness won’t change after the fluid path is completely blocked. The resonance frequency therefore stays at around 8.5Hz when the field keeps increasing after 100kA/m, for example to 250kA/m.
Figure 5-1: Simulation result of effect of flow mode only without control.

Figure 5-2 shows the effect of the squeeze working mode. When the magnetic field for the squeeze mode is turned on, the peak value decreases because the squeeze mode provides additional damping to the fluid mount. The squeeze mode acts to increase the stiffness mainly when the magnetic field for the squeeze mode is increased to 45 kA/m. The resonance frequency then moves to higher values. Squeeze mode acts to increase damping at low frequencies and acts to increase stiffness at higher frequencies (comparatively speaking).

Figure 5-1 shows the best peak value of transmissibility is 3.5 dropped from around 4.6 when the flow working mode is activated. The resonant frequency from the flow mode changes from 7 Hz to 8.5 Hz. Figure 5-2 shows that the peak value of transmissibility drops from around 4.6 to 1 when the magnetic field is increased to more than 45kA/m. Compared to the flow working mode, the squeeze working mode performs more effectively. The resonant frequency from squeeze mode changes from 7 Hz to 15.5 Hz.
Figure 5-2: Simulation result of effect of squeeze mode only without control.

Figure 5-3 shows the transmissibility plot when skyhook control law is applied to the flow mode. The maximum value for the magnetic field for flow mode is chosen to be 100kA/m for two reasons. As shown in figure 5-1, the value of the magnetic field for flow mode doesn’t influence the transmissibility very much after 100kA/m. The second reason is that this value provides the lowest transmissibility at the lower frequency before it crosses over the transmissibility line for no field. Figure 5-3 shows that the skyhook control tracks the lowest transmissibility between without field and with field. By skyhook control on flow mode itself, lower peak value is obtained and the resonant frequency is around 8 Hz.

Figure 5-4 shows the transmissibility plot when skyhook control law is applied to the squeeze mode. 250kA/m is chosen to be the maximum value for the magnetic field for squeeze mode due to the MR fluid working field limit. From this figure the increase of the magnetic field always influences the shape of the transmissibility plot. As shown in this result, the transmissibility values decreases below 1. The lowest transmissibility is
obtained and the peak disappears when the skyhook control for the squeeze mode is used. Compared to the skyhook control on flow mode, skyhook control on squeeze mode provides better vibration isolation observed from the plots.

Figure 5-3: Simulation result of skyhook control on flow mode only.

Figure 5-4: Simulation result of skyhook control on squeeze mode only.
Figure 5-5 to Figure 5-7 show the combined skyhook control on both flow mode and squeeze mode. When the magnetic field for squeeze mode is small, turning on the magnetic field for flow mode can show some obvious influence on the transmissibility as shown in Figure 5-5. Increasing the magnetic field for the squeeze mode, the effect of flow mode becomes less apparent as observed in Figure 5-6 and Figure 5-7. When both the flow mode and squeeze mode are activated and controlled simultaneously, the squeeze mode exhibits a dominant effect in contributing to the decrease of transmissibility. However, the combined skyhook control can provide better isolation at higher frequencies, though the difference between squeeze mode and both modes is minor. The minor difference is observed more clearly in the logarithmic scale as shown in Figure 5-8. Figure 5-7 shows the best transmissibility which can be achieved by skyhook control given the maximum applicable magnetic fields for flow mode and squeeze mode.

Figure 5-5: Simulation result of skyhook control on both modes H_Smax=30kA/m & H_Fmax=100kA/m.
Figure 5-6: Simulation result of skyhook control on both modes $H_{S_{\text{max}}}=45\text{kA/m}$ & $H_{F_{\text{max}}}=100\text{kA/m}$.

Figure 5-7: Simulation result of skyhook control on both modes $H_{S_{\text{max}}}=250\text{kA/m}$ & $H_{F_{\text{max}}}=100\text{kA/m}$. 
Figure 5-8: Simulation result of Skyhook control on both modes on log scale

\[ H_{S\text{max}}=250\text{kA/m} & H_{F\text{max}}=100\text{kA/m}. \]

5.2 Simulation results of modeling and control in TDOF

The previous simulations with single DOF system allowed for investigation of the control in the displacement transmissibility. In order to investigate force transmissibility, simulations have been performed to investigate the effect of the flow mode and the squeeze mode separately and simultaneously. These simulations cover both displacement transmissibility and force transmissibility. The excitation is a chirp signal of force.

Figure 5-9 presents the displacement transmissibility and force transmissibility without activating either the flow mode or the squeeze mode. The significant changes of transmissibility take place in the 0-20 Hz range. The MR fluid mount works as a hydraulic mount. The peaks of these two kinds of transmissibility happened at different
frequencies. The force transmissibility represents the resultant force effect which combined the motion of the engine and chassis, and therefore appears two peaks, while the displacement transmissibility shows one peak because the effect of the excitation force on both engine and chassis could be uncoupled. The effect of MR fluid mount and skyhook control law are investigated for the displacement transmissibility and force transmissibility respectively.

![Graph showing transmissibility](image)

**Figure 5-9**: Displacement transmissibility and force transmissibility without fields.

**FORCE TRANSMISSIBILITY**

Figure 5-10 to Figure 5-14 demonstrate the simulation results for the force transmissibility. Figure 5-10 shows the effect of flow working mode. The activation of flow mode doesn’t decrease the first peak significantly but obviously contribute to the decrease of the second peak. The extra damping is supplied when the magnetic field for the flow mode is turned on which caused the drop in the peak value. The stiffness is also increased at a higher frequency caused by the field. Increasing the field continuously leads to the solidification of the MR fluid inside the mount and the damping become
decreases; which results in the increase of the peak again. Figure 5-11 shows the effect of squeeze working mode. The activation of the squeeze mode reduces the both peaks when the magnetic field for the squeeze mode is turned on to a small value. The effect of squeeze mode is more obvious than the flow mode to some extent. However, the increase of the field for squeeze mode causes a parallel increase of the transmissibility for the whole working frequency range.

![Graph showing the effect of flow mode for force transmissibility.](image)

*Figure 5-10: Effect of flow mode for force transmissibility.*

![Graph showing the effect of squeeze mode for force transmissibility.](image)

*Figure 5-11: Effect of squeeze mode for force transmissibility.*
Figure 5-12 and Figure 5-13 show the transmissibility plot when skyhook control law is applied to the flow mode and squeeze mode. The principle of choosing the value of the applied field for the skyhook control is based on the simulation results for the effect of the field for each working mode. The field value is chosen which can produce the lowest transmissibility before the crossover point of the transmissibility without field and the one with field. Both figures demonstrate that the skyhook control can track the lowest transmissibility for the whole working frequency. Skyhook control on flow mode decreases the second peak and increases the resonant frequency. Skyhook control on squeeze mode decreases the first peak and makes the second peak disappear. From the observation of these two figures, skyhook control on squeeze mode provides better vibration isolation compared to the skyhook control on flow mode.

