Magneto-Rheological Actuators for Human-Safe Robots: Modeling, Control, and Implementation

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Graduate Program in Electrical and Computer Engineering
A thesis submitted in partial fulfillment of the requirements for the degree in Doctor of Philosophy
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MAGNETO-RHEOLOGICAL ACTUATORS FOR HUMAN-SAFE ROBOTS: MODELING, CONTROL, AND IMPLEMENTATION

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by

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Graduate Program in Electrical and Computer Engineering

A thesis submitted in partial fulfillment of the requirements for the degree of Doctor of Philosophy

The School of Graduate and Postdoctoral Studies The University of Western Ontario London, Ontario, Canada

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Abstract

In recent years, research on physical human-robot interaction has received considerable attention. Research on this subject has led to the study of new control and actuation mechanisms for robots in order to achieve intrinsic safety. Naturally, intrinsic safety is only achievable in kinematic structures that exhibit low output impedance. Existing solutions for reducing impedance are commonly obtained at the expense of reduced performance, or significant increase in mechanical complexity. Achieving high performance while guaranteeing safety seems to be a challenging goal that necessitates new actuation technologies in future generations of human-safe robots.

In this study, a novel two degrees-of-freedom safe manipulator is presented. The manipulator uses magneto-rheological fluid-based actuators. Magneto-rheological actuators offer low inertia-to-torque and mass-to-torque ratios which support their applications in human-friendly actuation. As a key element in the design of the manipulator, bi-directional actuation is attained by antagonistically coupling MR actuators at the joints. Antagonistically coupled MR actuators at the joints allow using a single motor to drive multiple joints. The motor is located at the base of the manipulator in order to further reduce the overall weight of the robot. Due to the unique characteristic of MR actuators, intrinsically safe actuation is achieved without compromising high quality actuation. Despite these advantages, modeling and control of MR actuators present some challenges. The antagonistic configuration of MR actuators may result in limit cycles in some cases when the actuator operates in the position control loop. To study the possibility of limit cycles, describing function method is employed to obtain the conditions under which limit cycles may occur in the operation of the system. Moreover, a connection between the amplitude and the frequency of the potential limit cycles and the system parameters is established to provide an insight into the design of the actuator as well as the controller. MR actuators require magnetic fields to control their output torques. The application of magnetic field however introduces hysteresis in the behaviors of MR actuators. To this effect, an adaptive model is developed to estimate the hysteretic behavior of the actuator. The effectiveness of the model is evaluated by comparing its results with those obtained using the Preisach model. These results are then extended to an adaptive control scheme in order to compensate for the effect of hysteresis. In both modeling and control, stability of proposed schemes are evaluated using Lyapunov method, and the effectiveness of the proposed methods are validated with experimental results.

Keywords: Human Safe Robots, Smart Actuators, Magneto-Rheological Fluids, Adaptive Modeling and Control, Nonlinear Systems.
Co-Authorship Statement

The work presented in this thesis involved in part collaboration with Alex Shafer who provided dimensional parameters of the Magneto-Rheological clutches used in implementation of the safe robot manipulator developed as part of my PhD studies. I would also like to acknowledge the contribution of University Machine Services’ staffs Ian J. Vinkenveugel and Josh Taylor in the design and manufacturing of the manipulator.
To my family for their love, inspiration, and guidance.
Acknowledgements

I would like to express my sincere appreciation and gratitude to Dr. Mehrdad Kermani for supervising my research during my Ph.D. studies. I would also like to thank my lab-mates, Wenjun Li, Mahdi Anooshahpour, Alex Shafer, Vahid Setoudeh Nejad, Nima Najmaei, and Dr. Ali Asadian for sharing their experience and knowledge with me.
# Contents

**Abstract** ii  
**Co-Authorship Statement** iii  
**Acknowledgements** v  
**List of Tables** x  
**List of Figures** xi  

## 1 Introduction 1  
1.1 Motivation 1  
1.2 Why are most existing industrial robots unsafe? 4  
1.3 Related Works: Human-Safe Robots 7  
1.3.1 Safeguarding and Collision Avoidance Using Sensors 7  
1.3.2 Lightweight Robots 8  
1.3.3 Collision Force Suppression 10  
  - Active Compliance 10  
  - Passive Compliance 11  
1.3.4 Variable Damper Actuators 13  
  - Friction Dampers 13  
  - Eddy Current Dampers 14  
  - Fluid Dynamics Dampers 14  
  - Magneto-Rheological and Electro-Rheological Dampers 15  
1.4 Objectives 16  
1.4.1 Main Contributions 16  
  - Development of a New Safe Robot 16  
  - Analysing Limit Cycles in Antagonistically Coupled MR Actuators 16  
  - Modeling Hysteresis in MR Actuators 17  
  - Hysteresis Compensation for MR Actuators 17
### 1.4.2 Thesis Outlines

<table>
<thead>
<tr>
<th>Thesis Outlines</th>
<th>17</th>
</tr>
</thead>
</table>

### Bibliography

<table>
<thead>
<tr>
<th>2</th>
<th>Design and Development of a New Single-Motor, 2-DOF Safe Robot</th>
<th>28</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Introduction</td>
<td>29</td>
</tr>
<tr>
<td>2.2</td>
<td>The State-of-the-Art Human-Safe Robots</td>
<td>31</td>
</tr>
<tr>
<td>2.3</td>
<td>Design and Development of A Two DOF prototype Manipulator</td>
<td>33</td>
</tr>
<tr>
<td>2.3.1</td>
<td>Actuation Concept</td>
<td>33</td>
</tr>
<tr>
<td>2.3.2</td>
<td>2–DOF Manipulator Design: Proof-Of-Concept</td>
<td>35</td>
</tr>
<tr>
<td>2.3.3</td>
<td>Remarks On Motor Velocity Control</td>
<td>37</td>
</tr>
<tr>
<td>2.4</td>
<td>Safety Analysis</td>
<td>38</td>
</tr>
<tr>
<td>2.5</td>
<td>Preliminary Experimental Results</td>
<td>40</td>
</tr>
<tr>
<td>2.6</td>
<td>Conclusion</td>
<td>45</td>
</tr>
</tbody>
</table>

| Bibliography | 46 |

<table>
<thead>
<tr>
<th>3</th>
<th>Study of Limit Cycle in Antagonistically Coupled Magneto-Rheological Actuators</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td>Introduction</td>
<td>51</td>
</tr>
<tr>
<td>3.2</td>
<td>Magneto-Rheological Fluid-Based Actuation</td>
<td>52</td>
</tr>
<tr>
<td>3.2.1</td>
<td>Models for MR Fluids</td>
<td>53</td>
</tr>
<tr>
<td>3.2.2</td>
<td>A-DASA Model</td>
<td>54</td>
</tr>
<tr>
<td>3.3</td>
<td>Limit Cycle Analysis</td>
<td>57</td>
</tr>
<tr>
<td>3.3.1</td>
<td>A Practical Example</td>
<td>58</td>
</tr>
<tr>
<td>3.3.2</td>
<td>Assumptions and Preliminaries</td>
<td>59</td>
</tr>
<tr>
<td>3.3.3</td>
<td>Analysis</td>
<td>61</td>
</tr>
<tr>
<td>3.3.4</td>
<td>Discussion</td>
<td>64</td>
</tr>
<tr>
<td>3.4</td>
<td>Experimental Validations and Simulation</td>
<td>65</td>
</tr>
<tr>
<td>3.4.1</td>
<td>Model-based Simulations</td>
<td>66</td>
</tr>
<tr>
<td>3.4.2</td>
<td>Preliminary Experiments</td>
<td>68</td>
</tr>
<tr>
<td>3.5</td>
<td>Conclusion</td>
<td>68</td>
</tr>
</tbody>
</table>

| Bibliography | 69 |

<table>
<thead>
<tr>
<th>4</th>
<th>Adaptive Modeling of Magneto-Rheological Actuators</th>
<th>73</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1</td>
<td>Introduction</td>
<td>74</td>
</tr>
<tr>
<td>4.2</td>
<td>Human-friendly Actuators</td>
<td>76</td>
</tr>
</tbody>
</table>
# Bibliography

## 5 Adaptive Control of Magneto-Rheological Actuators

5.1 Introduction ................................. 98
5.2 Magneto-Rheological Actuators .................. 100
5.3 MR Actuator Model ......................... 101
5.3.1 MR Clutch Magnetic Circuit Model .......... 102
5.3.2 MR Clutch Mechanical Model ............... 103
5.4 Torque Control for MR Clutch Based Actuation .... 106
5.5 Experimental Validation ...................... 109
5.5.1 Validation of Torque Estimation .......... 110
5.5.2 Torque Control Results .................... 112
5.5.3 Comparison Between Adaptive and PID Controllers . 114
5.6 Conclusion .................................. 115

## Bibliography

## 6 Conclusion and Future Works

6.1 Conclusion .................................. 123
6.2 Future Works .............................. 124
Reconsideration of active drives in the design of MR-actuators ........ 124
Safety analysis and experiments ........................................... 125
Safety-oriented control ...................................................... 126

## Bibliography

## A Injury Criteria

A.1 Safety Criteria .............................. 130
A.1.1 3 ms Criterion .............................. 130
A.1.2 Head Injury Criterion .................. 131
A.1.3 Viscous Criterion ....................... 131
A.1.4 Injury Criteria for the Neck .............. 132
List of Tables

1.1 Inertial parameters of several industrial robots. ......................... 6
1.2 Typical Motor Inertias [38] ................................................. 9
1.3 Gear ratios and actuator inertia in several industrial robots. .......... 9

2.1 2–DOF Safe Manipulator Characteristics ................................ 36
2.2 Specifications of the MR clutch ............................................. 36
2.3 Normalized Effective Masses ................................................ 40
2.4 Right Hand Characteristics of an Average US Male Civilian [32] ....... 40

3.1 Specifications of the A-DASA system ...................................... 58

4.1 RMSE values of the Magnetic field-current modeling and torque-current prediction ..................................................... 90

5.1 Specifications of the MR clutch ............................................... 110
5.2 RMSEs of Torque Estimations ............................................... 111
5.3 RMSEs of Torque Control Results .......................................... 113

A.1 Definition of the abbreviated injury scale ................................. 131
A.2 Acting force and torque limits specified for the human neck .......... 132
List of Figures

1.1 Demographic features of the developed countries [5] ........................................ 3
1.2 Mass-spring collision model. ............................................................................. 5
1.3 The maximum force in an impact. ...................................................................... 5

2.1 DASA Configuration. $G$ is the gear ratio. $J_r$ and $J_l$ represent rotor and link’s inertias, respectively. ................................................................. 33
2.2 Bingham model for MR clutches; a Coulomb friction element $f_c$ in parallel with a viscous damper $g_v$, where $q_m$ and $q_l$ are angular rotations of input and output shafts of the clutch, respectively, associated to the motor and link rotations. ............................... 34
2.3 Antagonistic DASA Configuration. $G$ is the gear ratio. $J_r$ and $J_l$ represent rotor and link’s inertias, respectively. .............................................. 34
2.4 Pluralized Antagonistic DASA Configuration. $G_r$ is the gear ratio. $J_{li}$, $i = 1, 2, \ldots, n$, represents i-th link’s inertia. ......................................................... 35
2.5 2–DOF safe manipulator. .................................................................................... 36
2.6 The set of gearbox and belt transmission for antagonistic actuation. .............. 37
2.7 The second generation of MR clutches. .............................................................. 37
2.8 Manipulators configurations for effective mass calculation. .......................... 39
2.9 (a) Effective Masses, (b) Zoomed-in by 4x ....................................................... 41
2.10 Manipulator Safety Index (MSI) ........................................................................ 42
2.11 Developed 2-DOF robot manipulator ............................................................... 42
2.12 Position tracking control results for square command with constant motor velocity ........................................................................................................... 43
2.13 Velocities of joints along with motor velocity with constant motor velocity. 44
2.14 Joint 1 position tracking control results for sinusoidal inputs. ...................... 44
2.15 Joint 2 position tracking control results for sinusoidal inputs. ...................... 45

3.1 DASA Configuration. $G$ is the gear ratio. $J_r$ and $J_l$ represent rotor and link’s inertias, respectively. ................................................................. 52
3.2 Antagonistic DASA Configuration. $G$ is the gear ratio. $J_r$ and $J_l$ represent rotor and link’s inertias, respectively.

3.3 Visco-plastic models of MR fluids.

3.4 Shear stress vs. shear rate in a sample MR fluid.

3.5 Biviscous model of MR fluids.

3.6 Cross-section of a multi-disk MR clutch.

3.7 Torque-velocity relationship of the A-DASA actuator.

3.8 Position control block diagram using A-DASA actuation.

3.9 Presence of limit cycles in position control of A-DASA actuator; (a) $k_p = 1$, $k_d = 0$, (b) $k_p = 2$, $k_d = 0.005$, (c) $k_p = 2.5$, $k_d = 0.005$, and (d) $k_p = 2.5$, $k_d = 0.01$. The circles show initial conditions.

3.10 Equivalent position control block diagram of the A-DASA actuation.

3.11 A snapshot of the MR clutch during fabrication and the 2-DOF MR-actuator robot manipulator.

3.12 (a) Simulated angular position and velocity of the 1st joint, (b) Phase portrait of the A-DASA in position control; $k_p = 0.1$, $k_d = 0$. The initial condition is marked by a circle.

3.13 (a) Simulated angular position and velocity of the 1st joint,(b) Phase portrait of the A-DASA in position control; $k_p = 0.5$, $k_d = 0$. The initial condition is marked by a circle.

3.14 Phase portrait of the A-DASA in position control; (a) $k_p = 0.5$, $k_d = 0.01$, (b) $k_p = 0.5$, $k_d = 0.026$. The initial condition is marked by a circle.

3.15 Experimental results using the 2-DOF robot; (a) $k_p = 1$, (b) $k_p = 2$.

4.1 The proposed modeling structure.

4.2 Cross-section of a multi-disk MR clutch.

4.3 Snapshots of the MR clutch and the actuation mechanism that use MR clutches as part of the actuator.

4.4 Block diagram of the proposed model for a MR based actuator.

4.5 Current-Magnetic field map for the MRF-based actuator.

4.6 Characteristics of the LORD MRF-140CG (Adopted from manufacturer datasheet [44]).

4.7 The prototype platform.

4.8 Measured magnetic field and output torque v.s. predicted values.

4.9 Estimated magnetic field and predicted output torque corresponding to Multi-Sinusoids input current.
4.10 Predicted magnetic field and torque values corresponding to exponentially decaying sinusoid input current. 87
4.11 A sample of uneven surface require grinding 88
4.12 Gridding application results; predicted magnetic field and torque values from the proposed model and the corresponding values from actual measurements and Preisach model. 88

5.1 Three operation modes of MR fluids; (a) flow mode, (b) direct shear mode, and (c) squeeze film mode, where $F, P, \varphi, D, \omega, \text{and} \ B$ represents force, pressure, flow, displacement, velocity, and the applied magnetic field, respectively. 100
5.2 A possible arrangement for robot joint actuation using an MR clutch. 101
5.3 Cross-section of a multi-disk MR clutch. 102
5.4 Magnetic circuit of an MR clutch. 102
5.5 Rate-dependent hysteresis for an MR clutch. 104
5.6 Characteristics of the LORD MRF-140CG (Adopted from manufacturer datasheet [48]). 105
5.7 MR clutch torque-current model. 106
5.8 A snapshot of the MR clutch during fabrication and the 2-DOF MR-actuator robot manipulator. 110
5.9 Actual torque measurements versus estimations. 111
5.10 Control block diagrams of the adaptive controller that uses magnetic field measurements and Bingham model and the PID controller that uses output torque measurements. 112
5.11 Torque control results for a sinusoidal desired torque trajectory. 113
5.12 Frequency spectrum of the Multi-Sinusoid desired torque trajectory. 114
5.13 Torque control results for a Multi-Sinusoid desired torque trajectory. 114
5.14 Control block diagram of adaptive and PID controllers based on output torque estimation using Bingham model. 115
5.15 Comparison of the adaptive and PID controllers in hysteresis compensation. 116

A.1 Taxonomy of neck motions. 133
Chapter 1

Introduction

1.1 Motivation

Robots will pervade our everyday life as well as the manufacturing environment in near future. According to International Federation of Robotics (IFR),[1] 3 million service robots were sold for domestic, personal, and professional use in 2012, 20% more than in 2011. Applications of service robots include medical, construction, cleaning, and rescue. It is predicted that sales of service robots will exceed 22 million units in the period of 2013-2016. Industrial robots has also gained in prevalence quickly as they help in streamlining manufacturing and mass production without compromising quality and precision. In 2012, sales for industrial robots grew by 4% to 159,346 units. The total accumulated sales amounted to be more than 2,460,000 units by the end of 2012, counted since the installation of the first industrial robots in 1961. Considering average 12 years of service life, the total stock of operational robots is estimated in the range of 1,235,000 and 1,500,000 units at the end of 2012. Sales for industrial robots is expected to grow by 6% per year till 2016. Comparing statistical data, the population of operational robots in industry has been increased 3 times in the last two decades, and has almost doubled in 10 years. Along this growth in the use of robots, economical factors including decreasing life-cycles of products, growing consumer markets, and efficiency call for more flexible and customizable automation. Whereas mass production was traditionally the main objective in industry, today’s demand for customization and shorter life cycles of products require mass customization. As mass production shifts to mass customization, a new generation of robots are required to facilitate more agile and adaptive manufacturing lines. This would not be fully adapted without cooperation of human and robot in close proximity.

Human cognitive ability surpasses robots’ intelligence in certain cases. Products assembly and packaging are two examples in which robots were not yet able to replace human in manufacturing lines. Dense sensory requirements, programming complexity, and cost have hindered these tasks to be fully automated by robots. Human-robot collaboration can be a viable response to such complex and expensive tasks. While human’s ability in interpreting sensory data and reasoning cannot be competed by existing artificial intelligence, robots can perform repeated and tedious tasks requiring precision and speed. The collaboration of human and robot can not only rationalize automation of complex and expensive tasks, but accommodates flexible automation at a quick pace. In situation where occasional alternation of manufacturing process is required, an operator - with unbeatable perception and decision-making - can easily guide the robot for a new task without the need for complete reprogramming of the robot. By the mean of human-robot collaboration, it is also expected to use robots in the applications that they have not ever been introduced to.

Human-robot collaboration can be carried out either remotely in a tele-manipulation fashion or in close proximity, sharing workspace with robots. In teleoperation, an operator uses vision, motion, and/or force feedbacks to control a distant robot. While effective, teleoperation involves several practical difficulties including low-level perception, force feedback, and communication delay. Despite recent advancements, widely implementation of teleoperation systems is still a challenging task. In contrast, cooperation of human and robot in close proximity takes less cognitive effort and allows more delicate tasks that requires higher level of perception. While it was conventionally intended to avoid the proximate cooperation of human and robot, a proximate collaboration allows an operator to have more realistic sensation and command motion/force in the same coordinate as end-effector acts in. In addition, human working alongside robots is in fact needed in variety of applications. In many applications, coexistence of human with robot is needed for changing tools, gaging, and fine adjustments. Assembly of rear-axle differentials is being done at BMW through cooperation of human and robot. To assemble rear-axle differentials, the robot lifts and pre-positions the differentials, while the operator adjusts the precise positions of parts. Similarly, mounting truck instrument panels on a moving assembly line and placing cylinder heads on top of engines are being done cooperatively by human and robots at Ford Motor Company. The similar concept is being used in Toyota Motor Company for installing spare tires. Experiments showed significant improvements in task completion time and reduction of operator fatigue.

The desire for co-existence of human and robot poses a fundamental problem of ensuring safety. According to the United Auto Workers (UAW) report [1], various injuries has occurred during physical human-robot interaction. Injuries included cuts, abrasions, and/or
1.1. Motivation

Skeletal fractures. Injuries can be caused by contacts with a sharp or abrasive edges, impact load, and/or trapping in the manipulators pinch points. Extensive experiments performed on dummy in [2-4] showed that serious injuries, such as cranial and facial bones fractures may occur by blunt impacts even at moderate velocities (about 2 m/s). It was also shown that robots with large masses can be life-threatening in case of clamping accidents such that both head and chest can be severely injured. Sharp edges of robots and/or tools can also be other sources of hazard in human-robot interaction. Several soft-tissue experimental tests were performed in [4] on swine legs. Experiments indicated that severity of injuries caused by stabbing or cutting was very high even for velocities lower than 1 m/s. These studies showed that existing industrial robots are too far from being safe for human-robot physical interaction. Hence, the next generation of robots require a radical reformation in their designs to enable safe human-robot interaction.

From different perspective, the elderly share of population - including 60-year old people or older - is increasing (see Fig. 1.1), and projected to form 32% of population by 2050, exceeding even the population of potential workers [5]. Consequently, the ratio of the dependent people to the workers is foreseen to increase substantially. Given the fact that the birth rate is decreasing [2] substitutions for current workers will be a challenging issue in near future. This shift in the age distribution forecasts increasing demands for industrial, assisting, and rehabilitation robots. Undoubtedly, safety is the fundamental requirements for assisting and rehabilitation robots. Also, with the increase in robots surrounding human workspace, the need for safe industrial robots will increase greatly, demanding more research in this area.

Figure 1.1: Demographic features of the developed countries [5].

---

\(^2\)Birth rate in 2010 will be decreased from 1% to 0.5% by 2050 in the world, while it will be even worse in the developed countries by decreasing from +0.3% to -0.3% per year.
Currently, safety for industrial robots is typically attained by segregation of the robot and human workspace using either safety cages or light barriers [6]. In either case, the robot immediately stops if the operator crosses the restricted area. It is clear that segregation of human and robot is orthogonal to the concept of human-robot cooperation. Although, for some specific applications, human-robot safe interaction can yet be achieved partially through control-based techniques, new schemes are required to provide safety for general tasks.

Recently, the concept of cage-free robots has drawn large attention in the robotics community, and led to introduction of a new paradigm shift in the design of industrial robots. The key component behind cage-free robots is intrinsic safe manipulation, allowing human-robot physical interaction for a wide range of applications. As an added benefit, cage-free robots can help in cutting the manufacturing costs by eliminating safety cages and conveyors as well as saving in floor space. According to [7], only 30-40% save of floor space can save more than $100K per robot in manufacturing costs. Reducing the production cost also accommodates the use of robots in small and medium-size enterprises, opening a potential market for robot industry.

In response to the increasing demand for human-safe robots, this study was aimed at developing a new safe robot by using a new actuation concept. The actuation concept was originally disclosed in [8–11] without practical implementation. For the first time, a new 2-Degree-Of-Freedom (DOF) robot is developed as part of this study to validate the actuation concept. The new robot uses Magneto-Rheological (MR) actuators as core of its actuation mechanism. MR actuators offer high torque-to-mass and torque-to-inertia ratios, making them a proper candidate for human-friendly actuation. The use of MR actuators however entails new challenges in modeling and control of the robot actuation that were sought to be addressed in this thesis.

1.2 Why are most existing industrial robots unsafe?

According to Occupational Safety & Health Administration technical manual [12], injuries during a physical interaction between humans and robots can be caused due to several sources of hazards. These sources include unexpected impacts or collisions, crushing and trapping between robot’s arms and/or peripheral equipments, mechanical part break-down, and environmental hazards associated with equipments that either supply power to robots or robots work with. Examples of environmental hazards are radio-frequency interfaces, metal spatter, and high pressure fluids/air. All these sources of hazards can be considered in a comprehensive safety analysis to evaluate safety in human-robot interaction. However,
1.2. **Why are most existing industrial robots unsafe?**

What makes robots different from other industrial machines is perhaps the possibility of unexpected impacts. As such, this section will focus on unexpected collisions only, and discuss why most existing industrial robots do not meet safety requirements in this regard.

Impact collisions include in-contact accidents caused by unpredictable movements of robots, and mainly occur due to components malfunctions and/or unpredictable program errors. To estimate acting forces in an impact, a mass-spring model shown in Fig. 1.2 can be considered, where \( I_r \) represents the effective inertia of the robot at the point of impact, \( K_e \) is the effective interface stiffness, \( I_h \) is human head effective inertia, and \( v_r \) is the impact velocity. According to this model, the maximum acting force in impact can be obtained as follows,

\[
F_{\text{max}} = v_r \sqrt{\frac{I_r I_h K_e}{I_r + I_h}}. \tag{1.1}
\]

An average human head mass is 5.1 kg. The effective interface stiffness is a series combination of the robot link stiffness, \( K_l \), and the stiffness of human skull, \( K_h \). The robot link

![Figure 1.2: Mass-spring collision model.](image)

![Figure 1.3: The maximum force in an impact.](image)
Table 1.1: Inertial parameters of several industrial robots.

