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Design of Magneto- Rheological Damper for Suspension System

A Thesis

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By

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بسم الله الرحمن الرحيم

وَوَهَبْنَا لَهُمْ مِنْ رَحْمَتِنَا وَجَعَلْنَا لَهُمْ لِسَانَ صِدْقٍ عَلِيًّا

صدق الله العلي العظيم

سورة مريم آية ﴿50﴾

Dedication

To those who Gave Me Support, Inspiration, Courage, and Strength, Especially My Father and My Mother

Mahdi Qahrman Fakhrulddin

2022

Acknowledgments

First and foremost, my limitless thanks to (**Allah)** for his help and blessing. The most kind for giving me the power and the courage, as well as patience to accomplish this work.

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Abstract

Suspension has been seen as the vehicle's most significant component. The suspension architecture affects driving comfort and car handling efficiency. Over recent decades, automobile technology has continually introduced advances to give end users greater driving comfort. This study uses pattern search optimization techniques to achieve multi-target optimization of MR damper with objective damping force maximization by MR Damper with geometric parametric limit function.

MR dampers are semi-active systems that gain higher-level, controllable strengths and efficiency, while maintaining passive system functionality and low energy demands. However, a semi-active component of an MR Damper's highly non-linear behavior increases the complexity of the numerical modeling behavior, notably when using parametrical models.

Research focuses on semi-active suspension design, modeling, and simulation with regard to the hysteresis behavior of one-quart car model using non-linear theory of the Magneto-Rheological (MR) damper. The research is based on the assumption that the same disturbance is caused by each wheel. Bingham models are used for analysis of hysteresis. Research involves simulation of Bingham and passive models. For the three road profiles the modelled system is analyzed and the model comparison study for maximum comfort with a reduced shot and a remediation time of the vehicle's jump weight is performed. In terms of damping power requirements, the

Bingham model is more comfortable that the passive suspension and has less overhead and less time for settling.

A Magneto-Rheological fluid damper is used in the damper system of the suspension system. In order to simplify, the suspension system is described by a quarter-car model. The complex model of the observed form of a four-car is used in Newton's second motion theorem. The Bingham Model for Magnet-rheological Liquid Dampers expresses a complex hysteresis power in this article. Several simulations for the suspension system have been performed in an automotive application; different excitations (square, bump, and random) have been performed as road profiles to demonstrate the effect of the force generated in the damper for each excitation signal. Also in this study use PID controller with the system and study its effect. The simulation results showed the efficacy of the quarter car model derived, including the MR fluid damper.

Nomenclature

Symbol	Definition
MR	Magneto-Rheological
ER	Electro- Rheological
MRF	Magneto-Rheological Fluid
MRD	Magneto-Rheological Damper
M	Mass (kg)
Z	Displacement (m)
$\dot{z}$	Velocity (m/s)
$\ddot{z}$	Acceleration (m/s^2)
k	Stiffness (N/m)
c	Coefficients of damping (N s/m)
r	Displacement in longitudinal direction(m)
$\dot{r}$	Velocity in longitudinal direction(m/s)
U	Force generated (N)
D	Form Factor
PID	Proportional-Integral-Derivative
K	Gain
τ	Time period
Subscript	
C	Controlled
O	Offset

S	Sprung mass
P	Proportional
I	Integral
D	Derivative
U	Un-sprung mass

Contents

Chapter One

Introduction

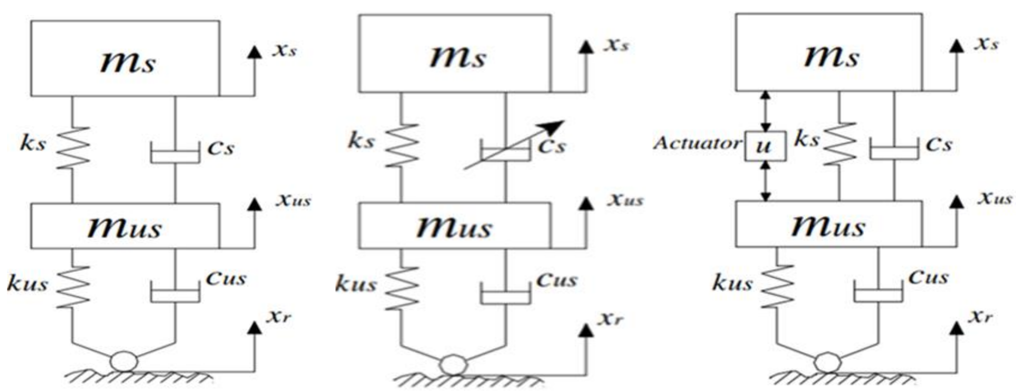
1.1 Background

Automotive suspension systems have a great importance in making ground automobile more reliable, comfortable and with less wear due to road roughness. The Suspension is a collective term given to the system of springs, damper and linkages that isolates a vehicle body from the wheel assembly [1].

The fundamental reasons for disturbances of the passengers are the oscillations that influence the passengers from vibrations. These vibrations caused by the roadway excitation in each element of the chassis. Therefore, the suspension system plays an important role to isolate the passenger from the noise of road as much as possible to provide comfort for the passenger and in order to prevent the transmission and penetration of disturbances into the passenger's compartment, while, keeping contact of tire with road to improve the road handling and movement [2].

1.2 Types of Suspension System

Suspension systems can be classified, based on their function, into three categories, passive, semi-active, and active. Passive traditional suspensions system contain only passive elements, spring in parallel with a damper, and have good characteristics in achieving passenger comfort and automobile control, i.e., ride, but not for all road and loading situations (fig1.1 a, show the passive suspension system digram). Semi-active suspension systems have shown better performance since they have the ability to vary their damping value according to control scheme through using Electro-Rheological or Magneto-Rheological dampers (fig1.1 b show the semi-active suspension system digram). These suspensions need some amount of power to be fed to it. In general, increasing damper stiffness of a suspension gives better handling but with less passenger comfort and vice versa. Usually, suspension design should compromise between this conflicting trade-off and also is related to cost [3]. Active suspension system have additional hydraulic actuators to the passive elements of traditional ones (fig 1.1 c show the active suspension system digram). These actuators give the suspension the ability to generate needed forces to improve performance characteristics of automobile ride for all driving situations. Active suspensions, in addition to their cost, require additional power to be injected to them and this is an important issue to be justified when selecting an active suspension for some automobile and power consumption of suspension must be taken into consideration [4].



(a) (b) (c)

Fig 1.1 schematics digram for: a- passive, b- semi-active, c- active suspension system [5]

1.3 Magnetorheological (MR) Fluids Damper

Controllable fluid dampers generally utilise either electrorheological (ER) fluids or MR fluids, whose viscosity properties can be altered dramatically by applying an electric field (ER) or a magnetic field (MR). These fluids were first discovered by 27 the inventor Willis Winslow, who achieved a US patent regarding these fluids in 1947 [15] and published a scientific article in 1949 [16]. The yield stress of these fluids can be controlled very precisely by changing the field intensity to generate a continuously variable damping force. An MR fluid consists of micro-sized magnetically polarisable particles, such as iron particles, suspended in a carrier liquid such as mineral oil, synthetic oil, water or glycol. A typical MR fluid will contain 20 to 40 percent by volume of relatively pure iron particles around 3 to 10 micron diameter in size. These fluids have found several successful applications in field of vibration control. This thesis is focussed on MR fluid dampers.

MR dampers typically consist of a piston, magnetic coils, accumulator, bearing, seal, and damper reservoir filled with MR fluid [8]. These types of dampers have several remarkable features such as stability, reliability, low power requirements. By applying a magnetic field, it is possible to form chains with magnetic particles in the fluid see Fig 1.2. It reacts in less than a few milliseconds related to the viscosity of a magnetorheological fluid [9].

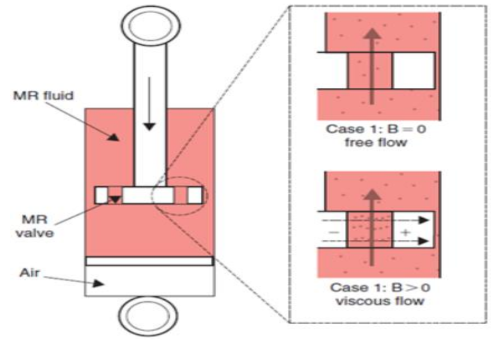


Figure 1.2. Schematic representation of a magnetorheological damper behavior[10]

MR damper have a disadvantage such as sedimentation, low wear resist for working part as compared to conventional damper, and more expensive than conventional damper. MR damper is unemployed commercially in-vehicle shock absorbers, landing gear, vibration isolation systems, helicopter lag dampers, vehicle seat suspension systems, mounts, bearings, and military equipment, and brake, fig 1.3 shwoed some of MR damper, for more application see [11-14].

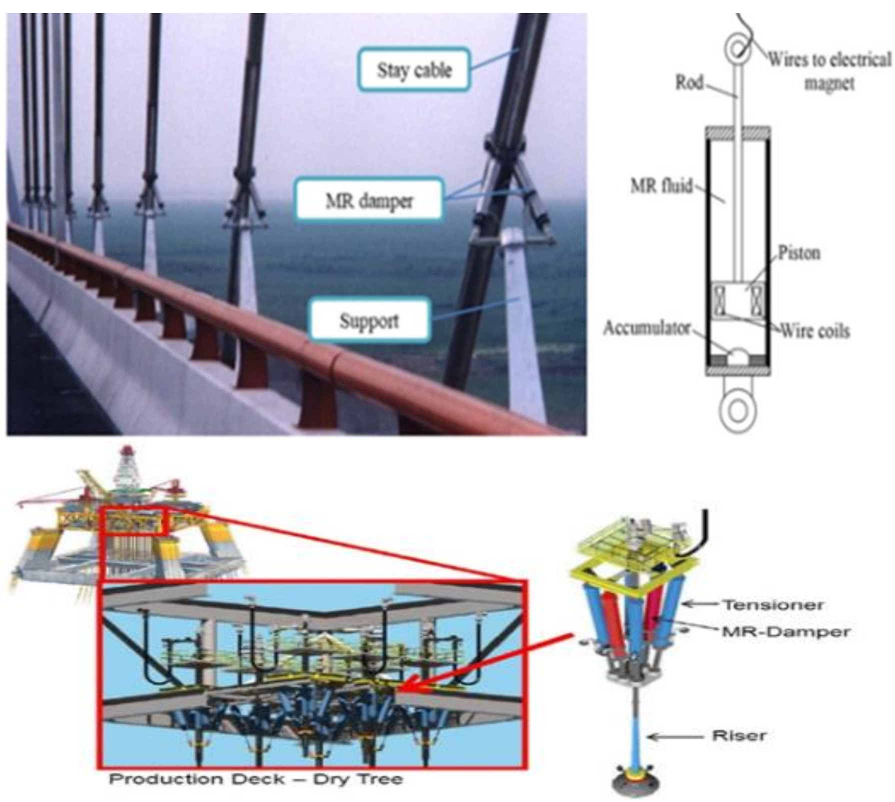


Figure 1.3 Some application figure

1.4 Modeling of MR Fluid Damper

The MR damper is a nonlinear and hysteretic system. By “hysteretic” is meant that the output is dependent not just on the instantaneous values of the inputs, but also on the history of the output [15]. There are several type of model including, Bingham, Dahl, LuGre and Bouc-Wen models [16]. In general, all of them have described the developed damping force. In this study focuses on the Bingham model.

Bingham model model is the simple to modeling and most effective hysteresis model used for MR dampers [17]. This model assumes that a body acts as a solid until minimum yield stress is reached, at which point it exhibits a linear relationship between stress and shear or deformation intensity [18]. The Bingham model considers Coulomb (dry) friction, plain (constant) damping, stiffness, and a constant offset force, as well as how the control force direction changes concerning the sprung mass velocity value. Fig 1.4 show Bingham model digram.

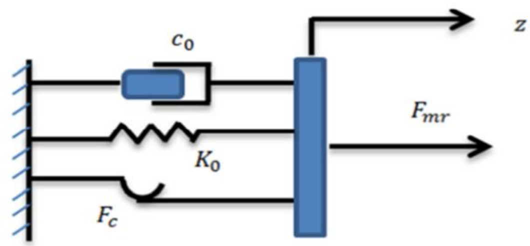


Figure 1.4 Mechanical model by Bingham [19]

1.5 Research Objectives

This study aims to design a control system for a semi-active suspension system based on a quarter car model with an MR damper. Design and modeling a PID controller for a quarter car model semi-active suspension system with MR damper based on Bingham model using MATLAB/Simulink have been achieved and analyzed with applied different road profiles including (step, bump, and random).

1.6 Scopes of Research: The scopes of this project are:

a. present the mathematical modeling of a suspension system of a quarter car based on Bingham-plastic model for damping force presentation using MATLAB/Simulink.

b. Design and simulate a PID control for a semi-active suspension system with MR damper.

c. An analysis of the controller's performance for semi-active suspension system using MR damper in comparison with a passive suspension system.

Chapter Two

Literature Review

2.1 Introduction

Vehicle stability and ride comfort are the important parameters for any suspension system behavior under road roughness. Overshoot and settling time are the dynamic characteristics of a semi-active suspension system. In this study, design a PID controller to obtain batter response and to reduce the overshoot and settling time. The study focuses on semi-active suspension with MR damper. The previous work on MR damper for semi-active suspension system will be represented and discussed in this chapter.

2.2 Uncontroller Case

(Eshkabilov) (2016)[16] used different models including Bingham, Bouc-Wen, Dahl, and LuGre to model MR dampers in the semi-active suspension system. The behavior investigated under several road inputs, using MATLAB Simulink to determine the result and compared them. The result showed that the Bingham model gives better performance as compared with other models to reduce overshoot and settling time.

(vaishnav et al,) [20] modeled and simulated of two degree of freedom quarter car model. Compared between passive and active suspension system in terms of riding quality and vehicle handling. The mathematical modelling was developed and the differential equations of motion were derived from it and solved it using MATLAB software. An Active suspension provides better ride quality and better load distribution on each axle of the car. The results included the displacement and acceleration of the sprung and unsprung mass after hitting a bump. It was also includes the stability analysis of the system.

(Solovyov et al.) (2017) [21] Studed the dynamics of an external mechanical device using a damping mechanism, taking the hysteretic dimension of the damping mechanism into account. used the Bingham model as the hysteric damper's mathematical basis. The simulation takes the form of a force transfer function and displays the hysterical damper's "efficiency" in relation to the non-linear viscous damper. The findings include a distinction between the various types of viscous (linear and nonlinear) and hysterical dampers. Based on empirical studies.

(Razali et al.) (2018) [22] presented a compassion study for five models of the MRD, Hyperbolic Tangent Function, Simple Bouc-Wen, Bingham, Modified Bouc-Wen, and Nonlinear Biviscous model. The nonlinear least square method was used for the fitting process to obtain the constants in each model. The resulted parameters were tested and fitted for several input current values. The nonlinear behavior observed for all of the studied parameters and models. They concluded that the Bingham model was the best model that can be used for describing the MRD behavior, because it presented the lowest value of error.

(Sharma et al,) (2018) [23] analyzed the MR damper in semi-active suspension system for a passenger vehicle and compared it with passive suspension system. Bingham model was use for modeling MR damper. Quarter car model with two degree of freedom has been use for modeling semi-active suspension system and simulated with MATLAB/Simulink. The results showed that the semi-active suspension system performed better than the passive suspension system in terms of vehicle stability.

(Palomares et al.) (2019) [24] Numerous models of magnetorheological dampers, including classification, simulation and contrast, have been reviewed. The efficiency of the various models has been tested by multiple experimental experiments and simulations in a basic and straightforward semi-active control case study. The findings obtained indicated that most models typically fail to forecast correct low-speed behavior and, as a result, can lead to inaccurate estimates when used in control schemes due to modelling and charting errors.

(Yang et al.) (2020) [25] The stiffness system used in the semi-active suspension using a magneto-rheological (MR) damper offered a way to increase the vibration-reduction performance of the semi-active suspensions to the degree attainable by active suspensions. The experimental results revealed that the new suspension provided better vibration-reduction efficiency than the passive suspension and the traditional MR suspension and that the vibration-reduction efficiency is comparable to that indicated by some of the active systems. The new MR suspension can be viewed as a new type of semi-active suspension capable of achieving comparable performance with active suspension but with a much lower cost, lower power consumption and a robust feature. This concept is intended to greatly boost the semi-active vibration control technology.

(Jamadar et al.) (2020) [26] Studied the hysteresis loops of the modified Bouc-wen model output model controlling the internal dynamics of the Magneto-Rheological (MR) damper. The measurement of the energy dissipated by the vibration process is carried out by means of an MR damping dynamic characterization on a dynamic test system in order to analyze the influence of the frequency, the amplitude of the displacement and the excitation current on the dissipated energy and the corresponding damping coefficient. Complex reactions of the modified Bouc-wen numerical model will be correlated with experimental responses and then recalibrated with the new Bouc-wen numeriscal model, thus changing the parameters of control.

(Yahaya et al,) (2021) [27] studied the passive suspension system and compared between mathematical modelling and experimental work for the ride performance of a passive quarter car model suspension system was showed and discussed. The mathematical modelling of double passive quarter car model suspension system was formulated and solved numerically using the second-order linear differential equation by used the MATALB Software. The vertical displacement was produced from the mathematical modelling was at 0.015 m identified as a maximum point while the minimum point was at-0.015 m. The vertical displacements from the experimental work were at 0.02 m and-0.03 m at the minimum point.

(Barethiye et al,) [28] studied the shock absorber model with passive suspension system and compared it with piecewise liner model. Half car suspension system used to investigate the influence of damping forces on the response characteristics of a vehicle. MATLAB/Simulink used to simulate the model with applied bump road profile. The simulation results showed that the performance characteristics of vehicle suspension using hybrid shock absorber model improves the road comfort of the passengers.

2.3 Controller Case

(Phalke et, al.) (2016) [29] used a PID controller with a semi-active suspension system to improve the ride comfortable . A quarter car mathematical model with MATLAB Simulink to determent the result, and compared between PID semi-active suspension system with a passive suspension system. The result showed that the PID semi-active give batter performance with different road input as, half-sine bump 51%, double sine bump 40%, trapezoidal bump 42.3%, step bump 34.9%, and square bump 48.8% as compared with the passive case.

(Pepe) (2016)[30] used vibrational feedback controller with MR damper in the semi-active suspension system. This study used a quarter car mathematical model and, simulate it by using MATLAB/Simulink with applied different road profile. These controllers are compared with other types such as skyhook and, ground hook controllers. The simulation result showed that the VFC controller has batter performance as compared with the sky hood and, ground hook controller.

(Amiruddin et, al) (2017) [31] In this study PID controller was studied with MR damper in the semi-active suspension system. Two degrees of freedom quarter car model system simulated using MATLAB/Simulink. Special model of MR damper design to simulate Patron Preve car specification. The simulation result showed that the PID controller enhances raid comfortability and, car stability by 60% as compared with the passive case under applied step profile as road exaction.

(Huang et, al) (2018)[32] studied effect of a PID controller with a semi-active suspension system. Design a quarter car model with two degrees of freedom. The system is simulated with MATLAB/Simulink and, an experimental test. Use the Genetic algorithm to tune the PID controller. The result shows that the PID controller improves the riding comfortability by (26.342%), car stability by (9.964%), and overall performance by (27.62%) as compared with the passive case.

(Saad et, al) (2018)[33] uses a PID controller with a semi-active suspension system with an MR damper to enhance it. PID controller was used with Bouc-Wen model for MR damper and, quarter car model. the results are simulated with using of MATLAB Simulink also applied different road profile such as sinusoidal, and random input. The simulation result showed enhance in the performance of the MR damper and an increase in car handling.

(Jamil et, al) (2018) [34] studied MR damper for a semi-active suspension system with a quarter car model to improve the response, vehicle stability, and ride comfort. Design PID controller for the semi-active suspension system. The result simulate the by using MATLAB/Simulink and it showed that the PID controller showed batter performance with lower overshoot and faster settling time as compared with a passive suspension system.

(Soliman) (2019) [35] studied the semi-active suspension system instead of the active suspension system required less power and chipper and he studied the semi-active suspension system to improve it. For this reason, he used many types of controller to enhance the performance of semi-active suspension systems such as skyhook, ground-hook, optimal controller, and fuzzy logic. The result was simulated by MATLAB Simulink, and the result showed that using the controller with the semi-active suspension gives high enhancement on performance in case of car stability and ride comfort.

(Kumar E et, al) (2019) [36] used different forms of PID controller such as PID, PI-D, and I-PD controller with a semi-active suspension system. This study used a quarter car mathematical model with MATLAB/Simulink. Ziegler-Nichols method use to tune PID controllers form. Applied step input as road profile. The simulation result showed that the I-PD has batter performance in reducing overshoot and, settling time as compared with another form.

(Shehata et, al) (2019) [37] used two different control techniques including Fuzzy logic, and a Fuzzy self-tuning PID controller with four degrees of freedom semi-active suspension system. Applied bump and random road input. Simulate the system in MATLAB/

Simulink, the study showed that the Fuzzy self-tuning PID controller gives higher performance compared with the Fuzzy logic controller.

(Bashir et, al) (2019) [38] used a hybrid fuzzy and, fuzzy-PID (HFFPID) controller with an MR damper in a semi-active suspension system to reduce overshoot and, road comfortably. Quarter car mathematical model was used with six-degree of freedom to simulate it with MATLAB/Simulink. Applied different road profiles (bump and, random) and compare them with different controller strategies. The simulation result showed that (HFFPID) has better performance in improving roads comfortably and, reducing overshoot.

(Yoon et, al) (2020)[39] studied the enhancement of MR damper for full car semi-active suspension system time response. For that using robust sliding controller to controller MR damper, and the parameter of this controller was found by using a different method including Bode plot and Nyquist analysis, to be compared the controller result with a passive semi-active suspension system. The result showed that using of controller showed a faster time response of MR damper with an improved in performance.

(Jibril) (2020)[40] used an H∞ controller to compare active and, semi-active suspension systems. For that a quarter car model was used and, simulated with MATLAB/Simulink under a sinewave road profile. The simulation result showed that the active suspension system is better than that of the semi-active suspension system.

(Negash et, al) (2021) [41] used skyhook and skyhook –acceleration damper with MR damper in the semi-active suspension system. A quarter car model was built with MATLAB/Simulink. The simulation result showed that the skyhook controller and, skyhook –acceleration damper controller improve the response by 58.49% with skyhook –acceleration damper and, 54.94% with skyhook.

(Walavalker et, al) (2021) [42] investigated different controller methods including PI, PID, internal model control, , internal model control with filter, FUZZY, and adaptive network-based fuzzy inference system. The controllers used the semi-active suspension system to improve raid comfortability and, car stability. A quarter car mathematical model was used and simulated with MATLAB/Simulink under road exaction (bump and, sin wave). The simulation result from different controller methods and, passive cases in terms of (settling time and overshoot) respectively. Passive (8.564, 69.806), PI (3.1248, 65.7), PID (2.084, 50.3148), IMC-PID (2.084, 60.6402), internal model control -PID with filter (1.840, 72.026), FUZZY(1.642, 34.5382) and, adaptive network-based fuzzy inference system (1.4473, 456255). The result showed that FUZZY has better performance as compared with another method.