Figure 5-12: Skyhook control on flow mode for **force** transmissibility.
Figure 5-13: Skyhook control on squeeze mode for **force** transmissibility.

Figure 5-14 shows the combined skyhook control on both flow mode and squeeze mode. The skyhook control on either mode can reduce the transmissibility with no doubt. However, the combined skyhook control can provide better isolation at higher frequencies even though the difference between squeeze mode and both modes is minor. It is also observed that when both the flow mode and squeeze mode are activated and controlled simultaneously, the squeeze mode exhibits a more influential effect in contributing to the decrease of transmissibility. It can be concluded that to achieve the lowest force transmissibility by skyhook control, the magnetic field for flow mode and squeeze mode should be set to 50kA/m and 15kA/m, respectively.
Figure 5-14: Skyhook control on combined mode for force transmissibility.

**DISPLACEMENT TRANSMISSIBILITY**

Figure 5-15 to Figure 5-19 demonstrate the simulation results for the displacement transmissibility. The same investigations for the effect of the flow mode, the effect of the squeeze mode, skyhook control on flow mode and squeeze mode as well as on both modes has been conducted. These simulation results show a similar pattern compared to those of the force transmissibility. The displacement transmissibility for skyhook control on squeeze mode is presented in the logarithmic scale as shown in Figure 5-18. The whole 100 Hz is shown so that the higher frequencies can be observed better. It can be concluded that to achieve the lowest displacement transmissibility by skyhook control, the magnetic field for flow mode and squeeze mode should be set to 30kA/m and 5kA/m, respectively.
Figure 5-15: Effect of flow mode for displacement transmissibility.

Figure 5-16: Effect of squeeze mode for displacement transmissibility.
Figure 5-17: Skyhook control on flow mode for **displacement** transmissibility.

Figure 5-18: Skyhook control on squeeze mode for **displacement** transmissibility.
Figure 5-19: Skyhook control on combined mode for **displacement** transmissibility.

From the simulation results of Figure 5-14 and Figure 5-19, it is not difficult to discover that the conditions to achieve the lowest displacement transmissibility and force transmissibility by skyhook control are not the consistent with each other (flow mode max 50kA/m & squeeze mode max 15kA/m vs. flow mode max 30kA/m & squeeze mode max 5kA/m). So these two types of transmissibility cannot obtain the lowest values at the same time. *There exists a trade-off between these two types of transmissibility if the skyhook controller is adopted.* Further research therefore involves finding a controller than can minimize both types of transmissibility.
5.3 Simulation results of hierarchical control for MR mount in TDOFs

The Simulink® model of hierarchical control of the MR mount in two degree of freedom structure has been built as shown in Figure 5-20. There are four main components in the model: excitation block in red, suspension and chassis block in yellow, MR mount and engine in blue and controller in green.

![Figure 5-20: Model of hierarchical control of the MR mount in TDOF.](image)

Inside the excitation block, there are two kinds of excitation as illustrated in Figure 5-21: the displacement excitation from the road and the force excitation from the engine (or pump/motor). It depends on the simulation need to choose which excitation will be used or if both excitations are required.
Figure 5-21: Model of excitation block.

The suspension and chassis block is the subsystem of the following items as shown in Figure 5-22. The suspension is simplified as a damper and a spring and the chassis can be influenced by displacement excitation and the reaction force created by MR fluid mount. The chassis displacement, velocity and acceleration are the outputs of this block.
Figure 5-22: Model of suspension and chassis block.

The content of MR mount and engine block can be seen in Figure 5-23. The MR mount model is the same as the one in previous sections. The inputs of this block are the magnetic field for flow mode and squeeze mode, and the force excitation, while the outputs are the transmitted force, the relative velocity between the chassis and the engine, and the velocity of the engine.
Figure 5-23: Model of MR mount and engine block.

The last part is the controller block as illustrated in Figure 5-24. This block takes chassis acceleration, relative velocity between chassis and engine and engine velocity as inputs and produces the appropriate magnetic field for flow mode and squeeze mode. The controller is a hierarchical control with two layers: the basic layer control is to switch between the maximum magnetic field value and the minimum magnetic field value (decided by the skyhook algorithm); the second layer control is switch between the lowest force transmissibility and lowest displacement transmissibility.
Figure 5-24: Model of MR Controller block.

With the hierarchical control design, the simulation result can be observed in Figure 5-25. The lowest displacement transmissibility is pursued as the primary law. Whenever the acceleration of the chassis is over \(0.2 \, m/s^2\), the values to achieve the lowest force transmissibility are implemented. If the acceleration value falls under \(0.2 \, m/s^2\), the values to achieve the lowest displacement transmissibility are chosen to be used again. The switch acceleration value is chosen based on observed simulation result. The acceleration values are under \(0.3 \, m/s^2\), the \(0.2 \, m/s^2\) is chosen to protect the chassis.
from vibrating acceleration less than that. From this figure, the tradeoff between the lowest displacement transmissibility and force transmissibility is realized by switching to the proper values. The principle behind the tradeoff between the lowest displacement transmissibility and force transmissibility is the tradeoff between the transient state response and the steady state response of the chassis.

Figure 5-25: Tradeoff between two types of transmissibility.

For the two layers of hierarchical control, the skyhook control in first layer switch values of between maximum and minimum to achieve the lowest transmissibility (either displacement or force); the control in the second layer switch the sets of values between the lowest displacement transmissibility and lowest force transmissibility. Every switch in the system makes the running of the system unsmooth because this logic causes oscillation between two values. From the control perspective, it requires time to reach certain values and also may cause “overshoot”. This is unpleasant feather of switch logic. Fuzzy logic control, on the other hand, is known for provision of smooth control surface.
Therefore, a fuzzy controller has also been designed to imitate the effect of the hierarchical controller. The modeling of the whole structure is shown in Figure 5-26. The parts in the red ring are the fuzzy logic controller for flow mode and squeeze mode. The rest parts are identical to each part in Figure 5-20 except the controller part. In another word, the only part in Figure 5-26 which is different from Figure 5-20 is the controller. Here is the fuzzy logic controller.
Figure 5-26: Fuzzy logic controller for MR mount in TDOF.
The fuzzy inference system editor and rule surface for the flow mode are shown in Figure 5-27 and Figure 5-28.

Figure 5-27: Fuzzy logic controller FIS editor for flow mode.