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<td>Inertial Parameters† [kg]</td>
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<td>67</td>
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</tr>
</tbody>
</table>

† The values represent reflected mass at the end-effector in the direction of impact.

stiffness has been considered commonly as $K_I = 1 \times 10^5$ N/m in the literature and the human skull stiffness varies between $1 \times 10^5$ N/m and $1 \times 10^6$ N/m depending on the area of contact\(^3\). Considering the maxilla bone stiffness, maximum impact force is plotted in Fig. 1.3 as a function of the robot effective inertia and the impact velocity. Reflected link inertia of several industrial robots with low to high payloads are given in Table 1.1. As can be seen from the Fig. 1.3 the maximum impact force for velocity of 1 m/s is higher than 485 N, and it linearly increases with respect to impact velocity. Considering 6.45 cm\(^2\) (1 in\(^2\)) area of impact, the resulting force at 1.5 m/s is 6 times higher than pain threshold in temporal region\(^4\) and it is enough to cause fracture in Maxilla bone. Increasing impact velocity to 4 m/s, the impact force will exceed 1.9 kN, large enough to break all facial bones\(^5\). Comparing to punches forces in martial arts, the estimated impact force at moderate impact velocity of 3 m/s is larger than punches in karate. Average impact force in so-called reverse punch measured as 1.45 kN, and short-range punches averaged 790 N\([15]\). Extending simulation for frontal bone showed even more sever potential injuries. At impact velocity of 2 m/s, it can be shown that the impact force exceed 4 kN, large enough to break a concrete slap with 3 cm thickness. Not only it can cause fractures in skull, but is enough to accelerate human’s head at the rate of 58 g linearly and 6343 rad/s\(^2\) rotationally, as measured with dummies in sport research (e.g. [16]). Rotational acceleration in excess of 1800 rad/s\(^2\) can lead to a 50% chance of concussion [17].

Recalling [1.1] three factors are effective in an impact force; a) the effective inertia of robot arm, b) the effective stiffness at the point of impact, and c) the impact velocity. Therefore, to achieve intrinsic safety for a robot, one requires to design a robot with low stiffness and inertia, or to achieve safety by limiting the robot velocity. Industrial robots are however intended to be as fast, precise, and as strong as possible to address the payload and performance requirements. The precision in robots depends on robots’s stiffness [19, ch.8],

\(^3\)For instance, the stiffness of the frontal bone $\approx 10^6$ and the stiffness of the maxilla $\approx 10^5$ N/m.

\(^4\)The pressure-pain threshold in the temporal region is 171 kPa [13].

\(^5\)Fracture force for facial bones varies between 0.66 kN to 1.78 kN [14].
requiring high stiffness in both the actuation system and the robot’s structure. The common approach to gain precision and maximum performance is to employ stiff actuation at joints and to use heavy and stiff materials in the robots’ structure. As a consequence, industrial robots possess high inertia and bulky structure. Hence, the mechanical structures of existing robots are the limit to achieve intrinsic safety, that must be addressed by revisiting the actuation and mechanical design of industrial robots.

1.3 Related Works: Human-Safe Robots

Safety for industrial manipulators can be achieved using one or a combination of three different approaches; a) Safeguarding and collision avoidance, b) Light weight design, and c) Collision force suppression. In what follows, each approach and its employments in existing robots will be reviewed from different perspectives.

1.3.1 Safeguarding and Collision Avoidance Using Sensors

Early attempts to ensure robot safety involved the use of safeguarding systems in order to inhibit a robot in presence of humans in its workspace. These methods were mainly oriented to reduce (prevent) unexpected collisions between robots and human, and are widely being used in industrial applications \[20, 21\]. A comprehensive review on the safeguarded robots and related topics can be found in \[22\]. However, these methods are based on isolating human from robot workspace, and are not suitable for applications which include human-robot interaction. Apart from safeguarding systems, a variety of technologies have been utilized in the past two decades to prevent and detect collisions in robot applications.

The prevention techniques are mainly based on non-contact sensors. These techniques were initially utilized in object recognition applications to detect objects in close proximity. Examples of non-contact sensors are infrared and ultrasonic sensors \[23, 24\], capacitance-based systems \[25, 26\], and vision \[27, 28\]. In general, ultrasonic and vision based systems suffer from fundamental problems such as disturbance and dead angles. Compared to these systems, capacitance-based systems possess higher reliability \[29\], however they can only detect objects in close proximity that restricts their usability in high velocity applications.

The most common approach in collision detection is to use force/torque sensors (e.g. see \[30, 31\]). The effectiveness of sensor-based methods are restricted to the placement of sensors on robots, so that these methods can only detect collision at the link/joint that sensors are mounted on. Alternatively, tactile sensors were suggested to achieve sensitivity

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\(^6\)Such as cages, laser fencing, and visual acoustic signals.
on all surfaces of a robot [32]. Difficulties with mounting tactile sensors on the moving parts and deformation are of problems with such techniques. The main problem with torque/force sensor-based techniques is the real-time realization of such techniques. It was experimentally observed that variations of the joint torques in response to an impact were only observable when the impact was completely over [4]. This is because the deflection caused by an impact has to propagate from the impact point to the joints to be detectable by torque sensors at the joints, and the propagation time can be longer than the duration of an impact depending on the length and stiffness of the robot links. A similar problem exists with techniques using force sensors. Most available industrial force sensors have about 1 ms latency, and require anti-aliasing filtering. As a result, force sensors cannot detect high frequency impacts, and detection techniques based on force sensors will carry on at least 1 ms delay, not including the processing time. Using acceleration signals obtained by high gain observers or Kalman filters can be alternatives to this problem. However, even if impacts can be detected shortly after occurrence, the motors at joints cannot react fast enough to compensate the impact force. Hence, no collision detection techniques based on existing technologies can be effective in reality. Experimental tests performed in [3, 4] also confirmed that the severity of injury would not differ with or without collision detection mechanisms.

Although systems for collision detection and avoidance are essential for safe human-robot interactions, most existing methods require imposing velocity restrictions on the operation of robots [33]. Moreover, none of the existing methods can guarantee collision detection or avoidance with sufficient reliability. Even the most reliable systems are subject to unpredictable sensor failure and/or software errors. Furthermore, distinguishing between impacts and interactions is still an unsolved problem, that further adds complexity of applying such techniques in human-robot interaction.

1.3.2 Lightweight Robots

As mentioned earlier, effective mass/inertia of robots are a role-playing factor in safety. Robots with lower masses/inertias are potentially less harmful than the ones with higher mass/inertia, to such a degree most commercial safe robots are aimed to be lighter. Towards this objective, the DLR LWR–III [34] attained a fully integrated light–weight design by utilizing light–weight carbon composite, featuring 1.1 m length, total weight of 13.5 kg, and up to a 15 kg (7 kg nominal) payload. The approximate 1:1 weight-to-payload ratio was obtained by employing harmonic drives, and small on-joint motors. As a comparison

\footnote{For instance, ATI sensors have 800 to 2585 $\mu$s latency, and require 235 Hz anti-aliasing filter.}
1.3. **Related Works: Human-Safe Robots**

Table 1.2: Typical Motor Inertias [38].

<table>
<thead>
<tr>
<th>Motor type</th>
<th>10 hp</th>
<th>1 hp</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC Induction</td>
<td>21.015</td>
<td>5.875</td>
</tr>
<tr>
<td>Induction Servo</td>
<td>21.015</td>
<td>5.875</td>
</tr>
<tr>
<td>Inverter</td>
<td>21.015</td>
<td>5.875</td>
</tr>
<tr>
<td>Vector</td>
<td>21.015</td>
<td>5.875</td>
</tr>
<tr>
<td>DC Servo</td>
<td>21.015</td>
<td>5.875</td>
</tr>
<tr>
<td>Brushless</td>
<td>21.015</td>
<td>5.875</td>
</tr>
<tr>
<td>Ferrite</td>
<td>2.712</td>
<td>0.881</td>
</tr>
<tr>
<td>Rare Earth</td>
<td>2.712</td>
<td>0.881</td>
</tr>
</tbody>
</table>

Table 1.3: Gear ratios and actuator inertia in several industrial robots.

<table>
<thead>
<tr>
<th>Manipulator</th>
<th>Gear ratio</th>
<th>Rotor inertia [kg.m$^2$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Harmonic drives</td>
<td>&gt; 30</td>
<td>N/A</td>
</tr>
<tr>
<td>PUMA 560</td>
<td>53~107</td>
<td>3E-4~4E-4</td>
</tr>
<tr>
<td>Cincinnati Milacron T3-726</td>
<td>96</td>
<td>1.4E-3</td>
</tr>
</tbody>
</table>

To conventional robots, the weight-to-payload ratio in PUMA-560 is 22:1. Kuka’s LBR iiwa is the counterpart of DLR LWR-III, weighing 23 kg with the same design and payload of the LWR-III. The LBR iiwa is currently being used at Mercedes-Benz to assemble rear-axle gear boxes. The Universal Robots 5 and 10 (UR-5 and UR-10) are other examples of light weight robots with respectively 5 and 10 kg payloads. The weight-to-payload ratios in UR-5 and UR-10 are 3.6:1 and 3.7:1, respectively. These robots should not, however, be confused with those robots which achieved safety by reducing payload. An example of such robots is ABB dual arm [35], whose payload is as small as 100 grams.

In the mentioned robots, reducing the effective inertia was mainly achieved by reducing weight of motors and links. However, the effective inertia in robots is in fact summation of the link’s inertia and the reflected inertia of the motor, as expressed below,

$$ J_{\text{eff}} = J_l + G_r^2 J_m, $$

(1.2)

where $G_r$ represents gear ratio and $J_{\text{eff}}$, $J_l$, and $J_m$ are the effective inertia of the robot, inertia of the link, and inertia of the motor, respectively. In most cases, the reflected inertia of the motor becomes the dominant term in (1.2), resulting in a very high effective inertia. Tables [1.2 and 1.3] provide typical values of motors inertia and gear ratios used in robots. Considering Cincinnati Milacron T3-726 robot as an example, the reflected rotor inertia of its first joint is 11.98 kg.m$^2$, while the link inertia itself has been reported in the range of 7.5 to 8.5 kg.m$^2$ [36, 37]. Hence, true effective inertia reduction relies on reducing both robots weight and reflected motor inertia.

Reducing the reflected inertia of motors can be attained by decoupling the motor inertia from the link. The WAM robot was the first robot utilizing this concept by replac-
ing gears with a set of cable and pulleys. Using this configuration motors were also located at the base, further reducing the overall weight of the robot. Using similar approach, the mass/inertia-to-payload was significantly reduced in BioRob-X4 \cite{39} robot arm. The BioRob-X4 can handle up to 2 kg payload, with 3.75 kg total weight. However, cable drives generally result in a limited control bandwidth due to the fundamental resonance frequency of the elastic transmission, which can be as low as 10 Hz or even less in some cases \cite{40}. This issue was considered in a 2-DOF arm \cite{41} and a two-arm Human-Friendly Robot (HFR) \cite{42}, whereby mini on-joint motors were collocated at the joints to improve the high frequency characteristics. These attempts resulted in the development of a bio-inspired robot, so called Stanford Safety Robot \((s2p)\) \cite{43}. The effective mass-to-payload ratio in this robot was close to that of an average US male civilian arm, whereas this ratio can be 36 times larger in a conventional robot such as PUMA560. Reducing effective mass however may achieve by compromising performance. Therefore, a care must be paid in the actuation design in addition to safety considerations so as to maintain high performance.

1.3.3 Collision Force Suppression

Collision force can be reduced by introducing compliance in structure of robots. Compliance can be incorporated either by covering the links by compliant materials or by adding compliant elements in the joints.

Several studies showed that using deformable substances can significantly reduce the impact forces \cite{44–46}. Two problems present with using compliant covers; a) covering joints without limiting the robot movability and b) the required amount of covering. The amount of covering materials is typically substantial\cite{8}. Due to these critical problems, the focus in robot community has recently shifted towards using compliant elements at joints. To this effect, compliance can be generally achieved in two different ways; a) active compliance and b) passive compliance. These two approaches have been reviewed in the following section.

Active Compliance

Active compliance methods involve control-oriented methods to obtain a task-specified compliance. Active compliance can be achieved using either force feedback control or impedance control. Both approaches were predominantly oriented towards controlling forces in contact tasks. The main application of such approaches included automan.

\[\text{As calculated for PUMA 560, the thickness of compliant material required to reduce the chance of severe injury is between 5 to 10 inches, depending on the desired level of safety.} \]
milling and deburring. While they were not initially designed to control collision forces, they can potentially be considered as the means to provide safety to some extent.

Force control methods (e.g. [19, 47]) use force/torque sensors to regulate the applied torques at joints, and hence keep the potential force at the end-effector within a safe region. However, this methodology is only applicable in a situation where the robot is in contact with a surface, that means they can only be employed to guarantee the post-collision safety. Alternatively, compliant actuation can be achieved using impedance control schemes. Impedance control has been realized in three different manners in the literature; stiffness control [48], damping control [49], and general impedance control [50–52]. Despite potential benefit of impedance control in safe actuation, the reliability of the method is restricted by the reliability of sensors and software. Ideally, robots are required to be safe regardless of failure and errors in robot’s component and software, necessitating methods to provide intrinsic safety for robots.

**Passive Compliance**

The passive compliance methods refer to the structural compliance such as springs and dampers employed in robots. Advantages of compliant devices was first described by Whitney in 1982 in peg-in-hole insertion tasks [53]. Laurin-Kovitz et. al. [54] showed that the impedance of a robot can physically be altered using adjustable mechanical elements in the robot actuator. Sugano et. al. [55] also described a single-joint finger prototype consisting of mechanical adjustable spring. A combination of a pneumatic bladder and a DC motor was utilized in [56] to adjust manipulator stiffness. Further, the two 1-DOF and 4-DOF prototypes were developed by Morita and Sugano in [57] and [58] using a combination of adjustable springs and dampers in the mid 90’s. A general formulation of Series Elastic Actuators (SEAs) were enclosed in 1995 by Pratt and Williamson [59]. It was also shown that SEA concept offers shock tolerance, and can result in lower reflected inertia, more accurate and stable force control, and less damage to the environment in comparison to traditional stiff actuators.

Particularly in robot safety, the first application of passive compliance should be credited to the work performed by Lim et. al. [60, 61], in which a visco-elastic trunk was utilized in the design of a human-friendly robot. The visco-elastic trunk was composed of a set of mechanical spring and dampers. This study was extended to a robot with a movable base in [33], where friction was introduced between the base and ground. In case of unexpected collision, the trunk and movable base could be controlled in order to reduce the produced collision forces.
In early systems the actuators impedance were only adjustable before the task. An example of such actuators is SEA. At high frequencies, the actuator can be adjusted to act as a spring so as to reduce impacts caused by collisions or any unexpected interactions. However, this comes at the expense of limited high-frequency control performance due to use of an elastic element. Presence of elastic coupling dramatically limits the control bandwidth of the system. Control performance can be improved by using stiffer coupling, but it adversely affects the output impedance and safety characteristics. In reponse to this trade-off, VSA was proposed with the ability to alter its compliance during task execution. The fundamental limitation of SEAs on control bandwidth still remains a limiting factor in torque performance of VSAs. These attempts led to the development of VIA, which took advantage variable elastic and variable damping elements. This approach was an extension of VSA concept. By being able to vary both elastic and damping element, VIA concept made it possible to enhance the performance, while ensuring safety. The need for additional actuators to vary coupling parameters of VIA is a shortcoming in this technique.

More recently, DM² was introduced to address the shortcoming of SEA and its counterparts. DM² is, in nature, a combination of series elastic and cable drive ideas. The actuation mechanism in DM² consists of two actuators in parallel at each joint; a high speed/low torque actuator and a low speed/high torque actuator. In high frequency, a high speed actuator provides high frequency components of the desired torque with limited torque. This is where the other actuator provides high amplitude components of the desired torques, but at a lower speed. The second actuator introduces a large inertia, hence high impedance. A series elastic actuator is therefore used to achieve low impedance [41]. This concept has been employed in the actuation mechanisms of a 2–DOF arm [41] and a 3-DOF "Human-Friendly Robot" (HFR) [42]. However, there are many practical issues in terms of the design and manufacturing of robots based on DM² concept [62]. DM² is, in fact, a modification over Parallel Coupled Micro-Micro Actuator (PaCMMA), where cable drive transmission is used in place of gears in order to mitigate the effect of backlash. In general, using cable drive transmission may add to the complexity of the robot design. S2ρ [62] addressed this issue by replacing heavy electrical motors (actuators) with pneumatic artificial muscles. A small on-joint motor is still required to compensate the low dynamics of the pneumatic muscles for higher frequencies. Potential hazards of using high pressured air is, however, a weakness of this design. This robot was also modified by incorporating antagonist actuation that led to further performance improvement. The modified S2ρ [43] was able to achieve a force control bandwidth of 26 Hz at 5 N variation of force. The position control bandwidth was also increased to 6 Hz for 10° peak-to-peak motion. Despite these successful accomplishments, the use of artificial muscles may limit the industrial ap-
plication of such robots. Artificial muscles dictate a bound on the maximum achievable payload as well as a limited range of motion due to the limited size and contraction. Muscle creep, high threshold pressure, and substantial hysteresis are also other drawbacks of artificial muscles that can limit the performance of such actuators [63, 64].

In regard with safety characteristics, experiments performed in [33] showed that passive elastic elements are only effective for reducing the post-collision forces, and does not significantly affect the acting forces in impacts. Similar results were simulated in [3], indicating that adding elastic element may not alter the impact characteristics, at least for robots with mass/inertia characteristics as in LWR-III. Passive elastic joint may even adversely effect on collision forces, by storing and realising energy. As such, velocity in link can exceed motor velocity for some trajectories [65, 66]. This should be mentioned that these results were carried out in situations where elastic elements were pre-adjusted, hence the results may not necessarily be valid for variable compliant actuators (e.g. VSA and VIA) and/or other types of compliant actuators.

1.3.4 Variable Damper Actuators

Replacing an elastic element with a damper can be considered as an alternative to SEAs [67]. Series Damping Actuation (SDA) has gained in popularity recently, and has extended to Variable Damping Actuation (VDA) [68–70]. Variable dampers can fundamentally categorized into four categories depending on their operation principles; friction dampers, Eddy current dampers, fluid dynamics dampers, and Electro-Rheological and Magneto-Rheological dampers.

Friction Dampers

Friction Damper (FD), as is evident by its name, is based on friction principles, applying normal force on its output shaft. A simple Coulomb friction model can be used to express its behavior

\[ F_d = -f F_n \text{sign}(\dot{q}_r), \]  

where \( F_n \) is the normal force, \( f \) represents the friction coefficient and \( \dot{q}_r \) the relative speed between the actuator and the output shaft. The Coulomb friction model, however, does not represent hysteretic behavior of the damper. To this end, a more realistic model based on the Bouc-Wen model is proposed in [71]. The Variable Physical Damper Actuator (VPDA)

\footnote{For an example, the maximum contraction in Mckibben muscles is only 20-30% of their initial lengths.}
[69] is an example of an actuator in robotic based on FD’s principles. Hysteretic behaviors, low bandwidth, and high power consumption can be mentioned as drawbacks of FDs [71].

**Eddy Current Dampers**

Eddy Current Dampers (ECDs) employ conductive materials moving through a magnetic field. The damping force in this type of actuator can be explained by the following equation,

\[ f_d = -D(d, B, \sigma)\dot{q}_r, \]

where the damping coefficient \( D \) depends on the geometrical dimension of the actuator \( d \), magnetic field \( B \), and the conductivity of materials \( \sigma \). Several variable dampers were developed based on the concept of ECD (e.g. [72], [73]). As an advantage of ECD, the actuator is fluid-free and contact-free, thereby typical issues with oil leakage and frictional wear are not present in this actuator. The main drawback of ECD is, however, its low output torque, which makes this actuator undesirable in some robot applications.

**Fluid Dynamics Dampers**

Fluid Dynamics Dampers use viscous fluids therein designs to provide damping forces. These devices can be classified into two types of linear and quadratic dampers, depending on their fluid characteristics. As it can be deduced from their names, damping force is linearly proportional to the relative speed between input and output shafts in the linear fluid dynamics damper, while the damping force is proportional to the square of the relative velocity in the quadratic one. Damping coefficient in this type of actuator is a function of geometrical dimensions of the actuator, the surface of contact with the fluid, and viscosity of the fluid. Quadratic relationship between the relative velocity and the output force leads to long lasting oscillations, which renders the quadratic actuator unsuitable for robotic applications [70]. In response to this drawback, a linear fluid dynamics damper was designed and developed recently in [70]. The actuator was however realized using non-Newtonian fluid in order to recuperate the low viscosity range of Newtonian fluids. The use of non-Newtonian fluid may result in non-Newtonian behavior such as shear thinning and time varying behaviors. As an added drawback, manufacturing limits as well as non-Newtonian behavior of the fluid impose geometrical constraints on the design of the actuator, which has limited the maximum achievable output torque.
Magneto-Rheological and Electro-Rheological Dampers

Magneto-Rheological (MR) and its counterpart Electro-Rheological (ER) fluids are kinds of smart materials with varying viscosity respective to applied magnetic and electric fields, respectively. MR and ER based devices have been widely used in many applications, such as MR throttle valve [74], ER/MR long-stroke vibration damper for control of ground vehicles [75], [76], and rotary MR fluid devices for controlling the force[77]. The MR and ER based actuators have several advantages over conventional actuation methods. Specifically in robotic application, Furusho et al. [78] improved the robot arm stability by developing an ER damper. Also, Takesue et al. [79] employed an ER damper in a direct–drive motor system to improve the gain margin of the control system. Similarly, MR devices can be used to obtain high bandwidth actuation. Benefits of a controllable actuator utilizing MR fluids has been recognized in multiple robot applications. Design and development of several haptic devices based on MR fluids were presented in [80–82]. It has been shown that the actuation performance can be enhanced using MR actuators both in robot manipulators [83] and haptic devices [84]. The advantages of MR actuators in variable impedance actuation were discussed in [85, 86]. The MR operating principle was used in realization of VDA in robotics [67] and vehicles [87]. Moreover, the ability of MR actuators to enhance the safety of human-friendly robots based on the intrinsic passivity of MR actuators was discussed in [11, 88, 89].

The behaviors of MR and ER fluids-based dampers can be represented by the Bingham visco-plastic model described by,

\[
F_d = -\text{sign}(\dot{q}_r)(g(\mu, |\dot{q}_r|)) + Cu, \tag{1.5}
\]

where \(g(\mu, |\dot{q}_r|)\) represents the viscous component of the damping force, \(\mu\) is the viscosity of the fluid, \(u\) is either the electric field or the magnetic field, respectively, for ER and MR, and \(C\) denotes a coefficient that depends on the physical properties of the fluid and the geometry of the actuator.

As is evident from [1.5] MR and ER actuators require magnetic and electric circuits, respectively, to form magnetic and electric field required to control the actuators. In particular, using magnetic circuits for MR actuators entails hysteresis in the operation of MR actuators. Obviously, the presence of hysteresis in the actuation often leads to such problems in control systems as tracking errors, limit cycles, and inefficient power consumption. To avoid these effects, a hysteresis compensation scheme is required to enable accurate control of MR actuators.
1.4 Objectives

In the face of the notable progress mentioned above to achieve safe human robot interaction, the desire to develop new safe industrial robots still remains high. Existing solutions has often obtained by compromising velocity and accuracy requirements, and hence performance. For robots to be useful, new designs are required to offer the desirable characteristics of both safety and performance. Achieving this aim, in the author’s strong opinion, falls in developments of new actuation systems for robots. Developments of new actuators may lead to new challenges in modeling and control. However, it is the author’s contention that most undesirable behaviors stemming from new actuators can be addressed by means of control theory. In this respect, this dissertation focuses on development of a new actuation approach and the manner in which the undesirable behaviors of the actuator can be recuperated so as to reconcile the demands for safety and performance.

1.4.1 Main Contributions

The major contributions of this dissertation are as follows:

Development of a New Safe Robot

A new intrinsically safe 2-DOF manipulator is developed. The manipulator is based on the Pluralized Active-Semi Active (PA-DASA) actuation disclosed in precious works in our research group. While the actuation borrows the idea from other works, the developed manipulator is the first implementation of the PA-DASA concept. PA-DASA configuration allows using a unidirectional motor to provide a bi-directional actuation for multiple joints. That is, the actuation of multiple joints can be actively driven by a single motor located at the base. In addition, for the first time, the MR actuator is implemented in an under-actuated manner using parallel combination of an MR actuator and a constant-force spring to provide bi-directional actuation. The performance and suitability of the robot and its new actuation mechanism are validated. Moreover, the MR clutch used in this design is the second generation of MR clutches introduced in previous works. The second generation of MR clutches offer reduced sealing friction as well as reduced weight by using lighter materials in their design.