(Ma et, al) (2021)[43] used a fuzzy skyhook controller with an MR damper for a semi-active suspension system and, tuning using Grey Wolf optimizer. A Neuro-Invers model was used for the MR damper and simulated with MATLAB/Simulink. The simulation result showed that the skyhook- Grey Wolf optimizer improves the vertical acceleration by 22.65%, deflection by 15.46%, and, dynamic load by 7.75% as compared with the passive case.

(Ab Talib et, al) (2022)[44] studied the Skyhook-Differential equation (DE) controller with MR damper for the semi-active suspension system. A modified Boc-Wen model was used with quarter car mathematical model. This system was simulated using MAT5LAB/Simulink and experimental test. The result showed that the skyhook-DE controller enhances the performance of semi-active suspension by 63.04% as compared with another control method.

Chapter Three

Mathematical Analysis

3.1 Introduction

In this chapter the design and, the analysis of car suspension system with MR damper models exposed to input displacement. It uses to determine the time response for the semi-active and passive suspension system and compared between them by a set of governing differential equations of both cases for the quarter car mathematical model. Also this chapter presents a design of a PID controller for a semi-active suspension system.

3.2 Quarter Car Suspension System Modeling

To find out the dynamic characteristics for different road exactions because it's simple and easy to model. Fig (1.1 a, and b) show the quarter car diagram for the passive and semi-active suspension system. The equation of motion for both passive and semi-active suspension systems was derived by using Newton's law of motion to describe the relative motion between sprung and unsprung mass as the follow [45]:

a- passive suspension system:

$$m_s\ddot{z}_s + c_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) = 0 \tag{3.1}$$

$$m_u\ddot{z}_u + c_s(\dot{z}_u - \dot{z}_s) + k_s(z_u - z_s) + c_u\dot{z}_u + k_u z_u = k_u r + c_u\dot{r} \tag{3.2}$$

b- semi-active suspension system:

$$m_s\ddot{z}_s + c_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) = -U_c \tag{3.3}$$

$$m_u\ddot{z}_u + c_s(\dot{z}_u - \dot{z}_s) + k_s(z_u - z_s) + c_u\dot{z}_u + k_u z_u = U_c + k_u r + c_u\dot{r} \tag{3.4}$$

Where, m_s and m_u are the sprung and unsprung mass, c_s damping coefficient of sprung mass, c_u damping coefficient for unsprung mass. In this study $c_u = 0$ in this study since the tire have internal stiffness so neglecting its value for simplification, k_s and k_u are stiffnesses of sprung and unsprung masses, r and $\dot{r}$ is displacement and velocity in the longitudinal direction due to roughness of terrain (disturbance), z_s and z_u are the displacements of sprung and unsprung mass respectively, $\dot{z}_s$ and $\dot{z}_u$ are the speeds of sprung and unsprung mass, $\ddot{z}_s$ and $\ddot{z}_u$ are the acceleration of sprung and unsprung mass respectively, and U_c is the force generated as a function of time that depends on the speed of the vehicle. The following assumptions have been assumed to study the oscillations of the automobile:

1- The velocity of automobile is constant rectilinear motion.

2- The contact between the wheels and ground is continuous

3- Initial values of z_s, z_u, $\dot{z}_s$, and $\dot{z}_u$ are zero.

Assume state variables as:

$x_1(t) = z_s(t) - z_u(t)$

$x_2(t) = z_u(t) - r$

$x_3(t) = \dot{z}_s(t)$

$x_4(t) = \dot{z}_u(t)$

Where:

$(z_s - z_u)$ is the suspension relative displacement, $(z_u - r)$ is the wheel deflection, $\dot{z}_s$ is body velocity and, $\dot{z}_u$ wheel velocity.

Now sub-state variables in equation 3.3 and, 3.4

$$m_s \dot{x}_3(t) + c_s(x_3(t) - x_4(t)) + k_s(x_1(t)) = -U_c(t) \tag{3.5}$$

$$m_u \dot{x}_4(\mathrm{t}) + c_s\big(x_4(t) - x_3(t)\big) - k_s(x_1(t)) + k_u(x_2(t) = U_c(t) \tag{3.6}$$

Therefore:

$$\dot{x}_1(t) = x_3(t) - x_4(t)$$

$$\dot{x}_2(t) = x_4(t) - \dot{r}$$

$$\dot{x}_3(t) = -\frac{k_s * x_1(t)}{M_s} - c_s * \frac{x_3(t)}{M_s} + c_s * \frac{x_4(t)}{M_s} - \frac{U_c(t)}{M_s}$$

$$\dot{x}_4(t) = -\frac{k_s * x_1(t)}{M_u} - k_t * \frac{x_2(t)}{M_u} c_s * \frac{x_3(t)}{M_u} - c_s * \frac{x_4(t)}{M_u} + \frac{U_c(t)}{M_u}$$

Now written state space equation as:

$$\dot{x}(t) = Ax(t) + BU_c(t) \tag{3.7}$$

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & -1 \\ 0 & 0 & 0 & 1 \\ -k_s/M_s & 0 & -c_s/M_s & c_s/M_s \\ k_s/M_u & -k_t/M_u & c_s/M_u & -c_s/M_u \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ -1/M_s \\ 1/M_u \end{bmatrix} U_c \tag{3.8}$$

Where:

$$\mathrm{A} = \begin{bmatrix} 0 & 0 & 1 & -1 \\ 0 & 0 & 0 & 1 \\ -k_s/M_s & 0 & -c_s/M_s & c_s/M_s \\ k_s/M_u & -k_t/M_u & c_s/M_u & -c_s/M_u \end{bmatrix}, \mathrm{B} = \begin{bmatrix} 0 \\ 0 \\ -1/M_s \\ 1/M_u \end{bmatrix}$$

The numerical values of all parameters mentioned in eq.3.1 through eq. 3.4 are presented in Table 3.1

Table (3- 1) Numerical values of quarter-car model [45]

parameter	**Value**
Spurung mass (m_s)	450 kg
Unspurung mass (m_u)	68 kg
Spurung mass stiffness (k_s)	28500 [N/m]
Unspurung mass stiffness (k_u)	293.9 [KN/m]
Damping coefficient of spurung mass (c_s)	2700 [N.s/m]
Damping coefficient of unspurung mass (c_u)	0

The quarter car mathematical have some disadvantages like it doesn't show the effect of full car geometry. The quarter car model assumes that the wheel is always in contact with the road, so it is accurate at the low range of frequency, but not in the high-frequency range [46].

3.3 Bingham Model

The Bingham model is the simplest and most effective hysteresis model used for MR dampers. The following equation shows the mathematical modeling of the Bingham model [45]:

$$F_{mr} = F_c sgn(\dot{z}) + c_0\dot{z} + k_0 z + F_0 \tag{3.9}$$

The alternative form of the Bingham model [47]

$$F_{mr} = \frac{2F_c \tan^{-1}(d.\dot{z})}{\pi} + c_0\dot{z} + k_0 z + F_0 \tag{3.10}$$

Where z is a displacement of the MR damper, F_c is the controlled force, F_o is the constant force of the MR damper, c_o is the damping coefficients of the MR damper, k_o is the stiffness of the MR damper and d is the form factor, The form factor enhances the accuracy of the hysteresis loops of the equation and sgn function represented to reduce direction and motion for the controlled frictional force with relative velocity $\dot{z}$[48]. Bingham plastic model diagram is shown in figure 3.1 as an approximate introduction of the hysteresis loop expressed by eq (3.5), and a working form enhances the rounding of hysteresis loops of equation (3.5) to a large extent. The ratio of change in force to change in velocity is called the damping constant $\frac{\Delta F}{\Delta \dot{Z}}$. The form factor enhances the accuracy of the hysteresis loops of equation (3.5) considerably. Bingham model parameters are F_o offset force is assumed to equal 0 N , for d takes the following value for each current value 20 and c_o, F_c is both depends on the input current to MR damper. The damping force depended on the previous parameter in the Bingham model, to control it we need to control these parameters. Their values are given as a function of the current as in the following equation[48].

$$f_c(I) = -910.09I^3 + 986.49I^2 + 663.56I + 52.19$$
$$c_0(I) = 48.74I^4 - 106.39I^3 + 66.00I^2 + 1.43I + 0.53 \qquad (3.11)$$

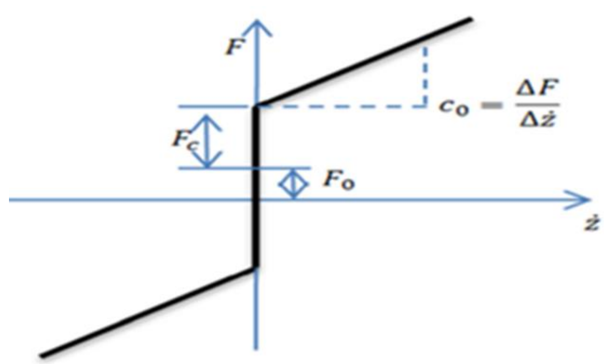


Fig 3.1 Bingham plastic model proposed[19]

3.4 Road profile

The road is the most intensive source of excitation of the vehicle. The vehicle vertical dynamic behavior (oscillations) depends on a series of factors, most of them related to the road: length, height, shape, irregularities frequency, etc. Every road has a profile of irregularities (small up and downs), which can be periodic or random (stochastic). Most of the real roads have a random profile or irregularities. Therefore, to study the suspension system behavior accurately research use three different road signals including Step signal, bump signal and random profile input.

3.4.1 Square profile input

Square input profile is the simplest road profile which is a sudden change in amplitude for short time. This study uses 10 cm step profile as shown in figure 3.2.

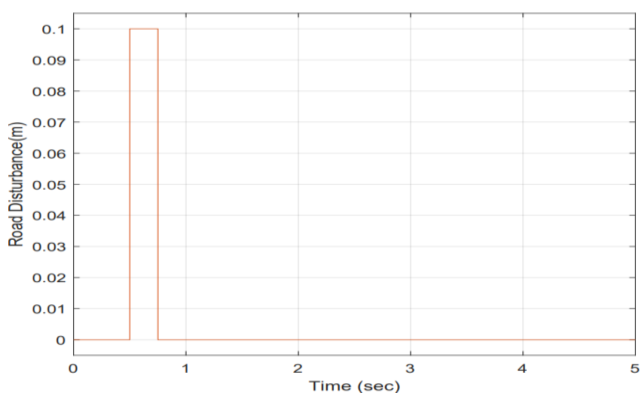


figure 3.2. Square road profile.

3.4.2 Bump Profile

The bump profile similar to the case of that car passing through a bump. In this study with 10cm amplitude, and 79 radian per second frequency as shown in figure 3.3.

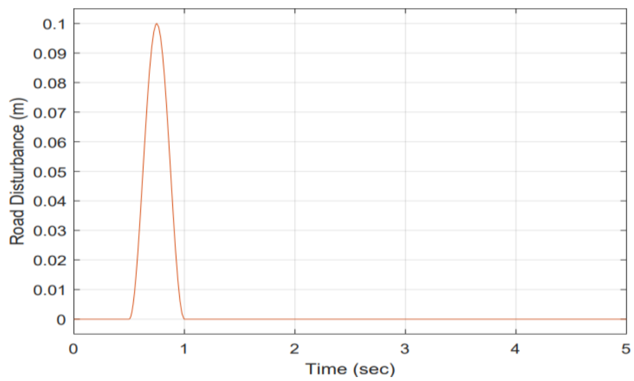
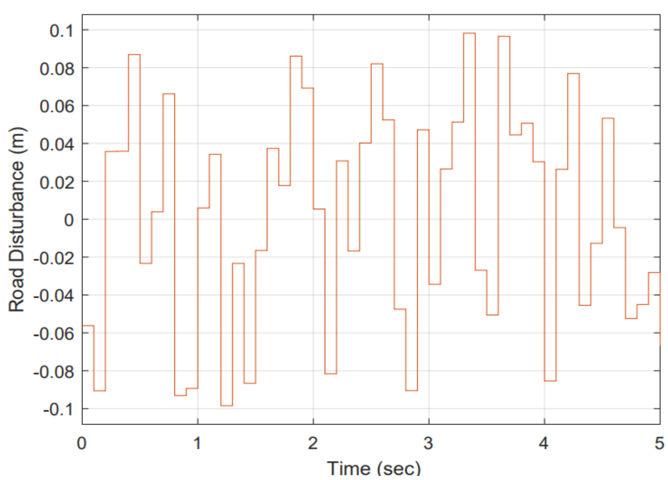


Figure 3.3. bump road profile.

3.4.3 Random Road Profile

Random road profile is the actual road profile and, cannot be described mathematically. Therefore, Gaussian distribution was used to describe the random road profile. Figure 3.4. shows -1cm to 1 cm amplitude.



Fiquer3.4. random road profile [45]

3.5 Controller

The controller decides the amount of MR damper force that is needed to achieve the system best state. Feedback management has been drastically improved, It affects the system natural frequencies, as well as its dynamics, both in terms of transient response and stability [48].

3.6 PID Controller

Proportional–integral–derivative (PID) controllers are a wide range used in engineering applications. Because of their simplicity, good control efficiency, and excellent robustness to uncertainties, PID controllers are the most commonly used in industry [49]. When a proportional term is obtained after an error is processed by proportional gain, it is fed through integral and derivative gain, which increases the overshoot when compared to a PID controller [50]. One of the main goals of this study is to minimize overrides due to road disruptions by using the shortest possible time change. PID controllers have three gains, each of which affects the manipulated variables and control mechanism, as discussed below. This study used a PID controller to control the MR damper because of its simplicity, ease to use, and good performance. By the following equation [49].

Proportional (P) part : $u_p(t) = k_c\big(y_s(t) - y(t)\big)$ (3.12)

Integral (I) part : $u_I(t) = \frac{k_c}{\tau_i}\int_0^t \big(y_s(\tau) - y(\tau)\big)d\tau$ (3.13)

Derivative (D) part : $u_D(t) = k_c\tau_d \frac{d(y_s(t)-y(t))}{dt}$ (3.14)

$$u(t) = u_p(t) + u_I(t) + u_D(t) \quad (3.15)$$

Where, y_s(t),y(t), and u(t) are the set point, the process output, and the PID controller output respectively . k_c,$\tau_I$and τ_d are the proportional gain, integral time, and derivative time.

3.6.1 Proportional Gain

The difference between the set point and the proportional gain is determined by the difference between the set point and the proportional gain, (desired operating point) and the state of the process variable (value) at the time [51]. The control system responds faster when the proportional gain is higher, but it creates process variable oscillations when the proportional gain is higher[50].

3.6.2 Integral Gain

The steady-state error eliminator is often referred to as integral gain. In a PID Controller, the integral is a representation of the Variation in error caused by the time [52]. The integral Function brings the process closer to its set point, avoiding the steady-state error variation that can occur with a pure preoperational controller[52].

3.6.3 Derivative Gain

The rate of change in the error term interacts with the PID's derivative gain. The decrease in system output is caused by system variables at a higher rate of change derivative gain[50]. As the derivative time parameter is increased, the controller works more forcefully and quickly against the term error. Because it is sensitive to the noisy signal, the derivative gain is usually kept low [50].

3.7 Zeigler-Nicholas method.

Zeigler-Nicholas method is the most used tuning method because it's very simple and, give good controller result[53]. Zeigler-Nicholas method is about to find out the ultimate gain K_u and, ultimate period P_u. Ultimate gain is a minimum gain value that sustains sinusoidal oscillations and, the ultimate period is the distance between two pikes of oscillation. Ultimate gain determine by changing the controller gain until reaches self-sustaining oscillations, then determine ultimate period, and go to Ziegler-Nicholas table to obtain the proportional - integral - derivative gains as shown in table 3.2 [53].

Table 3.2 Ziegler-Nicholas table

Controller	**Tuning parameter**[a]		
	k_c	τ_i	τ_d
P	$k_u/2.0$	-	-
PI	$k_u/2.2$	$p_u/1.2$	-
PID	$k_u/1.7$	$p_u/2.0$	$p_u/8.0$

Chapter Four

Result and Discussion

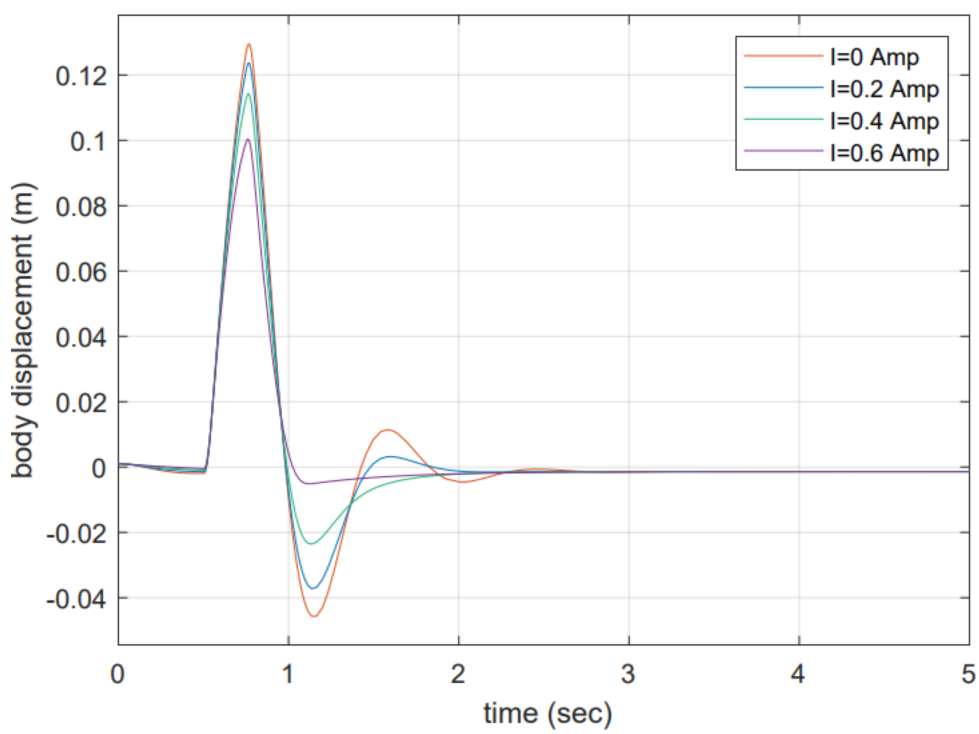
INTRODUCTION

This chapter introduces the results and discusses the simulation results. The simulations result that determine using MATLAB/Simulink software and discuss it for both cases controlling and, uncontrolling case. Applied different road profile (square with 10cm amplitude, bump wave 10 cm, and, random wave with -1cm to 1 cm) With applied range of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp).

4.1 Uncontrolled Case Passive Case

4.1.1 Current Effect on Response

In this case the effect of current value on MR damper response and damping force under different road profiles (square, bump, and, random).



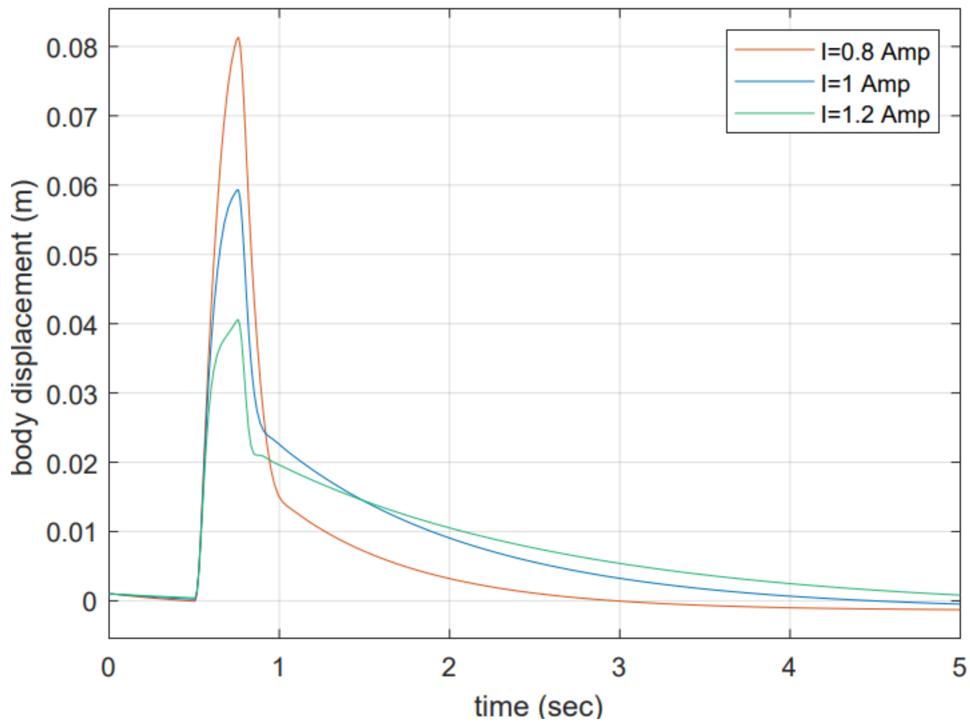
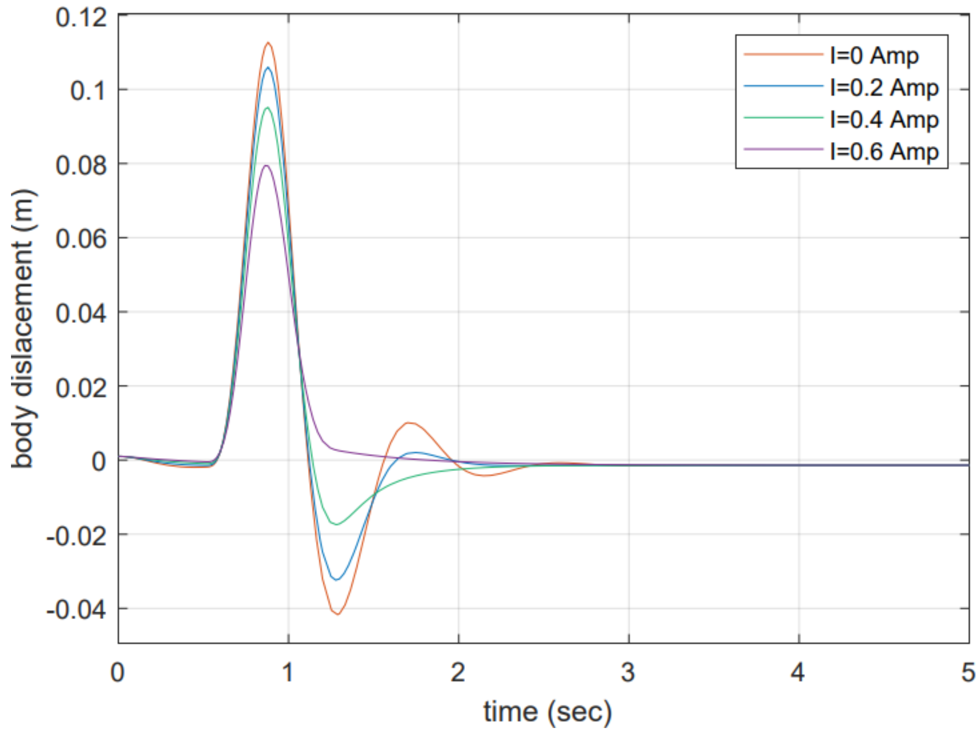


Figure 4.1.a Displacement of car body under square excitation with the different input current.

