

**Design and Adaptive Control of Magnetorheological Damper for Washing
Machine**

by

Abasin Ulasyar

**A Dissertation Submitted to the
Graduate School of Sciences and Engineering
in Partial Fulfillment of the Requirements for
the Degree of**

Doctor of Philosophy

in

Electrical & Electronics Engineering



August 2018

**Design and Adaptive Control of Magnetorheological Damper for Washing
Machine**

Koç University

Graduate School of Sciences and Engineering

This is to certify that I have examined this copy of a doctoral thesis by

Abasin Ulasyar

and have found that it is complete and satisfactory in all respects,

and that any and all revisions required by the final

examining committee have been made.

Committee Members:

Prof. Dr. Hakan Ürey, (Advisor)

Prof. Dr. İsmail Lazoğlu, (Co-Advisor)

Prof. Dr. Sedat Nizamoğlu

Prof. Dr. Funda Acar Yağci

Prof. Dr. Ali Fuat Ergenç

Prof. Dr. Seyed Ehsan Layegh Khavidaki

Date:



DEDICATIONS



To my Dearest Family

ABSTRACT

This thesis primarily focuses on the design, analysis, active and adaptive control of a new magneto rheological damper for the vibration attenuation of washing machine. The MR damper provides variable and controlled damper for all the frequency ranges of washing machine. The damping force of MR damper is selected according to the desired value of force required inside washing machine. The design of MR damper is made by using finite element analysis and reluctance method. This research has worked on three different prototypes of MR damper. Furthermore, a special experimental test-bed is established for testing all the prototypes of MR damper. The MR damper is tested with commercially available MR fluid. The performance of MR damper is also evaluated using different samples of MR fluid synthesized by the Koç University chemistry lab.

This thesis introduces a novel design of adaptive controller for the MR damper inside the washing machine, which is not available in the literature and in any commercial washing machine yet. A sliding mode control theory is used to develop a controller for the MR damper. In the sliding mode control, a sliding surface is defined in terms of the state error and convergence parameter. A nonlinear terminal sliding surface is proposed for the whole system in order to counter any nonlinearities, disturbances and unmodeled dynamics. All the three prototypes of MR dampers are tested in the washing machine with significant reduction of vibrations. Initially, the first prototype of MR damper is tested and observed that the MR damper attenuate 75 % vibrations in terms of displacement as compared to the passive dampers in washing machine. However after using second prototype of MR dampers and the designed adaptive control, it is seen that the MR dampers have reduced 85 % vibrations of washing machine in terms of displacement with fast response time. Besides this, the temperature of MR dampers are also measured successfully at inside and outside surfaces of MR damper for its full cycle.

ÖZET

Bu tez, amařır makinesinin titreřiminin zayıflatılması iin yeni bir manyeto reolojik sönümleyicinin tasarımı, analizi, aktif ve uyarlamalı kontrolüne odaklanmaktadır. MR damper, amařır makinesinin tüm frekans aralıkları iin deęişken ve kontrollü bir sönümleme sağlar. MR damperin sönümleme kuvveti, amařır makinesinde istenen kuvvet deęerine göre seçilir. MR damperin tasarımı sonlu elemanlar analizi ve isteksizlik metodu kullanılarak yapılmıştır. Bu arařtırmada, üç farklı MR damper prototipi üzerinde alışılmıştır. MR damperin tüm prototiplerini test etmek iin özel bir deneysel test yataęı kurulmuştur. MR damper, öncelikle, ticari olarak satılan MR sıvısı ile test edilmiştir. MR damperin performansı, ayrıca da Ko Üniversitesi kimya laboratuvarının sentezledięi farklı MR sıvısı örnekleri kullanılarak da deęerlendirilmiştir.

Bu tezde, amařır makinesinde bulunan MR damper iin, literatürde ve henüz herhangi bir ticari amařır makinesinde bulunmayan yeni bir adaptif denetleyici tasarımı tanıtılmıştır. MR dampere kontrolör geliřtirmek iin kayar mod kontrol teorisi kullanılmıştır. Kayma modu kontrolünde, kayan bir yüzey, durum hatası ve yakınsama parametresi olarak tanımlanır. Doğrusal olmayan bir terminal kayma yüzeyi, doğrusalsızlık durumunu, düzensizlięi ve modellenmemiş dinamikleri dengelemek iin önerilmiştir. MR damperlerin üç prototipi de amařır makinesinde test edilmiş ve titreřimler önemli ölçüde azaltılmıştır. MR damperin ilk prototipi amařır makinesindeki pasif amortisörler ile karşılaştırıldığında, yer deęiřtirme aısından %75 titreřimin azaldıęı gözlenmiştir. MR damperin ikinci prototipi, tasarlanan uyarlanabilir kontrolör ile kullanıldığında MR damperin daha hızlı tepki süresi sayesinde amařır makinesindeki titreřiminin %85 oranında azaldıęı görölmüştür. Bunun yanında MR damperlerin sıcaklıęı da tam döngü iin damperin i ve dıř yüzeylerinde başarıyla ölçölmüştür.

ACKNOWLEDGEMENTS

First, I would thank my Great Almighty ALLAH for giving me great strength, courage and success to bear the hardships and difficulties during four years of my PhD studies. I am also grateful to Prof. Dr. Ismail Lazoglu for giving me a platform like MARC, which has really developed, transformed and nurtured my mind for the research. The achievements which I got during the past four years were impossible without the facilities provided by him. I am truly inspired by his research methodology, entrepreneurship and discipline. He supported me in every difficult time of my challenging research.

I am also very much thankful to Prof. Dr. Hakan Ürey, Prof. Dr. Sedat Nizamoğlu and Prof. Dr. Funda Acar Yagci for their great support, patience, effort and time during my research and write-up of my thesis. In addition to this, I would thank my research group partners Anas Gillani and Mostafa Saleh for making this project a success by working equally hard throughout 4 years. Furthermore, I am also very much thankful to KOC University Istanbul, Turkey, The Scientific and Technological Research Council of Turkey (TÜBİTAK) and Arçelik A.Ş for funding the research project and my PhD studies.

I would give my sincere thanks to my family especially my parents, wife, brother and sisters for supporting and giving me courage during my PhD studies.

I am thankful to my friend Haris Sheh Zad for his guidance, patience and support throughout all trouble filled four years. His love, entertainment and dynamic personality really helped me in my difficult times of PhD studies.

I am also very thankful to my MARC members and friends especially Talha, Akmal, Sarmad, Adnan, Armine, Caglar, Erdem, Ugur, Abbas, Shahid, Taha, Anjum, Sheharyar, Omer, Atacan, Hosein, Enes, Başar, Isa and Muzaffer Abi.

In the end, I am thankful to my ALLAH for giving me health and great life. I wish a successful and wonderful life for all of you.

Abasin Ulasyar!

TABLE OF CONTENTS

LIST OF TABLES.....	vi
LIST OF FIGURES	vii
Chapter 1 INTRODUCTION	15
1.1 Magneto Rheological (MR) Fluid Devices.....	15
1.2 Design and Force Modeling of Shear Mode MR Damper	19
1.3 Control of Shear Mode MR Dampers	25
1.4 Application of MR Dampers in Washing Machine	26
1.5 Aims and Objectives of Research	27
Chapter 2 LITERATURE REVIEW	30
2.1 Overview	30
2.2 Operational Modes and Types of MR Dampers	30
2.3 Applications of MR devices.....	40
2.4 Control of MR Devices	42
2.5 Reduction of Vibrations Using MR Dampers.....	45
Chapter 3 DESIGN AND ANALYSIS OF FIRST PROTOTYPE OF MR DAMPER.....	47
3.1 Introduction.....	47
3.2 Design and Analysis	47
3.3 Summary	61
Chapter 4 EXPERIMENTAL TESTRIGS AND RESULTS OF FIRST PROTOTYPE OF MR DAMPER FOR WASHING MACHINE.....	63

4.1 Introduction.....	63
4.2 Experimental setups of MR Damper for Washing Machine.....	63
4.3 Results and Discussion	65
4.4 Summary	73

Chapter 5 DESIGN AND CONTROL OF SECOND PROTOTYPE OF MR DAMPER FOR WASHING MACHINE..... 75

5.1 Introduction.....	75
5.2 Design	75
5.3 Adaptive Control of MR Damper	81
5.4 Control Simulation Results	86
5.5 Analytical Model and Simulation Results of Washing Machine.....	92
5.6 Summary	97

Chapter 6 EXPERIMENTAL SETUPS AND RESULTS DISCUSSION OF SECOND PROTOTYPE OF MR DAMPER FOR WASHING MACHINE..... 98

6.1 Introduction.....	98
6.2 Test Rig and Results of Second Prototype of MR Damper	98
6.3 Test Rig and Results of Vibration Control of Washing Machine	110
6.4 Summary	124

Chapter 7 DESIGN, TESTING AND VIBRATION CONTROL OF THIRD PROTOTYPE OF MR DAMPER FOR WASHING MACHINE..... 125

7.1 Introduction.....	125
7.2 Design of 3 rd Prototype of MR Damper.....	125
7.3 Vibration Control of Washing Machine	132
7.4 Summary	135

Chapter 8 CONCLUSION	137
BIBLIOGRAPHY	141



LIST OF TABLES

Table 3.1: Specifications of commercial MR fluid [139].....	49
Table 3.2: Optimization results of MR damper using FEM and GSA.....	56
Table 4.1: MR damper comparison in terms of performance with other designs of MR damper in the literature.	73
Table 5.1: Optimized parameters of Double Coil MR damper.....	79
Table 6.1: Damping force results using different samples of MR fluid	106
Table 7.1: Final geometric variables of 3 rd prototype of MR damper	129

LIST OF FIGURES

Figure 1.1: Operation of MR fluid in shear mode [3].	15
Figure 1.2: Operation of MR fluid in flow mode [3].	16
Figure 1.3: Operation of MR fluid in squeeze mode [3].	16
Figure 1.4: Operation of MR fluid in pinch mode [4].	17
Figure 1.5: MR damper with its flow of magnetic flux in magnetic circuit [5].	18
Figure 1.6: Position feedback based Magnetorheological brake [7].	19
Figure 1.7: 2D axis symmetric model of single coil MR damper [8].	20
Figure 1.8: Meshing of the magnetic circuit of MR damper [9].	20
Figure 1.9: Lines of magnetic flux flowing between the poles of the electromagnet of the MR damper [9].	21
Figure 1.10: (a) Force vs velocity hysteresis curve using bouc-wen model (b) Force vs velocity hysteresis curve using bouc-wen model [10].	22
Figure 1.11: (a) Force vs velocity hysteresis curve using hysteretic model (b) Force vs velocity hysteresis curve using hysteretic model [10].	22
Figure 1.12: Behavior of yield stress as magnetic field intensity varies [11].	23
Figure 1.13: BH curve of the silicon steel showing the original and approximated trends [11].	24
Figure 1.14: BH curve of the MR fluid showing the original and approximated trends [11].	24
Figure 1.15: Control of a plant in the presence of disturbances by using MR dampers [12].	25
Figure 1.16: Transmitted force curve of washing machine versus RPM [13].	26
Figure 2.1: A control valve of MR damper showing shear mode operation [14].	32
Figure 2.2: A control valve of MR damper showing flow mode operation [14].	32

Figure 2.3: A control valve of MR damper showing squeeze mode operation [15].	33
Figure 2.4: A control valve of MR damper showing mix mode operation [16].	33
Figure 2.5: A design of MR damper having mono tube assembly [31].	34
Figure 2.6: A design of MR damper having twin tube assembly [32].	34
Figure 2.7: A design of MR damper having double assembly [34].	35
Figure 2.8: A design of MR damper having hybrid hydraulic arrangement [30].	35
Figure 2.9: The MR damper showing the internal coil configuration [35].	36
Figure 2.10: The MR damper showing the external coil configuration [36].	36
Figure 2.11: The MR damper showing the single coil configuration [37].	37
Figure 2.12: The MR damper showing the double coil configuration [38].	38
Figure 2.13: The MR damper showing the multi turn coil configuration [39].	38
Figure 2.14: The MR damper showing the inner bypass configuration [40].	39
Figure 2.15: The MR damper showing the external bypass configuration [16].	39
Figure 2.16: Seat suspension system using MR dampers [68].	41
Figure 2.17: MR dampers are used for the bed seat of the patient in the ambulance [70].	41
Figure 3.1: 2D CAD model of the front loaded washing machine.	48
Figure 3.2: 3D CAD model of the MR shear mode sponge damper.	48
Figure 3.3: 2D cross- sectional view of the MR shear mode sponge damper showing its geometric parameters.	49
Figure 3.4: 2D axis symmetric model of MR damper showing the magnetic flux lines and distribution.	51
Figure 3.5: Behavior of magnetic flux density at a pole for the changing length of coil.	52
Figure 3.6: Behavior of magnetic flux density at a pole for the changing thickness of cylinder.	52
Figure 3.7: Behavior of magnetic flux density at a pole for the number of turns of a coil.	53
Figure 3.8: Behavior of magnetic flux density at a pole for changing the radius of core.	53

Figure 3.9: Behavior of magnetic flux density at a pole for changing thickness of sponge.	54
Figure 3.10: Effect of weight percent of particles on the yield stress of MR fluid.	54
Figure 3.11: Golden section technique results for the optimization of geometric variables.	57
Figure 3.12: Two dimensional cross-sectional view of MR damper magnetic circuit with geometric variables.	58
Figure 3.13: Magnetic circuit of MR damper showing all the ten reluctances with magnetic flux flowing throughout the circuit.	59
Figure 3.14: Analysis of hysteresis behavior using hysteretic model of MR damper.	61
Figure 3.15: Complete flowchart of shear mode MR damper and its implementation in washing machine.	62
Figure 4.1: Testing setup of 1 st prototype of MR damper in washing machine.	64
Figure 4.2: Optimization strategy for identifying the hysteresis behavior of MR damper. .	65
Figure 4.3: Hysteresis curves for force with respect to velocity for various current values.	66
Figure 4.4: Hysteresis curves for force with respect to displacement for various current values.	66
Figure 4.5: Behavior of MR damping force with respect to input current.	67
Figure 4.6: Behavior of dissipation of energy versus current for various values of amplitude.	68
Figure 4.7: Behavior of dissipation of energy versus amplitude for various values of current.	68
Figure 4.8: Behavior of equivalent damping coefficient versus current for various values of amplitude.	69
Figure 4.9: Behavior of equivalent damping coefficient versus amplitude for various values of current.	69
Figure 4.10: Complete mechanism of control of vibrations using MR dampers.	70

Figure 4.11: Vibration comparison of MR dampers with passive dampers.	73
Figure 5.1: Two dimensional axis symmetric model of magnetic circuit of MR damper. ..	76
Figure 5.2: Three dimensional equivalent FEM model of magnetic circuit of MR damper.	76
Figure 5.3: Trend of magnetic flux density with respect to active pole for changing values of number of turns.	77
Figure 5.4: Trend of magnetic flux density with respect to active pole for changing values of core radius.	78
Figure 5.5: Trend of magnetic flux density with respect to active pole for changing values of cylinder thickness.....	79
Figure 5.6: Three dimensional CAD model of 2 nd prototype of MR damper.....	80
Figure 5.7: A two dimensional cross section view of 2 nd prototype of MR damper showing its geometric variables.	81
Figure 5.8: A complete block diagram for the adaptive control of current through MR dampers.	82
Figure 5.9: Electric current response of MR damper for various values of α	86
Figure 5.10: Voltage response of MR damper for various values of α	87
Figure 5.11: Electric current response of MR damper tracking its desired value equal to 0.495 A at $\alpha=0.846$	87
Figure 5.12: Electric current response of MR damper tracking its desired value equal to 0.165 A at $\alpha=0.846$	88
Figure 5.13: Trajectories of error current and its derivative of MR damper for various values of α	88
Figure 5.14: Electric current response of MR damper with the $dist = 50$ mA at $t = 0.3$ s. ..	89
Figure 5.15: Electric current response of MR damper with the $dist$ equal to $0.02\sin(1273.24\pi t)$	89

Figure 5.16: Electric current response of MR damper due to its variations of the parameters.	90
Figure 5.17: Trajectory between error current and its derivative of MR damper for new adaptive value of $\alpha = 165.63$.	90
Figure 5.18: Behavior of frequency of switching with respect to desired current.	91
Figure 5.19: Behavior of frequency of switching with respect to voltage source input.	91
Figure 5.20: Washing machine CAD model schematic.	92
Figure 5.21: Amplitude of vibrations in terms of displacement by using low damping and MR damping.	95
Figure 5.22: Behavior of transmissibility of washing machine by using low damping, high damping and MR damping.	96
Figure 6.1: Damping force of 2 nd Prototype of MR damper using 1 st sample at 0.3 A.	99
Figure 6.2: Damping force of 2 nd Prototype of MR damper using 1 st sample at 0.4 A.	100
Figure 6.3: Damping force of 2 nd Prototype of MR damper using 1 st sample at 0.5 A.	100
Figure 6.4: Damping force of 2 nd Prototype of MR damper using 1 st sample at 0.62 A.	101
Figure 6.5: Damping force of 2 nd Prototype of MR damper using 2 nd sample at 0.3 A.	101
Figure 6.6: Damping force of 2 nd Prototype of MR damper using 2 nd sample at 0.4 A.	102
Figure 6.7: Damping force of 2 nd Prototype of MR damper using 2 nd sample at 0.5 A.	102
Figure 6.8: Damping force of 2 nd Prototype of MR damper using 2 nd sample at 0.62 A.	103
Figure 6.9: Damping force of 2 nd Prototype of MR damper using commercial MR fluid at 0.3 A.	103
Figure 6.10: Damping force of 2 nd Prototype of MR damper using commercial MR fluid at 0.4 A.	104
Figure 6.11: Damping force of 2 nd Prototype of MR damper using commercial MR fluid at 0.5 A.	104

Figure 6.12: Damping force of 2 nd Prototype of MR damper using commercial MR fluid at 0.62 A.....	105
Figure 6.13: Damping force of 2 nd Prototype of MR damper using different samples.	105
Figure 6.14: The magnitudes and phases plots of 2 nd prototype of MR damper using impact hammer test.....	108
Figure 6.15: The FRF and power spectrum plots of 2 nd prototype of MR damper using impact hammer test.....	108
Figure 6.16: Hysteresis graph of force vs velocity for changing value of input current. ..	109
Figure 6.17: Hysteresis graph of force vs displacement for changing value of input current.	110
Figure 6.18: Experimental testing setup of adaptive vibration control of washing machine using 2 nd prototype of MR damper.	111
Figure 6.19: Installation of two force sensors in washing machine for force measurement.	112
Figure 6.20: Vibration adaptive control technique developed for the washing machine using 2 nd prototype of MR dampers.	113
Figure 6.21: Electric current tracks $I_{fd} = 0.495$ A at $\beta = 0.846$	114
Figure 6.22: Commanded input control signal with duty cycle of 75 %.	114
Figure 6.23: Electric current tracks $I_{fd} = 0.329$ A at $\beta = 0.846$	115
Figure 6.24: Commanded input control signal with duty cycle of 50 %.	115
Figure 6.25: Measurement of unbalanced force by load cell 1.....	116
Figure 6.26: Measurement of unbalanced force by load cell 2.....	116
Figure 6.27: Displacement of vibrations along z axis.....	117
Figure 6.28: Displacement of vibrations along y axis.	117
Figure 6.29: Acceleration of vibrations along x axis.....	118
Figure 6.30: Acceleration of vibrations along y axis.....	118

Figure 6.31: Acceleration of vibrations along z axis.	119
Figure 6.32: Washing machine angular velocity for its complete cycle.	120
Figure 6.33: Load cells are utilized at two sides of 2 nd prototype of MR damper.	120
Figure 6.34: Load cells are used at two sides of 2 nd prototype of MR damper for transmissibility measurement.	121
Figure 6.35: Load cell 1 measurement of force in the washing machine.	121
Figure 6.36: Load cell 2 measurement of force in the washing machine.	122
Figure 6.37: Transmissibility measurement by the load cells in the washing machine for the complete cycle.	122
Figure 6.38: Behavior of temperature at the outside surface of MR damper's cylinder. ..	123
Figure 6.39: Behavior of temperature at the inside surface of MR damper's piston.	123
Figure 7.1: Magnetic flux density distribution in two dimensional axis symmetric model of 3 rd prototype of MR damper.	126
Figure 7.2: Magnetic flux density distribution in three dimensional axis equivalent model of 3 rd prototype of MR damper.	126
Figure 7.3: Magnetic flux density behavior versus length of active pole for changing values of number of turns.	127
Figure 7.4: Magnetic flux density behavior versus length of active pole for changing values of radius of core.	127
Figure 7.5: Magnetic flux density behavior versus length of active pole for changing values of cylinder thickness.	128
Figure 7.6: Three dimensional CAD model of 3 rd prototype of MR damper.	130
Figure 7.7: Measurement of force at 0.3 A for the 3 rd prototype of MR damper.	131
Figure 7.8: Measurement of force at 0.5 A for the 3 rd prototype of MR damper.	131
Figure 7.9: Experimental behavior of force with respect to electric current for the 3 rd prototype of MR damper.	132

Figure 7.10: Two new 3 rd prototypes of MR dampers are used in washing machine.....	133
Figure 7.11: Vibrations measurement of washing machine by using 3 rd prototypes of MR dampers.	133
Figure 7.12: Three thermocouples are fixed to the cylinder of 3 rd prototype of MR damper.	134
Figure 7.13: Behavior of temperature at the outside surface of MR damper's cylinder. ..	134
Figure 7.14: Behavior of temperature at the inside surface of MR damper's piston.....	135

Chapter 1

INTRODUCTION

1.1 Magneto Rheological (MR) Fluid Devices

The magneto rheological fluid devices consist of MR fluid that makes the characteristics of the device very versatile and dynamic [1]. The MR fluid consists of micron-sized magnetic particles that are suspended in a carrier fluid. The properties of MR fluid can easily be changed by the magnetic field intensity which is generated due to the applied electric current. The magnetic particles can be used with other types of materials which give rise to other MR materials such as; MR elastomers, MR foams, MR sponges MR grease, MR plastomers, and MR polymer gel [2]. The MR fluid is widely used in different types of devices such as; dampers, vehicle suspension systems, seat suspension systems, prosthesis applications, rotary brakes and etc. The MR fluid devices are divided into four main modes. 1). Shear mode, 2). Squeeze mode, 3). Valve mode and 4). Gradient pinch mode. The orientation and operation of MR fluid in all these four modes are presented in Figure 1.1 to Figure 1.4.

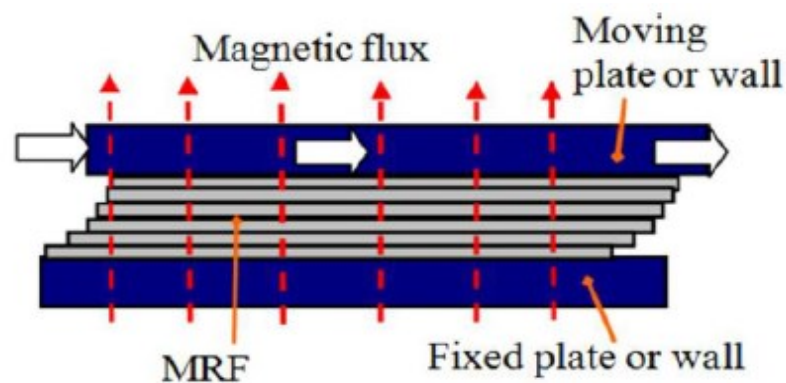


Figure 1.1: Operation of MR fluid in shear mode [3].

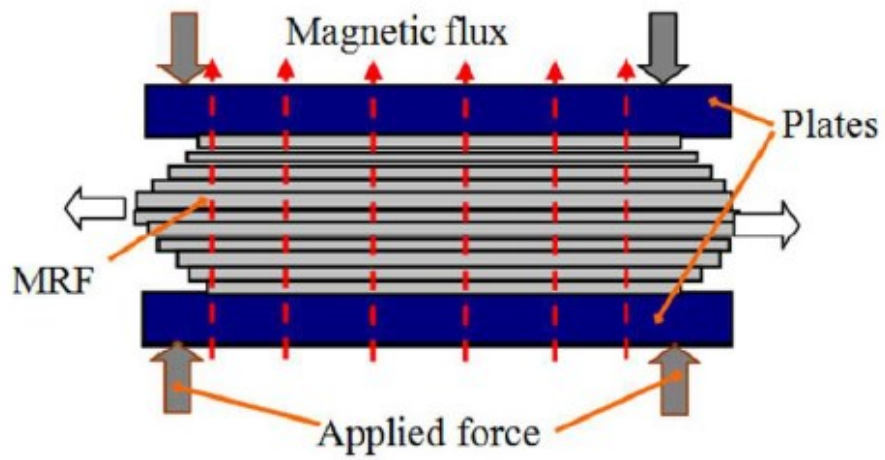


Figure 1.2: Operation of MR fluid in flow mode [3].

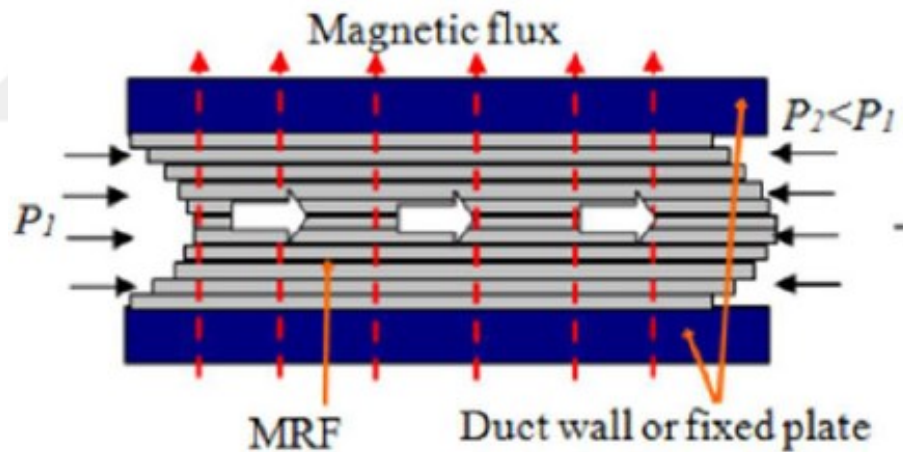


Figure 1.3: Operation of MR fluid in squeeze mode [3].

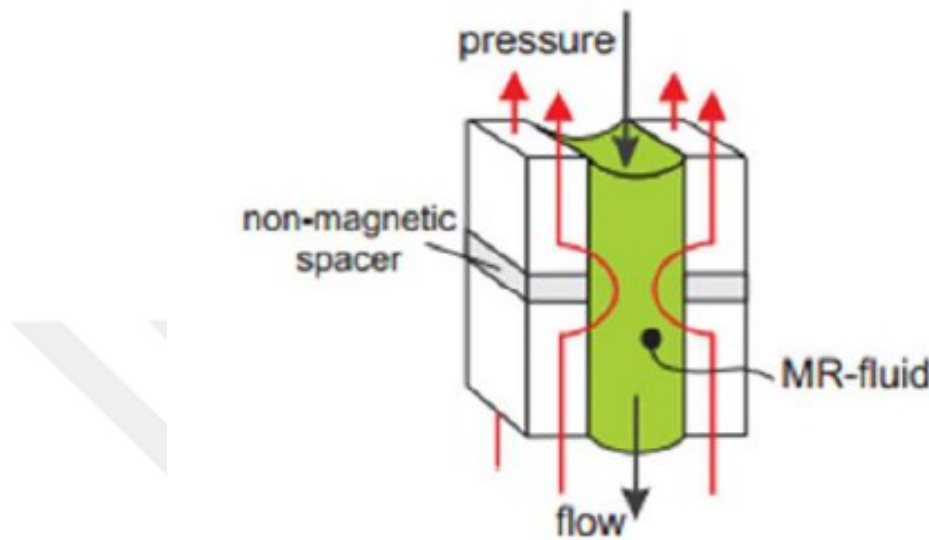


Figure 1.4: Operation of MR fluid in pinch mode [4].

The MR devices can be manufactured in different ways by using numerous and wide variety of designs. The MR devices should meet the requirements from the consumer market so that it efficiently work for the desired application. The MR devices are characterized into three different types such as; 1). MR brake, 2). MR valve and 3). MR damper. The MR damper is a semi-active device that is widely employed in variety of applications. It has drawn significant amount of attention from the consumer sector due to its dynamic characteristics. The MR damper with the flow of magnetic flux in the magnetic circuit is shown in Figure 1.5. These can be used in semi-active vibration control mechanisms due to its fast and dynamic response. They can be manufactured for different ranges of force applications with different capacities. They can also be designed according to the required power and energy requirements. The MR damper consist of the MR fluid, piston, piston rod and the cylinder. The piston of the MR damper consist of magnetic circuit which plays a crucial role in the overall operation of MR damper. The

applied magnetic field intensity at the poles of the electromagnet inside the MR damper results into chain like structures in few milliseconds which provide the desired damping force. A low carbon steel plays very important role in the damping force. Therefore, the overall MR damper's magnetic circuit consist of a low carbon steel, copper wire which forms the turns of the coil and the MR fluid. The magnetic field intensity is dependent upon the applied input current that can be controlled in numerous ways. However, when there is no electric current the MR damper just depends on its passive force.

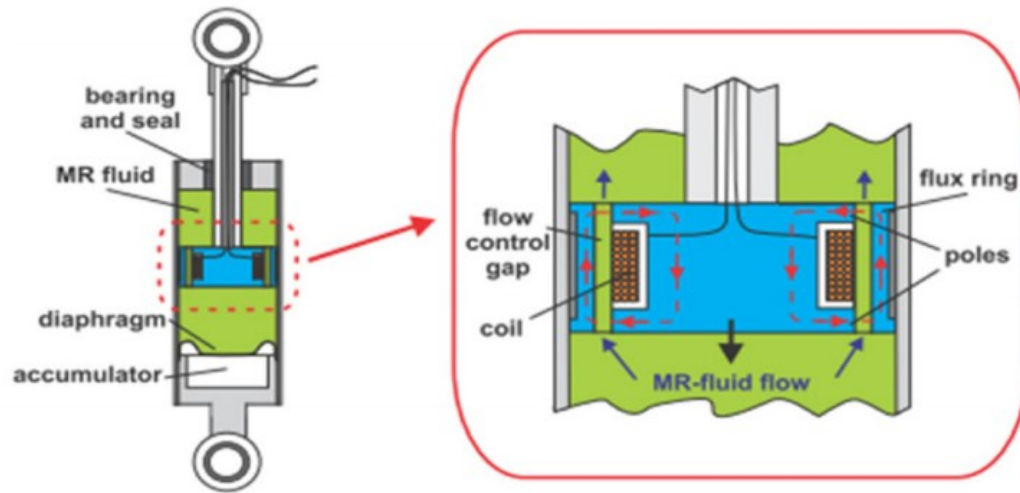


Figure 1.5: MR damper with its flow of magnetic flux in magnetic circuit [5].

On the other hand, MR brakes consists of coil, MR fluid, stator and rotor. These are utilized in the exercise equipments [6]. The MR brakes come in different shapes which is mainly dependent upon the rotor's shape and size. The MR brakes that consumes very less power is shown in Figure 1.6. The MR brakes help in minimizing the angular speed of a particular shaft.

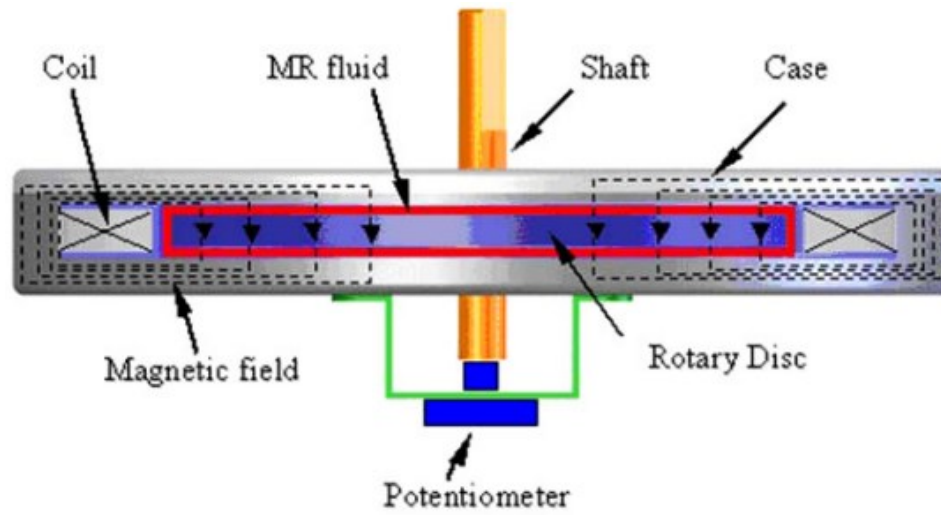


Figure 1.6: Position feedback based Magnetorheological brake [7].

1.2 Design and Force Modeling of Shear Mode MR Damper

The shear mode MR dampers are used in low force applications which require less response time and large range of controllability of damping force. The design of shear mode MR damper is very important for the overall operation of MR damper. The piston, MR fluid with sponge and cylinder of the MR damper makes the overall magnetic circuit of the MR damper. The optimization of the magnetic circuit for the desired damping force is usually done using finite element analysis. The magnetic circuit can also be analytically evaluated using the reluctance method. However, to practically design the MR damper, it is very essential to analyze and optimize the magnetic circuit of the MR damper using finite element analysis techniques. A two dimensional axis symmetric model of the magnetic circuit of a single coil MR damper is shown in Figure 1.7. Each and every part such as piston, cylinder, MR fluid gap or sponge, coil and areas of different regions are labeled to show its contribution to the magnetic circuit of MR damper. The two dimensional axis symmetric model is studied by applying the meshing feature of a FEM software on the model as shown

in Figure 1.8. An appropriate size of the mesh should be chosen so that the analysis is efficiently carried out. The distribution of magnetic flux between the active poles inside the MR damper is shown in Figure 1.9. The magnetic flux lines are flowing from the north to south poles via the cylinder which acts as a return path for the magnetic flux lines.

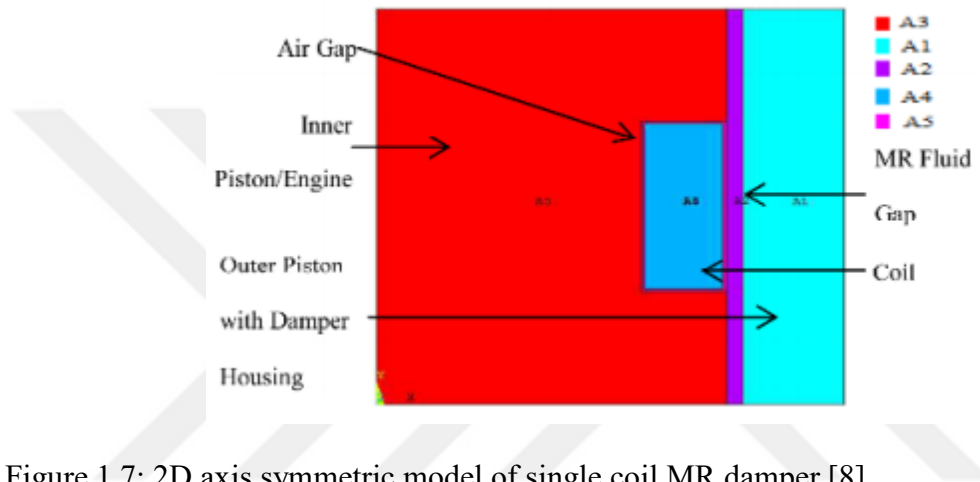


Figure 1.7: 2D axis symmetric model of single coil MR damper [8].

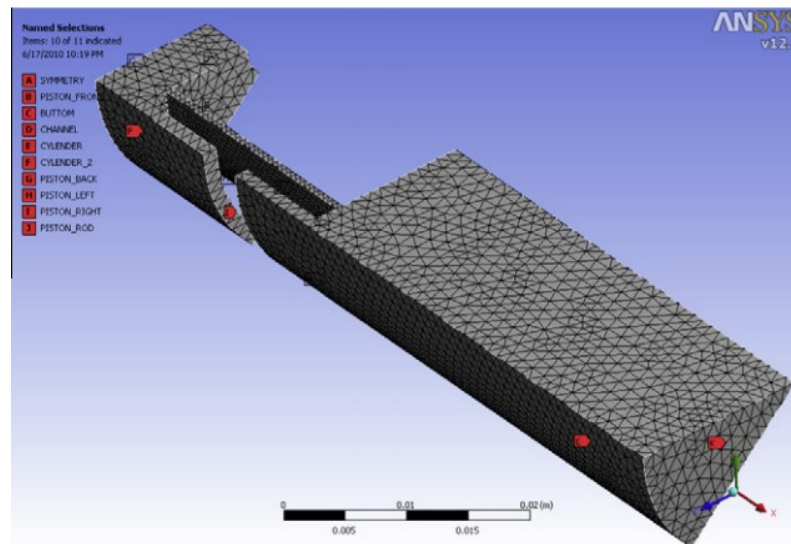


Figure 1.8: Meshing of the magnetic circuit of MR damper [9].

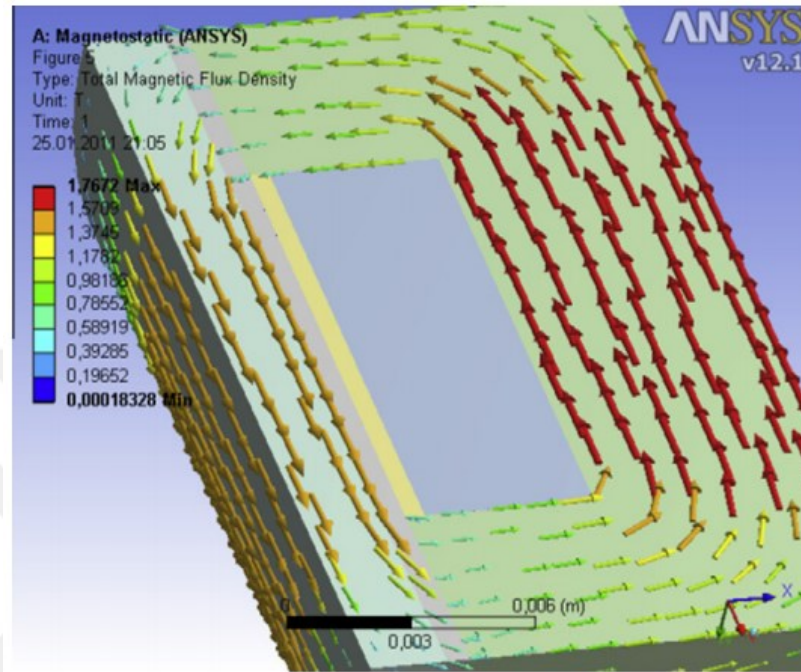


Figure 1.9: Lines of magnetic flux flowing between the poles of the electromagnet of the MR damper [9].

The parametric modeling of MR damper is very essential due to the nonlinear behavior of MR dampers. The hysteresis behavior of the MR dampers can easily be studied using these models. These models tries to efficiently model the hysteresis behavior of MR dampers using less computational time and complexity. The hysteresis curves should accurately approximate the experimental hysteresis curves with minimum error between them. The two most famous models such as; bouc-wen model and hysteretic model are used to simulate the hysteresis responses of MR dampers. The hysteresis curves of force versus velocity and force versus displacement using bouc-wen model are shown in Figure 1-10. Similarly, the same curves are generated in Figure 1-11 using hysteretic model. Both these models are compared in terms of their computational time and accuracy. However, the sharpness of the hysteresis

curve is well achieved by using hysteretic model. The model also simplifies the computational time as compared to bouc-wen model with less error.

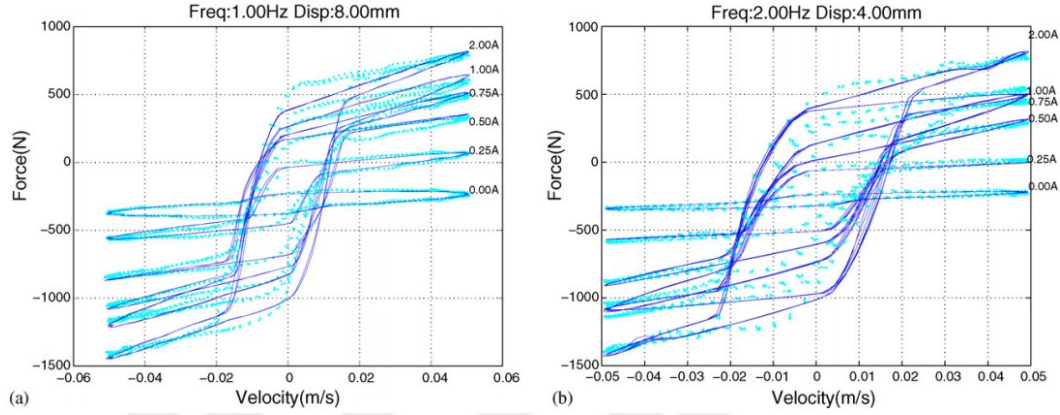


Figure 1.10: (a) Force vs velocity hysteresis curve using bouc-wen model (b) Force vs velocity hysteresis curve using bouc-wen model [10].

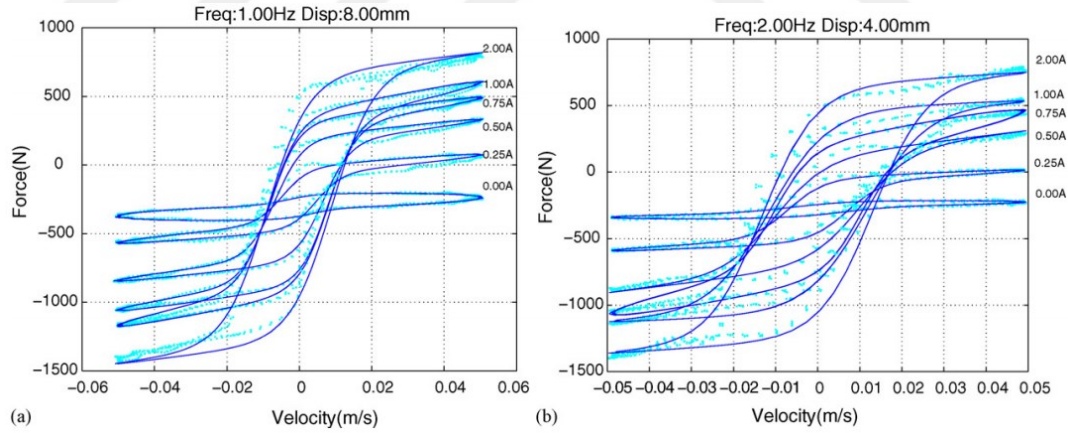


Figure 1.11: (a) Force vs velocity hysteresis curve using hysteretic model (b) Force vs velocity hysteresis curve using hysteretic model [10].

On the other hand, the relation of damping force with respect to electric current is modeled using reluctance method. In this method, the magnetic circuit is analyzed using meshing loops of magnetic flux. The targeted value of magnetic field intensity is calculated

at the active poles of the MR damper. The value of magnetic field intensity is used in the evaluation of the yield stress of the MR damper. The relationship between the yield stress of commercial MR fluid with magnetic field intensity is given in Figure 1-12. After evaluating the yield stress, the overall damping force of MR damper is evaluated by using the yield stress and area of active regions of MR damper. The damping force is dependent upon the applied current. However, the damping force becomes saturated due to material of low carbon steel or silicon steel. The low carbon steel exhibits a nonlinear behavior as can be seen from its BH curve in Figure 1-13. The saturation is also due to the nonlinear behavior of BH curve of MR fluid as given in Figure 1-14. Therefore, the damping force doesn't increase after the applied current changes. It is important to mention that the maximum value of current consumes much power which results into heating and rising temperature of the MR dampers. The heat generated inside the MR damper makes the contents of MR fluid less responsive which degrades the performance of MR damper.