Figure 5-28: Fuzzy logic controller rule surface for flow mode.
The fuzzy logic controller takes three input membership functions which are Relative Displacement, Relative Velocity, and Acceleration. The rule surface is not able to fully show the relationship between the flow field and these three inputs so the rules are also given:

1. If (Relative Displacement is N) and (Relative Velocity is N) and (Acceleration is N) then (Flow Field is B) (1)

2. If (Relative Displacement is N) and (Relative Velocity is N) and (Acceleration is Z) then (Flow Field is B) (1)

3. If (Relative Displacement is N) and (Relative Velocity is N) and (Acceleration is P) then (Flow Field is B) (1)

4. If (Relative Displacement is N) and (Relative Velocity is Z) and (Acceleration is N) then (Flow Field is B) (1)

5. If (Relative Displacement is N) and (Relative Velocity is Z) and (Acceleration is Z) then (Flow Field is M) (1)

6. If (Relative Displacement is N) and (Relative Velocity is Z) and (Acceleration is P) then (Flow Field is B) (1)

7. If (Relative Displacement is N) and (Relative Velocity is P) and (Acceleration is N) then (Flow Field is B) (1)

8. If (Relative Displacement is N) and (Relative Velocity is P) and (Acceleration is Z) then (Flow Field is M) (1)

9. If (Relative Displacement is N) and (Relative Velocity is P) and (Acceleration is P) then (Flow Field is B) (1)

10. If (Relative Displacement is Z) and (Relative Velocity is N) and (Acceleration is N) then (Flow Field is B) (1)

11. If (Relative Displacement is Z) and (Relative Velocity is N) and (Acceleration is Z) then (Flow Field is M) (1)

12. If (Relative Displacement is Z) and (Relative Velocity is N) and (Acceleration is P) then (Flow Field is B) (1)
13. If (RelativeDisplacement is Z) and (RelativeVelocity is Z) and (Acceleration is N) then (FlowField is S) (1)

14. If (RelativeDisplacement is Z) and (RelativeVelocity is Z) and (Acceleration is Z) then (FlowField is S) (1)

15. If (RelativeDisplacement is Z) and (RelativeVelocity is Z) and (Acceleration is P) then (FlowField is S) (1)

16. If (RelativeDisplacement is Z) and (RelativeVelocity is P) and (Acceleration is N) then (FlowField is S) (1)

17. If (RelativeDisplacement is Z) and (RelativeVelocity is P) and (Acceleration is Z) then (FlowField is S) (1)

18. If (RelativeDisplacement is Z) and (RelativeVelocity is P) and (Acceleration is P) then (FlowField is B) (1)

19. If (RelativeDisplacement is P) and (RelativeVelocity is N) and (Acceleration is N) then (FlowField is M) (1)

20. If (RelativeDisplacement is P) and (RelativeVelocity is N) and (Acceleration is Z) then (FlowField is M) (1)

21. If (RelativeDisplacement is P) and (RelativeVelocity is N) and (Acceleration is P) then (FlowField is B) (1)

22. If (RelativeDisplacement is P) and (RelativeVelocity is Z) and (Acceleration is N) then (FlowField is S) (1)

23. If (RelativeDisplacement is P) and (RelativeVelocity is Z) and (Acceleration is Z) then (FlowField is S) (1)

24. If (RelativeDisplacement is P) and (RelativeVelocity is Z) and (Acceleration is P) then (FlowField is B) (1)

25. If (RelativeDisplacement is P) and (RelativeVelocity is P) and (Acceleration is N) then (FlowField is M) (1)

26. If (RelativeDisplacement is P) and (RelativeVelocity is P) and (Acceleration is Z) then (FlowField is B) (1)

27. If (RelativeDisplacement is P) and (RelativeVelocity is P) and (Acceleration is P) then (FlowField is B) (1)
The fuzzy inference system editor and rule surface for the squeeze mode are shown in Figure 5-29 and Figure 5-30.

![Figure 5-29: Fuzzy logic controller rule surface for squeeze mode.](image)

![Figure 5-30: Fuzzy logic controller rule surface for squeeze mode.](image)
Similarly, the rule surface is also not able to fully show the relationship between
the squeeze field and these three inputs and the rules follow:

1. If (RelativeDisplacement is N) and (RelativeVelocity is N) and (Acceleration is N)
then (SqueezeField is B) (1)

2. If (RelativeDisplacement is N) and (RelativeVelocity is N) and (Acceleration is Z)
then (SqueezeField is B) (1)

3. If (RelativeDisplacement is N) and (RelativeVelocity is N) and (Acceleration is P)
then (SqueezeField is B) (1)

4. If (RelativeDisplacement is N) and (RelativeVelocity is Z) and (Acceleration is N)
then (SqueezeField is M) (1)

5. If (RelativeDisplacement is N) and (RelativeVelocity is Z) and (Acceleration is Z)
then (SqueezeField is S) (1)

6. If (RelativeDisplacement is N) and (RelativeVelocity is Z) and (Acceleration is P)
then (SqueezeField is S) (1)

7. If (RelativeDisplacement is N) and (RelativeVelocity is P) and (Acceleration is N)
then (SqueezeField is B) (1)

8. If (RelativeDisplacement is N) and (RelativeVelocity is P) and (Acceleration is Z)
then (SqueezeField is S) (1)

9. If (RelativeDisplacement is N) and (RelativeVelocity is P) and (Acceleration is P)
then (SqueezeField is B) (1)

10. If (RelativeDisplacement is Z) and (RelativeVelocity is N) and (Acceleration is N)
then (SqueezeField is B) (1)

11. If (RelativeDisplacement is Z) and (RelativeVelocity is N) and (Acceleration is Z)
then (SqueezeField is S) (1)

12. If (RelativeDisplacement is Z) and (RelativeVelocity is N) and (Acceleration is P)
then (SqueezeField is S) (1)

13. If (RelativeDisplacement is Z) and (RelativeVelocity is Z) and (Acceleration is N)
then (SqueezeField is S) (1)

14. If (RelativeDisplacement is Z) and (RelativeVelocity is Z) and (Acceleration is Z)
then (SqueezeField is S) (1)
15. If (RelativeDisplacement is Z) and (RelativeVelocity is Z) and (Acceleration is P) then (SqueezeField is S) (1)