Figure.4.1.a, indicated the value of body displacement for the suspension system with variable value of current (0, 0.2,0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper. When a current flow through the electrical circuit in the piston head generates a magnetic flux in MR fluid which lead to a change it from a liquid phase to a semi-solid phase that obstructs the piston head moves as a result of this lead to reduce the velocity due to equation 3.3 and 3.4 reduce the overshoot.



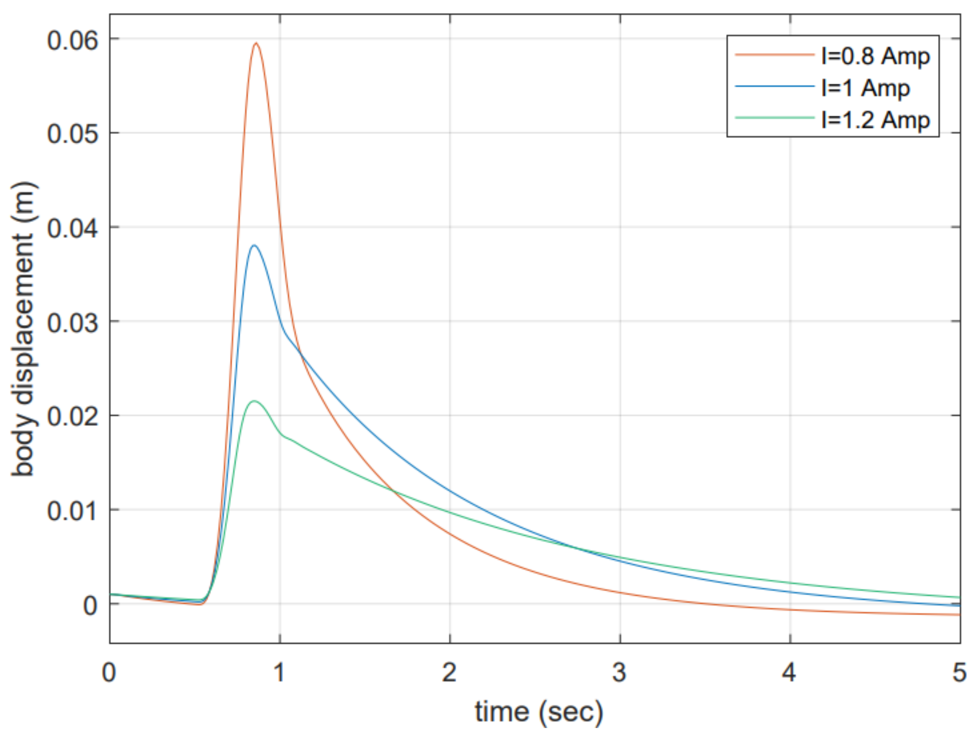
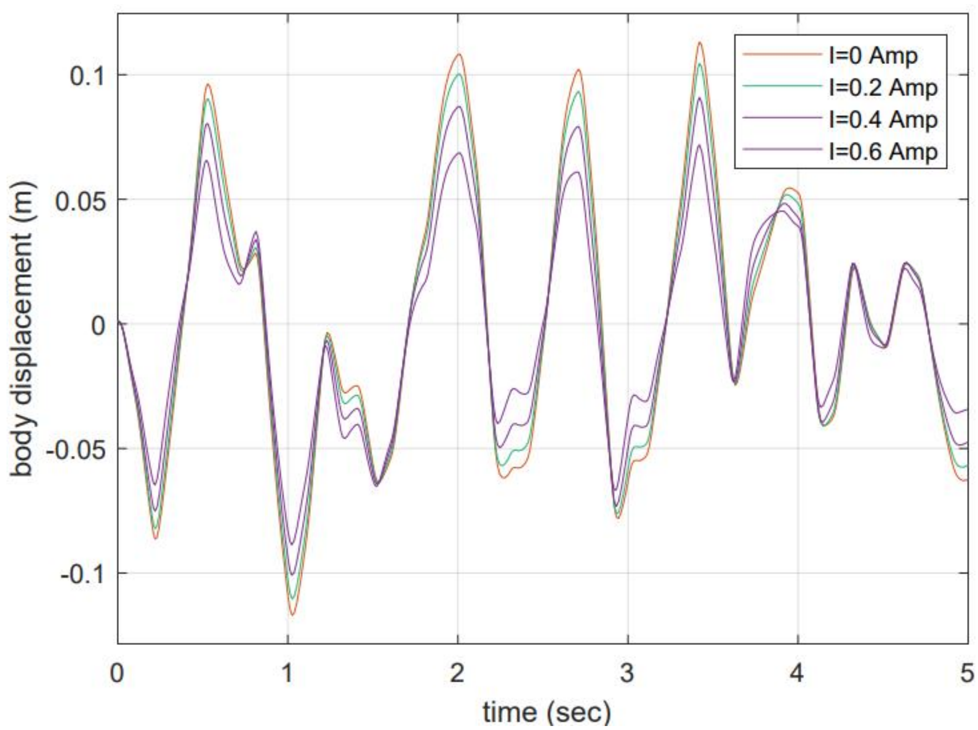


Figure 4.1.b Displacement of car body under bump excitation with the different input current.

Figure 4.1.b indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for bump exaction.



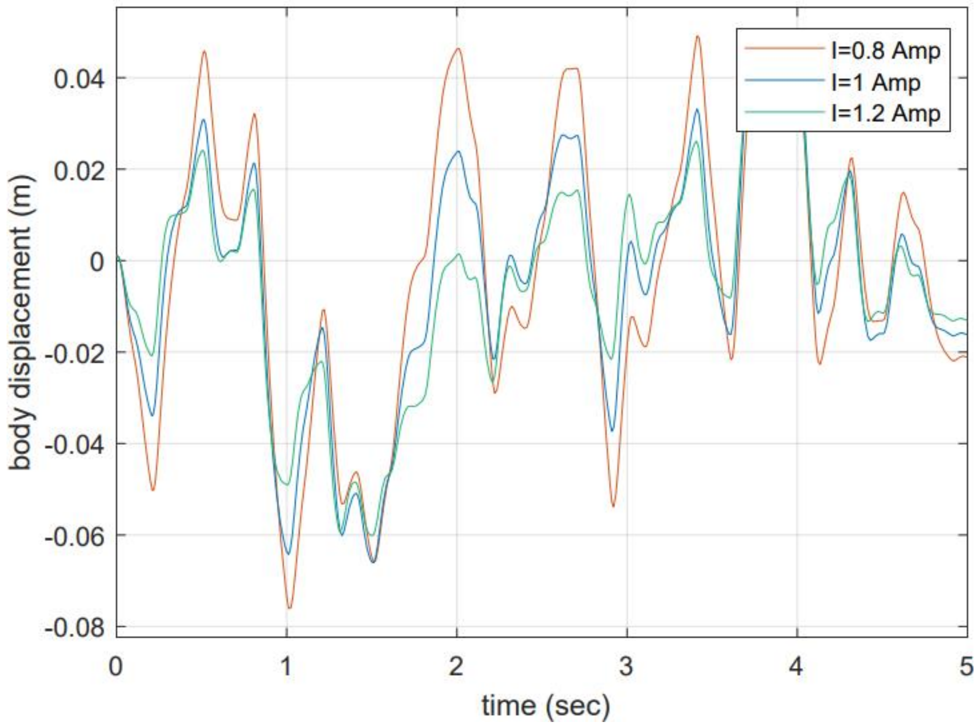


Figure 4.1.c Displacement of car body under random excitation with the different input current.

Figure 4.1.c indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for random exaction.

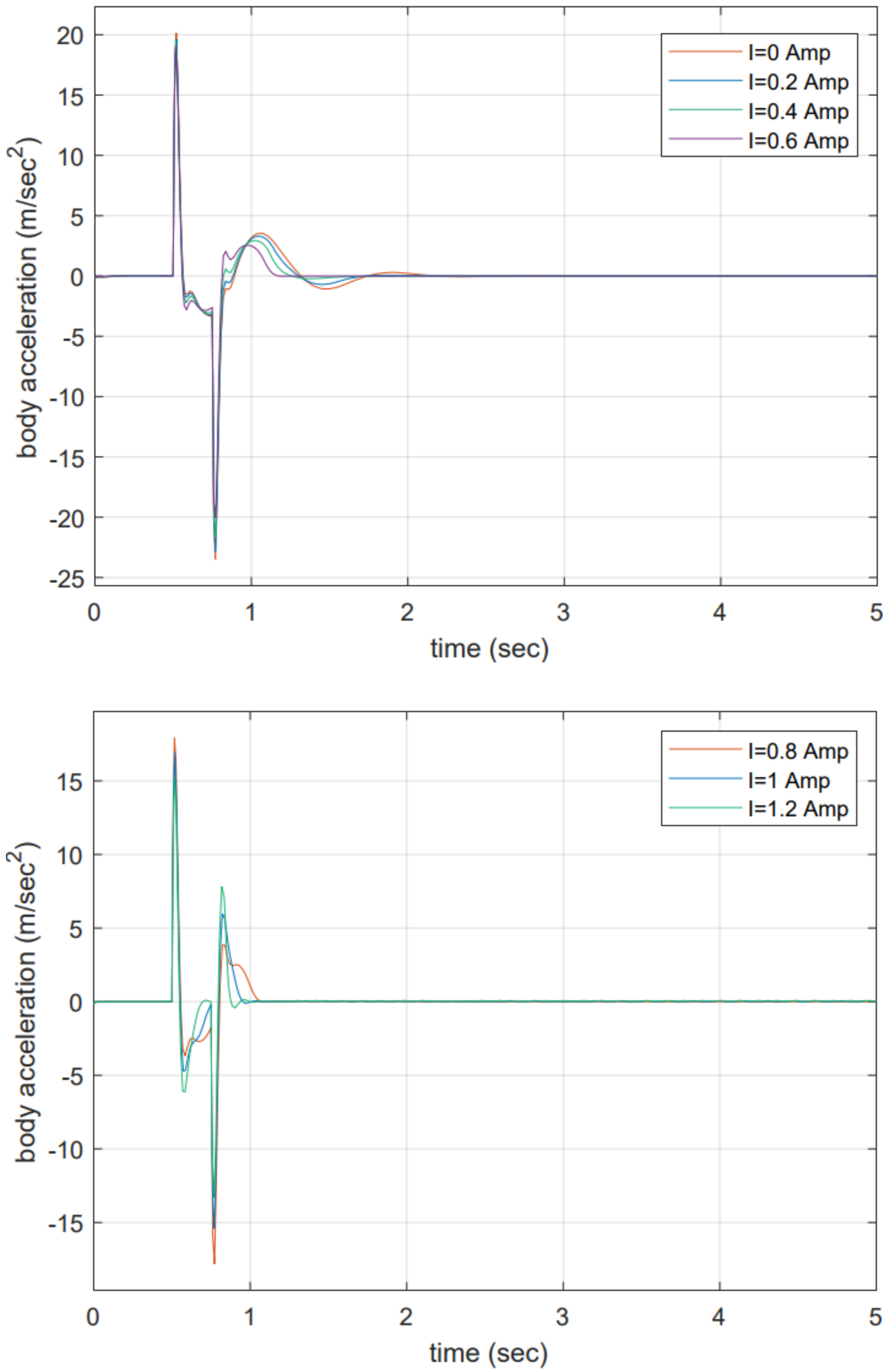
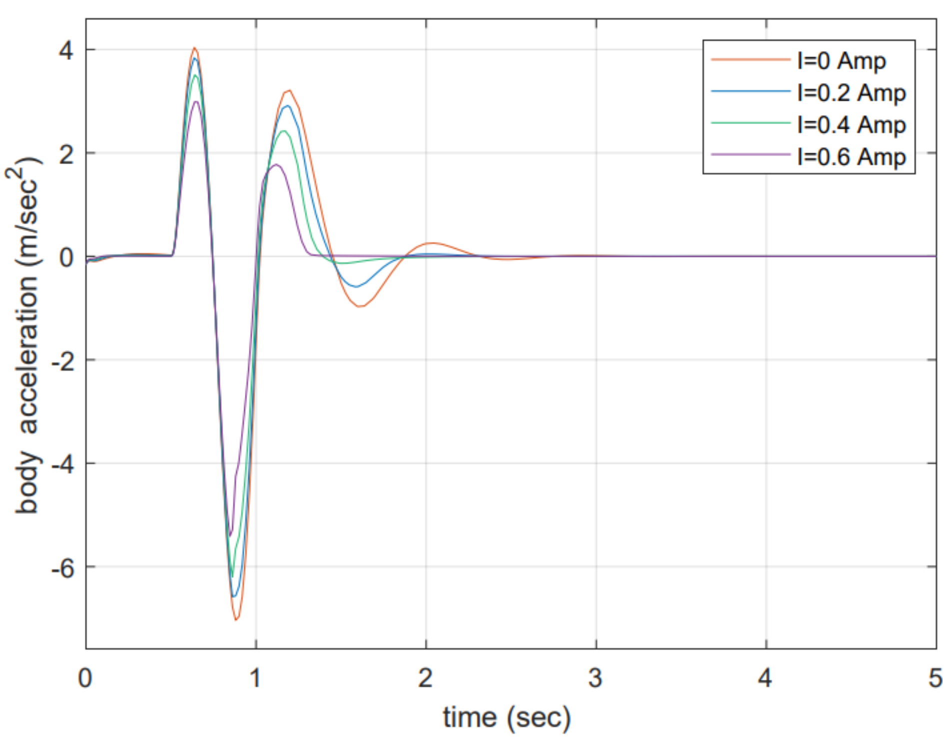


Figure 4.2.a Acceleration of car body under square excitation with the different input current

Figure.4.2.a indicated the value of acceleration of body for the suspension system with variable value of current (0, 0.2, 0 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper, improves the ride comfortability need to reduce the acceleration of car body. Figure 4.2 show that the acceleration and the body response are inversely proportional to the input current. Which reduce the body acceleration under square exaction.



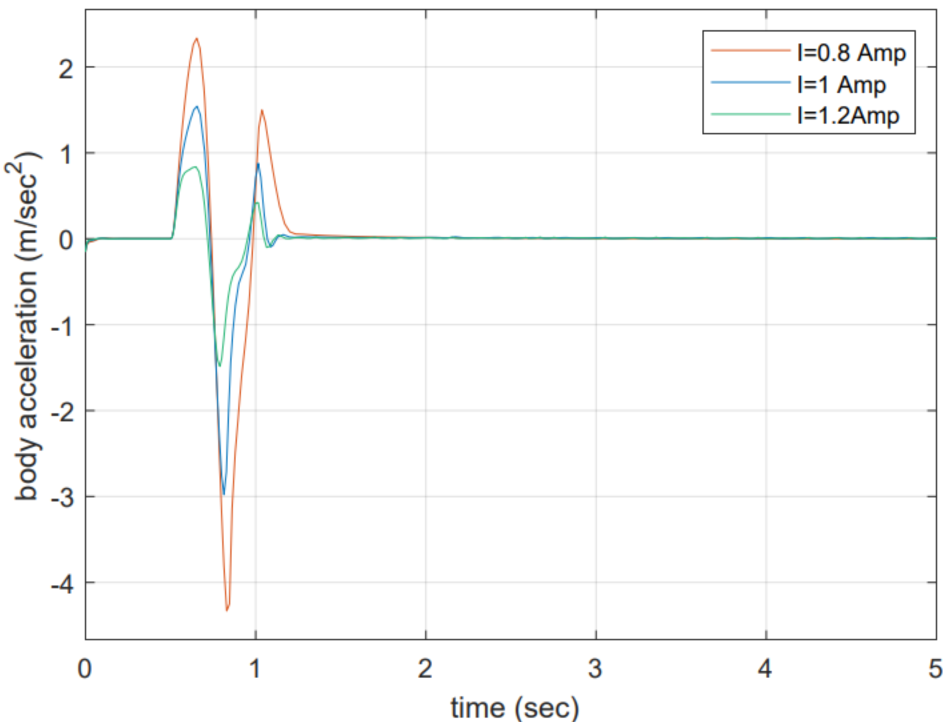
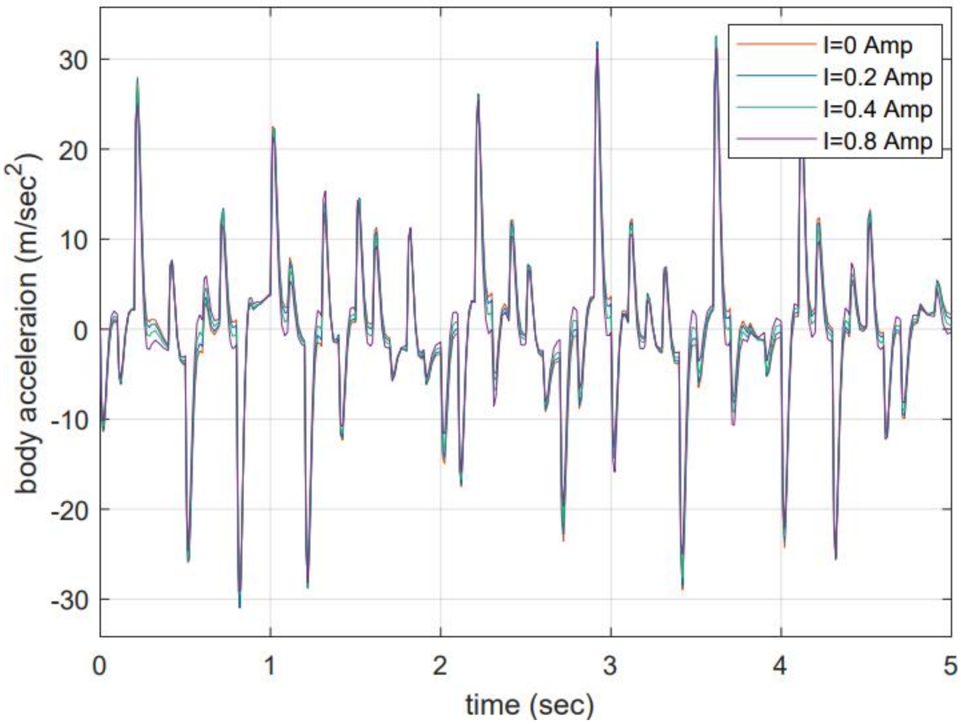


Figure 4.2.b Acceleration of car body under bump excitation with the different input current

Figure 4.2.b indicated the value of acceleration of body for the suspension system with variable value of current (0, 0.4, 0.8, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper lead to reduce the body acceleration in bump exaction.



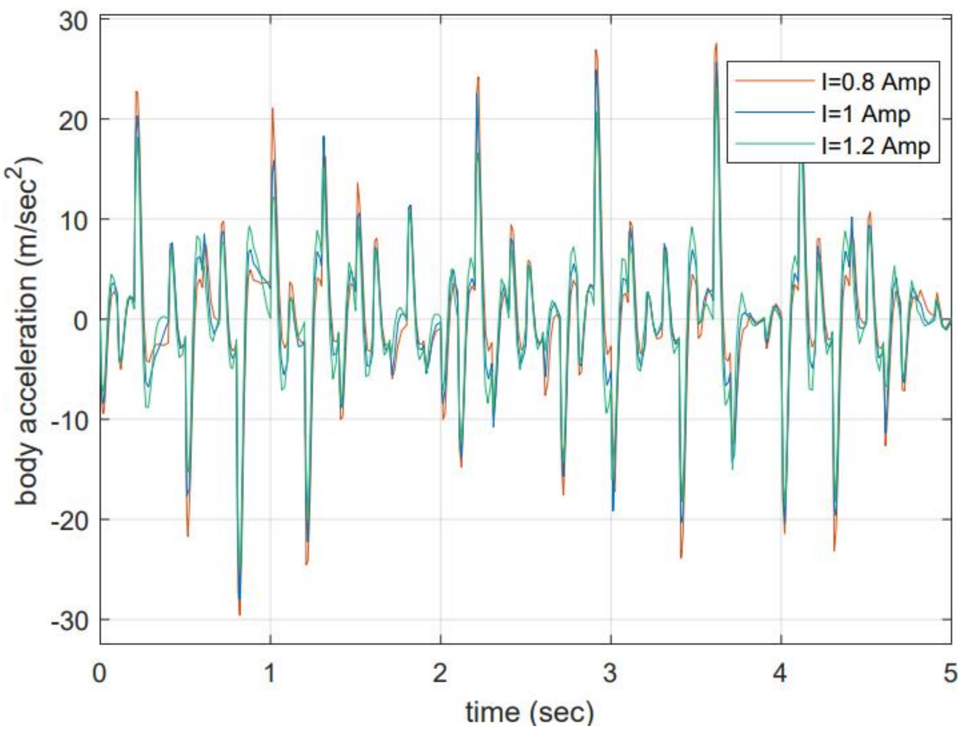
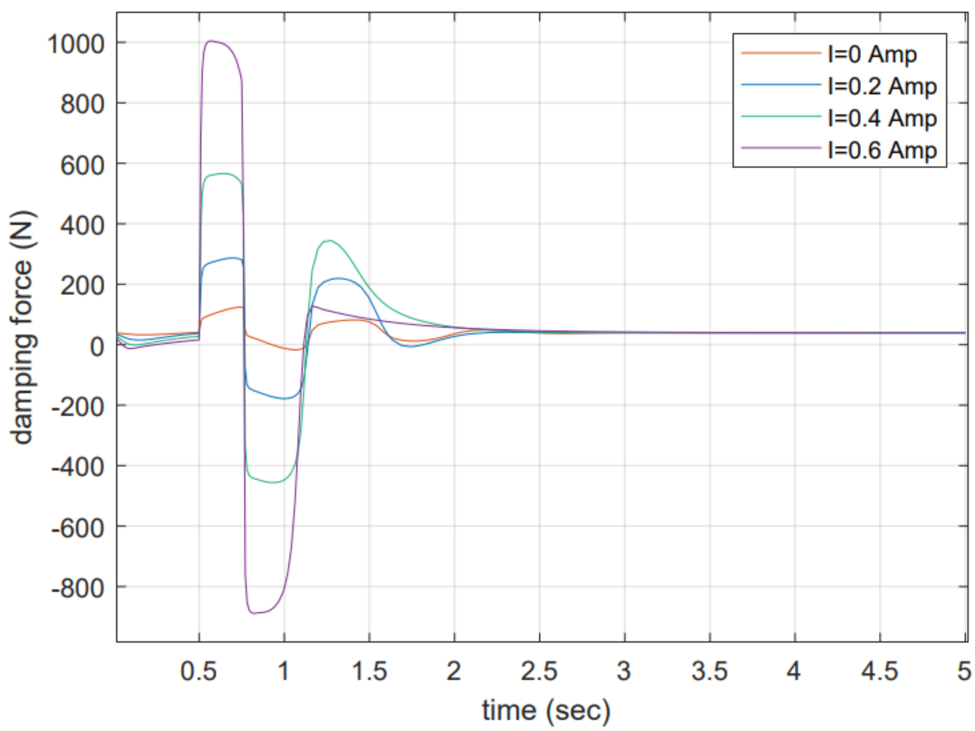


Figure 4.2.c Acceleration of car body under random excitation with the different input current

Figure 4.2.c indicated the value of acceleration of body for the suspension system with variable value of current (0, 0.4, 0.8, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper lead to reduce the body acceleration in random exaction.