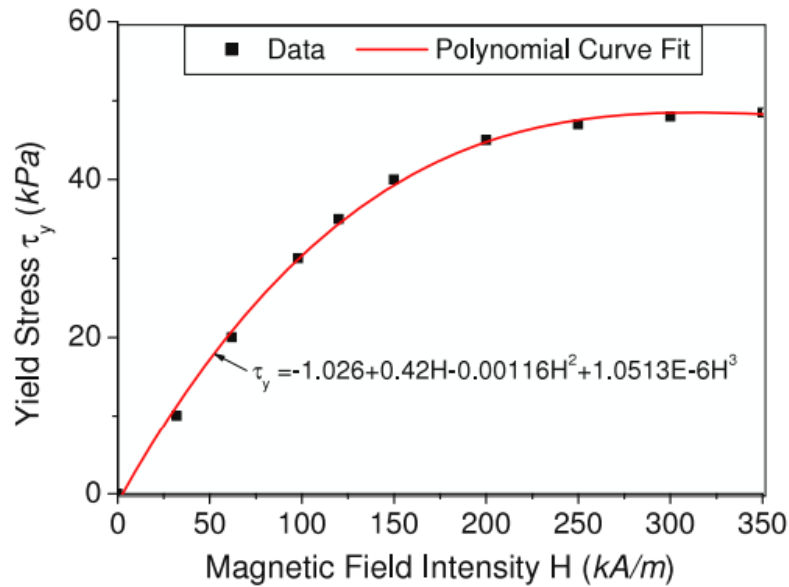


Figure 1.12: Behavior of yield stress as magnetic field intensity varies [11].

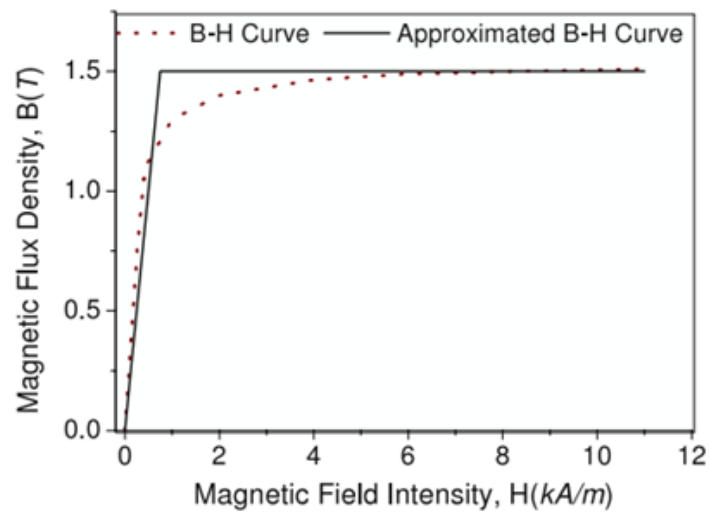


Figure 1.13: BH curve of the silicon steel showing the original and approximated trends [11].

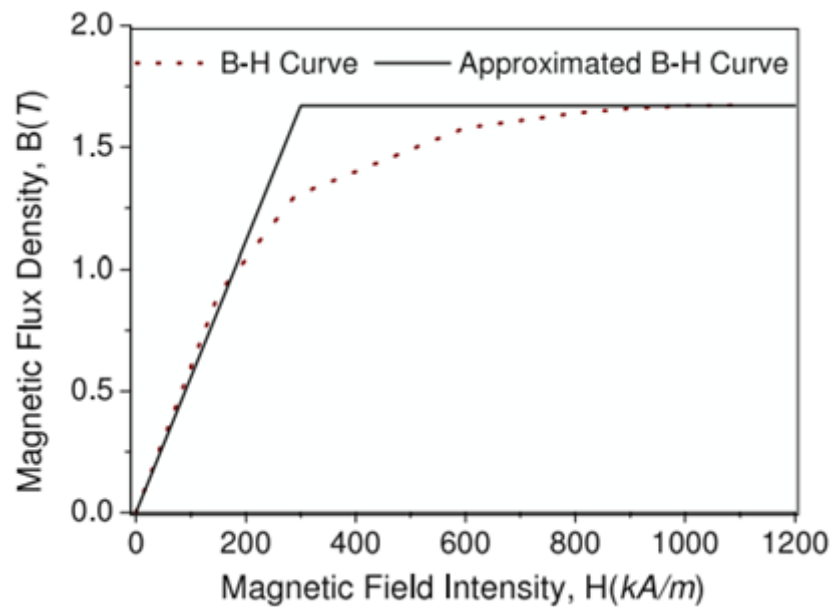


Figure 1.14: BH curve of the MR fluid showing the original and approximated trends [11].

1.3 Control of Shear Mode MR Dampers

The control of shear mode MR dampers is the crucial step when it comes to reducing the response time and fast switching between the force values. The control of MR dampers require a nonlinear, robust and adaptive control algorithm which significantly achieve the desired results. The control algorithm is implemented using a power electronic amplifier. The control input to the power amplifier is given in the form of pulse width modulation (PWM). The power amplifier uses the PWM and changes the current through the MR damper. However, the relation between damping force and electric current requires the information regarding the physical parameters of MR dampers. The damping force should be derived in terms of the physical parameters of MR damper such as; resistance and inductance of the winding of electromagnet, area of active poles and yield stress of MR fluid inside the MR damper. A typical flow of semi-active control using MR dampers is illustrated in Figure 1.15.

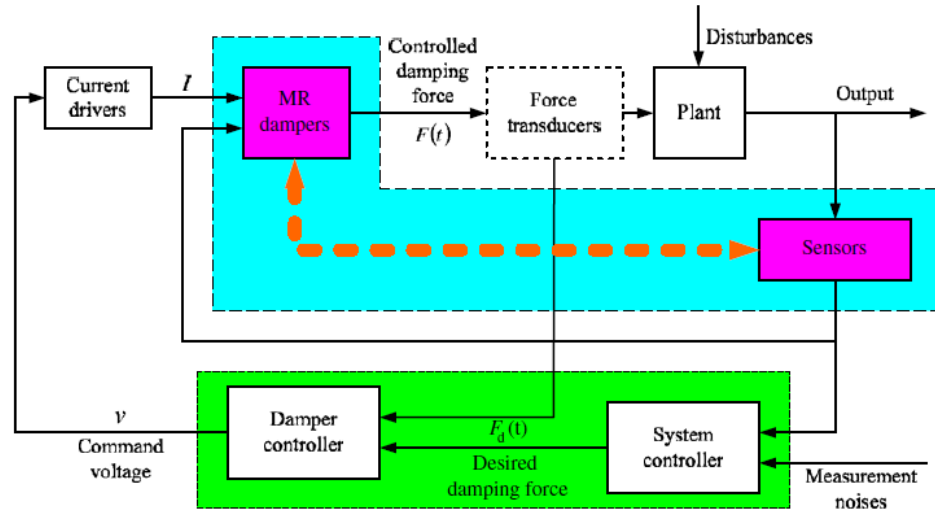


Figure 1.15: Control of a plant in the presence of disturbances by using MR dampers [12].

1.4 Application of MR Dampers in Washing Machine

The operation of washing machine is strongly dependent upon the behavior of its vibration which it experiences when the machines passes through different ranges of RPMs. The mass of the wet clothes are distributed inside the tub of the washing machine. When the washing machine operates in the spinning mode, the wet clothes stick with the surface of the drum which creates huge amplitude of vibrations in different ranges of RPMs. As the washing machine consist of suspension system, the vibrations are transferred to its base and cabinet. The vibrations from the base and cabinet are much visible at high RPMs. The transmissibility curve of the force show the overall behavior of the washing machine throughout its spinning cycle as shown in Figure 1.16.

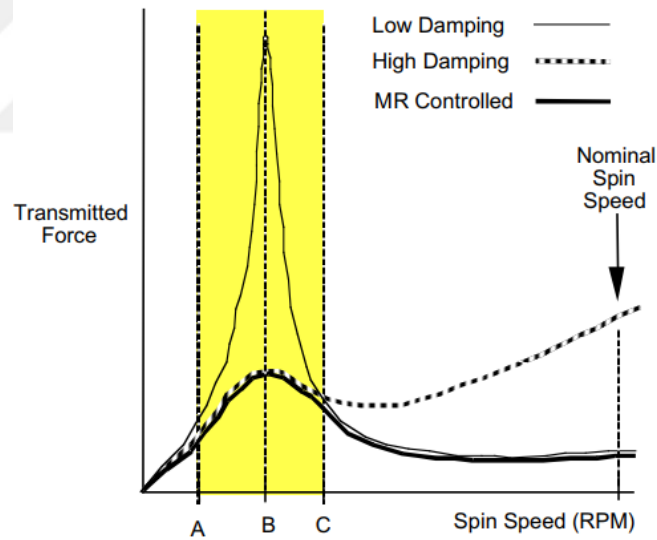


Figure 1.16: Transmitted force curve of washing machine versus RPM [13].

The curve shows the application of low, high and MR damping for the full cycle of washing machine. The low damping performs badly between A and C points. The region between points A and C show the low resonance region of washing machine. However, the performance is improved when it crosses the point C. The high damping gives effective

suppression of transmitted force between A and C points. However, the high damping result helps in increasing the vibration after point C. Therefore, a variable and controlled damping is required to effectively reduce the transmitted force of washing machine for all ranges of RPMs.

1.5 Aims and Objectives of Research

The passive dampers in the washing machine are not effective to minimize the vibrations that are experienced by the washing machine. The active dampers reduces the vibrations in the low resonance region but it gives bad performance at high RPMs. Therefore, a semi-active device in the form of MR damper counters the vibrations in all regions of spinning cycle of washing machine. The unbalanced mass inside the tub generates centrifugal force which result into severe vibrations. The MR dampers should be used instead of passive dampers for the suppression of vibrations of washing machine.

The main of this research is to focus on the design, analysis and control of MR dampers for the reduction of vibrations of washing machine. The selection of the appropriate mode of the MR damper is very important in order to reduce the size, cost and improve the performance of MR damper. In addition to this, the MR damper should have very high value of bandwidth so that it can work at wide range of frequencies of washing machine. The shear mode of MR damper is chosen for the design of the MR damper. The shear mode incur less cost and the usage of MR fluid is also in very less amount as compared to its counterpart MR fluid mode dampers. The geometry of the MR damper is very crucial for the optimum size. The size of the MR damper is optimized using single coil and double coil configurations. The 3D CAD model is generated for both the configurations. Both the configurations are solved analytically using reluctance method. However, the method only consider few loops which are not sufficient to model the desired value of magnetic flux density and damping force. Therefore, in order to design the MR damper for the desired value of magnetic flux and force, it is essential to optimize its geometry by using finite element analysis techniques. The design

parameters of the shear mode MR damper are studied and analyzed in terms of magnetic flux density. The value of magnetic flux density is parametrically varied with respect to the length of active regions for different values of design parameters. The geometry is also optimized by using first order techniques. The low carbon steel is used for the cylinder and piston part of the MR damper. The piston rod and cover of cylinder is selected to avoid the leakage of flux. All the parts of the MR damper are manufactured using CNC machining. The winding of the coil is done using copper wire. A sponge is selected so that MR fluid is kept inside in the sponge. The piston of the MR damper is covered with the sponge having MR fluid. Commercial MR fluid is used for the MR damper to initially test it. However, this research aims to use MR fluid that is made in the laboratory. The performance of both the MR fluids are investigated by testing them inside the MR damper.

The first prototype of shear mode MR damper is made by using single coil configuration. The MR dampers are tested inside the washing machine and they successfully reduce the vibrations by using On/Off controller. The vibrations are measured and controlled using laser sensors and accelerometers. However, electronic control is required in order to swiftly control the flow of current through MR dampers with less response time. Moreover, the size of first prototype of MR damper needs to be reduced. For this purpose, a second prototype of shear mode MR damper is designed using double coil configuration. The damper is tested and it achieves the same desired value of high damping force and minimum force when no electric current is applied. Besides this, a power electronic converters and amplifiers are studied in order to effectively control the electric current. Different DC-DC converters are analyzed and implemented for the MR dampers. The MR damper is modeled as resistor and inductor and mathematically modeled with the DC-DC buck converter. The buck converter is a step down converter which operates on the PWM signal. The PWM signal has to be controlled by an adaptive and robust control algorithm. A sliding mode control algorithm is designed and the convergence parameter of the sliding surface is formulated in terms of the

physical parameters of the MR damper for the washing machine. By using nonlinear sliding surface and designed power converter, the PWM signal is controlled in real time which directly controls the flow of current using the power converter. By using the second prototype together with designed controller, the vibrations are more reduced as compared to the first prototype. The third prototype of MR damper is also manufactured with much reduced size as compared to second prototype. The third prototype is also tested inside the washing machine by using the designed adaptive control algorithm and power converter. The temperature of both the second and third prototypes are measured using thermocouples in order to see the heating effect during the operation of washing machine.

The force sensors are used inside the washing machine to observe the transmitted force generated due to unbalanced mass. The transmissibility is validated with the analytical model that is developed for the washing machine. The unbalanced force of the tub of washing machine acting on the MR dampers is measured by using the force sensors. A feedback control is developed using force sensors in order to counter the unbalanced forces in low resonance region. The MR dampers applies the desired values of damping forces for countering the unbalanced forces. However, at high range of RPMs the washing machine requires low value of force. Therefore, by using acceleration sensor and laser sensors the MR dampers are switched off and the MR dampers use its passive force in order to decrease the unwanted vibrations.

Chapter 2

LITERATURE REVIEW

2.1 Overview

This thesis explains the design optimization of shear mode MR damper and its application for the washing machine. The design optimization is done using finite element analysis and reluctance method. A current driver is designed for the MR damper that consist of a DC-DC power converter and an algorithm which is robust and adaptive. The purpose of the current driver is to reduce the response time as well as to maintain the required value of current for the MR dampers. A novel analytical model of the adaptive sliding mode control is proposed for the current control of MR damper inside washing machine. All the results for the controller are verified analytically and experimentally. This chapter deeply reviews the MR dampers and its application in different fields of engineering in general and for the washing machine in particular. The methodologies for the vibration reduction of washing machine are discussed. The thesis in detail review different control strategies of MR damper. In addition to this, different nonlinear control algorithms such as back-stepping and sliding mode control are discussed. Finally, a detail discussion is provided for the state of the art application of MR dampers and its control for the washing machine.

2.2 Operational Modes and Types of MR Dampers

The MR damper is one of the most widely used semi-active device as compared to its other counterparts. There are three different modes of operation for the MR damper. In the shear mode, the piston moves with respect to the stationary cylinder as shown in Figure 2.1. The MR fluid is sheared between the two surfaces. This type of MR damper is used in variety of different low force applications. The shear mode MR damper is usually manufactured using MR sponge or foams which is ideal for the applications such as; washing machines,

brakes, clutches and locking devices. On the other hand, the flow mode MR damper works on the principle of flow of MR fluid due to the gradient of the pressure between the two surfaces of piston and cylinder respectively as given in Figure 2.2. However, there is a sealing problem in the flow mode MR damper and also the MR fluid is used in very high volume. On the other hand, the shear mode MR damper doesn't need any sealing and very less volume of MR fluid is used. Also the shear mode MR damper doesn't need any gas chamber or accumulator as compared to flow mode MR damper. Besides these two modes of MR damper, there is another type of MR damper that is squeeze mode MR damper as shown in Figure 2.3. In this mode, the MR fluid is operated between the two surfaces that are perpendicular to the surfaces. This type of MR damper is used in applications which generates very less vibrations. The squeeze mode damper is also used in applications which require impact MR dampers [2] and [17-23]. Besides, this many other modes of MR dampers can be created by using any two basic modes such as flow and squeeze mode to meet the needs and demands of a particular application as shown in Figure 2.4. Moreover, a pinch mode is also developed for the operation of MR fluid in the MR damper [24-29].

Basically, the MR dampers consists of a magnetic circuit which is made using the cylinder, MR fluid and the piston. The piston is the electromagnet which can be single coil or multicoil. The MR damper works on the principle of forming chain like structures of magnetic flux density which originates from one pole that travels and reaches to other pole via the cylinder. In between the two surfaces of piston and cylinder, there is an MR fluid that provides support in forming strong chains of magnetic particles between the surfaces. There are different types of the designs of MR damper which has its own unique application. The mono tube MR damper is shown in Figure 2.4. It is a flow mode MR damper that consists of a single cylinder, piston and a gas chamber. When the piston moves the MR fluid comes to the extension chamber to generate the damping force by changing the viscosity of MR fluid. The gas chamber is filled with nitrogen gas. The accumulator manages the volume of MR

fluid due to the movements of the piston. It also manages the pressure in the control valve section of the MR damper [30].

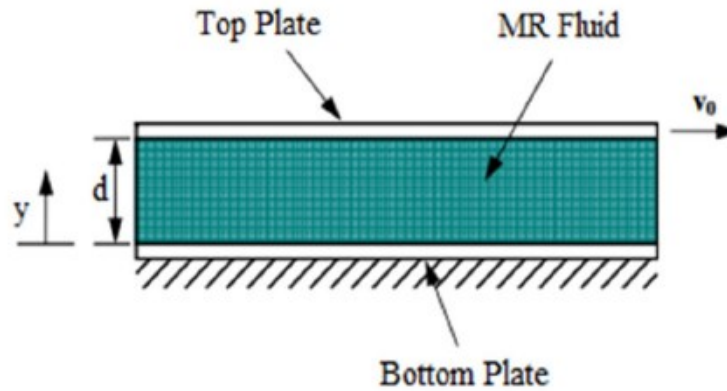


Figure 2.1: A control valve of MR damper showing shear mode operation [14].

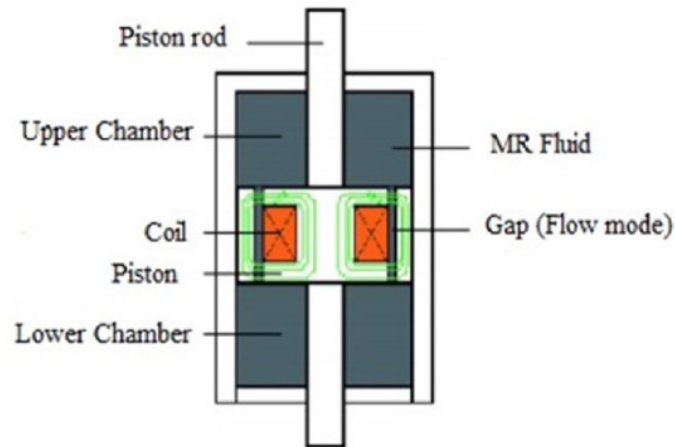


Figure 2.2: A control valve of MR damper showing flow mode operation [14].

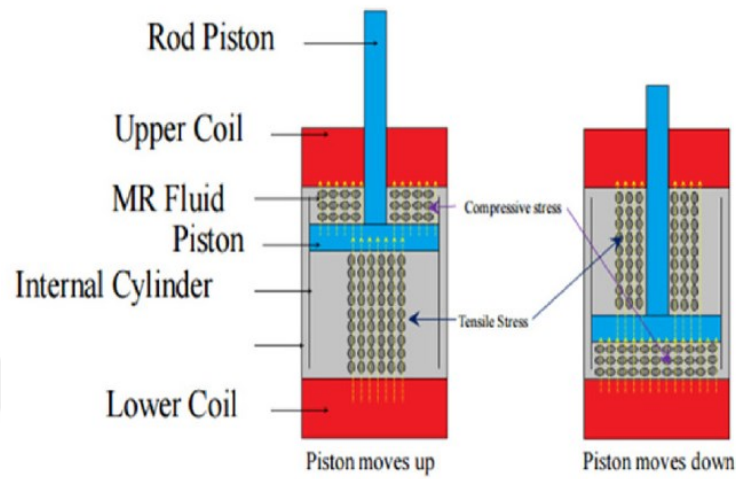


Figure 2.3: A control valve of MR damper showing squeeze mode operation [15].

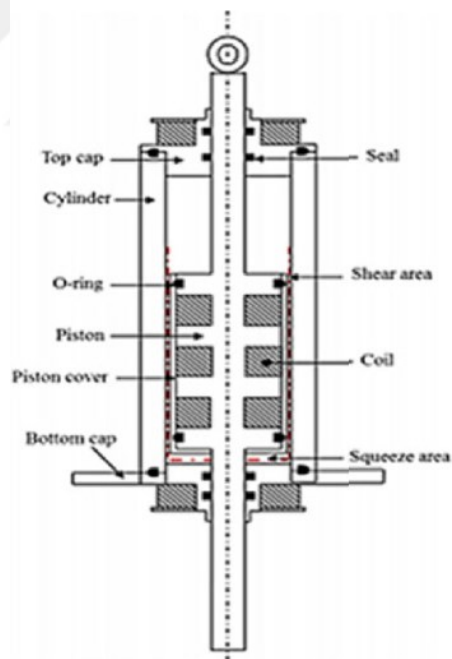


Figure 2.4: A control valve of MR damper showing mix mode operation [16].

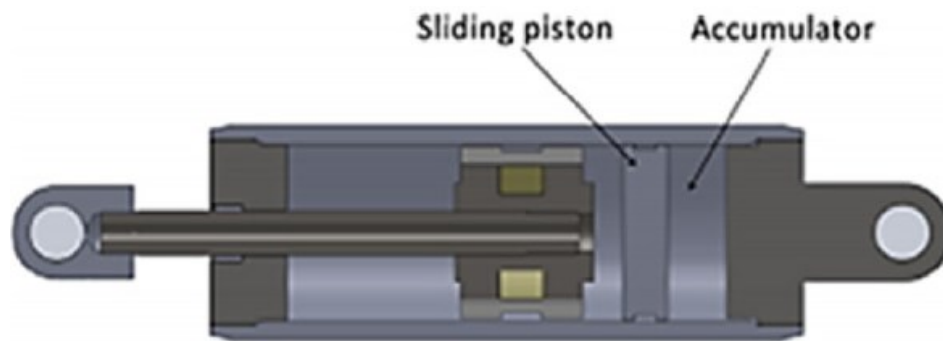


Figure 2.5: A design of MR damper having mono tube assembly [31].

There is another design that is called a twin tube MR damper. In this type of design, there are two housings such as outer and inner as shown in Figure 2.6. The MR fluid is kept in the inner housing. The outer housing has also some MR fluid but less than the inner housing. This less volume of MR fluid inside the outer housing helps in compensation of change in volume of MR fluid due to motion of the piston.

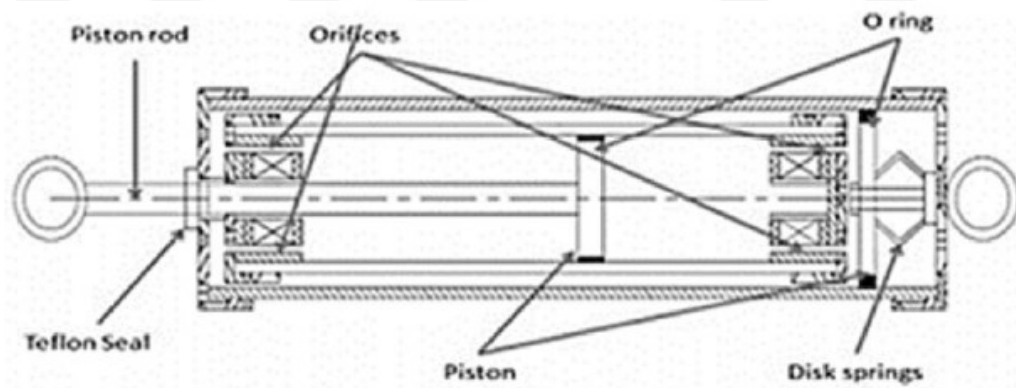


Figure 2.6: A design of MR damper having twin tube assembly [32].

A double ended MR damper is just like a mono tube damper as shown in Figure 2.7. However, in double ended MR damper, it doesn't need any accumulator due to which no effect of spring can be observed [30] and [33]. A hydraulic hybrid type of MR damper is given in Figure 2.8.

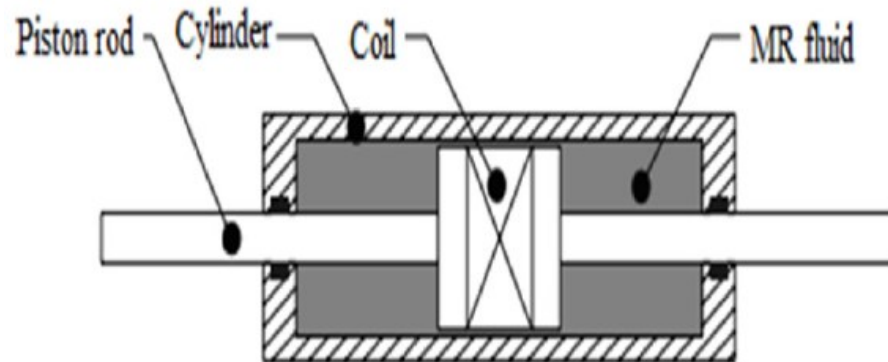


Figure 2.7: A design of MR damper having double assembly [34].

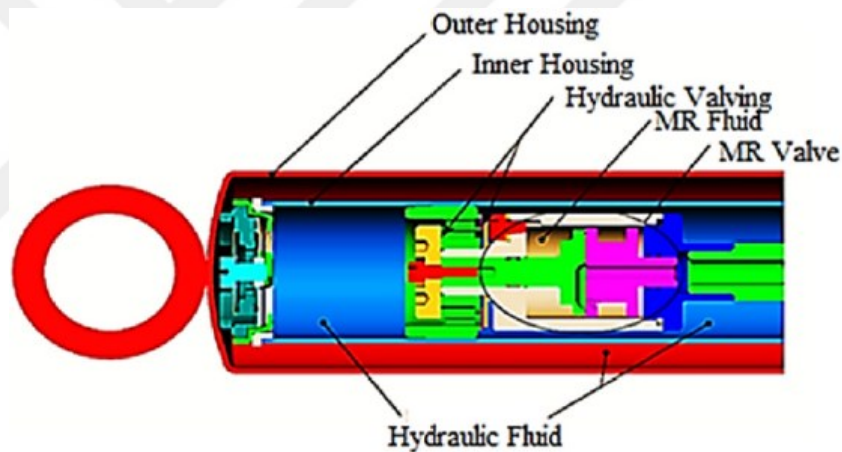


Figure 2.8: A design of MR damper having hybrid hydraulic arrangement [30].

The coil of MR damper can be designed in different configurations such as internal coil and external coil. Figure 2.9 shows the internal coil configuration for the MR damper. The coil is wound internally on the designed piston of the MR damper. In the internal coil configuration, the direction of magnetic flux lines are perpendicular to the flow of MR fluid. All modes of operation can be implementable with this type of arrangement. On the other hand, when the coil is wound in such a way that it doesn't have any direct contact with the MR fluid region then this is called external coil winding as shown in Figure 2.10. In external coil configuration, the direction of magnetic flux lines are parallel to the flow of MR fluid.

However, maximum capacity of the pressure is achieved with fast response by using internal coil as compared to the external coil configuration [1].

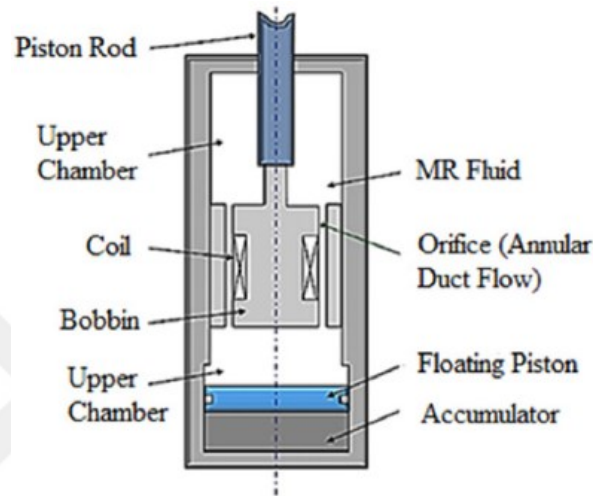


Figure 2.9: The MR damper showing the internal coil configuration [35].

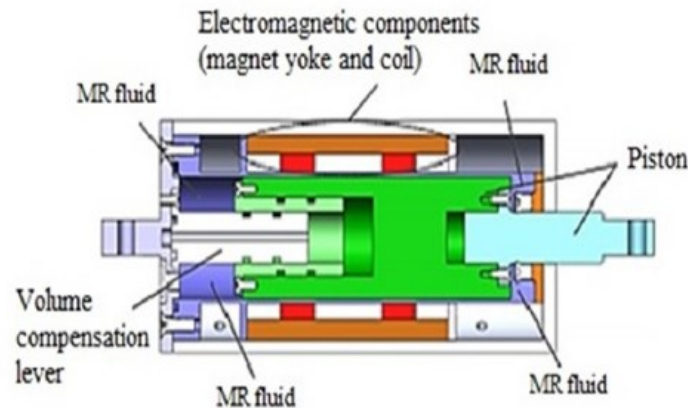


Figure 2.10: The MR damper showing the external coil configuration [36].

The number of coils have also great impact on the overall magnetic flux density and the damping of the MR damper. The MR damper can be designed as single coil, double coil and multi turn coil. A single coil arrangement is a simple geometry in which a coil is wound with particular number of turns as shown in Figure 2.11. A double coil configuration is designed

using the middle pole with twice of length as compared to the remaining two poles of the electromagnet of MR damper. The number of turns are selected in clockwise and counter clockwise directions to maximize the magnetic flux density at the middle pole of the MR damper. The MR damper can be designed with more than two coil configurations such as multi turn coil as shown in Figure 2.13. The coils can be placed in such a way so that the overall magnetic flux density and force values are achieved. The activation areas are also increased by using multi turn coil configuration. The multi turn coil are used in those applications which require high damping force under some constraints. In multi turn coil design, a faster response can be achieved by using parallel connection of coils as compared to the series connection [33]. However, by increasing the number of coils the stroke length is reduced which affects the performance of the MR damper used in the system.

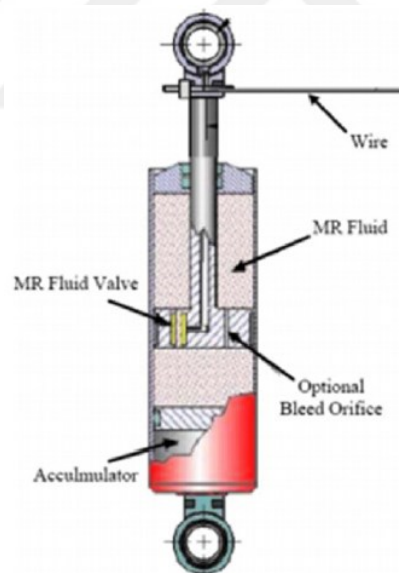


Figure 2.11: The MR damper showing the single coil configuration [37].

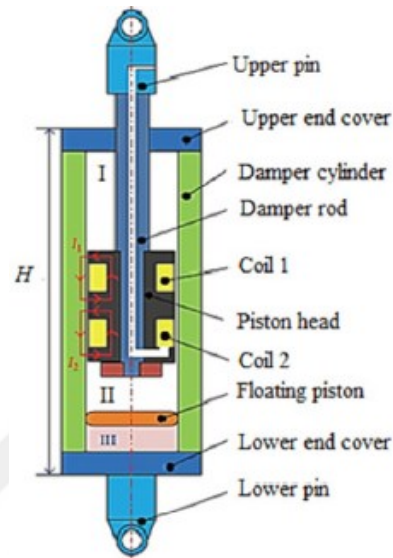


Figure 2.12: The MR damper showing the double coil configuration [38].

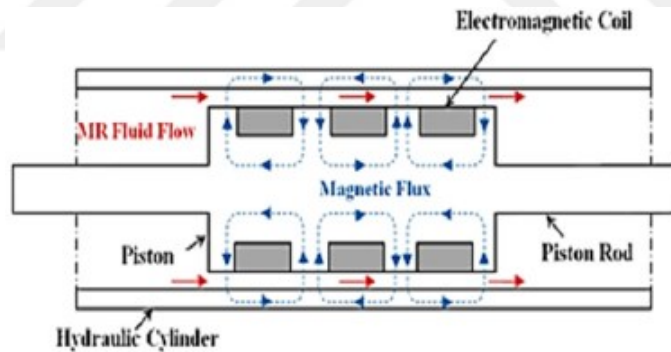


Figure 2.13: The MR damper showing the multi turn coil configuration [39].

The inner bypass MR damper has two channels for the flow of MR fluid as shown in Figure 2.14. The first channel which is outer channel that is created between the sleeve of the shielding and the cylindrical part of the MR damper and is not affected due to magnetic flux density. The other channel which is the inner channel that is formed between the piston and yoke part of the MR damper and is controlled by the applied magnetic flux density. The external bypass damper is designed in which the coil assembly is placed outside the housing

of MR damper as shown in Figure 2.15. This helps in reducing the heat generated inside the other designs of MR damper due to which the MR fluid is not affected.

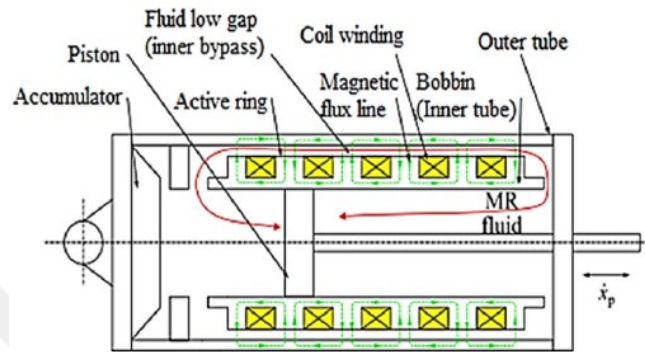


Figure 2.14: The MR damper showing the inner bypass configuration [40].

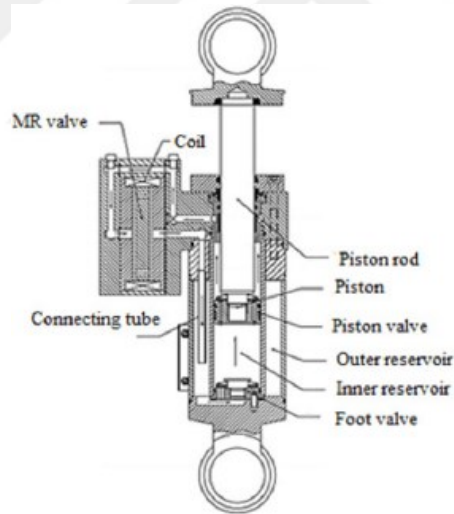


Figure 2.15: The MR damper showing the external bypass configuration [16].

A finite element analysis technique is used to optimize the geometry of the MR damper. The magnetic circuit is thoroughly analyzed using the finite element analysis technique [8] and [41]. The value of magnetic flux density and force of the MR damper is investigated at various values of input electric current. By using finite element analysis, the optimized

design is turned out to be very realistic as compared to the reluctance method which take few magnetic loops for the solution of magnetic circuit.

2.3 Applications of MR devices

The MR devices are vastly used in different fields of application and engineering. In mechanical engineering, it is used in damping of the motor, shock absorber, seat suspension of the operator and other heavy vehicles used for the construction [42-48]. It can be used in other applications which need both linear and rotational motion [49]. It is also used in making fixtures for other mechanically oriented tasks which helps in reducing the machining time with better accuracy [50]. The MR devices are used in systems which finishing and polishing. In power transmission system, the MR devices are also utilized efficiently [51-53]. The MR dampers are used in different fields of technologies such as shock test machine, highway bridge and drawing press [54-60]. Likewise, they are also used in wheeled tractor seat suspension, prosthesis knee and cable-stayed bridge [61-67]. Figure 2.16 shows the suspension of a seat using MR dampers for the drivers. This suspension consists of air reservoir, MR damper and controller and position sensor. This facility for the drivers protects their health and safety. This increases the efficiency and productivity of the drivers. The MR dampers are also used in seats of the crew in helicopter [69]. The MR damper is also implemented for the ambulance bed seats for the patients as shown in Figure 2.17. The MR damper is utilized in several biomedical engineering purposes such as; prosthesis, movement of artificial limb, manipulator of artificial muscle and to give strength to the rehabilitation systems [71-73]. The MR damper helps in the attenuation of tremors [74]. The MR dampers are used in the drilling of the deep holes [75]. The MR damper is also used for the attenuation of chatter vibrations of the boring bar [76]. The MR dampers are widely used in the absorption of vibrations which are generated due to the earthquake. For this purpose, large MR dampers of high damping force in kN are used to attenuate the shocks and vibrations

caused in the structures [77]. The structural beam is controlled optimally in the presence of disturbances by using MR fluid mount which has the operation of mixed mode [78].

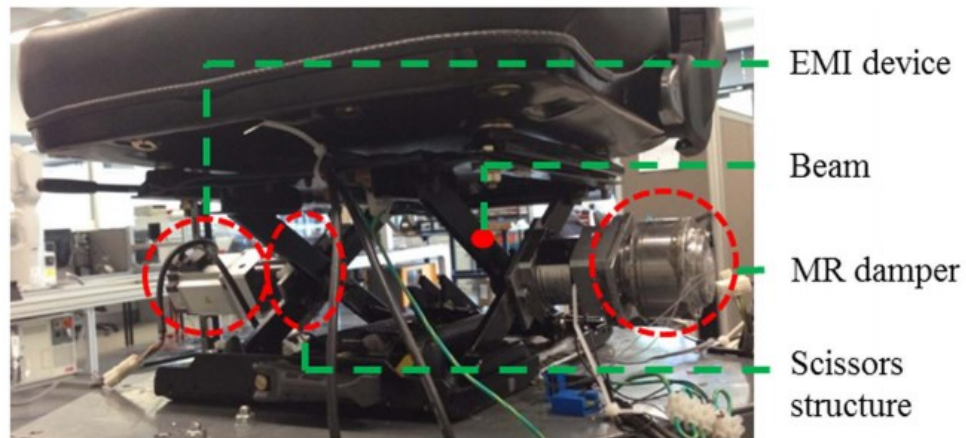


Figure 2.16: Seat suspension system using MR dampers [68].

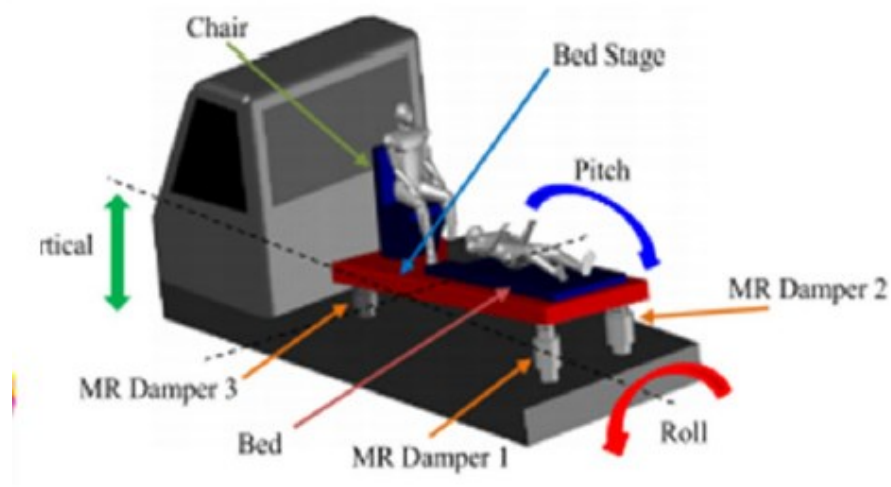


Figure 2.17: MR dampers are used for the bed seat of the patient in the ambulance [70].

2.4 Control of MR Devices

This research further focuses on the design and implementation of adaptive controller for MR dampers. In adaptive controller, the parameters of the controller adjust itself according to the desired output of the system. Different mechanisms have been proposed for controlling the MR devices. However, appropriate modelling and selection of control algorithm for the controller is crucial for the desired results. The control schemes for the MR devices are divided into different types such as; classical control, adaptive and hybrid control, adaptive fuzzy control, optimal control and H-infinity control. In Skyhook control, the control gain takes any arbitrary value [79-81]. However, the gain can't be changed in the real time. Another modified version of skyhook controller is proposed which is skyhook but with no jerk [82]. The new modified controller is better than the traditional skyhook controller. Optimal control such as LQG and sliding mode control is implemented on the quarter model of suspension of car [83]. A skyhook controller is also implemented on the MR based landing gear [84]. A fuzzy skyhook control is utilized for the MR dampers when the impact load applied force on the MR dampers [85]. Another classical control such as PID control is implemented for industrial purposes [86]. The gains of the PID controller plays great role for the dynamic response of the output. [33] has used a PID controller together with the current driver for decreasing the response time as shown in Figure 2.18. In the vehicle suspension, a PID controller is proposed by using MR dampers [87]. A PD controller together with H-infinity method is used for the desired value of damping force in [88]. A fuzzy based PID is presented for the vehicle suspension using MR dampers [89]. The vibrations of a truss is controlled using MR damper and LQR control technique [90]. The LQR controller is designed using MR dampers for the structures of the beam [91]. The LQG control is used for the structures of civil engineering using MR dampers in order to decrease the vibrations [92]. On the other side, due to the nonlinear response and uncertainties of MR damper, a sliding mode control is proposed due to several disadvantages associated with the optimal controllers

and classical controllers. The sliding mode control is a robust control which performs very well in the case of half modeled dynamics and disturbances. A sliding mode control is suggested for the comfort of ride in a vehicle using MR dampers [93]. A sliding mode based observer is proposed for the suspension of the vehicle in [94]. The sliding mode is also utilized for the MR based mounts and the brakes [95]. A fuzzy based sliding mode controller is presented for the MR based rotary dampers and mounts [96-97]. A sliding mode control is used for the squeeze mode MR damper and the performance of the robust controller is successfully compared with PID controller [98]. The fuzzy controller is implemented for the control of semi-active plants [99]. [100] has done work on the usage of fuzzy controller for the quarter model of the car. [101] has done research on the reduction the response of the seismic using MR damper and fuzzy controller.

There are different adaptive control algorithms that are chosen according to the adjustment of parameters. The adaptive control methods are divided into model-based and non-model based control algorithms. Information of model parameters is required in model-based algorithms. On the other side, non-model-based algorithms have used the neural networks as well as fuzzy logic [102-103]. These types of models are very much used for the non-linear systems. These models are also implemented in the presence of any uncertainty occurred in the system [104-105]. Feed-forward and feedback are two adaptive controller strategies. The adaptive controllers are recognized based on these two types. The control input is evaluated using the desired and output signals [106]. On the other side, the regulation in adaptive based feed-forward controller is dependent on the processing of information or the measurement of disturbances [107-108]. There are also direct and indirect adaptive controllers. In direct adaptive controller, the parameters are estimated and used in the controller directly. On the other hand, the estimated parameters in indirect type are used to evaluate the targeted controller parameters.

The robust control method is used to make the desired changes in the parameters according to the system output. These changes should be in some particular bounds [109-111]. The adaptive controller is different from the robust controller because it requires information about the bounds of the parameters [112]. On the other hand, the control law in adaptive control is changed according to the modifications made in the system [113]. Gain scheduling is a technique used in model-based adaptive controllers. In this technique, the controller is designed for various points of operation. A feedback controller is designed at each point which has constant gain to achieve the targets of system. The gain in this type of controller is dependent on time and is evaluated by using varying sets of gain for the points in operation [114-116]. An unstable behaviour is observed when the sudden change occur in the gain of controller. The Lyapunov stability is used intensively rigorously for the adaptive controllers in order to derive the convergence and laws of adaptation. A sliding mode control method is a model-based adaptive control technique [117]. In this strategy, the trajectory of the response of the output is derived using Lyapunov method [118-119]. In addition to this, the sliding surface is formed by using the convergence variable and state variables. On the other side, a back stepping technique is used which is a recursive control algorithm for the stability of nonlinear plants [120]. In this algorithm, Lyapunov stability criteria is utilized in order to derive virtual control and the error signal is needed for rewriting equations of system dynamics. In the field of MR device technology, an adaptive controller is designed for the MR fluid based systems [121]. A hybrid adaptive controller is built by combining type 2 interval fuzzy controller with modified SMC [122]. The chattering of the system is attenuated by modifying the surface of switching of SMC and the adaptive laws are evaluated using theory of Lyapunov. An adaptive neural based direct and indirect controllers are designed for the suspension of vehicle using MR dampers [123-124]. An adaptive neural based controller is used for the civil structures using MR dampers [125-126]. A direct adaptive controller is implemented on MR brakes for the targeted value of torque [127-130].

However, this research work focuses on the design of adaptive terminal sliding mode control for the minimization of the response time of MR damper. For this purpose, a nonlinear sliding surface is designed using terminal sliding mode technique [131]. A complete modeling of the electronic converter, MR damper and control algorithm is done, which can't be found in the literature. The complete model is analytically evaluated and then simulated. The experimental results are verified by the simulation of the model.