16. If (RelativeDisplacement is Z) and (RelativeVelocity is P) and (Acceleration is N) then (SqueezeField is S) (1)

17. If (RelativeDisplacement is Z) and (RelativeVelocity is P) and (Acceleration is Z) then (SqueezeField is S) (1)

18. If (RelativeDisplacement is Z) and (RelativeVelocity is P) and (Acceleration is P) then (SqueezeField is B) (1)

19. If (RelativeDisplacement is P) and (RelativeVelocity is N) and (Acceleration is N) then (SqueezeField is M) (1)

20. If (RelativeDisplacement is P) and (RelativeVelocity is N) and (Acceleration is Z) then (SqueezeField is S) (1)

21. If (RelativeDisplacement is P) and (RelativeVelocity is N) and (Acceleration is P) then (SqueezeField is M) (1)

22. If (RelativeDisplacement is P) and (RelativeVelocity is Z) and (Acceleration is N) then (SqueezeField is S) (1)

23. If (RelativeDisplacement is P) and (RelativeVelocity is Z) and (Acceleration is Z) then (SqueezeField is S) (1)

24. If (RelativeDisplacement is P) and (RelativeVelocity is Z) and (Acceleration is P) then (SqueezeField is M) (1)

25. If (RelativeDisplacement is P) and (RelativeVelocity is P) and (Acceleration is N) then (SqueezeField is M) (1)

26. If (RelativeDisplacement is P) and (RelativeVelocity is P) and (Acceleration is Z) then (SqueezeField is B) (1)

27. If (RelativeDisplacement is P) and (RelativeVelocity is P) and (Acceleration is P) then (SqueezeField is B) (1)

Because different field limitations and effects of the fields on the flow mode and squeeze mode, the rules are set different accordingly. Take rule 11 (on page 97) and the one (on page 100) for example, the flow field is set to middle for the flow mode, but the squeeze mode is set to small for the squeeze mode.
The simulation results between these two controllers are shown in Figure 5-31. As it appears, the fuzzy logic controller can also achieve similar transmissibility for both displacement and force. Compared to the hierarchical controller, fuzzy logic controller shows better performance at higher frequencies. However, at lower frequencies, the hierarchical controller can provide slightly better vibration isolation. During the design of the fuzzy logic controller, neither the concept of displacement transmissibility or force transmissibility is considered, but they can achieve similar results of the hierarchical control. As seen in Figure 5-28 and 5-30, the control surfaces are smooth and continuous. However, the rule development requires professional knowledge of certain area.

Figure 5-31: Comparison between two controllers.
5.4 Experimental setup

Dynamic stiffness is the major indicator of the transmissibility. Therefore in the experiments this figure of merit has been used as a measurement for vibration transmissibility.

As the simulations for the control of MR fluid mount in the single degree of freedom and two degree of freedom have been conducted, one question is left: “can the desired dynamic stiffness (which corresponds to the lowest transmissibility) be achieved by the prototype mount and testing machine?” If the desired dynamic stiffness at certain frequency can be obtained by the control of the current, it is equivalent to achieve the desired lowest transmissibility.

The experimental setup is illustrated in Figure 5-32:

Figure 5-32: Experimental setup for the desired dynamic stiffness.
There are four main parts in this setup. The first part is ElectroForce® 3330 test instrument by BOSE® Corporation. The test instrument is able to generate displacement excitation and measure the transmitted force by a load cell as shown in Figure 5-32. By the PCI_82 Control Box as shown in Figure 5-33, the measurement of the excitation displacement and transmitted force can be extracted and processed.

![PCI-82 control box](image)

Figure 5-33: PCI-82 control box.

The displacement and force signals are taken into the dSPACE1104 box, which is an interface between the analog signals and digital signals as seen in Figure 5-34. There are four columns of the channels, two of them are analog to digital converter channels called ADCH 1-8, and the others are digital to analog converter channels called DACH 1-8. This experiment uses two analog to digital converter channels to input the displacement and force signals (ADCH 5 and ADCH 6) into MATLAB/Simulink® environment. Also, one digital to analog converter channel (DACH 1) is used to output the desired current the computer calculated to the power amplifier.
The calculated current is not large enough to be supplied by the dSPACE system alone; therefore an Agilent 6543A DC power supply is also used shown in Figure 5-35. The power supply is used in the current control mode such that it amplifies the current values calculated from the computer.

Figure 5-34: dSPACE 1104 box.

Figure 5-35: Agilent 6543A DC power supply.
The amplified current is output to the appropriate magnetic coil; i.e. either the flow mode coil or the squeeze mode coil. This completes the closed loop control. The ControlDesk by dSPACE and Simulink® by MathWorks® can easily work together by using the dSPACE’s Real-Time Interface (RTI). The RTI is the link between the real-time hardware by dSPACE and the Simulink® model from MathWorks®. It extends the C code generator Real-Time Workshop® in order that the Simulink® model can be implemented on dSPACE real-time hardware. The ControlDesk test and experiment software can provide virtual instruments, automation and parameter set handling. For a hardware-in-the-loop (HIL) test, dSPACE provides an ideal experiment environment.

The Simulink® model with Real-Time Interface for the control of current is shown in Figure 5-36. The control law adopted here is a proportional control. Based on the envelop values for a certain frequency, a linear increase of the dynamic stiffness is assumed due to the increase of the current. For instance, 20Hz, the dynamic stiffness without any field is about 255 N/mm as a low base line, while the dynamic stiffness with the current 0.4A for the flow mode is about 380 N/mm as a high base line. Therefore the difference of dynamic stiffness 380 N/mm-255/mm=125 N/mm is caused by 0.4 A. Then a unit increase of dynamic stiffness can be caused by 0.4/125 A. For the desired dynamic stiffness, the desired current can be calculated by the operation that the difference between its value and the low base line times the current causing the unit increase of dynamic stiffness. If 350N/m dynamic stiffness is desired to be achieved at 20Hz, the desired current to realize that will be (350-255)*0.4/125=0.304A. The controller is realized in a Lookup Table as shown in Figure 5-37.
Figure 5-36: Simulink model with RTI.

Figure 5-37: Lookup Table to control the current.
The ControlDesk by dSPACE interface is shown in Figure 5-38. Any parameter in the Simulink® model can be observed by the virtual instruments in the layout.

Figure 5-38: The interface from ControlDesk by dSPACE.

5.5 Experimental results

5.5.1 Dynamic stiffness – fixed point

The experiments have been conducted at the frequencies under 30 Hz. Flow mode is chosen to achieve the desired dynamic stiffness (DS) as shown in Table 5.1.