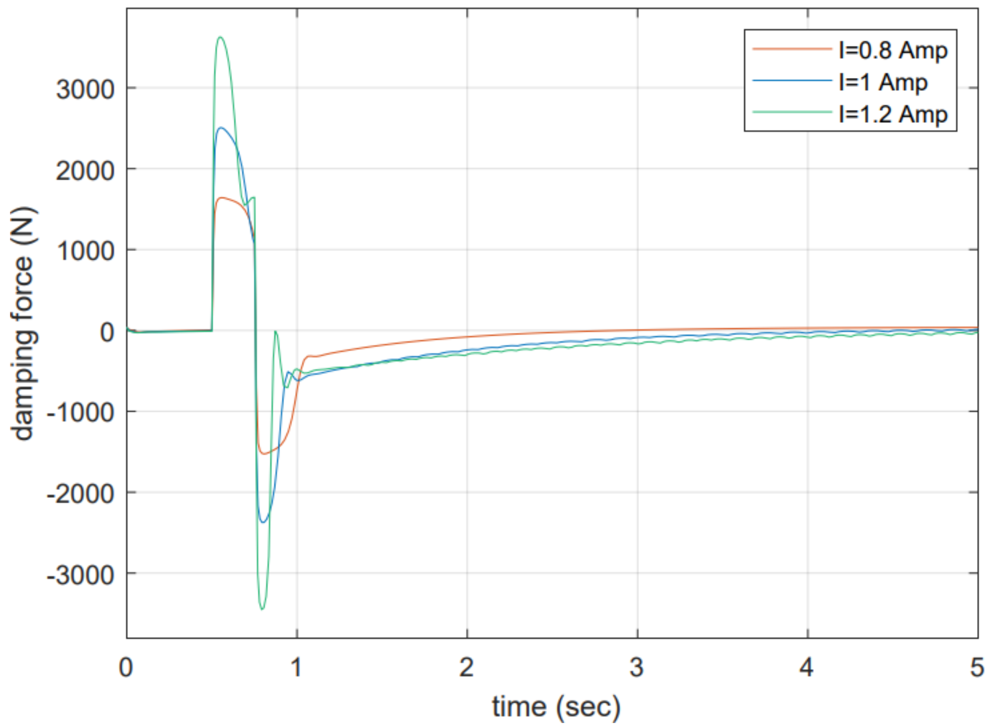


Figure 4.3.a Damping Force of car body under square excitation with the different input current

Figure 4.3.a represent the value of force generated by MR damper after supply the current where the force generated increase with increased the current to drag the force generated by excitation road and to decrease the car body displacement to save the suspension system and passengers from square excitation. The increase of force with time as equation (3-10) in chapter three, where the force is function of some parameters ($c_0$andF_c) as function of time according equation (3-11) in chapter three.

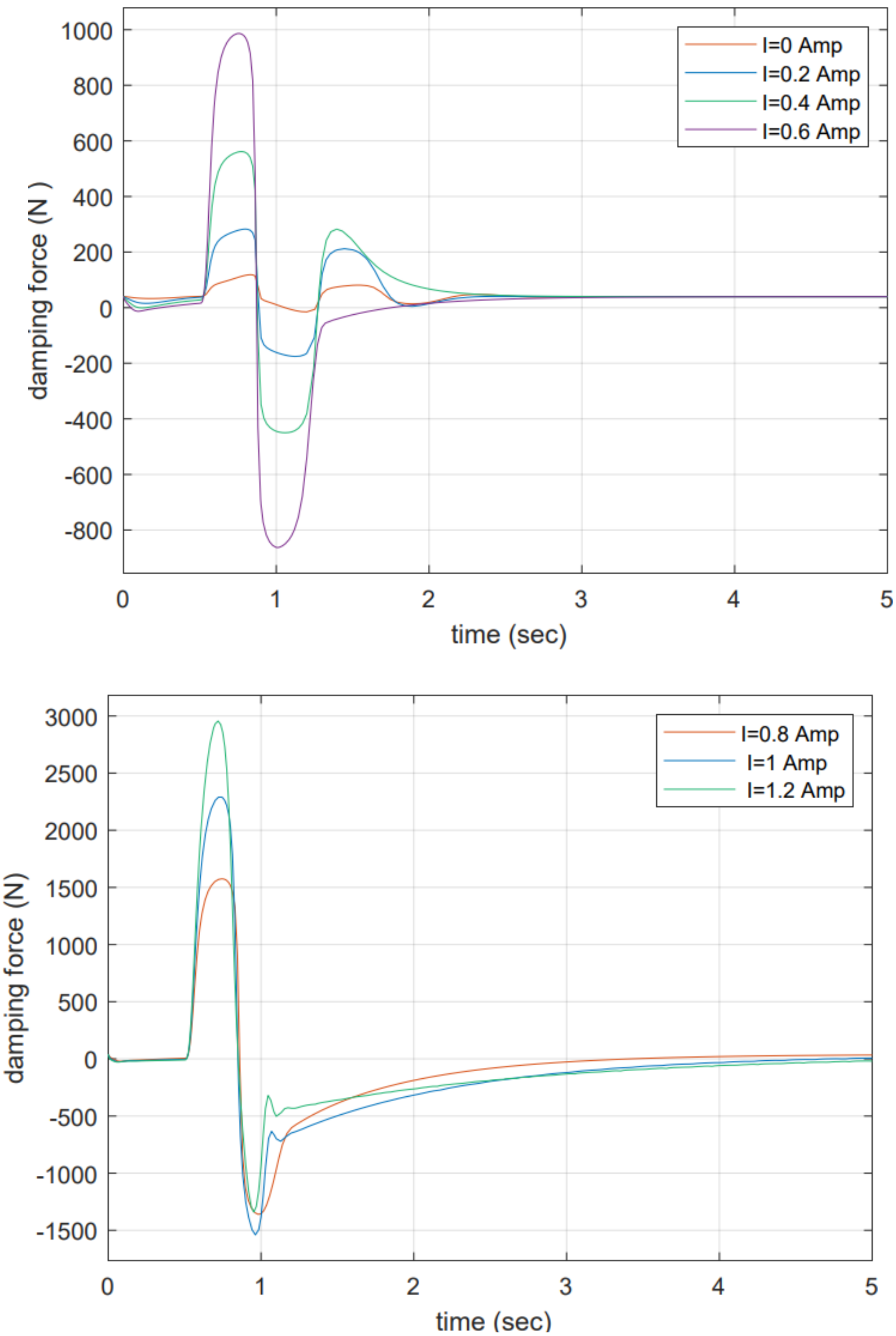
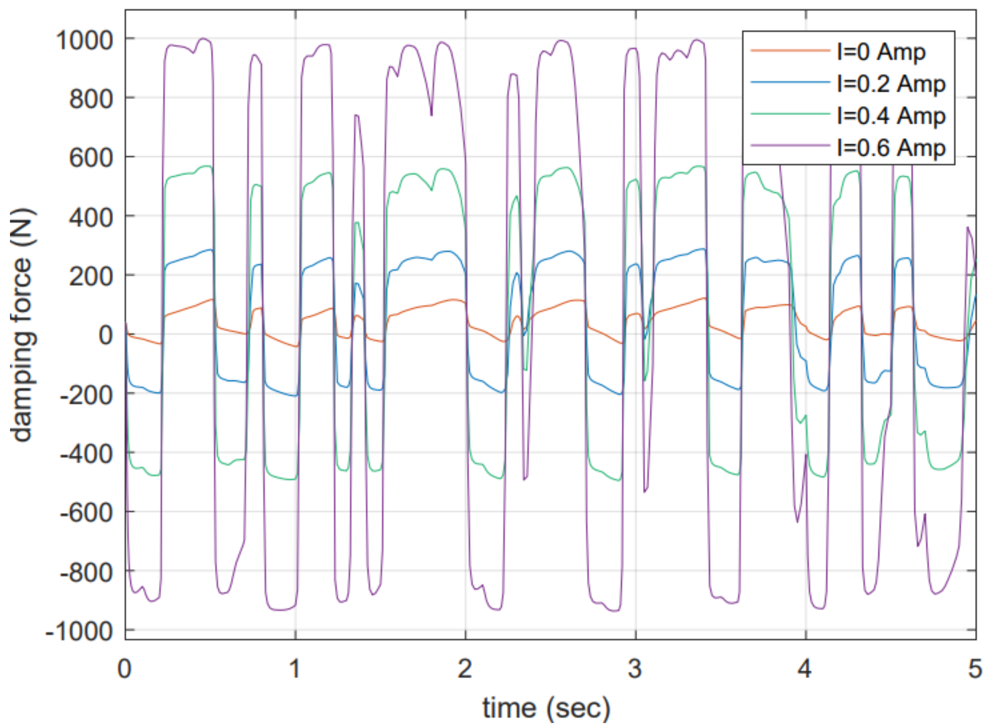


Figure 4.3.b Damping Force of car body under bump excitation with the different input current

Figure 4.3.b represent the value of force generated by MR damper after supply the current where the force generated increase with increased the current to drag the force generated by excitation road and to decrease the car body displacement to save the suspension system and passengers from bump excitation.



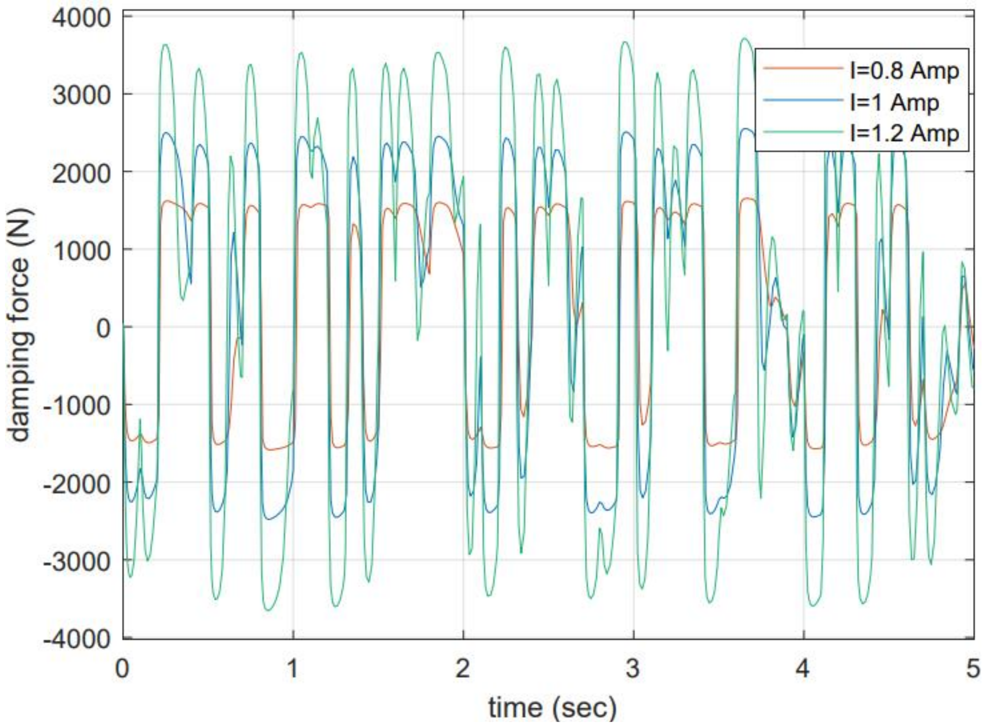


Figure 4.3.c Damping Force of car body under random excitation with the different input current

Figure 4.3.c represent the value of force generated by MR damper after supply the current where the force generated increase with increased the current to drag the force generated by excitation road and to decrease the car body displacement to save the suspension system and passengers from random excitation.

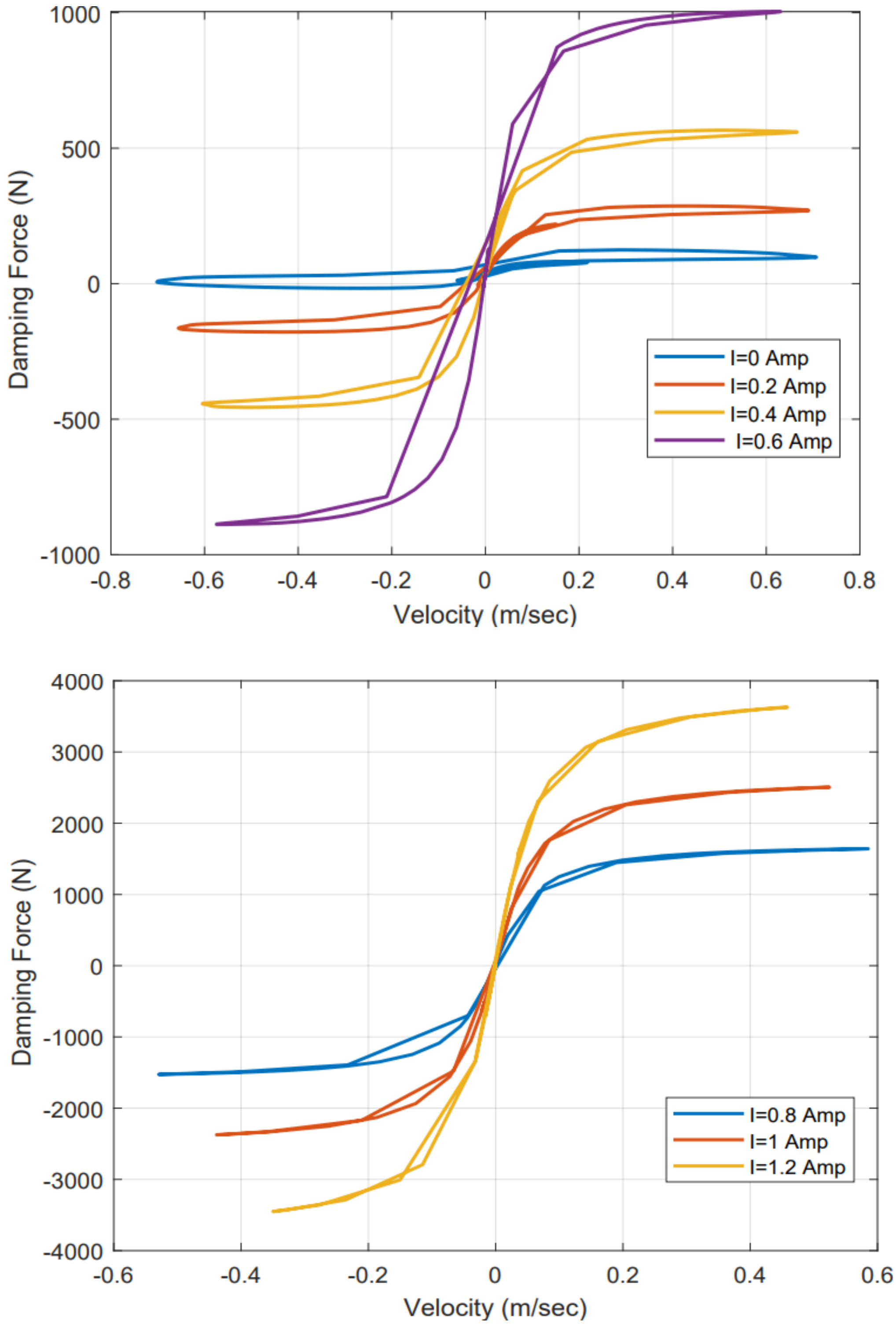
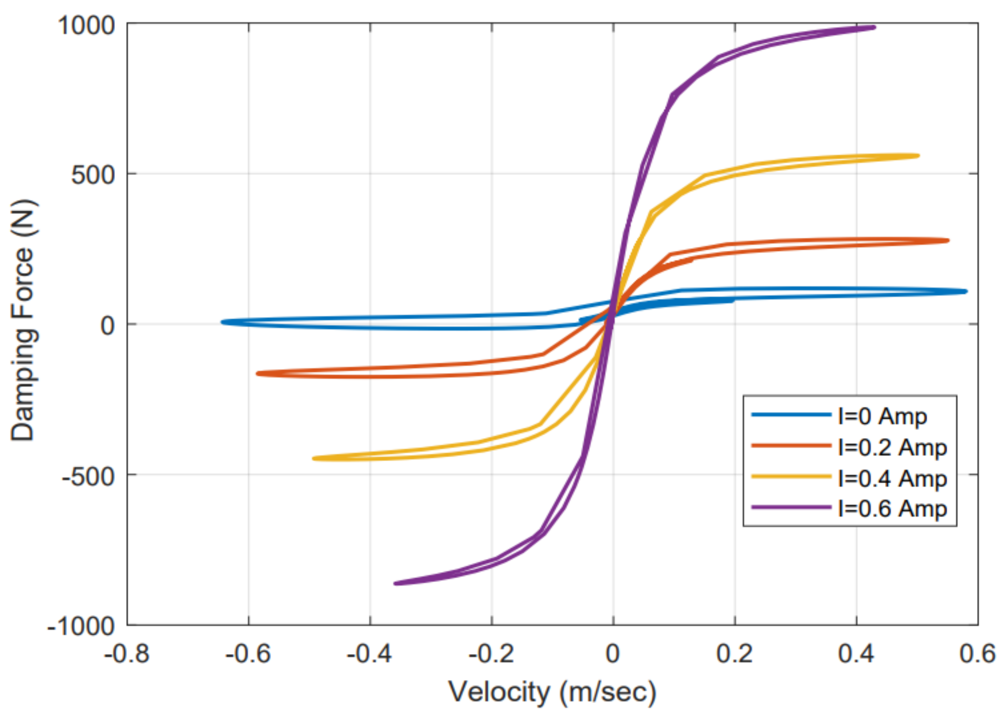


Figure 4.4.a the damping force along with damper relative velocity (histories) System behavior under sinusoidal

The force-velocity hysteresis loop for the MR damper at different value of current are shown in figure 4.4.a It is observable that the MR damper behaves approximately as a friction damper where the friction force level can be changed by changing the applied damper current under square exaction. The hysteresis loop of force-velocity is increasing from 0Amp up to the 1.2Amp. By increasing the current, the Magneto-rheological (MR) damper generate more force, and velocity of the MR damper decreases that lead to the vibration of the body goes to the stable state. the primary factor is that an increase in current will result in a rise in the magnetic flux density in the damping gap.



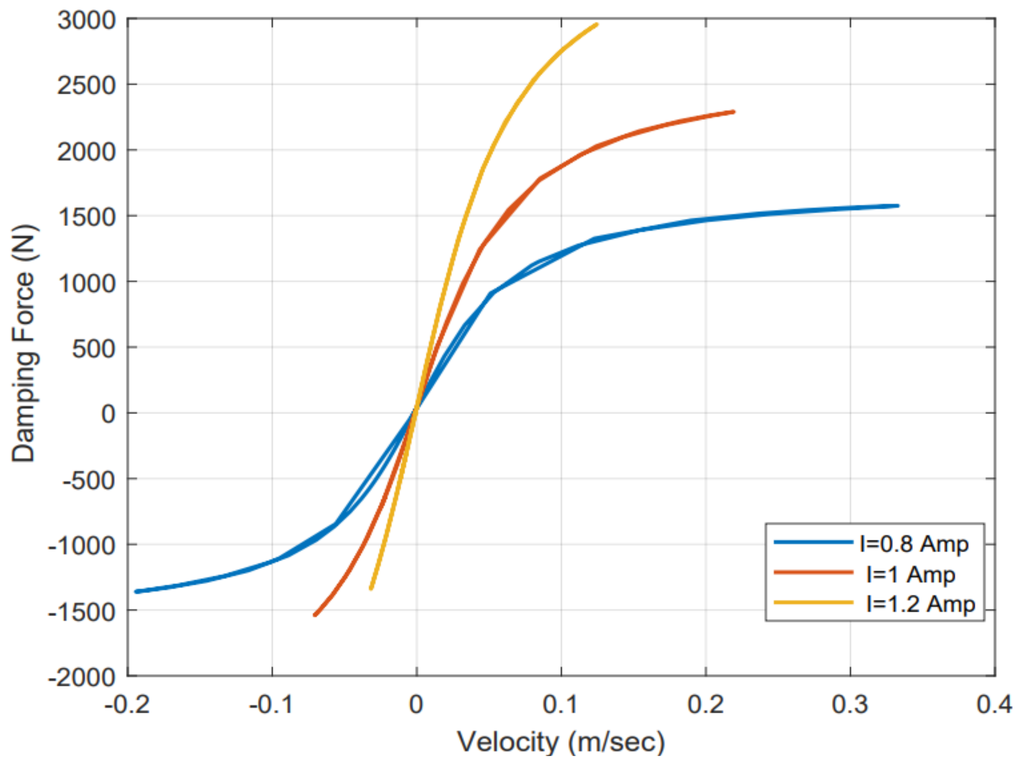


Figure 4.4.b the damping force along with damper relative velocity (histories) System behavior under sinusoidal

The force-velocity hysteresis loop for the MR damper at different value of current are shown in Figure 4.4.b. It is observable that the MR damper behaves approximately as a friction damper where the friction force level can be changed by changing the applied damper current under bump exaction. The hysteresis loop of force-velocity is increasing from 0A up to the 1.2A. By increasing the current, the Magneto-rheological (MR) damper generate more force, and velocity of the MR damper decreases that lead to the vibration of the body goes to the stable state.

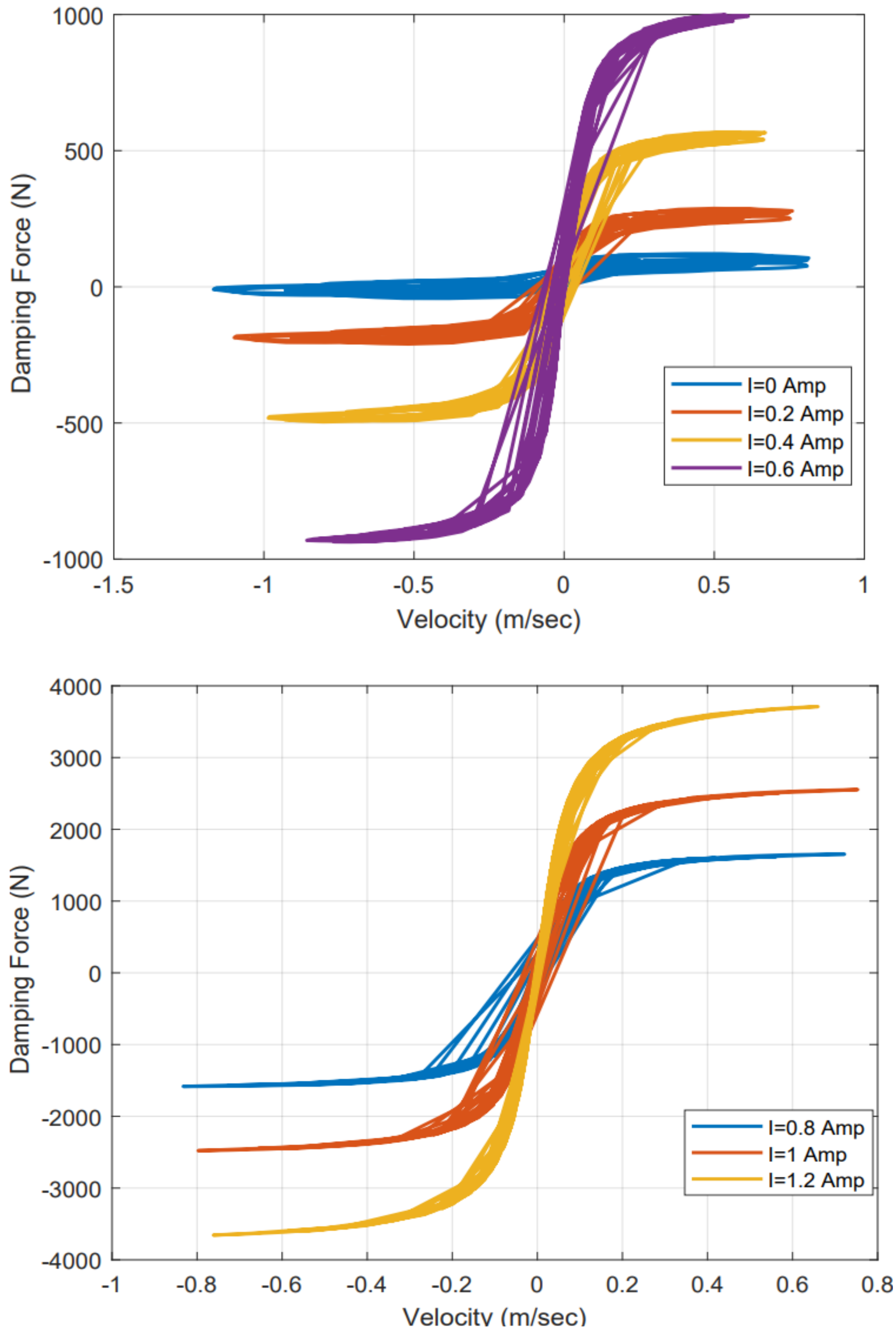


Figure 4.4.c the damping force along with damper relative velocity (histories) System behavior under random exaction

The force-velocity hysteresis loop for the MR damper at different value of current are shown in Figure 4.4.b. It is observable that the MR damper behaves approximately as a friction damper where the friction force level can be changed by changing the applied damper current under random exaction. The hysteresis loop of force-velocity is increasing from 0A up to the 1.2A. By increasing the current, the Magneto-rheological (MR) damper generate more force, and velocity of the MR damper decreases that lead to the vibration of the body goes to the stable state.