Analysing Limit Cycles in Antagonistically Coupled MR Actuators

The key design element of PA-DASA concept relies on using antagonistically coupled MR actuators. Despite benefits of this concept in improving performance and removing back-
lash, the antagonistically coupled actuation may result in limit cycles in position control loop. To this effect, the dynamic model of the antagonistic actuator is formulated and an in-depth analysis of the occurrence of limit cycles in the operation of antagonistically coupled MR actuators is studied. To predict possibility of limit cycles, the Describing Function (DF) method is employed, and a connection between frequency and amplitude of oscillation and the system parameters is established. Simulation and experimental results validate the analysis.

**Modeling Hysteresis in MR Actuators**

A new open-loop model for hysteretic behaviors of MR actuators is proposed. The model is based on a novel nonlinear adaptive model using the internal magnetic field and the input current of MR actuators. The adaptive model uses polynomial approximation to deal with unknown dynamics of the system. The uniformly ultimately boundedness of the estimation error is proven using Lyapunov’s direct method. Moreover, a widely accepted hysteresis modeling approach, known as Preisach model is constructed to compare and evaluate the results of adaptive model.

**Hysteresis Compensation for MR Actuators**

Application of adaptive control in compensating hysteresis in MR actuators is explored. To this end, a new adaptive controller is proposed to estimate and eliminate the hysteretic dynamics of MR actuators. The controller uses both geometrical parameters of the actuators and physics of the system. As the main advantage of the proposed scheme, it eliminates the need for any additional force/torque sensors, offering significant advantages in terms of cost reduction and alleviating the fundamental problems with using rigid force/torque sensors in contact with other rigid environments. The stability of the closed-loop system is evaluated using Lyapunov method and the conditions on which the error dynamic remains stable are derived. The proposed scheme is experimentally validated and compared to conventional PID control.

**1.4.2 Thesis Outlines**

The rest of this thesis is organized as follows, while the overlap between chapters are inevitable due to the integrated nature of the thesis.

Chapter 2 describes the new actuation concept along with the design and development of the 2–DOF safe manipulator. Safety analysis of the developed 2-DOF manipulator is also
presented in this chapter. Moreover, several position tracking control results are provided in this chapter to demonstrate the performance of the actuation concept.

Chapter 3 introduces a nonlinear model of antagonistically coupled MR actuators, and presents limit cycle analysis. Relation between amplitude and frequency of the resultant limit cycle with system parameters is also obtained in this chapter. A set of simulation and experimental results is provided in this chapter to validate the analysis.

Chapter 4 discusses the mechanical and magnetic models of MR actuators, and develops the proposed adaptive model for MR actuators. Chapter 4 also compares the modeling results with those obtained by the Preisach model.

Chapter 5 discusses the characteristics of MR fluids along with their controllable yield stress, and presents the proposed adaptive control scheme for compensating hysteresis within MR actuators. Dynamic models of the magnetic circuit and the torque-current relationship in MR actuators are also described in this chapter. In addition extensive experimental results are presented in this chapter to validate the proposed control method and compare the results with those obtained by PID controller.

Chapter 6 concludes this thesis and indicates new directions for future works.
Bibliography


Chapter 2

Design and Development of a New Single-Motor, 2-DOF Safe Robot

The material presented in this chapter is published in “IEEE/ASME Transactions on Mechatronics,” doi:10.1109/TMECH.2013.2281598. A part of this work has also been published in the proceeding of “IEEE International Conference of Robotics and Automation (ICRA),” pp. 337-342, Karlsruhe, 2013.

This chapter presents the design and development of a novel two degrees-of-freedom safe robot manipulator. Magneto-rheological clutches are incorporated in the design to enable antagonistic actuation at the joints. A single unidirectional motor supports bidirectional actuation of all joints. Unlike most current safe robots, high quality actuation is preserved, while the manipulator weight and effective inertia are reduced. This is achieved by relocating the driving motor to the base of the robot. Moreover, magneto-rheological clutches have been shown to pose superior torque to mass, and torque to inertia characteristics over conventional servo motors, further contributing to the reduction of manipulator mass and inertia. The manipulator exhibits both high performance and intrinsic safety as a result of mechanically passive dynamics. A set of experiments is performed to validate the manipulator performance.
2.1 Introduction

In recent years, research on physical human-robot interactions has received considerable attentions. Of particular interest is the subject of robot safety within the context of interactive environments. The study of this subject has led to the development of new robot control and actuation mechanisms to achieve intrinsic safety. Naturally, intrinsic safety is only achievable in kinematic structures that exhibit low output impedance. Commonly, solutions presenting reduced impedance at the expense of reduced performance, or significant increase in the mechanical complexity. Achieving high performance while guaranteeing safety seems to be a challenging goal that necessitates new actuation technologies in future generations of human-safe robots.

Methods for preventing collisions were exordial attempts to achieve safety in anthropic environments. These methods typically relied on vision and/or proximity sensoring [1–3]. Safe-oriented control techniques have also tackled this issue [4–6]. Control-based strategies switch robot to a safe mode as soon as a collision is detected. However, these strategies are only effective for frequencies that are below the closed loop bandwidth of the system [7]. Even though, safety can be achieved by considering strict limitations on the power and/or velocity, as in medical/surgical robots, new schemes must be developed in attempts to guarantee the safety of the humans within a shared workspace [8].

In the event of controller or critical sensor failures, safety relies entirely on the mechanical properties of the manipulator. The addition of compliant material to cover the links of a manipulator can significantly reduce the severity of collisions. However, the quantity of material required to render conventional rigid manipulators (e.g. PUMA) safe for human interaction prohibits practical implementation in this realm [9]. The mass redistribution concept proposed by the design for control methodology in [10] can be applied in the development of safe robots. The development of light weight manipulators as a technique to overcome the inherent dangers of rigid manipulators have gained prominence. To this effect, removing the actuators from the links in order to reduce the link mass, hence the associated link inertia has been successfully experienced. Several notable examples of commercially available robots have adopted this technique. The Whole Arm Manipulator (WAM) [11] is one of the first examples of such light-weight manipulators. This approach, however, resolves only half of the problem. The effective inertia of a link is the sum of reflected actuator’s and link’s impedances. The reflected inertia of the actuator is obtained by multiplying the actuator’s output inertia by the square of the gear reduction ratio. Notably, the reflected actuator inertia is commonly much larger than the link inertia [9]. The new DLR/KUKA Lightweight Robot III (LWR–III) [12] was able to achieved this requirement partially with
approximately a 1:1 payload-to-mass ratio having a total mass of roughly 15 kg. Actuation techniques based on decoupling the reflected inertia from the link, at least partially, have been extensively researched in the literature. A wide range of robots have been developed during the past two decades using compliant actuation systems. Among them are robots that employed Series Elastic Actuation (SEA) [13], Parallel-Coupled Micro-Macro Actuation (PaCMMA) [14], Variable Stiffness Actuation (VSA) [15], and Variable Impedance Actuation (VIA) [16]. While, these approaches successfully enhance the safety, their limited control bandwidth restricts their application, especially in tasks that require versatile and high performance motion. A 3-Degrees-Of-Freedom (DOF) platform using Distributed Macro-Micro (DM\textsuperscript{2}) actuation [9] was developed in [17]. Despite its novel actuation mechanism, this robot was unable to provide large torques without overheating. The Stanford Human Safety Robot (S2\textsuperscript{\rho}) [8] successfully addressed this drawback using artificial muscles. In the face of these efforts, demand for developing safe industrial robots still remains high.

The main contribution of this chapter is the development of a new intrinsically safe 2-DOF manipulator. The manipulator is based on the Distributed Active-Semi Active (DASA) actuation introduced in [18][19] and the Pluralized Antagonistic DASA (PA-DASA) actuation disclosed in [20][21]. Here, the PA-DASA has been utilized and implemented for the first time in a 2-DOF proof-of-concept robot manipulator. The performance and suitability of the robot and its new actuation mechanism are validated. DASA actuation mechanism locates a power source at the base, while uses Magneto–Rheological (MR) clutches to distribute torque at the joints. The power source provides a constant rotational motion while MR clutches control the delivery of output torque. This concept can be reconciled to an antagonistic configuration thanks to unique properties of MR clutches. The antagonistic configuration allows using a unidirectional active drive (motor) to provide a bi-directional actuation. Torque transmission direction is independent from motor direction in this configuration. This independent motion of the power source allows driving multiple antagonistic clutches. That is, the actuation of multiple joints can be actively driven by a single motor located at the base. This concept is the key strategy behind the development of the manipulator presented in this chapter. The overall weight of the manipulator is reduced by relocating a single motor at the base, while MR clutches filter unwanted perturbations caused by friction and compliance of the power transmission. To facilitate the pluralised antagonistic DASA actuation, the manipulator has employed two MR clutches at the first joint and a combination of an MR clutch and a spring at the second joint. The MR clutch has a significantly higher torque-to-mass ratio than conventional electric motors, which makes it a good candidate to be used in human-safe robots. In addition, the effective inertia
of the manipulator is reduced by decoupling the reflected inertia of the actuator from link’s own inertia. The MR clutch used in this design is the second generation of MR clutches introduced in [19]. The second generation of MR clutches offer reduced friction introduced by the sealing as well as reduced weight by using lighter materials in their design. To the best of our knowledge, this is the world first multi degrees-of-freedom safe robot manipulator that uses a single motor to provide multi joints actuation.

The rest of this chapter is organized as follows: Section 2.2 provides briefly a literature background on previous human-safe robot designs. Section 2.3 describes the new actuation concept along with the design and development of the 2–DOF safe manipulator. Section 2.4 provides safety analysis of the developed 2-DOF manipulator based on a collision model. Section 2.5 presents position tracking results. Finally, Section 2.6 concludes this chapter.

2.2 The State-of-the-Art Human-Safe Robots

Traditional robots provided safety by avoiding collision. Collision prevention approaches include real-time obstacle avoidance [22], employing sensitive skin [2], image based collision detection [3], and integrating proximity sensors along with energy-absorbing layers [1]. Although, these approaches are necessary in human-safe robot applications, major hazards result from mechanical characteristics of robots.

Conventional robots employ electric servo motors. Servomotor can only provide very low torque/force which is not appropriate in most robotic applications. Therefore, gear reduction is often used in high torque applications. Gear reduction has major drawbacks including significant friction and backlash. In addition, the reflected-motor shaft inertia increases by the square of the gear reduction, which would result in extremely large impedance. An alternative mechanism to improve geared reduction is to use a set of cable and pulleys in place of gears. This concept is successfully implemented in WAM robot. Although, cable drive transmissions offer low backlash and low static friction, the control bandwidth of the cable drive transmission is limited by the fundamental resonant frequency of the system, which can be as low as 10Hz or even less in most cases [9]. The LWR–III [12] also attained a fully integrated light–weight design by utilizing light–weight carbon composite along with incorporating backdrivable harmonic drives. But, the safety still relies on control-based strategies in detecting and impeding a collision.

Safety can, however, be achieved inherently by employing compliant actuators in the design of robots. SEA was perhaps one of the first attempts of achieving compliant actuators. SEAs have low impedance and friction. There is, however, a trade-off between high performance and safety in using SEA concept. At high frequencies, the actuator acts as
a spring, that reduces impacts caused by collisions or any unexpected interactions. However, this comes at the expense of poor high-frequency torque performance due to the use of an elastic element. The presence of an elastic coupling dramatically limits the control bandwidth of the system. Torque control performance can be improved using stiffer coupling, but it adversely affects output impedance and safety characteristics. To address the mentioned trade-off with SEAs, VSA was proposed with the ability to alter its compliance during task execution. The fundamental limitation of SEAs on control bandwidth still remains a limiting factor in torque performance of VSAs. Replacing an elastic element with a damper was utilized as an alternative in Series Damper Actuation (SDA) [23]. This concept also suffered from similar trade-offs between safety and performance as in SEA and VSA. These attempts led to the development of VIA, which took advantage of both variable elastic and variable damping elements. This approach was an extension of VSA concept. By being able to vary both elastic and damping element, VIA concept made it possible to enhance the performance, while ensuring safety. The need for additional actuators to vary coupling parameters of VIA is a shortcoming in this technique.

More recently, DM² was introduced to address the shortcoming of SEA and its counterparts. DM² is, in nature, a combination of series elastic and cable drive ideas. The actuation mechanism in DM² consists of two actuators in parallel at each joint; a high speed/low torque actuator and a low speed/high torque actuator. In high frequency, a high speed actuator provides high frequency components of the desired torque with limited torque. This is where the other actuator provides high amplitude components of the desired torques, but at a lower speed. The second actuator introduces a large inertia, hence high impedance. A series elastic actuator is therefore used to achieve low impedance [24]. This concept has been employed in the actuation mechanisms of a 2-DOF arm [24] and a 3-DOF “Human-Friendly Robot” (HFR) [17]. However, there are many practical issues in terms of the design and manufacturing of robots based on DM² concept [8]. DM² is, in fact, a modification over PaCMMA, where cable drive transmission is used in place of gears in order to mitigate the effect of backlash. In general, using cable drive transmission may add to the complexity of the robot design. S2ρ addressed this issue by replacing heavy electrical motors (actuators) with pneumatic artificial muscles. A small on-joint motor is still required to compensate the low dynamics of the pneumatic muscles for higher frequencies. Potential hazards of using high pressured gas is, however, a weakness of this design.
2.3 Design and Development of A Two DOF prototype Manipulator

2.3.1 Actuation Concept

The main contribution of this chapter is to present a proof-of-concept of the pluralised antagonistic DASA actuation concept. DASA actuation shown in Fig. 2.1 locates an active drive at the base of the robot, and a semi-active MR clutch at the joint. The active drive provides power to the joint, where the MR clutch limits the output torque. The behavior of MR clutches can be represented by the Bingham visco-plastic model described by

\[ T = \text{sign}(\omega_r) f_c(B) + g_v(\eta, \omega_r), \quad (2.1) \]

where \( f_c(B) = c \tau_y(B) \) is the Coulomb damping torque depending on the yield stress of the fluid \( \tau_y \) and the geometrical dimensions of the clutch \( c \), \( B \) is the magnetic field, \( \omega_r \) is the angular velocity between input and output shafts of the clutch, \( g_v(\cdot) \) is a viscous term depending on the geometrical parameters of the clutch and the viscosity of the fluid \( \eta \). Given the Bingham model, an MR clutch can be considered as a series damper with a geared motor in the DASA actuation, decoupling the output shaft form the input shaft (see Fig. 2.2). Moreover, MR actuators can be designed in such way that the viscous term in (2.1) becomes insignificant compared to the effect of the magnetic field (i.e. \(|g_v| \ll |f_c|\)). Therefore, the output torque becomes less sensitive to the change in the motor velocity. As such, the output torque can independently be controlled by adjusting the applied magnetic field that results in an independent motion of the link. Consequently, unlike servo motors, the MR clutch facilitates a uniform torque transmission, filtering unwanted perturbations caused by bringing mechanical power from the base to the joint. This allows to reduce the robot weight by relocating the active drive at the base, while MR clutch retains the high performance of a “stiff” direct drive.
Chapter 2. Design and Development of a New Single-Motor, 2-DOF Safe Robot

Figure 2.2: Bingham model for MR clutches; a Coulomb friction element $f_c$ in parallel with a viscous damper $g_v$, where $q_m$ and $q_l$ are angular rotations of input and output shafts of the clutch, respectively, associated to the motor and link rotations.

Figure 2.3: Antagonistic DASA Configuration. $G$ is the gear ratio. $J_r$ and $J_l$ represent rotor and link’s inertias, respectively.

Given the fact that active drive rotation is independent from the robot’s link motion, the DASA concept is capable of being used in an antagonistic configuration (see Fig. 2.3). Antagonistic-DASA (A-DASA) provides bi-directional actuation by utilizing only a single unidirectional drive (motor). A-DASA has the added benefit of rectifying a possible dead-zone in a single-clutch DASA configuration due to motor direction change. Two MR clutches receive torques in opposite direction from a motor that only provides a unidirectional rotation. This can be simply achieved using differential gearing. In this way, the net torque delivered to the joint can be changed without altering motor direction. Unidirectional rotation of the motor also eliminates the occurrence of backlash resulting from the use of gears and belts in the transmission. This is due to the fact that neither gears nor belts lose their engagements with the driven component. In addition, the motor can deliver its torque to multiple MR clutches, as opposed to two, allowing to extend the antagonist DASA to a pluralised antagonistic configuration. Fig. 2.4 shows the Pluralised Antagonist-DASA (PA-DASA) actuation schematic. In the PA-DASA actuation, multiple joints will be driven by a single motor located at the base. The PA-DASA mechanism is intended to increase robots’ performance, while offering reduced weight and effective inertia at all joints over using conventional servo motors. Torque-to-mass ratio in our MR clutch is 25:1, while this ratio can be as low as 1:3 for a servomotor of the same weight (Maxon EC60). In addi-
2.3. Design and Development of A Two DOF prototype Manipulator

2.3.2 2–DOF Manipulator Design: Proof-Of-Concept

Fig. 2.5 shows a prototype of a 2–DOF manipulator that uses MR actuators. The manipulator incorporates three MR clutches. Two clutches are mounted at the first joint to provide antagonistic actuation for the first link, while a constant force spring and a clutch are employed at the second clutch. A set of a brushless motor, a gear head, and a gearbox is utilized in the base to provide rotational drive motions for all clutches. The first clutch is mounted underneath the first link and is driven directly by the gearbox. The differential gearbox also provides reversal rotational motion to the second clutch on top through a belt transmission system (see Fig. 2.6). The antagonistic mechanism at the first joint enables us to apply torque in opposite directions without altering the direction of the motor. This configuration also eliminates backlash of the geared motor. Lastly, the third clutch is coupled to the second link. This clutch provides torque in one direction while a constant force spring is used to provide reverse torque. This configuration was intended to reduce the weight/inertia of the end effector for improved safety criteria. The drive input for the third clutch is also delivered through a belt from the first clutch. The specifications of the developed 2–DOF manipulator are presented in Table 2.1.

The first generation of MR clutches developed in our research group was introduced in [19]. The clutch showed high performance in terms of transient responses in position and
Table 2.1: 2–DOF Safe Manipulator Characteristics

<table>
<thead>
<tr>
<th></th>
<th>Length</th>
<th>Weight</th>
<th>Range of Motion</th>
<th>Inertia ($I_{zz}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Link 1</td>
<td>0.32m</td>
<td>5.78kg</td>
<td>85°</td>
<td>0.13kg.m$^2$</td>
</tr>
<tr>
<td>Link 2</td>
<td>0.45m</td>
<td>2.77kg</td>
<td>220°</td>
<td>0.03kg.m$^2$</td>
</tr>
</tbody>
</table>

Table 2.2: Specifications of the MR clutch

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer diameter [mm]</td>
<td>69</td>
</tr>
<tr>
<td>Width [mm]</td>
<td>44</td>
</tr>
<tr>
<td>Disk thickness [mm]</td>
<td>1.0</td>
</tr>
<tr>
<td>MR fluid gap thickness [mm]</td>
<td>0.5</td>
</tr>
<tr>
<td>No. of input disks</td>
<td>2</td>
</tr>
<tr>
<td>Maximum torque [Nm]</td>
<td>15</td>
</tr>
<tr>
<td>Total mass [kg]</td>
<td>2.3</td>
</tr>
<tr>
<td>Output inertia [kg-m$^2$]</td>
<td>$3.7 \times 10^{-3}$</td>
</tr>
</tbody>
</table>

Figure 2.5: 2–DOF safe manipulator.
Figure 2.6: The set of gearbox and belt transmission for antagonistic actuation.

Figure 2.7: The second generation of MR clutches.

torque control. The suitability of the clutch in control applications was studied in [25]. In the second generation of the MR clutch, the weight and the sealing friction are reduced. The configuration of the new MR clutch is shown in Fig. 2.7. The Aluminum spacers used between disks in the first design are replaced with plastic spacers in order to reduce the weight. In addition, the sliding sealing in the first design caused a moderate friction, which limited the minimum output torque. This issue is also addressed in the second generation of the MR clutch by using oil sealing in lieu of ring sealing. The specifications of the new clutch are listed in Table 2.2.

2.3.3 Remarks On Motor Velocity Control

The DASA implementation can be controlled to operate in a region in which torque transmission is relatively immune to perturbation in the relative angular velocity within the
clutch, effectively allowing the clutch to act as a mechanical power filter. The rheological behavior of the MR fluid can be divided into two separate pre- and post-yield regions, depending on the shear strain of the fluid. In pre-yield region, the shear stress of the fluid exhibits Newtonian characteristics. In this region, the shear stress proportionally grows with the shear rate. However, the post-yield behavior is more desirable in most applications since the shear stress can only be controlled by the applied magnetic field without being significantly affected by changes of the shear rate. That is the MR-clutch behavior will remain independent from motor and/or link velocity perturbations. To guarantee this condition, two motor velocity constraints were given in [26] for DASA and A-DASA configurations, as follows

\[
\text{DASA: } |\omega_m| = |\omega_j - \omega^*| + \epsilon^*, \\
\text{A-DASA: } |\omega_m| = \max \{|\omega_j - \omega_1^*|, |\omega_2^* - \omega_j|\} + \epsilon^*,
\]

where \(\omega_m\) and \(\omega_j\) are the motor velocity and the angular velocity of the joint, respectively. \(\omega^*, \omega_1^*, \text{ and } \omega_2^*, \) defined in [26], are the thresholds above which operation in the post-yield region is ensured. Technically, the post-yielding thresholds can differ from clutch to clutch. \(\epsilon^*\) is a field–dependant error margin selected to ensure that the motor have enough time to react to rapid changes of \(\omega_j\). The motor velocity conditions can be further extended to PA-DASA, where the following condition should be satisfied to ensure post-yielding behavior of MR-clutches in this arrangement,

\[
|\omega_m| = \max_i \{|\omega_j_i - \omega_1^*|, |\omega_2^* - \omega_j|\} + \epsilon^*, i = 1, ..., n,
\]

where \(\omega_j_i\) is the angular velocity of the \(i\)-th joint, \(n\) is the number of joints, and \(\omega_\alpha^*, \alpha \in \{1, 2\} \) are the post-yielding thresholds of clutches at the \(i\)-th joint. A proper motor velocity control should be considered in accordance to (2.4) in parallel with the positioning or force control to benefit from all outstanding characteristics of PA-DASA actuation mechanism.

### 2.4 Safety Analysis

Several quantitative metrics have been developed in order to evaluate the safety of mechanical systems in motion. Among them are the Gadd Severity Index [27], the Head Injury Criterion (HIC) [28], the 3 ms criterion, the Viscous Injury Response (VC) [29], and Thoracic Trauma Index (TTI) [30]. Among all, HIC is the most popular and used standard index in the car industries as well as robotics to asses head injury in impacts. This index is mainly correlated with a tolerance curve established at Wayne State University, the
so called Wayne State University Tolerance Curve (WSUTC). WSUTC relates the head acceleration and impact duration to the severity of the brain injury. The curve is obtained experimentally from collision tests for animals and cadaver heads. It is shown that tolerable head acceleration is inversely correlated to impact duration, such that higher acceleration can be tolerated for a shorter period of time, and vice versa.