2.5 Reduction of Vibrations Using MR Dampers

The washing machine experiences vibrations when the laundry filled drum rotates at specific RPMS [132]. This rotation creates centrifugal force which results into unbalanced force generated from the tub [133]. For decreasing these vibrations, there are two main methods proposed. One method use hydraulics balancer to make the tub balance [134-135]. This method uses salt with water in the balancer to make the tub stable [134]. On the other side, two masses can also be used to balance the tub [135]. Overall, the technique is very good for decreasing vibrations, However, the cost is very high as well as the system becomes more complex. There is another method which uses variable damping implemented with some control to kill the vibrations in the washing machine. The low damping will not help to reduce the energy of vibrations at low RPMS but it works well at high RPMS. On the other side, high damping solves the problem at low RPMS but it increases the vibrations at high RPMS. A variable damping with control technique solve the problem.

As discussed earlier, the MR foam or sponge damper is very good for applications which require low value of force. This type of MR damper operates in shear mode. The complexities are very less in this type of MR damper as compared to MR flow mode damper which is basically MR fluid damper. Many MR fluid dampers are used for decreasing the vibrations in the washing machine [136-141]. The MR foam dampers are also used for washing machine [13] and [142-145]. However, these designs generate the damping force in very small value as well as the passive force is also very high in value. This research focuses on the design of

MR shear mode based damper. Three prototypes of MR damper are made and tested along with the adaptive controller for the minimization of vibrations of the washing machine. A finite element analysis (FEA) is used for the finalization of overall geometry and the experimental results are validated by the FEA results. The controller results are also matched with the simulation analysis. Overall, the vibrations are reduced by all the MR dampers and controller.



Chapter 3

DESIGN AND ANALYSIS OF FIRST PROTOTYPE OF MR DAMPER

3.1 Introduction

This chapter focuses on the complete design optimization of the first prototype of MR shear mode damper. First, the CAD model is presented with its cross-sectional view. The optimization is performed using FEA and golden section algorithm which is a first order technique. The magnetic circuit of the first prototype is analyzed using the reluctance method. Based on that method, the damping force of the MR damper is also evaluated. In the end, complete procedure of optimization of MR damper and vibration mechanism is discussed.

3.2 Design and Analysis

The two dimensional model of the washing machine is presented in Figure 3.1. The assembly of the tub of the washing machine has two springs and two passive damper from the company of washing machine. Tub assembly experiences vibrations when it passes through different frequencies. The vibrations are experienced by the washing machine due to the unbalanced laundry inside the drum. The passive dampers are not able to suppress all the vibrations inside the washing machine. They are less effective at low and high frequencies of resonance, which sometimes caused unwanted noise and also sometimes it makes the washing machine out of order. A three dimensional model of the MR sponge damper is given in Figure 3.2. The MR damper comprises of piston with its rod, ball joint, clamp for the cylinder and the cylinder and its cover. The MR damper piston is covered with a sponge having commercial MR fluid. Its physical properties are given in Table 3.1. This type of MR damper functions in shear mode. The MR damper piston makes motion inside the housing of MR damper with a stroke length of ± 15 mm. A low carbon steel is required for the design

of the MR damper. Aluminum is used for the clamp and piston rod for avoiding the leakage of magnetic flux.

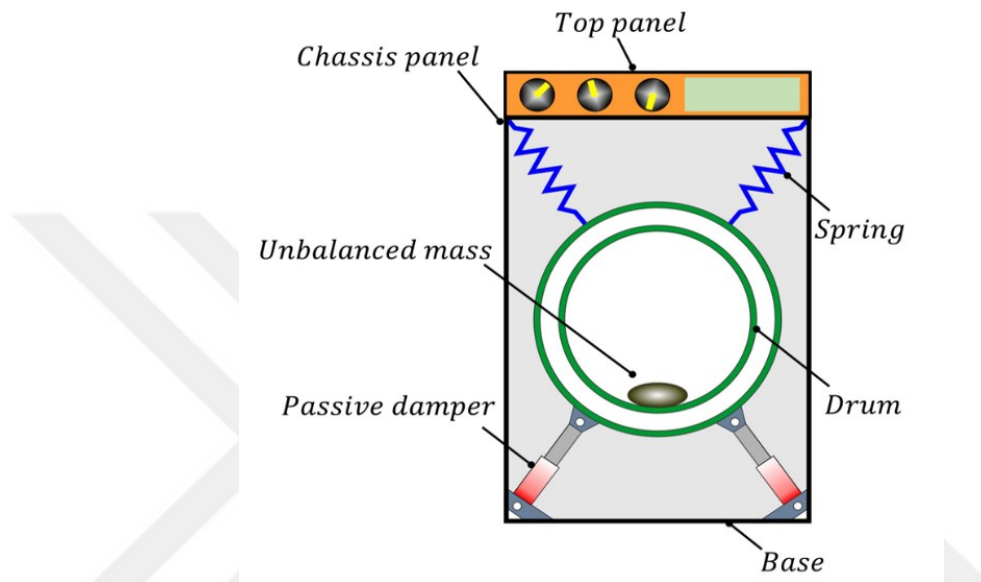


Figure 3.1: 2D CAD model of the front loaded washing machine.

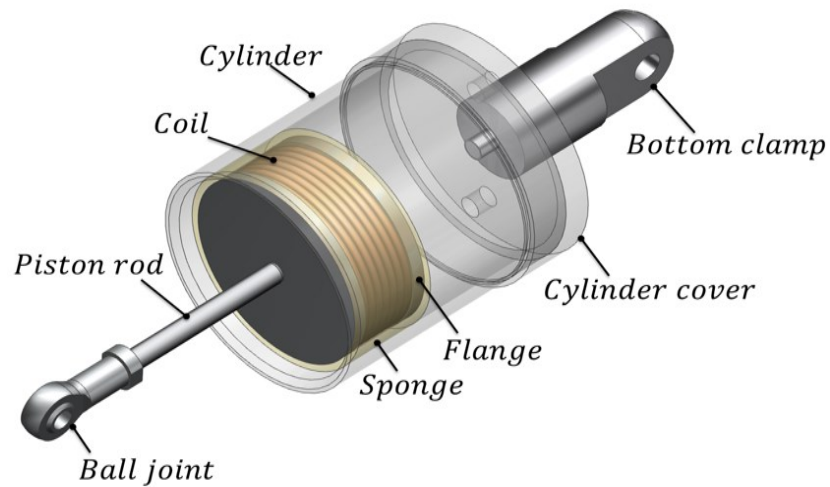


Figure 3.2: 3D CAD model of the MR shear mode sponge damper.

A two dimensional cross sectional view of MR damper is given in Figure 3.3. The cross sectional view shows the geometric parameters which need to be designed such as length of flange, length of coil, thickness of sponge, radius of core and thickness of cylinder. These parameters play very critical role in the overall performance of MR damper.

Table 3.1: Specifications of commercial MR fluid [139]

Appearance	Dark gray liquid
Viscosity, Pa-s @ (40°C (140°F) Calculated as slope 800-1200 sec^{-1}	0.280 $\pm$ 0.070
Density $\frac{g}{cm^3}$ ($\frac{lb}{gal}$)	3.54 – 3.74 (29.5-31.2)
Solid Content by Weight, %	85.44
Flash Point, °C(°F)	> 150(> 302)
Operating Temperature, °C (°F)	–40 to + 130 (–40 to + 266)

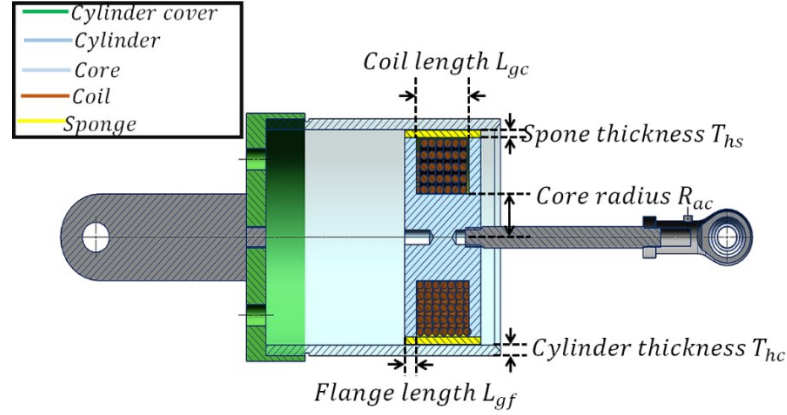


Figure 3.3: 2D cross- sectional view of the MR shear mode sponge damper showing its geometric parameters.

The finite element analysis (FEA) is performed on the 2D axis symmetric mode of MR damper. The meshing of the magnetic circuit of MR damper is done by using 2216 elements of boundary and 112725 elements of the domain. The input current of 0.25 A is applied during the FEA. The diameter of copper wire is fixed at 0.33 mm. This diameter handles the 0.25 A current very well. The damping force of the MR shear mode damper relies on the magnetic flux density at the poles of the MR damper. Moreover, it depends on the MR fluid and the area of active poles. A 2D axis symmetric model of MR damper is given in Figure 3.4. A magnetic flux is evaluated at the active poles by doing the parametric analysis of all the geometric parameters. The area of active poles is dependent upon the length of flange. Surface area of the poles is inversely proportional to the value of magnetic flux density [139]. By increasing the length of flange, very large volume of the MR fluid would be used at the active poles. But with high value of length of flange, the piston mass increases which results into the low value of natural frequency. The length of coil is in direct relationship with the magnetic flux density [139]. The low value of length of coil gives less MR effect through the sponge of MR damper. On the other side, the high length of coil pushes the cylinder thickness value to a very small number. This also brings magnetic saturation in the cylindrical part of the MR damper. The high value of length of coil also affects the flange length which results in the reduction of the MR fluid volume at the active poles of the MR damper. The final value of the length of the coil is chosen according to the targeted value of the length of the stroke of the piston. In Figure 3.5, a sweep in terms of length coil is performed in the FEA for the range of 20 to 50 mm. The length of the coil is chosen in between the range of 20 to 30 mm. The coil length is selected for the targeted value of length of the stroke and magnetic flux density. The thickness of the cylinder is changed between 2 to 8 mm as given in Figure 3.6. It is seen that the magnetic flux density increases with the increase in thickness value as discussed in [11], [139] and [141]. But on the other hand, it will result into saturation as the magnetic flux increases. Therefore, 4 mm is chosen as thickness of cylinder for the MR

damper. On the other side, high value of thickness result into high value of passive force of MR shear mode damper. The radius of the core increases with respect to the magnetic flux density at the active poles. The value of damping force is increased with high value of radius of the core. But on the other hand, the core radius also gives high value of passive force of MR damper [139]. It also contributes in the mass of the damper. The value of the sponge thickness also affects the performance of MR damper. The thickness of the sponge changes in the inverse relation with respect to magnetic flux density as discussed in [11], [139] and [141]. By using low value of thickness, the MR effect increases which increases the damping force. In the same manner, all other geometric variables are analyzed as shown in Figure 3.7 to Figure 3.9. The maximum value of the yield stress is gained by using the perfect alignment of carbonyl iron particles inside the sponge under the effect of magnetic flux density. The particles form chain structures that are perpendicular to the cylindrical surface. The yield stress increases with the MR fluid particles weight percent. The weight percent increases with the number of particles of MR fluid. In rheometer, the yield stress is observed in relation with the commercial MR fluid particles weight percent. The commercial MR fluid has 85.44 weight percent particles. It is clearly observed in Figure 3.10, that by decreasing particles weight percent to 80 percent, the yield stress significantly decreases.

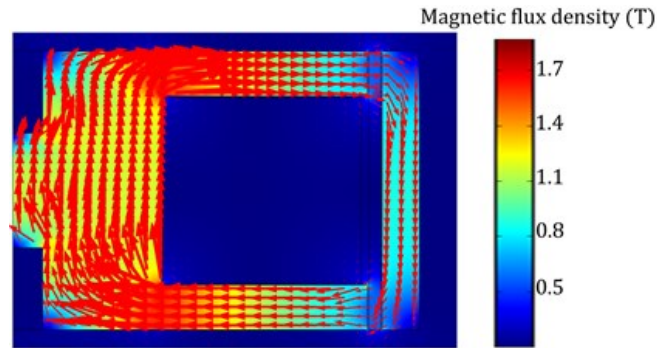


Figure 3.4: 2D axis symmetric model of MR damper showing the magnetic flux lines and distribution.

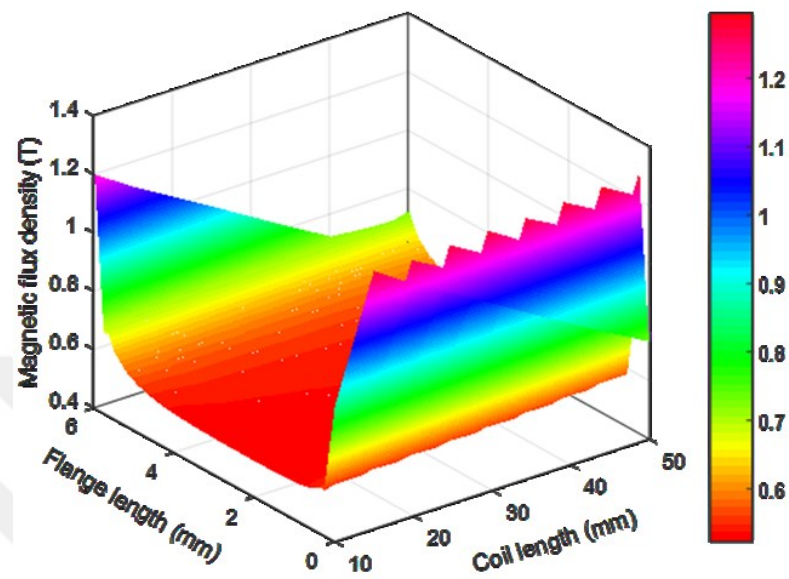


Figure 3.5: Behavior of magnetic flux density at a pole for the changing length of coil.

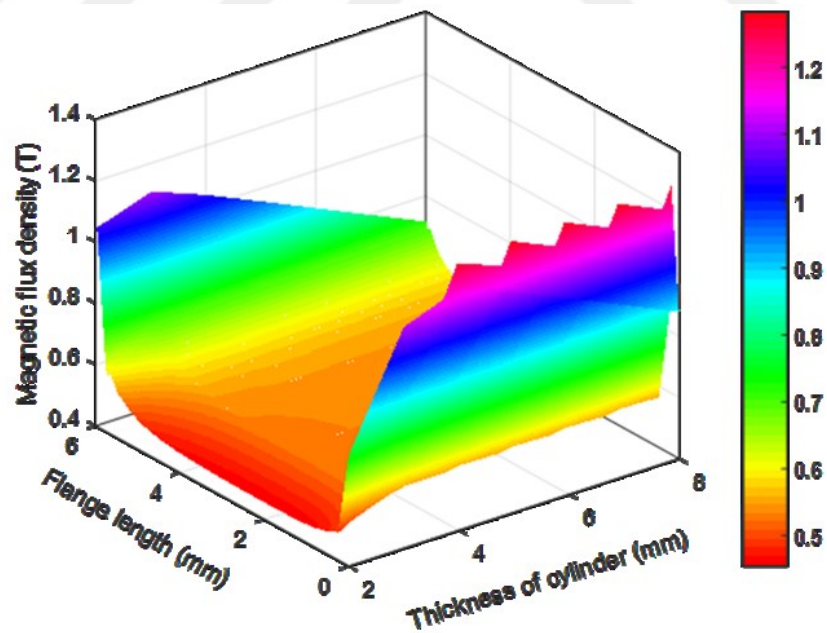


Figure 3.6: Behavior of magnetic flux density at a pole for the changing thickness of cylinder.

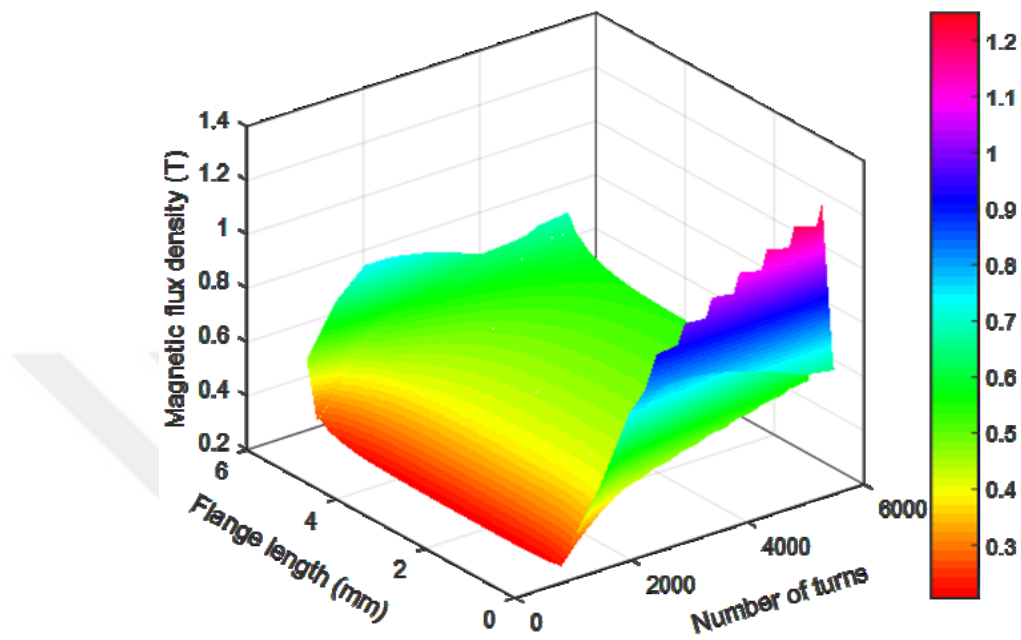


Figure 3.7: Behavior of magnetic flux density at a pole for the number of turns of a coil.

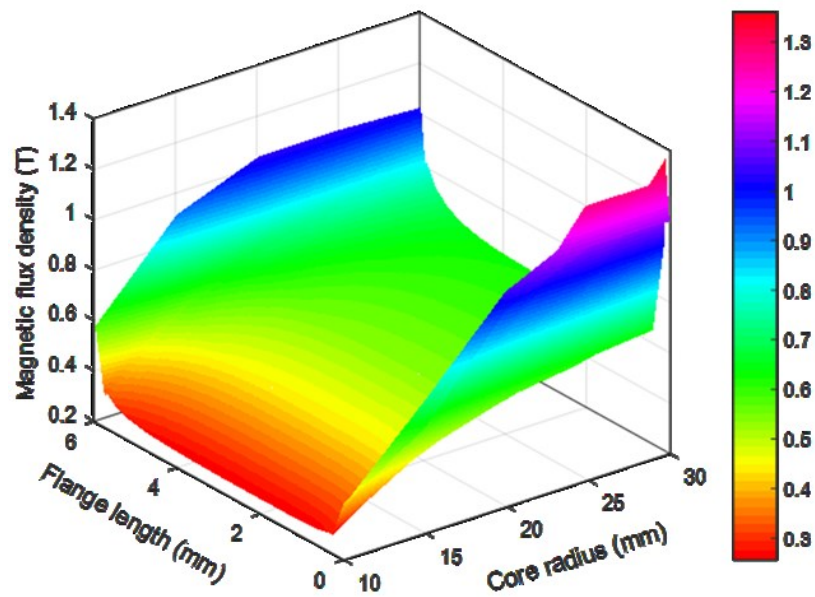


Figure 3.8: Behavior of magnetic flux density at a pole for changing the radius of core.

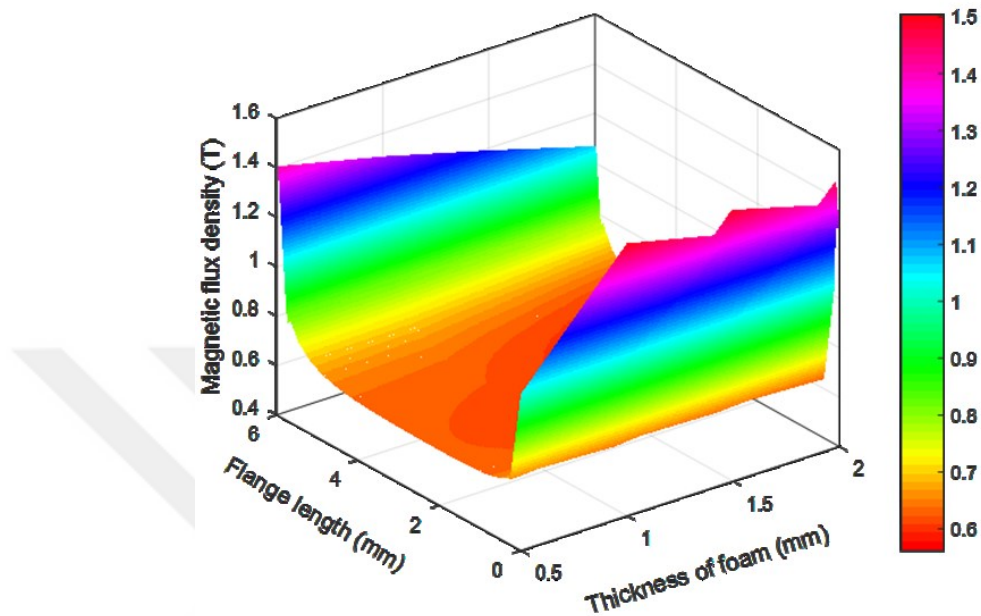


Figure 3.9: Behavior of magnetic flux density at a pole for changing thickness of sponge.

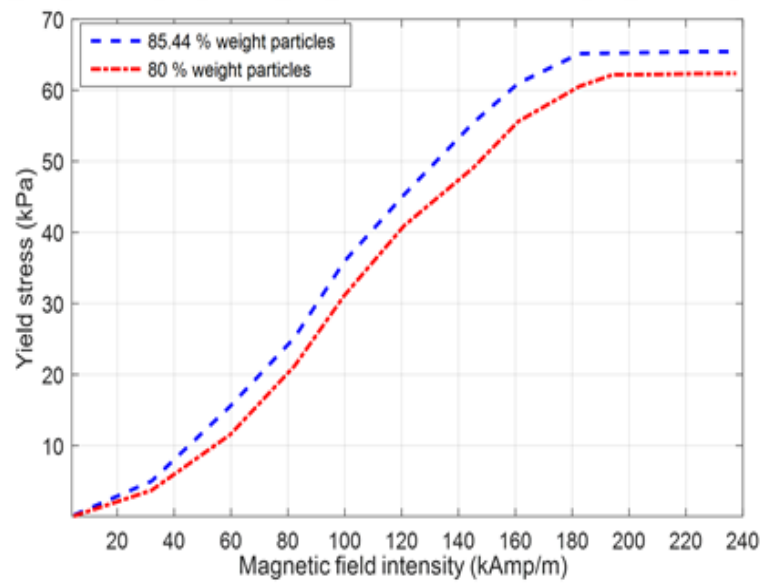


Figure 3.10: Effect of weight percent of particles on the yield stress of MR fluid.

The design optimization of the MR damper is also performed using first order, second order and non-derivative techniques. The second order has highest rate of convergence, but for large geometry this technique is very expensive for the solution of linear equations. The non-derivative methods has low rate of convergence. This technique is suitable for the problems which are very random behavior. But on the other side, it is not consider good for the problems where the design parameters are not dependent on each other. The non-derivative techniques are not advisable for the systems which uses the MR fluid. The first order derivative technique has low rate of convergence as compared to the second order but the cost of computation is not expensive. Therefore, a golden section algorithm (GSA) is applied on Figure 3.12. The steps for the optimization is given below and the results are shown in Figure 3.11.

- (1) The geometric parameters G_p are initialized such as; L_{gf} , L_{gc} , R_{ac} , T_{hc} and T_{hs} .
- (2) The values of l_g , O_r , W_a , l_{g1} , M_a , A_{r1} , l_{g10} , A_{r10} , l_{g9} , A_{r9} , l_{g7} , A_{r7} , l_{g6} , A_{r6} are calculated, where W_a and M_a are the wire cross-sectional and surface areas of active poles respectively.
- (3) The magnetic flux density B_{dm} , should be maximized by using the objective function as shown by Equation (3.1).

$$B_{dm}(G_p) = \frac{\mu_{s4}(l_g - 2L_{gf})(O_r - R_{ac} - T_{hs} - T_{hc})I}{W_a \left(2T_{hs} + \frac{l_{g1}\mu_{mf}M_a}{\mu_{lst}A_{r1}} + \frac{2l_{g10}\mu_{mf}M_a}{\mu_{lst}A_{r10}} + \frac{2l_{g9}\mu_{mf}M_a}{\mu_{lst}A_{r9}} + \frac{2l_{g7}\mu_{mf}M_a}{\mu_{lst}A_{r7}} + \frac{l_{g6}\mu_{mf}M_a}{\mu_{lst}A_{r6}} \right)}$$

Subject to:

$$\begin{aligned} L_{gfmin} &\leq L_{gf} \leq L_{gfmax} \\ L_{gcmin} &\leq L_{gc} \leq L_{gcmax} \\ R_{acmin} &\leq R_{ac} \leq R_{acmax} \\ T_{hcmin} &\leq T_{hc} \leq T_{hcmax} \\ T_{hsmin} &\leq T_{hs} \leq T_{hsmax} \end{aligned} \tag{3.1}$$

Where I is the value of the electric current, μ_{lst} and μ_{mf} are the permeability of steel and MR fluid respectively.

(4) In the $(n+1)$ iteration, the values of the geometric parameters are updated by using golden section technique as given by Equation (3.2). Where Q_n is the search parameter for the line and J^n is the vector for direction which is equated to the negative gradient of $B_{dm}(G_p)$ at $n = 0$ iteration.

$$G_p^{n+1} = G_p^n + Q_n J^n \quad (3.2)$$

(5) The optimization procedure is stopped when the geometric parameters G_p are converged, otherwise step number 4 is performed again till the rate of convergence is completed.

The finite element method (FEM) and GSA results are very near to each other after performing the optimization procedure. The final geometry is presented in Table 3.2 which is used for manufacturing and testing of MR damper inside washing machine.

Table 3.2: Optimization results of MR damper using FEM and GSA

Variables	FEM	GSA	Optimized value
L_{gf} (mm)	6.013	5.975	6
L_{gc} (mm)	25.109	25.014	25
R_{ac} (mm)	20.052	19.891	20
T_{hc} (mm)	5.027	4.954	5
T_{hs} (mm)	1.608	1.583	1.6
B_{dm} (T)	0.7204	0.7157	0.72

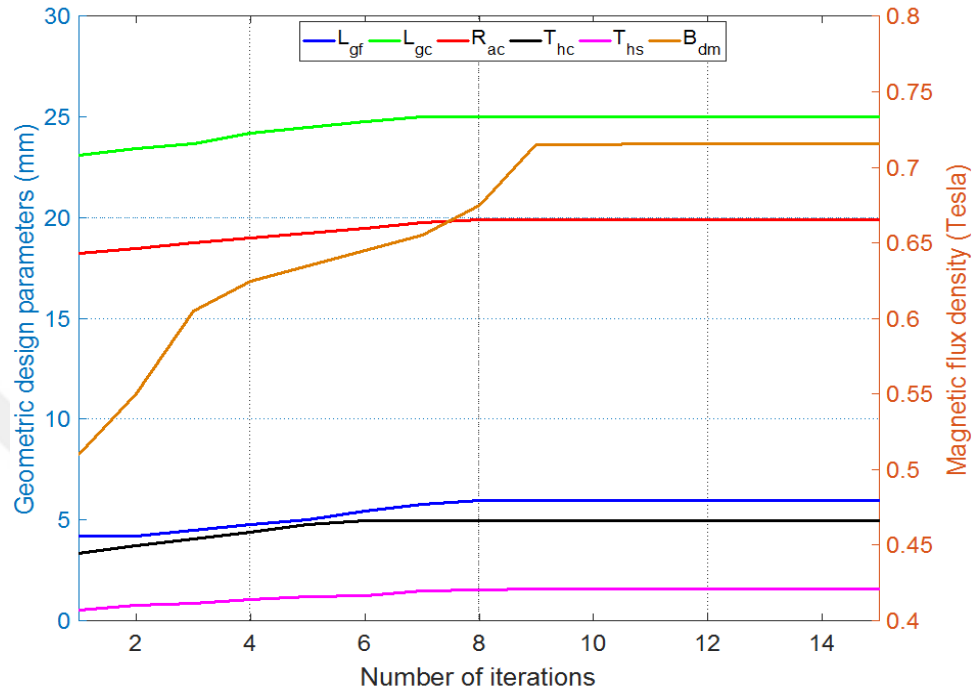


Figure 3.11: Golden section technique results for the optimization of geometric variables.

The MR damper magnetic circuit is studied analytically by using the cross-sectional diagram of a single coil as shown in Figure 3.12. The circuit is segmented into ten various regions of reluctance $\mathfrak{R}$. The total reluctance $\mathfrak{R}_{rt}$ is given by Equation (3.3).

$$\mathfrak{R}_{rt} = \sum_{n=1}^k R_{rn} = \sum_{n=1}^k \frac{l_n}{\mu_n A_n}; \text{ where } k = 10$$

$$\mathfrak{R}_{rt} = \mathfrak{R}_{r1} + \mathfrak{R}_{r2} + \mathfrak{R}_{r3} + \mathfrak{R}_{r4} + \mathfrak{R}_{r5} + \mathfrak{R}_{r6} + \mathfrak{R}_{r7} + \mathfrak{R}_{r8} + \mathfrak{R}_{r9} + \mathfrak{R}_{r10} \quad (3.3)$$

Where l_{gn} is the n th region length, μ_n is the permeability of the material of the n th region and A_n is the area of cross section at the center point of n th region. All the areas and lengths of 10 regions are given in Equation (3.4).

$$l_{g6} = l_g - L_{gf}; O_r = R_{ac} + W_{aw} + T_{hs} + T_{hc}; I_{nr} = R_{ac} + W_{aw} + T_{hs}; A_{r6} = \pi(O_r^2 - I_{nr}^2)$$

$$l_{g8} = l_{g4} = T_{hs}; R_{r8} = R_{r4} = R_{ac} + W_{aw} + \frac{T_{hs}}{2}$$

$$\begin{aligned}
A_{r8} &= A_{r4} = 2\pi R_{r8} L_{gf} \\
l_{g7} &= l_{g5} = \frac{T_{hc}}{2}; R_{r7} = R_{r5} = O_r - \frac{3}{4} T_{hc}; A_{r7} = A_{r5} = 2\pi R_{r7} L_{gf} \\
l_{g1} &= l_g - 2t_{hr}; A_{r1} = \pi R_{ac}^2 \\
l_{g9} &= l_{g3} = \frac{R_{ac}}{2} + W_{aw}; R_{r9} = R_{r3} = \frac{W_{aw}}{2} + \frac{3}{4} R_{ac}; A_{r9} = A_{r3} = 2\pi R_{r9} L_{gf} \\
l_{g10} &= l_{g2} = \frac{t_{hr}}{2}; A_{r10} = A_{r2} = \pi(R_{ac}^2 - t_{ra}^2)
\end{aligned} \tag{3.4}$$

The MR damper magnetic circuit is analyzed by using magnetic based Kirchoff voltage law on Figure 3.13. Where Φ is the magnetic flux, $\mathcal{F}$ is the magnetomotive force, $\mathfrak{R}_{rt}$ is the total reluctance, N is the turns of the coil and I is the electric current. The resultant equation for the KVL is given in Equation (3.5).

$$NI = \Phi(\mathfrak{R}_{r1} + \mathfrak{R}_{r2} + \mathfrak{R}_{r3} + \mathfrak{R}_{r4} + \mathfrak{R}_{r5} + \mathfrak{R}_{r6} + \mathfrak{R}_{r7} + \mathfrak{R}_{r8} + \mathfrak{R}_{r9} + \mathfrak{R}_{r10}) \tag{3.5}$$

Where $\mathfrak{R}_{r2} = \mathfrak{R}_{r10}$, $\mathfrak{R}_{r3} = \mathfrak{R}_{r9}$, $\mathfrak{R}_{r4} = \mathfrak{R}_{r8}$, $\mathfrak{R}_{r5} = \mathfrak{R}_{r7}$, $\mu_{s2} = \mu_{s10}$, $\mu_{s3} = \mu_{s9}$, $\mu_{s4} = \mu_{s8}$, $\mu_{s5} = \mu_{s7}$. The solution for Φ is given in Equation. (3.6).

$$\Phi = \frac{NI}{\frac{l_{g1}}{\mu_{s1}A_{r1}} + \frac{2l_{g10}}{\mu_{s10}A_{r10}} + \frac{2l_{g9}}{\mu_{s9}A_{r9}} + \frac{2l_{g4}}{\mu_{s4}A_{r4}} + \frac{2l_{g7}}{\mu_{s7}A_{r7}} + \frac{l_{g6}}{\mu_{s6}A_{r6}}} \tag{3.6}$$

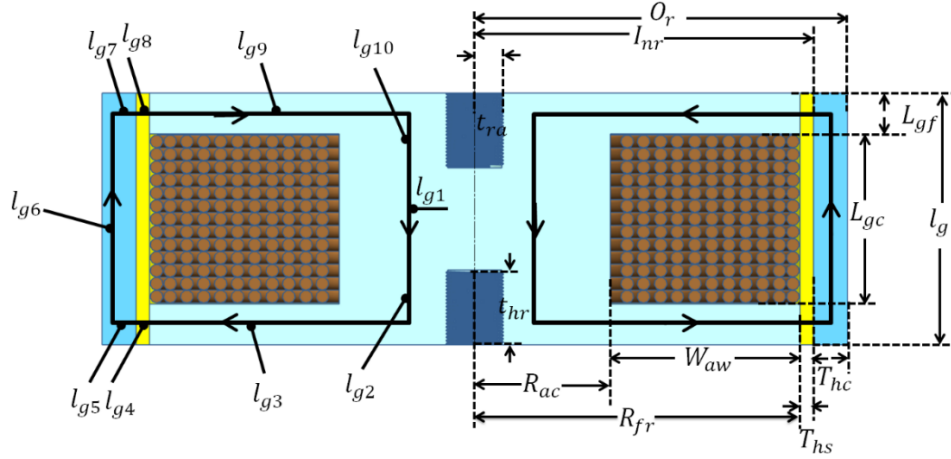


Figure 3.12: Two dimensional cross-sectional view of MR damper magnetic circuit with geometric variables.

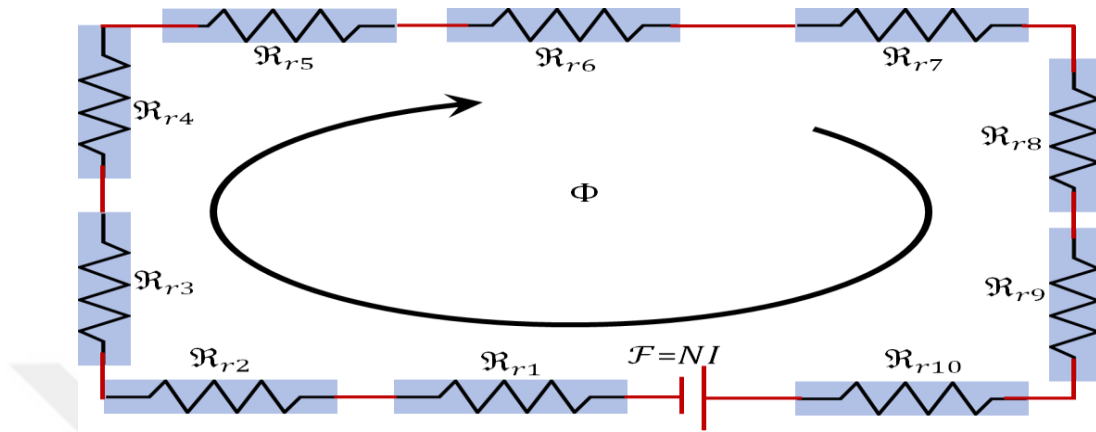


Figure 3.13: Magnetic circuit of MR damper showing all the ten reluctances with magnetic flux flowing throughout the circuit.

The H_{r4} and H_{r8} are the magnetic field intensities at the active poles of the MR damper. All the material permeabilities of steel regions are equal to μ_{lst} and all the fluid permeabilities of active poles regions are equal to μ_{mf} . However, μ_{mf} is less than μ_{lst} and μ_{mf} is negligible. Therefore, the relation of Φ is related to H_{r4} and H_{r8} in Equation (3.7).

$$B_{r4} = B_{r8} = \frac{\Phi}{A_{r4}}$$

$$H_{r4} = H_{r8} = \frac{\Phi}{\mu_{s4}A_{r4}} = \frac{NI}{2l_{g4}} = \frac{NI}{2T_{hs}} \quad (3.7)$$

The shear stress of the MR fluid is defined in Equation (3.8). Where, τ is the shear stress, $\dot{\gamma}$ is the shear strain rate, η is the plastic viscosity and τ_y is the yield stress. The yield stress is the function of H and formulated in Equation (3.7). The relation of yield stress with magnetic field intensity for the commercial MR fluid is shown in Equation (3.9) [141]. Where d_3 , d_2 , d_1 and d_0 are the coefficients which are evaluated using curve fitting method based on least square technique. The area of active poles of MR damper is shown by Equation (3.10). Where, R_{fr} and L_{gf} are active poles radius and active poles length. The damping force of MR damper is shown by Equation (3.11).

$$\tau = \tau_y(H) \operatorname{sgn}(\dot{\gamma}) + \eta \dot{\gamma} \quad (3.8)$$

$$\tau_y = d_3 H^3 + d_2 H^2 + d_1 H + d_o \quad (3.9)$$

$$A_r = 4\pi R_{fr} L_{gf} \quad (3.10)$$

$$F = \tau A_r \quad (3.11)$$

The viscous damper is analyzed in the relation with MR damper by formulating c_{eqv} [139]. The dissipation of energy W_d in a single cycle by the MR damper as shown in Equation (3.12). Where w_{ex} is the frequency of excitation and v is the velocity of MR damper. An excitation of simple harmonic of $x(t) = X_m \sin(w_{ex}t)$, where X_m is the amplitude given as peak to MR damper. The c_{eqv} is the equivalent coefficient of damping as given in Equation (3.13).

$$\begin{aligned} W_d &= \int_0^{2\pi/w_{ex}} F dx = \int_0^{2\pi/w_{ex}} c_{eqv} v dx = \int_0^{2\pi/w_{ex}} c_{eqv} (w_{ex} X_m \cos(w_{ex}t))^2 dt \\ W_d &= c_{eqv} w_{ex}^2 X_m^2 \int_0^{2\pi/w_{ex}} \left(\frac{1 - \cos(2w_{ex}t)}{2} \right) dt \\ W_d &= c_{eqv} w_{ex} X_m^2 \pi \end{aligned} \quad (3.12)$$

$$c_{eqv} = \frac{W_d}{X_m^2 \pi w_{ex}} \quad (3.13)$$

The hysteresis behavior of MR damper is found using various models [146-147]. However, a hysteretic model is used for MR damper and compared its performance with Bouc-Wen model [10]. The hysteretic model offers fine smoothness of curves of hysteresis due to function of tangent hyperbolic which is very efficient when it comes to computation [10]. The model provides less value error and it is also very simple with few number of variables needs to be identified. The hysteretic model comprises elements of dashpot and spring as wells as it consists of hysteresis part as given in Figure 3.14. The damping force of the hysteretic model is shown by Equation (3.14) and (3.15).

$$F = c\dot{x} + kx + \alpha z + f_o \quad (3.14)$$

$$z = \tanh(\beta \dot{x} + \delta \operatorname{sign}(x)) \quad (3.15)$$

The variable c shows the damping of viscous, k represents stiffness, α shows the factor of

scale and hysteresis behavior is modeled using function of tangent hyperbolic and is shown by z . The offset force f_o of MR damper is negligible due to the accumulator absence in the shear mode MR damper.

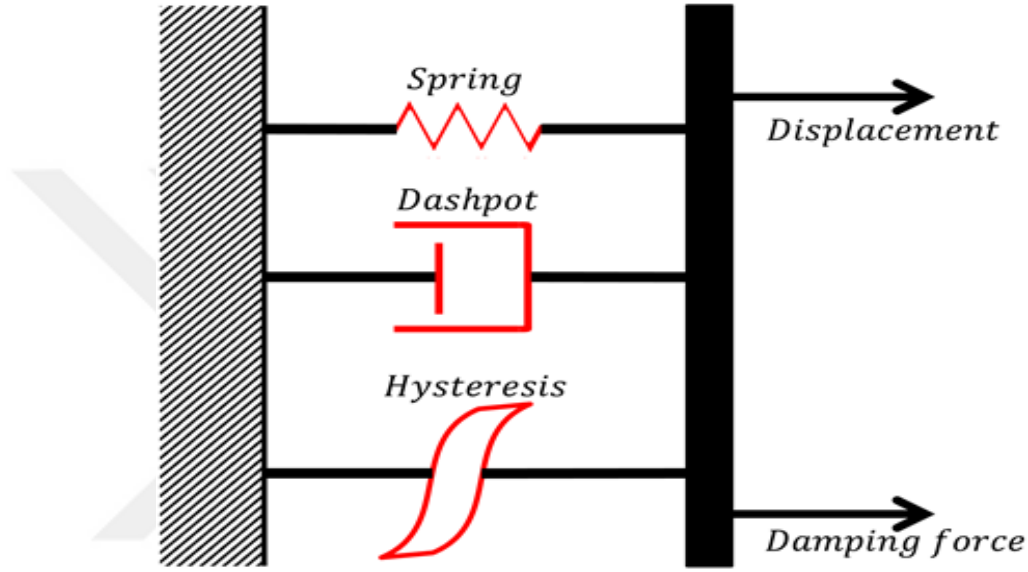


Figure 3.14: Analysis of hysteresis behavior using hysteretic model of MR damper.

3.3 Summary

The chapter discusses in detail the design optimization of shear mode MR damper by using finite element analysis and golden section algorithm techniques. The whole geometry is optimized by using these two methods. The magnetic circuit of MR damper is analyzed using reluctance method. The magnetic flux density is evaluated and relationship of damping force with shear stress and magnetic field intensity is clearly formulated. The hysteresis analysis of MR damper is done using hysteretic model. The model efficiently analyze all the components which are needed for the hysteresis behavior of MR damper. The complete flow chart of the design optimization of MR damper and its further implementation in washing machine for decreasing the vibration is given in Figure 3.15.

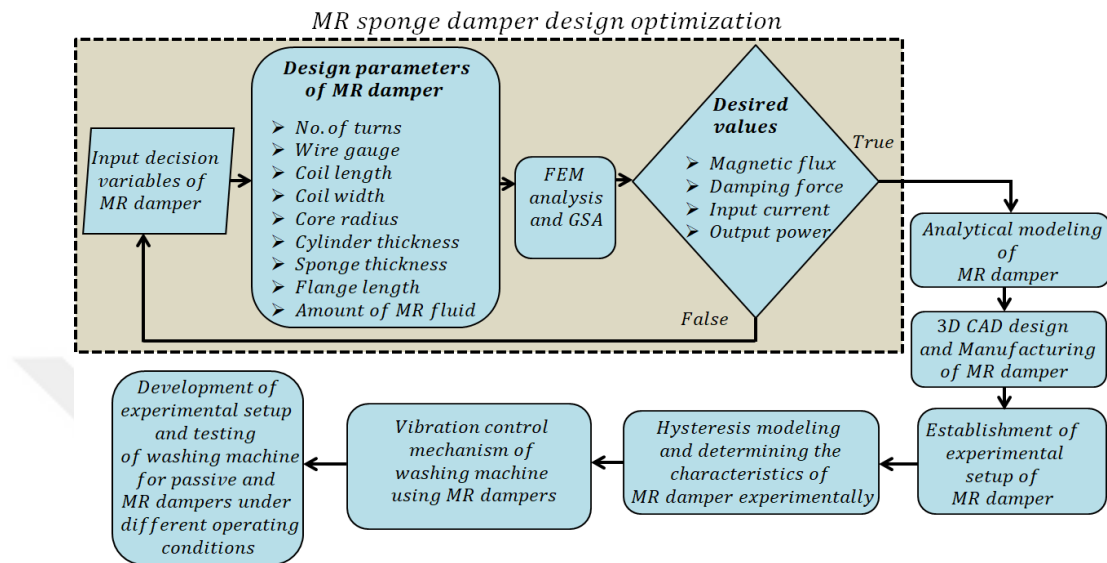


Figure 3.15: Complete flowchart of shear mode MR damper and its implementation in washing machine.