Table 5.1: Test data for dynamic stiffness

<table>
<thead>
<tr>
<th>Frequency</th>
<th>Desired DS</th>
<th>Experimental DS</th>
<th>Error</th>
</tr>
</thead>
<tbody>
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<td>325</td>
<td>324.5477</td>
<td>0.14%</td>
</tr>
<tr>
<td>2</td>
<td>325</td>
<td>320.4184</td>
<td>1.41%</td>
</tr>
<tr>
<td>3</td>
<td>325</td>
<td>316.7083</td>
<td>2.55%</td>
</tr>
<tr>
<td></td>
<td></td>
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<tr>
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<td>0.15%</td>
</tr>
<tr>
<td>12</td>
<td>350</td>
<td>356.4935</td>
<td>1.86%</td>
</tr>
<tr>
<td>13</td>
<td>350</td>
<td>341.4909</td>
<td>2.43%</td>
</tr>
<tr>
<td>14</td>
<td>350</td>
<td>337.1238</td>
<td>3.68%</td>
</tr>
<tr>
<td>15</td>
<td>350</td>
<td>342.9332</td>
<td>2.02%</td>
</tr>
<tr>
<td>16</td>
<td>350</td>
<td>328.8489</td>
<td>6.04%</td>
</tr>
<tr>
<td>17</td>
<td>350</td>
<td>347.66</td>
<td>0.67%</td>
</tr>
<tr>
<td>18</td>
<td>350</td>
<td>335.4969</td>
<td>4.14%</td>
</tr>
<tr>
<td>19</td>
<td>350</td>
<td>330.2319</td>
<td>5.65%</td>
</tr>
<tr>
<td>20</td>
<td>350</td>
<td>321.5749</td>
<td>8.12%</td>
</tr>
<tr>
<td>21</td>
<td>375</td>
<td>370.5858</td>
<td>1.18%</td>
</tr>
<tr>
<td>22</td>
<td>375</td>
<td>364.1501</td>
<td>2.89%</td>
</tr>
<tr>
<td>23</td>
<td>375</td>
<td>360.2053</td>
<td>3.95%</td>
</tr>
<tr>
<td>24</td>
<td>375</td>
<td>366.9657</td>
<td>2.14%</td>
</tr>
<tr>
<td>25</td>
<td>375</td>
<td>372.2782</td>
<td>0.73%</td>
</tr>
<tr>
<td>26</td>
<td>375</td>
<td>377.2625</td>
<td>0.60%</td>
</tr>
<tr>
<td>27</td>
<td>375</td>
<td>376.696</td>
<td>0.45%</td>
</tr>
<tr>
<td>28</td>
<td>375</td>
<td>393.6588</td>
<td>4.98%</td>
</tr>
<tr>
<td>29</td>
<td>375</td>
<td>384.9155</td>
<td>2.64%</td>
</tr>
<tr>
<td>30</td>
<td>375</td>
<td>382.6816</td>
<td>2.05%</td>
</tr>
</tbody>
</table>
Compared the experimental dynamic stiffness and desired dynamic stiffness, the error rate is calculated.

\[
Error = \left[ \frac{\text{Desired } DS - \text{Experimental } DS}{\text{Desired } DS} \right] \times 100\%
\]

The largest error that occurred in the tests was 8.12%, and the smallest error was 0.14%. The average error was 2.69%. The error is an indicator of good tracking ability for the desired dynamic stiffness. The close match between desired DS and achieved DS could be observed in Figure 5-39.

![Comparison between desired DS and achieved DS for fixed points.](image)

One of the difficulties of this experimental design is the comparison the data between time domain and frequency domain. It is noted that the desired dynamic stiffness is in frequency domain, the signals obtained from the PCI control box which are the displacement and load are in time domain. The ratio of the load divided by the
displacement is also in time domain which is the dynamic stiffness at the moment. The solution is shown in the Figure 5-36. The signals over a period of time are stored by the Delay Line block in Simulink®, the maximum and minimum values of the stored data are chosen to obtain the peak to peak value for displacement and load. Through Unbuffer block, the dynamic stiffness in frequency domain is acquired by the division of peak to peak displacement value from peak to peak load value. Then, this value can be compared with the desired value and the difference will be sent to the controller to calculate the appropriate amount of current.

As it shows, for each frequency, the desired dynamic stiffness is set; take 24 Hz for an example, the target is 375 N/mm, experimental result is about 367 N/mm. The dynamic stiffness at a certain frequency can be closely achieved. This makes possible the desired dynamic stiffness profile over the whole working frequency. As stated before, the corresponding transmissibility is also achievable which indirectly verified the effectiveness of the controller design in single degree of freedom model and in two degree of freedom model.

5.5.2 Dynamic stiffness – variable points

For sine wave is tested for 10Hz to verify that the variable dynamic stiffness can be achieved for certain frequency. The squeeze mode is chosen for this test. The sine wave is

\[ \text{sine wave } = 25 \times \sin(\pi t) + 315 \]
The desired dynamic stiffness profile fluctuates around 315 N/mm and the fluctuation range is 25 N/mm as shown in the layout of ControlDesk software, see the red line in Figure 5-40.

![Desired dynamic stiffness profile](image)

Figure 5-40: Desired dynamic stiffness profile.

The dynamic stiffness achieved is also shown in black. The range of the achieved dynamic stiffness is from about 275 to 315. The mean value is 295. The error can be roughly calculated by the following:

\[
\frac{(340 - 315)}{340} \times 100\% = 7.35\%
\]

\[
\frac{(315 - 295)}{315} \times 100\% = 6.35\%
\]

\[
\frac{(290 - 275)}{290} \times 100\% = 5.17\%
\]

The largest error is about 7.35%, the smallest error is about 5.17%, and the average error is about 6.35%. These results are acceptable for the experiment.

This part of experiment proves that the variable dynamic stiffness at a certain frequency can be approximately achieved. It can be inferred that the variable dynamic stiffness can be achieved over a range of frequency. Consider the two degree of freedom model; the lowest displacement transmissibility and lowest force transmissibility
correspond to different dynamic stiffness profiles. The experiment show that the
switching between two dynamic stiffness profiles can be closed obtained.

**Error Analysis**

Besides the effectiveness of the controller, these errors could be caused by many factors which mainly include the following:

- Instrumental limitation
- Numerical solver limitation
- Human error
- Mechanical delays
- Synchronization of different devices

In summary, the experiments complete the verification of the model and control of the mixed mode MR fluid mount. It can be concluded by simulations and by experiments that the proposed MR fluid mount is able to provide a certain range of dynamic stiffness over a large range of frequency and also the desired dynamic stiffness could be achieved in reality.
Chapter 6

Discussion

The last chapter of this dissertation summarizes the modeling, simulation and experimental accomplishments and contributions for the control system of the MR fluid mount. The chapter covers the whole process and is followed by the concluding remarks. Future work is also recommended based on this research.

6.1 Discussion

Several simulations and experiments have been conducted with a uni-axial MR fluid mount. Discussion in this section includes modeling and design aspects of the MR mount, the modeling of the MR fluid mount in two degree of freedom, the types of transmissibility, comparison of the two controllers, experimental results, and direction for future work.