In general, safety in robotics is a function of several parameters including impact velocity, interface stiffness between human and robot, and effective mass/inertia of the robot [24]. Considering these parameters, a two-degrees-of-freedom mass-spring model can be utilized to predict the resulting human acceleration in a collision. Subsequently, the severity of a possible injury can be estimated using the predicted acceleration using WSUTC or other associated metrics. For a given impact velocity, the interface stiffness and effective mass/inertia can be optimised accordingly to ensure safety. The interface stiffness depends on robot’s cover. However, the required amount of cover might adversely effect the effective mass/inertia of the robot in some cases. Therefore, the effective mass/inertia seems to be of paramount importance in the design of a robot intended to mitigate the severity of the impacts. The effective mass/inertia depends on the robot configuration as well as the direction of force (impact). A geometrical representation of the effective mass/inertia is given in [31] that displays the mass/inertia perceived at the end effector along different direction. Fig. 2.9 illustrates the effective mass at the same configurations of $q_1 = 20^\circ$ and $q_2 = -90^\circ$ (see Fig. 2.8) for a PUMA560, LWR-III, human arm, and the 2–DOF robot. The diagram shows that LWR-III, the 2–DOF robot, and human arm have maximum effective masses of 17.142 kg, 5.452 kg, and 1.998 kg respectively, while PUMA560 as a conventional robot example has the effective mass of 39.992 kg. The effective mass of 2–DOF robot is comparable to that of lightweight LWR-III. This is not, however, a fair comparison unless the effective mass is normalised by each arm’s payload. The normalized effective masses are given in Table 2.3. To obtain these results, the human biomechanics and performance capabilities data of an average US male civilian [32], given in Table 2.4, were used.
Table 2.3: Normalized Effective Masses

<table>
<thead>
<tr>
<th></th>
<th>Effective Mass</th>
</tr>
</thead>
<tbody>
<tr>
<td>PUMA560</td>
<td>1.5997</td>
</tr>
<tr>
<td>2-DOF</td>
<td>0.1704</td>
</tr>
<tr>
<td>Human arm</td>
<td>0.0322</td>
</tr>
<tr>
<td>LWR-III</td>
<td>0.2449</td>
</tr>
</tbody>
</table>

As observed the 2–DOF robot has a normalized effective mass that is close to LWR-III.

Table 2.4: Right Hand Characteristics of an Average US Male Civilian

<table>
<thead>
<tr>
<th></th>
<th>Length</th>
<th>Weight</th>
<th>Center of mass</th>
<th>Inertia ($I_{zz}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper Arm</td>
<td>0.366 m</td>
<td>2.5 kg</td>
<td>0.15m</td>
<td>0.0151 kg.m$^2$</td>
</tr>
<tr>
<td>Forearm</td>
<td>0.454 m</td>
<td>1.45 kg</td>
<td>0.11m</td>
<td>0.0014 kg.m$^2$</td>
</tr>
</tbody>
</table>

Further, the Manipulator Safety Index (MSI) is computed and illustrated in Fig. 2.10 to evaluate the severity of an uncontrolled impact. The safety simulations are performed under assumptions that constant impact velocity is 3 m/s, an average human head weight is 5.1 kg, and the interface stiffness between the head and the manipulator is 20000 N/m. Although, 2–DOF manipulator displays slightly more normalized effective mass than LWR–III light weight robot, the MSI of the 2–DOF robot is lower than the LWR–III under same conditions. The MSI is a safety index based on the HIC, in which the manipulator effective mass is incorporated in the impact mass-spring model. The direction of maximum MSI value coincides with the direction of maximum end-effector effective mass. A frontal collision in this direction will yield the greatest likelihood of brain injury. When the MSI or equivalent HIC$_{15}$ is less than 10, the probability of minor brain injury is zero.

2.5 Preliminary Experimental Results

To analyze the performance of the developed 2–DOF manipulator shown in Fig. 2.11, several position tracking experiments were performed. To this end, a brushless motor

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1 It should be acknowledged that LWR-III is however a 7–DOF robot, and the extra weight added by other joints should also be considered in the comparison.
2.5. Preliminary Experimental Results

Figure 2.9: (a) Effective Masses, (b) Zoomed-in by 4x
Figure 2.10: Manipulator Safety Index (MSI)

Figure 2.11: Developed 2-DOF robot manipulator.
2.5. Preliminary Experimental Results

Figure 2.12: Position tracking control results for square command with constant motor velocity.

(BLWRPG235D-36V-4000-R13) was used to provide the rotational inputs to the MR clutches. The motor was driven by a driver in velocity control mode. The manipulator incorporates two encoders (HEDS-9000). Three high-power motor drivers, set in current mode, provide the command currents to the MR clutches (AMC-AZ12A8). In our experiment, a PID controller, implemented on a desktop computer, controlled the manipulator via a dSPACE (DS 1103) controller board. The sampling frequency for gathering experimental data was set to 1 kHz.

Position tracking control experiments were performed to evaluate the performance and feasibility of the proposed actuation mechanism. Fig. 2.12 shows the results of the position tracking control of the 1st and 2nd joints, respectively. The motor was set to rotate at a constant velocity of 1800 RPM. PID controller was used to control the MR clutches using feedback from encoders located at the joints. Both joints were commanded simultaneously and moved simultaneously. These results clearly, demonstrate independent motion of the links. The joints velocities of the manipulator are also shown in Figs. 2.13 along with the motor velocity to further assess the feasibility of the proposed actuation concept. As shown in this figure, the direction of joint rotation can be altered, while the motor continues rotating in one direction. It should be pointed that the motor was command with a constant velocity in all experiments and even though the motor velocity occasionally dropped by few percentage when both joints loaded the motor simultaneously, the manipulator was able to
Figure 2.13: Velocities of joints along with motor velocity with constant motor velocity.

Figure 2.14: Joint 1 position tracking control results for sinusoidal inputs.
2.6 Conclusion

The development of a new intrinsically safe 2–DOF manipulator has been presented. The manipulator utilises a novel actuation approach referred to as Pluralized Antagonistic Distributed Active-Semi Active (PA-DASA) actuation, which is an extension of our earlier work (i.e. A-DASA). The design integrates MR clutches as the semi-active elements of the actuation system. This approach leverages key properties of MR clutches, namely low impedance and backdrivability. MR clutches provide an additional benefit of decoupling the motor rotor inertia from the effective inertia of the link, lending themselves to provide intrinsic safety. The PA-DASA actuation arrangement locates a single driving motor at
the base of the manipulator, reducing the overall mass as well as the inertia of the robot, further improving the inherent safety of the manipulator. Relevant values for mass and effective inertia were given along with MSI to provide benchmark metrics in regard to intrinsic safety. A set of experiments were conducted to show the feasibility and performance of the actuation approach. The results showed that the approach could in fact provide bi-directional actuation to all joints using unidirectional motion generated by a single motor. Furthermore, the results of position control experiments demonstrated the performance of the actuation concept in accurate tracking. Further results on position and force control will be provided as part of our ongoing research. Moreover, collision based experiments will be performed in our future work to further assess the manipulator’s so called intrinsic-safety value.
Bibliography


Chapter 3

Study of Limit Cycle in Antagonistically Coupled Magneto-Rheological Actuators

The material presented in this chapter is submitted in “Control Engineering Practice.” A part of this work has also been submitted to the “IEEE International Conference of Robotics and Automation (ICRA),” Hong Kong, 2014.

In this chapter, the presence of limit cycles in the behavior of antagonistically coupled Magneto-Rheological (MR) actuators is investigated. The actuator considered in this chapter was developed and described in [1] and [2]. This actuator offers high torque-to-mass and torque-to-inertia ratios, for inherent safe actuation. While the antagonistic arrangement is beneficial in improving the actuator performance and eliminating backlash, it may result in limit cycles when the actuator operates in a position control loop. The occurrence of limit cycle depends on the parameters of the actuator as well as the controller. An in-depth analysis is carried out in this chapter to establish a connection between the system parameters and the limit cycle occurrence. Moreover, sufficient conditions for avoiding limit cycle are derived specifically for a Proportional-Derivative (PD) controller. Simulations and experimental results validate the analysis and provide insights into the limit cycle observed in the operation of antagonistic MR actuators.
3.1 Introduction

Limit cycles are self-excited oscillations in nonlinear systems, the characteristics of which are irrespective to initial conditions. Such a periodic solution emerges as a closed trajectory in the phase portrait of a nonlinear system. In particular, the limit cycle oscillation is an undesirable response to positioning control systems that can cause mechanical failure. As experimentally observed, antagonistic actuation based on Magneto-Rheological (MR) fluids can give rise to limit cycles in position control loops. This chapter aims at establishing the dependency of limit cycle occurrence on the parameters of MR actuators.

MR fluids are non-homogenous suspensions of micrometer-sized ferromagnetic particles in a carrier fluid. The apparent viscosity of MR fluids can be adjusted by an external magnetic field. The suspended particles in the fluid form columns (chains) aligned to the direction of the applied field that results in shearing or flow resistance in the fluid. The degree of the resistive force is related to the strength of the magnetic field, resulting in a field dependent yield stress in MR fluids \[3, 4\]. In the absence of a magnetic field, MR fluids act as Newtonian fluids whose viscosity change proportional to the shear rate.

Controllability and fast response of MR fluids to external magnetic field have made them an attractive technology for a broad range of applications from civil engineering to automotive, robotics, and rehabilitation applications (e.g. see \[5-7\]). Particularly, in robotics, MR actuators can be employed in series with active drives (e.g. motors) to control the delivery of the output torque at the joints. Benefits of a controllable actuator utilizing MR fluids have been recognized in various robotic applications including haptic devices \[8-10\] and human-robot interaction \[2, 11-13\].

As the main contribution of this chapter, we perform a careful and in-depth analysis of the occurrence of limit cycles in the operation of an antagonistic MR actuator. To this end, the dynamic model of the antagonistic actuator is formulated. Considering the system in a feedback loop, a set of simulations is carried out to provide insights to the occurrence of the limit cycle in the actuator response. The describing function method \[14\] Ch. 7 is then employed to predict the possibility of limit cycle oscillation in the actuator response. The frequency and amplitude of the oscillations are analytically derived and their dependencies on the system parameters and PD controller are studied. Sufficient conditions under which limit cycle oscillations can be prevented are obtained. The results are experimentally validated using a planar robot utilizing MR actuators as a test bench.

The rest of this chapter is organized as follow: Section 3.2 introduces antagonist actuators based on MR fluids along with discussion on MR fluids characteristics. A nonlinear model of an antagonistic MR actuator is also derived in this section. Section 3.3 presents
the main contribution of this chapter, where the occurrence of limit cycles is analyzed, and the amplitude and frequency of the resultant limit cycle is obtained. In Section 3.4, simulation and experimental results are provided to validate our theoretical analysis. Finally, Section 3.5 concludes the chapter.

3.2 Magneto-Rheological Fluid-Based Actuation

The torque-velocity behavior of an MR actuator is formulated in this section. Fig. 3.1 depicts the Distributed Active-Semi Active (DASA) actuation concept [12]. In this arrangement, the active drive provides power to the joint, via an MR clutch that controls the output torque. A drawback of DASA actuation is the need for the motor velocity to reverse for bi-directional actuation, resulting in backlash in geared actuation. Moreover, the performance of the actuation will be limited by the performance of the active drive. To address the problem, the DASA actuation can be extended to an antagonistic configuration [2].

Antagonistic actuation provides bi-directional actuation by utilizing a single unidirectional motor (see Fig. 3.2). In this configuration, two MR clutches receive rotational motion in opposite direction from a motor that provides a unidirectional rotation. This can be simply achieved using differential gearing. In this way, the net torque delivered to the joint in both direction can be changed without altering the motor direction. While eliminating backlash in the gears and/or transmission belts. This is due to the fact that neither gears nor the transmission belts lose their engagements with the motor.

In general, the behavior of MR actuators depends on the MR fluid characteristics and the geometry of the actuator. The effects of the MR fluids and geometrical parameters of the actuator on the operation of A-DASA actuation are explained in the following subsections.
3.2. Magneto-Rheological Fluid-Based Actuation

3.2.1 Models for MR Fluids

Several visco-plastic models have been proposed to describe the behavior of MR fluid-based devices. Among them are the Bingham plastic model [14, 15, 16], the Herschel-Bulkley model [17, 18], and the biviscous model [19, 20].

According to the Bingham visco-plastic model, MR fluids can be regarded as plastic materials, in which their shear stress $\tau$ changes with respect to the field-dependent yield stress $\tau_y$ and shear rate $\dot{\gamma}$, as described below,

$$
\tau = \tau_y(H)\text{sgn}(\dot{\gamma}) + \eta\dot{\gamma},
$$

(3.1)

where $H$ is the magnetic field, $\eta$ is the effective Newtonian viscosity of the carrier oil of MR fluids, and $\text{sgn}(\cdot)$ is the signum function. The fluid yield stress $\tau_y$ and viscosity $\eta$ are typically provided by the manufacturer of MR fluids (e.g. see [21]).

Closely related, the Herschel-Bulkley phenomenological model was proposed to account for shear thinning and thickening effects. Shear thinning and thickening effects may appear in the behavior of MR fluids depending on the operational region of MR-based devices [17]. The Herschel-Bulkley model is a modification to the Bingham model, and can be represented by,

$$
\tau = [\tau_y(H) + K|\dot{\gamma}|^m^{\frac{1}{m-1}}]\text{sgn}(\dot{\gamma}),
$$

(3.2)

where $m$, and $K$ are the fluid index parameters, which can be adjusted using experimental measurements. For $m = 1$ the Herschel-Bulkley model reduces to the Bingham model, whereas it represents the shear thinning and thickening effects for $m > 1$ and $m < 1$, respectively. The non-Newtonian behaviors of the Bingham and Herschel-Bulkley models are depicted in Fig. 3.3.

Despite the widely use of the Bingham and Herschel-Bulkley models due to their simplicity, the true rheological behavior of MR fluids degrades from the ideal Bingham and Herschel-Bulkley models. Fig. 3.4 shows the shear stress-shear rate behavior of a sample MR fluid, which qualitatively represents the typical behavior of MR fluids [22]. As
observed, the behavior of an MR fluid can be divided into two main phases; pre-yielding and post-yielding regions. The behavior of MR fluids varies from viscoelastic in the pre-yielding region to plastic in the post-yielding, and it is visco-plastic in the transition through yielding. This type of behavior suggests a biviscous characteristic (see Fig. 3.5), where the fluid behavior switches between pre- and post-yielding behaviours, as can be expressed by,

\[
\tau = \begin{cases} 
\tau_y(H) \text{sgn}(\dot{\gamma}) + \eta \dot{\gamma}, & |\dot{\gamma}| \geq \dot{\gamma}_y \\
\eta_r \dot{\gamma}, & |\dot{\gamma}| < \dot{\gamma}_y
\end{cases}
\]  

(3.3)

where \(\eta_r\) is the elastic property of the fluid, and \(\dot{\gamma}_y\) has the following relationship with the yield stress,

\[
\dot{\gamma}_y = \frac{\tau_y(H)}{\eta_r - \eta}.
\]  

(3.4)

It should be mentioned that more complex models have also been proposed for MR fluids considering the fluid mechanics at the particle level [23–26]. These studies were carried out using either the kinetic or thermodynamic theories considering the effects of the carrier fluid flow, external magnetic field, interparticle forces, and Brownian motion. While effective, practical implementation of such models relies on some particle-level parameters, which may not be available for commercial MR fluids\(^1\).

### 3.2.2 A-DASA Model

Fig. 3.6 shows the cross-section of a typical multi-disk MR clutch. The input shaft breaks out into a set of input disks which are aligned in parallel to a set of output disks attached to

---

\(^1\)In most cases, only yield stress and viscosity of MR fluids are available.
Figure 3.4: Shear stress vs. shear rate in a sample MR fluid.

Figure 3.5: Biviscous model of MR fluids.

Figure 3.6: Cross-section of a multi-disk MR clutch.
the output shaft. MR fluid fills the volume between input and output disks. By energizing the electromagnetic coil, the shear stress of MR fluids, thereby the output torque of the clutch can be controlled. Considering this arrangement, the shear rate of MR fluids at radius $\rho$ equals to

$$\dot{\gamma}(\rho) = \dot{\theta}_r \rho l_f^{-1}, \quad (3.5)$$

where $\dot{\theta}_r$ is the angular velocity between the input and output shafts, and $l_f$ is the gap between the input and output disks. The torque produced by a circumferential element at a radius $\rho$ is given by,

$$dT = 2\pi \rho^2 \tau d\rho, \quad (3.6)$$

where $\tau$ is the shear stress as defined in (3.5). Assuming that the clutches in antagonistic configuration are identical, and each clutch has $N$ output disks, the torque transmitted through either of the clutches can be obtained after substituting (3.3) into (3.6) and integrating across both faces of each output disk, i.e.,

$$T_i = \begin{cases} 
2N \int_{R_1}^{R_2} 2\pi \left( \tau_y(H_i)\text{sgn}(\dot{\theta}_r)\rho^2 + \eta \frac{\dot{\theta}_r \rho^3}{l_f} \right) \, d\rho, & |\dot{\theta}_r| \geq \dot{\theta}_y \\
2N \int_{R_1}^{R_2} 2\pi \left( \tau_y(H_i)\text{sgn}(\dot{\theta}_r)\rho^2 + \eta \frac{\dot{\theta}_r \rho^3}{l_f} \right) \, d\rho, & O.W. \\
+2N \int_{R_1}^{R_2} 2\pi \left( \eta \frac{\dot{\theta}_r \rho^3}{l_f} \right) \, d\rho, & O.W. 
\end{cases} \quad (3.7)$$

where $T_i, i \in \{1, 2\}$ is the output torque of the $i$-th clutch, $H_i, i \in \{1, 2\}$ is the applied magnetic field to the corresponding clutch, $\dot{\theta}_r = (\dot{\theta}_m - \dot{\theta}_i)$ and $\dot{\theta}_r = (-\dot{\theta}_m - \dot{\theta}_i), \dot{\theta}_m$ and $\dot{\theta}_i$ are, respectively, the input and output shafts velocities, $R_1$ and $R_2$ are the inner and outer radii of the disks, respectively, $\dot{\theta}_y = l_f \dot{\gamma}_y R_1^{-1}$, and $R = \min\{R_2, l_f \dot{\gamma}_y |\dot{\theta}_r|\}$. All other parameters are as defined previously. Subsequently, we have,

$$T_i = \begin{cases} 
\varphi_1 \tau_y(H_i)\text{sgn}(\dot{\theta}_r) + \varphi_2 \dot{\theta}_r, & |\dot{\theta}_r| \geq \dot{\theta}_y \\
\varphi_3 \tau_y(H_i)\text{sgn}(\dot{\theta}_r) + \varphi_4 \dot{\theta}_r, & O.W. 
\end{cases} \quad (3.8)$$

where the coefficients $\varphi_i, i \in \{1, 2, 3, 4\}$ are defined as,

$$\begin{array}{l}
\varphi_1 = 4N\pi(R_2^3 - R_1^3)/3, \\
\varphi_2 = N\pi \eta(R_2^4 - R_1^4)/l_f, \\
\varphi_3 = 4N\pi(R_2^3 - R_3^3)/3, \\
\varphi_4 = N\pi \left( \eta R_2^4 - \eta R_1^4 + R_1^4(\eta - \eta) \right)/l_f
\end{array} \quad (3.9)$$

Then, the total torque of the actuator $T$ equals to the summation of $T_1$ and $T_2$, i.e.,

$$T(H_1, H_2, \dot{\theta}_i, \dot{\theta}_m) = T_1(H_1, \dot{\theta}_i, \dot{\theta}_m) + T_2(H_2, \dot{\theta}_i, \dot{\theta}_m). \quad (3.10)$$
3.3 Limit Cycle Analysis

The behavior of A-DASA actuation in feedback control loop is studied in this section. This study requires a dynamic model of the controlled system. Fig. 3.8 shows a position control scheme for the A-DASA actuation, in which $G_c(s)$ and $G_f(s)$ are the transfer functions of the controller and the output shaft mechanical subsystem, respectively. The dynamic equations of the system can be expressed as,

$$
\dot{H}_1 = -\lambda_1 H_1 + c_1 S|i(t)|,
$$

$$
\dot{H}_2 = -\lambda_2 H_2 + c_2 (1 - S)|i(t)|,
$$

$$
\dot{\theta}_l = -b \dot{\theta}_l + \frac{1}{J} T(H_1, H_2, \dot{\theta}_l, \dot{\theta}_m),
$$

(3.11)

where $i(t)$ is the input current commanded by the controller, $\lambda_i$ and $c_i, i \in \{1, 2\}$ are the magnetic circuit parameters, $J$ is the inertia of the output shaft, $b$ represents the viscous

Considering (3.8), the output torque of each clutch in an A-DASA actuator is depicted in Fig. 3.7, where the effect of the fluid viscosity is neglected, i.e. $\eta \approx 0$. Since $|\dot{\theta}_r| < \dot{\gamma}$ corresponds to the pre-yielding phase of MR fluids ($|\dot{\gamma}| < \dot{\gamma}$), a roll-off phenomenon occurs in the operation of the actuator for $|\dot{\theta}_r| < \dot{\gamma}$, when the output torque of the actuator decreases in absolute value, depending on the value of $\varphi_4$. In a typical MR actuator, this phenomenon results in zero acceleration at velocities closed to the motor velocity, as such the output shaft velocity $\dot{\theta}_l$ remains in a bounded region, i.e. $|\dot{\theta}_l| \leq \dot{\theta}_m$.

In general, the output torque of the actuator can be controlled by adjusting the applied magnetic fields to the clutches and varying the motor velocity. It is however favourable to set the motor velocity to a constant value greater than the maximum desired velocity and control the torque solely by the applied magnetic fields, i.e. $H_1$ and $H_2$. 

![Figure 3.7: Torque-velocity relationship of the A-DASA actuator.](image)
friction coefficient, and the switching variable $S$ is defined as below,

$$S = \begin{cases} 
1, & i(t) \geq 0 \\
0, & i(t) < 0
\end{cases}$$

(3.12)

Since the output torques of the clutches counteract with each other, it is more efficient to power only one clutch at a time. Note that the yield stress of the MR fluid is independent of the applied magnetic field, hence the applied current.

### 3.3.1 A Practical Example

Let us now consider a practical example before proceeding with the limit cycle analysis. In this example, the A-DASA actuator with a PD controller is considered. The parameters of the system are provided in Table 3.1. Considering a regulating task, the controlled system described in (3.11) was simulated in Matlab, where $i(t)$ was defined as

$$i(t) = k_p(\theta_d - \theta_l) - k_d \dot{\theta}_l,$$

(3.13)
3.3. Limit Cycle Analysis

Figure 3.9: Presence of limit cycles in position control of A-DASA actuator; (a) $k_p = 1$, $k_d = 0$, (b) $k_p = 2$, $k_d = 0.005$, (c) $k_p = 2.5$, $k_d = 0.005$, and (d) $k_p = 2.5$, $k_d = 0.01$. The circles show initial conditions.

in that $\theta_d$ and $\theta_l$ are the desired and the output shaft position, respectively, and $k_p$ and $k_d$ are the controller gains corresponding to the proportional and derivative terms. In all simulations, the desired angle $\theta_d$ was set to zero. Fig. 3.9 shows the phase portrait of the system for a set of proportional and derivative gains. As observed, the response of the system converges to various limit cycles depending on the values of $k_p$ and $k_d$. Similar results can be obtained for a PID controller. It can be shown that the amplitude and frequency of oscillations depend on the controlled system parameters. To this effect, the describing function method can be used to establish the connection between the system parameters and the resulting limit cycles. The describing function method is applicable to quasi-linear systems that contain one or multiple nonlinearities [27].

3.3.2 Assumptions and Preliminaries

In order to use describing function method, the system is required to be presented as an interconnection of linear subsystems and a nonlinearity. Referring to Fig. 3.8, the dynamic model of the A-DASA subsystem is required. We define the post-yielding torque $T_y$ as
follows,
\[
T_y = \begin{cases} 
\varphi_1 \tau_y(H_1), & i(t) \geq 0 \\
-\varphi_1 \tau_y(H_2), & i(t) < 0 
\end{cases} \quad (3.14)
\]

Linearizing (3.14) around \( H_1 = H_2 = 0 \) yields,
\[
T_y \approx \begin{cases} 
\varphi_1 \nu H_1, & i(t) \geq 0 \\
-\varphi_1 \nu H_2, & i(t) < 0 
\end{cases} \quad (3.15)
\]

where \( \nu \) is defined as,
\[
\nu = \frac{\partial \tau_y(H)}{\partial H} \bigg|_{H=0}.
\]

Considering (3.11) and (3.14), the time derivative of \( T_y \) can be obtained as,
\[
\dot{T}_y \approx \begin{cases} 
\varphi_1 \nu \dot{H}_1, & i(t) \geq 0 \\
-\varphi_1 \nu \dot{H}_2, & i(t) < 0 
\end{cases} \\
\approx \begin{cases} 
-\varphi_1 \nu \lambda_1 H_1 + \varphi_1 \nu c_1 i(t), & i(t) \geq 0 \\
+\varphi_2 \nu \lambda_2 H_2 + \varphi_1 \nu c_2 i(t) & i(t) < 0 
\end{cases}
\]
\[
\approx \begin{cases} 
-\lambda_1 T_y + \varphi_1 \nu c_1 i(t), & i(t) \geq 0 \\
-\lambda_2 T_y + \varphi_1 \nu c_2 i(t) & i(t) < 0 
\end{cases} \quad (3.16)
\]

Considering that both clutches and their magnetic circuits are identical, one can assume,
\[
\lambda_1 = \lambda_2 = \lambda, \quad c_1 = c_2 = c,
\]
which leads to,
\[
\dot{T}_y \approx -\lambda T_y + c_T i(t), \quad (3.17)
\]

for \( c_T = \varphi_1 \nu c \).