Chapter 4

EXPERIMENTAL TESTRIGS AND RESULTS OF FIRST PROTOTYPE OF MR DAMPER FOR WASHING MACHINE

4.1 Introduction

This chapter deals with the experimental test setups for the first prototype of MR damper. The chapter also focuses on the simulation results and they are validated by the experimental results. The MR damper is investigated for the modeling of hysteresis with changing values of current. For this task, a complete procedure of optimization is followed. The relation of total damping force with electric current is also shown for the experimental data, FEM data and analytical data. The dissipation of energy and equivalent damping coefficient are experimentally evaluated for varying amplitudes and electric current. A vibration control procedure is designed for washing machine using On/Off controller. A detailed comparison of the characteristics of MR damper and reduced vibrations are done with the available literature.

4.2 Experimental setups of MR Damper for Washing Machine

The first test setup for testing MR damper consist of linear actuator having its own controller, a displacement measuring laser, a force sensor having its charge amplifier, and a DC power supply for the purpose of recording displacement and energizing linear actuator. The single coil of MR damper is powered by separate DC supply of power for generating the magnetic flux density. The displacement is recorded by the laser sensor. The force is measured by the dynamometer. The MR damper force is totally dependent on the supply of input current as compared to passive dampers which rely on the velocity of stroke given to it by the linear actuator. The MR damper give a peak force of 220 N with a maximum current of 0.25 A.

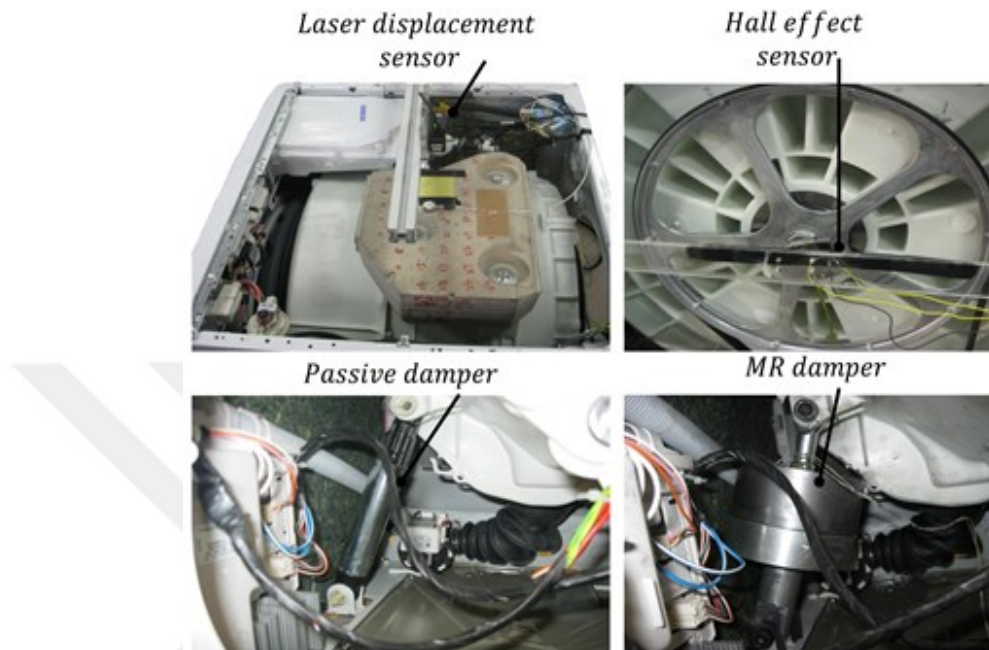


Figure 4.1: Testing setup of 1st prototype of MR damper in washing machine.

On the other hand, another experimental test setup is developed for the comparison of performance of MR damper with passive dampers as shown in Figure 4.1. A washing machine is used in this research which has the tub attached to the side panels of the washing machine through two dampers and two springs. The tub has drum, laundry, water and a motor attached with the drum through pulley. The drum of the tub rotates in counter clock wise and clock wise direction at angular velocity of 1400 RPMs. The passive dampers has 100 N of damping force at maximum. First the passive dampers are studied and then they are replaced with two MR dampers in the washing machine. A power supply is used for supplying current to both MR dampers. A laser sensor is attached to the beam and the beam is further fixed with optic table to measure the displacement. The RPMs are measured using hall-effect sensor.

4.3 Results and Discussion

The hysteretic model is selected for studying the hysteresis behavior of MR damper. The model is used to approximate the hysteresis curves of force with respect to velocity and displacement for changing values of input electric current. For this task, an optimization procedure is developed for the minimization of error between the experimental and simulated result as given in Figure 4.2. The experimental data of force, velocity along with displacement are measured and loaded for different values of input current. The data is recorded for the commanded amplitudes at different frequencies. All the data is taken at specific commanded amplitude and frequency with a particular value of current. The optimization is started by loading the parameters of hysteretic model with some arbitrary values. The optimization is stopped by finding the optimized values of parameters and minimum error of the objection function O . The resultant hysteresis curves for both the velocity and displacement in relation with force for varying electric current are presented in Figure 4.3 and Figure 4.4.

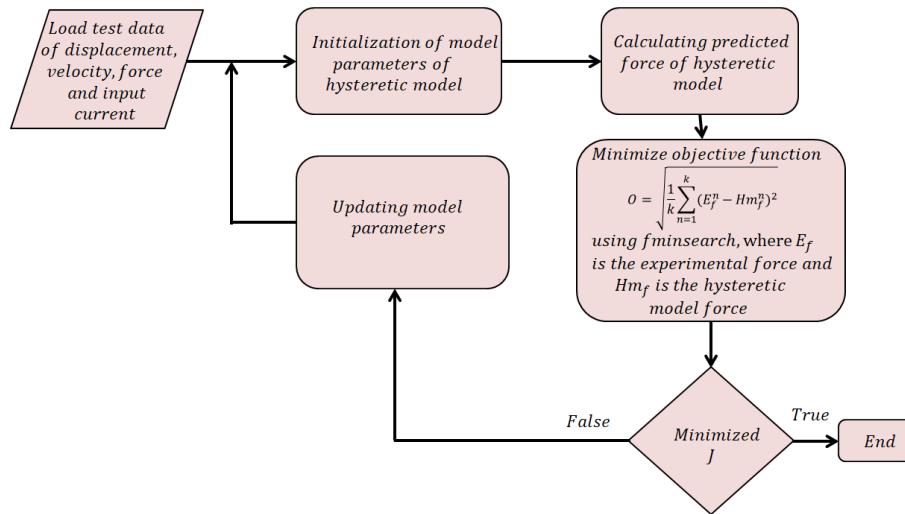


Figure 4.2: Optimization strategy for identifying the hysteresis behavior of MR damper.

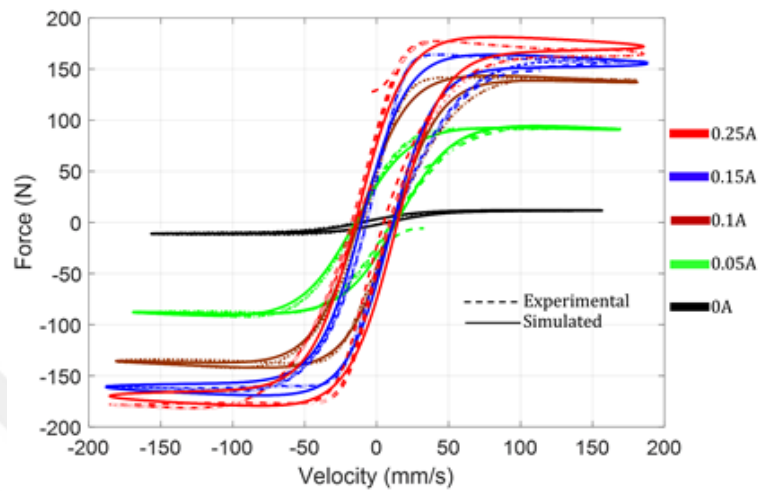


Figure 4.3: Hysteresis curves for force with respect to velocity for various current values.

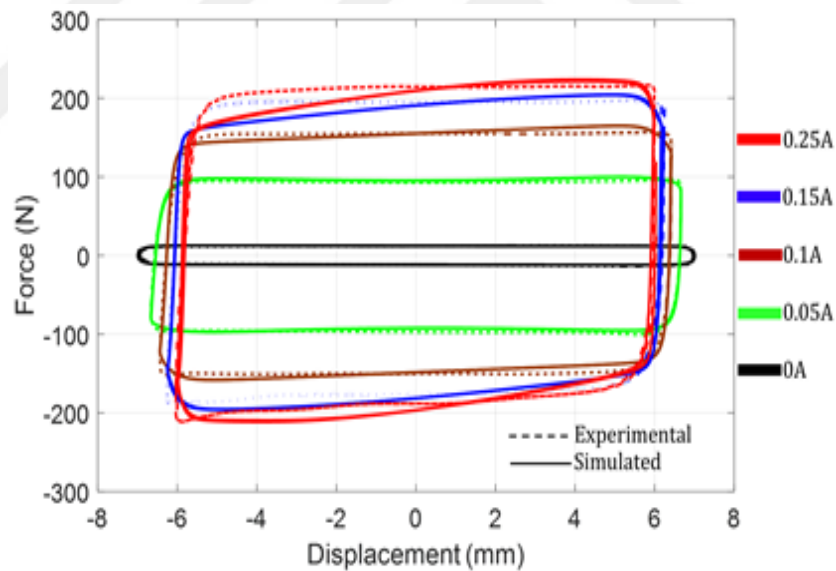


Figure 4.4: Hysteresis curves for force with respect to displacement for various current values.

The behavior of total damping with respect to the value of electric current is investigated by simulating the mathematical model of force of MR damper in Matlab. First by using Equation (3.7), at active poles the magnetic field is evaluated by changing the value of electric current. That value of magnetic field intensity is used in Equation (3.8) to calculate the shear stress. After that, the value of shear stress is utilized in Equation (3.11) along with surface area in order to find the MR damping force. On the other hand, the damping force is experimentally determined with respect to electric current by exciting the MR damper with a linear actuator. Besides this, the damping force is also found by using FEM model as discussed in chapter 3. The relation of MR damping with current for mathematical mode, FEM model and experimental data is given in Figure 4.5.

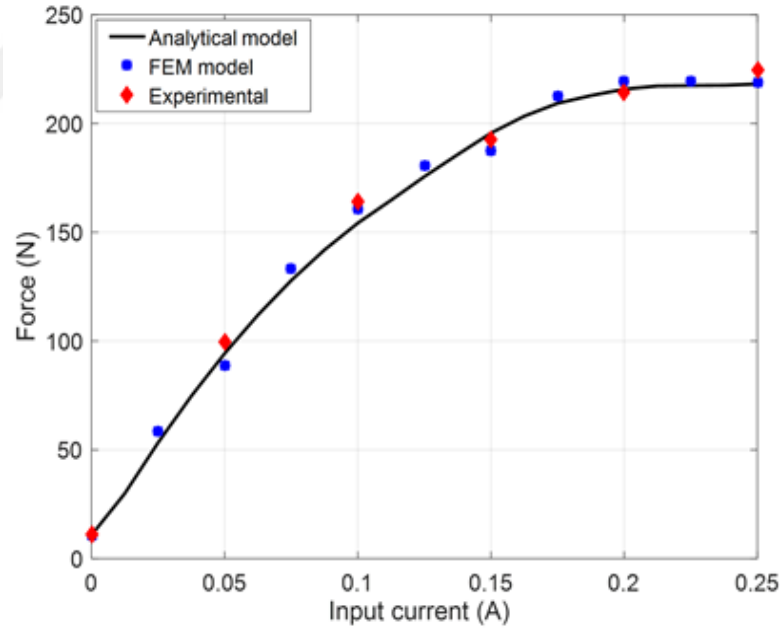


Figure 4.5: Behavior of MR damping force with respect to input current.

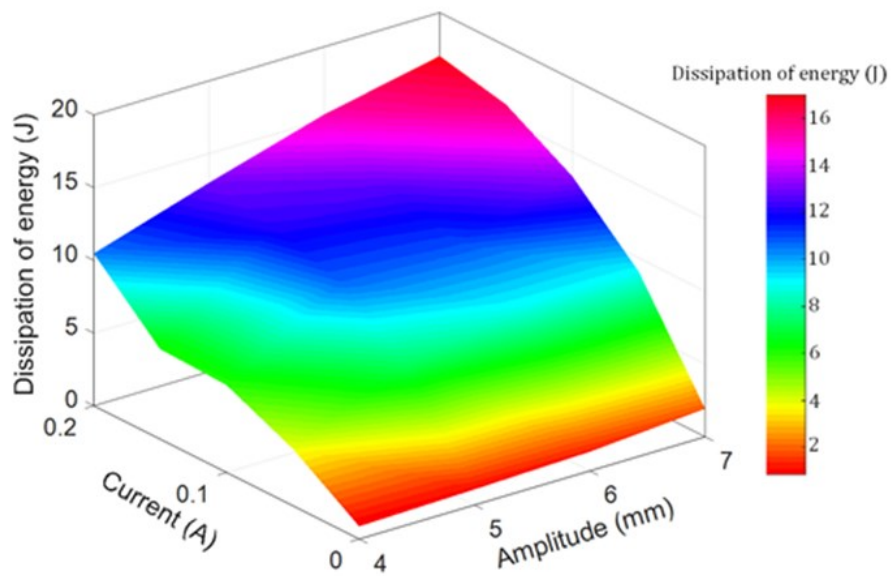


Figure 4.6: Behavior of dissipation of energy versus current for various values of amplitude.

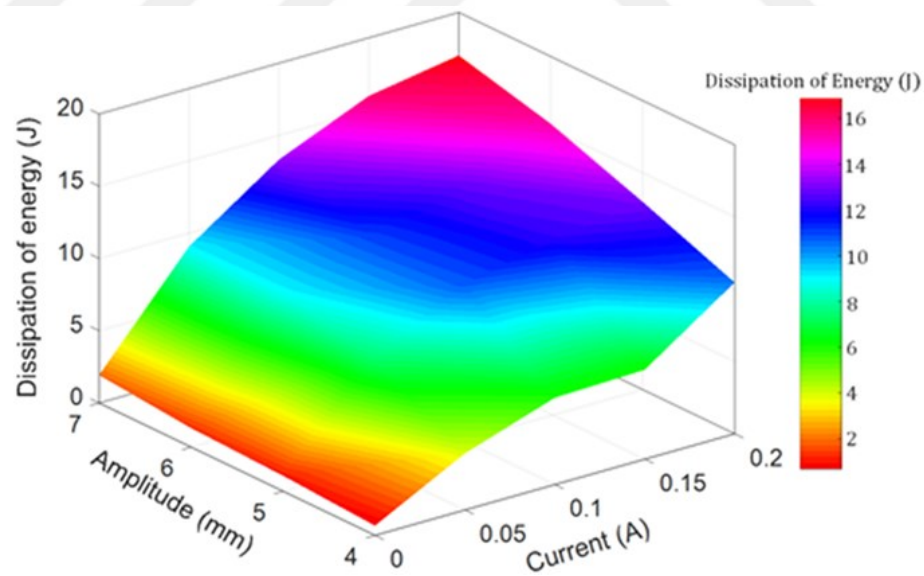


Figure 4.7: Behavior of dissipation of energy versus amplitude for various values of current.

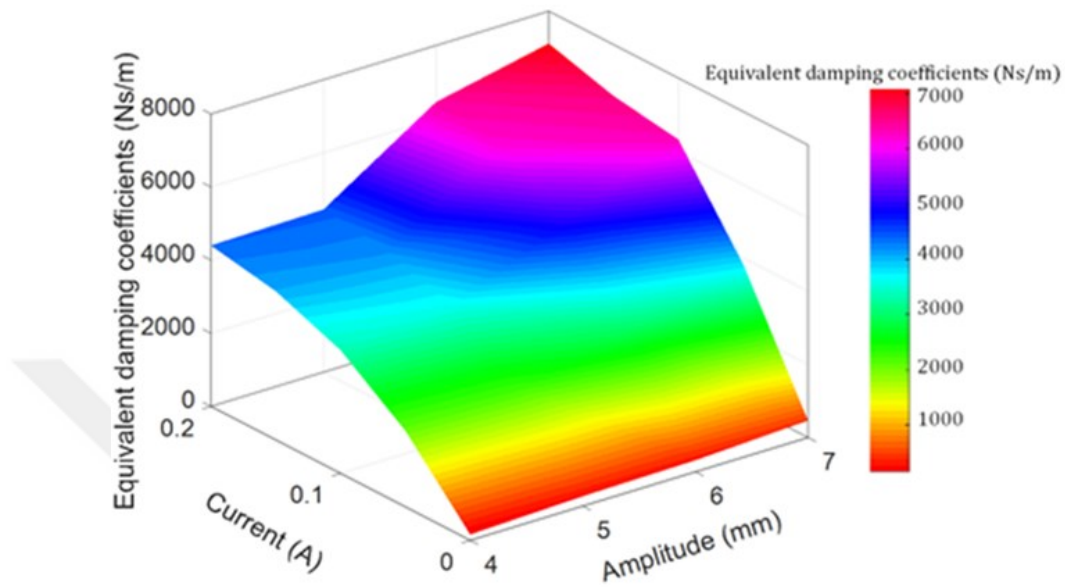


Figure 4.8: Behavior of equivalent damping coefficient versus current for various values of amplitude.

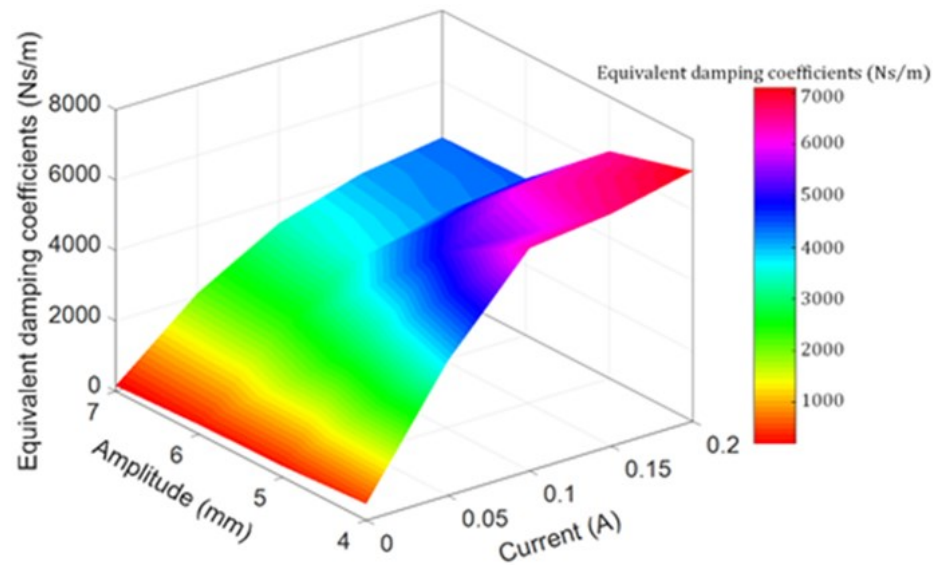


Figure 4.9: Behavior of equivalent damping coefficient versus amplitude for various values of current.

The dissipation of energy W_d and coefficient of equivalent damping c_{eqv} of MR damper are studied for different values of input electric current supplied by the DC power supply and for different values of commanded amplitudes. The frequency of commanded signal is set fixed during the operation. The input current to the MR damper is varied from 0 to 0.2 A. The W_d of MR damper during a cycle is found from the area of force and displacement hysteresis graph. In Figure 4.6 and Figure 4.7, it is seen that dissipation of energy has increasing behavior with respect to different electric current values and commanded amplitude values. This energy dissipation increases due to high value of electric current. The coefficient of equivalent damping shows increasing behavior with respect to electric current for different amplitude values as shown in Figure 4.8. This trend is more visible at low commanded amplitude values. The reason behind is this that magnitude of velocity is less at certain frequency for very low value of displacement. However, the c_{eqv} decreases with respect to commanded amplitude values for changing values of electric current as given in Figure 4.9. This decreasing behavior is more clearer at very high electric current values. Moreover, this behavior is due to c_{eqv} , which has inverse relationship with commanded amplitude signal.

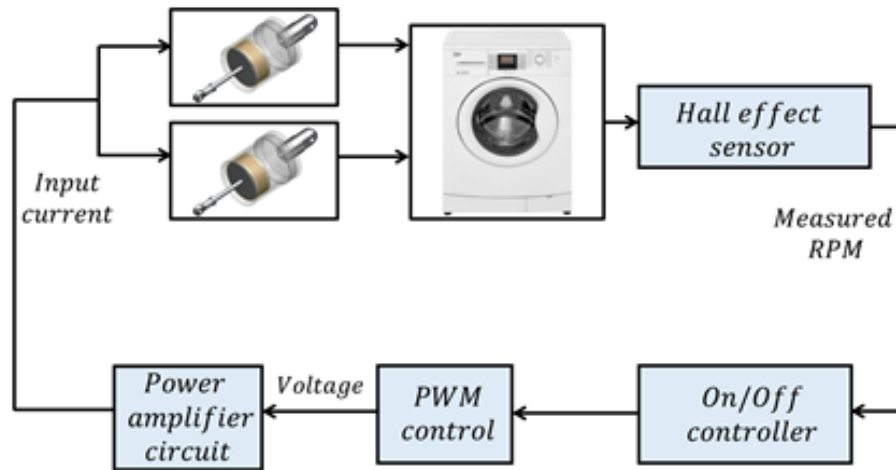


Figure 4.10: Complete mechanism of control of vibrations using MR dampers.

Figure 4.10 presents a complete strategy for vibration control of washing machine. All the data acquisition is done through DAQ cards. The frequency of sampling is set to 10 kHz. The hall-effect is used to measure the angular velocity. The On/Off control which is implemented in Matlab takes the decision based on the recorded value of angular velocity. A circuit of power amplifier is designed and driven by PWM signal. The testing of washing machine is performed in its spinning mode using laundry masses ranging from 14 kg to 20 kg with a step of 2 kg. The washing cycle is selected with a maximum value of 1400 RPM. The water and clothes in the drum define the unbalanced mass. The clothes are stuck to drum surface when it spins in the range of 80 to 200 RPM. The displacement of vibrations are measured using laser sensor for MR and passive dampers of a total duration of 400 seconds. It is seen that severe vibrations are observed at 20 kg. Figure 4.11 presents the displacement of vibrations at 20 kg for MR and passive dampers. The RPM which ranges from the values of 200 to 400 represents the resonance band of low frequency. The passive dampers are inefficient in absorbing the vibrations. Therefore, the MR damper generates 220 N force each at 0.25 A to decrease the vibrations. The MR dampers absorbs 75.61 % average vibrations comparing to passive dampers. On the other side, after low resonance the vibrations are transmitted to the top and side panels of washing machine. This increases the transmissibility of washing machine. The RPM range in this region consist of values from 400 to 1400 RPM. For this purpose, the MR dampers utilizes their passive force which is less than the passive dampers. The reason behind is this that a very low force of 10.76 N is required which the passive dampers fall short of providing it. In this particular region of washing cycle, the MR dampers minimizes the 30.57 % average vibrations with no electric current provided by the DC power supply.

The MR dampers are compared in terms of their performance with other existing designs of MR dampers for washing machine as discussed in [13] and [136-145]. The sponge based MR dampers are designed for washing machine [144-145]. In [137-141], the fluid based MR

dampers are developed for washing machine. The commercial MR dampers such as RD-1097-01 and RD-1005-3 are used inside washing machine [140,145]. A batch optimization technique is used as a control technique for MR dampers [36]. But, the vibration data is presented as normalized index. In [139] and [141], the vibration data is not presented. Therefore, the performance comparison of MR dampers is given in Table 4.1 with other designs available in literature [136-138], [13] and [143-144]. At full loading of washing machine for example at 20 kg, a maximum of 220 N force F_{total} is applied. This force is high in value as compared to other designs [136-145]. On the other side, the MR damper passive force F_{pas} is 10.76 N. This value of force is less than other designs except [143-144] which has 7 N force. Moreover, a force of 8 N is recorded of three coil MR damper [139]. But as the coil number increases the passive force reduces with increasing length of stroke of MR damper [139]. The power and current ratings of MR damper are lowered than [136-138] and [13,143-144]. But in [141], the power dissipation is 1.9 W and current usage is 2.5 A. In three coil MR damper mentioned in [139], the current rating is 1.5 A and power consumption is 16 W. The acceleration Acc of average vibrations using MR dampers is 0.916 m/s^2 for full cycle which is less than other designs [137-138] and [144]. On the other side, the MR dampers reduce 0.807 mm displacement Ds which is the average vibrations recorded during the washing cycle. This value is also lowered than the design presented in [13].

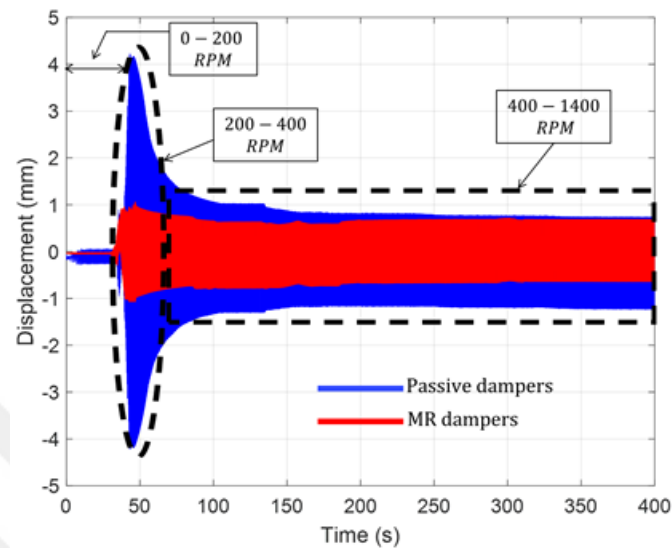


Figure 4.11: Vibration comparison of MR dampers with passive dampers.

Table 4.1: MR damper comparison in terms of performance with other designs of MR damper in the literature.

Parameters	MR damper	[136] & [137]	[138]	[13]	[143] & [145]
F_{total} (N)	220	150	80	150	125
F_{pas} (N)	10.76	65	23.5	12.5	10
I_w (A)	0.25	3	0.4	2	2.5
P_w (W)	17.25	35	-	40	24.375
Ds (mm)	0.807	-	-	3.07*	-
Acc (g)	0.916	2.790*	1.25*	0.25*	0.933*
* Evaluation of data is done using the results which are provided					

4.4 Summary

This chapter explains the experimental setups of MR damper and their testing inside

washing machine. A detail hysteresis analysis of MR damper is carried out for changing input currents using the hysteretic model. The trend of damping force with respect to input current is determined experimentally and it is validated by FEM and analytical models. The MR damper dissipation of energy and coefficient of equivalent damping is shown experimentally. A strategy of control for vibration minimization is developed for MR dampers testing in washing machine and its comparison with passive dampers. It is shown that 1st prototype of MR damper significantly decreases the vibrations. Moreover, the characteristics of 1st prototype of MR damper are better in terms of its performance as compared to other designs in literature.

Chapter 5

DESIGN AND CONTROL OF SECOND PROTOTYPE OF MR DAMPER FOR WASHING MACHINE

5.1 Introduction

This chapter deals with the design optimization of MR damper and its control in washing machine. The design of 1st prototype is improved by introducing double coil configuration for the MR damper. This configuration helps in reducing the size of the MR damper while keeping the force value at 200 N. The optimization is performed in the same manner as it is done for the 1st prototype. Besides this, a detailed control strategy is explained for the MR damper. A sliding mode control is developed for the current control of MR damper. The terminal sliding mode control (TSMC) is used which is a nonlinear control algorithm for the control of current through MR dampers. The converging parameter is derived in terms of the parameters of MR damper in order to make the parameter adaptive. Stability analysis is investigated using the Lyapunov theory. A complete simulation study is performed for the control model and MR damper.

5.2 Design

The magnetic circuit of 2nd prototype of MR damper is investigated using finite element analysis. It is essential to optimize the design of magnetic circuit for the targeted value of magnetic flux density and force. For this task, a two dimensional axis symmetric finite element analysis mode is built for the 2nd prototype of MR damper. The magnetic flux is distributed throughout the model as given in Figure 5.1. The 3D equivalent FEM model is shown in Figure 5.2. It is important to mention that during finite element analysis, different parts of the model are assigned different materials. The cylinder and piston are assigned steel

parts, the winding part is assigned with copper material and the sponge part is assigned with MR fluid material designed in finite element analysis software.

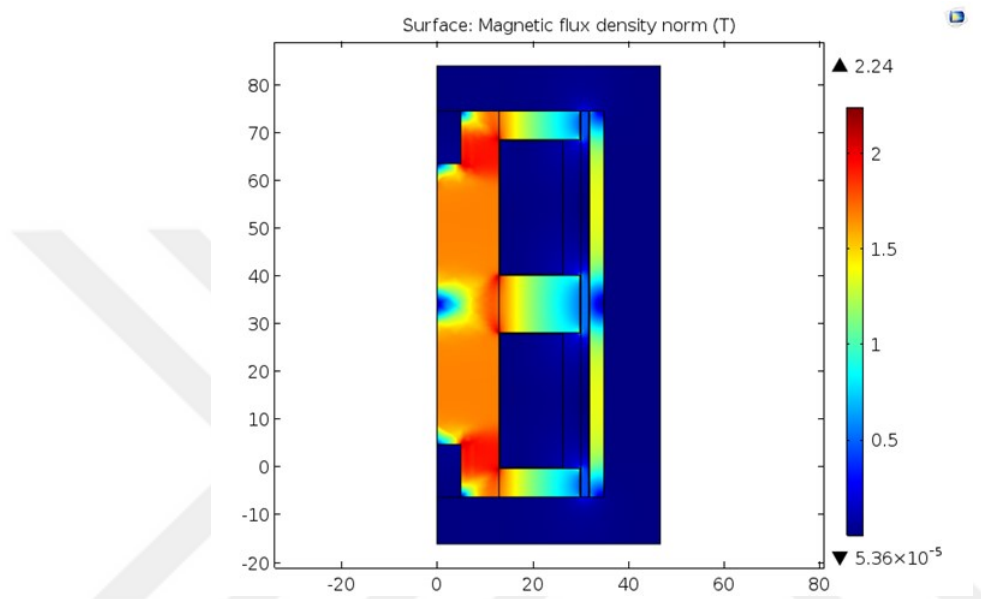


Figure 5.1: Two dimensional axis symmetric model of magnetic circuit of MR damper.

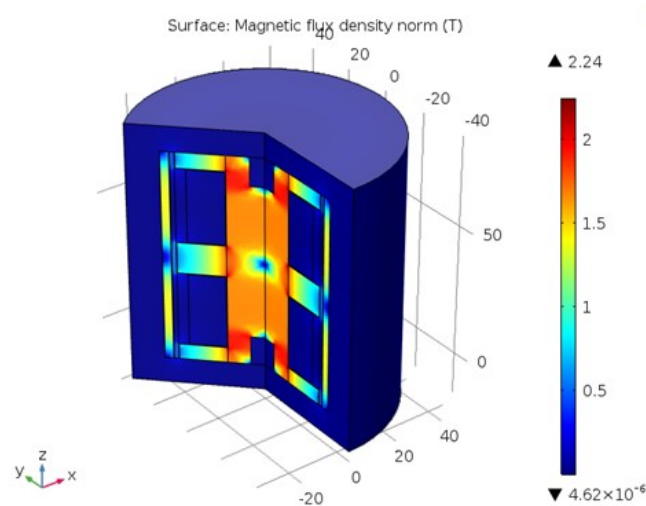


Figure 5.2: Three dimensional equivalent FEM model of magnetic circuit of MR damper.

The geometric variables of MR damper such as total length, outer diameter, length of stroke are fixed to a particular value due to space provided in washing machine for the MR dampers. A wire gauge is chosen so that it can bear 0.5 A electric current. A parametric sweep is executed for all the geometric variables. The number of turns are chosen in the range of 1000 to 3500 turns and a parametric sweep is performed on it with a fixed current of 0.5A. It is seen that the magnetic flux increases but it saturates after 3000 turns as given in Figure 5.3.

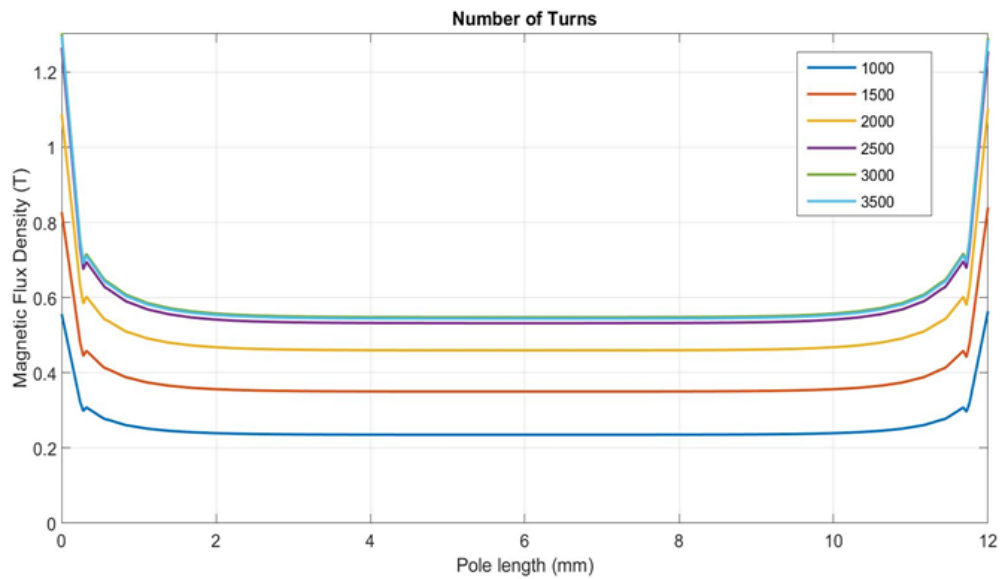


Figure 5.3: Trend of magnetic flux density with respect to active pole for changing values of number of turns.

The magnetic flux density is investigated between 10 to 15 mm for different value of core radius with respect to the length of active pole as shown in Figure 5.4. It is seen that the flux increases as the core radius increases. However, due to the limitation of outer diameter of MR damper, 13 mm is selected for the core radius which satisfies the targeted value of magnetic flux density and also it is suitable for the overall diameter and size of MR damper. On the other side, a parametric sweep is implemented for the cylindrical thickness of MR damper between 1 to 4 mm. The flux doesn't show any increasing trend after 3 mm as given

in Figure 5.5. However, 2 mm also shows good value of magnetic flux, but it is very difficult to select as thickness for cylinder from the manufacturing point of view. A complete geometry of MR damper after the optimization is given in Table 5.1

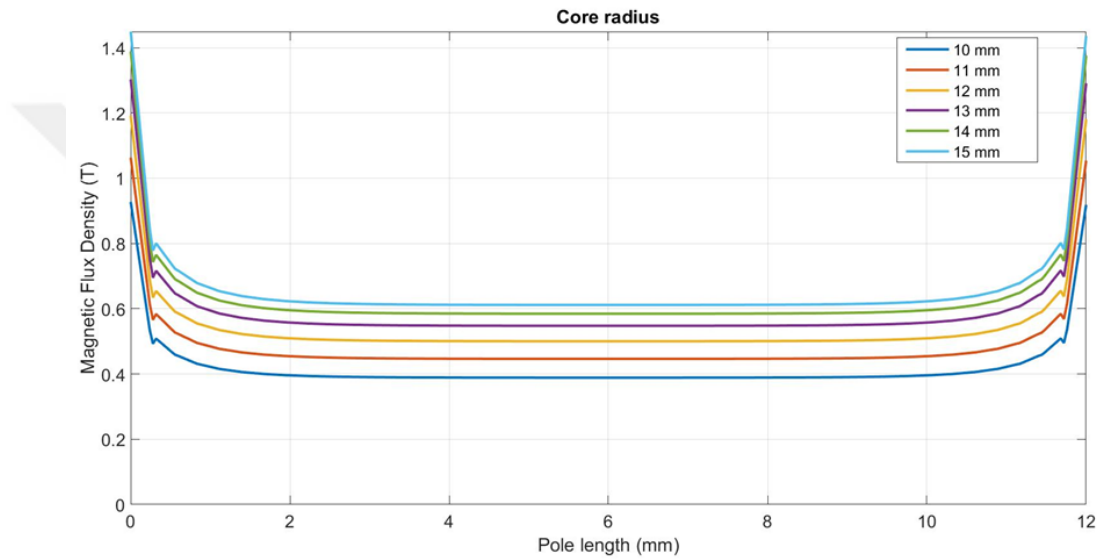


Figure 5.4: Trend of magnetic flux density with respect to active pole for changing values of core radius.

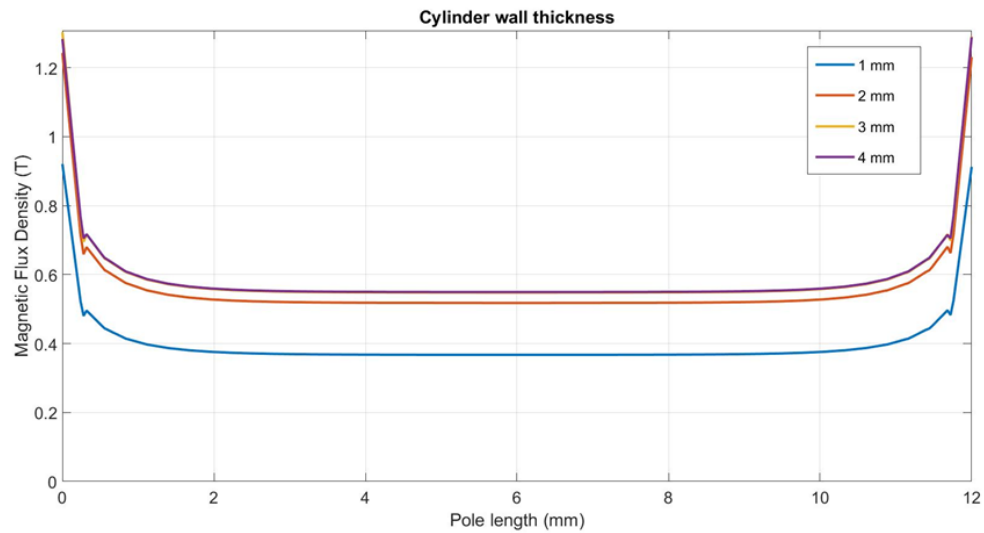


Figure 5.5: Trend of magnetic flux density with respect to active pole for changing values of cylinder thickness.

Table 5.1: Optimized parameters of Double Coil MR damper

Parameter	Value
Piston Diameter (mm)	59.8
Outer Diameter (mm)	69.8
Cylinder Wall Thickness (mm)	3
Stroke (mm)	± 22.5

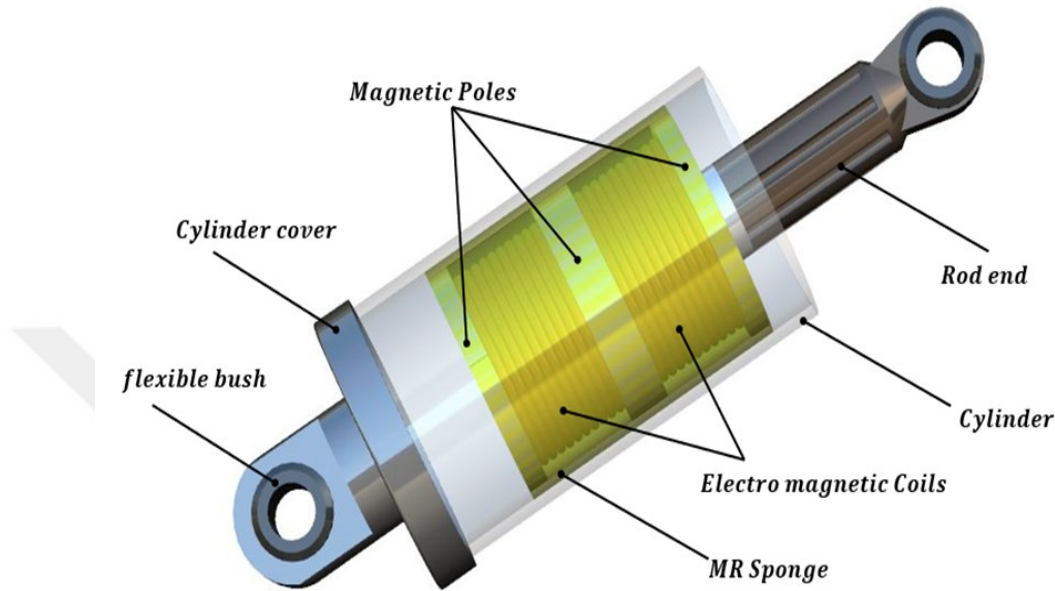


Figure 5.6: Three dimensional CAD model of 2nd prototype of MR damper.

The mechanical design of 2nd MR damper consists of piston which has double coil configuration, rod of the piston, a cylinder for the return path of magnetic flux, a cover of cylinder and bushes as shown in Figure 5.6. A two dimensional cross section view of 2nd prototype of MR damper is shown in Figure 5.7. The geometric variables are clearly shown in the figure which are optimized by using finite element analysis. The material of AISI 1010 which is a low carbon steel, this material is used for the manufacturing of the cylindrical and piston of MR damper. The length of the middle pole is kept twice as compared to the outside poles in order to enhance the flux at the middle pole. Then each winding in double coil is done in opposite directions. This is due to the fact for maximizing flux at the center pole of the MR damper. The cylinder cover is designed in such a way to take the heat out which is built sometimes due to the friction between the cylindrical part of the MR damper and its piston.

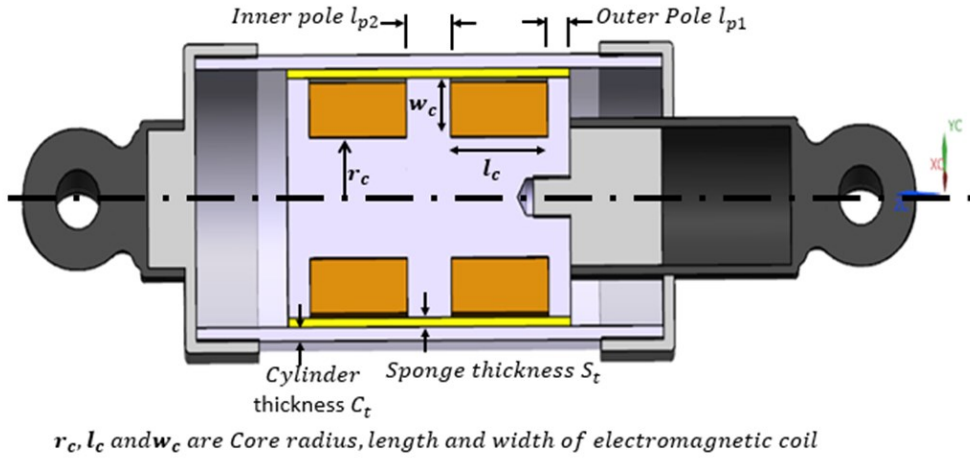


Figure 5.7: A two dimensional cross section view of 2nd prototype of MR damper showing its geometric variables.