4.2 Controller Case

For controlled cases using PID (proportional, integral, and derivative) controller. The reason for using of PID controller is to simplify, excellent controlling performance, good robustness and easy to apply in a computer or microprocessor [49]. Using Ziegler-Nicholas method of tuning to tune the P, PI, and, PID controllers and compared between them. The PID controllers structure and diagram of the control system are shown in Figure.4.5 [54-56].

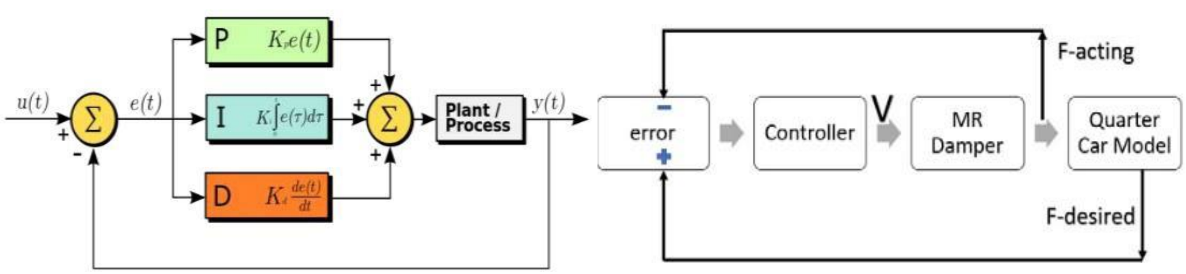


Figure 4.5 (a) PID controller configuration[56] **, (b)** Flowchart of the control system [57]**.**

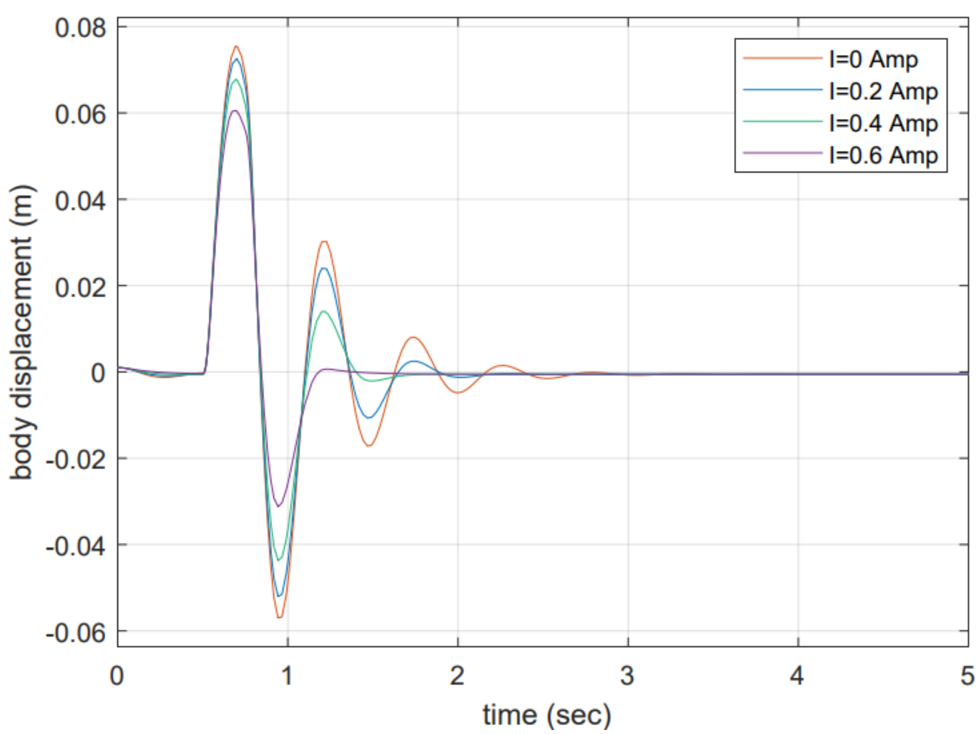
From the Ziegler-Nicholas method determine the value of ultimate gain and frequency, K_U=290 and P_U=578.66. Using table 5.1 to find out P, PI, and, PID controller gain.

Table 4.1 Ziegler-Nicholas table to determent gain vale [53]

Controller	K_p	K_i	k_d
P	$K_u/2$	-	-
PI	$K_u/2.2$	$P_u/1.2$	-
PID	$K_u/1.7$	$P_u/2$	$P_u/8$

4.2.1 P controller

The simplest form of PID controller. From the Ziegler-Nicholas method find proportional gain K_P=145.



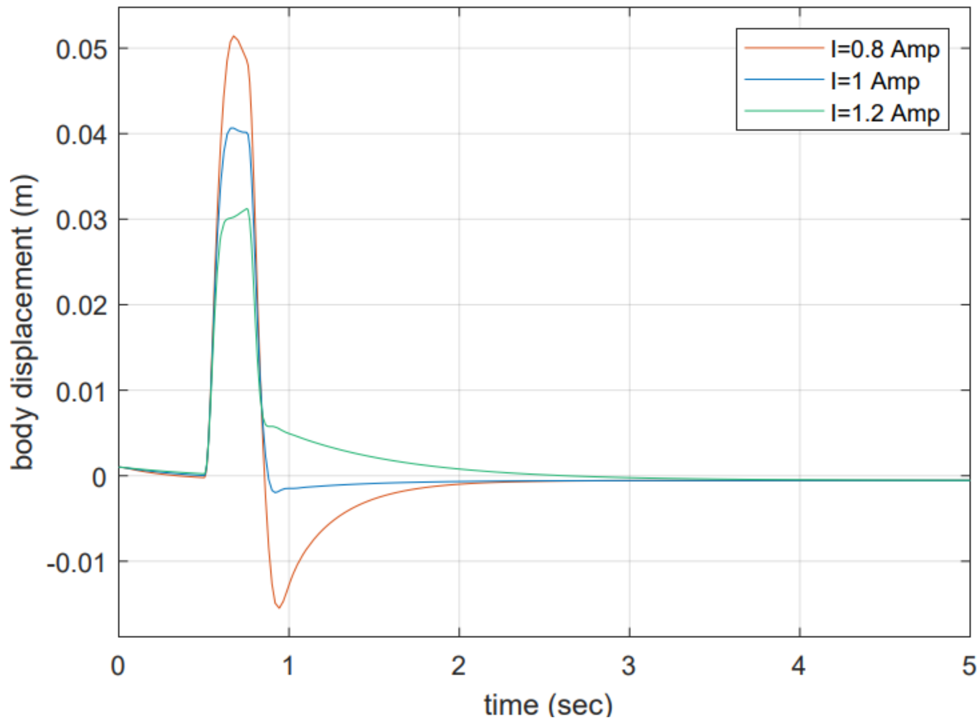


Figure 4.5.a Displacement of car body under square excitation with the different input current in P controller case

Figure 4.5.a shows the effect of the P controller on MR damper response, indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for square exaction.

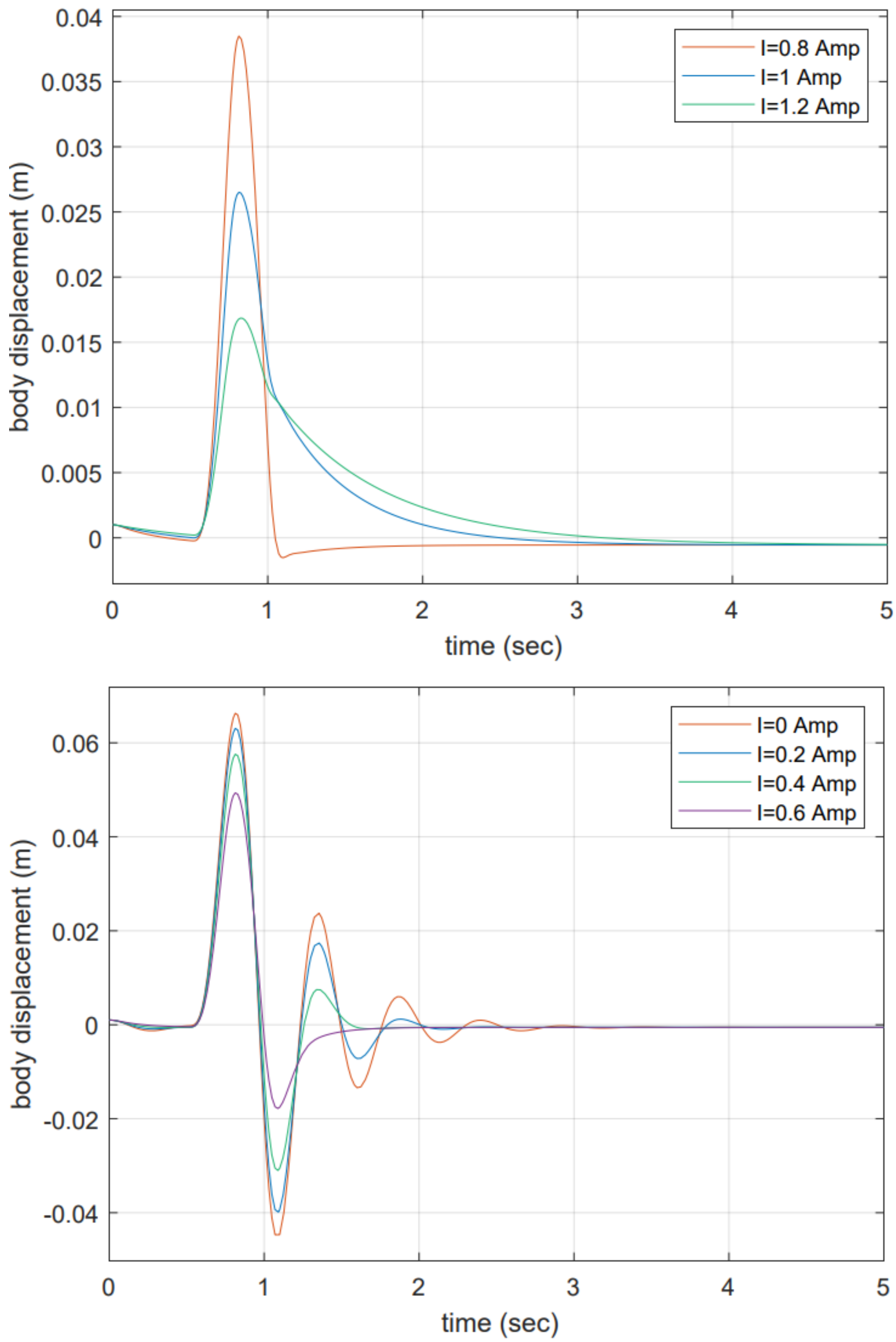
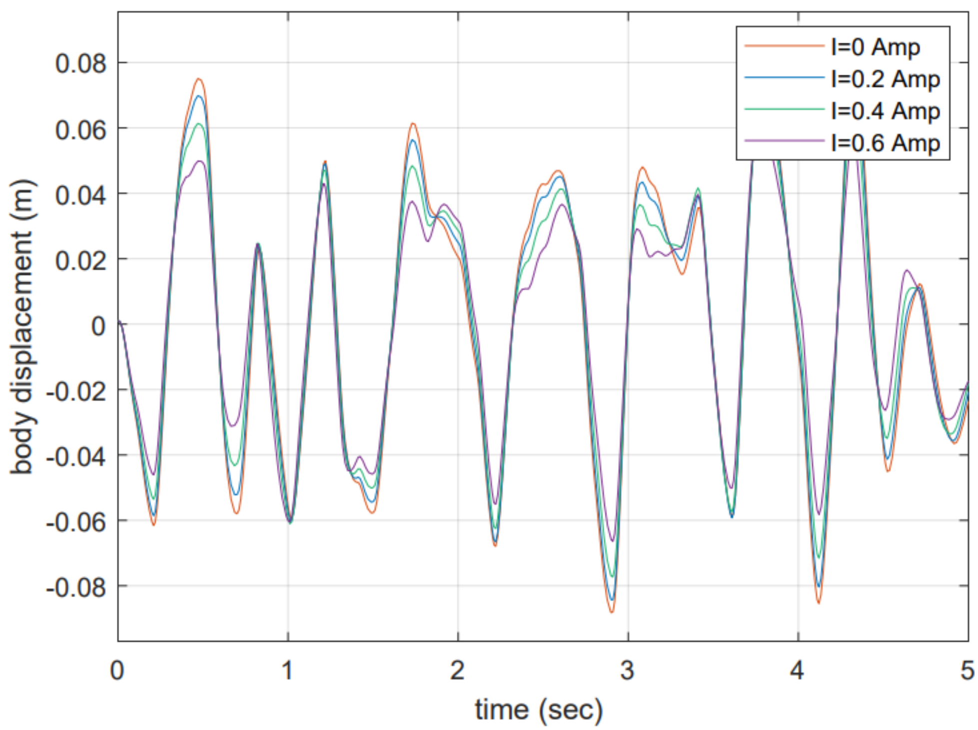


Figure 4.5.b Displacement of car body under bump excitation with the different input current in P controller case

Figure 4.5.b shows the effect of the P controller on MR damper response, indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for bump exaction.



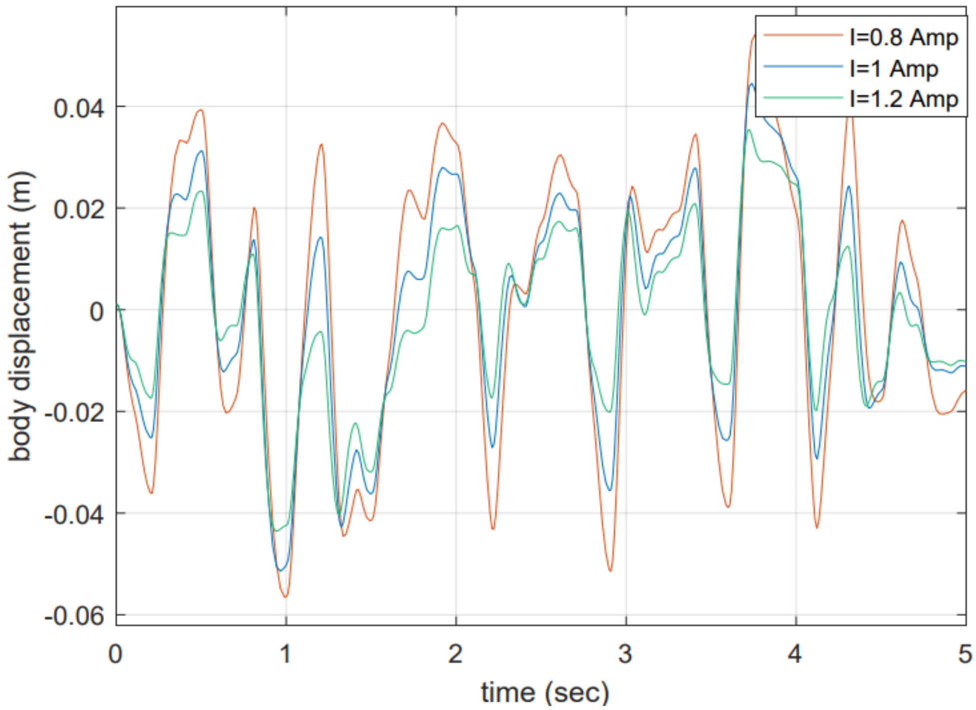


Figure 4.5.c Displacement of car body under random excitation with the different input current in P controller case

Figure 4.5.c shows the effect of the P controller on MR damper response, indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for random exaction.

4.2.2 PI Controller

The second form of PID controller which includes proportional and integral gain. By using the Ziegler-Nicholas method determine the value of the proportional and integral gain which is $K_P = 130.5$ and, $K_I = 482.22$.

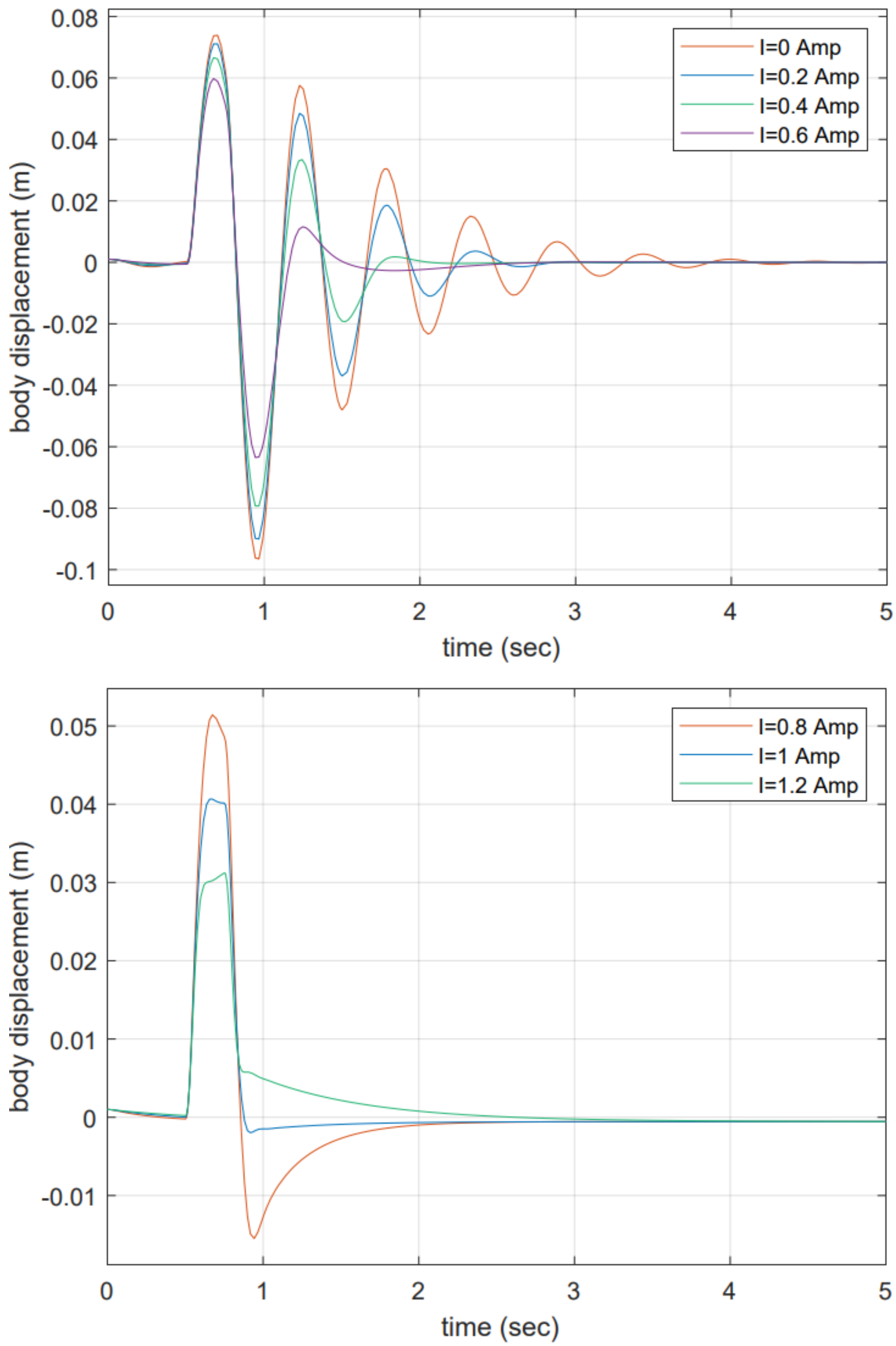
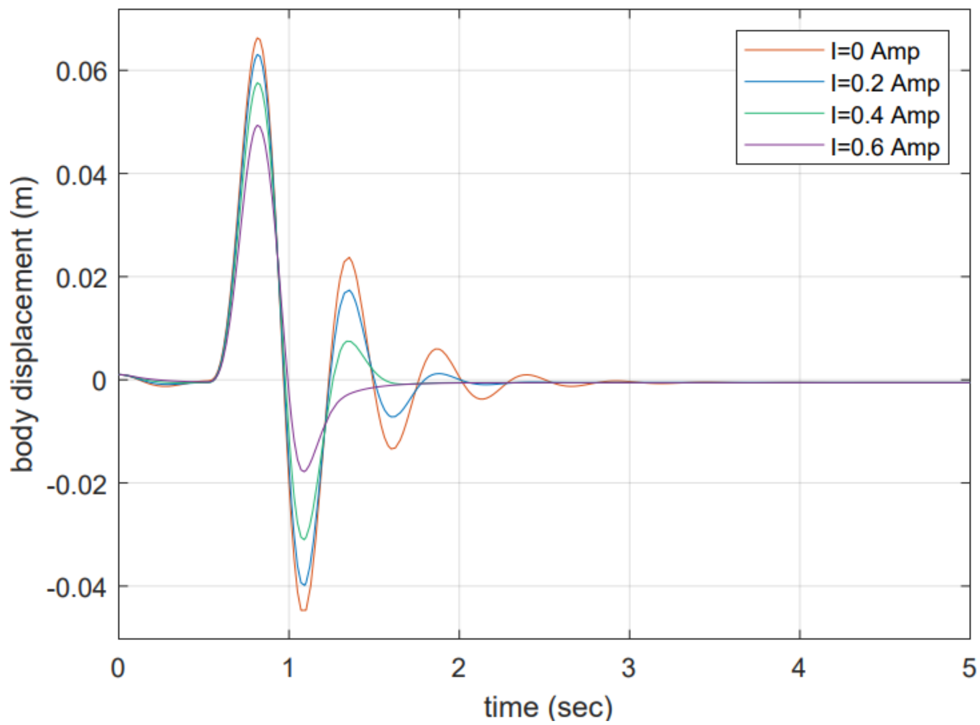


Figure 4.6.a Displacement of car body under square excitation with the different input current in PI controller case

Figure 4.6.a shows the effect of the PI controller on the MR damper response. indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for square exaction.