Given the fact that the effect of the fluid viscosity is neglectable,\(^2\) the output torque of the A-DASA actuator is approximately equal to \( T_y \) when the actuator operates in the post-yielding region. In the event that the actuator enters the pre-yielding region, the output torque will decrease in absolute value reciprocal to the growth of the output shaft velocity due to roll-off (see (3.8)). The position control block diagram of the system can be represented as shown in Fig. 3.10, where \( G_T(s) \) is the transfer function between the post-yielding...

\(^2\)The viscosities of MR fluids are typically in the range of 0.1 to 0.3 Pa-s.
3.3. Limit Cycle Analysis

Figure 3.10: Equivalent position control block diagram of the A-DASA actuation.

torque and the input current, and the nonlinearity \(\psi(\dot{\theta}_l)\) imitates the roll-off phenomenon defined as,

\[
\psi(\dot{\theta}_l) = \begin{cases} 
0, & |\dot{\theta}_l| \leq \dot{\theta}_m - \dot{\theta}_y \\
-\varphi_4 \dot{\theta}_l, & \text{O.W.}
\end{cases}
\]  
(3.18)

The describing function method can now be used to analyze the possibility of limit cycles. In this method, the system is assumed to be a free system; that is, the system is only excited by the initial conditions and the system inputs are zero. The basic idea behind the analysis is to derive conditions under which the system is capable of regenerating a sinusoid, if the input to the nonlinearity is sinusoidal. This condition is met when the closed-loop gain of the system from the nonlinearity input back to the nonlinearity equals to unity. Hence, the system exhibits limit cycle behavior, regenerating the same sinusoidal response.

3.3.3 Analysis

To derive the conditions for limit cycle, we define a function \(\Psi(A)\) as,

\[
\Psi(A) = \frac{2\omega}{\pi A} \int_0^{\pi \omega} \psi(A \sin \omega t) dt.
\]  
(3.19)

where \(\Psi(A)\) is called the describing function of \(\psi(\cdot)\), and can be regarded as an equivalent gain of the nonlinearity when exposed to a sinusoidal input \(A \sin(\omega t)\). Introducing \(3.18\) into \(3.19\), the describing function \(\Psi(\omega, A)\) can be obtained as,

\[
\Psi(A) = \begin{cases} 
0, & A \leq \dot{\theta}_m - \dot{\theta}_y \\
-\varphi_4 \left\{1 - \Psi_s \left(\frac{A}{\dot{\theta}_m - \dot{\theta}_y}\right)\right\}, & A > \dot{\theta}_m - \dot{\theta}_y
\end{cases}
\]  
(3.20)
where

\[ \Psi_s(x) = \frac{2}{\pi} \left\{ \sin^{-1} \frac{1}{x} + \frac{1}{x} \cos(\sin^{-1} \frac{1}{x}) \right\}. \tag{3.21} \]

According to (3.11) and (3.17), \( G_l(s) \) and \( G_T(s) \) can be described as,

\[ G_l(s) = \frac{k_l}{\tau_l s + 1}, \quad G_T(s) = \frac{k_T}{\tau_T s + 1}, \tag{3.22} \]

where

\[ k_l = J^{-1}, \quad \tau_l = Jb^{-1}, \tag{3.23} \]
\[ k_T = cT \lambda^{-1}, \quad \tau_T = \lambda^{-1}. \]

Considering \( G_c(s) \) as a PD controller, the closed-loop transfer function of the system can then be expressed as,

\[ G(j\omega) = \frac{-\left( k_p + jk_d\omega \right) G_T(j\omega)G_l(j\omega)}{j\omega(1 + \Psi(A)G_l(j\omega))}. \tag{3.24} \]

Introducing (3.22) into (3.24), one can obtain,

\[
G(j\omega) = \frac{-\left( k_p + jk_d\omega \right) k_T k_l}{j\omega(j\omega\tau_l + k_l\Psi(A) + 1)(j\omega\tau_T + 1)}
= \frac{-\left( k_p + jk_d\omega \right) k_T k_l}{-\alpha\omega^2 + j\omega(\beta - \omega^2\tau_l\tau_T)}
= \frac{k_T k_l \left( k_p + jk_d\omega \right) \left( \alpha\omega^2 + j\omega(\beta - \omega^2\tau_l\tau_T) \right)}{\alpha^2\omega^4 + \omega^2(\beta - \omega^2\tau_l\tau_T)^2}. \tag{3.25}\]

where \( \alpha = \tau_l + \tau_T\beta \) and \( \beta = 1 + k_l\Psi(A) \). A limit cycle probably exists if (3.25) equals unity. Equating (3.25) to unity results in the following conditions,

\[ \Re \{ G(j\omega) \} = 1, \quad \Im \{ G(j\omega) \} = 0 \tag{3.26} \]

where \( \Re \{ G \} \) and \( \Im \{ G \} \) are real and imaginary parts of the transfer function \( G(j\omega) \), as written below,

\[
\Re \{ G(j\omega) \} = \frac{k_T k_l \left( \alpha k_p - k_d(\beta - \tau_l\tau_T\omega^2) \right)}{\alpha^2\omega^2 + (\beta - \omega^2\tau_l\tau_T)^2},
\Im \{ G(j\omega) \} = \frac{k_T k_l \left( k_p\beta - \omega^2(k_p\tau_l\tau_T - k_d\alpha) \right)}{\alpha^2\omega^3 + \omega(\beta - \omega^2\tau_l\tau_T)^2}. \tag{3.27}\]
3.3. Limit Cycle Analysis

Considering the condition on the imaginary part in (3.26) yields the frequency of the limit cycle as follows,

\[
\omega = \sqrt{\frac{k_p \beta}{k_p \tau_l \tau_T - k_d \alpha}} = \sqrt{\frac{k_p (1 + k_l \Psi(A))}{k_p \tau_l \tau_T - k_d (\tau_l + \tau_T + \tau_T k_l \Psi(A))}}. 
\] (3.28)

Furthermore, substituting \( \omega \) in (3.27) and applying the condition on the real part in (3.26) results in the following condition, after some algebraic manipulations,

\[
p_4 \beta^4 + p_3 \beta^3 + p_2 \beta^2 + p_1 \beta + p_0 = 0, 
\] (3.29)

where

\[
p_0 = k_T k_i k_p \tau_i^2 (k_p \tau_T - k_d)^2, 
\]
\[
p_1 = \tau_i (k_d - k_p \tau_T) (2 k_p \tau_l \tau_T + 2 k_p k_T k_i \tau_T^2 k_d - k_d \tau_l), 
\]
\[
p_2 = \tau_l (k_d - k_p \tau_T) (2 k_p \tau_l \tau_T + 2 k_p k_T k_i \tau_T^2 k_d - k_d \tau_l), 
\]
\[
p_3 = \tau_T (k_p \tau_T - k_d) (k_T k_i k_d^2 \tau_T + 2 k_d \tau_l - k_p \tau_l \tau_T), 
\]
\[
p_4 = \tau_l^2 k_d (k_p \tau_T - k_d), 
\]

By solving (3.29) for \( \beta \), four solutions are obtained of which two of them are always negative, one is always positive, and one can be positive depending on the parameters of the system. Since \( \beta \geq 0 \) by definition, two acceptable solutions are,

\[
\beta = \begin{cases} 
\frac{k_p \tau_l}{k_d} & \tau_l + k_T k_i (k_T \tau_T) + \sqrt{\tau_T^2 - 2 k_T k_i (k_T \tau_T) + k_T^2 k_i^2 k_d^2 k_\tau^2 + 4 k_T k_i \tau_T^2 \tau_T k_p} \\
\end{cases} 
\] (3.30)

Subsequently, \( \Psi(A) \) can be calculated as,

\[
\Psi(A) = \frac{\beta - 1}{k_i}. 
\] (3.31)

A limit cycle is predicted for any non-zero positive \( A \) that satisfies the equality given in (3.31). This is equivalent to the transfer function of the closed-loop system to be unity for the specified \( A \) and \( \omega \). Consequently, an oscillation will be regenerated in the loop, resulting in a limit cycle behavior.
Example 1  Considering $\beta = k_p \tau_l / k_d$, and $\Psi(A)$ as,

$$\Psi(A) = \frac{k_p \tau_l - k_d}{k_d k_l}. \quad (3.32)$$

Assuming $\Psi(A) > \dot{\theta}_m - \dot{\theta}_y$, the amplitude of the limit cycle can be calculated by solving the following equation for $A$,

$$\Psi_s \left( \frac{A}{\dot{\theta}_m - \dot{\theta}_y} \right) = 1 + \frac{k_p \tau_l - k_d}{k_d k_l \phi_4}. \quad (3.33)$$

Since $\phi_4$ is proportional to the MR fluid elasticity $\eta_r$, and $\eta_r$ is typically large for MR fluid (see Fig. 3.4), $\Psi_s$ can be approximated as,

$$\Psi_s \left( \frac{A}{\dot{\theta}_m - \dot{\theta}_y} \right) \approx 1.$$

Hence, $A \approx \dot{\theta}_m - \dot{\theta}_y$. By substituting (3.32) into (3.28), the frequency of the limit cycle can also be obtained.

3.3.4 Discussion

It is well-known that the proportional gain $k_p$ in a PD controller corresponds to the closed-loop bandwidth of the system. Hence, a higher performance can be achieved by increasing $k_p$. It is, however, necessary to study the maximum allowable $k_p$ in the A-DASA actuation before entering to a limit cycle. If $\Psi(A)$ is positive, then only $\beta \geq 1$ will be acceptable in (3.30). Therefore, the necessary condition for the limit cycle can be obtained as,

$$k_p \geq k_p^*, \quad (3.34)$$

where

$$k_p^* = \min \left\{ k_d, \frac{(\tau_l + \tau_T)(1 + k_T k_l \phi_4)}{k_T k_l \tau_T \tau_l} \right\} = \frac{k_d}{\tau_l}. \quad (3.35)$$

This implies that, for a given derivative gain, the maximum proportional gain before entering into a limit cycle is limited by the value of $\tau_l$. The maximum achievable derivative gain depends on the resolution of the encoder and its noise level. Moreover, considering (3.23), for a maximum achievable derivative gain, shows that the maximum allowable $k_p$ for robots with large inertia can be very small. This limitation on the value of $k_p$ indicates that high performance control may not be achievable for heavy robots. It can, however, be shown that the condition derived in (3.34) is too conservative.
Recalling (3.26), no \( \omega \) can be found to satisfy the condition on the imaginary part of \( G(j\omega) \) given in (3.27), if, 

\[
k_d \geq \frac{k_p \tau_l \tau_T}{\alpha} = \frac{k_p \tau_l \tau_T}{\tau_l + \tau_l \beta}.
\]  

(3.36)

That is, no limit cycle can be occurred for any \( k_d \) satisfying (3.36). Since the minimum value of \( \beta \) equals to one, no limit cycle presents if 

\[
k_p \leq \frac{k_d (\tau_l + \tau_T)}{\tau_l \tau_T}.
\]  

(3.37)

Therefore, at least one of the system time constants should be designed to be as small as possible in order to increase the allowable bound for \( k_p \). The value of \( \tau_l \) depends on the output shaft inertia and the inertia of the robot link attached to the actuator. Decreasing \( \tau_l \) corresponds to reducing the robot link inertia, which may not be feasible. It is therefore required to minimize \( \tau_T \), which is a function of the actuator geometry alone\(^3\). In conclusion, high bandwidth magnetic circuit is essential if the goal is to avoid the occurrence of limit cycles and attain high performance. One can note that if \( k_p \) is chosen such that \( k_p < k_d/\tau_l \), then the condition in (3.37) will also be satisfied. However, because 

\[
\frac{\tau_l + \tau_T}{\tau_l \tau_T} \geq \frac{1}{\tau_l},
\]

the condition in (3.37) is less conservative with respect to the controller gains, while both conditions guarantee the limit cycle prevention. Therefore, the maximum bound for \( k_p \) can be enlarged by designing an actuator possessing a high-bandwidth magnetic circuit.

### 3.4 Experimental Validations and Simulation

In this section, a set of model-based simulations along with experimental results are provided to validate our analysis. The experimental results were carried out using a 2-DOF manipulator. The 2-DOF manipulator (see Fig. 3.11) utilizes MR clutches as part of its actuation system. Two MR clutches configured antagonistically are used to actuate the first joint, while the second joint is actuated using a single MR clutch and a spring (for more details on the design of the manipulator see [2]). The antagonistic joint was only used in this set of experiments. The specifications of the MR clutches used in this manipulator were given in Table 3.1. The robot link inertia is 0.13 kg.m\(^2\). The manipulator is driven by a brushless motor (BLWRPG235D-36V-4000-R13) to provide the rotational inputs to the

\(^3\)\( \tau_T \) depends on the magnetic field time constant. The magnetic field time constant is a function of the coil inductance and the reluctance of the magnetic circuit forming magnetic flux. Both parameters depend on the internal dimensions of the actuator, and can be optimised in the actuator design stage.
MR clutches. The motor is driven by a driver in the velocity control mode. The manipulator incorporates two encoders (HEDS-9000) to measure the angular positions of the joints. Three high-power motor drivers (AMC-AZ12A8), set in current mode, provide the command currents to the MR clutches. In our experiments, both controllers were implemented on a desktop computer connected to the manipulator via a dSPACE (DS 1103) controller board. The sampling frequency for gathering experimental data was set to 1 kHz.

![Figure 3.11: A snap shot of the MR clutch during fabrication and the 2-DOF MR-actuator robot manipulator.](image)

### 3.4.1 Model-based Simulations

The A-DASA actuation in Fig. 3.8 for position control was considered. The output torque of the actuator was modeled by (3.8) and (3.10). The response of the system was simulated by solving the ordinary differential equation (3.11) in Matlab, considering the inertia of the robot link. Fig. 3.12 shows the simulation results for $k_p = 0.1$ and $k_d = 0$. As observed, no limit cycle was occurred. Fig. 3.13 shows the results for $k_p = 0.5$ and $k_d = 0$ when a limit cycle occurs in the response. Based on the previous analysis, adding $k_d$ can eliminate the limit cycle if the value of $k_d$ is chosen to be greater than $k_d$. The calculated value of $k_d$ for the current system is $k_d = 0.025$. To validate the results, the response of the system was simulated using two different values for $k_d$. Fig. 3.14 compares the results for $k_d = 0.01$ ($k_d < k_d$) and $k_d = 0.026$. The results clearly show that limit cycle oscillations can be avoided by applying $k_d \geq k_d$. On the other hand for $k_d < k_d$, the response of the system is trapped in a limit cycle. Increasing the derivative gain further in this case will alleviate the problem and allow the response to converge to the origin. This validates the results derived in this chapter.
Figure 3.12: (a) Simulated angular position and velocity of the 1st joint, (b) Phase portrait of the A-DASA in position control; $k_p = 0.1$, $k_d = 0$. The initial condition is marked by a circle.

Figure 3.13: (a) Simulated angular position and velocity of the 1st joint,(b) Phase portrait of the A-DASA in position control; $k_p = 0.5$, $k_d = 0$. The initial condition is marked by a circle.
3.4.2 Preliminary Experiments

The occurrence of limit cycle was investigated using the 2-DOF robot. Fig. 3.15 shows the experimental results for two different proportional gains. The proportional gains were \( k_p = 1 \) and \( k_p = 2 \), respectively. To perturb the robot from its steady state, a 100 mA current was applied at \( t = 2s \) and removed at \( t = 4s \). As seen, the frequency of the response changes with respect to the controller gain. Referring to (3.28), it can be shown that the limit cycle frequency for a P controller (for \( k_d = 0 \)) can be approximated by,

\[
\omega \simeq \sqrt{\frac{k_p k_f k_l}{\tau_l}}.
\] (3.38)

According to (3.38), the frequency of the limit cycle is expected to grow proportionally with the root square of \( k_p \). The limit cycle frequencies for \( k_p = 1 \) and \( k_p = 2 \) were \( f = 1.9048 \) Hz and \( f = 2.4876 \) Hz, respectively. The limit cycle frequency for \( k_p = 2 \) is approximately \( \sqrt{2} \) times the frequency of \( k_p = 1 \), confirming the theoretical analysis.

3.5 Conclusion

In this chapter, the occurrence of limit cycles in the behavior of an antagonistically coupled MR actuators was discussed. A practical example was studied to demonstrate the possibility of limit cycle induced by the antagonistic arrangement of the actuators when operated in the position control loop. Using describing function method, the dependency of the limit cycle on the actuator parameters was thoroughly analyzed, and the results were demon-
Figure 3.15: Experimental results using the 2-DOF robot; (a) $k_p = 1$, (b) $k_p = 2$.

strated. Further, the frequency and amplitude of the resultant limit cycle were analytically derived as functions of the system parameters. It was shown that the limit cycle oscillations could be prevented for specific PD controller gains. As an important observation, it was discussed that high-bandwidth magnetic circuits were essential in eliminating the limit cycle. Numerical simulations along with experimental results validated the theoretical analysis. The insight gained in this chapter can be used as a foundation for the design of antagonistically coupled MR actuators as well as the design of controllers aiming at the limit cycle prevention.
Bibliography


Chapter 4

Adaptive Modeling of Magneto-Rheological Actuators

The material presented in this chapter will be appeared in “IEEE/ASME Transactions on Mechatronics.” A part of this work has also been published in the proceeding of “IEEE International Conference of Robotics and Automation (ICRA),” pp. 2698-2703, Saint Paul, MN, 2012.

In this chapter, a new open-loop model for a Magneto-Rheological (MR) based actuator is presented. The model consists of two parts relating the output torque of the actuator to its internal magnetic field, and the internal magnetic field to the applied current. Each part possesses its own hysteretic behavior. The first part uses a novel nonlinear adaptive model that relates the internal magnetic field to the applied current. The second part uses an open-loop Bingham model to relate the output torque to internal magnetic field. The model facilitates accurate control of the actuator using its input current. It also eliminates the need for force/torque sensors for providing feedback signals. The accuracy of the constructed model is validated through simulations. The model is assessed against a widely accepted hysteresis modeling approach, known as Preisach model and its advantages are highlighted. Experimental results using the prototyped actuation mechanism further verify the accuracy of the model and demonstrate its effectiveness.
4.1 Introduction

In recent years, research on physical human-robot interactions has received considerable attentions. Of particular interest is the subject of safety within the interactive environments. The study of this subject has led to the study of new control and actuation mechanisms for robots. It has been shown that new actuation technologies are essential components of the future generations of human-safe robots [1–3].

Toward this objective, a new Magneto–Rheological (MR) based actuation mechanism was designed and developed in our research group. The actuator is capable of providing both rigid and compliant actuation. The main rational behind using MR Fluids (MRFs) in this actuation mechanism was to instantaneously and reversibly control the compliancy of the actuator using an applied magnetic field [4].

The main difficulty in employing the actuator is the nonlinear behavior of the actuator due to the use of MRF and a magnetic circuit built from ferromagnetic materials to form a flux path for the fluid. The MRFs and magnetic circuit each introduce hysteretic behavior in the current–torque curve of the actuator. The Greek word “Hysteresis” means “to lag behind”, and describes a relationship between inputs and outputs of a certain system. For a single-input, single-output system, hysteresis is the presence of a nondegenerate input-output closed curve as the frequency of excitation tends toward dc [5]. The presence of hysteresis leads to such known problems in the control systems as tracking errors, unwanted harmonics, and instability [6]. A high gain feedback control can compensate the effects of the hysteresis behavior [7]. However, a high gain feedback results in more power consumption and poor control performance. To compensate for the hysteresis, a hysteresis model is often required in designing the control algorithm. This model is used in order to predict nonlinear characteristics of the MRFs and magnetic circuit.

A well-known hysteresis modeling technique for ferromagnetic as well as smart materials is the Preisach model [8–10]. While, this model is widely accepted, it suffers from implementation problem. Mainly, the inverse of the Preisach model cannot be obtained analytically in order to be used in a feedback control loop. To address this issue, several researchers attempted to obtain numerical inverse models [11–15]. However, these models are computation- and/or storage-intensive. Among other hysteresis models are the Prandtl-Ishlinskii [16–18] and Krasnosel’skii-Pokrovskii [19] models. The Prandtl–Ishlinskii can exhibit neither asymmetric hysteresis loop nor saturated hysteresis output [20]. The Krasnosel’skii-Pokrovskii model also requires extensive computational resources. The formulations of all of these modeling techniques are based on experimental data measured in advance, and not the actual dynamical process. Thus, none of these models can
guarantee the stability and/or robust performance of the control system under all operating conditions.

Alternative approaches, in which the dynamics of the hysteretic behavior is considered in the modeling, are the use of a special form of nonlinearity [21], neural networks [22], and polynomial approximation [24]. The first approach cannot be easily generalized, while the second one is usually difficult in real-time applications due to complicated learning rules. The modeling approach outlined in this chapter is based on the polynomial approximation. This approach is advantageous since it can deal better with uncertain nonlinearities.

Khambanonda et al. [25] first introduced the idea of using polynomial approximation for the stability analysis of the nonlinear systems. Later, [26] and [24] studied similar approximation approach in the control of nonlinear systems. The qualitative nature of the nonlinear process was an a priori requirement. Moreover, no strategy was proposed to obtain the approximate model [26]. This drawback was addressed in [24] using an off-line least square method. However, the convergence analysis of the error between the approximate model and the actual model were not discussed.

The main contribution of this chapter is a new open-loop model of an MR based actuation mechanism with hysteretic behavior. The model consists of two parts (see Fig. 4.1) relating the output torque of the actuator to its internal magnetic field, and the internal magnetic field to the applied current. The first part uses a novel nonlinear adaptive identification method for modeling the relationship between the internal magnetic field and the applied current of the actuator. The second part uses an open-loop Bingham model to relate the output torque to the internal magnetic field. The identification method in the first part is based on the polynomial approximation of unknown dynamics. To this end, an on-line adaption law is proposed. The uniformly ultimately boundedness of the estimation error is proven using Lyapunov’s direct method. The model presented in this chapter encompasses the hysteretic behavior of the MRFs and the magnetic circuit. The model is evaluated against a widely accepted hysteresis modeling approach, known as Preisach model and its advantages are highlighted. Simulation and experimental results to support the validity of the model are also presented.

The chapter is organized as follow: Section 4.2 provides a literature background on human-friendly actuators along with a description of an MR based actuation mechanism. Section 4.3 discusses the mechanical and magnetic models of the MR actuator that is used as a platform for presenting our modeling approach. Section 4.4 presents simulation and experimental results and compares the results. Finally, Section 4.5 concludes the chapter.
4.2 Human-friendly Actuators

Typical rigid manipulators utilize a stiff connection between the motor and the link. A stiff connection normally results in a high output impedance that reflects the rotor and the link inertia. Thus, rigid manipulators can have a high impact even at a low-speed collision. In order to overcome this problem, a number of safe-oriented control techniques have been suggested to reduce the system output impedance, \[27, 28\]. However, these strategies are only effective for frequencies that are below the closed loop bandwidth of the control system \[29\].

In contrary to the “stiff” approach, a wide range of studies during the past two decades, \[30,33\] have tackled the problem through novel compliant actuation systems.

A compliant actuators using Magneto– or Electro–Rheological (ER) fluids can bound the amount of transmitting torque through the intensity of an applied field. Fig. 4.2 shows the cross–section of a typical multi–disk MR clutch. The input shaft breaks out into a set of input disks which are aligned in parallel to a set of output disks attached to the output shaft. MRF fills the volume between input and output disks. By energizing the electromagnetic coil, the viscosity of MRFs, thereby the compliancy of the clutch is controlled. The details of a prototype actuation mechanism that uses MR clutches are discussed in \[34\].

Despite their explicit advantages over other types of actuators, MR based actuators
4.2. **Human-friendly Actuators**

Figure 4.3: Snapshots of the MR clutch and the actuation mechanism that use MR clutches as part of the actuator.

Figure 4.4: Block diagram of the proposed model for a MR based actuator.
suffer from nonlinear hysteretic relationships between the input current and output torque. This nonlinear behavior causes inaccuracy in the output response of the actuator. It also results in the instability of the closed loop system \[35\].

It is essential to study and model the current–torque relationship for reliable control and actuation. This has been the subject of many studies. However, as justified in the Introduction section, there is still a clear need for a dynamic model that can be used in practical applications. The aim of this chapter is to provide an adequately accurate model for these purposes.