5.3 Adaptive Control of MR Damper

The power converter such as buck converter is used with MR damper as given in Figure 5.8. It comprises of V_{pi} which is an input voltage, U_{is} is control input for the MOSFET M_o , D_o is the diode. L_{bc} is the inductor of converter and C_{bc} is the capacitance. The MR damper is modeled as R_{co} and L_{co} of a double coil winding. Equations (5.1)-(5.3) explains the power converter operation, Where I_{id} is current of the inductor, I_{pa} is capacitor current, I_{pt} and V_{pt} are the current and voltage of MR damper. The error between desired and output values of electric current is given in Equation (5.4).

$$\dot{I}_{id} = [U_{is}V_{pi} - V_{pt}]/L_{bc} \quad (5.1)$$

$$\dot{V}_{pt} = [I_{id} - I_{pt}]/C_{bc} \quad (5.2)$$

$$\dot{I}_{pt} = [V_{pt} - R_{co}I_{pt}]/L_{co} \quad (5.3)$$

$$e_r = I_{pt} - I_{fd}; \dot{e}_r = \dot{I}_{pt}; \ddot{e}_r = \ddot{I}_{pt} \quad (5.4)$$

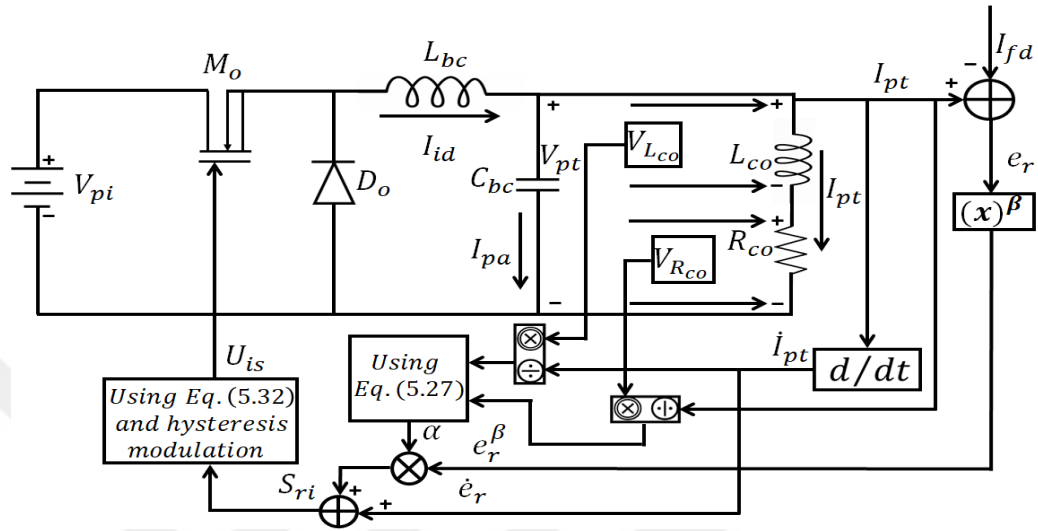


Figure 5.8: A complete block diagram for the adaptive control of current through MR dampers.

A sliding surface is shown in Equation (5.5), where α represents the slope as given by Equation (5.6), when $S_{ni} = 0$. The $\alpha > 0$ will results the system into stable system, otherwise the system remains unstable when $\alpha < 0$. Equation (5.7) is determined by using the Equations (5.1)-(5.6). Besides this, $b_{co} = (R_{co} - \alpha L_{co})$ and $w_{yr}^2 = 1/L_{bc}C_{bc}$.

$$S_{ni} = \dot{e}_r + \alpha e_r; \quad (5.5)$$

$$\dot{e}_r = -\alpha e_r; \quad (5.6)$$

$$\ddot{e}_r = w_{yr}^2 [U_{is}V_{pi} - (R_{co}I_{fd} + b_{co}e_r) - L_{bc}\dot{e}_r]/b_{co} \quad (5.7)$$

A modulation using hysteresis is used to keep the frequency of switching in the certain limits as given by Equation (5.8). When $S_{ni} < -h_{co}$, the U_{is} is high but $S_{ni} > h_{co}$, the U_{is} is low. The condition of existence is shown in in Equation (5.9), which give direction to the states of the system, so that the states come to the line of sliding. Equation (5.10) explains the derivative of Equation (5.5).

$$U_{is} = \begin{cases} 1 & \text{if } S_{ni} < -h_{co} \\ 0 & \text{if } S_{ni} > h_{co} \end{cases} \quad (5.8)$$

$$S_{ni}\dot{S}_{ni} < 0 \quad (5.9)$$

$$\dot{S}_{ni} = \ddot{e}_r + \alpha \dot{e}_r \quad (5.10)$$

After using Equation (5.1)-(5.7), for $S_{ni} < 0$ and $U_{is} = 1$ the inequality is shown in Equation (5.11). On other hand, for $S_{ni} > 0$ and $U_{is} = 0$ the inequality is given in Equation (5.12).

$$q_{s1} = (\alpha - 1/b_{co}C_{bc})\dot{e}_r + w_{yr}^2(V_{pi}/b_{co} - e_r - R_{co}I_{fd}/b_{co}) > 0 \quad (5.11)$$

$$q_{s2} = (\alpha - 1/b_{co}C_{bc})\dot{e}_r + w_{yr}^2(-e_r - R_{co}I_{fd}/b_{co}) < 0 \quad (5.12)$$

The surface of sliding S_{ni} divides the whole phase plane in two main zones. The lines are realized in the plane by taking q_{s1} and q_{s2} equal to zero which crosses the points $(V_{pi} - R_{co}I_{fd}/b_{co}, 0)$ and $(R_{co}I_{fd}/b_{co}, 0)$ with the slopes given by Equation (5.13). The α is given by Equation (5.14) which is the coefficient of sliding and it is represented in terms of the physical parameters of whole system.

$$j_{s1} = j_{s2} = w_{yr}^2/[\alpha - 1/b_{co}C_{bc}] \quad (5.13)$$

$$\alpha = \left[R_{co}/L_{co} \pm \sqrt{(R_{co}/L_{co})^2 - 4/(L_{co}C_{bc})} \right] / 2 \quad (5.14)$$

The α generates two real values α_1 and α_2 as given by Equation (5.15)-(5.16), where $\alpha_1 \gg \alpha_2$. So the value of α_1 is chosen for the α . The regions of the existence for the controller are dependent on $\alpha < \alpha_1$ and $\alpha > \alpha_1$. The intercepts of the line for the greater value of α are given by Equation (5.17)-(5.18).

$$\alpha_1 = \left[R_{co}/L_{co} + \sqrt{(R_{co}/L_{co})^2 - 4/(L_{co}C_{bc})} \right] / 2 \quad (5.15)$$

$$\alpha_2 = \left[R_{co}/L_{co} - \sqrt{(R_{co}/L_{co})^2 - 4/(L_{co}C_{bc})} \right] / 2 \quad (5.16)$$

$$z_{s1} = [w_{yr}^2(R_{co}I_{fd} - V_{pi})]/[b_{co}\alpha - 1/C_{bc}] \quad (5.17)$$

$$z_{s2} = [w_{yr}^2(R_{co}I_{fd})]/[b_{co}\alpha - 1/C_{bc}] \quad (5.18)$$

The intercepts of the line for the lower value of α are shown by Equation (5.19)-(5.20). The response of the system is made faster by using high value of α . But there is main obstacle comes in the form of time convergence. To solve this problem, another nonlinear TSMSC surface S_{ri} is selected as given by Equation (5.21). The value of β should be between 0 and 1. The time derivative of Equation (5.21) is given in Equation (5.22). After taking $S_{ri} = 0$, Equation (5.23) is generated in which e_r is the equilibrium point. The time convergence is given by Equation (5.24), which is derived by using integral of Equation (5.23).

$$z_{s1} = [w_{yr}^2(R_{co}I_{fd})]/[b_{co}\alpha - 1/C_{bc}] \quad (5.19)$$

$$z_{s2} = [w_{yr}^2(R_{co}I_{fd} - V_{pi})]/[b_{co}\alpha - 1/C_{bc}] \quad (5.20)$$

$$S_{ri} = \dot{e}_r + \alpha e_r^\beta \quad (5.21)$$

$$\dot{S}_{ri} = \ddot{e}_r + \beta\alpha\dot{e}_r e_r^{\beta-1} \quad (5.22)$$

$$\dot{e}_r = -\alpha e_r^\beta \quad (5.23)$$

$$t_{icg} = [|e_r(0)|^{1-\beta}]/[\alpha(1-\beta)] \quad (5.24)$$

In order to use the dynamic value of α , the R_{co} and L_{co} would be estimated to adjust the line of sliding in adaptive fashion as shown by Equation (5.25)-(5.26). Where, I_{pt} and $\dot{I}_{pt}$ are current and its derivative and $V_{R_{co}}$ and $V_{L_{co}}$ are the voltages across R_{co} and L_{co} respectively. The adaptive α is expressed in Equation (5.27).

$$\hat{R}_{co} = V_{R_{co}}/I_{pt} \quad (5.25)$$

$$\hat{L}_{co} = V_{L_{co}}/\dot{I}_{pt} \quad (5.26)$$

$$\alpha = \left[\hat{R}_{co}/\hat{L}_{co} + \sqrt{(\hat{R}_{co}/\hat{L}_{co})^2 - 4/(\hat{L}_{co}C_{bc})} \right] / 2 \quad (5.27)$$

The existence condition of TSMC is given by Equation (5.28). On the other side, a Lyapunov based function and its derivative are used as given by Equation (5.29)-(5.30) in order to satisfy Equation (5.28). These Lyapunov functions derive the equivalent

control input as given in Equation (5.31). In addition to this, a control of switching is also used to direct the trajectory to the sliding mode. The complete control input U_{is} is given by Equation (5.32). Where U_{si} is the gain of controller.

$$S_{ri}\dot{S}_{ri} < 0 \quad (5.28)$$

$$L_{pu}(t) = S_{ri}^2/2 \quad (5.29)$$

$$\dot{L}_{pu}(t) = S_{ri}(\ddot{e}_r + \beta\alpha\dot{e}_r e_r^{\beta-1}) < 0 \quad (5.30)$$

$$U_{cv} = [R_{co}I_{fd} + b_{co}e_r + L_{bc}\dot{e}_r - b_{co}L_{bc}C_{bc}\beta\alpha\dot{e}_r e_r^{\beta-1}]/V_{pi} \quad (5.31)$$

$$U_{is} = [R_{co}I_{fd} + b_{co}e_r + L_{bc}\dot{e}_r - U_{si}\text{sgn}(S_{ri}) - b_{co}L_{bc}C_{bc}\beta\alpha\dot{e}_r e_r^{\beta-1}]/V_{pi} \quad (5.32)$$

Due to the variations of parameters and other unmodeled dynamics, $d_{is}(t)$ is integrated in Equation (5.7), which results into Equation (5.33). By putting Equation (5.32) into Equation (5.33) for U_{is} , it gives Equation (5.34).

$$\ddot{e}_r = [w_{yr}^2 [U_{is}V_{pi} - R_{co}I_{fd} - b_{co}e_r - L_{bc}\dot{e}_r]/b_{co}] + d_{is}(t) \quad (5.33)$$

$$\ddot{e}_r = -\beta\alpha\dot{e}_r e_r^{\beta-1} - U_{si}\text{sgn}(S_{ri}) + d_{is}(t) \quad (5.34)$$

For the stability analysis, Equation (5.30) is presented in Equation (5.35) by using Equation (5.34), where $U_{si} > d_{is}(t)$. On the other hand, $\dot{L}_{pu}(t) < 0$ gives the system stability for $S_{ri} > 0$ or $S_{ri} < 0$.

$$\dot{L}_{pu}(t) = S_{ri}d_{is}(t) - S_{ri}U_{si}\text{sgn}(S_{ri}) < 0 \quad (5.35)$$

The frequency of switching of power converter for MR damper is very important for its operation. The t_{oint} and t_{oift} are the periods of time as given by Equation (5.36)-(5.37).

On the other hand, $\dot{S}_{ri}^{oint}$ and $\dot{S}_{ri}^{oift}$ are the derivatives of time during t_{oint} and t_{oift} . Equation (5.38)-(5.39) shows the steady state of time periods. After using Equation (5.38)-(5.39), the frequency of switching f_{sit} is found in Equation (5.40).

$$t_{oint} = 2h_{co}/\dot{S}_{ri}^{oint} \quad (5.36)$$

$$t_{oift} = -2h_{co}/\dot{S}_{ri}^{oift} \quad (5.37)$$

$$t_{oint} = b_{co}2h_{co}/[w_{yr}^2(V_{pi} - R_{co}I_{fd})] \quad (5.38)$$

$$t_{oift} = b_{co}2h_{co}/[w_{yr}^2(R_{co}I_{fd})] \quad (5.39)$$

$$f_{sit} = (w_{yr}^2 R_{co} I_{fd} / b_{co} 2 h_{co}) [1 - R_{co} I_{fd} / V_{pi}] \quad (5.40)$$

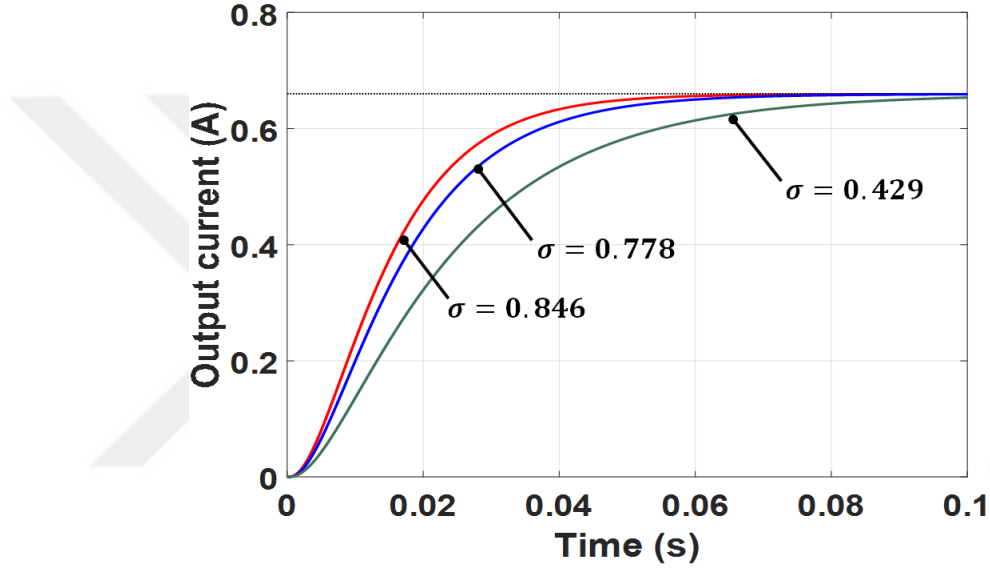


Figure 5.9: Electric current response of MR damper for various values of α .

5.4 Control Simulation Results

The simulation of the proposed controller is performed by taking the design parameters such that $I_{fd} = 0.66$ A, $C_{bc} = 500$ μ F, $L_{bc} = 10$ mH, $L_{co} = 1.06$ H, $R_{co} = 182$ Ω , $V_{pi} = 120$ V, $h_{co} = 100$ and $\alpha = 159.43$. The responses of the MR damper current along with the voltage for various values of β are shown in Figure (5.9) and Figure (5.10). The controller also tracks 0.165 A and 0.495 A step source currents at $t = 0.3$ ms as given in Figure (5.11) and Figure (5.12). The trajectories of electric current versus its derivatives for various value of β is given in Figure (5.13). The $\beta = 0.846$ shows highest slope while comparing

to another values of β . A $d_{is}(t)$ is the disturbance of 50 mA, which is injected into the system at $t = 0.3$ s.

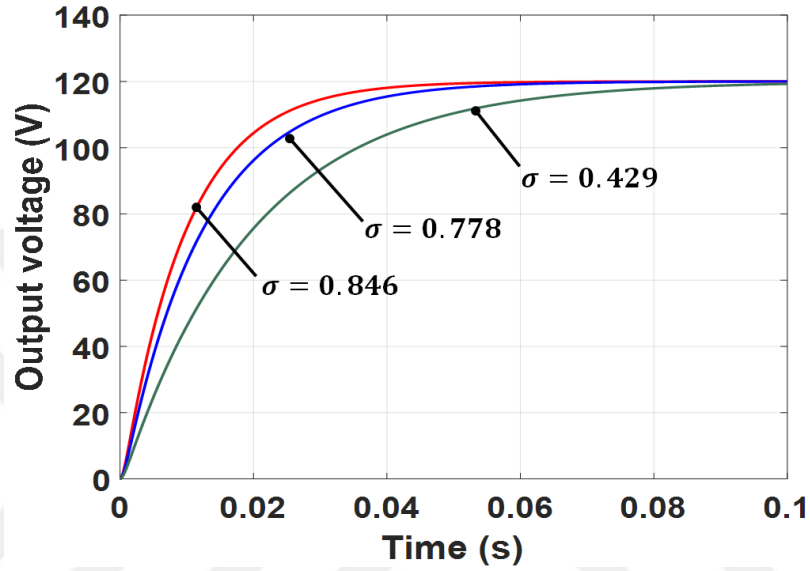


Figure 5.10: Voltage response of MR damper for various values of α .

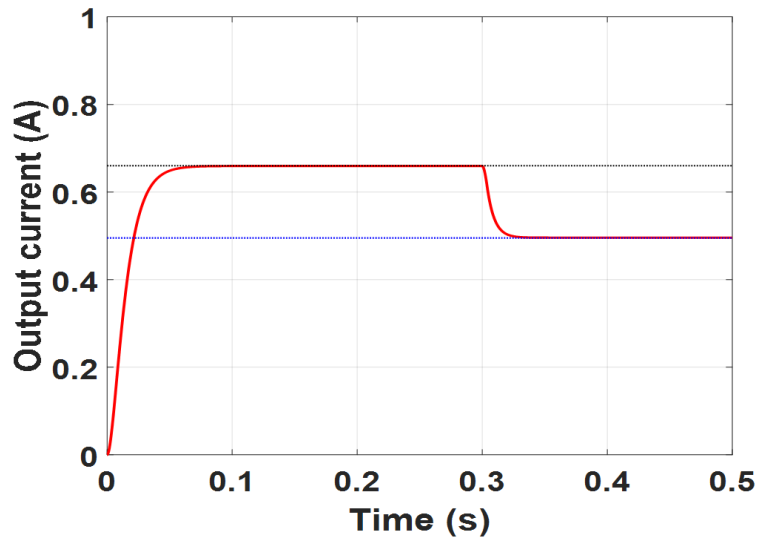


Figure 5.11: Electric current response of MR damper tracking its desired value equal to 0.495 A at $\alpha=0.846$.

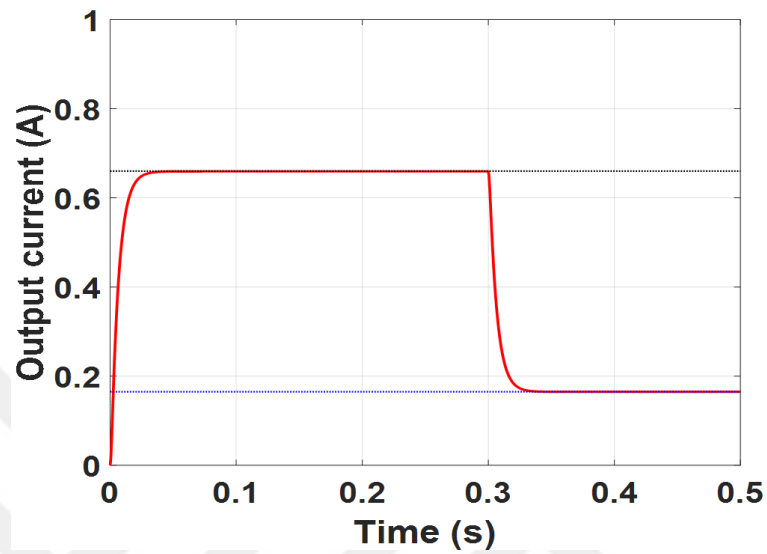


Figure 5.12: Electric current response of MR damper tracking its desired value equal to 0.165 A at $\alpha=0.846$.

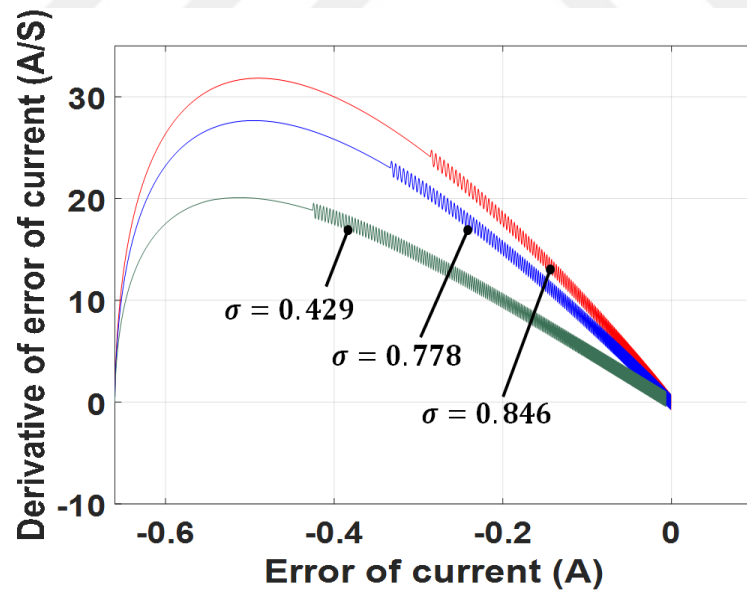


Figure 5.13: Trajectories of error current and its derivative of MR damper for various values of α .

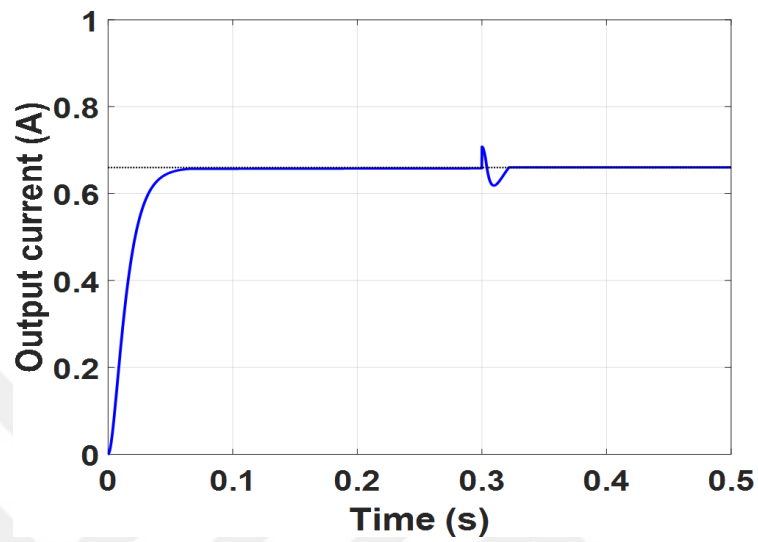


Figure 5.14: Electric current response of MR damper with the $d_{is}(t) = 50$ mA at $t = 0.3$ s.

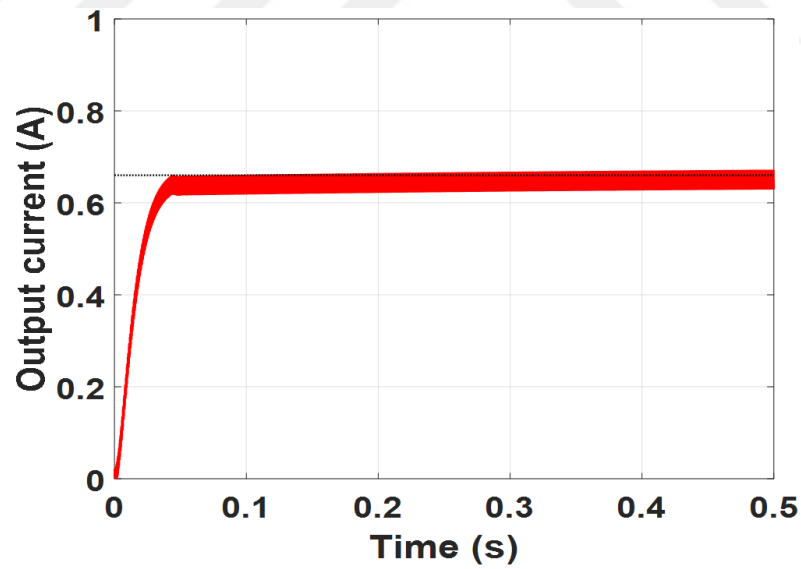


Figure 5.15: Electric current response of MR damper with the $d_{is}(t)$ equal to $0.02 \sin(1273.24\pi t)$.

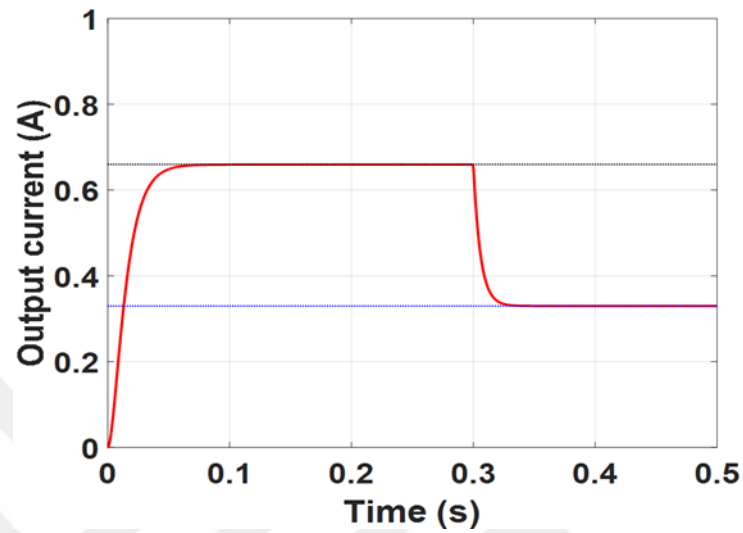


Figure 5.16: Electric current response of MR damper due to its variations of the parameters.

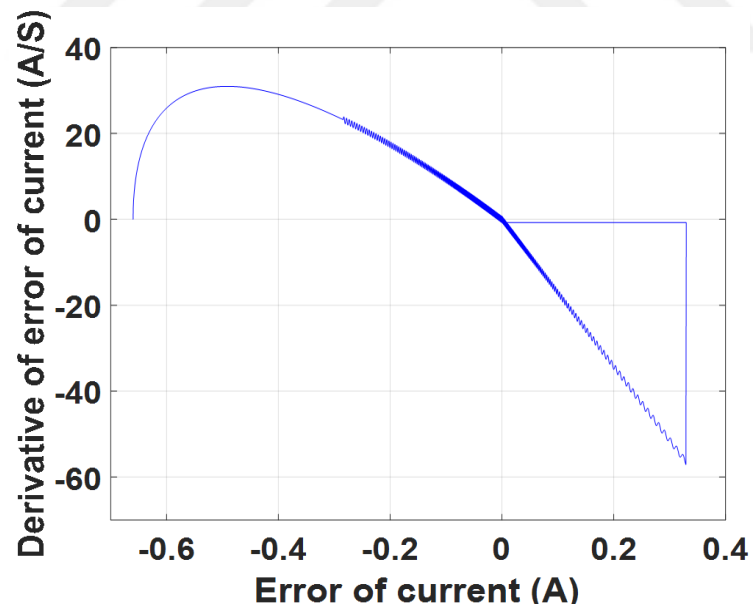


Figure 5.17: Trajectory between error current and its derivative of MR damper for new adaptive value of $\alpha = 165.63$.

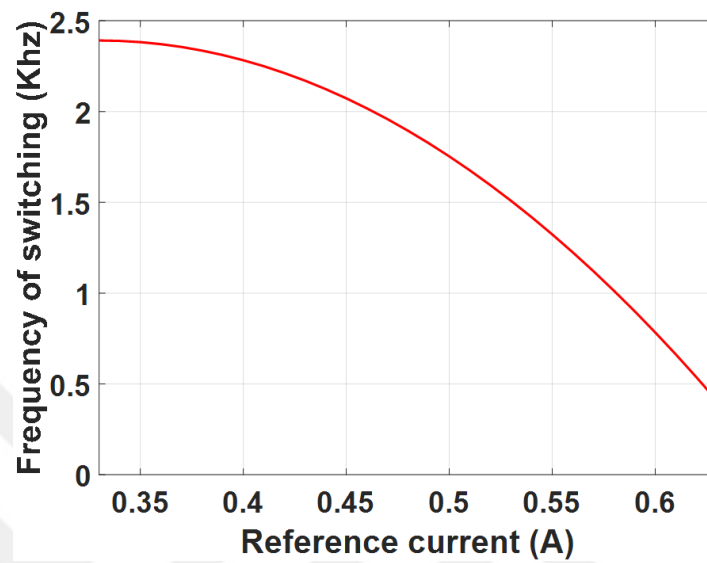


Figure 5.18: Behavior of frequency of switching with respect to desired current.

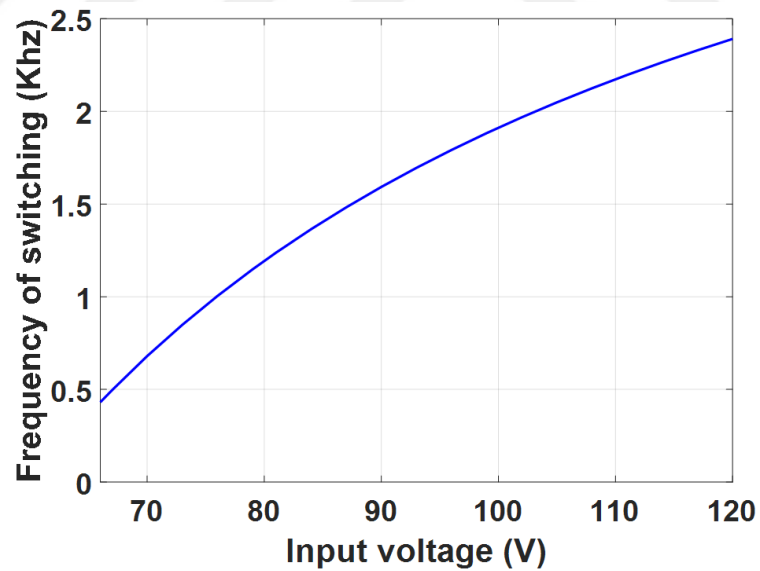


Figure 5.19: Behavior of frequency of switching with respect to voltage source input.

The controller rejects the disturbance very well as shown in Figure (5.14). On the other hand, the controller is also tested for the system in the presence of $0.02 \sin(1273.24\pi t)$ disturbance. This disturbance is also attenuated by the controller as given in Figure (5.15). The adaptive nature of the controller is determined by changing the values of R_{co} and L_{co} double of their original values. In response, the current changes with a new value of $\alpha = 165.63$ as given in Figure (5.16). The adaptive control make sure that the trajectory of current error reaches to the equilibrium point as shown in Figure (5.17). The behavior of f_{sit} with respect to desired current I_{fd} and V_{pi} which is the input voltage are given in Figure (5.18) and Figure (5.19). The desired current has inverse relation with frequency of switching. On the other side, the f_{sit} has direct relation with voltage source input.

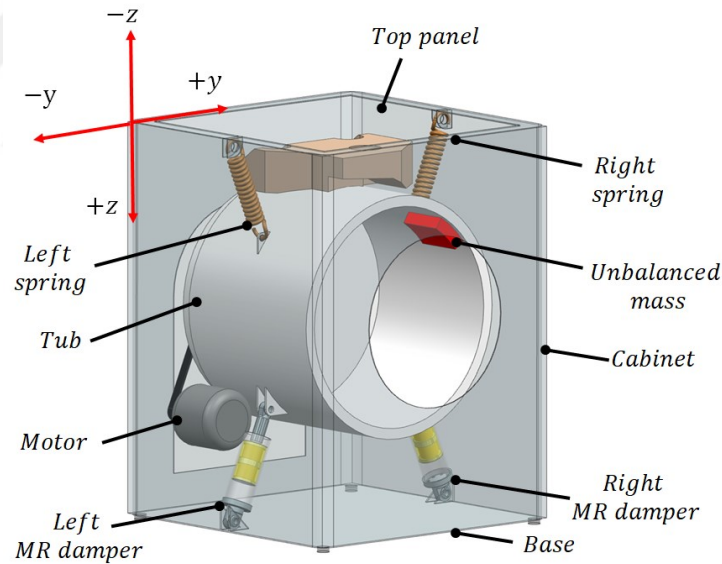


Figure 5.20: Washing machine CAD model schematic.

5.5 Analytical Model and Simulation Results of Washing Machine

The three dimensional CAD model of front loaded washing machine is shown in Figure 5.20. The model consists of tub assembly which is connected to the body of washing machine

using dampers and springs. The washing machine is analytically modeled according to specific degrees of freedom. A portable washing machine is discussed in [135]. The washing machine model with six degree of freedom is analyzed by taking rotational and translational motions along all three axes [148]. On the other hand, planar models are also utilized for the modeling of washing machine due to the rigidity of washing machine in axial direction. Another model of washing machine which is nonlinear with two degree of freedom is proposed and solved using genetic algorithm [149]. On the other side, some models of washing machine only consider the vertical motion of the tub [150-151]. However, this work has considered washing model with two degree of freedom in which the vertical motions are taken along z axis and horizontal motion is taken along y axis [151]. The washing machine is considered rigid along x axis according to the recorded value of acceleration. The springs of washing machine are equal in stiffness values and makes an angle β_s along vertical axis. The MR dampers make an angle α_m with the vertical axis. By using Newton's 2nd law of motion, the motion equations along zy axes are given in Equation (5.41) and Equation (5.42). Where f_{ud}^z is equal to $m_{ud}e\omega^2\sin(\omega t)$ and f_{ud}^y is equal to $m_{ud}e\omega^2\cos(\omega t)$ respectively. In addition to this, m_{ud} is drum's unbalanced mass, e represents eccentricity and ω shows the rotating speed.

$$f_{ud}^z - f_{me}^z - f_{pn}^z = m\ddot{z} \quad (5.41)$$

$$f_{ud}^y - f_{me}^y - f_{pn}^y = m\ddot{y} \quad (5.42)$$

By taking both the motions along vertical and horizontal axes independently, the coordinate of the spring q_{pn} which is inclined and it gives deflection due to the motion along vertical axis as given in Equation (5.43). The forces of the springs along vertical axis is shown in Equation (5.44). By putting Equation (5.43) into Equation (5.44), we get Equation (5.45). By taking $\delta\beta_s$ small as compared to β_s , the f_{pn}^z is simplified in Equation (5.46)

$$q_{pn} = Z\cos(\beta_s - \delta\beta_s) \quad (5.43)$$

$$f_{pn}^z = 2Kq_{pn}\cos(\beta_s - \delta\beta_s) \quad (5.44)$$

$$f_{pn}^z = 2KZ \cos^2(\beta_s - \delta\beta_s) \quad (5.45)$$

$$f_{pn}^z = 2KZ \cos^2(\beta_s) \quad (5.46)$$

The damping forces in the washing machine is expressed in Equation (5.47). The forces of MR dampers consists of viscous part and yield stress which plays its dominant part when the magnetic field intensity is applied as shown in Equation (5.48). The shear rate $\dot{\gamma}$ varies directly with the velocity between the cylinder and piston and changes in inverse manner to the height of gap as given by Equation (5.49). Furthermore, the viscous equivalent damping is given by Equation (5.50). Where f_{tym} is equal to $\tau_y A_{po}$ and ω is the vibration frequency. The MR damper has its also viscous part in terms viscosity, pole area and height as given by Equation (5.51). The total MR damping coefficient is represented in Equation (5.52). The damping forces of MR damper along vertical and horizontal axes are given in Equation (5.53) and Equation (5.54).

$$f_{me}^z = 2C_{mr}\dot{Z} \cos^2(\alpha_m) \quad (5.47)$$

$$f_{dm} = A_{po}(\eta\dot{\gamma} + \tau_y \text{sign}(\dot{\gamma})) \quad (5.48)$$

$$\dot{\gamma} = d\dot{q}/h \quad (5.49)$$

$$C_{dm} = 4f_{tym}/\pi\omega Z \cos(\alpha_m) \quad (5.50)$$

$$C_{vsoi} = \eta A_{po}/h \quad (5.51)$$

$$C_{mr} = C_{vsoi} + C_{dm} \quad (5.52)$$

$$f_{me}^z = (2C_{mr} \cos^2(\alpha_m))\dot{Z} \quad (5.53)$$

$$f_{me}^y = (2C_{mr} \cos^2(\alpha_m))\dot{Z} \quad (5.54)$$

By putting Equation (5.45)-(5.46) and Equation (5.53)-(5.54) into Equation (5.41)-(5.42), we get;

$$M_{ud}\ddot{Z} + (2C_{mr} \cos^2(\alpha_m))\dot{Z} + (2K \cos^2(\beta_s))Z = m_{ud}e\omega^2 \sin(\omega t) \quad (5.55)$$

$$M_{ud}\ddot{Y} + (2C_{mr} \sin^2(\alpha_m))\dot{Y} + (2K \sin^2(\beta_s))Y = m_{ud}e\omega^2 \cos(\omega t) \quad (5.56)$$

The vertical motion of the washing machine by restricting it to single degree of freedom as given by Equation (5.57). As the vibrations in the washing machine is due to unbalanced load of clothes inside the drum as shown by Equation (5.58). The transmissibility of washing machine is formulated in order to analyze the transmitted vibrations from the tub to the side panels, top and base of washing machine as given by Equation (5.59). Where F_{tr} is the transmitted force, ζ is the damping ratio and r is the frequency ratio. The transmitted force is dependent on the values of ζ and r . The transmissibility is high while using low damping at low frequencies, however, it is lowered at high frequencies. On the other hand, high damping shows the opposite behaviour as compared to the application of low damping.

$$z(t) = Z\sin(\omega t + \varphi) \quad (5.57)$$

$$MZ/me = r^2 / \sqrt{(1 - r^2)^2 + (2\zeta r)^2} \quad (5.58)$$

$$TRS = F_{tr}/me\omega^2 = \sqrt{1 + (2\zeta r)^2} / \sqrt{(1 - r^2)^2 + (2\zeta r)^2} \quad (5.59)$$

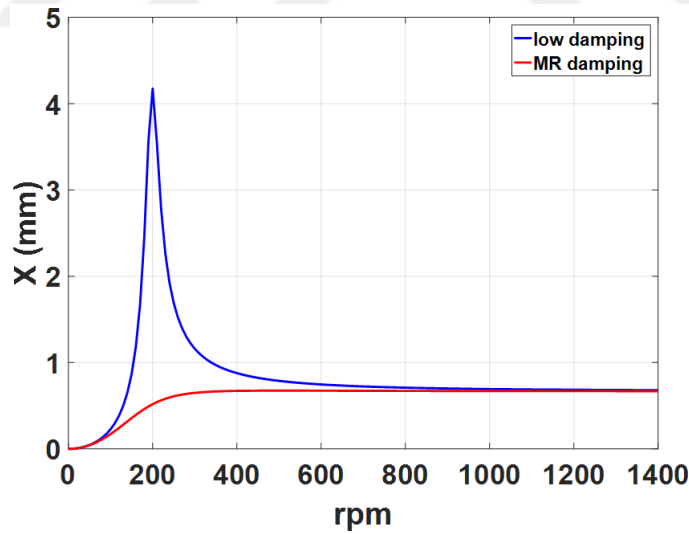


Figure 5.21: Amplitude of vibrations in terms of displacement by using low damping and MR damping.

The transmissibility is analytically modeled by using the mass of the oscillatory part equal to 25 kg, mass of laundry equal to 8 kg, stiffness of each spring equal to 8.6 kN/m, the angle of spring is selected from 20 to 30 degree. The model is simulated by using these values of the parameters. The response of the displacement by using low damping and MR damping is given in Figure 5.21. The transmissibility of washing machine is given in Figure 5.22. The low damping is not able to reduce the amplitude of vibrations in the low frequencies as compared to high damping. However, it gives good performance at high frequencies where high damping increase the vibrations of washing machine. Therefore variable and controlled MR damping is required for washing machine to perform effectively at all range of frequencies.

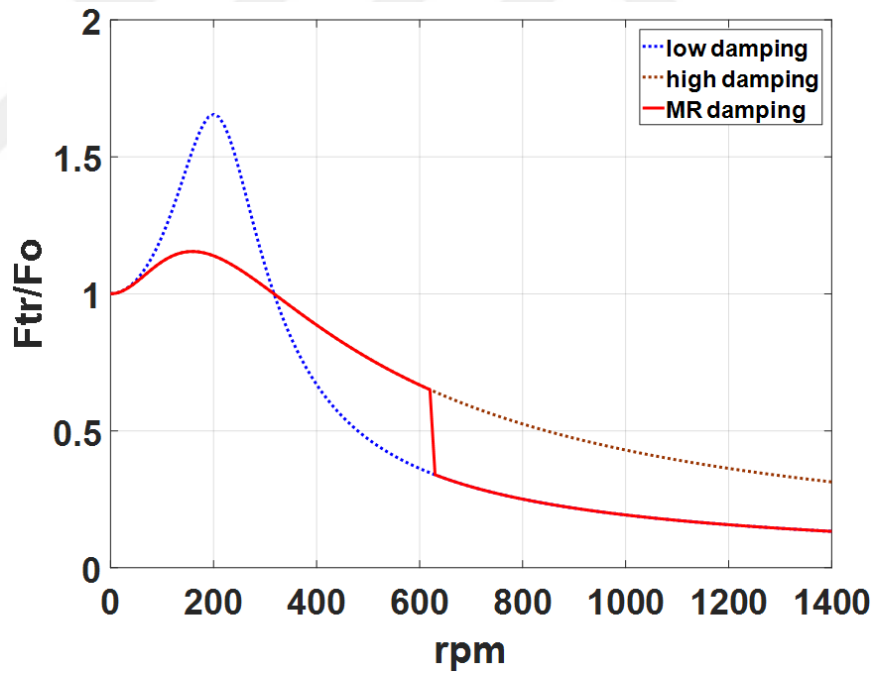


Figure 5.22: Behavior of transmissibility of washing machine by using low damping, high damping and MR damping.

5.6 Summary

This chapter presents the discussion regarding the design of the 2nd prototype of MR damper. The design improvements are done by analyzing the FEM model of 2nd prototype of MR damper. The size of the MR damper is reduced as compared to its 1st prototype single coil MR damper. The 2nd prototype of MR damper gives same value of force similar to the 1st prototype of MR damper. It becomes possible due to the successful analysis of geometric variable of the MR damper. After designing the 2nd prototype, it is desired to design an adaptive controller for the MR damper in order to reduce its response time. A terminal based sliding mode controller is designed and the parameter of its convergence is made adaptive for the variations in the parameters of MR damper. A simulation study is done for the whole system in order to validate the analytical model. It is clearly observed from the simulation graphs that the controller performs according to the specifications designed and mentioned in its mathematical mode. The modeling of the washing machine is performed in order to determine the forces along vertical and horizontal axes for the springs and MR dampers. Furthermore, the transmissibility of washing machine is also done analytically and simulated by using low damping, high damping and MR damping.

Chapter 6

EXPERIMENTAL SETUPS AND RESULTS DISCUSSION OF SECOND PROTOTYPE OF MR DAMPER FOR WASHING MACHINE

6.1 Introduction

This chapter introduces the experimental part of the MR damper and its testing with different samples of MR fluid synthesized by chemistry lab of Koç University. An experimental setup is designed for the 2nd prototype of MR damper in which the samples of MR fluid are tested and compared their results with commercial MR fluid. The hysteresis graphs of force with respect to velocity and displacement are found for various values of electric current. Another experimental setup is built for the adaptive vibration control of washing machine using MR dampers. An electronic circuitry along with data acquisition system is designed for the whole system. Different sensors such as force sensors, laser sensor, hall-effect sensor, current sensor and acceleration sensor are used to analyze the performance of the whole system.

6.2 Test Rig and Results of Second Prototype of MR Damper

The experimental test rig of second prototype of MR damper is developed for the measurement of force and performance testing of MR damper with different samples of MR fluid and hysteresis analysis of MR damper. The MR damper is made from AISI-1010 steel which is a low carbon steel. A double coil arrangement is used for the MR damper and the winding is done using copper wire. A power supply is connected with MR damper in order to supply the required value of current to coils. Different samples of MR fluid made by the chemistry lab of Koç University are used in MR damper along with the commercial MR fluid. The MR damper is excited by the linear actuator using its controller. A laser sensor is used along with its accessories to record the velocity of MR damper as well as its

displacement. A force sensor is used to record all ranges of force that the MR damper can absorb. The 2nd prototype of MR damper is tested with the first sample of MR fluid which is tested at different values of electric current ranging from 0 A to 0.62 A and its results are given in Figure 6.1-6.4.