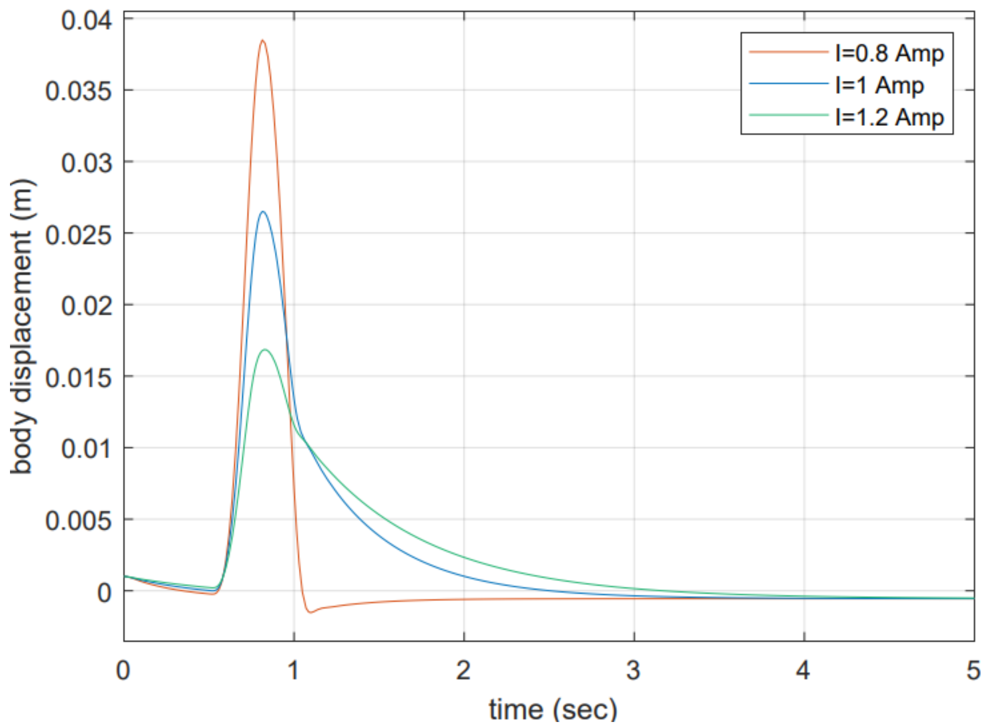


Figure 4.6.b Displacement of car body under bump excitation with the different input current in PI controller case

Figure 4.6.b shows the effect of the PI controller on the MR damper response. indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for bump exaction.

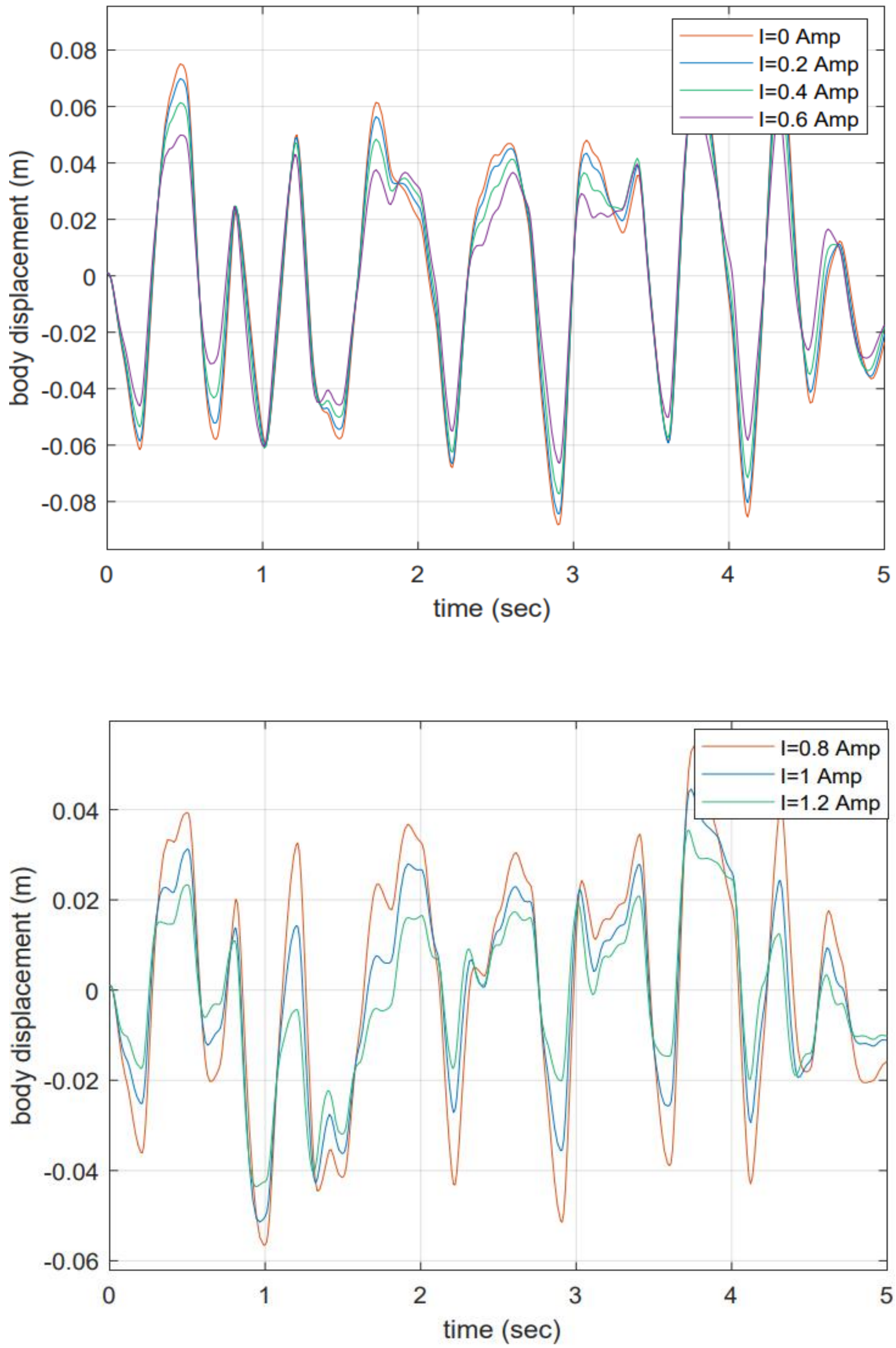


Figure 4.6.c Displacement of car body under random excitation with the different input current in PI controller case

Figure 4.6.c shows the effect of the PI controller on the MR damper response. indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for random exaction.

4.2.3 PID Controller

Its final form and including proportional-integral-derivative parts. By using the Ziegler-Nicholas method, determine the values of PID controller gains. $K_P = 174, K_I = 803.7,$ and $K_D = 9.42.$ Figure 5.7 shows the effect of the PID controller on MR damper response

From previous results for three different forms of PID controller including P, PI, and PID that the PID has better performance as compared with forms in terms of reducing the overshoot. As a result of this, this study will focus on the PID controller and find out its effect on damping force.

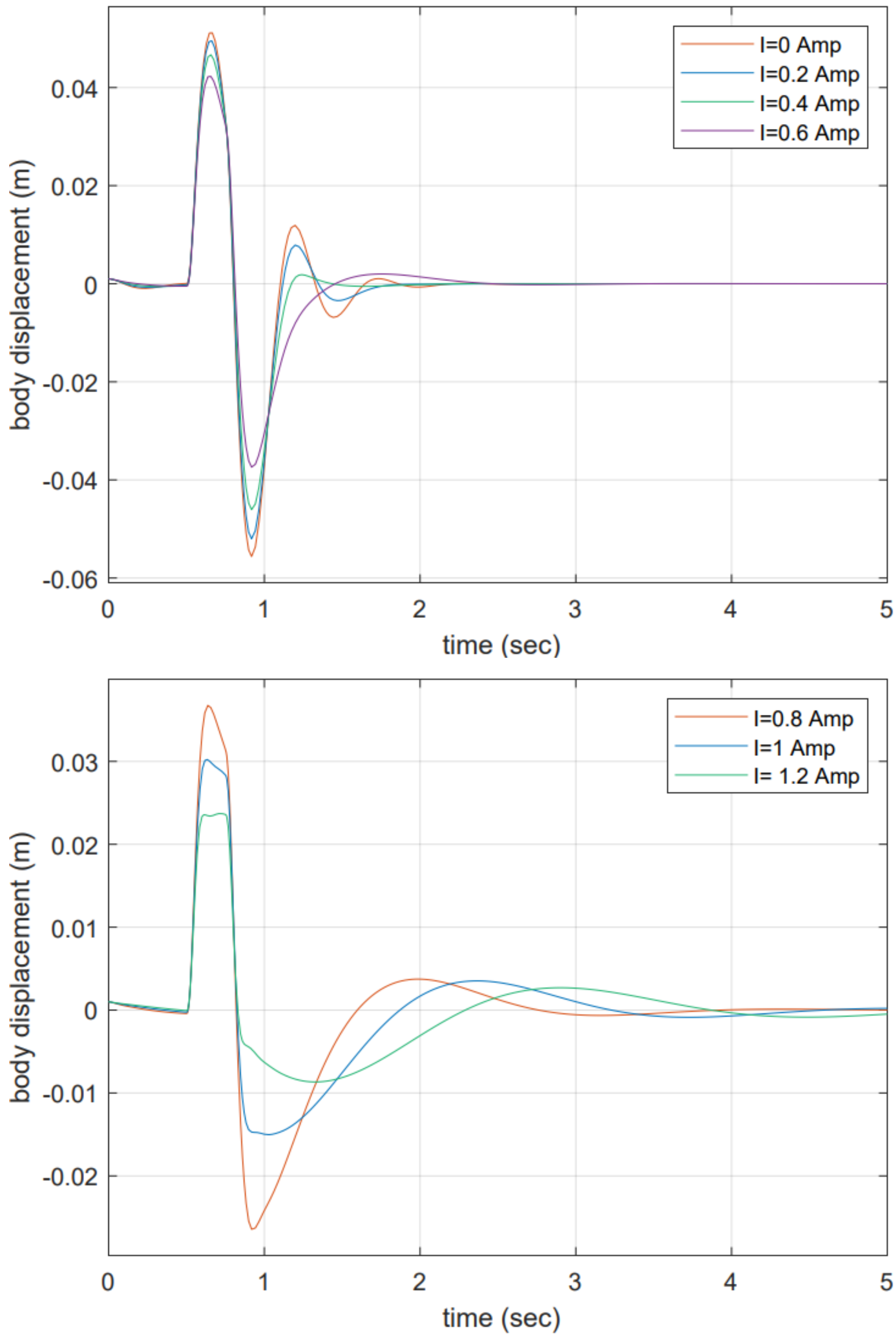
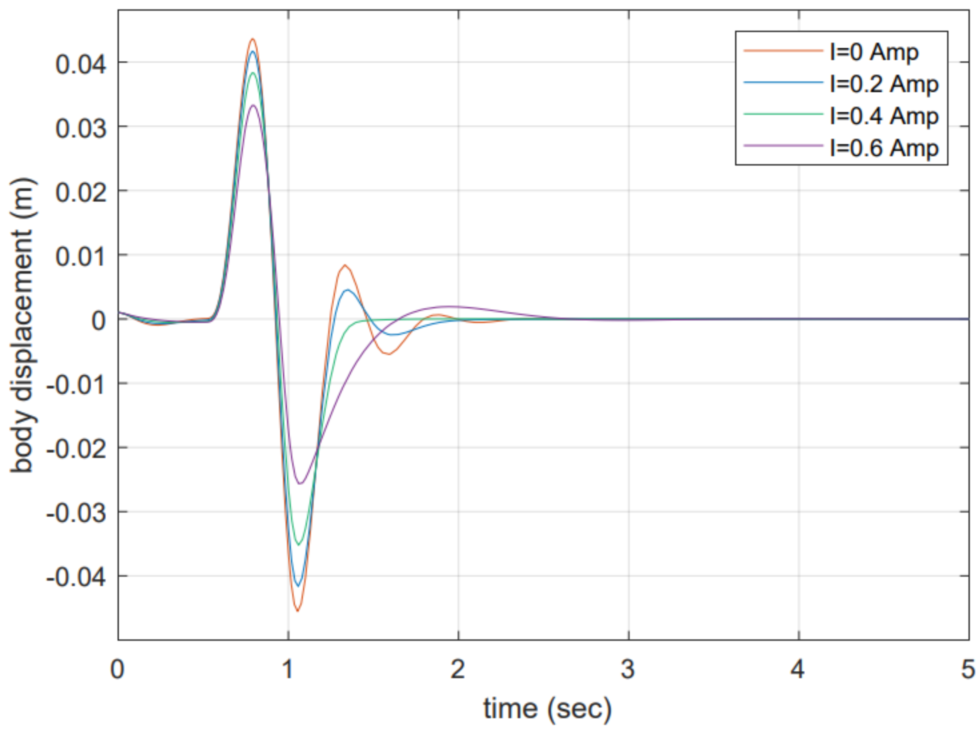


Figure 4.7.a Displacement of car body under square excitation with the different input current in PID controller case

Figure 4.7.a shows the effect of the PID controller on the MR damper response. indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for square exaction.



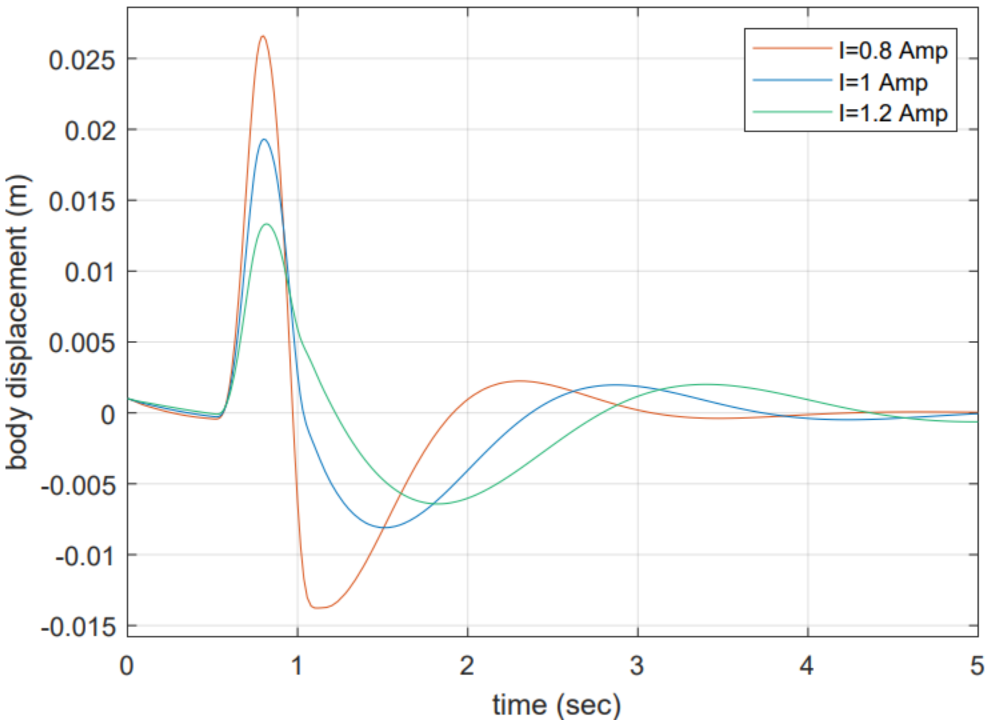


Figure 4.7.b Displacement of car body under bump excitation with the different input current in PID controller case

Figure 4.7.b shows the effect of the PID controller on the MR damper response. indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for bump exaction.

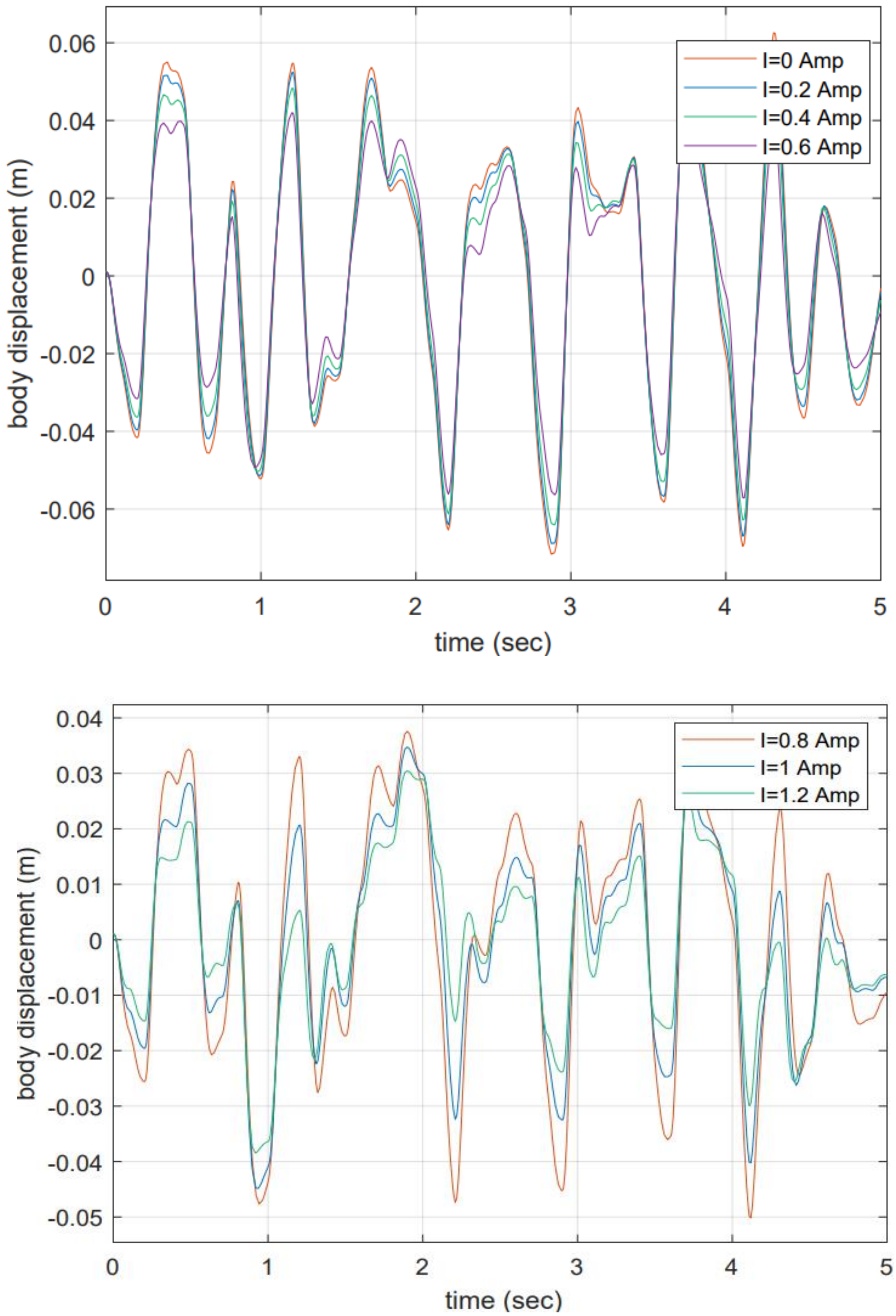
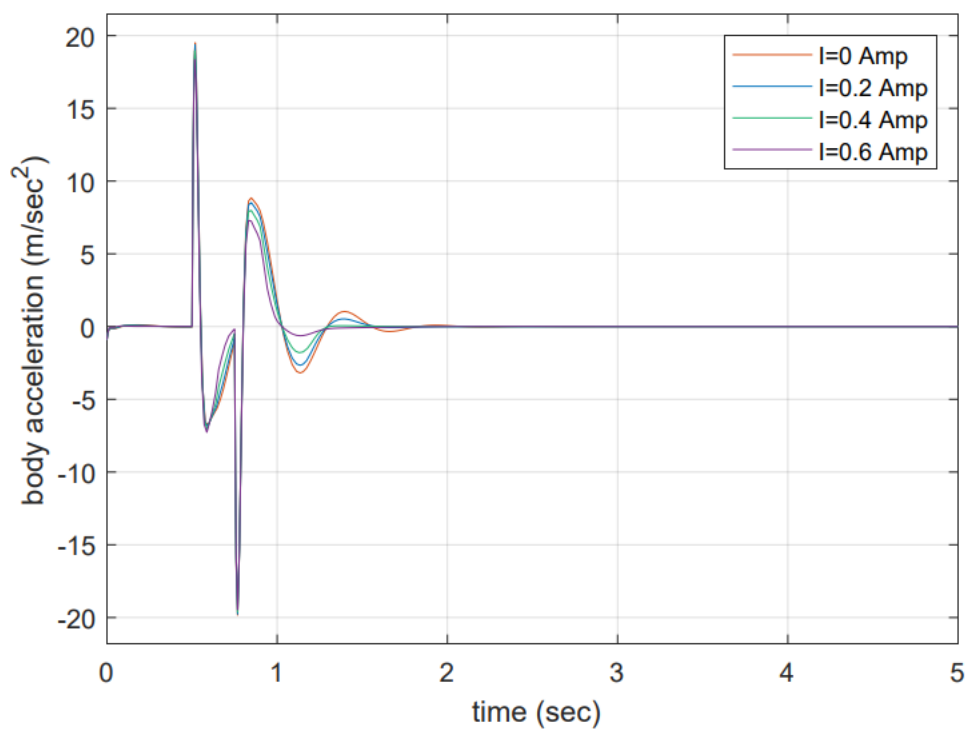


Figure 4.7.c Displacement of car body under random excitation with the different input current in PID controller case

Figure 4.7.c shows the effect of the PID controller on the MR damper response. Indicated the value of body displacement for the suspension system with variable value of current (0, 0.2, 0.4, 0.6, 0.8, 1, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper reduce the overshoot for random exaction.