### 4.3 MR Based Actuator Modeling

In this section, a model consisting of two parts for a MR based actuator is proposed. The model relates the output torque of the actuator to its internal magnetic field, and the internal magnetic field to the applied current. Fig. 4.3 depicts snapshots of a prototype MR clutch and MR based actuation mechanism.

The block diagram of the proposed model is shown in Fig. 4.4. The first part of the block diagram represents the actuator magnetic circuit. It provides the relationship between the actuator internal magnetic field and its input current. This part uses a novel adaptive nonlinear model that is based on polynomial approximation method. In what follows, the adaptation law and the stability of this model are discussed. To provide the feedback signal required for adaptation, a set of hall sensors are used. The sensors measure the internal magnetic field using a proprietary arrangement within the MR clutch. The second part of block diagram represents the visco-plastic properties of the actuator using a Bingham model. The output of this model is the estimated value of the actuator torque.

#### 4.3.1 Current–Magnetic Field: adaptive polynomial approximation model

As discussed earlier, the magnetic field-current behavior in the MRF-based actuator is hysteretic. In classical terminology \[36,38\], hysteresis is often categorised to two types; (a) rate-independent, and (b) rate-dependent. A hysteresis whose periodic input-output map is independent of input frequency is called rate-independent. While, rate-dependent hysteresis points to the hysteresis system whose periodic input-output map is dependant to input frequency. Hysteresis can usually be detected experimentally by applying a periodic signal \( u = u_0 \sin(\omega t) \), where \( u_0 \) is the amplitude of the signal, and \( \omega \) is the frequency. If input-output curves converge asymptotically to a periodic closed loop curve when \( \omega \to 0 \),
then there is a hysteresis in a certain system, and the periodic curve is commonly called hysteresis loop [39].

In order to investigate the hysteresis in the MRF-based actuator, current inputs with different frequencies are applied to the actuator. Fig. 4.5 reveals that input-output (i.e. current-magnetic field) curves are converging to a periodic closed loop curve near dc, that is, at asymptotically low frequency. It can also be inferred that the hysteresis in the magnetic circuit of the MRF-based actuator is rate-dependent, as input-output curves differ for different frequencies.

To model the rate-dependent current-magnetic field hysteresis, a novel model that relates the actuator input current to its internal magnetic field is presented in this section. This model uses polynomial approximation approach to model hysteretic behavior of a magnetic circuit. It should be pointed out that any universal approximation method can be used in lieu of the current polynomial approximation. The main purpose of modeling is to compensate for the system nonlinearities in real-time and to achieve linear actuation using an embedded controller unit (e.g. FPGA) within the actuator. Given the limited space inside the actuator, the simplicity of the approximation method for enabling the use of a smaller processing unit becomes of primary importance. The suggested polynomial approximation and its resulting adaption rules deem the most computationally efficient methods for embedded applications.

Consider an affine nonlinear system,

\[ \dot{x} = f(x) + Bu, \]  
(4.1)

where \( f(\cdot) \in \mathbb{R}^n \) is an unknown nonlinearity, \( B \in \mathbb{R}^{n \times 1} \) is an unknown vector, and \( u \in \mathbb{R} \) is the input signal.

The system described in (4.1) may be rewritten as follows,

\[ \dot{x} = Ax + g(x) + Bu, \]  
(4.2)

where \( A \in \mathbb{R}^{n \times n} \) is a Hurwitz matrix, and \( g(x) = f(x) - Ax \). Assuming \( Q \) is a symmetric positive definite matrix, there exists a symmetric positive definite matrix \( P = P^T \) that
satisfies the following Lyapunov equation,
\[ A^T P + PA = -Q. \] (4.3)

Let us now assume that \( p(x) \) is a summation of homogenous polynomials of degree \( m \) \[^{[24]}\], i.e.,
\[ p(x) = A_1 x^{[1]} + A_2 x^{[2]} + \ldots + A_m x^{[m]}, \] (4.4)
where \( A_i, (i = 1, 2, \ldots, m) \) are \( n \times \left( \binom{n+i-1}{i} \right) \)-matrices, and \( x^{[i]} \) is the \( \left( \binom{n+i-1}{i} \right) \)-tuple of the \( i \)-forms in the components of \( x \), i.e.,
\[ x^{[i]} = [x_1^{i}, \alpha_{i1}x_1^{i-1}x_2, \alpha_{i2}x_1^{i-1}x_3, \ldots, x_{i}^{i}]^T, \]
in that the weights \( (\alpha_{i1}, \alpha_{i2}, \ldots) \) are chosen such that, \( \|x^{[m]}\| = \|x\|^m \).

**Example** Let \( x = [x_1, x_2]^T \) and \( m = 3 \), then
\[ p(x) = A_1 x^{[1]} + A_2 x^{[2]} + A_3 x^{[3]}, \]
is a homogenous polynomial of degree 3, where
\[
\begin{align*}
x^{[1]} & = [x_1, x_2]^T, \\
x^{[2]} & = [x_1^2, \sqrt{2}x_1x_2, x_2^2]^T, \\
x^{[3]} & = [x_1^3, \sqrt{3}x_1^2x_2, \sqrt{3}x_1x_2^2, x_2^3]^T.
\end{align*}
\]

Let us now assume that the nonlinear smooth function \( g(x) \) can be represented as,
\[ g(x) = p(x) + \epsilon(x), \] (4.5)
where \( \epsilon(x) \) is a bounded approximation error, i.e. \( \|\epsilon(x)\| \leq \bar{\epsilon} \). Then, \( g(\cdot) \) given in (4.5) can be approximated as,
\[ \hat{g}(\cdot) = \hat{p}(\cdot). \] (4.6)

Substituting \( g(\cdot) \) in (4.2) with its approximated value from (4.6), the following model can be constructed,
\[ \dot{x} = A\hat{x} + \hat{p}(\hat{x}) + \hat{B}u, \] (4.7)
where \( \hat{B} \) is an estimated value of \( B \), \( \hat{p}(\hat{x}) = \hat{A}_1 \hat{x}^{[1]} + \hat{A}_2 \hat{x}^{[2]} + \ldots + \hat{A}_m \hat{x}^{[m]}, \) and \( \hat{A}_i, (i = 1, 2, \ldots, m) \) are the estimates of \( A_i, (i = 1, 2, \ldots, m) \).

Defining \( \tilde{x}(t) = x(t) - \hat{x}(t) \), the estimation error dynamics can be given using (4.2) and (4.6) as,
\[ \dot{\tilde{x}} = A\tilde{x} + g(x) - \hat{p}(\hat{x}) + \tilde{B}u, \] (4.8)
where \( \tilde{B} = B - \hat{B} \). Recall that \( A \) is a known Hurwitz matrix.

The main result of this section is given in the following theorem.
Theorem 4.3.1 Consider the estimation error dynamic system given in (4.8). If the polynomial coefficients $\hat{A}_i(i = 1, 2, ..., m)$ and the vector $\hat{B}$ are updated according to the following rules,

\[
\hat{B} = \varsigma(-\kappa_B\|\hat{x}\|\hat{B} + P\hat{x}u) \quad (4.9)
\]
\[
\hat{A}_i = \varsigma(-\kappa_A\|\hat{x}\|\hat{A}_i + P\hat{x}(\hat{x}^{[i]})^T), i = 1, 2, ..., m \quad (4.10)
\]

where $\varsigma$, $\varsigma$, $\kappa_B$, and $\kappa_A_i$, $(i = 1, 2, ..., m)$ are positive constants, then the estimation error dynamic system (4.8) is uniformly ultimately bounded on any compact subset of $\mathbb{R}^n$.

Proof Consider a Lyapunov function candidate as,

\[
V = \frac{1}{2}\hat{x}^TP\hat{x} - \frac{1}{2}\varsigma^2\hat{B}^T\hat{B} + \frac{1}{2}\sum_{i=1}^{m} tr(\hat{A}_i^T\varsigma^{-1}\hat{A}_i), \quad (4.11)
\]

where $\hat{A}_i = A_i - \hat{A}_i$. The time derivative of (4.11) is given by,

\[
\dot{V} = \frac{1}{2}\hat{x}^TP\hat{x} + \frac{1}{2}\hat{x}^TP\hat{x} + \frac{1}{2}\varsigma^2\hat{B}^T\hat{B} + \sum_{i=1}^{m} tr(\hat{A}_i^T\varsigma^{-1}\hat{A}_i). \quad (4.12)
\]

By substituting (4.3), (4.8), and (4.9) into (4.12), one can obtain,

\[
\dot{V} = -\frac{1}{2}\hat{x}^TQ\hat{x} + \hat{x}^TP(\hat{g}(x) - \hat{p}(\hat{x})) + \hat{B}^T(\kappa_B\|\hat{x}\|\hat{B}) + \sum_{i=1}^{m} tr(\hat{A}_i^T\varsigma^{-1}\hat{A}_i). \quad (4.13)
\]

Now, adding and subtracting $\hat{x}^TP\hat{p}(\hat{x})$ to and from (4.13), we have,

\[
\dot{V} = -\frac{1}{2}\hat{x}^TQ\hat{x} + \hat{x}^TP\hat{p}(\hat{x}) + \hat{x}^TP(p(x) - \hat{p}(\hat{x})) + \hat{x}^TP\hat{p}(\hat{x}) + \hat{B}^T(\kappa_B\|\hat{x}\|\hat{B}) + \sum_{i=1}^{m} tr(\hat{A}_i^T\varsigma^{-1}\hat{A}_i), \quad (4.14)
\]

where $\hat{p}(\hat{x}) = p(x) - \hat{p}(\hat{x}) = \sum_{i=1}^{m} \hat{A}_i\hat{x}^{[i]}$.

Furthermore, introducing (4.10) into (4.14) and given the fact that for any two column vectors $M$ and $N$, $tr(MN^T) = tr(N^TM)$, then it can be shown that,

\[
\dot{V} = -\frac{1}{2}\hat{x}^TQ\hat{x} + \hat{x}^TP(p(x) - \hat{p}(\hat{x})) + \hat{x}^TP\hat{e}(x) + \hat{B}^T(\kappa_B\|\hat{x}\|\hat{B}) + \|\hat{x}\|\sum_{i=1}^{m} \kappa_A_i tr(\hat{A}_i^T\hat{A}_i). \quad (4.15)
\]

The polynomial functions are locally Lipschitz on any compact subset of $\mathbb{R}^n$, i.e. $\|p(x) - p(\hat{x})\| \leq L\|x - \hat{x}\|$, where $L > 0$ is a Lipschitz constant. Consequently, the following inequality holds for any compact subset of $\mathbb{R}^n$,

\[
\dot{V} \leq -\|\hat{x}\| \left\{ \kappa_{B2}\|\hat{B}\|^2 + \frac{1}{2}\sum_{i=1}^{m} \kappa_{A_i} \|\hat{A}_i\|^2_F - \nu_1 \right\}, \quad (4.16)
\]
where $\varrho_1 = \frac{1}{2} (\lambda_{\min}(Q) - 2L\|P\|)$ and $\varrho_2 = \frac{e}{2} \|B\|^2 + \frac{1}{2} \sum_{i=1}^{m} \kappa_i \|A_i\|_F^2$. Defining $D$ as

$$D = \left\{ (\|\hat{x}\|, \hat{B}, \hat{A}_i) \mid \|\hat{x}\| \leq \frac{\|P\|}{\varrho_1}, \|\hat{B}\| \leq \|B\|, \|\hat{A}_i\|_F \leq \|A_i\|_F \right\},$$

$\dot{V}$ is negative semi-definite outside of $D$. Therefore, $\hat{x}$, $\hat{B}$, and $\hat{A}_i$ are uniformly ultimately bounded. This completes the proof of the theorem.

**Remark** As it can be inferred from (4.9) and (4.10), the proposed adaptive model requires the magnetic field measurement. While this measurement can be used to directly predict the output torque of the MR actuator, a model between the input current (not magnetic field) and the output torque is required for control purposes. The key point behind proposing the current model is to use the data from the magnetic field measurements to provide a model between the control command i.e., input current and the output torque of the MR actuator. This model enables the design of high performance model-oriented controllers in order to achieve high fidelity torque control. It is well understood that conventional non-model based controllers will result in poor control performance in the presence of hysteresis and other nonlinear behaviors.

**Remark** Two parameters are effective in the results of the proposed model; a) the order of the polynomial, and b) the adaption gain. In general, a polynomial with higher order will result in higher accuracy in estimating system nonlinearities, at the expense of more polynomial coefficients and longer adaption process. The adaption convergence rate depends on the adaption gains in (4.9) and (4.10), and more specifically on matrix $P$. A $P$ matrix with larger eigenvalues generally leads to a faster convergence. However, the larger the eigenvalues of $P$ are, the closer the eigenvalues of $A$ get to the imaginary axes, for a constant $Q$ matrix. This condition reduces the stability margin of the adaptive system. A matrix $P$ with larger eigenvalues also expands the bound on the model estimation error. Therefore a trade-off between the model accuracy and the model convergence rate should be considered when selecting matrix $P$ or $A$.

### 4.3.2 Magnetic field–Torque: Bingham model

It is recognized that the typical relationship between shear stress and shear rate of Bingham fluids can imitate typical MRFs behaviors when a magnetic field is applied [40–42]. In this regard, Shames and Cozzarelli [43] developed an idealized mechanical model known as Bingham visco–plastic model. This model describes the rheological properties of MRFs. Based on this model, a visco–plastic model for the clutch prototype discussed previously was obtained in [4].
According to this model, the shear stress of the fluid can be controlled with the applied field as,

\[
\tau = \tau_y(H) + \eta \frac{dv}{dz}, \quad \tau > \tau_y
\]

(4.17)

where \( \tau \) is the shear stress, \( \tau_y \) is the field dependent yield stress, \( H \) is the magnetic field intensity, \( \eta \) is the Newtonian viscosity, and \( \frac{dv}{dz} \) is the velocity gradient in the direction of the field.

The velocity gradient in (4.17) can be assumed constant under conditions discussed in [4], yielding,

\[
\tau = \tau_y(H) + \eta \dot{\gamma}(r), \quad \tau > \tau_y
\]

(4.18)

where the shear rate \( \dot{\gamma} \) is defined as \( \dot{\gamma} = \omega r l_f^{-1} \), where \( \omega \) is the angular velocity between input and output shafts of the clutch and \( l_f \) is the gap between input and output disks. It is easy to show that the torque produced by a circumferential element at a radius \( r \) is given by,

\[
dT = 2\pi r^2 \tau dr.
\]

(4.19)

Assuming that the clutch has \( N \) output disks, the torque transmitted through the clutch can be obtained after substituting (4.18) into (4.19) and integrating across both faces of each output disk, i.e.,

\[
T = 2N \int_{R_1}^{R_2} 2\pi \left( \tau_y(H)r^2 + \eta \frac{\omega r^3}{l_f} \right) dr
\]

(4.20)

\[
= 4N\pi \left( \tau_y(H) \left( \frac{R_2^3 - R_1^3}{3} \right) + \frac{\eta \omega (R_2^4 - R_1^4)}{4l_f} \right).
\]

where \( R_1 \) and \( R_2 \) are the inner and outer radii of the disks, respectively. All other parameters are as defined previously. The viscosity \( \eta \) of the carrier fluid is typically in the range of 0.1 to 0.3 Pa-s.

The yield stress \( \tau_y \) is controlled by varying the intensity of the magnetic field inside the clutch. The yield stress depends on the magnetization properties of the particles suspended in the fluid. Data relating the yield stress of MRFs to applied magnetic fields is generally provided by the manufacturers (LORD Co.). The data for the MRF used in our study is presented in Fig. 4.6. The yield stress of the fluid increases almost linearly with respect to the magnetic field, as more particle chains will be formed in fluid. Gradually, the yield stress starts to saturate around certain point (800 mT for MRF-140 manufactured by LORD Co.) indicating that no more chains can be formed in the fluid due to the limited number of particle in MRF. Using polynomial fitting function in Matlab, a third-order polynomial can
be reasonably fitted to this data so as to express the yield stress as a function of flux density \( B_f \). The result will be a model as,

\[
\tau_y = c_1 B_f + c_2 B_f^2 + c_3 B_f^3,
\]

where \( c_1 = 47.763 \), \( c_2 = 47.702 \), and \( c_3 = -32.442 \) are constants.

In summary, the proposed model estimates the output torque of the actuator as a function of its input current. In this regard, the adaptation laws given in (4.9) and (4.10) are employed in order to predict the strength of the magnetic field within the actuator as in (4.7). The predicted value of the magnetic field is then used to obtain the corresponding yield stress as in (4.21). Finally, the Bingham visco-plastic model in (4.20) is utilized to predict the output torque of the actuator.

### 4.4 Experimental Results

In this section the performance of the proposed model is examined by comparing the predicted behavior of the MR clutch with its actual behavior. We also implemented a Preisach model to compare the results. The MR based actuation mechanism introduced in Section 4.3 was utilized as our experimental platform. This actuation mechanism uses a MR clutch in its core. The MR clutch has the torque capacity of 75 Nm. The clutch weighs 2.8 Kg and has an inertia of \( 0.19 \times 10^3 \) Kg.m\(^2\) without its coil. With its coil, the mass and inertia of the clutch are 4.5 Kg and \( 5.0 \times 10^3 \) Kg.m\(^2\), respectively. The MR clutch is driven by a servo amplifier (Maxon 4-Q-DC Servo-amplifier ADS 50/5) set up in torque mode for provide the command current. The clutch is mounted on an experimental platform (see Fig.4.7) that incorporates a static load cell (Transducer Techniques SBO-1K) on the output shaft for torque measurements. A servo motor (Maxon EC 60) provides the rotational input to the MR clutch. In our experiment, a PID controller implemented on a desktop computer was
4.4. **Experimental Results**

![Prototype Platform](image)

Figure 4.7: The prototype platform

used to control the actuator via a National Instruments (NI USB-6229) multifunction I/O device. The sampling frequency for gathering experimental data is set to 500 Hz. Using this setup, we performed several experiments. A sinusoidal input current with two different frequencies was applied to the actuator. A set of experimental data for two frequencies of 1 and 5 Hz is shown in Fig. 4.8. This figure compares the predicted values of the internal magnetic field and the output torque using the proposed adaptive model with those obtained experimentally using hall sensors and load cell, respectively. The first column of the figure presents the predicted magnetic field versus current input. While, the second column depicts predicted torque as a function of time. The proposed adaptive model was implemented in MATLAB. The figure clearly demonstrates the hysteretic nature of the magnetic field-current relationship. The results also show that the proposed adaptive model is capable of accurately predicting the internal magnetic field. Furthermore, the predicted torque is computed using the Bingham model and the estimated values of the magnetic field using the proposed adaptive model and Preisach model. A comparison between the predicted torque and the corresponding actual values of the torque is shown in the second column in Fig. 4.8. This figure shows a well agreement between the estimated output torque and that of the actual measurement using the combination of the adaptive and the Bingham models. Fig. 4.8 also compares estimated torque values from a Preisach model. The Preisach model was identified using our experimental data and the numerical technique described in [45]. In this regard, the input current was partitioned into 16 sub-ranges between 0 A to 4 A range. The input current was applied in a sinusoidal, step-wise decaying manner, where the frequency of the sinusoidal inputs was set to 0.1 Hz. More detail on the Preisach identification procedure can be found in [46].
Moreover, to assess the proposed model against more complex input current, a standard Multi-Sinusoids is considered. Multi-Sinusoidal input is essentially a sum of sinusoids...
4.4. Experimental Results

Figure 4.10: Predicted magnetic field and torque values corresponding to exponentially decaying sinusoid input current.

\[ u(t) = \sum_{k=1}^{d} a_k \cos(\omega_k t + \phi_k), \]  
\[ \text{where } a_k, k = 1, 2, ..., d \text{ are amplitudes of sinusoids. } \omega \text{ is the frequency of sinusoids, and } \phi_k, k = 1, 2, ..., d \text{ are the phases. The current input is obtained using (4.22), where ten different frequencies in the range of 0.5 Hz to 10 Hz are selected. The amplitudes } a_k \text{ are selected equal, and the phases } \phi_k \text{ are spread as described in the following based on Schroeder phase [48] for having as much input power as possible:} \]

\[ \phi_1 \text{ arbitrary, } \phi_k = \phi_1 - \frac{k(k - 1)}{d} \pi; \quad 2 \leq k \leq d. \]

Then, a dc offset is added to input current in order to provide positive current all the time to the actuator. Fig. 4.9 shows the input current signal applied to the clutch. The estimated magnetic field and the corresponding predicted output torque are also depicted in this figure. As observed, the proposed adaptive model is capable of estimating the magnetic field very accurately after a reasonably short transition (adaptation) time. Also, there is a well agreement between the predicted output torque and its measured values. In comparison, the Preisach model fails to estimate the magnetic field as accurate as the adaptive model, mainly due to the rate-dependent hysteresis in current-magnetic field behavior. Consequently, the error in the estimation of the magnetic field using the Preisach model worsens the prediction of the output torque.
Figure 4.11: A sample of uneven surface require grinding

Figure 4.12: Griding application results; predicted magnetic field and torque values from the proposed model and the corresponding values from actual measurements and Preisach model.
Furthermore, to demonstrate and validate the ability of the model in capturing multi-loop characteristic of the hysteresis, an exponentially decaying sinusoid is used. Fig. 4.10 displays the magnetic field and the output torque estimations corresponding to the exponentially decaying sinusoid. The results clearly demonstrate the advantages of the adaptive model to the Preisach model in representing multi-loop characteristic of the current-magnetic field hysteresis in the actuator. As a result, more accurate prediction of the output torque can be obtained through the combination of the adaptive model and the Bingham model in comparison to the combination of the Preisach model and the Bingham model, as shown in the figure.

To evaluate the effectiveness of the proposed model, a real-life example of grinding an uneven surface is considered. Fig. 4.12 illustrates the associated input current required for the actuator in order to maintain a constant normal force on the surface of the metal shown in Fig. 4.11. The predicted magnetic field and the corresponding torque for this current are also shown in Fig. 4.12. Comparing the results clearly indicates that the predicted signal closely matches experimental measurements. These results are also compared with those obtained using the Preisach model. As observed, the combination of the proposed adaptive model and the Bingham model can predict the behavior of the MR based actuator more accurately. It is clear that the Preisach model does not perform well in this case. In order to assist drawing a conclusion, Table 4.1 lists the Root Mean Square Error (RMSE) for each sinusoidal inputs, Multi-Sinusoids, exponentially decaying sinusoid, and the input signal for the grinding application. It is clear that the prediction error of the proposed model is much smaller than the one from the Preisach model.
Table 4.1: RMSE values of the Magnetic field-current modeling and torque-current prediction

<table>
<thead>
<tr>
<th></th>
<th>1 Hz Sine</th>
<th>5 Hz Sine</th>
<th>Multi-Sines</th>
<th>Exp-Decay Sine</th>
<th>Grinding App.</th>
</tr>
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<tr>
<td>Magnetic field modeling</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>Preisach RMSE (mT)</td>
<td>7.4148</td>
<td>17.1596</td>
<td>7.1060</td>
<td>5.4110</td>
<td>8.7722</td>
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<tr>
<td>Adaptive model RMSE (mT)</td>
<td>4.4670</td>
<td>6.7444</td>
<td>3.1335</td>
<td>1.6474</td>
<td>1.0510</td>
</tr>
<tr>
<td>Output torque prediction</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Preisach &amp; Bingham RMSE (Nm)</td>
<td>3.7389</td>
<td>4.4889</td>
<td>3.4000</td>
<td>1.9954</td>
<td>1.9970</td>
</tr>
<tr>
<td>Hall sensor &amp; Bingham RMSE (Nm)</td>
<td>2.4074</td>
<td>1.7656</td>
<td>1.4481</td>
<td>1.1473</td>
<td>1.0947</td>
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<tr>
<td>Adaptive &amp; Bingham RMSE (Nm)</td>
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<td>0.9126</td>
<td>1.9076</td>
<td>1.1888</td>
<td>1.1305</td>
</tr>
</tbody>
</table>
4.5 Conclusion

In this chapter, a novel model for predicting the behavior of MR based actuators was proposed. The model consisted of an adaptive model based on polynomial approximation for estimating the internal magnetic field, and the Bingham model to predict the output torque without using torque/force sensor. The stability analysis of the proposed adaptive model using Lyapunov’s direct method and the adaptation laws for minimizing the prediction errors were discussed. The validity of the modeling results was verified using simulation and experimental results. The results illustrated an accurate and competitive approximation model. As part of our ongoing research, it is expected that this model will enable us to perform high fidelity force/torque control in such demanding applications as physical human-robot interactions. Moreover, the adaptive approximation model presented in this chapter can be applied to identify a wide class of unknown nonlinear dynamic systems including those built with hysteretic smart materials.
Bibliography


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Chapter 5

Adaptive Control of Magneto-Rheological Actuators

The material presented in this chapter is submitted in “IEEE/ASME Transactions on Control Systems Technology.” A part of this work has also been published in the proceeding of “IEEE/RSJ International Conference on Intelligent Robots and Systems (IROS),” Tokyo, 2013.