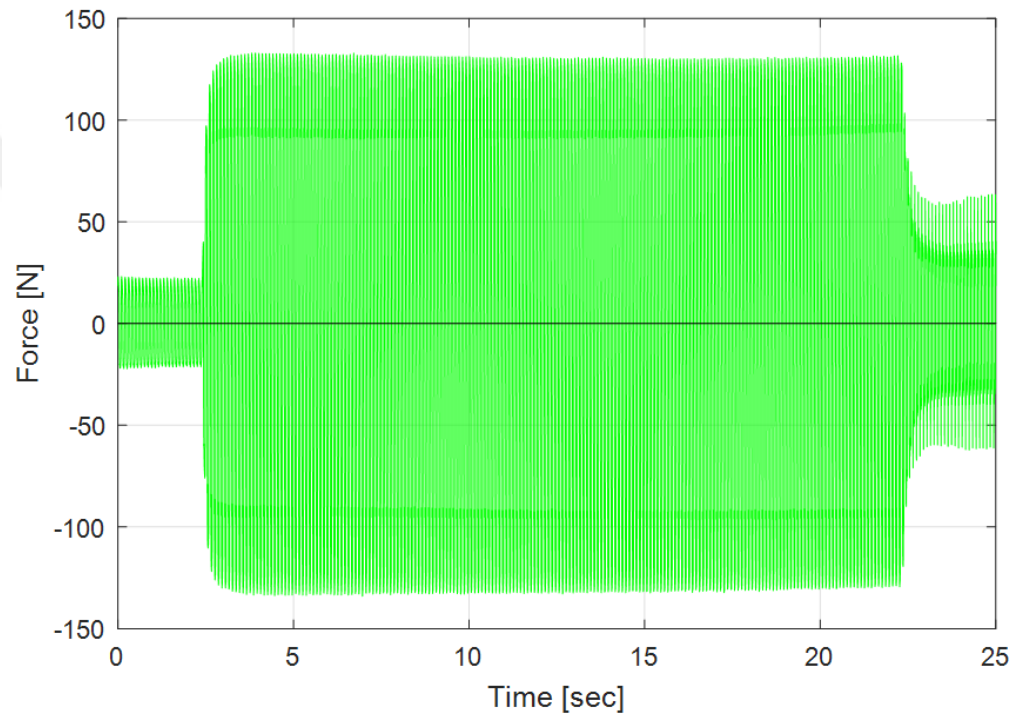


Figure 6.1: Damping force of 2nd Prototype of MR damper using 1st sample at 0.3 A.

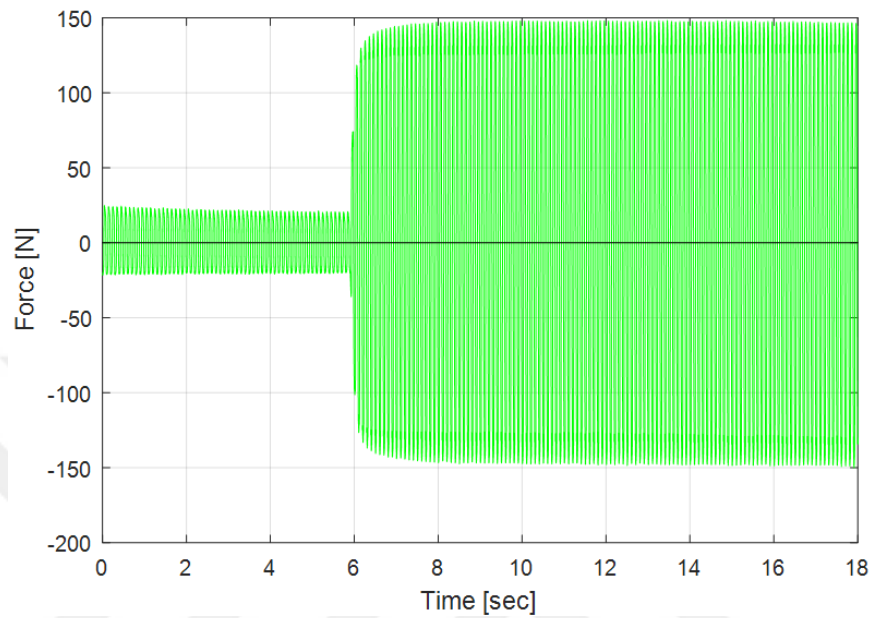


Figure 6.2: Damping force of 2nd Prototype of MR damper using 1st sample at 0.4 A.

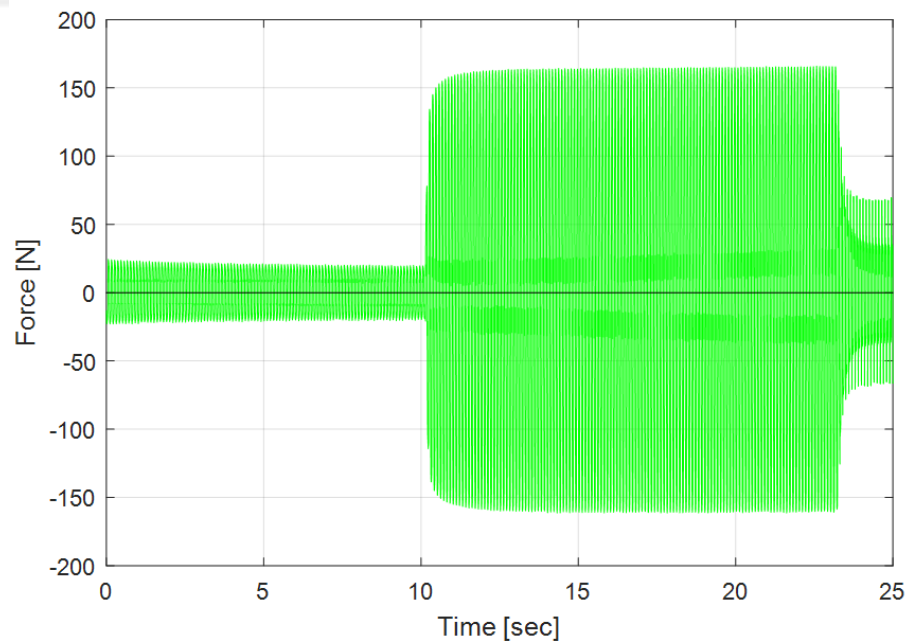


Figure 6.3: Damping force of 2nd Prototype of MR damper using 1st sample at 0.5 A.

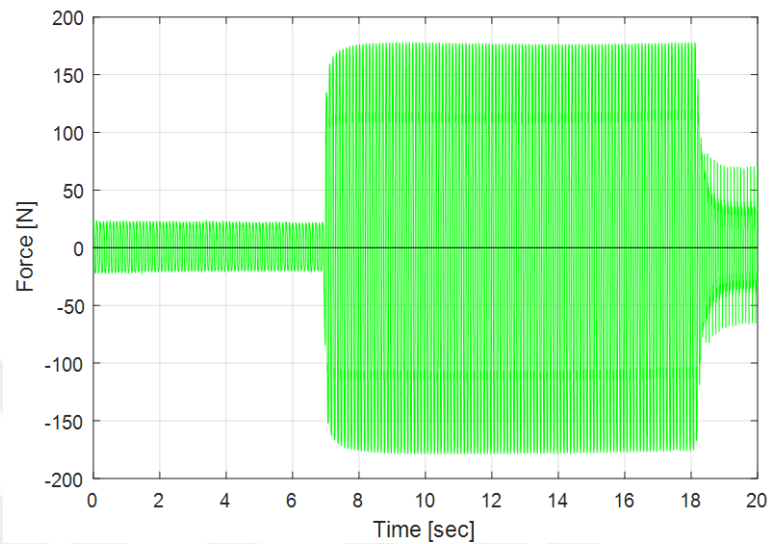


Figure 6.4: Damping force of 2nd Prototype of MR damper using 1st sample at 0.62 A. After the first sample, the 2nd prototype of MR damper is investigated with the 2nd sample of MR fluid. The 2nd sample is analyzed at different values of electric current which range from 0 A to 0.62 A and its results are shown in Figure 6.5-6.8.

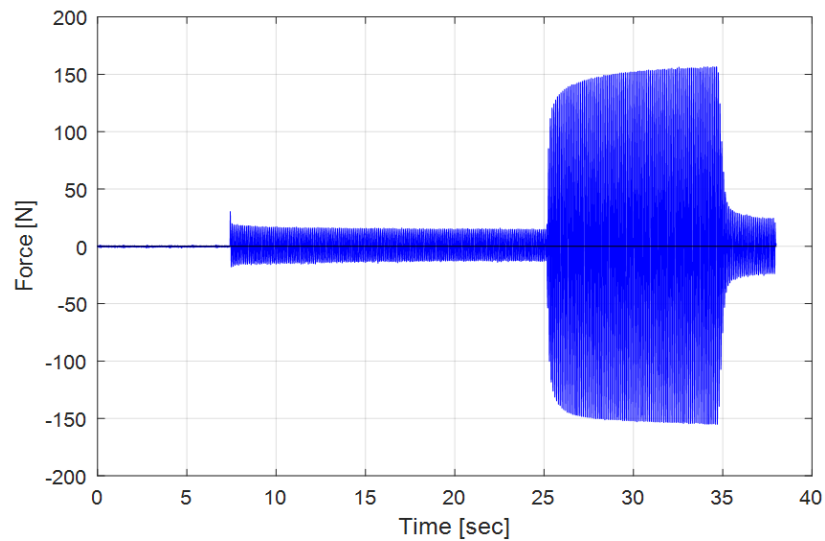


Figure 6.5: Damping force of 2nd Prototype of MR damper using 2nd sample at 0.3 A.

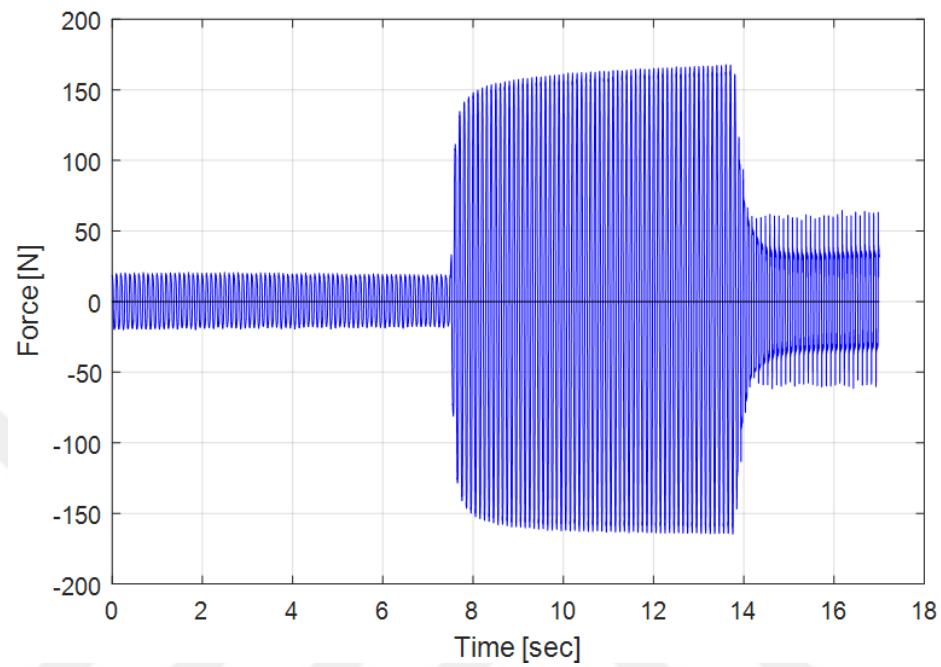


Figure 6.6: Damping force of 2nd Prototype of MR damper using 2nd sample at 0.4 A.

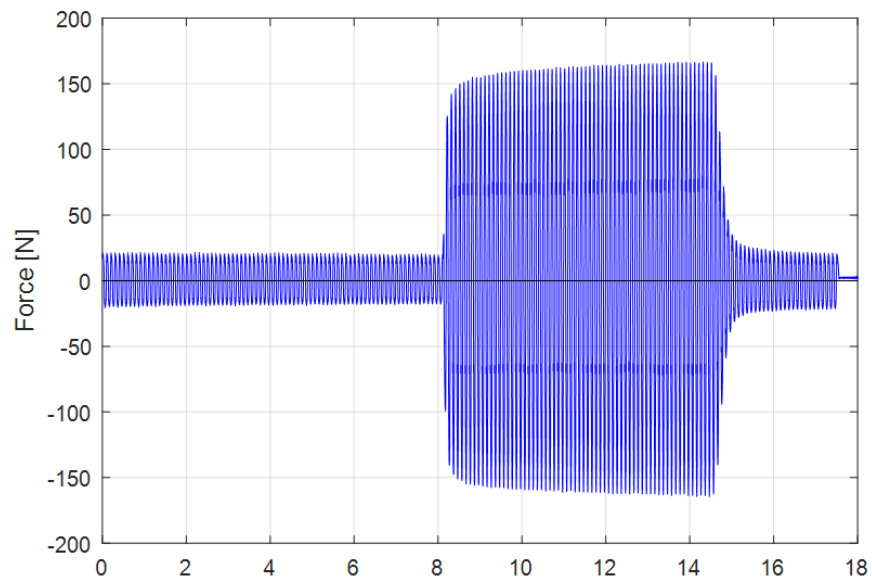


Figure 6.7: Damping force of 2nd Prototype of MR damper using 2nd sample at 0.5 A.

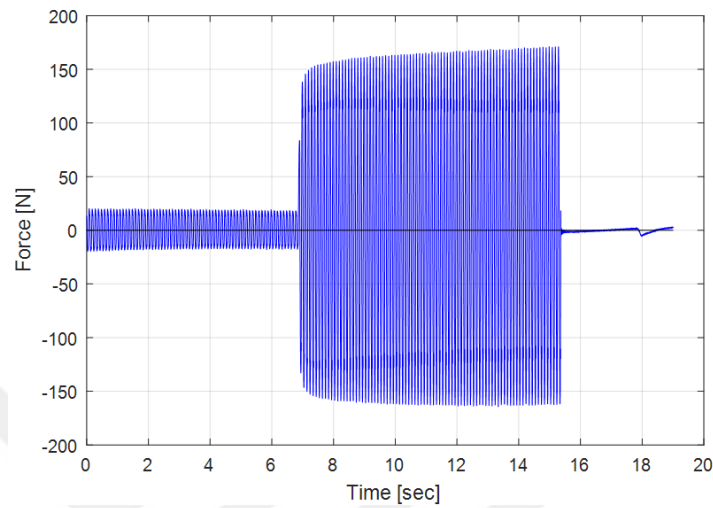


Figure 6.8: Damping force of 2nd Prototype of MR damper using 2nd sample at 0.62 A. After testing second sample, the 2nd prototype of MR damper is analyzed with the commercially available MR fluid. The MR fluid is tested at different values of electric current between 0 A to 0.62 A and their results are given in Figure 6.9-6.12.

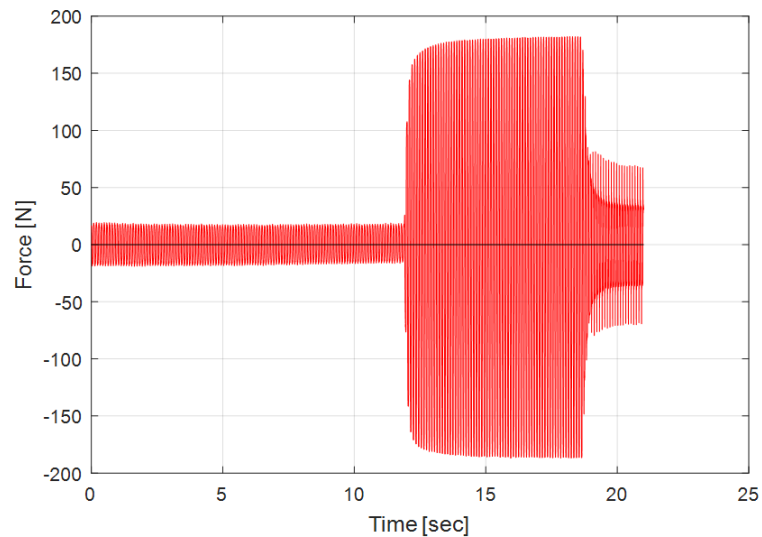


Figure 6.9: Damping force of 2nd Prototype of MR damper using commercial MR fluid at 0.3 A.

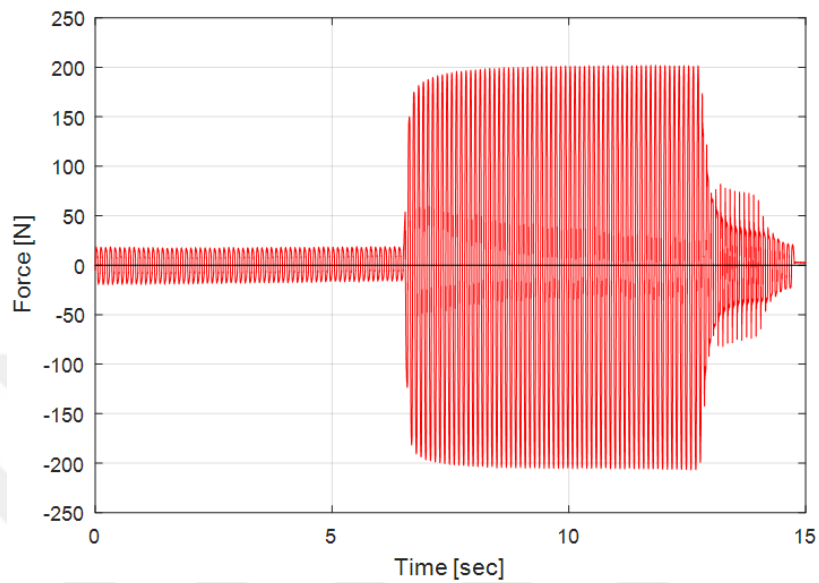


Figure 6.10: Damping force of 2nd Prototype of MR damper using commercial MR fluid at 0.4 A.

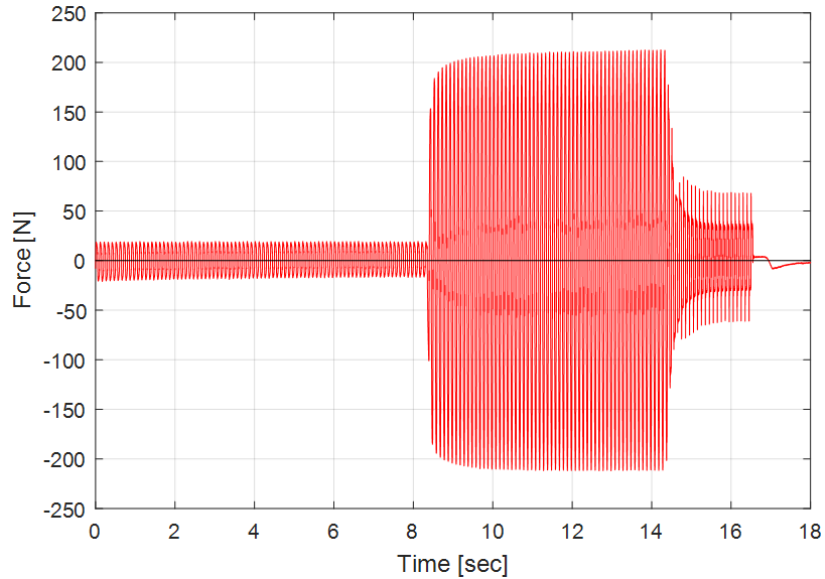


Figure 6.11: Damping force of 2nd Prototype of MR damper using commercial MR fluid at 0.5 A.

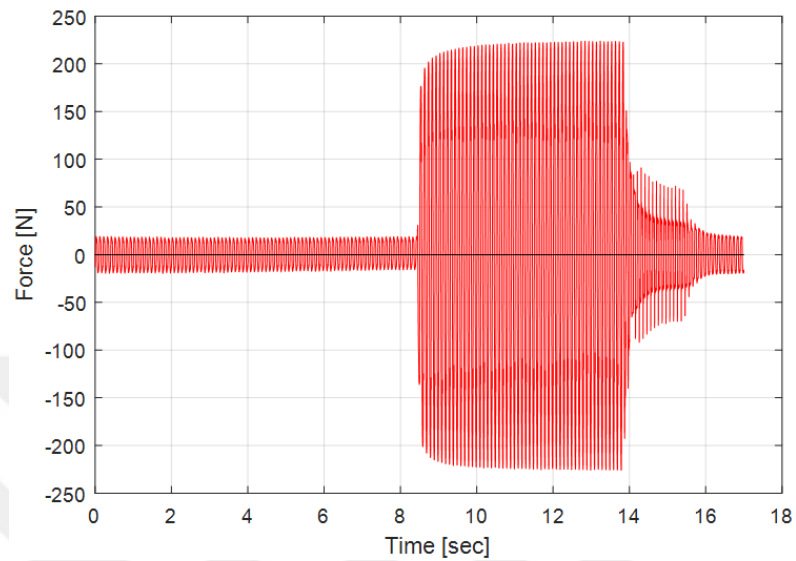


Figure 6.12: Damping force of 2nd Prototype of MR damper using commercial MR fluid at 0.62 A.

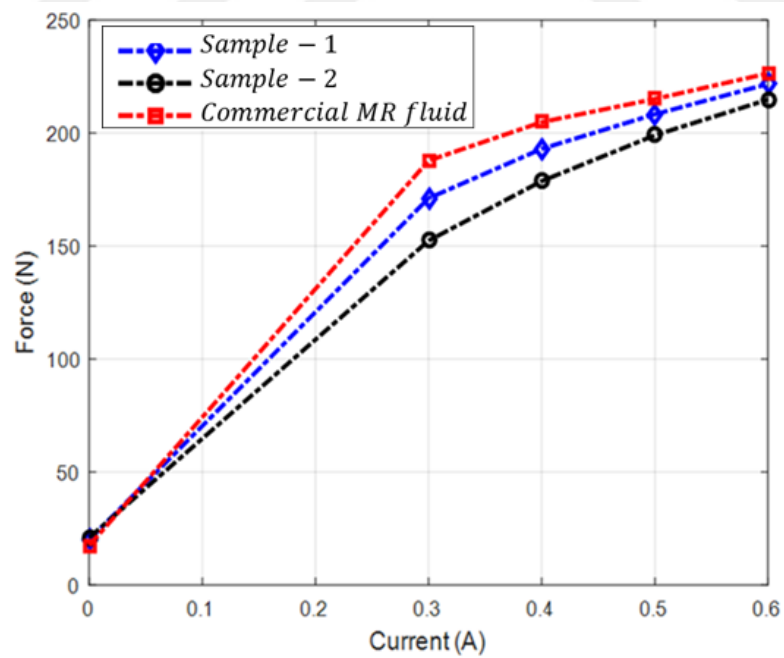


Figure 6.13: Damping force of 2nd Prototype of MR damper using different samples.

After testing two synthesized samples of MR fluid and commercial MR fluid, the 3rd sample is tested inside 2nd prototype of MR damper. A 4th sample of MR fluid is also analyzed inside the same MR damper. The performances of both the samples of MR fluid are compared with commercial MR fluid for different values of electric current from 0 A to 0.62 A as shown in Figure 6.13. Finally all the Koç University synthesized MR fluid samples and commercial MR fluid along with their corresponding damping forces are summarized in Table 6.1. After analyzing the results, it is proposed that Sample 3 has best performance in terms of minimum and maximum forces. In addition to this, the results of Sample 3 are very close to commercial MR fluid in terms of damping force. The response of Sample 3 is much better as compared to the rest of the samples. The oily effect of Sample 3 is negligible as it is found in other samples of MR fluid.

Table 6.1: Damping force results using different samples of MR fluid

Sample Name	Current (A)				
	0	0.3	0.4	0.5	0.6
Sample-1	22.1 N	130.74 N	146.92 N	164.8 N	172.3 N
Sample-2	14.03 N	155.2 N	165.6 N	166.17 N	175.8 N
Sample-3	20 N	170.3 N	195.11 N	206.43 N	222.16 N
Sample-4	21.01 N	150.51 N	183.92 N	195.24 N	215.05 N

Commercial MR Fluid	17.61 N	186.3 N	207.5 N	212.65 N	226.98 N
----------------------------	---------	---------	---------	----------	----------

The modal analysis of 2nd prototype of MR damper is performed in order to determine the bandwidth of the MR damper. For this purpose, an impact hammer test is utilized and the resultant response is recorded using acceleration sensor or laser sensor. The Fast Fourier Transform (FFT) is used to process the recorded data which helps in determining the frequency response function (FRF) of the complete system. The FRF is further used to find the stiffness, natural frequency and damping of the system. Therefore, the same procedure is implemented for the 2nd prototype of MR damper for the evaluation of its bandwidth. The displacement after the impact hammer test along the force are processed through data acquisition instruments and data analysis is performed in its corresponding softwares. Figure 6.14 shows the magnitude and phase plots of 2nd prototype of MR damper. In addition to this, Figure 6.15 clearly shows the frequency response and power spectrum of 2nd prototype of MR damper.

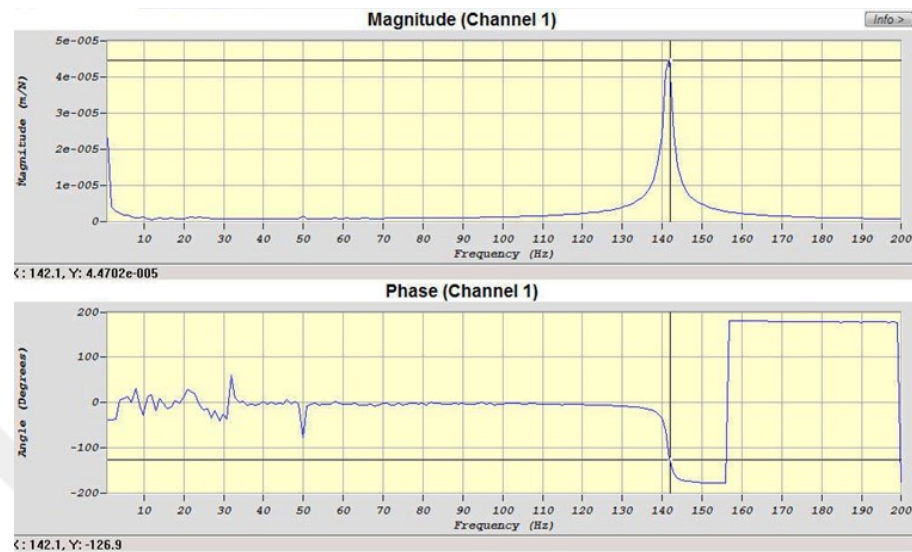


Figure 6.14: The magnitudes and phases plots of 2nd prototype of MR damper using impact hammer test.

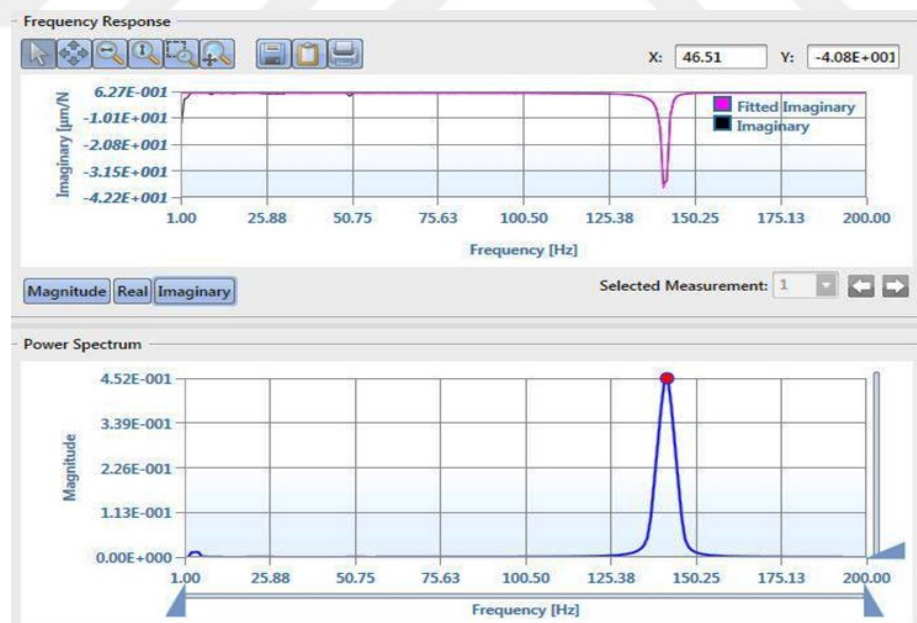


Figure 6.15: The FRF and power spectrum plots of 2nd prototype of MR damper using impact hammer test.

The hysteresis analysis of the 2nd prototype of MR damper is recorded experimentally for various values of electric current from 0 A to 0.6 A. The displacement, velocity and force are measured by using the sensors which are mentioned in the details of experimental setup. A data acquisition equipment is used for recording and processing all the data. The hysteresis graphs are given in Figure 6.16 and Figure. 6.17. It is quite clear from the graphs, that the MR damping force of 2nd prototype of MR damper increases as the input current also increases. This would result into large area of force versus displacement graph which means that high value of energy is dissipated at high value of electric current. Besides this, the hysteresis behavior of MR damper is more prominent in the pre-yield zone. But, the MR damping force shows very low variation with velocity when the MR damper functions in post-yield zone.

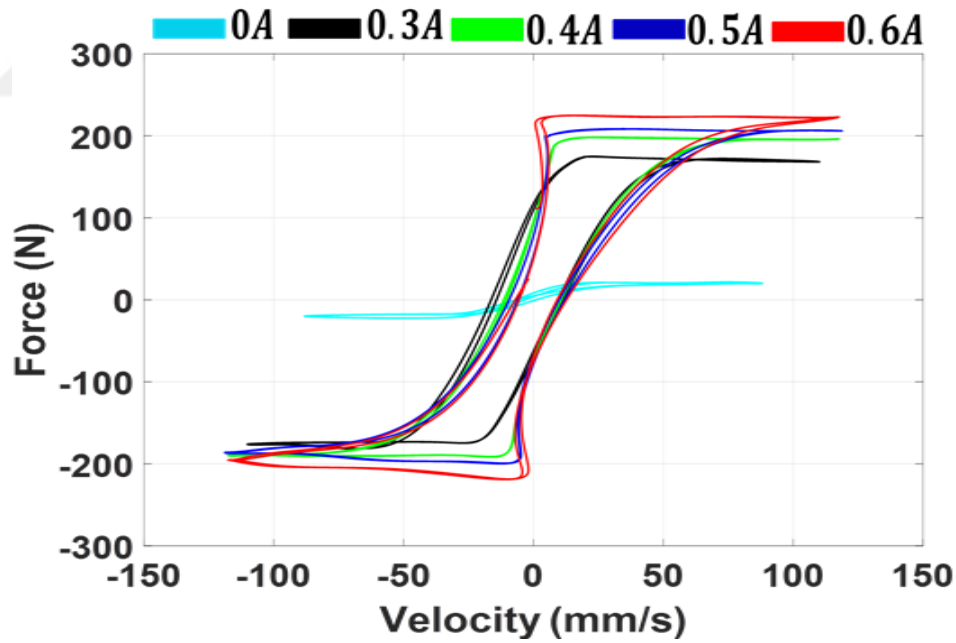


Figure 6.16: Hysteresis graph of force vs velocity for changing value of input current.

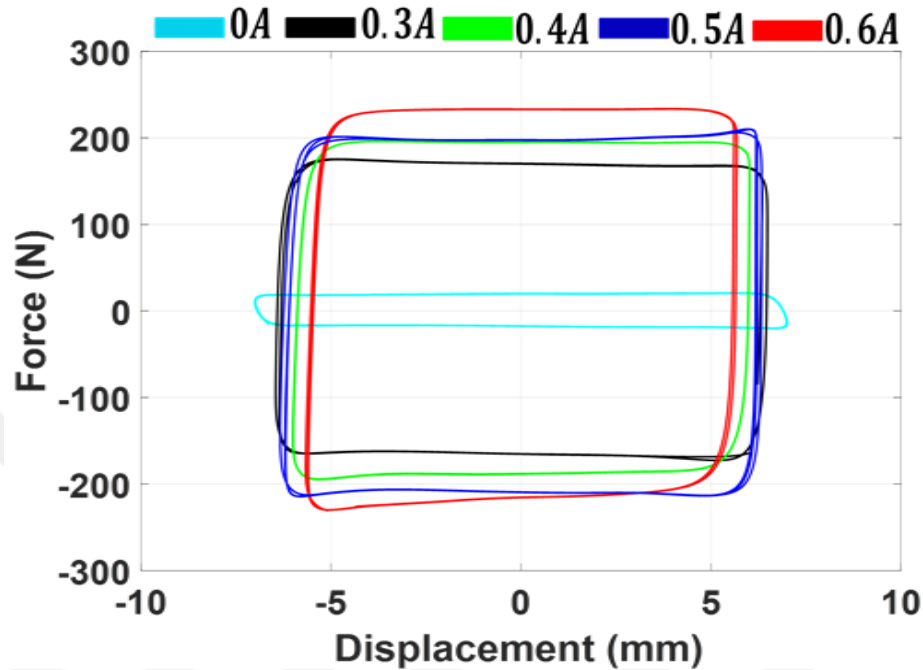


Figure 6.17: Hysteresis graph of force vs displacement for changing value of input current.

6.3 Test Rig and Results of Vibration Control of Washing Machine

A second experimental setup is made for the implementation of designed adaptive control on the 2nd prototype of MR damper inside washing machine as given in 6.18. The testing of MR damper is made in such a way so that to compare its vibrations results with the passive dampers. A power converter is designed having MOSFET, capacitor, inductor and diode. The DAQ cards and other controllers are used with real time environment of Matlab for the complete implementation of the adaptive control for MR damper. The power supplies are used to supply required power to the electronic board and MR dampers. The current sensors record the current value of MR dampers.

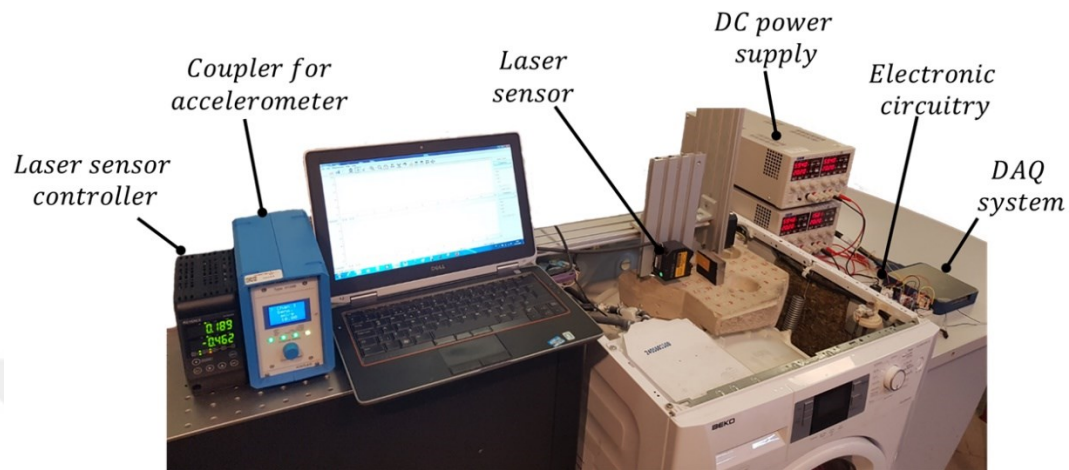


Figure 6.18: Experimental testing setup of adaptive vibration control of washing machine using 2nd prototype of MR damper.

A front loaded washing machine is chosen as testing machine for the 2nd prototype of MR damper. Inside washing machine, the mass which is suspended consists of clothes and rotating drum driven by the motor. This mass is further connected to the side panels through high stiffed springs and dampers. Originally, passive dampers of 100 N are installed in the machine. The two displacement sensors are used to record the values in the directions of yz axes. A three axes acceleration sensor is utilized for recording the acceleration. The washing machine rotates at maximum of 1400 RPMs in both directions. A hall-effect sensor is used to record the RPMs. The two force sensors are used to record the unbalanced force acting from the tub on the MR dampers is given in Figure 6.19.

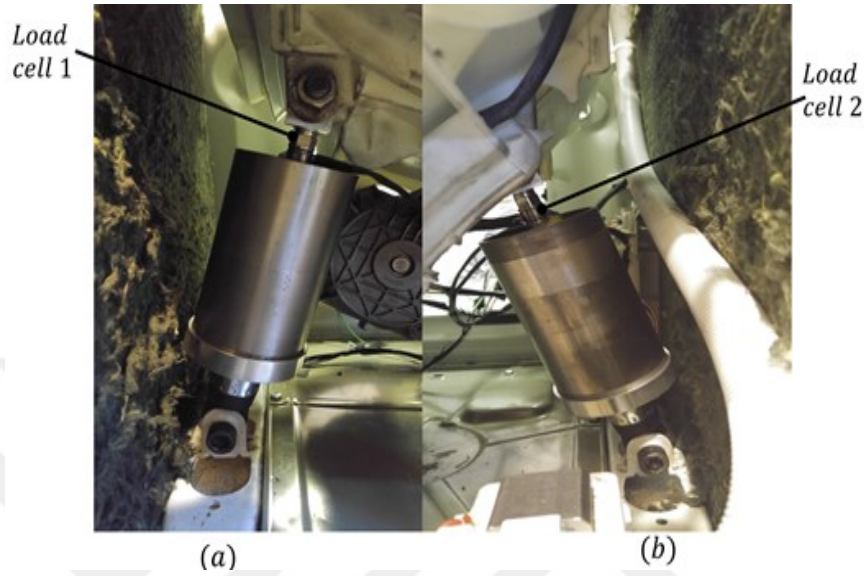


Figure 6.19: Installation of two force sensors in washing machine for force measurement.

The complete control strategy for the attenuation of vibrations is shown in Figure 6.20. The unbalanced forces such as f_{unb1} and f_{unb2} are recorded by the force sensors. The force error between the sensors data and the forces of the MR dampers are given to the controller. A PI control is built for the force based control technique according to the method discussed in [33]. The PI values for both the MR dampers are set to $c_p = 20.71$ and $c_i = 115.29$. A command based on the current is sent to the adaptive control where it adjusts the value of current through the MR dampers. The acceleration is recorded in all three directions and it is recorded by the acceleration sensor. The RPM or angular velocity is recorded by the hall-effect sensor and designed low pass filter which is used to suppress any unwanted high frequency noise. For this task, a unit gain filter is built in Equation (6.1). Where c_{1pr} is the gain of low pass filter and c_{2pr} is the time constant of the low pass filter. The frequency of the filter is designed to 750 Hz.

$$Q_{pr}(s) = c_{1pr}/1 + c_{2pr}s \quad (6.1)$$

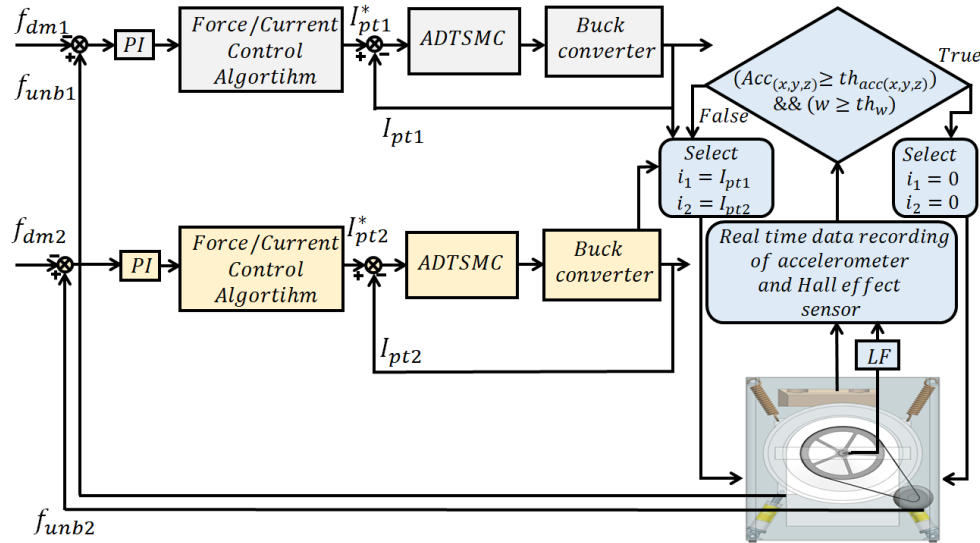


Figure 6.20: Vibration adaptive control technique developed for the washing machine using 2nd prototype of MR dampers.

The vibrations generated by the washing machine is investigated at various mass values of unbalanced load by using the dampers which are passive in nature. The unbalanced load comprises of clothes and water. On the other hand, the mass of clothes is varied from 2 to 8 kg. It is found that at 20 kg, washing machine experiences high vibrations. To counter these vibrations, 2nd prototype of MR dampers are inserted in washing machine instead of passive dampers. For the complete washing machine cycle, the 2nd prototype of MR dampers are tested with the adaptive control for washing machine. The adaptive control is investigated for various values of commanded electric current. Figure 6.21-6.24 clearly show that adaptive control follows the commanded electric current values along with its PWM inputs to the power converter. The commanded current values are 0.329 A and 0.495 A.

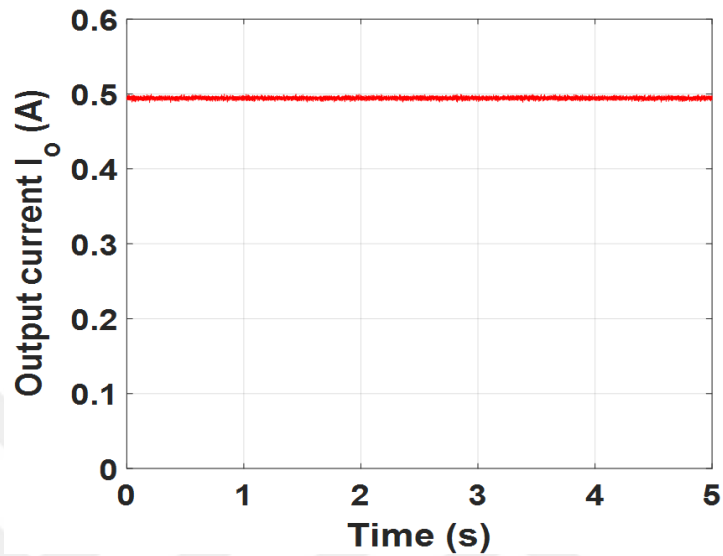


Figure 6.21: Electric current tracks $I_{fd} = 0.495$ A at $\beta = 0.846$.

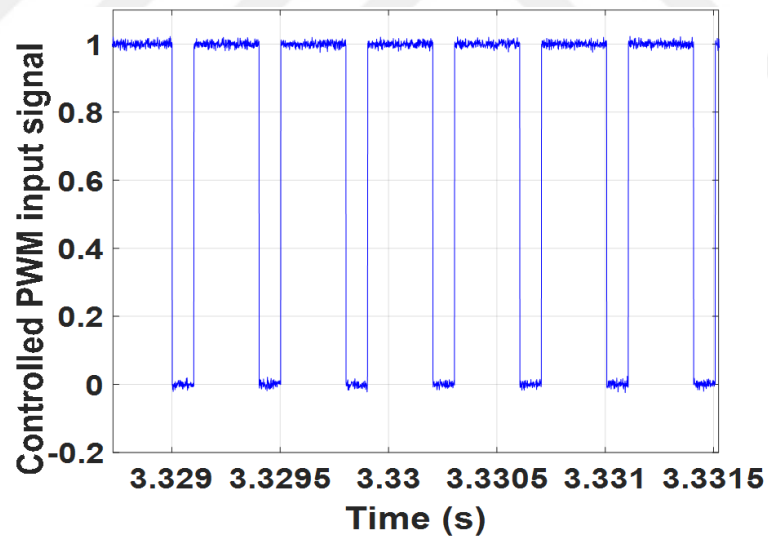


Figure 6.22: Commanded input control signal with duty cycle of 75 %.

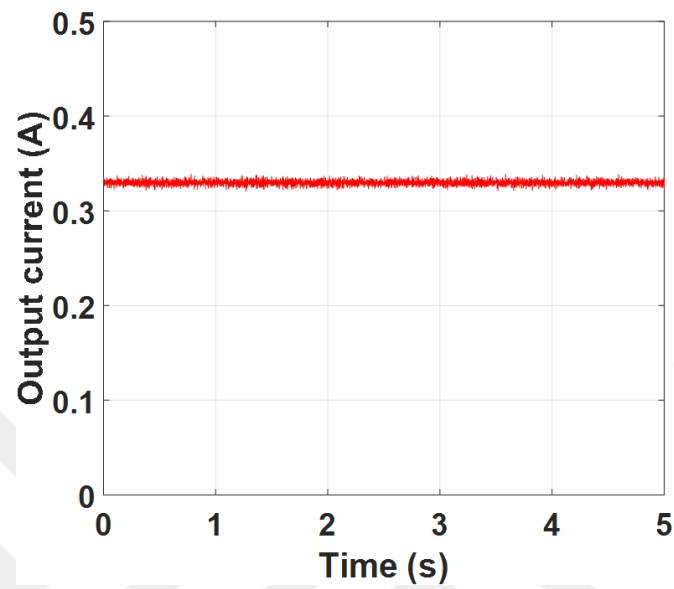


Figure 6.23: Electric current tracks $I_{fd} = 0.329$ A at $\beta = 0.846$.