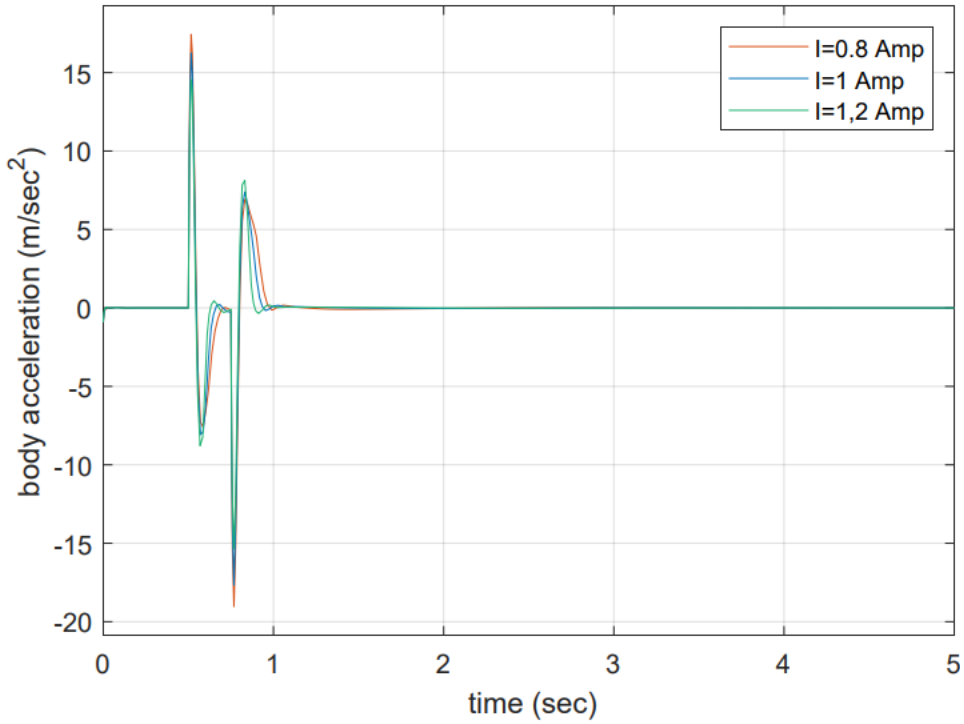
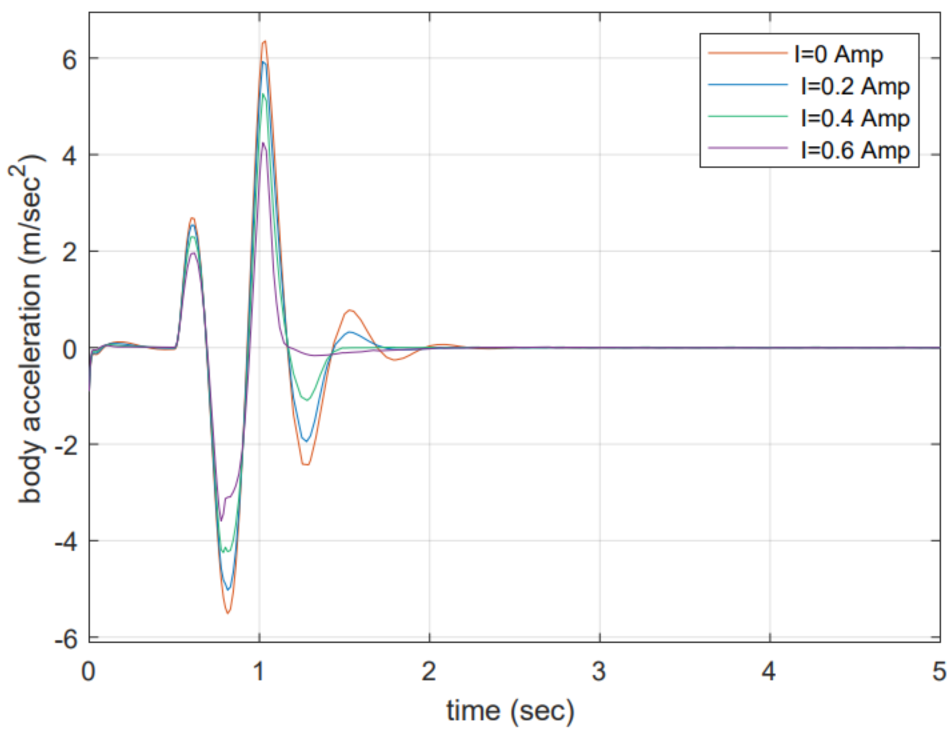


Figure 4.8.a Acceleration of car body under square excitation with the different input current in PID controller case.

Figure 4.8.a represent the effect of current on acceleration of car body with PID controller, Indicated the value of acceleration of body for the suspension system with variable value of current (0, 0.4, 0.8, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper lead to reduce the body acceleration in random exaction.



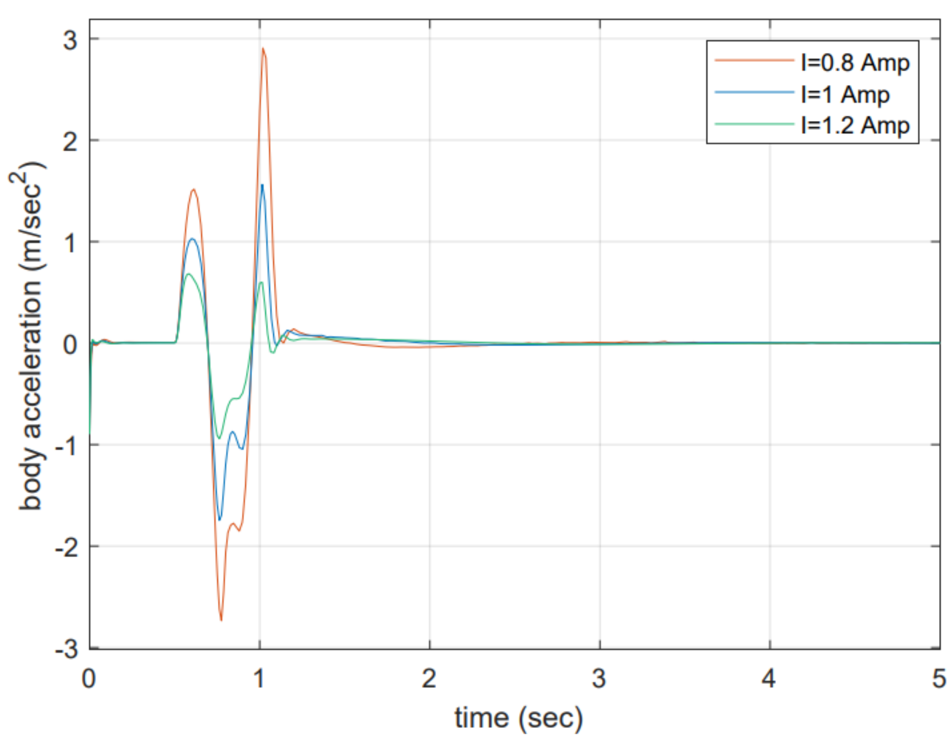
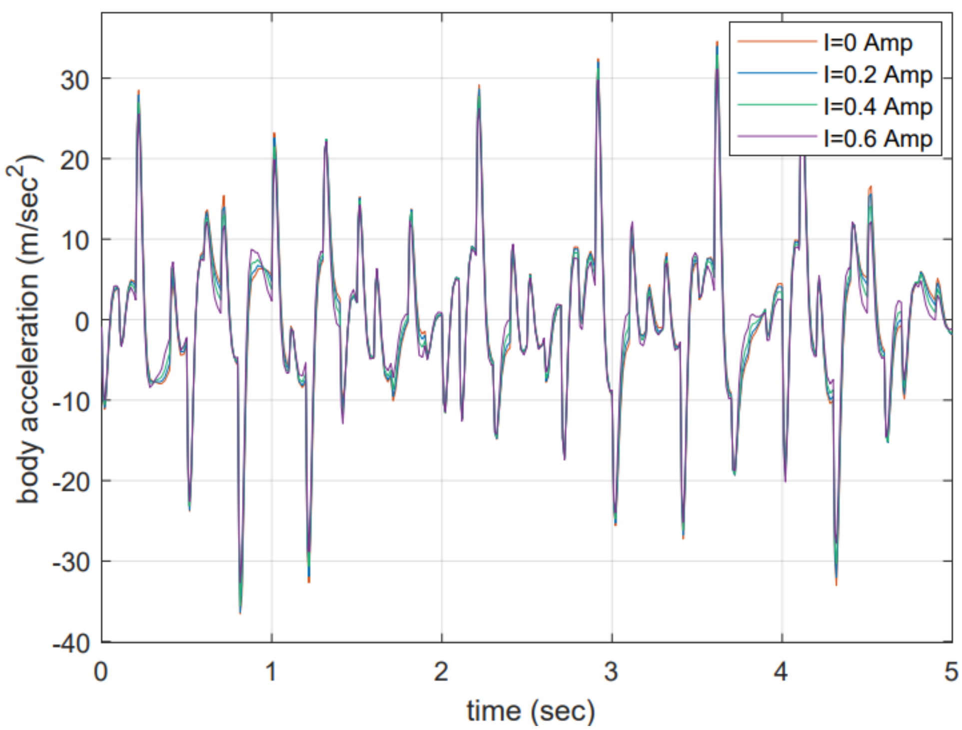


Figure 4.8.b Acceleration of car body under bump excitation with the different input current in PID controller case.

Figure 4.8.b represent the effect of current on acceleration of car body with PID controller, Indicated the value of acceleration of body for the suspension system with variable value of current (0, 0.4, 0.8, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper lead to reduce the body acceleration in bump exaction.



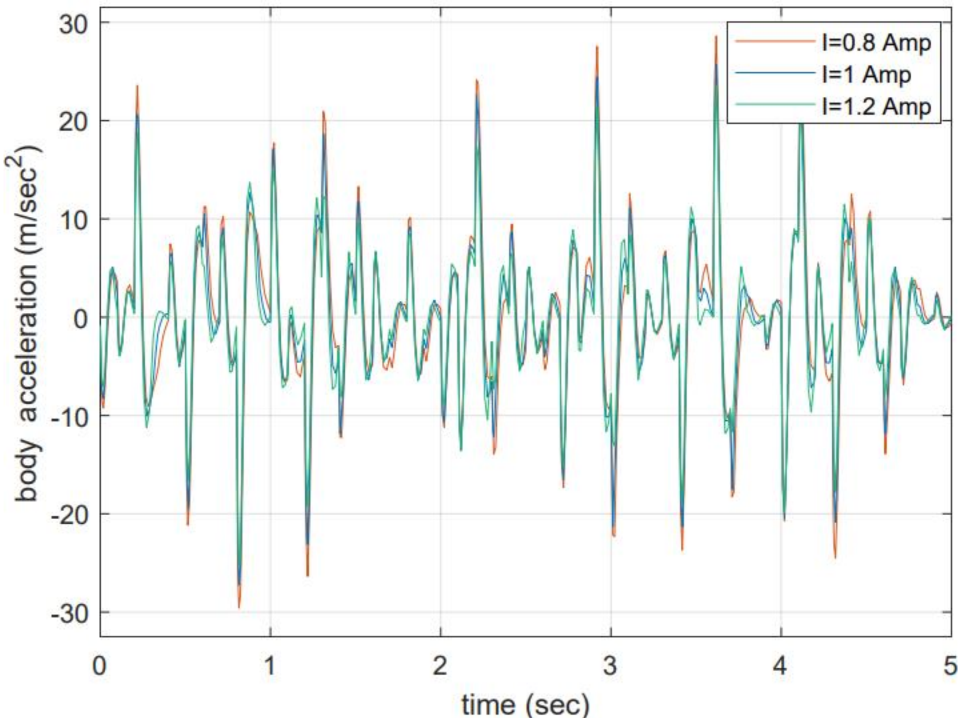
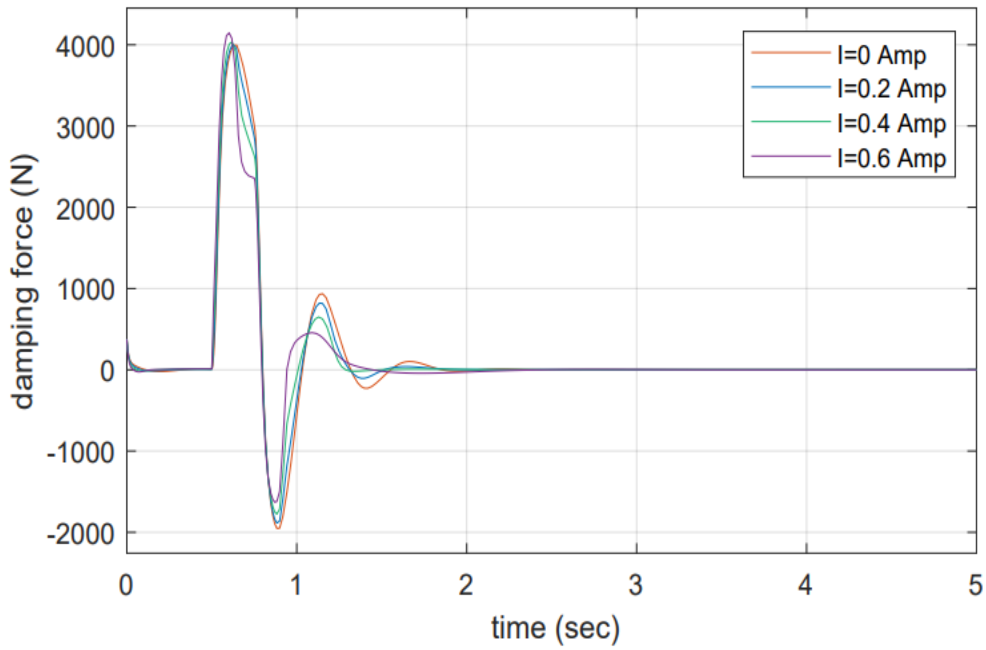


Figure 4.8.c Acceleration of car body under random excitation with the different input current in PID controller case.

Figure 4.8.b represent the effect of current on acceleration of car body with PID controller, Indicated the value of acceleration of body for the suspension system with variable value of current (0, 0.4, 0.8, 1.2 Amp), can be shown the effect of increase of current on the behavior of suspension system with MR damper lead to reduce the body acceleration in bump exaction.



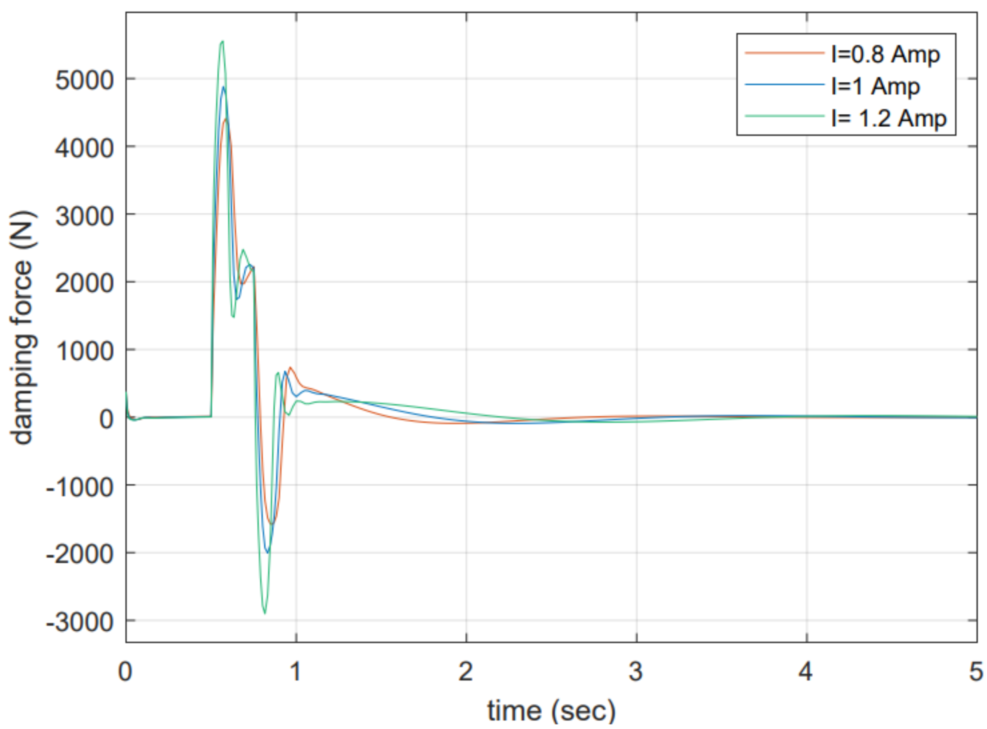
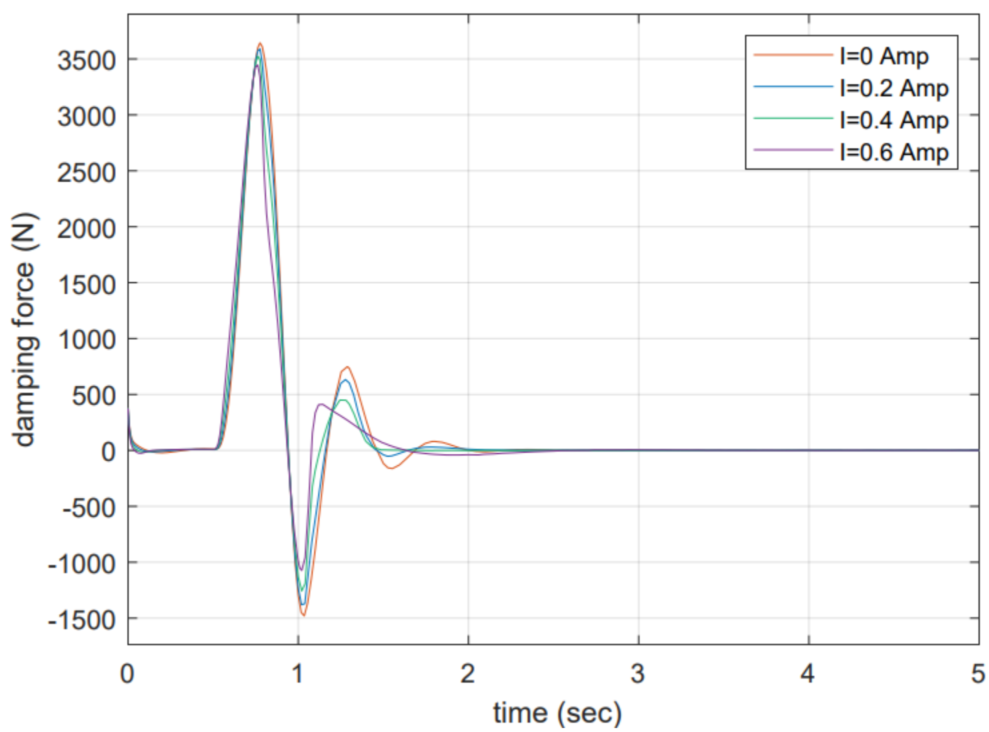


Figure 4.9.a Damping force of car body under square excitation with the different input current in PID controller

Figure 4.9.a represent the value of force generated by MR damper after supply the current with PID controller, where the force generated increase with increased the current to drag the force generated by excitation road and to decrease the car body displacement to save the suspension system and passengers from square excitation.



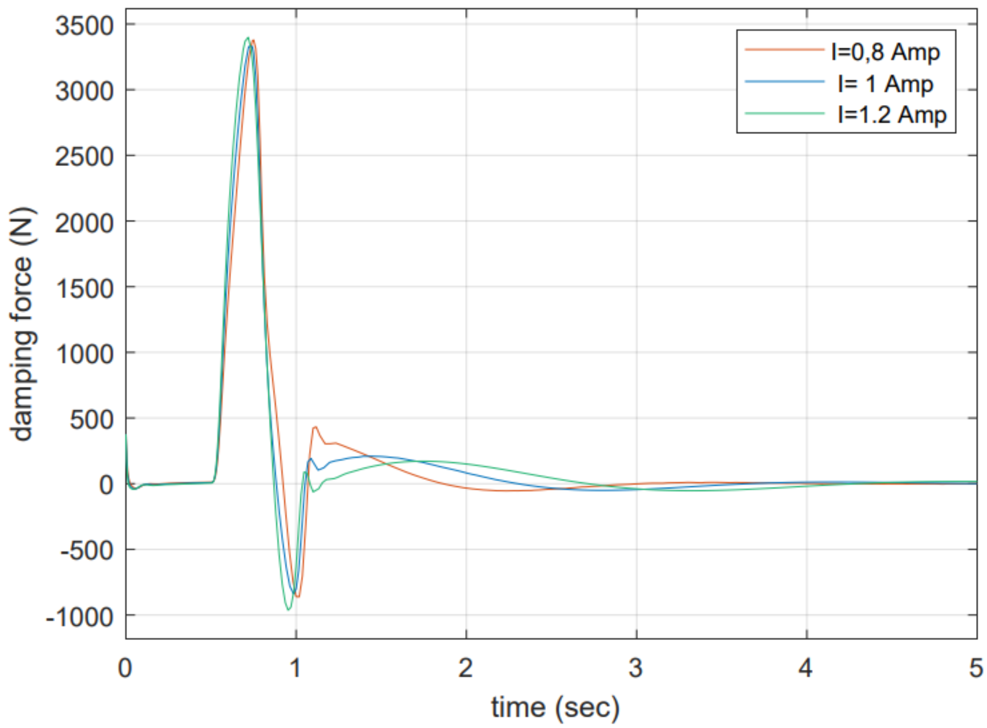
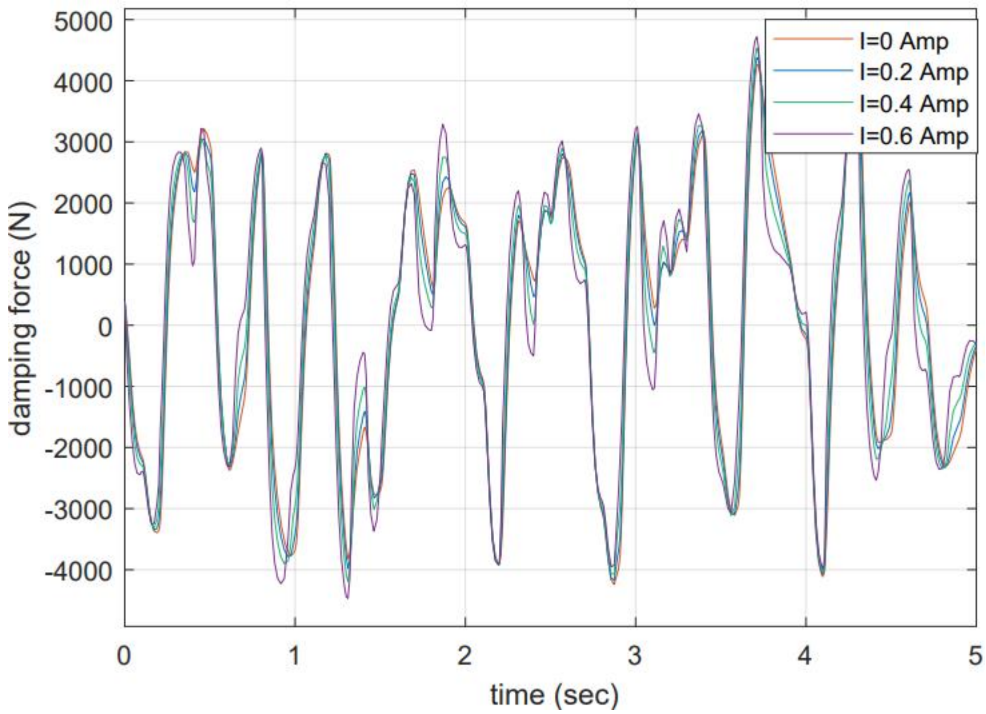


Figure 4.9.b Damping force of car body under bump excitation with the different input current in PID controller

Figure 4.9.b represent the value of force generated by MR damper after supply the current with PID controller, where the force generated increase with increased the current to drag the force generated by excitation road and to decrease the car body displacement to save the suspension system and passengers from bump excitation.