Modeling and design of the MR mount:

- The MR fluid mount in [21] has the unique design of flow mode and squeeze mode which can operate individually and simultaneously. The flow mode and squeeze mode influence the damping of the MR fluid mount in different ways.
The flow mode increases the dynamic stiffness in low frequency range and decreases the dynamic stiffness in high frequency range when the magnetic field is increased to the blocking value. The squeeze mode on the other hand increases the dynamic stiffness over the whole range of frequency when the magnetic field is increased. The two working modes therefore operating together can provide a large range of dynamic stiffness over the whole working frequencies. The main limitations of the MR mount investigated in this research can be summarized as follows:

- In many applications, the vibration is not from one direction. The design of the uni-axial MR fluid mount provides a foundation for the modeling and design of the multi-axial MR fluid mount which can isolate vibration from different directions simultaneously.

- Also a wide-bandwidth MR fluid mount can be developed based on results of this MR fluid mount [21]. The wide-bandwidth means a wide range of the frequencies in the isolation region due to the multiple flow valves. For instance, two of three flow valves are MR valves. Without any MR valve activated, the MR mount with three flow paths is equivalent to a hydraulic mount with a large flow path. With only one MR flow path as shown in this dissertation, the mount provides a limited isolation range. With two MR flow paths, another dynamic stiffness profile is obtained. By controlling the activation of the MR valves at the crossover points in the three situations, the best isolation can be obtained. Compared to the single valve MR fluid mount, the wide-bandwidth MR fluid mount can provide low
dynamics stiffness even at higher frequency after 100Hz which the single-valve mount is not able to achieve.

**Modeling of the MR mount in two degree of freedom:**

Chassis are subject to both road profile and engine or pump/motor vibration when a vehicle is moving on the road. Vibration isolators are developed to reduce the effect of the road conditions on the chassis. The semi-active MR fluid dampers are alternative solutions of the conventional isolators designed to reduce the vibration transmitted from the road to the chassis. The MR fluid mounts should also mitigate vibration transmission from the engine or pump/motor (in hybrid vehicles) to the chassis.

In chapter 4, the modeling approach of the MR fluid mount in two degree of freedom is inspired by the concept of quarter car model. In most papers concerning the MR fluid dampers, the quarter car model is adopted which include the sprung mass and the unsprung mass. The MR fluid damper is placed between the sprung mass and the unsprung mass. The excitation is from the road profile. This structure is applied to the modeling in this dissertation as shown Figure 6-1 (a). First, the engine and MR fluid mount is added upon the original structure which uses the tire and chassis as the unsprung mass and sprung mass as shown in Figure 6-1 (b). Second, the parts under the chassis are simplified as a spring and a damper. The reason for this simplification is that the effect of the MR fluid mount is the purpose of modeling. The main objective is control design for the MR fluid mount as shown Figure 6-1 (c). To this end, the excitation source is changed from the road profile (displacement excitation) to the engine vibration (force excitation) as shown in Figure 4-2.
The development of the model is specially to address the targets of the research: assessing effectiveness and controllability of the MR fluid mount. The advantage of this development is that the modeling is based on the real vehicle scenario yet without taking all the complexity of it. The disadvantage is taking the suspension as a passive one and taking only the engine effect on the chassis. However, the modeling can be extended to both excitations from the engine and road situation. Furthermore, the model can include the MR damping element which makes the suspension semi-active. For this research, the development is suitable and necessary.

Figure 6-1: The development of the modeling of the MR mount in TDOF: (a) quarter car model; (b) engine and MR mount added; (c) simplification for MR mount.

**Two types of transmissibility:**
Some current research of the vibration isolator focuses on displacement transmissibility, some investigates force transmissibility. Seldom have the researches taken into account both types of transmissibility. In this research both transmissibilities have been considered with the MR mount:

- The displacement transmissibility simulation results displays the isolator effectiveness is reducing the chassis displacement caused by the vibration source of engine.

- The force transmissibility simulation results displays the isolator effectiveness is reducing chassis acceleration. The ride comfort is directly related to the acceleration.

- No prior research has taken into account both displacement transmissibility and force transmissibility. This dissertation is the first to investigate both types of transmissibility and realize the trade-off between them. Therefore, no comparison can be made with other papers.

**Comparison of the two controllers:**

Two controllers designed in this dissertation for the MR fluid mount in two degree of freedom model: hierarchical controller and fuzzy logic controller. Both of the controllers can significantly decrease the peak value of the transmissibility.

- From the perspective of system and control, the hierarchical controller which adopted switch logic in the design is not preferred due to the oscillation at the switching point.
The fuzzy logic controller can avoid such a problem by providing a smooth control surface. Also, it can provide slightly better isolation than hierarchical controller in higher frequencies. However, the design of the fuzzy logic controller needs a good foreknowledge of the effect on the outputs from the inputs. As matter of fact, the development of the fuzzy logic controller is based on the knowledge of the design of the hierarchical controller.

On the other hand, the hierarchical controller is easier to comprehend and implement in practice due to its simpler logic.

**Experiments on the Mount:**

- A series of experiments were conducted with the MR mount to evaluate effectiveness of the mount in delivering a construable stiffness profile. To this end closed-loop control experiments were performed. In these experiments each controller was evaluated separately in a hardware-in-the-loop arrangement. To ensure consistent result each time the mount was filled through a vacuum filling process. This process ensured that the fluid did not contain any gas. The experiments showed the following results:
  - It is possible to achieve a wide range of dynamic stiffness with the mount
  - It is possible to track a time-dependent desired dynamic stiffness. This feature is specially interesting as in many advanced propulsion and hybrid vehicles excess noise and vibration is related to certain events in the power cycle. Events such as switching between power modes or switching between functionality of the main equipments (e.g. switching between pump and motor functions in hydraulic
hybrid vehicles). The ability of the mount in assuming time dependent dynamic stiffness is essential for achieving desired levels of transmissibility over the entire working conditions of these vehicles.

**Direction for future work:**

The mount has been designed, fabricated and verified. The control of the mount has also been developed and the controllability of dynamic stiffness has been experimentally verified in the laboratory. In the future special efforts can be dedicated for evaluating and modifying the design of the controller for a single degree of freedom model and also the same for the control system for a two degree of freedom model. After experiments, it will be the time for testing the controllers with the mount in a real hybrid vehicle. The problem may be encountered is the space limitation. The design of the MR fluid mount should be considered when the design of the hybrid vehicles is in the process. More modifications can be made to the controllers according to the testing results. After that the task will be to implement the program into an electronic control unit (ECU). The ECU will be tested with the mount in the hybrid vehicle as the finalization.