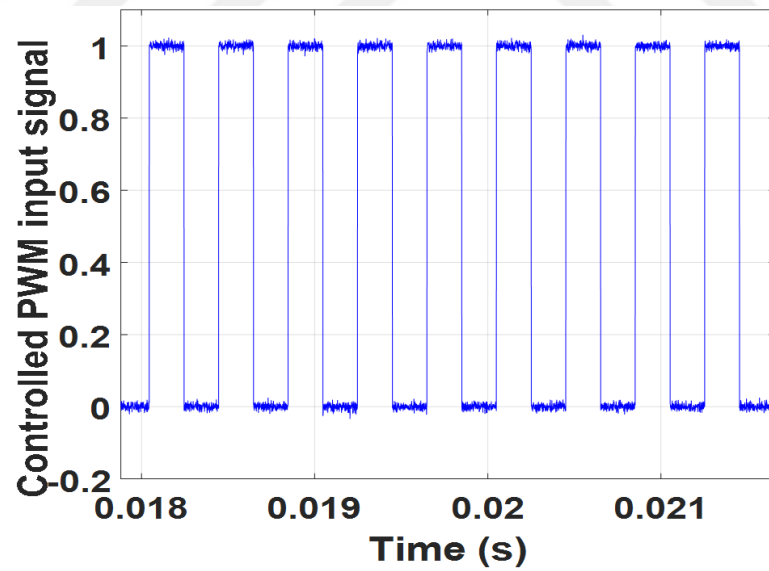


Figure 6.24: Commanded input control signal with duty cycle of 50 %.

The unbalanced forces from the tub are presented in Figure 6.25-6.26. The acceleration and displacement of vibrations for both types of dampers are given in Figure 6.27-6.31 as well as the angular velocity which is recorded by hall-effect sensor is shown in Figure 6.32.

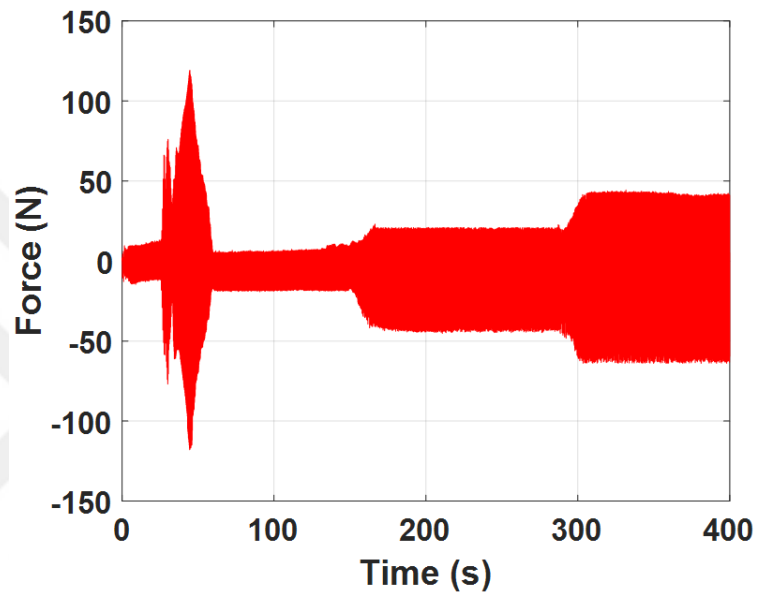


Figure 6.25: Measurement of unbalanced force by load cell 1.

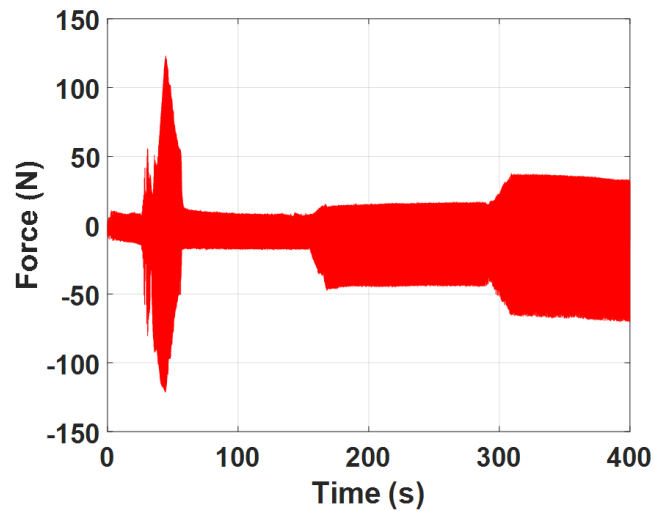


Figure 6.26: Measurement of unbalanced force by load cell 2.

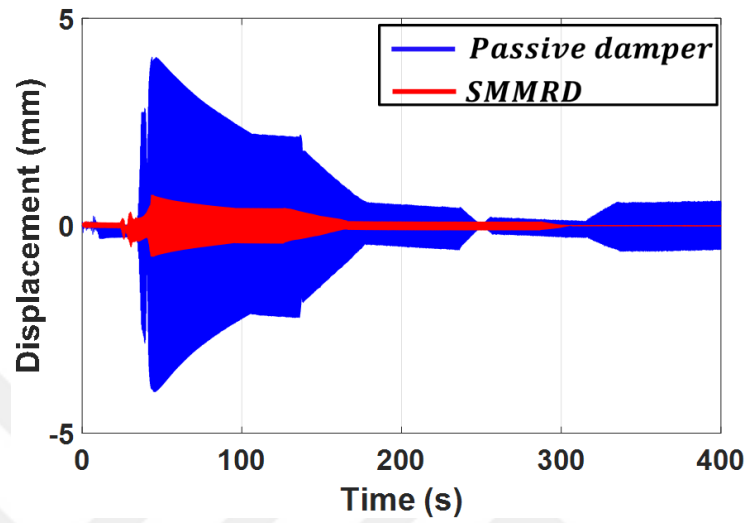


Figure 6.27: Displacement of vibrations along z axis.

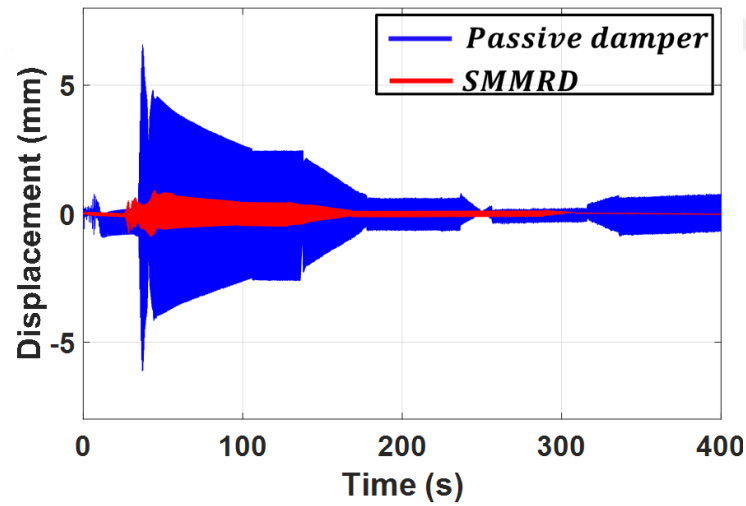


Figure 6.28: Displacement of vibrations along y axis.

The washing cycle of the machine is investigated into three main zones. The first zone is from 0 to 200 RPMs. A 7 N value of force is shown by the load cell as unbalanced force which is the maximum force in this zone. In y and z axes, 28.7 % and 22.6 % which are the

mean value of vibrations are recorded in first zone. The acceleration of mean values of vibration along x, y and z axes are recorded to 13.4 %, 17.2 % and 13.2 %. The adaptive control supplies mean current value of 0.6 A in the first zone to minimize the washing machine vibrations.

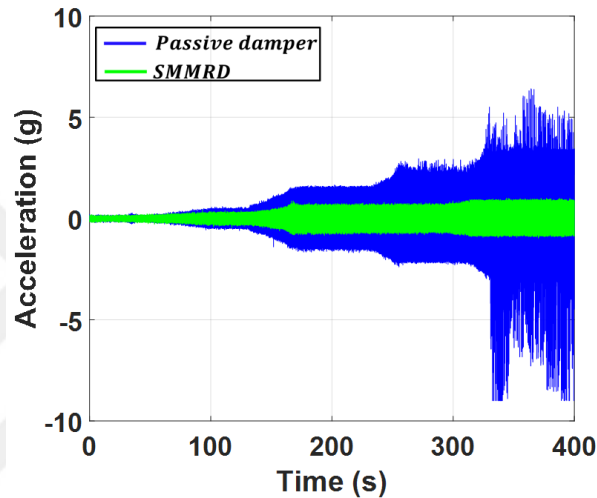


Figure 6.29: Acceleration of vibrations along x axis.

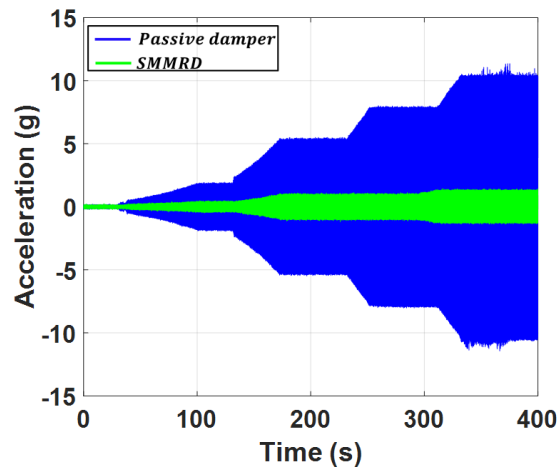


Figure 6.30: Acceleration of vibrations along y axis.

On the other side, the second zone shows the vibrations from 200 to 400 RPM. The tub applies 121 N force on the 2nd prototype of MR dampers. The passive dampers are unable to counter effectively the unbalanced forces which gives maximum value of vibrations in the second zone. In addition to this, the acceleration is also increased and they are measured in three axes such as x, y and z. In the second zone, high value of accelerations are observed. For the attenuation of these types of vibrations, a maximum of 0.62 A is given by the adaptive control to the 2nd prototype of MR dampers. Due to this action of controller, the mean displacement vibrations are decreased to 86.4 % and 85.5 % along y and z axes. The mean acceleration vibrations are attenuated to 13.8%, 66 % and 68.4 % along x, y and z axes.

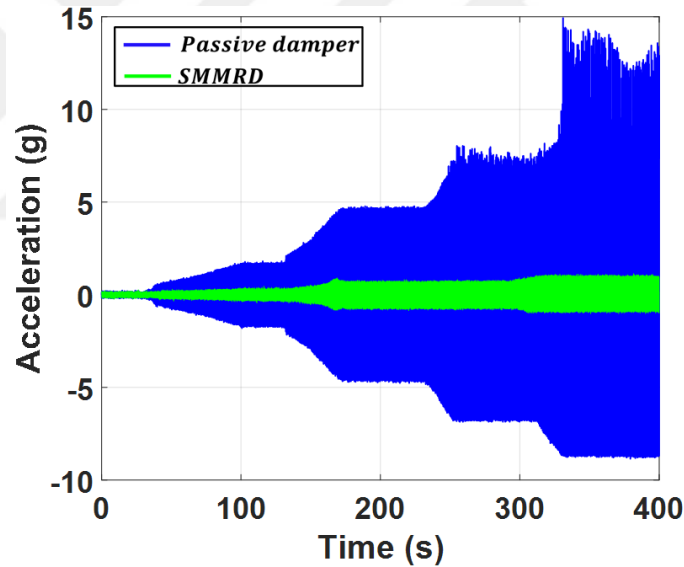


Figure 6.31: Acceleration of vibrations along z axis.

The adaptive control gives 0.45 A to the 2nd prototype of MR dampers for the variable damping in washing machine. In the second zone, the MR dampers generates 222.16 N force for countering the vibrations. On the other hand, the third zone shows the RPM range between 400 to 1400 RPMs. In the 3rd zone, the adaptive control doesn't give any current to the MR dampers and with the help of passive force of 20 N, the vibrations are countered. It

is also seen that the unbalance force start increasing as the RPM increases. A mean value of 30 N force are measured by the load cells in the 3rd zone. On the other hand, the displacement of mean vibrations are decreased to 75 % and 79.7 % along y and z axes. The acceleration of mean vibrations are suppressed to 56.4 %, 79.9 % and 82.9 % along x, y and z axes. In the 3rd zone, the acceleration of vibrations are more evident from the side panels and top surface of washing machine.

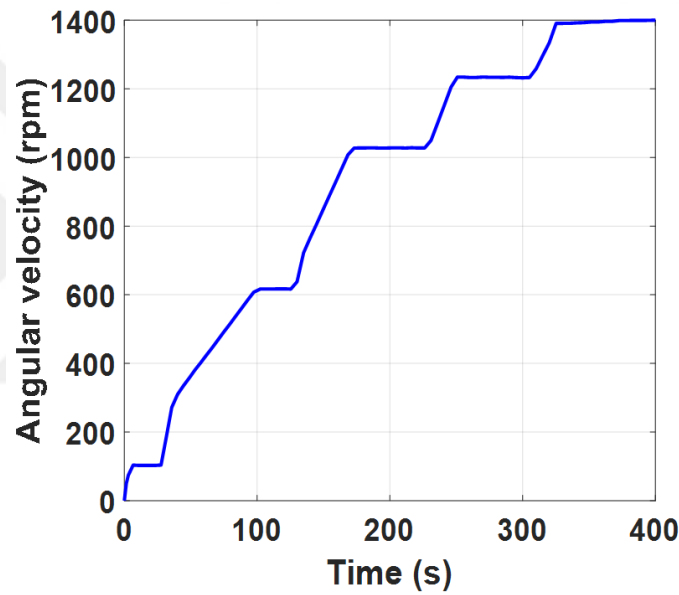


Figure 6.32: Washing machine angular velocity for its complete cycle.



Figure 6.33: Load cells are utilized at two sides of 2nd prototype of MR damper.

The transmissibility of washing machine is investigated by load cells that are fixed at both sides of 2nd prototype of MR damper as given in Figure 6.33. In addition to this, one load cell is fixed between tub and MR damper and another load cell is fixed between base of machine and MR damper as shown in Figure 6.34.

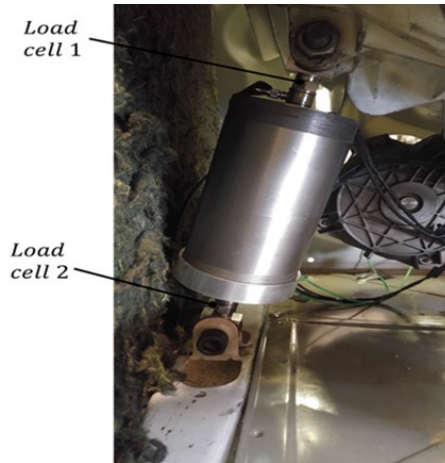


Figure 6.34: Load cells are used at two sides of 2nd prototype of MR damper for transmissibility measurement.

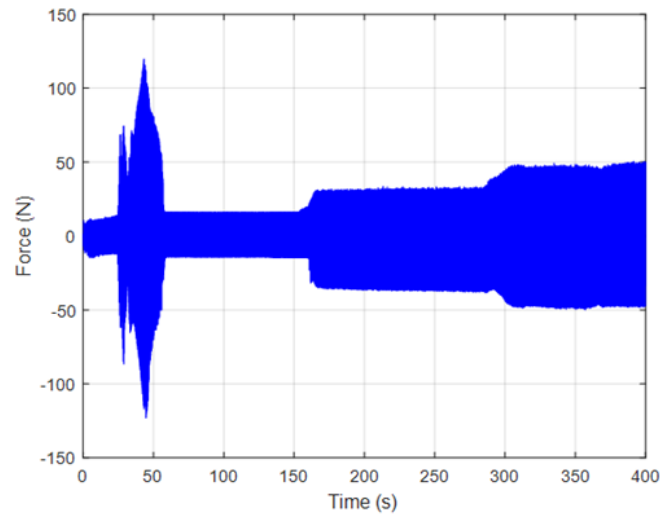


Figure 6.35: Load cell 1 measurement of force in the washing machine.

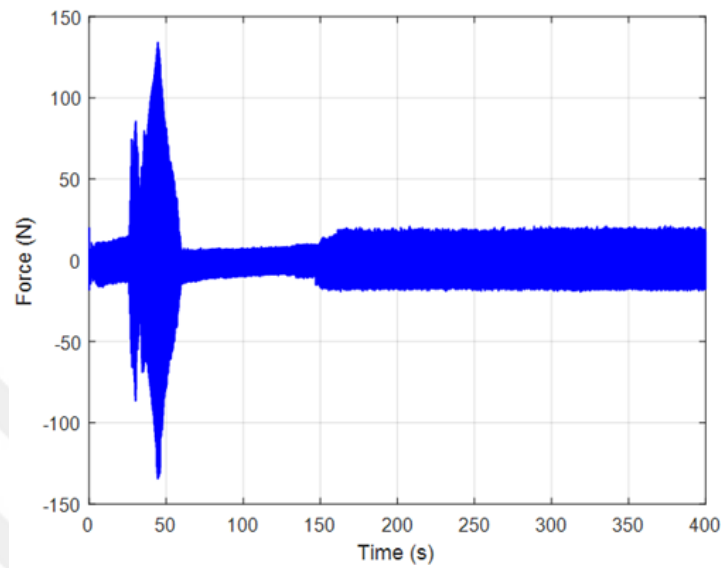


Figure 6.36: Load cell 2 measurement of force in the washing machine.

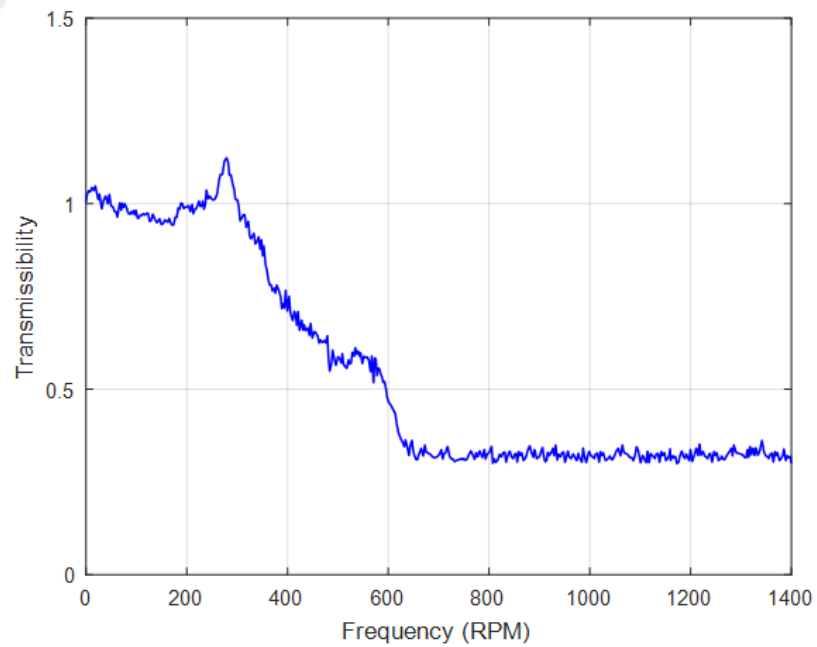


Figure 6.37: Transmissibility measurement by the load cells in the washing machine for the complete cycle.

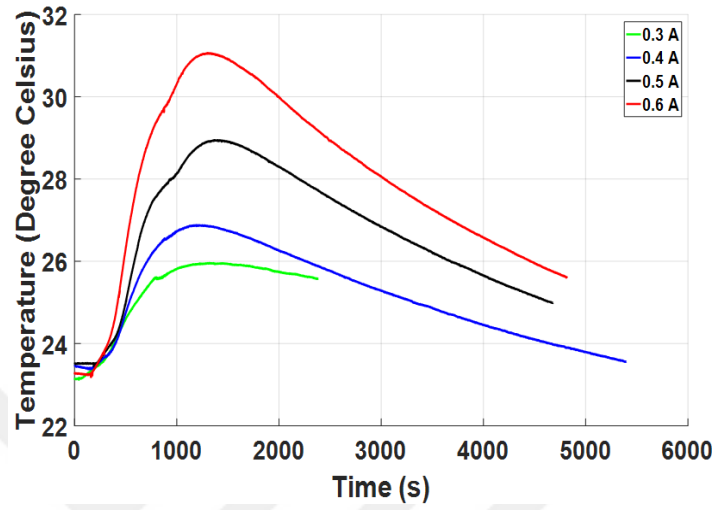


Figure 6.38: Behavior of temperature at the outside surface of MR damper's cylinder.

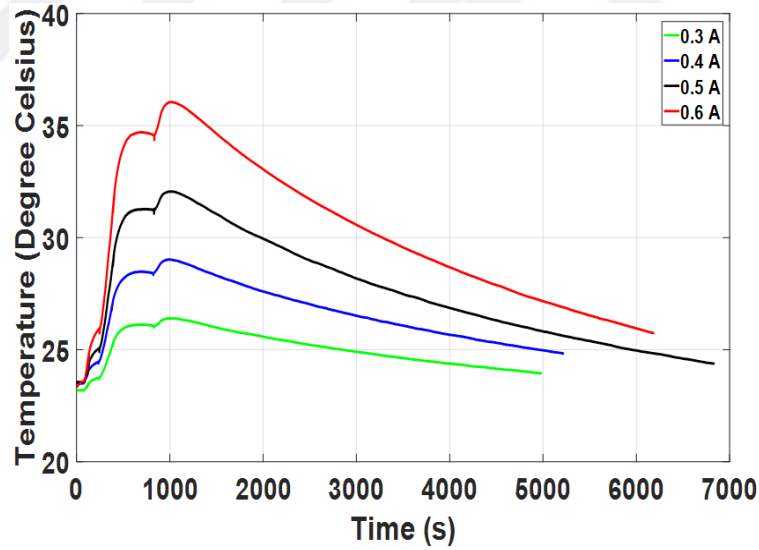


Figure 6.39: Behavior of temperature at the inside surface of MR damper's piston.

The two load cells measure the unbalanced forces as given in Figure 6.35-6.36. These forces are further used to experimentally determine the behavior of transmissibility versus RPM as given in Figure 6.37. From the observations, it is found that the transmissibility is attenuated in the low resonance region by using controlled damping of 2nd prototype of MR dampers.

However, the transmissibility starts reducing as the washing machine performs its operations at high RPMs. The temperature of 2nd prototype of MR damper is experimentally found using thermocouples. The thermocouples are fixed on the piston and cylinder of MR damper. The temperature is recorded at various values of electric current. From the results, it is seen that peak temperature value of 31.06 °C is measured at 0.6 A at the surface of MR damper's cylinder as shown in Figure 6.38. On the other side, 36.05 °C is the highest value measured at 0.6 A at the piston of 2nd prototype of MR damper as given in Figure 6.39.

6.4 Summary

This chapter in detail discusses the experimental test rig of 2nd prototype of MR damper. The 2nd prototype of MR damper is successfully tested with different sample of MR fluid and compare its performance with commercial MR fluid. The 2nd prototype of MR damper characteristics are measured in terms of damping forces and hysteresis behavior. A second experimental setup is developed for the vibration adaptive control of washing machine using 2nd prototype of MR damper. A special control strategy is built for the vibration control using load cells, displacement sensors, hall-effect sensor, current sensor and accelerometer. The vibrations are efficiently decreased by the 2nd prototype of MR damper as observed from the results. The transmissibility of washing machine is recorded by using load cells. Moreover, for the safe operation of MR dampers in washing machine, the temperature is measured for the complete washing cycle.

Chapter 7

DESIGN, TESTING AND VIBRATION CONTROL OF THIRD PROTOTYPE OF MR DAMPER FOR WASHING MACHINE

7.1 Introduction

This chapter discusses the third prototype of MR damper for the vibration attenuation of washing machine. A double coil arrangement is proposed for the 3rd prototype of MR damper. The design of the 3rd prototype of MR damper is analyzed by using finite element analysis. The geometric parameters are adjusted in such a way so that optimal design is realized for targeted force of 110 N. In addition to this, the 3rd prototype is tested experimentally for the targeted force value. The performance of 3rd prototype of MR dampers are successfully examined inside washing machine and compared with passive dampers. The temperature of 3rd prototype of MR dampers are also recorded during the washing cycle of machine.

7.2 Design of 3rd Prototype of MR Damper

The double coil sponge based MR damper is designed by investigating the magnetic circuit of 3rd prototype of MR damper using finite element analysis. The magnetic flux density at the active poles is fully relied on the geometric variables of the 3rd prototype of MR damper. The magnetic circuit is built in finite element analysis software by using the two dimensional axis symmetric model of 3rd prototype of MR damper as given in Figure 7.1. The distribution of magnetic field is presented in the equivalent three dimensional model of 3rd prototype of MR damper as shown in Figure 7.2.

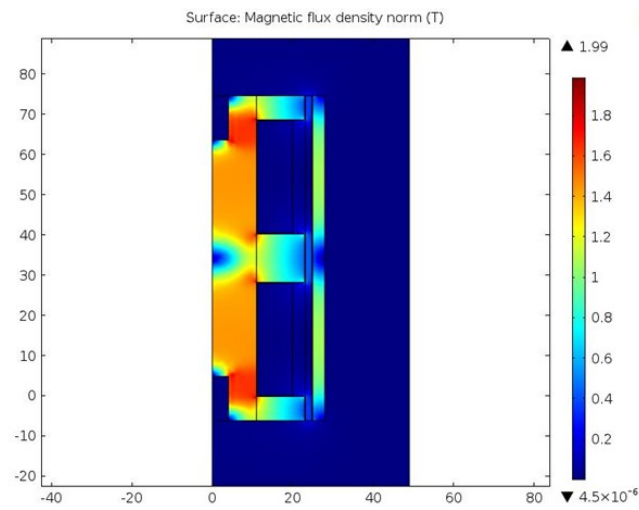


Figure 7.1: Magnetic flux density distribution in two dimensional axis symmetric model of 3rd prototype of MR damper.

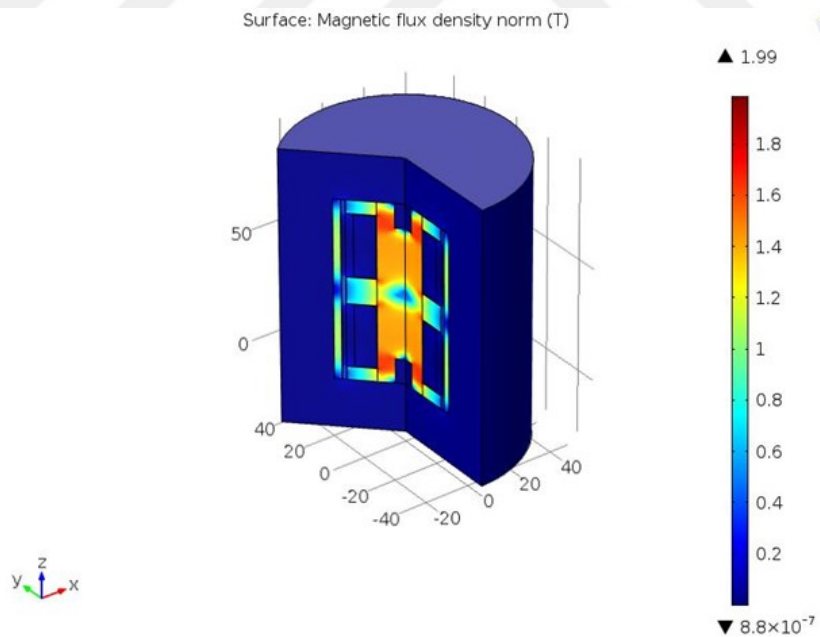


Figure 7.2: Magnetic flux density distribution in three dimensional axis equivalent model of 3rd prototype of MR damper.

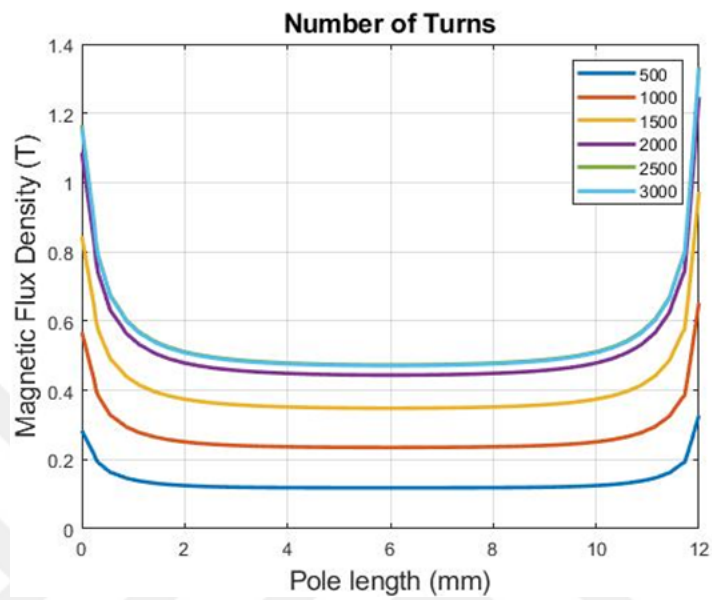


Figure 7.3: Magnetic flux density behavior versus length of active pole for changing values of number of turns.

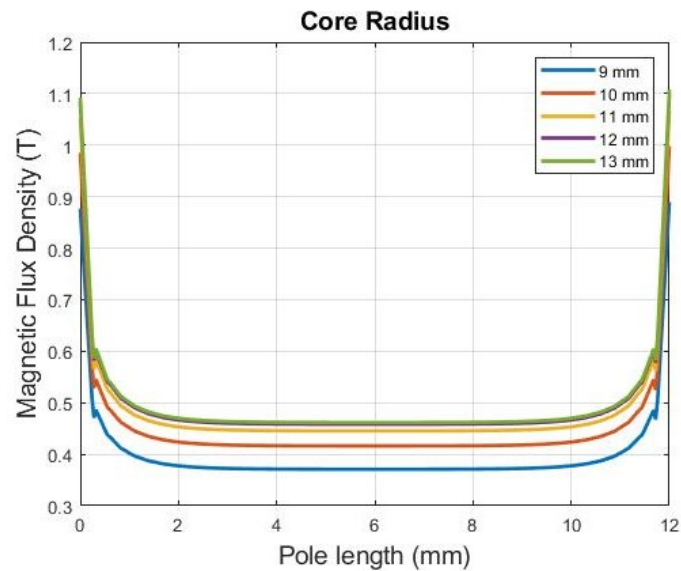


Figure 7.4: Magnetic flux density behavior versus length of active pole for changing values of radius of core.

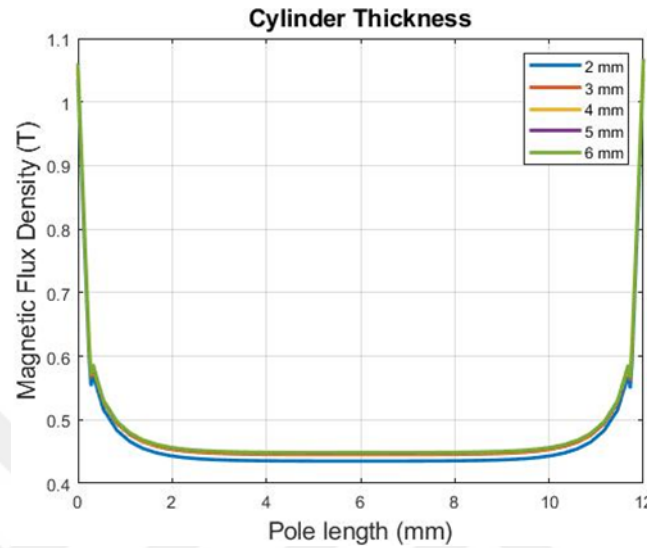


Figure 7.5: Magnetic flux density behavior versus length of active pole for changing values of cylinder thickness.

A parametric sweep technique available in the finite element analysis is implemented for the two dimensional axis symmetric model of 3rd prototype of MR damper. The total length, outer diameter and length of stroke of 3rd prototype of MR damper are decided based on the geometric variables. The optimal design of 3rd prototype of MR damper gives targeted results inside washing machine. A high permeable steel such as AISI-1010 is assigned to the cylinder and core of MR damper. However, aluminum is used for the side cover of MR damper so that maximum magnetic flux flows inside the magnetic circuit. A nonlinear BH curve is used in finite element analysis in order to model the saturation of the steel.

The optimal value of number of turns are investigated between 500 to 3000 turns as given in Figure 7.3. The gauge of copper wire is chosen in such a way so that the cross section is minimized and overall weight of MR damper is selected to the optimal value. It is seen that after 2000 turns the magnetic flux density at the active pole is saturated. On the other hand, the core radius is optimized between 9 to 13 mm as shown in Figure 7.4. It is seen that 11 mm is the optimum value for the maximum magnetic flux density at the active pole. In

addition to this, the value is chosen in such a way so that the outer diameter of 3rd prototype of MR damper doesn't exceed from the targeted size of the MR damper. Besides this, for the cylinder thickness the parametric sweep is run between 2 to 6 mm as given in Figure 7.5. The magnetic flux density saturates at the active pole when the value of thickness reaches to 3 mm. The final values of geometric variables of 3rd prototype of MR damper are presented in Table 7.1.

Table 7.1: Final geometric variables of 3rd prototype of MR damper

Parameter	Value
Piston Diameter (mm)	46
Outer Diameter (mm)	56
Cylinder Wall Thickness (mm)	3
Stroke (mm)	± 22.5

The three dimensional CAD model of 3rd prototype of MR damper is presented in Figure 7.6. The total diameter of the MR damper is decreased to 56 mm. However the length of stroke is fixed to ± 22.5 mm. The bush which should be flexible in nature is used in the connecting ends of the MR damper which helps in its easy installation inside washing machine. A copper wire is used for the double coil arrangement of the MR damper. The MR damper's cylinder and piston are made from low carbon steel respectively. The MR damper's piston rod and covers are made from aluminum. The MR fluid is retained in the thick sponge which is used to cover the piston of the MR damper as well as it is used to hold the MR fluid at the area of poles.

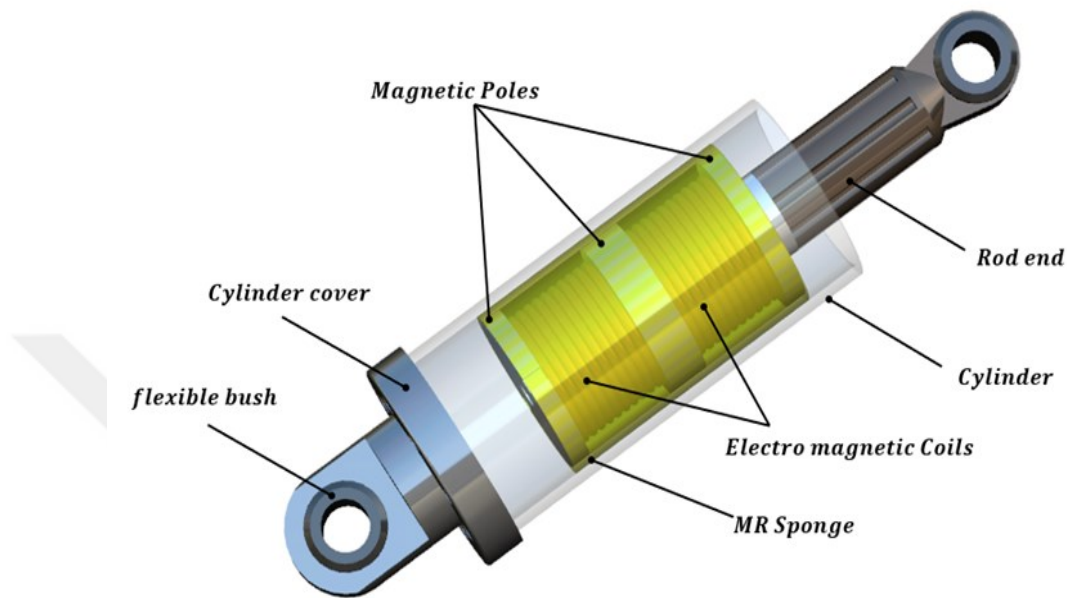


Figure 7.6: Three dimensional CAD model of 3rd prototype of MR damper.

An experimental test-bed is made for the testing of 3rd prototype of MR damper just like the previous two prototypes of MR damper discussed before. The linear actuator and force sensor are used to test the force generated by the manufactured 3rd prototype MR damper. A 110 N force is observed at the supplied current of 0.3 A as shown in Figure 7.7. In addition to this, a maximum force of 165 N is measured at 0.5 A current as given in Figure 7.8. Furthermore, the force of 3rd prototype of MR damper is analyzed in the On and Off states by changing the values of electric current as shown in Figure 7.9. Moreover, a passive force of 15 N is recorded for the 3rd prototype of MR damper.

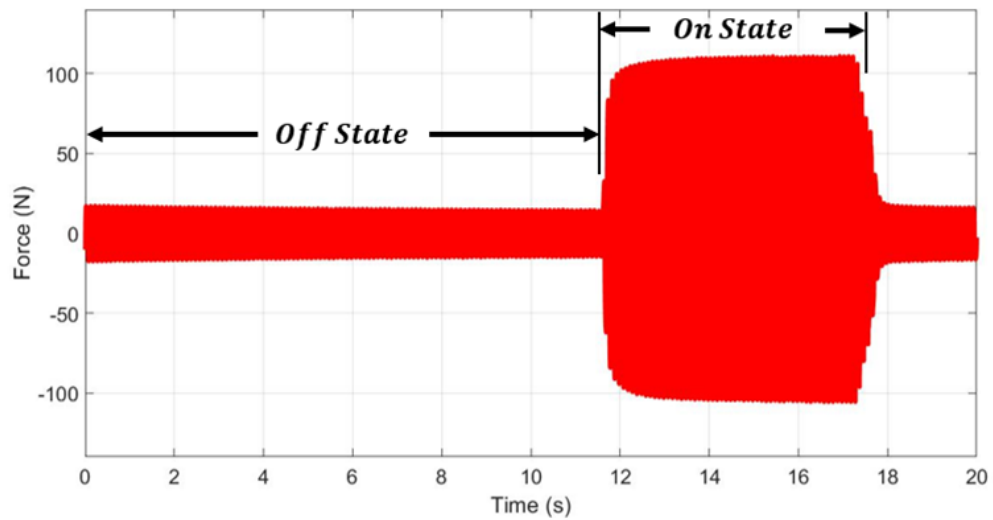


Figure 7.7: Measurement of force at 0.3 A for the 3rd prototype of MR damper.

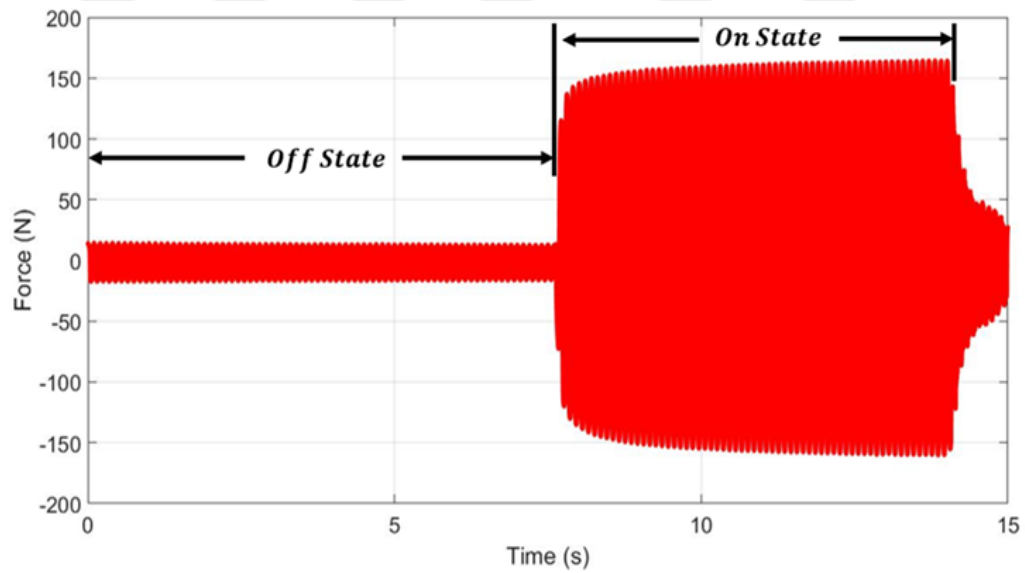


Figure 7.8: Measurement of force at 0.5 A for the 3rd prototype of MR damper.

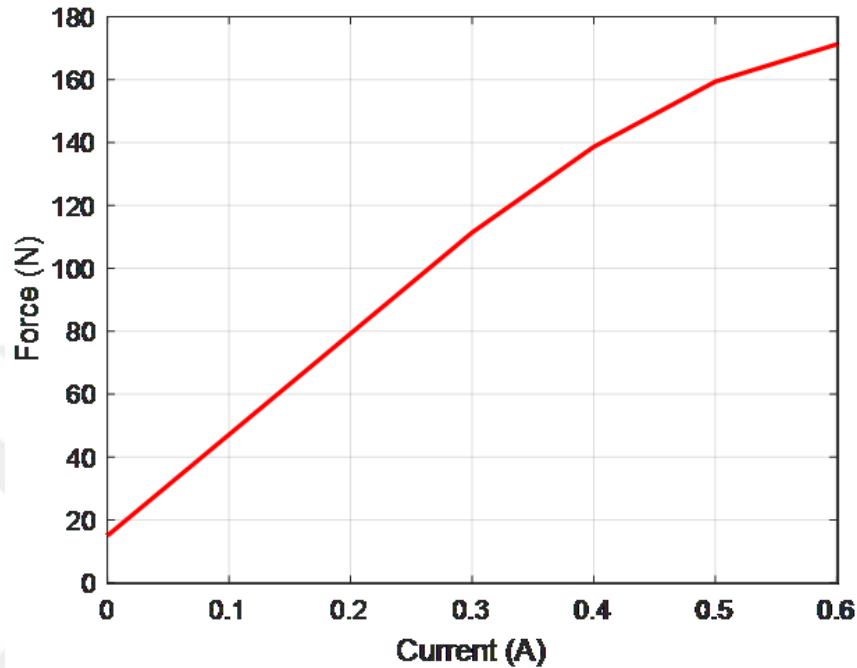


Figure 7.9: Experimental behavior of force with respect to electric current for the 3rd prototype of MR damper.

7.3 Vibration Control of Washing Machine

The designed adaptive control is implemented for the two new 3rd prototypes of MR damper of 110 N. The performance of the new MR dampers in the washing machine is investigated at different conditions as given in Figure 7.10. The vibrations are recorded by using passive dampers at the laundry mass of 8 kg for the full cycle. The new MR damper are fitted at the locations of passive dampers inside the washing machine. Both types of damper such as the new MR dampers and passive dampers are compared with each other in terms of its performance for various values of electric current as shown in Figure 7.11. The MR dampers are tested at each value of electric current and its corresponding performance is recorded in terms of displacement. It is clearly found that vibrations are minimized as the electric current rises from 0 to 0.6 A.



Figure 7.10: Two new 3rd prototypes of MR dampers are used in washing machine.