In this chapter, adaptive compensation of the hysteresis in a Magneto-Rheological (MR) fluid based actuators and its application for sensor-less high fidelity force/torque control is investigated. The MR actuator considered in this chapter was originally described in [1] and [2]. This actuator offers high torque-to-mass and torque-to-inertia ratios. Yet, as an essential component of the system, the magnetic circuit of the MR actuator experiences hysteresis between the input current and the resultant magnetic field. The hysteresis in the magnetic circuit results in a similar relationship between the input current and the output torque of the MR actuator. The control scheme used with the actuators possessing hysteresis often requires compensating for the hysteresis. To this end, we propose an adaptive control method based on feedback linearization that estimates both hysteresis and uncertain parameters of the magnetic circuit. A set of experiments is performed to validate the effectiveness of the proposed method.
5.1 Introduction

Magneto-Rheological (MR) fluids are a special class of fluids that exhibit variable yield stress with respect to an applied magnetic field. MR fluids offer unique properties enabling their use in variety of electro-mechanical devices. The states of MR fluids can vary from a fluid to a semi-solid (or plastic) state upon exposing to a magnetic field, with the ability to achieve up to 100 kPa yield stress in a matter of few milliseconds. Their controllability and fast responses to external magnetic field have made MR fluids an attractive technology for a broad range of applications from civil engineering to automotive, robots, and rehabilitation applications (e.g. see [3–5]).

Benefits of a controllable actuator utilizing MR fluids has been recognized in multiple robot applications. Design and development of several haptic devices based on MR fluids were presented in [6–8]. It has been shown that the actuation performance can be enhanced using MR actuators both in robot manipulators [9] and haptic devices [10]. The advantages of MR actuators in variable impedance actuation were discussed in [11][12]. Moreover, the ability of MR actuators to enhance the safety of human-friendly robots based on the intrinsic passivity of MR actuators was discussed in [13][14][15].

The main difficulty in employing MR actuators in robotic applications lies in the non-linear behavior of the actuator due to the use of ferromagnetic materials in the magnetic circuit of the fluid. The magnetic circuit introduces hysteresis in the current–torque curve of the actuator. The Greek word “hysteresis” means “to lag behind”, and it describes a relationship between the inputs and outputs of a certain system. For a single-input, single-output system, hysteresis is the presence of a non-degenerate input-output closed curve as the frequency of excitation tends toward a DC signal [15]. The presence of hysteresis leads to such known problems in control systems as tracking errors, unwanted harmonics, and instability [16]. A high gain feedback control can compensate for the negative effects of the hysteresis [17]. However, high gain feedbacks often result in more power consumption and poor control performance. To compensate for the hysteresis, a hysteresis model is often required in designing the control algorithm. The accuracy of the model plays an important role in the effectiveness of the control and enhancing the system performance.

In hysteresis modeling two main approaches, namely a) phenomenological modeling and b) physic-based modeling are often discussed. Phenomenological models include Preisach [18][19], Prandtl-Ishlinskii [20], and Krasnoselskii-Pokrovskii [21] models. Although these models are widely accepted and can successfully predict magnetic hysteresis, they present several implementation problems that restrict their applications for the systematic design of closed-loop controllers [22]. Moreover, the formulation of phenomenological
models is based on prior experimental measurements, that makes the use of such models controversial in applications in which reliability and/or robust performance of the control are of primary importance. Alternatively, analytical models are structured around the physical principles of magnetic systems. Jiles-Atherton’s model [23] and Hodgdon’s model [24] are examples of such models for ferromagnetic hysteresis. Jiles-Atherton’s model was applied in [25] to analyze the magnetic hysteresis of an MR actuator. Hodgdon’s hysteresis model was also used to predict the magnetic circuit behaviors of an MR actuator in [9].

While these models have been used for either off-line simulations or actuator designs, to the best of our knowledge, no real-time implementation of the above mentioned models has ever been reported for the control design purposes. Given the fact that most non-model based controllers (e.g. PID\footnote{Proportional-Integral-Derivative} controller in [26]) result in poor performance, the need for a model-based controller for utilizing MR actuators is apparent for achieve high performance and delivering high-fidelity force/torque control.

The main contribution of this chapter is the design of a new closed-loop control for actuators with hysteretic behavior. A new adaptive controller is proposed to deal with uncertainties of MR actuators. In the design of the controller, both geometrical parameters of the actuators and physics of the system are considered in order to compensate for the hysteretic and nonlinear behaviors of MR actuators. The main advantage of the proposed controller is that it requires only the estimation of the output torque, eliminating the need for any additional force/torque sensors in the control loop. This sensor-less force/torque control technique offers significant advantages in terms of cost reduction and performance improvement thanks to unique properties of MR fluids. The stability of the closed-loop system is rigorously analyzed and the conditions on which the error dynamic remains stable are derived. The proposed scheme is experimentally validated using a 2-DOF planar robot as a test bench. Furthermore, our sensor-less force/torque control results are compared to those obtained using direct force/torque measurements and the advantages are highlighted.

The rest of this chapter is organized as follow: Section 5.2 introduces MR fluids along with their applications in electro-mechanical actuators. Section 5.3 discusses the characteristics of MR fluids, and their controllable yield stress. Dynamic models of the magnetic circuit and the torque-current relationship in MR actuators are also described in this section. Section 5.4 presents our proposed adaptive control method. In Section 5.5 experimental results for torque control are presented. Finally, Section 5.6 concludes the chapter.
5.2 Magneto-Rheological Actuators

The Magneto-Rheological (MR) effect was discovered by Jacob Rabinow in 1940s who developed the first MR-based device \[27,28\] at the US National Bureau of Standards. MR fluids are non-homogenous suspensions of micrometer-sized ferromagnetic particles in a carrier fluid. The apparent viscosity of MR fluids can be adjusted by an external magnetic field. The suspended particles in the fluid form columns (chains) aligned to the direction of the applied field that results in shearing or flow resistance in the fluid. The degree of the resistive force is related to the strength of the magnetic field, resulting in a field dependent yield stress in MR fluids \[29,30\]. In the absence of a magnetic field, MR fluids act as Newtonian fluids, whose viscosity changes proportionally to the shear rate.

Actuators based on MR fluids have several advantages over conventional actuators including fast response time, high torque density, low power consumption (2-50 Watts), and intrinsic passivity \[31\]. The insensitivity to contamination, durability, and long service life of MR fluids further empower their commercial utilization.

The operational modes of MR-based devices can mainly be categorized into three different modes; flow mode (also know as valve mode), direct shear mode, and squeeze film mode \[32-35\], as demonstrated in Fig. 5.1. Linear and rotary MR actuators (e.g. clutches, brakes, and dampers) utilize either the flow mode \[36\] or the shear mode \[37\], and in some cases a combination of both \[38-40\]. The squeeze mode is often used for axial or rotary operations with limited torque/force capacity \[41\]. Recently, two new operation modes, the so called pinch mode and reversible jamming mode, have also been discovered with the improvements in zero-field friction and the fluid functionality \[42\]. The characteristics of an MR actuator highly depends on the mode in which the actuator operates, and differ from one type to another. The focus of this chapter is on MR clutches only and their application for force/torque control in robotic systems.
5.3 MR Actuator Model

MR clutches can be employed in the actuation mechanism of robot manipulators for controlling the delivery of output torque at the joints. Fig. 5.2 depicts a possible arrangement for the actuation of a robot joint. The active drive (i.e. motor) provides power to the joint via an MR clutch that controls the output torque. The advantages of using MR clutches in robot control were studied in [43]. The utilization of MR clutches at robot joints enhances the robot performance while reducing the impedance for more human-friendly actuation. The high torque-to-mass ratio and the fast transient response of the MR clutch in both position and torque control modes are among other benefits of MR fluid based actuation. While extensive studies in the literature report on MR actuator modeling, most models are predominantly focused on force-velocity behaviors of MR actuators. This is because of the fact that these models are primarily developed for MR dampers where the desired operation is vibration damping. In the case of MR clutches, however, the relation between the input current and the output torque becomes of higher importance. Hence, our attention in Section 5.3 will be on the modeling of torque-current behavior of MR clutches.

5.3 MR Actuator Model

Fig. 5.3 shows the cross-section of a typical multi-disk MR clutch. The input shaft breaks out into a set of input disks which are aligned in parallel to a set of output disks attached to the output shaft. MR fluid fills the volume between input and output disks. By energizing the electromagnetic coil, the apparent viscosity of MR fluids, thereby the compliance of the clutch can be controlled. A model for an MR clutch based actuation should consist of two parts; a) the MR clutch magnetic circuit that relates the input current to the resultant magnetic field, and b) the mechanical dynamics of the MR clutch that relates the output torque of the actuator to its internal magnetic field. In what follows we describe each of these parts in more details.
5.3.1 MR Clutch Magnetic Circuit Model

Fig. 5.4 shows a simplified model of the magnetic circuit of a typical MR clutch, where \( R_c \), \( R_f \), and \( R_d \) are the reluctance of the core, the gaps between the disks, and the disks, respectively. The reluctance of a simplified geometric model was derived parametrically in [44] for a multi-disk MR clutch. Assuming the magnetic field is uniform, steady, and perpendicular to the cross sectional area of \( A \), the magnetic field intensity \( H \) can be calculated as below,

\[
H = \frac{\Phi}{\mu A},
\]

(5.1)

where \( \Phi \) is the magnetic flux in the circuit obtained based on the total reluctance of the circuit \( R_t \), the number of winding \( N \), and the input current \( i(t) \). It should be pointed out that the design of our MR clutch was carried in such a way that these assumptions would hold as much as possible.

The flux variation \( \Phi \) caused by an alternating input current results in an induced magnetic field \( H_d \) in the opposite direction of the applied magnetic field based on the Farady-
5.3. MR Actuator Model

Lenz law. The flux variation is equivalent to the magnetic field density variation $\dot{B}$ that is related to the induced magnetic field as

$$H_d = -\lambda \dot{B},$$

(5.2)

where $\lambda$ depends on the geometry of the magnetic circuit. The effective magnetic field $H_{\text{eff}}$ can be represented as the sum of the applied field $H$ and the induced field $H_d$, i.e.,

$$H_{\text{eff}} = H - H_d = H - \lambda \dot{B}.$$  

(5.3)

In diamagnetic and paramagnetic materials, the magnetic field density $B$ at any point within a magnetic circuit is related to the magnetic field intensity and the permeability of the material through a linear B-H relationship, i.e. $B = \mu H_{\text{eff}}$, whereas ferromagnetic materials exhibit a nonlinear and hysteretic relation as,

$$B \in \mathcal{B}(H_{\text{eff}}),$$

(5.4)

where $\mathcal{B}$ is a suitable hysteresis operator [20, ch. 2]. Considering (5.3) and (5.4), the magnetic circuit behavior of an MR clutch indicates a rate-dependent hysteresis (also known as visco-hysteresis). This phenomenon is demonstrated in Fig. 5.5 where we have applied a sinusoidal input current with different frequencies to the MR clutch. Fig. 5.5 clearly displays the frequency dependent nature of the hysteresis of the MR clutch.

It is essential to mention that the hysteresis in the magnetic field corresponds to the hysteresis caused mainly by ferromagnetic components within an MR actuator including the shaft, disks, coil, etc., and not the MR fluid itself. The MR fluid exhibits relatively a linear magnetic behavior due to the soft iron used in the fluid suspension, and as such little or no hysteresis is observed in the B-H curve of the fluid[30].

5.3.2 MR Clutch Mechanical Model

It is recognized that the typical relationship between shear stress and shear rate in Bingham fluids can imitate the behavior of MR fluids under an applied magnetic field [30,45,46]. In this regard, Shames and Cozzarelli [47] developed an idealized mechanical model known as Bingham visco–plastic model. This model describes the rheological properties of MR fluids. Based on this model, a visco–plastic model for a typical multi-disk clutch can be obtained as a function of the yield stress in MR fluids and the relative velocity between the input and output shafts [1].

According to this model, the shear stress of the fluid is given by,

$$\tau = \tau_y(B) + \eta \frac{dv}{dz}, \quad \tau > \tau_y$$  

(5.5)
where \( \tau \) is the shear stress, \( \tau_y \) is the field dependent yield stress, \( B \) is the magnetic field density, \( \eta \) is the Newtonian viscosity, and \( \frac{dv}{dz} \) is the velocity gradient in the direction of the field.

Assuming the velocity gradient is constant\(^2\), then (5.5) yields,

\[
\tau = \tau_y(B) + \eta \dot{\gamma}(r), \quad \tau > \tau_y,
\]

where the shear rate \( \dot{\gamma} = \omega{l_f}^{-1} \), \( \omega \) is the angular velocity between the input and output shafts of the clutch, and \( l_f \) is the gap between the input and output disks. It is easy to show that the torque produced by a circumferential element at a radius \( r \) is given by,

\[
dT = 2\pi r^2 \tau dr.
\]

Assuming that the clutch has \( N \) output disks, the torque transmitted through the clutch can be obtained after substituting (5.6) into (5.7) and integrating across both faces of each

---

\(^2\)It can be argued that this assumption is not restrictive and can always be met if the radii of the disks are much greater than the thickness of the fluid filling the gaps between the disks.
output disk, i.e.,

\[ T = 2N \int_{R_1}^{R_2} 2\pi \left( \tau_y(B)r^2 + \eta \omega r^3 \right) dr \]

\[ = 4N\pi \left( \frac{\tau_y(B)(R_2^3 - R_1^3)}{3} + \frac{\eta \omega (R_2^4 - R_1^4)}{4l_f} \right), \]

where \( R_1 \) and \( R_2 \) are the inner and outer radii of the disks, respectively. All other parameters are as defined previously. The viscosity \( \eta \) of the carrier fluid is typically in the range of 0.1 to 0.3 Pa-s.

The yield stress \( \tau_y \) is controlled by varying the magnetic field inside the clutch. The data relating the yield stress of MR fluids to applied magnetic fields is generally provided by the manufacturers (LORD Co.). Fig. 5.6 shows the data for the MRF-140 fluid manufactured by LORD Corporation, as an example. The yield stress of the fluid increases almost linearly with respect to the magnetic field, as more particle chains will be formed within the fluid. Gradually, the yield stress starts to saturate around certain point (e.g., 800 mT for MRF-140) indicating that no more chains can be formed in the fluid due to the limited number of particles in the MR fluid. To express the yield stress of the fluid as a function of the flux density \( B \), a polynomial can be fitted to the data. Applying the polynomial fitting function in Matlab to the data for MRF-140 results in the following analytical model,

\[ \tau_y = c_1B + c_2B^2 + c_3B^3, \]

where \( c_1 = 47.763 \), \( c_2 = 47.702 \), and \( c_3 = -32.442 \) are constants.

Figure 5.6: Characteristics of the LORD MRF-140CG (Adopted from manufacturer datasheet [48])

It should be pointed out that MR fluids can exhibit a nonlinear roll-off phenomenon in their torque-shear rate near zero relative velocity [49-51]. However, this phenomenon is unique to MR dampers, and can be easily avoided in MR clutches. By controlling the input shaft velocity of an MR clutch, the relative angular velocity (shear rate) can be maintained within the linear torque-shear rate region.
In summary, with insight into the physics of magnetic fields and fluid mechanics, an MR clutch model can be derived to predict the output torque of the actuator as a function of its input current (see Fig. 5.7). In this regard, the magnetic field density within the actuator is determined using (5.1)-(5.4) while (5.9) gives the fluid yield stress corresponding to the obtained magnetic field. Finally, the Bingham model in (5.8) predicts the output torque of the actuator.

### 5.4 Torque Control for MR Clutch Based Actuation

In this section force/torque control using an MR actuator is discussed. To this effect, the torque-current relationship of the MR clutch can be obtained after solving (5.3) for \( \dot{B} \) and using the values of \( H \) and \( H_{\text{eff}} \) as in (5.1) and (5.8), respectively, i.e.,

\[
\dot{B} = -\lambda^{-1}H(B) + \theta i(t),
\]

\[
T = \varphi_1 \tau_y(B) + \varphi_2 \omega,
\]

where \( H = B^{-1} \) is a hysteresis operator to be determined, \( \theta = \lambda^{-1}N/(\mu AR_y) \), and \( \varphi_1 \) and \( \varphi_2 \) are the geometrical parameters defined as,

\[
\varphi_1 = 4N\pi(R_2^3 - R_1^3)/3,
\]

\[
\varphi_2 = \eta N\pi(R_2^4 - R_1^4)/l_f.
\]

Defining \( e_T(t) = T_d(t) - T(t) \) as the control error, where \( T_d \) is the desired torque, the control error dynamics are given by,

\[
\dot{e}_T(t) = \dot{T}_d(t) - \dot{T}(t),
\]

\[
= \dot{T}_d(t) - \varphi_1 \dot{B} \left( \partial B_y(B)/\partial B \right) - \varphi_2 \dot{\omega},
\]

\[
= \dot{T}_d(t) - \varphi_2 \dot{\omega}
\]

\[
+ (\varphi_1 \lambda^{-1}H(B) - \varphi_1 \partial i(t)) \left( \partial \tau_y(B)/\partial B \right).
\]

(5.11)
Assuming all parameters of the system are known, \( i(t) \) can be chosen so as to linearize the system described in (5.11), i.e.,

\[
i(t) = \left( \varphi_1 \partial \left( \partial \tau_y(B)/\partial B \right) \right)^{-1} (\varphi_1 \lambda^{-1} \mathcal{H}(B)(\partial \tau_y(B)/\partial B)
+ \hat{T}_g(t) - \varphi_2 \dot{\omega} + ae_T),
\]

(5.12)

where \( a > 0 \) is a constant scalar. Introducing (5.12) into (5.11) results in the following stable linear dynamics,

\[
\dot{e}_T(t) = -ae_T(t)
\]

(5.13)
in that, the error approaches zero as \( t \rightarrow \infty \). In practice, however, \( \mathcal{H}(B), \lambda^{-1}, \) and \( \vartheta \) are unknown, and the input control law given in (5.12) cannot be constructed.

To deal with this issue, let us assume that the nonlinear function \( \lambda^{-1} \mathcal{H}(B) \) can be presented as,

\[
\lambda^{-1} \mathcal{H}(B) = \mathcal{H}_a(B) + \epsilon(B),
\]

(5.14)

where \( \epsilon(B) \) is a bounded approximation error, i.e. \( \| \epsilon(B) \| \leq \bar{\epsilon} \), and \( \mathcal{H}_a(B) \) is a polynomial of degree \( m \), i.e.,

\[
\mathcal{H}_a(B) = \sum_{i=0}^{m} \alpha_i B^i,
\]

(5.15)
in that \( \alpha_i, (i = 1, 2, \ldots, m) \) represents the unknown parameters to be estimated. As such, \( \mathcal{H}_a(B) \) can be approximated as,

\[
\hat{\mathcal{H}}_a(B) = \sum_{i=0}^{m} \hat{\alpha}_i B^i,
\]

(5.16)

Using the estimated values of \( \lambda^{-1} \mathcal{H}(B) \) and \( \vartheta^{-1} \) results in a new input control law given by,

\[
\hat{i}(t) = \left( \varphi_1 \hat{\vartheta} \left( \partial \tau_y(B)/\partial B \right) \right)^{-1} (\hat{T}_d(t) - \varphi_2 \dot{\omega}
+ ae_T + \varphi_1 \hat{\mathcal{H}}_a(B)(\partial \tau_y(B)/\partial B)).
\]

(5.17)

Introducing the control law given in (5.17) into (5.11) leads to the following control error dynamics,

\[
\dot{e}_T(t) = \hat{T}_d(t) - \varphi_2 \dot{\omega}
+ \left( \varphi_1 \lambda^{-1} \mathcal{H}(B) - \varphi_1 \vartheta \hat{i}(t) \right) \left( \partial \tau_y(B)/\partial B \right)
+ \varphi_1 \left( \partial \tau_y(B)/\partial B \right) \left( \mathcal{H}_a(B) - \hat{\mathcal{H}}_a(B) \vartheta \hat{\vartheta}^{-1} \right)
- \hat{\vartheta} \hat{\vartheta}^{-1} e_T + \varphi_1 \epsilon(B) \left( \partial \tau_y(B)/\partial B \right).
\]

(5.18)
By adding and subtracting \( ae_T + \varphi_1(\partial \tau_y(B)/\partial B)\dot{H}_i(B) \) to and from (5.18) and after some algebraic manipulations the new control error dynamics can be written as follows using (5.15) and (5.16),

\[
\dot{e}_T(t) = -ae_T + u_c(1 - \vartheta \hat{\vartheta}) + \varphi_1(\partial \tau_y(B)/\partial B)\left( \sum_{i=0}^{m} \tilde{\alpha}_i B^i + e(B) \right),
\]

(5.19)

where \( \hat{\vartheta} = \vartheta^{-1} \) and \( u_c = \dot{T}_d - \varphi_2 \dot{\omega} + \varphi_1 \dot{H}_i(B)(\partial \tau_y(B)/\partial B) + ae_T \) and \( \tilde{\alpha}_i = \alpha_i - \hat{\alpha}_i, (i = 0, 1, ..., m) \).

The main result of this section is given in the following theorem.

**Theorem 1** Assuming the sign of \( \vartheta \), i.e. \( \text{sgn}(\vartheta) \), is known, \(^3\) if the control input (5.17) is applied to the system described in (5.10), and the polynomial coefficients \( \hat{\alpha}_i, (i = 0, 1, ..., m) \) and the parameter \( \hat{\vartheta} \) are updated according to the following rules,

\[
\dot{\hat{\vartheta}} = \iota \left( -\kappa_\vartheta |e_T| \hat{\vartheta} + \text{sgn}(\vartheta) u_c e_T \right),
\]

(5.20)

\[
\dot{\hat{\alpha}}_i = \sigma \left( -\kappa_{\alpha_i} |e_T| \hat{\alpha}_i + \varphi_1 \left( \partial \tau_y(B)/\partial B \right) e_T B^i \right),
\]

(5.21)

where \( \iota > 0, \sigma > 0, \kappa_\vartheta, \) and \( \kappa_{\alpha_i} \) are constant gains, then the control error \( e_T \) is uniformly ultimately bounded, i.e. \( |e_T| \leq \rho \), in that \( \rho \) is inversely proportional to the control parameter \( a \) in (5.17), and can be arbitrarily small.