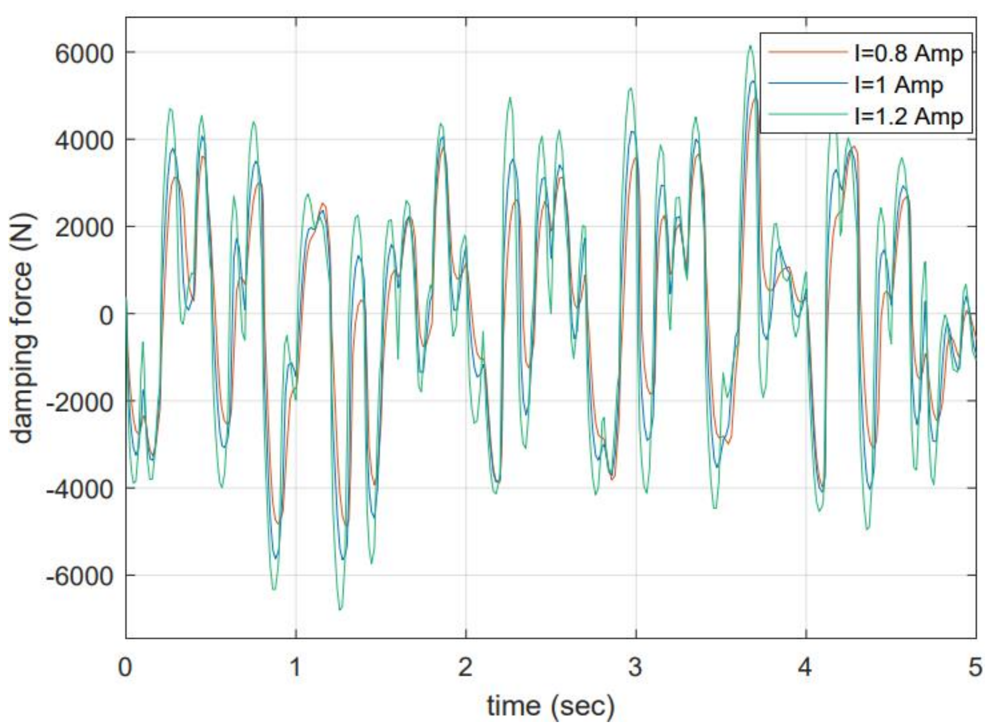
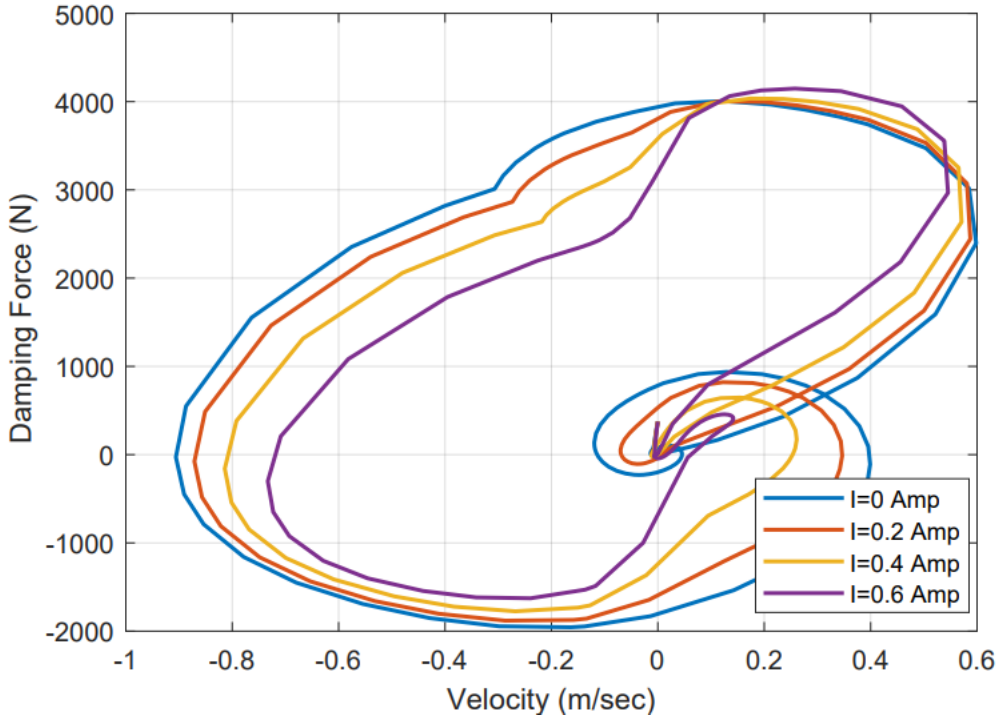


Figure 4.9.c Damping force of car body under random excitation with the different input current in PID controller

Figure 4.9.c represent the value of force generated by MR damper after supply the current with PID controller, where the force generated increase with increased the current to drag the force generated by excitation road and to decrease the car body displacement to save the suspension system and passengers from random excitation.



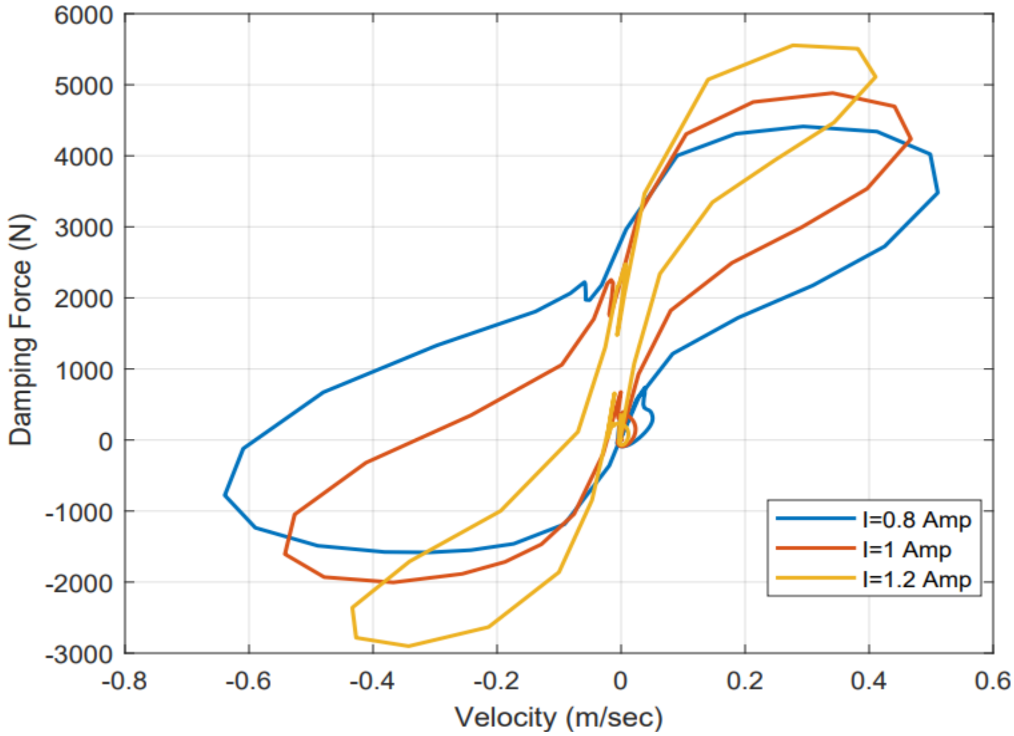


Figure 4.10.a the damping force along with damper relative velocity (histories) System behavior under square exaction in PID controller case.

Figure 4.10.a represent the effect of the PID controller on the damping force, The force-velocity hysteresis loop for the MR damper at different value of current are shown in Figure 4.4.a. It is observable that the MR damper behaves approximately as a friction damper where the friction force level can be changed by changing the applied damper current under square exaction. The hysteresis loop of force-velocity is increasing from 0A up to the 1.2A. By increasing the current, the Magneto-rheological (MR) damper generate more force, and velocity of the MR damper decreases that lead to the vibration of the body goes to the stable state.

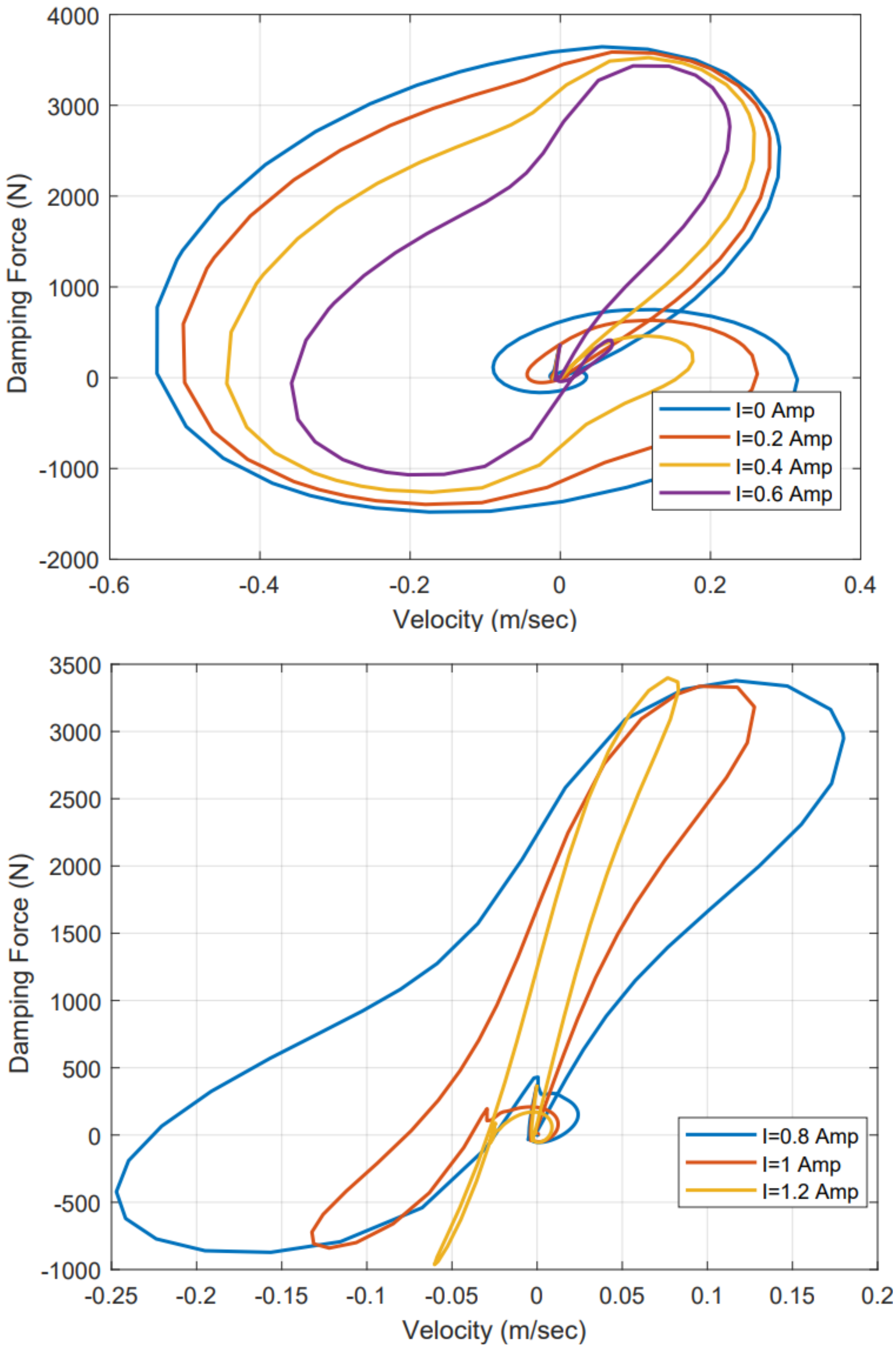
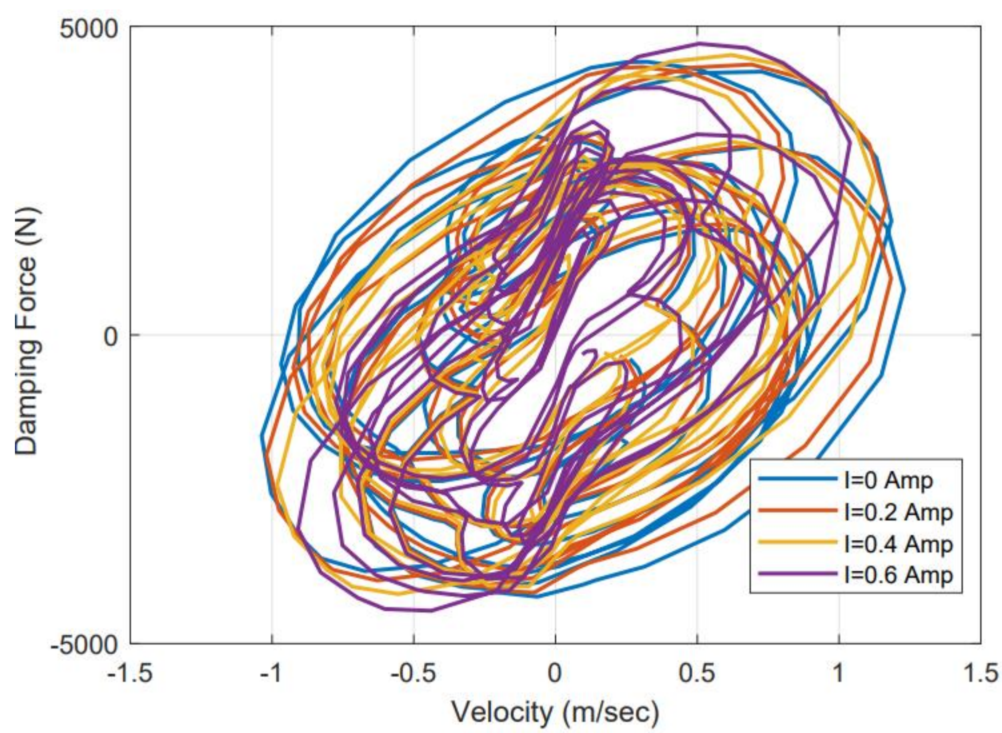


Figure 4.10.b the damping force along with damper relative velocity (histories) System behavior under bump exaction in PID controller case.

Figure 4.10.b represent the effect of the PID controller on the damping force, The force-velocity hysteresis loop for the MR damper at different value of current are shown in Figure 4.4.b. It is observable that the MR damper behaves approximately as a friction damper where the friction force level can be changed by changing the applied damper current under square exaction. The hysteresis loop of force-velocity is increasing from 0A up to the 1.2A. By increasing the current, the Magneto-rheological (MR) damper generate more force, and velocity of the MR damper decreases that lead to the vibration of the body goes to the stable state.



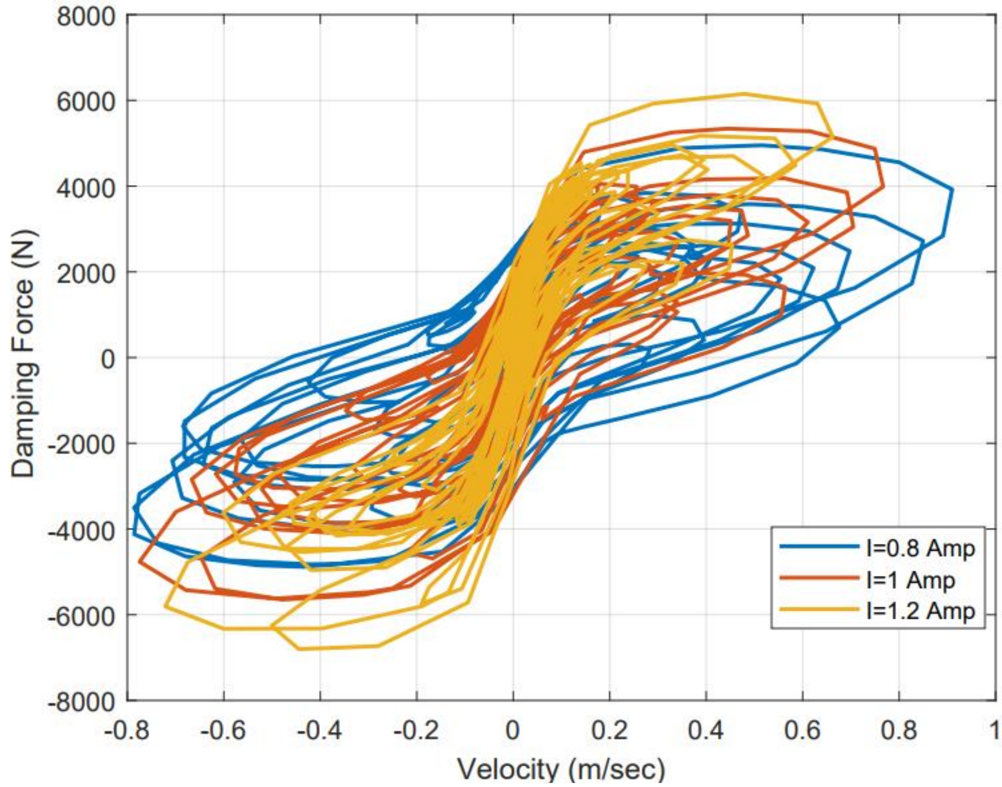


Figure 4.10.c the damping force along with damper relative velocity (histories) System behavior under random exaction in PID controller case.

Figure 4.10.b represent the effect of the PID controller on the damping force, The force-velocity hysteresis loop for the MR damper at different value of current are shown in Figure 4.4.b. It is observable that the MR damper behaves approximately as a friction damper where the friction force level can be changed by changing the applied damper current under square exaction. The hysteresis loop of force-velocity is increasing from 0A up to the 1.2A. By increasing the current, the Magneto-rheological (MR) damper generate more force, and velocity of the MR damper decreases that lead to the vibration of the body goes to the stable state.

4.3 Comparison of The Results

Compared this work with previous research such as Saad et, al [33] which use PID controller with bouc-wen model using following parameter as shown in table 4.1. The result at 1.2 Amp for time response showed that in uncontrolled case the overshoot reduced by 63.8 % in this study and 60.65% for pervious as compared with passive case. In controller case time response at 1.2 Amp the overshoot reduced by 75.5% for this study and 77.5% for pervious case as compared with passive case under random case.

Table 4.1 suspension system parameter.

Sr.No.	Parameter	Symbol	Quantities
1	Mass of vehicle body	M_s	2500 kg
2	Mass of tyre	M_u	320 kg
3	The coefficint of suspension spring	K_s	80000 N/m
4	The coefficint of tyre material	K_t	50000 N/m
5	Damping coefficent of the dampers	C_s	15020 N-s/m

Chapter Five

Conclusions and Recommendation Work

5.1 Concluded

This study is to improve the behavior of a semi-active suspension system with MR-damper. The outcomes of this study are expected to accelerate the use of these dampers in quarter-car suspension systems. This study may obtain the following conclusion.

1- In a semi-active suspension system in case of increasing current value is decreased the overshoot, and increasing damping force for different road exaction.

2- In the controller case, PID controller with increasing current value leads to reduce overshot and settling time for different road exaction. As compared with uncontroller case the PID show batter performance.

3- From previous works, it is noted the following notes

a-The semi-active suspension system with MR damper was studied under different cases and, for different applications.

b-Different control methods were used to enhance the performance of MR dampers for semi-active suspension systems. Including (PID, skyhook,…). All showed batter performance as compared with a semi-active suspension system.

c-Experiment tests were also applied in many studies and built different test rig structures to test different cases, as, compared with the simulation result.

5.2 Recommendations for Future Work

The following extents of coming research are projected to additionally assess the ideas displayed in this proposal and to upgrade their performance potential.

1- Built a quarter car test rig for experimental testing of semi-active suspension system with MP damper.

2- An optimal control algorithm for designed non-linear suspension system can give more robust and accurate behavior. Optimal control theories such skyhook, H_∞, will result into better dynamic control for sprung mass displacement minimization against the road surface disturbance

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الخلاصة

كان التعليق أهم نظام فرعي للمركبة يُنظر إليه على أنه نظام. تتأثر راحة القيادة وأداء التعامل مع السيارة بتصميم التعليق. كانت تكنولوجيا السيارات تدمج باستمرار التطورات على مدى العقود القليلة الماضية لتزويد المستخدمين النهائيين براحة أفضل في القيادة. تم تحقيق التحسين متعدد الأهداف لمثبط MR مع وظيفة موضوعية لزيادة قوة التخميد الناتجة عن مثبط MR مع وظيفة القيد الهندسي في هذا البحث باستخدام تقنية تحسين البحث عن الأنماط.

مخمدات MR هي أجهزة شبه نشطة يمكنها تحقيق قوى يمكن التحكم بها على مستوى عالٍ وأداء أفضل من الأنظمة السلبية مع الحفاظ على موثوقية أداء الأجهزة المنفعلة ومتطلبات طاقة منخفضة. ومع ذلك ، فإن السلوك غير الخطي للغاية لـ MR Damper المرتبط بطبيعته شبه النشطة يزيد من مستوى تعقيد النمذجة العددية للسلوك ، خاصة عند استخدام النماذج البارامترية.

يركز البحث على تصميم ونمذجة ومحاكاة نظام التعليق شبه النشط باستخدام النظرية غير الخطية للمثبط المغنطيسي الريولوجي (MR) مع مراعاة سلوك التباطؤ لنموذج ربع السيارة. يعتمد البحث على افتراض أن كل عجلة تواجه نفس إثارة الاضطراب. يتم تحليل التباطؤ باستخدام نماذج بينغهام. يشمل البحث محاكاة نماذج المبني للمجهول وبنغهام. يتم تحليل النظام المصمم على شكل نماذج للطرق الثلاثة ، ويتم تنفيذ الدراسة المقارنة للنماذج للحصول على أعلى راحة مع تقليل التجاوز ووقت الاستقرار لكتلة نوابض السيارة. يعتبر طراز Bingham أكثر راحة من التعليق السلبي من حيث متطلبات قوة التخميد ولديه تجاوز أقل ووقت استقرار أقل.

تم تصميم نظام المثبط لنظام التعليق على شكل مثبط مائع مغناطيسي ريولوجي. من أجل التبسيط ، تم استخدام طراز ربع سيارة لوصف نظام التعليق. تم اشتقاق النموذج الديناميكي لربع السيارة قيد الدراسة باستخدام قانون نيوتن الثاني للحركة. يتم التعبير عن قوة التباطؤ الديناميكي لمخمدات السائل المغناطيسية الريولوجية في هذه المقالة باستخدام نموذج بينغهام. تم إجراء العديد من عمليات المحاكاة لنظام التعليق لتطبيق السيارة ؛ تم تطبيق الإثارات المختلفة (مربعه ، مطب ، وعشوائية) كملف جانبي للطريق لإظهار تأثير القوة المتولدة في المثبط لكل إشارة إثارة. ايضا تم استخدام نظام سيطرة PID ودارسة تاثيرها. أظهرت نتائج المحاكاة المقدمة فاعلية النموذج المشتق لطراز ربع السيارة متضمن مثبط السوائل MR

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رسالة

مقدمة إلى جامعة بابل / كلية الهندسة وهي جزء من متطلبات نيل شهادة الماجستير في الهندسة/ الهندسة الميكانيكية (تطبيقي)

من قبل الطالب

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بكالوريوس هندسة ميكانيك

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بأشراف

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