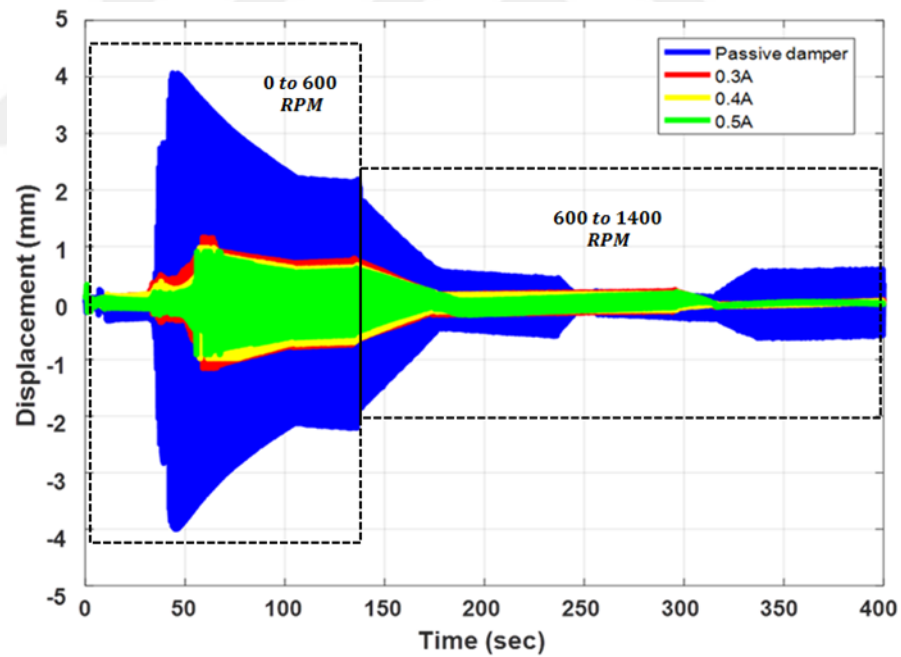


Figure 7.11: Vibrations measurement of washing machine by using 3rd prototypes of MR dampers.

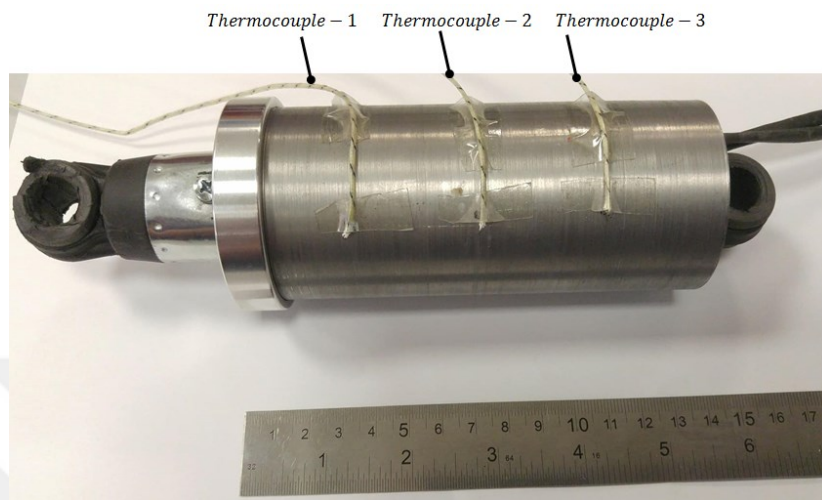


Figure 7.12: Three thermocouples are fixed to the cylinder of 3rd prototype of MR damper.

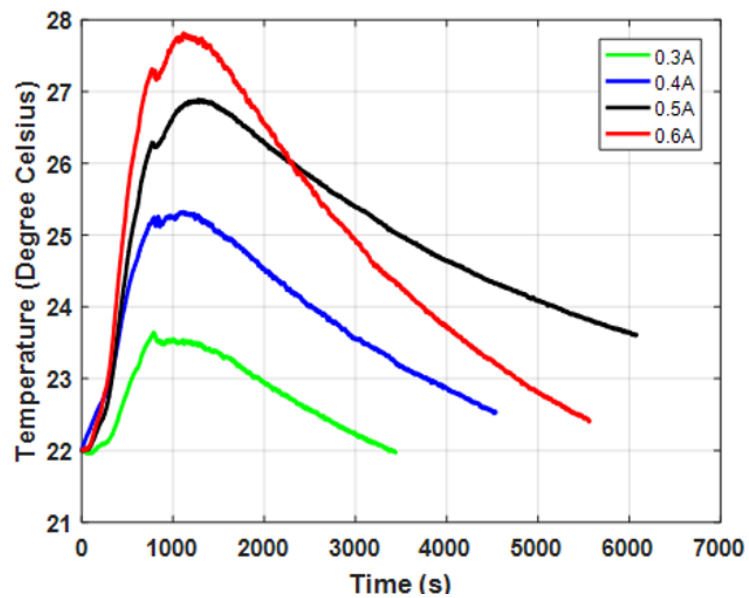


Figure 7.13: Behavior of temperature at the outside surface of MR damper's cylinder.

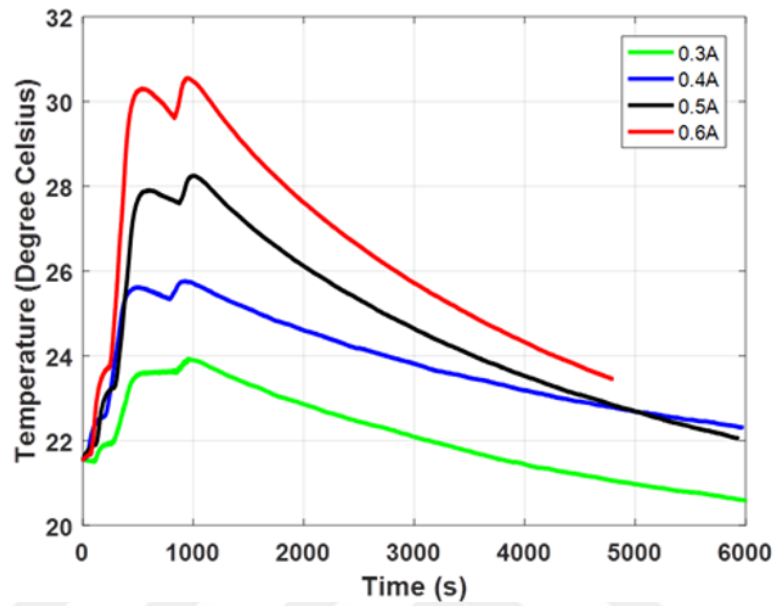


Figure 7.14: Behavior of temperature at the inside surface of MR damper's piston.

The temperature of the 3rd prototype of MR damper is measured using thermocouples. The thermocouples are fixed at cylinder's outer surface at three various places as shown in Figure 7.12. On the other hand, another thermocouple is used at the piston surface of MR damper. The temperature is measured at various values of electric current for both the surfaces of MR damper. Furthermore, the temperature is analyzed for the complete washing cycle at changing values of electric current. It is clearly seen from the recorded data of temperatures that, 27.81 °C is measured for the cylinder's outer surface as shown in Figure 7.13. Moreover, 30.54 °C is recorded for the piston surface as given in Figure 7.14.

7.4 Summary

The chapter focuses on the design and testing of 3rd prototype of MR damper for the washing machine. The new MR dampers successfully give 110 N force at 0.3 A. The size of the MR dampers are more compact comparing to 1st and 2nd prototype of MR dampers. The whole MR damper is studied using finite element analysis technique. The new MR dampers are

tested in the washing machine along with the adaptive control. The MR dampers are analyzed at various values of electric current and its vibrations are evaluated and compared with passive dampers. The vibrations are significantly reduced by using new MR dampers. Besides this, the temperature of 3rd prototype of MR dampers is also analyzed with by using thermocouples for the complete cycle of washing machine.



Chapter 8

CONCLUSION

Magneto Rheological dampers are utilized in many engineering applications such as civil structures, bridges, automobile suspension, seat suspension, prosthesis, reduction of vibrations in different shock absorbers. These dampers have wide range of advantages in terms of its dynamics and versatility. They are cost effective as well as giving its optimal performance in terms of electric current and power consumption. They provides variable damping which is very important for the performance of different applications which require the use of MR dampers.

MR dampers can be manufactured in variety of ways depending upon the desired application. The design of MR damper is solely dependent on the modes of MR fluid which is used in the MR damper. The orientation of the particles of MR fluid decides the characteristics of MR damper. The MR fluid is operated in shear, flow, squeeze and pinch modes. Every mode has its own advantages and disadvantages depending upon the specific application. In this research, a shear mode based MR damper is designed and manufactured for the vibrations attenuation in washing machine.

The design of shear mode MR damper is very important for the targeted value of magnetic flux density and damping force. The three dimensional CAD model is decided based on the optimization of geometric variables of the MR damper. For this task, a two dimensional axis symmetric model of MR damper is designed and built by using finite element analysis softwares. Each and every geometric variable is studied for the values of magnetic flux density and damping force. The magnetic circuit is also studied using the reluctance method by taking different loops and then approximating it with the finite element analysis data.

This research work focuses on the design and development of three prototypes of MR damper. In the first prototype of MR damper, a single coil arrangement is used for the

targeted force value of 200 N. A low carbon steel is utilized for the cylinder and piston part of the MR damper, while the piston rod and covers of cylinder are made from aluminum. The winding of the coil is made using copper wire of particular wire gauge. A sponge is used in order to retain the MR fluid inside in it. The piston of the MR damper is covered with the sponge. The design is manufactured and tested for the desired magnetic flux density. However due to the size of the first prototype of MR damper, it is desired to reduce its size keeping in view the availability and space inside the washing machine. For this task, a second prototype of MR damper is manufactured for the same value of damping force and magnetic flux. However, this time a double coil configuration is introduced in the design of MR damper by keeping the length of the center pole twice of the length of outer poles. The winding of the coil is made in such a way so that one winding is in clockwise direction and another one is in anticlockwise direction. Besides this a third prototype of MR damper is also made and analyzed successfully.

The experimental setups are established for testing of MR dampers. Different sensors are used while measuring the force, displacement and velocity of MR damper. These experimental results are used for studying the behavior of hysteresis of MR dampers. In addition to this, the dissipation of energy and equivalent coefficient damping is also experimentally found. The behavior of force of MR damper with respect to electric current is also determined experimentally and validated with the analytical and finite element analysis models. This research also focuses on testing different samples of MR fluid synthesized in the Koç University chemistry laboratory. The samples of MR fluid are synthesized and are tested in the prototypes of MR damper in order to compare its performance with commercial MR fluid. The performance of best sample in terms of damping force for various values of current is selected and used in the MR damper for the vibration control of washing machine.

This thesis also work on the design of adaptive control of MR dampers for decreasing the vibrations inside washing machine. A controller is required in order to reduce the response time of MR damper inside washing machine. Different methods of controller has been proposed in literature, however very little work is done on the response time of MR damper. For this task, a sliding mode control theory is proposed for designing the controller for the MR damper. The sliding mode control defines a sliding surface and using that surface it finds the control input for the MR damper and whole system. This work explains in detail a terminal sliding mode control (TSMC) for the MR damper in washing machine. The TSMC is a nonlinear control technique used for the MR damper in order to make sure the performance of MR damper in terms of disturbances and unmodeled dynamics. An analytical model is built for the controller and implemented for the MR damper. A detailed simulation study is done for the controller and MR damper. The controller is tested in terms of step and sinusoidal disturbances. The electric current response of MR dampers tracks very well the desired values given to the controller as reference. Furthermore, the adaptive behavior of MR damper is realized by making the convergence parameter dynamic. The controller is successfully checked in terms of its robustness and the expression of frequency of switching is also derived.

The washing machine is tested with all three prototypes of MR damper. In the case of first prototype of MR damper, the vibrations are decreased comparing to passive dampers by using On/Off controller. However, the size is so bulky that it touches the panels of washing machine. Therefore, a second prototype of MR damper is tested along with the adaptive control for the minimization of vibrations. The vibrations are successfully decreased as compared to first prototype of MR damper. Furthermore, there is no issue of size with second prototype of MR damper as its size is very compact. Besides this, a third prototype is successfully implemented in washing machine. The vibrations are successfully measured at various values of electric current and compared with original passive dampers. The

temperature of the MR dampers are experimentally recorded for the complete washing cycle of washing machine.



BIBLIOGRAPHY

- [1] Grunwald, A., & Olabi, A. G. (2008). Design of magneto-rheological (MR) valve. *Sensors and Actuators A: Physical*, 148(1), 211-223.
- [2] Carlson, J. D., & Jolly, M. R. (2000). MR fluid, foam and elastomer devices. *mechatronics*, 10(4-5), 555-569.
- [3] Nguyen, Q. H., & Choi, S. B. (2012). Optimal design methodology of magnetorheological fluid based mechanisms. In *Smart Actuation and Sensing Systems-Recent Advances and Future Challenges*. InTech.
- [4] Avraam, M. T. (2009). MR-fluid brake design and its application to a portable muscular rehabilitation device. *Doktora Tezi, Université Libre De Bruxelles, Faculté De Sciences Appliquées*.
- [5] Ahamed, R., Choi, S. B., & Ferdous, M. M. (2018). A state of art on magneto-rheological materials and their potential applications. *Journal of Intelligent Material Systems and Structures*, 1045389X18754350.
- [6] Carlson, J. D. (2005). MR fluids and devices in the real world. *International Journal of Modern Physics B*, 19(07n09), 1463-1470.
- [7] Liu, B., Li, W. H., Kosasih, P. B., & Zhang, X. Z. (2006). Development of an MR-brake-based haptic device. *Smart materials and structures*, 15(6), 1960.
- [8] Ferdous, M. M., Rashid, M. M., Hasan, M. H., & Rahman, M. A. (2014). Optimal design of Magneto-Rheological damper comparing different configurations by finite element analysis. *Journal of mechanical science and technology*, 28(9), 3667-3677.
- [9] Parlak, Z., Engin, T., & Çallı, İ. (2012). Optimal design of MR damper via finite element analyses of fluid dynamic and magnetic field. *Mechatronics*, 22(6), 890-903.
- [10] Kwok, N. M., Ha, Q. P., Nguyen, T. H., Li, J., & Samali, B. (2006). A novel hysteretic model for magnetorheological fluid dampers and parameter identification using particle swarm optimization. *Sensors and Actuators A: Physical*, 132(2), 441-451.

-
- [11] Nguyen, Q. H., Choi, S. B., Lee, Y. S., & Han, M. S. (2009). An analytical method for optimal design of MR valve structures. *Smart Materials and Structures*, 18(9), 095032.
- [12] Wang, D. H., & Liao, W. H. (2005). Semiactive controllers for magnetorheological fluid dampers. *Journal of Intelligent Material Systems and Structures*, 16(11-12), 983-993.
- [13] Chrzan, M. J., & Carlson, J. D. (2001, July). MR fluid sponge devices and their use in vibration control of washing machines. In *Smart Structures and Materials 2001: Damping and Isolation* (Vol. 4331, pp. 370-379). International Society for Optics and Photonics.
- [14] Wereley, N. M., Cho, J. U., Choi, Y. T., & Choi, S. B. (2007). Magnetorheological dampers in shear mode. *Smart Materials and Structures*, 17(1), 015022.
- [15] Kim, K., Chen, Z., Yu, D., & Rim, C. (2016). Design and experiments of a novel magneto-rheological damper featuring bifold flow mode. *Smart Materials and Structures*, 25(7), 075004.
- [16] Alghamdi, A. A., & Olabi, A. G. (2012, July). Novel design concept of magnetorheological damper in squeeze mode. In *International conference on experimental mechanics* (Vol. 2607).
- [17] Carlson JD (2009) Magnetorheological fluids, in Smart Materials, CRC Press 2009, Taylor & Francis Group, LLC. eBook ISBN: 978-1-4200-4373-0
- [18] Hoyle, A., Arzanpour, S., & Shen, Y. (2010). A novel magnetorheological damper based parallel planar manipulator design. *Smart Materials and Structures*, 19(5), 055028.
- [19] Kim, K. J., Lee, C. W., & Koo, J. H. (2008). Design and modeling of semi-active squeeze film dampers using magneto-rheological fluids. *Smart Materials and Structures*, 17(3), 035006.
- [20] Olabi, A. G., & Grunwald, A. (2007). Design and application of magneto-rheological fluid. *Materials & design*, 28(10), 2658-2664.

-
- [21] Sung, K. G., Choi, S. B., Lee, H. G., Min, K. W., & Lee, S. H. (2005). Performance comparison of MR dampers with three different working modes: shear, flow and mixed mode. In *Electrorheological Fluids And Magnetorheological Suspensions (Ermr 2004)* (pp. 645-651).
- [22] Tse, T., & Chang, C. C. (2004). Shear-mode rotary magnetorheological damper for small-scale structural control experiments. *Journal of structural engineering*, 130(6), 904-911.
- [23] Wereley, N. M. (2008). Nondimensional Herschel—Bulkley analysis of magnetorheological and electrorheological dampers. *Journal of Intelligent Material Systems and Structures*, 19(3), 257-268.
- [24] Brigley, M., Choi, Y. T., Wereley, N. M., & Choi, S. B. (2007). Magnetorheological isolators using multiple fluid modes. *Journal of Intelligent Material Systems and Structures*, 18(12), 1143-1148.
- [25] Choi, S. B., Hong, S. R., Sung, K. G., & Sohn, J. W. (2008). Optimal control of structural vibrations using a mixed-mode magnetorheological fluid mount. *International Journal of Mechanical Sciences*, 50(3), 559-568.
- [26] Hong, S. R., Wereley, N. M., Choi, Y. T., & Choi, S. B. (2008). Analytical and experimental validation of a nondimensional Bingham model for mixed-mode magnetorheological dampers. *Journal of Sound and Vibration*, 312(3), 399-417.
- [27] Unsal, M. (2006). Semi-active vibration control of a parallel platform mechanism using magnetorheological damping. *University of florida*.
- [28] Yu, M., Dong, X. M., Choi, S. B., & Liao, C. R. (2009). Human simulated intelligent control of vehicle suspension system with MR dampers. *Journal of Sound and Vibration*, 319(3-5), 753-767.

-
- [29] Goncalves, F. D., & Carlson, J. D. (2009). An alternate operation mode for MR fluids—magnetic gradient pinch. In *Journal of Physics: Conference Series* (Vol. 149, No. 1, p. 012050). IOP Publishing.
- [30] Poynor, J. C. (2001). *Innovative designs for magneto-rheological dampers* (Doctoral dissertation, Virginia Tech).
- [31] Parlak, Z., Engin, T., & Şahin, İ. (2013). Optimal magnetorheological damper configuration using the Taguchi experimental design method. *Journal of Mechanical Design*, 135(8), 081008.
- [32] Avinash, B., Sundar, S. S., & Gangadharan, K. V. (2014). Experimental study of damping characteristics of air, silicon oil, magneto rheological fluid on twin tube damper. *Procedia Materials Science*, 5, 2258-2262.
- [33] Yang, G., Spencer Jr, B. F., Carlson, J. D., & Sain, M. K. (2002). Large-scale MR fluid dampers: modeling and dynamic performance considerations. *Engineering structures*, 24(3), 309-323.
- [34] Wang, Q., Ahmadian, M., & Chen, Z. (2014). A novel double-piston magnetorheological damper for space truss structures vibration suppression. *Shock and Vibration*, 2014.
- [35] Sohn, J. W., Oh, J. S., & Choi, S. B. (2015). Design and novel type of a magnetorheological damper featuring piston bypass hole. *Smart Materials and Structures*, 24(3), 035013.
- [36] Hong, H., Tang, S., Sheng, Y., & Cui, Q. (2015, August). Magnetic circuit design and computation of a magnetorheological damper with exterior coil. In *Mechatronics and Automation (ICMA), 2015 IEEE International Conference on* (pp. 60-64). IEEE.
- [37] Ali, S. F., & Ramaswamy, A. (2009). Testing and modeling of MR damper and its application to SDOF systems using integral backstepping technique. *Journal of Dynamic Systems, Measurement, and Control*, 131(2), 021009.

-
- [38] Hu, G., Liu, F., Xie, Z., & Xu, M. (2016). Design, analysis, and experimental evaluation of a double coil magnetorheological fluid damper. *Shock and Vibration*, 2016.
- [39] Singh, H. J., & Wereley, N. M. (2014). Optimal control of gun recoil in direct fire using magnetorheological absorbers. *Smart materials and Structures*, 23(5), 055009.
- [40] Bai, X. X., Wereley, N. M., & Hu, W. (2015). Maximizing semi-active vibration isolation utilizing a magnetorheological damper with an inner bypass configuration. *Journal of Applied Physics*, 117(17), 17C711.
- [41] Rashid, M. M., Ferdaus, M. M., Hasan, M. H., & Rahman, A. (2015). ANSYS finite element design of an energy saving magneto-rheological damper with improved dispersion stability. *Journal of mechanical science and technology*, 29(7), 2793-2802.
- [42] Gravatt, J. W. (2003). *Magneto-rheological dampers for super-sport motorcycle applications* (Doctoral dissertation, Virginia Tech).
- [43] Lee, H. S., & Choi, S. B. (2000). Control and response characteristics of a magneto-rheological fluid damper for passenger vehicles. *Journal of Intelligent Material Systems and Structures*, 11(1), 80-87.
- [44] Prabakar, R. S., Sujatha, C., & Narayanan, S. (2009). Optimal semi-active preview control response of a half car vehicle model with magnetorheological damper. *Journal of sound and vibration*, 326(3-5), 400-420.
- [45] Rahman, M., Ong, Z. C., Julai, S., Ferdaus, M. M., & Ahamed, R. (2017). A review of advances in magnetorheological dampers: their design optimization and applications. *Journal of Zhejiang University-SCIENCE A*, 18(12), 991-1010.
- [46] Song, X., Ahmadian, M., & Southward, S. C. (2005). Modeling magnetorheological dampers with application of nonparametric approach. *Journal of intelligent material systems and structures*, 16(5), 421-432.

-
- [47] Stewart, B. B. (2015). Application of magnetorheological dampers in motorcycle rear swing arm suspension. *Cullowhee, NC: Western Carolina University*.
- [48] Yao, G. Z., Yap, F. F., Chen, G., Li, W., & Yeo, S. H. (2002). MR damper and its application for semi-active control of vehicle suspension system. *Mechatronics*, 12(7), 963-973.
- [49] Wang, J., & Meng, G. (2001). Magnetorheological fluid devices: principles, characteristics and applications in mechanical engineering. *Proceedings of the Institution of Mechanical Engineers, Part L: Journal of Materials: Design and Applications*, 215(3), 165-174.
- [50] Tang, X., Zhang, X., & Tao, R. (2000). Flexible fixture device with magneto-rheological fluids. In *Electro-Rheological Fluids And Magneto-Rheological Suspensions* (pp. 700-708).
- [51] Chew, C. M., Hong, G. S., & Zhou, W. (2006). Series damper actuator system based on MR fluid damper. *Robotica*, 24(6), 699-710.
- [52] Ashour, O., Rogers, C. A., & Kordonsky, W. (1996). Magnetorheological fluids: materials, characterization, and devices. *Journal of intelligent material systems and structures*, 7(2), 123-130.
- [53] Neelakantan, V. A., & Washington, G. N. (2005). Modeling and reduction of centrifuging in magnetorheological (MR) transmission clutches for automotive applications. *Journal of Intelligent Material Systems and Structures*, 16(9), 703-711.
- [54] Choi, K. M., Jung, H. J., Cho, S. W., & Lee, I. W. (2007). Application of smart passive damping system using MR damper to highway bridge structure. *Journal of mechanical science and technology*, 21(6), 870-874.
- [55] Yun, Y. W., Bae, H. S., & Park, M. K. (2010). A study of the control of the blank holding force using an MR damper in a drawing press. *Journal of mechanical science and technology*, 24(11), 2281-2288.

-
- [56] Kim, T. H., Bae, J. S., & Shul, C. W. (2016). Investigation on the parameter determination and implementation of a dual-pulse shock test machine using a MR damper. *Journal of Mechanical Science and Technology*, 30(12), 5381-5387.
- [57] Yun, Y. W., Lee, S. M., & Park, M. K. (2010). A study on the efficiency improvement of a passive oil damper using an MR accumulator. *Journal of mechanical science and technology*, 24(11), 2297-2305.
- [58] Shin, Y. J., You, W. H., Hur, H. M., & Park, J. H. (2012). Semi-active control to reduce carbody vibration of railway vehicle by using scaled roller rig. *Journal of mechanical science and technology*, 26(11), 3423-3431.
- [59] Kamalakkannan, K., Elayaperumal, A., & Managlamam, S. (2013). Input advanced control of semi active half car heave model. *Journal of Mechanical Science and Technology*, 27(5), 1225-1231.
- [60] Aguib, S., Nour, A., Djedid, T., Bossis, G., & Chikh, N. (2016). Forced transverse vibration of composite sandwich beam with magnetorheological elastomer core. *Journal of Mechanical Science and Technology*, 30(1), 15-24.
- [61] Liu, H., Chen, M., Yu, B., & Fang, Z. (2016). Configuration design and accuracy analysis of a novel magnetorheological finishing machine tool for concave surfaces with small radius of curvature. *Journal of Mechanical Science and Technology*, 30(7), 3301-3311.
- [62] Poznic, A., Miloradovic, D., & Juhas, A. (2017). A new magnetorheological brakes combined materials design approach. *Journal of Mechanical Science and Technology*, 31(3), 1119-1125.
- [63] Seo, J. U., Yun, Y. W., & Park, M. K. (2011). Magneto-rheological accumulator for temperature compensation in hydropneumatic suspension systems. *Journal of mechanical science and technology*, 25(6), 1621.

-
- [64] Sayyaadi, H., & Zareh, S. H. (2017). Intelligent control of an MR prosthesis knee using of a hybrid self-organizing fuzzy controller and multidimensional wavelet NN. *Journal of Mechanical Science and Technology*, 31(7), 3509-3518.
- [65] Mohammadi, N., Mahjoob, M. J., Kaffashi, B., & Malakooti, S. (2010). An experimental evaluation of pre-yield and post-yield rheological models of magnetic field dependent smart materials. *Journal of mechanical science and technology*, 24(9), 1829-1837.
- [66] Cho, S. W., Koo, J. H., Jo, J. S., & Lee, I. W. (2007). Vibration control of a cable-stayed bridge using electromagnetic induction based sensor integrated MR dampers. *Journal of mechanical science and technology*, 21(6), 875-880.
- [67] Zheng, E., Fan, Y., Zhu, R., Zhu, Y., & Xian, J. (2016). Prediction of the vibration characteristics for wheeled tractor with suspended driver seat including air spring and MR damper. *Journal of Mechanical Science and Technology*, 30(9), 4143-4156.
- [68] Sun, S. S., Ning, D. H., Yang, J., Du, H., Zhang, S. W., Li, W. H., & Nakano, M. (2017). Development of an MR seat suspension with self-powered generation capability. *Smart Materials and Structures*, 26(8), 085025.
- [69] Hiemenz GJ, Hu W and Wereley NM (2008) Semi-active magnetorheological helicopter crew seat suspension for vibration isolation. *Journal of Aircraft* 45: 945–953.
- [70] Chae, H. D., & Choi, S. B. (2014). A new vibration isolation bed stage with magnetorheological dampers for ambulance vehicles. *Smart Materials and Structures*, 24(1), 017001.
- [71] J. D. Carlson, W. Matthis, and J. R. Toscano, “Smart prosthetics based on magnetorheological fluids,” *Proc. SPIE*, vol. 4332, pp. 308–316, 2001.
- [72] Tomori, H., Midorikawa, Y., & Nakamura, T. (2013). Vibration control of an artificial muscle manipulator with a magnetorheological fluid brake. In *Journal of Physics: Conference Series* (Vol. 412, No. 1, p. 012053). IOP Publishing.

-
- [73] Dong, S., Lu, K. Q., Sun, J. Q., & Rudolph, K. (2006). Adaptive force regulation of muscle strengthening rehabilitation device with magnetorheological fluids. *IEEE Transactions on neural systems and rehabilitation engineering*, 14(1), 55-63.
- [74] Case, D., Taheri, B., & Richer, E. (2014). Dynamical modeling and experimental study of a small-scale magnetorheological damper. *IEEE/ASME Transactions on mechatronics*, 19(3), 1015-1024.
- [75] Kong, L., Chin, J. H., Li, Y., Lu, Y., & Li, P. (2014). Targeted suppression of vibration in deep hole drilling using magneto-rheological fluid damper. *Journal of Materials Processing Technology*, 214(11), 2617-2626.
- [76] D. Mei, T. Kong, A. J. Shih, and Z. Chen, "Magnetorheological fluid-controlled boring bar for chatter suppression," *J. Mater. Process. Technol.*, vol. 209, no. 4, pp. 1861–1870, 2009.
- [77] Spencer Jr, B. F., Yang, G., Carlson, J. D., & Sain, M. K. (1998, June). Smart dampers for seismic protection of structures: a full-scale study. In *Second World Conference on Structural Control, Kyoto* (pp. 417-426).
- [78] Choi, S. B., Hong, S. R., Sung, K. G., & Sohn, J. W. (2008). Optimal control of structural vibrations using a mixed-mode magnetorheological fluid mount. *International Journal of Mechanical Sciences*, 50(3), 559-568.
- [79] Choi, S. B., Nam, M. H., & Lee, B. K. (2000). Vibration control of a MR seat damper for commercial vehicles. *Journal of Intelligent Material Systems and Structures*, 11(12), 936-944.
- [80] Yao, G. Z., Yap, F. F., Chen, G., Li, W., & Yeo, S. H. (2002). MR damper and its application for semi-active control of vehicle suspension system. *Mechatronics*, 12(7), 963-973.
- [81] Liao, W. H., & Lai, C. Y. (2002). Harmonic analysis of a magnetorheological damper for vibration control. *Smart Materials and Structures*, 11(2), 288.

-
- [82] Ahmadian, M., Song, X., & Southward, S. C. (2004). No-jerk skyhook control methods for semiactive suspensions. *Journal of vibration and acoustics*, 126(4), 580-584.
- [83] Dong, X. M., Yu, M., Liao, C. R., & Chen, W. M. (2010). Comparative research on semi-active control strategies for magneto-rheological suspension. *Nonlinear dynamics*, 59(3), 433-453.
- [84] Lee, D. Y., Nam, Y. J., Yamane, R., & Park, M. K. (2009). Performance evaluation on vibration control of MR landing gear. In *Journal of Physics: Conference Series* (Vol. 149, No. 1, p. 012068). IOP Publishing.
- [85] Cho, J., Kim, H. Y., Jung, T., & Joh, J. (2007, January). Fuzzy control of shear-mode MR damper under impact force. In *Smart Structures, Devices, and Systems III* (Vol. 6414, p. 64140O). International Society for Optics and Photonics.
- [86] Karl J. Astrom, Bjorn Wittenmark 2008 Adaptive control, second edition Dover publications, INC. Mineola, New York.
- [87] Choi, S. B., Lee, H., Hong, S. R., & Cheong, C. (2000, June). Control and response characteristics of a magnetorheological fluid damper for passenger vehicles. In *Smart Structures and Materials 2000: Smart Structures and Integrated Systems* (Vol. 3985, pp. 438-444). International Society for Optics and Photonics.
- [88] Tyan, F. (2008, December). H_∞ -PD controller for suspension systems with MR dampers. In *Decision and Control, 2008. CDC 2008. 47th IEEE Conference on* (pp. 4408-4413). IEEE.
- [89] Kasemi, B., Muthalif, A. G., Rashid, M. M., & Fathima, S. (2012). Fuzzy-PID controller for semi-active vibration control using magnetorheological fluid damper. *Procedia Engineering*, 41, 1221-1227.
- [90] Oh, H. U., & Onoda, J. (2002). An experimental study of a semiactive magnetorheological fluid variable damper for vibration suppression of truss structures. *Smart Materials and Structures*, 11(1), 156.

-
- [91] Rajamohan, V., Sedaghati, R., & Rakheja, S. (2011). Optimal vibration control of beams with total and partial MR-fluid treatments. *Smart materials and structures*, 20(11), 115016.
- [92] Pohoryles, D. A., & Duffour, P. (2015). Adaptive control of structures under dynamic excitation using magnetorheological dampers: an improved clipped-optimal control algorithm. *Journal of Vibration and Control*, 21(13), 2569-2582.
- [93] Yokoyama, M., Hedrick, J. K., & Toyama, S. (2001). A model following sliding mode controller for semi-active suspension systems with MR dampers. In *American Control Conference, 2001. Proceedings of the 2001* (Vol. 4, pp. 2652-2657). IEEE.
- [94] Yokoyama, M., Hedrick, J. K., & Toyama, S. (2001). A model following sliding mode controller for semi-active suspension systems with MR dampers. In *American Control Conference, 2001. Proceedings of the 2001* (Vol. 4, pp. 2652-2657). IEEE.
- [95] Park, E. J., Stoikov, D., da Luz, L. F., & Suleman, A. (2006). A performance evaluation of an automotive magnetorheological brake design with a sliding mode controller. *Mechatronics*, 16(7), 405-416.
- [96] Li, L. Y., Song, G., & Ou, J. P. (2008). Observer and controller design for vibration suppression of a structure with MR brakes as nonlinear hinges. In *Earth & Space 2008: Engineering, Science, Construction, and Operations in Challenging Environments* (pp. 1-10).
- [97] Phu, D. X., Quoc, N. V., Park, J. H., & Choi, S. B. (2014). Design of a novel adaptive fuzzy sliding mode controller and application for vibration control of magnetorheological mount. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, 228(13), 2285-2302.
- [98] Hemmatian, M., & Ohadi, A. (2013). Sliding mode control of flexible rotor based on estimated model of magnetorheological squeeze film damper. *Journal of Vibration and Acoustics*, 135(5), 051023.

-
- [99] Ali, S. F., & Ramaswamy, A. (2009). Optimal fuzzy logic control for MDOF structural systems using evolutionary algorithms. *Engineering Applications of Artificial Intelligence*, 22(3), 407-419.
- [100] Titli, A., Roukieh, S., & Dayre, E. (1993, December). Three control approaches for the design of car semi-active suspension (optimal control, variable structure control, fuzzy control). In *Decision and Control, 1993., Proceedings of the 32nd IEEE Conference on* (pp. 2962-2963). IEEE.
- [101] Zhou, Y. (2015). Fuzzy semi-active control and analysis of wind-induced vibration of a ship lift. *Materials and Structures*, 48(10), 3307-3316.
- [102] Rigatos, G. G. (2009). Model-based and model-free control of flexible-link robots: A comparison between representative methods. *Applied Mathematical Modelling*, 33(10), 3906-3925.
- [103] Morlacchi, M., Resta, F., Ripamonti, F., & Tomasini, G. (2013, April). An adaptive non-model-based control strategy for smart structures vibration suppression. In *Active and Passive Smart Structures and Integrated Systems 2013* (Vol. 8688, p. 86882J). International Society for Optics and Photonics.
- [104] Al-Dawod, M., Samali, B., & Li, J. (2006). Experimental verification of an active mass driver system on a five-storey model using a fuzzy controller. *Structural Control and Health Monitoring: The Official Journal of the International Association for Structural Control and Monitoring and of the European Association for the Control of Structures*, 13(5), 917-943.
- [105] Soleymani, M., & Khodadadi, M. (2014). Adaptive fuzzy controller for active tuned mass damper of a benchmark tall building subjected to seismic and wind loads. *The Structural Design of Tall and Special Buildings*, 23(10), 781-800.

-
- [106] Hu, Q., Ma, G., & Xie, L. (2008). Robust and adaptive variable structure output feedback control of uncertain systems with input nonlinearity. *Automatica*, 44(2), 552-559.
- [107] Ma, T. W., & Yang, H. T. (2004). Adaptive feedback–feedforward control of building structures. *Journal of engineering mechanics*, 130(7), 786-793.
- [108] Fei, J., & Fang, Y. (2010). Robust adaptive feedforward vibration control for: Flexible structure. *International Journal of Innovative Computing, Information and Control*, 6(5), 2189-2197.
- [109] Hu, Y. R., & Ng, A. (2005). Active robust vibration control of flexible structures. *Journal of sound and vibration*, 288(1-2), 43-56.
- [110] Chen, Y., Ulker, F. D., Wickramasinghe, V., & Zimcik, D. G. (2014). Development of robust control law for active buffeting load alleviation of smart fin structures. *Journal of Intelligent Material Systems and Structures*, 25(7), 818-831.
- [111] Fallah, A. Y., & Taghikhany, T. (2014). Robust semi-active control for uncertain structures and smart dampers. *Smart Materials and Structures*, 23(9), 095040.
- [112] Ackermann, J. (2012). *Robust control: Systems with uncertain physical parameters*. Springer Science & Business Media.
- [113] Åström, K. J., Neumann, L., & Gutman, P. O. (1987). A comparison between robust and adaptive control of uncertain systems. In *Adaptive Systems in Control and Signal Processing 1986* (pp. 43-48).
- [114] Prempain, E., & Postlethwaite, I. (2008). L2 and H2 performance analysis and gain-scheduling synthesis for parameter-dependent systems. *Automatica*, 44(8), 2081-2089.
- [115] McKernan, A., Sala, A., Ariño, C., & Irwin, G. W. (2009). Sampled-data gain scheduling of continuous LTV plants. *Automatica*, 45(10), 2451-2453.

-
- [116] Körlin, R., Boonto, S., Werner, H., & Starossek, U. (2012). LMI-based gain scheduling for bridge flutter control using eccentric rotational actuators. *Optimal Control Applications and Methods*, 33(4), 488-500.
- [117] Utkin, V. (1977). Variable structure systems with sliding modes. *IEEE Transactions on Automatic control*, 22(2), 212-222.
- [118] Yang, J. N., Wu, J. C., & Agrawal, A. K. (1995). Sliding mode control for seismically excited linear structures. *Journal of engineering mechanics*, 121(12), 1386-1390.
- [119] Matheu, E. E., Singh, M. P., & Beattie, C. (1998). Output feedback sliding-mode control with generalized sliding surface for civil structures under earthquake excitation. *Earthquake engineering & structural dynamics*, 27(3), 259-282.
- [120] Ikhouane, F., & Krstic, M. (1998). Robustness of the tuning functions adaptive backstepping design for linear systems. *IEEE transactions on Automatic Control*, 43(3), 431-437.
- [121] Song, X., Ahmadian, M., Southward, S., & Miller, L. R. (2005). An adaptive semiactive control algorithm for magnetorheological suspension systems. *Journal of vibration and acoustics*, 127(5), 493-502.
- [122] Phu, D. X., Quoc, N. V., Park, J. H., & Choi, S. B. (2014). Design of a novel adaptive fuzzy sliding mode controller and application for vibration control of magnetorheological mount. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, 228(13), 2285-2302.
- [123] Haiyan, G. D. H. (2002). Semi-Active Control of Vehicle Suspensions Based on Magnetorheological Damper: Modeling and Directive Adaptive Control [J]. *Journal of Vibration Engineering*, 1, 001.
- [124] Haiyan, G. D. H. (2002). Semi-Active Control of Vehicle Suspensions Based on Magnetorheological Damper Part II: Indirect Adaptive Control and Experiments [J]. *Journal of Vibration Engineering*, 3, 006.

-
- [125] Laflamme, S., Slotine, J. J. E., & Connor, J. J. (2011). Wavelet network for semi-active control. *Journal of Engineering Mechanics*, 137(7), 462-474.
- [126] Tong, S., Wang, T., & Tang, J. T. (2000). Fuzzy adaptive output tracking control of nonlinear systems. *Fuzzy Sets and Systems*, 111(2), 169-182.
- [127] Chen, J. Z., & Liao, W. H. (2010). Design, testing and control of a magnetorheological actuator for assistive knee braces. *Smart Materials and Structures*, 19(3), 035029.
- [128] Chen, J., & Liao, W. H. (2008, April). Development and testing of a magnetorheological actuator for an assistive knee brace. In *Active and Passive Smart Structures and Integrated Systems 2008* (Vol. 6928, p. 69281G). International Society for Optics and Photonics.
- [129] Chen, J., & Liao, W. H. (2007, December). Design and control of a magnetorheological actuator for leg exoskeleton. In *Robotics and Biomimetics, 2007. ROBIO 2007. IEEE International Conference on* (pp. 1388-1393). IEEE.
- [130] Guo, H. T., & Liao, W. H. (2010, April). Design and control of multifunctional magnetorheological actuators for assistive knee braces. In *Active and Passive Smart Structures and Integrated Systems 2010* (Vol. 7643, p. 764310). International Society for Optics and Photonics.
- [131] Komurcugil, H. (2012). Adaptive terminal sliding-mode control strategy for DC–DC buck converters. *ISA transactions*, 51(6), 673-681.
- [132] Chen, H. W., Ji, W. X., Zhang, Q. J., Cao, Y., & Fan, S. Y. (2012). A method for vibration isolation of a vertical axis automatic washing machine with a hydraulic balancer. *Journal of Mechanical Science and Technology*, 26(2), 335-343.
- [133] Jung, C. H., Kim, C. S., & Choi, Y. H. (2008). A dynamic model and numerical study on the liquid balancer used in an automatic washing machine. *Journal of mechanical science and technology*, 22(9), 1843-1852.

-
- [134] Bae, S., Lee, J. M., Kang, Y. J., Kang, J. S., & Yun, J. R. (2002). Dynamic analysis of an automatic washing machine with a hydraulic balancer. *Journal of Sound and vibration*, 257(1), 3-18.
- [135] Papadopoulos, E., & Papadimitriou, I. (2001). Modeling, design and control of a portable washing machine during the spinning cycle. In *Advanced Intelligent Mechatronics, 2001. Proceedings. 2001 IEEE/ASME International Conference on* (Vol. 2, pp. 899-904). IEEE.
- [136] Aydar, G., Evrensel, C. A., Gordaninejad, F., & Fuchs, A. (2007). A low force magneto-rheological (MR) fluid damper: design, fabrication and characterization. *Journal of Intelligent Material Systems and Structures*, 18(12), 1155-1160.
- [137] Aydar, G. (2007). *A new magneto-rheological fluid (MRF) washing machine damper and a novel two-way-controllable MRF valve* (Doctoral dissertation, University of Nevada, Reno).
- [138] Cha, S. T., & Baek, W. K. (2013). Vibration Attenuation of a Drum-typed Washing Machine Using Magneto-Rheological Dampers. *Journal of the Korea Society For Power System Engineering*, 17(2), 63-69.
- [139] Nguyen, Q. H., Choi, S. B., & Woo, J. K. (2014). Optimal design of magnetorheological fluid-based dampers for front-loaded washing machines. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, 228(2), 294-306.
- [140] Tyan, F., Chao, C. T., Tu, S. H., & Cooperation, S. L. (2009, November). Modeling and vibration control of a drum-type washing machine via MR fluid dampers. In *Proceedings of 2009 CACS International Automatic Control Conference, National Taipei University of Technology, Taipei, Taiwan*.

-
- [141] Nguyen, Q. H., Nguyen, N. D., & Choi, S. B. (2014). Optimal design and performance evaluation of a flow-mode MR damper for front-loaded washing machines. *Asia Pacific Journal on Computational Engineering*, 1(1), 3.
- [142] Carlson, J. D. (2000). *U.S. Patent No. 6,151,930*. Washington, DC: U.S. Patent and Trademark Office.
- [143] Minorowicz, B., & Stefanski, F. (2014). Proposal of a new group of magnetorheological dampers. *Przegląd Elektrotechniczny*, 7, 263-267.
- [144] Minorowicz, B., Stefański, F., Pittner, G., & Regulski, R. (2015). Share mode magnetorheological dampers for vibration attenuation in domestic washing machines. In *Progress in Automation, Robotics and Measuring Techniques* (pp. 147-156). Springer, Cham.
- [145] Spelta, C., Previdi, F., Savaresi, S. M., Fraternale, G., & Gaudiano, N. (2009). Control of magnetorheological dampers for vibration reduction in a washing machine. *Mechatronics*, 19(3), 410-421.
- [146] Spencer Jr, B. F., Dyke, S. J., Sain, M. K., & Carlson, J. (1997). Phenomenological model for magnetorheological dampers. *Journal of engineering mechanics*, 123(3), 230-238.
- [147] Butz, T., & Von Stryk, O. (2002). Modelling and Simulation of Electro-and Magnetorheological Fluid Dampers. *ZAMM-Journal of Applied Mathematics and Mechanics/Zeitschrift für Angewandte Mathematik und Mechanik: Applied Mathematics and Mechanics*, 82(1), 3-20.
- [148] Lim, H. T., Jeong, W. B., & Kim, K. J. (2010). Dynamic modeling and analysis of drum-type washing machine. *International Journal of Precision Engineering and Manufacturing*, 11(3), 407-417.

-
- [149] Boyraz, P., & Gündüz, M. (2013). Dynamic modeling of a horizontal washing machine and optimization of vibration characteristics using Genetic Algorithms. *Mechatronics*, 23(6), 581-593.
- [150] Yorukoglu, A., & Altug, E. (2009, July). Determining the mass and angular position of the unbalanced load in horizontal washing machines. In *Advanced Intelligent Mechatronics, 2009. AIM 2009. IEEE/ASME International Conference on* (pp. 118-123). IEEE.
- [151] Hassaan, G. A. (2015). Vibration analysis of a horizontal washing machine, Part I: Vibration displacement and velocity. *International Journal of Engineering and Advanced Research Technology*, 3(4).