### 6.2 Conclusions

The purpose of this research was to develop a control system for the mixed mode magnetorheological fluid mount in different scenarios. This research has accomplished the following objectives:

- The modeling of the MR fluid mount in a single degree of freedom structure was developed. The skyhook control has been developed for both flow mode and
squeeze mode of the mount in order to achieve possible lowest displacement transmissibility.

- The modeling of the MR fluid mount in a two degree of freedom structure was developed. The skyhook control has been developed for both flow mode and squeeze mode of the mount in order to achieve the lowest possible displacement transmissibility or force transmissibility.

- The hierarchical control of the MR fluid mount in a two degree of freedom structure was developed to realize the tradeoff between the lowest displacement transmissibility and the lowest force transmissibility. The modeling of the control is completed in Matlab/Simulink® and the simulation result shows effectiveness of the control method.

- A fuzzy logic controller was also developed to imitate the effect of the hierarchical control system and to provide a continuous control surface at the same time. The modeling of the control is completed in Matlab/Simulink® and the simulation result shows effectiveness of fuzzy logic control method. Furthermore, the simulation result is compared with the one by the hierarchical controller.

- Experiments were performed to verify that the desired dynamic stiffness is realizable in order to achieve the desired transmissibility. Hardware-in-the-loop tests are taken to prove the controllability of the MR fluid mount.
6.3 Future work

Even though the control completed the research of the mixed mode MR fluid mount in general, there are still some extensions of this research to be recommended based on previous discussion:

- **Further experiments** – A hardware-in-the-loop tests were conducted to verify that desired dynamic stiffness can be achieved. However, a more complex setup should be built for the single degree of freedom and two degree of freedom scenarios. More experiments can be further conducted to realize those scenarios. Moreover, the MR fluid mount can be tested in the vehicle on the road.

- **Control for wide-bandwidth MR mount** – Wide-bandwidth MR mount has been designed in [21] and the notch frequency is able to be placed in different locations. This is realized by a suitable controller to be designed to manage the number of the open orifices.

6.4 Publications

Several publications are the direct and indirect results from this research.

i. Book chapter


ii. Journal publications

iii. Conference publications


References

[1] Bellis, M.,: “Automobile history- The history of cars and engines”.
http://inventors.about.com/od/cstartinventions/a/Car_History.htm


http://images.google.com/igres?imgurl=http://www.aircraftspruce.com/catalog/graphics/10-01557.jpg&imgrefurl=http://www.aircraftspruce.com/catalog/eppages/10-01557.php&usg=__21UKBXWEBgDr7FY-dgK1P9zLWY=&h=218&w=400&sz=78&hl=en&start=3&um=1&tbni=0ASZVE7sm9d0M:&tbnh=68&tbnw=124&prev=/images%3Fq%3Drubber%2Bengine%2B mounts%26hl%3Den%26sa%3DN%26um%3D


Conference and Automotive & Transportation Technology Conference, July 9-11, 2002.


Appendix

A. Analytical MATLAB® program

% This program is to simulate the MR mount with Displacement excitation
% from Top to calculate the Force vs. Motion and dynamic stiffness of the
% mount, or from Bottom to calculate the TRANSMISSIBILITY

clear all
clc

% -------------------------------------------

% HYDRAULIC PROPERTIES
% -------------------------------------------
rho = 2.66e3; %density kg/m^3
KL = .5; %loss coefficient
viscosity = 1*0.06;%MRF126CD, %0.092 MRF132AD;

% -------------------------------------------

% GEOMETRY OF FLOW CHANNEL
% -------------------------------------------
M=100;
Mx = 400; %70; %kg %NO effect
Mz = 1000;
h = 1*3e-3;
b = 4*45e-3;
L = 16e-3;
Di = 2*b*h/(b+h); %diameter of orifice
A = h*b;
Ii = 1*3.5*rho*L/A; \%kg/m^4 \%increase shifts left and up
Ri = 1*1.5*128*viscosity*L/pi/Di^4;

h1 = h; h2 = h; h3 = h;
b1 = b; b2 = b; b3 = b;
A1 = b1*h1; A2 = b2*h2; A3 = b3*h3;
L1 = L; L2 = L; L3 = L;

%GEOMETRY OF SQUEEZE PLATE

h0 = 3e-3; \%m
R = .025; \%0.05 \%m

%ELASTOMER

Kr = .250e6; \%0.225e6; \%N/m
Br = 10; \%300; \%N.s/m
C1 = 9.5*2.5e-12; \%3e-11; \%m^5/N \%increase shifts left and down, lowers the slope 
% increase lowers the tail
C2 = 1*2.4e-9; \%2.6e-9; \%m^5/N \%increase shifts left and up, up saturated at 1e-5
Ap = .7*25e-4; \%m^2 \%increase -> increase the peak height and tail height
% decrease -> increase the notch frequency, peak frequency unchanged

Kr1=100000;
Br1=1000;

%EXCITATION AMPLITUDE

Exc = [.1 .2 .4 .5 .6 .8 1.0]*10^-3;
% X = Exc(7); \%m Excitation from TOP
\begin{verbatim}
Y = Exc(7); %m Excitation from BOTTOM
% Y = 0;
% F = 100; %N Force excitation from TOP
G = 1;
% F = 0;
%
% SIMULATION CONFIGURATIONS
%
T = 100; %s
dt1 = .0001;
% dt2 = .001; %s
w1 = 0; %Hz
w2 = 100; %Hz
%
% Original fields
% H_F1 = 00; % kA/m
% H_S = 00; % kA/m
%
% Current & Field relationship
% C_F = 0; % A
% C_S = 0; % A
%
% H=0;
% Bang-bang control & skyhook

H_Fmin = 0; % kA/m
% H_Fmax = 0; % kA/m
H_Smin = 0; % kA/m
% H_Smax = 0; % kA/m
%
% H_Fmin_D = 0; % kA/m
\end{verbatim}
% H_Fmax_D = 30; % kA/m
% H_Smin_D = 0; % kA/m
% H_Smax_D = 5; % kA/m
% H_Fmin_F = 0; % kA/m
% H_Fmax_F = 50; % kA/m
% H_Smin_F = 0; % kA/m
% H_Smax_F = 15; % kA/m
% Thh = 0.2;
%
% Fuzzy logic control
%
% asymmetric
% fismatflow=readfis('MagneticField_Flow');
% fismatsqueeze=readfis('MagneticField_Squeeze');

% fismatflow=readfis('MagneticField_Flow_1');
% fismatsqueeze=readfis('MagneticField_Squeeze_1');

fismatflow=readfis('MagneticField_Flow_2');
fismatsqueeze=readfis('MagneticField_Squeeze_2');