**Proof** Consider the following Lyapunov function candidate,

\[
V = \frac{1}{2} \left\{ e_T^2 + |\vartheta| \tilde{\vartheta}^2 + \sigma^{-1} \sum_{i=0}^{m} \tilde{\alpha}_i^2 \right\},
\]

(5.22)

where

\[
\tilde{\vartheta} = \vartheta - \hat{\vartheta} = \vartheta^{-1} - \hat{\vartheta}^{-1},
\]

the time derivative of (5.22) is given by,

\[
\dot{V} = e_T \dot{e}_T + |\vartheta| \tilde{\vartheta} \tilde{\vartheta}^2 + \sigma^{-1} \sum_{i=0}^{m} \tilde{\alpha}_i \dot{\hat{\alpha}}_i,
\]

(5.23)

\(^3\)The sign of \( \vartheta \) can be easily found by applying a step input current to the coil (see (5.10)).
By substituting (5.19) and (5.20) into (5.23), one can obtain,

\[ \dot{V} = -ae^2_T + u_c e_T (1 - \theta \hat{\theta}^{-1}) - u_c \dot{e}_T \theta (\dot{\theta}^{-1} - \hat{\theta}^{-1}) + \varphi_1 e_T \left( \frac{\partial \tau_y(B)}{\partial B} \right) \left( \sum_{i=0}^{m} \tilde{\alpha}_i \tilde{B}_i + \epsilon(B) \right) + \kappa_\nu |\theta| e_T |\ddot{\nu}| + \sigma^{-1} \sum_{i=0}^{m} \tilde{\alpha}_i \dot{\hat{\alpha}}_i \]

Further, introducing (5.21) into (5.24) and given the fact that \( \left( \frac{\partial \tau_y(B)}{\partial B} \right) \) is a bounded function of \( B \), the following inequality holds,

\[ \dot{V} = -ae^2_T + \kappa_\nu |\theta| e_T |\ddot{\nu}| + \sigma^{-1} \sum_{i=0}^{m} \tilde{\alpha}_i \dot{\hat{\alpha}}_i + \varphi_1 e_T \left( \frac{\partial \tau_y(B)}{\partial B} \right) \left( \sum_{i=0}^{m} \tilde{\alpha}_i \tilde{B}_i + \epsilon(B) \right) \leq -|e_T| \left\{ a|e_T| + \frac{\kappa_\nu}{2} |\ddot{\nu}|^2 + \frac{1}{2} \sum_{i=0}^{m} \kappa_\alpha \tilde{\alpha}_i^2 - \varrho \right\} \]

Furthermore, introducing (5.21) into (5.24) and given the fact that \( \left( \frac{\partial \tau_y(B)}{\partial B} \right) \) is a bounded function of \( B \), the following inequality holds,

\[ \dot{V} = -ae^2_T + \varphi_1 e_T e(B) \left( \frac{\partial \tau_y(B)}{\partial B} \right) + \kappa_\nu |\theta| e_T |\ddot{\nu}| + |e_T| \sum_{i=0}^{m} \kappa_\alpha \tilde{\alpha}_i \dot{\hat{\alpha}}_i \]

where \( |\partial \tau_y(B) / \partial B| \leq \tilde{\delta}_r \) and \( \varrho = \varphi_1 \tilde{e} \tilde{\delta}_r + \frac{\kappa_\nu}{2} |\nu| + \frac{1}{2} \sum_{i=0}^{m} \kappa_\alpha \alpha_i^2 \). Defining \( D \) as

\[ D = \left\{ (e_T, \dot{\nu}, \tilde{\alpha}) | \ |e_T| \leq \frac{\varphi_1 \tilde{e} \tilde{\delta}_r}{a}, \ |\ddot{\nu}| \leq |\nu|, \ |\tilde{\alpha}_i| \leq |\alpha_i| \right\} \]

\( \dot{V} \) is negative semi-definite outside of \( D \). Therefore, \( e_T, \dot{\nu}, \) and \( \tilde{\alpha} \) are uniformly ultimately bounded. This completes the proof of the theorem.

### 5.5 Experimental Validation

In this section, the performance of the proposed control method is experimentally evaluated using a 2-DOF manipulator as our experimental platform. The 2-DOF manipulator (see Fig. 5.8) utilizes MR clutches as part of its actuation system. Two MR clutches configured antagonistically are used to actuate the first joint, while the second joint is actuated using a single MR clutch and a spring (for more details on the design of the manipulator see [2]). The specifications of the MR clutches used in this manipulator are given in Table 5.1. The manipulator is driven by a brushless motor (BLWRPG235D-36V-4000-R13) to provide the rotational inputs to the MR clutches. The motor is driven by a driver in the velocity
control mode. The manipulator incorporates two encoders (HEDS-9000) to measure the angular positions of the joints. Three high-power motor drivers (AMC-AZ12A8), set in current mode, provide the command currents to the MR clutches. In our experiments, both controllers were implemented on a desktop computer connected to the manipulator via a dSPACE (DS 1103) controller board. A force/torque sensor (ATI Gamma US-15-50) was employed for torque measurements. The sampling frequency for gathering experimental data was set to 1 kHz.

![Figure 5.8](image)

Figure 5.8: A snapshot of the MR clutch during fabrication and the 2-DOF MR-actuator robot manipulator.

### 5.5.1 Validation of Torque Estimation

The precision of the controlled system relies on the accuracy of Bingham model described in Section 5.3.2. This section presents the results of several experiments that were carried out...
Figure 5.9: Actual torque measurements versus estimations.

5.5. EXPERIMENTAL VALIDATION

out to assure Bingham model accuracy. Fig. 5.9 shows torque estimations along with actual torque measurements. The output torque of the MR clutch was controlled to track a sinusoidal trajectory with various frequencies ranging from 0.5 Hz to 5 Hz. The output torque was estimated based on Bingham model using magnetic field measurements. The Root Mean Square Errors (RMSEs) between the estimations and measurements are given in Table 5.2. As observed, it is clear that Bingham model provides an accurate estimation of the output torque.

Table 5.2: RMSEs of Torque Estimations

<table>
<thead>
<tr>
<th>Frequency</th>
<th>0.5 Hz</th>
<th>1 Hz</th>
<th>2 Hz</th>
<th>5 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMSE (Nm)</td>
<td>0.0858</td>
<td>0.0785</td>
<td>0.0689</td>
<td>0.1126</td>
</tr>
</tbody>
</table>
5.5.2 Torque Control Results

Having verified the accuracy of Bingham model, in this set of experiments, we evaluate the notion of torque control without the use of external force/torque sensors. The proposed method requires only the magnetic field measurements to control the output torque of the actuator via its input current. No force feedback is used in the construction of the proposed controller, and the torque measurements are only provided for the sake of comparison. To this end, we only used the second joint of the manipulator. To highlight the advantages of our proposed method, we compare the results to those obtained using a PID controller that regulated the output torque of the actuator using force feedback signals (see Fig. 5.10). The parameters of the PID controller were optimized in each specific experiment so as to obtain the best tracking results.

![Control block diagrams of the adaptive controller that uses magnetic field measurements and Bingham model and the PID controller that uses output torque measurements.](image)

Figure 5.10: Control block diagrams of the adaptive controller that uses magnetic field measurements and Bingham model and the PID controller that uses output torque measurements.

Fig. 5.11 compares the results of the proposed adaptive controller with those from the PID controller for 0.5 Hz, 1 Hz, 2 Hz and 5 Hz sinusoidal desired torques. The RMSEs for each controller are provided in Table 5.3. These results clearly demonstrate the ability of the adaptive controller to perform accurate torque control and its advantages to conventional techniques. Even though no torque feedbacks are used in the structure of the proposed controller, the performance of the controller competes with a PID controller that uses direct torque measurements. It should be pointed out that the performance of conventional force feedback-based controllers are generally limited because of inherent limitations of force/torque sensors including noise, latency in measurements, and instability in rigid
contacts. As such, most conventional methods for direct force control may fail to retain their performance in high frequencies. Alternatively, the proposed sensor less method can be used to achieve high fidelity force/torque control in robots that uses MR actuators.

Figure 5.11: Torque control results for a sinusoidal desired torque trajectory.

Table 5.3: RMSEs of Torque Control Results

<table>
<thead>
<tr>
<th>Frequency</th>
<th>0.5 Hz</th>
<th>1 Hz</th>
<th>2 Hz</th>
<th>5 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMSE (Nm)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Adaptive</td>
<td>0.1223</td>
<td>0.0945</td>
<td>0.1152</td>
<td>0.1853</td>
</tr>
<tr>
<td>PID</td>
<td>0.0941</td>
<td>0.3081</td>
<td>0.2742</td>
<td>0.2009</td>
</tr>
</tbody>
</table>

In order to further evaluate the ability of the proposed controller for tracking multi-frequency trajectories, a Multi-Sinusoid signal was used as the desired torque trajectory of the actuator. Fig. 5.12 shows the frequency spectrum of the desired trajectory, where

\[ T_d(t) = \sum_{k=1}^{n} \mu_k \cos(\omega_k t + \phi_k), \]

where \( \mu_k, \omega_k, \) and \( \phi_k \), for \( k = 1, \ldots, n \), are the amplitudes, frequencies, and phases of each sinusoid, respectively.
10 different frequencies between 0.75 to 7.5 Hz were used in generating the trajectory. In this signal, the amplitudes of all sinusoids, $\mu_k$, were selected equal, and the phases $\phi_k$ were spread based on Schroeder phase\textsuperscript{6} to have the maximum power for exciting all possible dynamics of the system. Fig. 5.13 depicts the tracking results for the Multi-Sinusoid desired torque. The results clearly demonstrate the fast adaptation and improved accuracy of the proposed control scheme. The results are compared with those obtained from the PID controller. The RMSE for the adaptive controller is 0.1569 Nm, while this value for the PID controller that uses direct torque measurements exceeds 0.1681 Nm.

5.5.3 Comparison Between Adaptive and PID Controllers

In this section, the abilities of our proposed controller and the PID controller for eliminating hysteresis are compared. In order to draw a fair comparison, the structure of the PID controller is given by:

\[ \phi_1 \text{ is arbitrary, and } \phi_k = \phi_1 - \pi (k - 1) / d, \quad 2 \leq k \leq d. \]
control loop was modified to match the adaptive control loop. In this way, both controllers use torque estimation based on Bingham model in the control loop (see Fig. 5.14). The parameters of the PID controller were optimized to obtain the best results. The output torque estimations for both controller are plotted in Fig. 5.15 against a sinusoid desired torque. As observed, it is clear that the PID controller cannot compensate for the hysteresis and its performance degrades as the frequency increased. This is where the proposed adaptive controller can effectively eliminate the hysteresis within all applied frequencies resulting in a linear relationship between the actuator input and output.

5.6 Conclusion

In this chapter, a new closed-loop control was proposed for actuators with hysteretic behavior. A new adaptive controller was designed to deal with the un-modeled hysteresis and uncertainties in MR actuators with insight into physical principles of the actuators. The hysteresis was approximated using a polynomial function, where adaption rules were provided to estimate the unknown parameters. The controller was constructed based on magnetic field measurements only, and no force feedback measurements were used. The stability of the closed-loop system was evaluated using Lyapunov method. Not only can the proposed technique provide high fidelity force/torque control, it also significantly reduces the manufacturing cost by eliminating the need for additional force/torque sensors in the control loop. This technique is also expected to alleviate the issues often arising when
Figure 5.15: Comparison of the adaptive and PID controllers in hysteresis compensation.
rigid force/torque sensors come into contact with other rigid environments. The proposed scheme was evaluated experimentally using a test bench and the sensor-less torque control results were compared to those obtained using direct torque measurements. The results clearly demonstrated the efficacy and advantages of the proposed technique. Further results on the use of the proposed controller for impedance control as well as hybrid force/position control will be provided as part of our ongoing research.
Bibliography


Chapter 6

Conclusion and Future Works

6.1 Conclusion

In this thesis, first, a new single motor, intrinsically safe 2–DOF manipulator was developed. A new actuation approach so called Pluralized Antagonistic Distributed Active-Semi Active (PA-DASA) actuation was incorporated for actuation of the manipulator. PA-DASA actuation utilises MR clutches as the semi-active elements in series with an active drive to leverage key properties of MR clutches including low impedance and backdrivability. As an added benefit, MR clutches offer decoupling the motor rotor inertia from the effective inertia of the link, accommodating intrinsic safety. Moreover, since the PA-DASA actuation locates a single motor at the base of the manipulator, it results to the reduction of the overall mass and inertia of the robot. As a result, the reduction in the mass/inertia will further improve safety characteristics of the manipulator. To provide a metric, mass and effective inertia of the manipulator were given along with safety analysis. Experimental results showed the feasibility and performance of the actuation approach. As the main outcome of this study, the feasibility of providing bi-directional actuation to all joints using a single unidirectional motor was validated. The manipulator developed in this study was also the first single motor driven manipulator that provides independent motion of the link without considering special path planning strategies. Furthermore, the results of position control experiments demonstrated the performance of the actuation concept in accurate tracking.

Second, the occurrence of limit cycles in the behavior of an antagonistically coupled MR actuator was studied. It was demonstrated that limit cycle can be induced by the antagonistic arrangement of MR actuators when operated in the position control loop. The dependency of the limit cycle on the controlled system was analyzed using describing function method. It was shown that the limit cycle oscillations could be avoided by choosing
specific gains for a PD controller. As an important aspect of the analysis, it was shown that high-bandwidth magnetic circuit was an essential element in preventing the limit cycle. In fact, eliminating limit cycle may not practically be achievable without a high-bandwidth magnetic circuit, dictating new constraints in the design of MR actuators. Numerical simulations along with experimental results validated the theoretical analysis.

Third, a novel model for predicting the behavior of MR actuators was developed. The model consisted of an adaptive model for modeling the magnetic field dynamics and the Bingham model to predict the output torque. As a key element, the adaptive model presented in this study utilised polynomial approximation for the first time to mimic the hysteretic dynamic of magnetic circuits. Unlike existing solutions, the new approach offered predicting the hysteretic behavior without the need for adjusting initial states. The stability analysis of the proposed adaptive model using Lyapunov’s direct method and the adaptation laws for minimizing the prediction errors were discussed. The accuracy of the modeling approach was verified using simulation and experimental results. The results were also compared to the well-known Preisach model, and its advantages were highlighted.

Forth, a new closed-loop control was constructed to compensate hysteresis in MR actuators. The control scheme presented in this study was developed based on physical principles of the actuators and utilised adaptive techniques to deal with the un-modeled hysteresis and uncertainties in MR actuators. The adaptive controller was in fact an extension to the modeling approach presented in this study, where hysteresis element was approximated by polynomial function. Adaption rules were provided to estimate the unknown parameters. The controller was constructed based on magnetic field measurements only, and no force feedback measurements were used. The stability of the tracking error dynamic was guaranteed using Lyapunov method. The proposed scheme was experimentally evaluated against a set of torque trajectory tracking experiments. These results were also compared with PID controller when PID controller used torque measurements. The results clearly demonstrated the efficacy and advantages of the proposed technique over the conventional method using force feedbacks. The results of this study also validated the feasibility of sensor-less force/torque control, offering significant reduction in the manufacturing cost.

6.2 Future Works

Reconsideration of active drives in the design of MR-actuators

MR actuators require an active drive to provide power to the actuator. As a common approach, all existing MR actuators use electric motors as the active drive. However, electric
motors can only provide very low torque, insufficient for most robotic applications. As a solution, gear reduction is necessary to attain high torques. However, using gear reduction introduces backlash in the operation of the system. As discussed in this study, antagonistic configuration was developed to address this drawback. However, the antagonistic configuration requires implementation of two MR clutches at joints, which can increase the overall weight of multi-DOF robots by itself. The PA-DASA configuration also adds to complexity of the design, and results in bulky structure for robots with higher than 2-DOF. In this respect, the author believes that the active drive in MR actuators must be substituted by other alternatives such as rotary hydraulic and air motors. Hydraulic actuators have the highest torque capacity and power density among other actuation mechanisms. They are capable of providing a few thousand Newton-meter torque and tens of kilowatts power output. Hydraulic actuators also feature very large torque-to-mass and power-to-mass ratios in comparison to electric motors\footnote{Force-to-weight ratio is typically as small as 16:1 in electric motors} For instance, Rotac 26R-5 hydraulic actuator can provide up to 2.79 kNm output torque at maximum power of 8.4 kW, where the actuator weighs 28.9 kg\footnote{\cite{2}}. Air motors also offer high force-to-mass ratio along with inherent compliance due to gas compressibility. Cleco MR70 rotary air motor offers up to 23 Nm with 12.5 kg mass without gearing. Hence, in the author’s opinion, employment of on-joint small hydraulic and air motors can be considered as an alternatives to PA-DASA concept. Due to the use of on-joint drive, the design complexity will be reduced significantly. The need for mounting two MR clutches at each joint will be eliminated. However this substitution may arise several challenges in control of robots. Without antagonistic actuation, the active drives require to revers direction, while reversing direction cannot be done instantly due to physical limitation. Hence, the performance of hydraulic and air motors should be considered in this regard. Moreover, in both hydraulic and air motors, significant friction presents due to sealing, that can limit the performance of the motors in reversing their direction.

\section*{Safety analysis and experiments}

Safety analysis for robotic applications are mainly based on research in car industry, and may not be valid in reality. Conditions in car crash experiments can be far different from situation where a robot is involved. For example, trapping a human between a robot and an obstacle is not comparable with car accidents. For trapping situation, the resulting injury can be, if not fatal, highly sever even at low velocity or power if the robot continues to follow its desired trajectory and no overgrowing stop is considered. In addition, many parameters such as using belt and/or airbags have been considered in safety analysis in car
industry that play no role in robot safety. For example, the maximum chest compression differs for different belt loading and blunt impact. The maximum chest compression for a 50% chance of serious injury is reported as 50 mm under belt loading \(^2\), while it is 61 mm in case of blunt loading \(^4\). Moreover, as discussed, impact velocity is an effective parameter in the severity of injury. Most impact-tests in the car industry have been done at velocities higher than 10 m/s (36 km/h) which is much higher than maximum achievable velocity in an industrial robot. Hence, most reported safety criteria in the car industry may not be valid for robot applications\(^3\). Therefore, not only careful attention should be paid when using existing severity indices, new safety analysis should be studied for robotic applications. Related to this subject, various severity indices\(^3\) were reviewed by Haddadin in \([6, 7]\). Crash tests using average male dummy in \([8–10]\) showed that industrial robots with velocity of up to 2 m/s pose no threat for humans in blunt impacts with respect to typical severity indices. However, in reality, serious injuries, including facial and cranial bones fractures may occur at this range of speed, confirmed by measurements of impact forces in \([9]\), and thus injury severity indices may fail in representing actual danger for robotic applications. Apart from single impact, another source of hazard in human-robot interaction can be repetitive impacts, which has not been considered in any severity indices. Borrowing results from boxing, an athlete in boxing can be exposed to repetitive impacts during a fight. While critical level for the Head Injury Criteria (HIC) is 700 for adults, the HIC level for a punch in boxing with maximum power was measured only as 119 \([11]\). Based on HIC, a single punch in boxing should not cause any kind of injury. However, boxing with no doubt includes high risk of brain, neck, eye, and head injuries \([11]\). Furthermore, safety analysis in robot applications were mostly conducted based on the human skull or bone fractures, while in the author’s opinion, any cause of pain should also be considered if the goal is to facilitate proximate human-friendly cooperation.

**Safety-oriented control**

To provide safety for general robots, a trivial solution, though inefficient, is to restrict the robot velocity. However, it can be argued that the amount of impact force exerted by robot depends on the robot configuration, and unsafe impact forces can only be reached in small subspaces of the workspace. Therefore, robots can operate faster in other configurations without exceeding safety limits. A similar approach was proposed based on limiting the robot’s kinetic energy \([12]\). This method also suffers from the same drawback of limiting

\(^2\)To provide an intuitive example, based on safety analysis in car industry, impacts with velocities lower than 3 m/s cannot cause any chest injuries \([3]\), which intuitively may not be valid for robots impact.

\(^3\)Refer to Appendix A for an overview on injury criteria.
the robot performance. Moreover, specifically in active and passive compliance methods, it is not clear how the compliance parameters should be adjusted during robot regular operation to provide safety and performance. Generally, regular control schemes (e.g. PID) would render the joints stiff for ordinary position tracking tasks, eliminating the benefits of the compliant actuation. Therefore, a new control paradigm is highly desirable to provide safety while maintaining performance. Embedding robot configuration and safety analysis in the design of controllers can be considered for future works. In the author’s opinion, the safety-oriented control can be achieved either by imposing necessary restrictions on a robot’s velocity in specific configurations or by a proper trajectory planning.
Bibliography


Appendix A

Injury Criteria

A.1 Safety Criteria

Several quantitative metrics are developed in industry in order to evaluate safety of mechanical systems in motion. Among them are the Gadd Severity Index (GSI) \[1\], the Head Injury Criterion (HIC) \[2\], the 3 ms criterion, the Viscous Criterion (VC), and Thoracic Trauma Index (TTI). The first three criteria are linked to human head injuries, while VC and TTI are chest safety indexes. Head injury-related criteria are mainly correlated with a tolerance curve established at Wayne State University, so called Wayne State University Tolerance Curve (WSUTC), which relates the head acceleration and impact duration to the severity of the brain injury. The curve is obtained experimentally from collision tests for animals and cadaver heads. It is shown that tolerable head acceleration is inversely correlated to impact duration, such that higher acceleration can be tolerated for a shorter period of time, and vice versa. Chest injury indexes, e.g. VC and TTI, are typically based on force and deflection measurements on the dummies in car crash tests.

A.1.1 3 ms Criterion

The 3 ms Criterion is based on the maximum allowable head acceleration. According to this criterion, the acceleration of the human head must be less than 72 g for any impact acts longer than 3 ms. Any impacts shorter than 3 ms will cause negligible, if not none, brain damage. This criterion is related to the human response to acceleration tests in the late 1950s that led to the 60 g standards for vehicles as the maximum spinal acceleration that human can withstand in 50 Km/h crash tests \[3\].
A.1. Safety Criteria

Table A.1: Definition of the abbreviated injury scale

<table>
<thead>
<tr>
<th>AIS</th>
<th>Security</th>
<th>Type of injury</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>None</td>
<td>None</td>
</tr>
<tr>
<td>1</td>
<td>Minor</td>
<td>Superficial Injury</td>
</tr>
<tr>
<td>2</td>
<td>Moderate</td>
<td>Recoverable</td>
</tr>
<tr>
<td>3</td>
<td>Serious</td>
<td>Possibly recoverable</td>
</tr>
<tr>
<td>4</td>
<td>Severe</td>
<td>Not fully recoverable without care</td>
</tr>
<tr>
<td>5</td>
<td>Critical</td>
<td>Not fully recoverable with care</td>
</tr>
<tr>
<td>6</td>
<td>Fatal</td>
<td>Unsurvivable</td>
</tr>
</tbody>
</table>

A.1.2 Head Injury Criterion

The HIC is the most popular and used standard index in the car industries as well as robotics to assess head injury in a head impact. The HIC is a weighted average of the head acceleration over the impact duration time, given by the following equation,

$$\text{HIC}_{\Delta T} = \Delta T \left[ \frac{1}{g \Delta T} \int_{t_1}^{t_2} \dddot{x}_h \, dr \right]^{2.5},$$  \hspace{1cm} (A.1)

where $\Delta T = t_2 - t_1$ is the impact duration time, $g$ is the gravity acceleration, and $\dddot{x}_h$ is the human head acceleration in an impact.

An impact with HIC$_{35}$ value of equal or greater than 1000 can result in a sever head injury. The level of injury is typically based on a qualitative scale so called the Abbreviated Injury Scale (AIS) \[4]. AIS is developed by the Association for the Advancement of Automotive Medicine and the American Medical Association, that scales an injury into seven levels of severity from "none" to "fatal" (see Table A.1).

A.1.3 Viscous Criterion

Rib fracture as well as internal injuries of the thorax are two possible dangers in an impact. Internal injuries can be however of a greater threat than skeletal injuries for human. For instance, a patient can die within an hour due to rupture or transection of the thoracic aorta caused by an impact \[5]. It is, however, realized through cadaver experiments that the acceleration based criteria cannot be a precise measure for the chest injuries. The chest injury is typically a function of chest deflection and impact force. Experimental studies show that the chest deflection should be less than 22 mm to avoid rib fractures.
Table A.2: Acting force and torque limits specified for the human neck

<table>
<thead>
<tr>
<th>Load</th>
<th>@0 ms</th>
<th>@25-35 ms</th>
<th>@45 ms</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shearing: $F_x, F_y$</td>
<td>1.9/3.1 kN</td>
<td>1.2/1.5 kN</td>
<td>1.1/1.1 kN</td>
</tr>
<tr>
<td>Tension: $F_z$</td>
<td>2.7/3.3 kN</td>
<td>2.3/2.9 kN</td>
<td>1.1/1.1 kN</td>
</tr>
<tr>
<td>Extension: $M_y$</td>
<td>42/57 Nm</td>
<td>42/57 Nm</td>
<td>42/57 Nm</td>
</tr>
</tbody>
</table>

[6]. However, in case of injuries of internal organs, the rate of intrusion can be a better descriptor for a potential injury. Accordingly, the Viscous Criterion (VC) [7] uses both the thoracic deflection and its velocity to characterize the thoracic injury likelihood, formulated as follow,

$$VC = c_c \frac{\|\Delta \dot{x}_C\|_2}{l_c},$$

where $\Delta x_C$ is the chest deflection, $c_c$ is the scaling factor, and $l_c$ is the deformation constant. Typically, a chest viscous tolerance of $VC_{\text{max}} = 0.5 m/s$ is recommended to minimize the likelihood of severe injury in a collision.

### A.1.4 Injury Criteria for the Neck

The neck injuries in human are typically caused by forces and bending torques acting on the spinal column. Accordingly, EuroNCAP [8] has provided the acting force and torque limits given in Table A.2 which above will cause significant risk of injury. The limits are based on positive cumulative exceedance diagram, and are functions of time. A linear interpolation can be used to calculate the limiting forces/torques at any time instant. The corresponding forces/torque are illustrated in Fig. A.1.
Figure A.1: Taxonomy of neck motions.
Bibliography


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