% ------------------------------------------------------------------------
% SIMULATION OF DISPLACEMENT EXCITATION
% ------------------------------------------------------------------------

% From TOP for TRANSMITTED FORCE and DYNAMIC STIFFNESS
% open('SWmr_mount_DISP_exc_TOP_08_27_09_MIX_1_valve')
% sim('SWmr_mount_DISP_exc_TOP_08_27_09_MIX_1_valve')
% sim('SWmr_mount_DISP_exc_TOP_08_27_09_MIX_2_valves')
% sim('SWmr_mount_DISP_exc_TOP_08_27_09_MIX_3_valves')
% sim('SWmr_mount_DISP_exc_TOP_10_21_09_MIX_1_valve_Current')
% sim('SWmr_mount_DISP_exc_TOP_10_22_09_MIX_1_valve_bangbang')
% sim('SWmr_mount_DISP_exc_TOP_10_22_09_MIX_1_valve')
% sim('SWmr_mount_DISP_exc_TOP_11_02_09_MIX_1_valve_bangbang')
% sim('SWmr_mount_DISP_exc_BOTT_11_05_09_Skyhook')
% sim('SWmr_mount_DISP_exc_BOTT_12_04_09')
% sim('SWmr_mount_DISP_exc_BOTT_12_21_09')
% sim('mr_mount_rectangular_orifice_DISP_exc_BOT_01_30_08')
% sim('SWmr_mount_DISP_exc_BOTT_1_12_10')
% sim('SWmr_mount_DISP_exc_BOTT_1_28_10_Groundhook')
% sim('SWmr_mount_DISP_exc_TOP_2_4_10_Groundhook')
% sim('SWmr_mount_Quarter_car_DISP_exc_BOTT_02_23_10')
% sim('SWmr_mount_Quarter_car_DISP_exc_BOTT_03_05_10_Skyhook')
% sim('SWmr_mount_Quarter_car_03_09_10')
% sim('SWmr_mount_Quarter_car_03_25_10_fft')
% sim('SWmr_mount_Quarter_car_03_25_10_Cap')
% sim('SWmr_mount_Quarter_car_03_29_10_Skyhook')
% sim('SWmr_mount_Quarter_car_04_13_10_within_calculation')
% sim('SWmr_mount_Quarter_car_04_18_10_within_calculation_direct')
% sim('SWmr_mount_Quarter_car_04_30_10_Skyhook')
% sim('SWmr_mount_Quarter_car_05_03_10_Skyhook')
% sim('SWmr_mount_Quarter_car_05_05_10_Groundhook')
% sim('SWmr_mount_Quarter_car_05_07_10_Skyhook')
% sim('SWmr_mount_Quarter_car_05_10_10_Switch_acceleration')
% sim('SWmr_mount_Quarter_car_05_11_10_Switch_acceleration_Cap')
% sim('SWmr_mount_DISP_exc_BOTTOM_05_13_10_Fuzzy')
sim('SWmr_mount_DISP_exc_BOTTOM_05_18_10_Fuzzy_Skyhook')
% sim('SWmr_mount_DISP_FORCE_exc_05_26_10_Fuzzy')
% sim('SWmr_mount_DISP_FORCE_exc_05_27_10_Fuzzy_Skyhook')
% sim('SWmr_mount_Quarter_car_05_28_10_Fuzzy')
% sim('SWmr_mount_Quarter_car_06_02_10_Fuzzy_Skyhook')
% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% FREQUENCY DOMAIN
Fs1 = 1/dt1;
blocksize1 = length(tout);
f1 = (0:blocksize1-1)*Fs1/blocksize1;
f1 = f1';

Fout_fft = fft(Ft);
index = find(Fout_fft == 0);
Fout_fft(index) = 1e-17;

Y_fft = fft(y);
index = find(Y_fft == 0);
Y_fft(index) = 1e-17;

% DSTime=abs(Fout./x)/1e3;
% plot(f1,DSTime)
% xlim([0 100])
% xlabel('Frequency (Hz)','Fontsize',14)
% ylabel('Dynamic stiffness (N/mm)','Fontsize',14)
% hold on

% Z_fft = fft(z);
% index = find(Z_fft == 0);
% Z_fft(index) = 1e-17;

% X_fft = fft(x);
% index = find(X_fft == 0);
% X_fft(index) = 1e-17;

% TR=abs(Z_fft./Y_fft);
% plot(f1,TR,'r')
% xlim([0 100])
% xlabel('Frequency (Hz)','Fontsize',14)
% ylabel('Transmissibility (Z/Y)','Fontsize',14)
% hold on

% TR=abs(X_fft./Y_fft);
% plot(f1,TR,'g')
% xlim([0.1 20])
% xlabel('Frequency (Hz)','Fontsize',14)
% ylabel('Transmissibility (X/Y)','Fontsize',14)
% hold on

% TR=abs(Z_fft./X_fft);
% plot(f1,TR)
% xlim([0 20])
% xlabel('Frequency (Hz)','Fontsize',14)
% ylabel('Transmissibility (Z/X)','Fontsize',14)
% hold on

% ForceInput = fft(Fin);
% index = find(ForceInput == 0);
% ForceInput(index) = 1e-17;
% ForceOutput = fft(Fout);
% index = find(ForceOutput == 0);
% ForceOutput(index) = 1e-17;

% ForceTR=abs(ForceOutput./ForceInput);
% plot(f1,ForceTR)
% xlim([0 20])
% xlabel('Frequency (Hz)','FontSize',14)
% ylabel('Transmissibility ','FontSize',14)
% hold on

% DS=abs(ForceOutput./Y_fft)/1e3;
DS=abs(Fout_fft./Y_fft)/1e3;
plot(f1,DS,'r')
xlim([0 100])
xlabel('Frequency (Hz)','Fontsize',14)
ylabel('Dynamic stiffness (N/mm)','Fontsize',14)
hold on

% ForceTr_fft = fft(ForceTr);
% index = find(ForceTr_fft == 0);
% ForceTr_fft(index) = 1e-17;
% plot(f1,ForceTr_fft)
% xlim([0 100])
% hold on

% TR_F = abs(Fout./Fin);
% plot(tout,TR_F)
% xlim([0 100])
% hold on

% DS=abs(Fout_fft./Y_fft)/1e3;
% plot(f1,DS,'b')
% xlim([0 20])
% xlabel('Frequency (Hz)','Fontsize',14)
ylabel('Dynamic stiffness (N/mm)','Fontsize',14)
% hold on
B. *The Simulink® Model of Hierarchical Control in Full Scale*

Figure A-1: Hierarchical controller in full scale.