MODELING AND SIMULATION OF SKYHOOK CONTROLLER FOR ACTIVE SUSPENSION SYSTEM

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I certify that the project entitled “Modeling and Simulation of Skyhook Controller for Active Suspension System” is written by Mohd Ezee bin Zaidee Ee. I have examined the final copy of this project and in our opinion; it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. I herewith recommend that it be accepted in partial fulfillment of the requirements for the degree of Bachelor of Mechanical Engineering with Automotive Engineering.

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Report submitted in partial fulfillment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Automotive Engineering

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SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive Engineering.

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I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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DEDICATION

This work is dedicated to all those who have inspired me throughout my life, with special thanks to my family and friends

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I am greatly indebted to ALLAH SWT on His blessing for making this project successful.

I would like to express my sincere gratitude to my supervisor Dr. Gigih Priyandoko for his brilliant ideas, invaluable guidance, continuous encouragement and constant support in making this project possible. I appreciate his consistent support from the first day I start this project to these concluding moments

My sincere thanks go to all my colleagues under the same supervision for sharing the ideas and information, advice and guidance throughout this project. My sincere thanks also go to all members of the staff of the Mechanical Engineering Department, UMP, who helped me in many ways and made my stay at UMP pleasant and unforgettable.

I acknowledge my sincere indebtedness and gratitude to my parents and for their love, dream and sacrifice throughout my life. I acknowledge the sincerity of my brother and sister who consistently encouraged me to work hard for this project.

ABSTRACT

This paper presents modeling and simulation force tracking control of a hydraulic actuator applied in a quarter car of the active suspension system using skyhook control system. The controller structure of the active suspension system was decomposed into two loops namely outer loop and inner loop controllers. Outer loop controller is used to calculate the optimum target force to reject the effects of road disturbances by using skyhook control and proportional-integral-derivative (PID) control system, while, the inner loop controller is used to keep the actual force close to this desired force. The results of the study show that the inner loop controller is able to track well the target force ranging from sinusoidal, square, saw-tooth and step functions of target force. The performance of outer loop controller also shows significant improvement in terms of body acceleration, body displacement and tire displacement, and spring deflection as compared to the passive suspension system.

ABSTRAK

Tesis ini membentangkan pemodelan dan simulasi pengesan kawalan kuasa dari aktuator hidraulik yang diaplikasikan dalam satu perempat struktur model kereta suspensi aktif menggunakan sistem kawalan “skyhook”. Struktur kawalan dari sistem suspensi aktif dibahagikan kepada dua bahagian iaitu kawalan pusingan luar dan kawalan pusingan dalam. Kawalan pusingan luar digunakan untuk mengira kuasa optimum yang disasarkan untuk menyingkirkan kesan daripada gangguan permukaan jalan dengan menggunakan sistem kawalan “skyhook” dan juga sistem kawalan “proportional-integral-derivative (PID)”, sementara kawalan pusingan dalam digunakan untuk memastikan kuasa sebenar menghampiri kuasa yang dikehendaki. Keputusan yang diperolehi berdasarkan analisis mendapati, sistem kawalan dalam mempunyai kebolehan untuk mengesan kepelbagaian jenis kuasa disasarkan merangkumi fungsi “sinusoidal”, “square”, “saw-tooth”, dan “step”. Begitu juga dengan prestasi sistem kawalan luar menunjukkan penambahbaikan dalam aspek pecutan badan, pemindahan badan, pemindahan tayar, dan pesongan pegas seperti yang dibandingkan dengan sistem suspensi pasif.

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LIST OF SYMBOLS

A_p	piston area, m^2
C_d	discharge coefficient
C_s	damping constant, N.s/m
C_{tm}	leakage coefficient
F_a	actuator force, N
K_s	spring stiffness, N/m
K_t	tire stiffness, N/m
M_s	sprung mass, kg
M_u	unsprung mass, kg
P_L	pressure induced by load
P_s	supply pressure, kN/m
Z_r	road profile
Z_s	sprung mass displacement
Z_u	unsprung mass displacement
$\dot{Z}_s$	vertical velocity of the body
$\dot{Z}_u$	vertical velocity of the wheel
u_1	spool valve position
u_2	bypass valve area
α	hydraulic coefficient, N/m^5
ρ	specific gravity of hydraulic fluid

τ	time constant
w	spool valve width, m

LIST OF ABBREVIATIONS

AC	Automatic Control
PI	Proportional Integral
PID	Proportional Integral Derivative

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

A car suspension system is the mechanism that physically separates the car body from the wheels of the car. The performance of the suspension system has been greatly increased due to increasing vehicle capabilities. Appleyard and Wellstead (1995) have proposed several performance characteristics to be considered in order to achieve a good suspension system. These characteristics deal with the regulation of body movement, the regulation of suspension movement and the force distribution. Ideally the suspension should isolate the body from road disturbances and inertial disturbances associated with cornering and braking or acceleration. The suspension must also be able to minimize the vertical force transmitted to the passengers for their comfort. This could be achieved by minimizing the vertical car body acceleration.

The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system. A passive suspension system is a conventional suspension system consists of a non-controlled spring and shock-absorbing damper. The commercial vehicles today use passive suspension system as means to control the dynamics of a vehicle's vertical motion as well as pitch and roll. Passive indicates that the suspension elements cannot supply energy to the suspension system. The suspension spring and damper do not provide energy to the suspension system and control only the motion of the car body and wheel by limiting the suspension velocity according to the rate determined by the designer. Hence, the performance of a passive suspension system is variable subject to the road profiles.

The semi-active suspension has the same elements but the damper has two or more selectable damping rate. In early semi-active suspension system, the regulating of the damping force can be achieved by utilizing the controlled dampers under closed loop control, and such is only capable of dissipating energy (Williams, 1994). Two types of dampers are used in the semi- active suspension namely the two state dampers and the continuous variable dampers. The disadvantage of these dampers is difficulties to find devices that are capable in generating a high force at low velocities and a low force at high velocities, and be able to move rapidly between the two.

An active suspension is one in which the passive components are augmented by hydraulic actuators that supply additional force. Active suspensions differ from the conventional passive suspensions in their ability to inject energy into the system, as well as store and dissipate it. The active suspension is characterized by the hydraulic actuator that placed in parallel with the damper and the spring. Since the hydraulic actuator connects the unsprung mass to the body, it can control both the wheel hop motion as well as the body motion. Thus, the active suspension now can improve both the ride comfort and ride handling simultaneously.

Although various control laws have been proposed to control the active suspension system, the methods were successful applied in computer simulations based only but not in real applications. Therefore, a real active suspension system is needed to implement and test the developed control strategy. A quarter car models are chosen as an initial model of controlling the active suspension system due to the simplicity of the model. Modeling of the quarter car suspension as well as the non-linear hydraulic actuator including its force tracking controller for an active suspension system is investigated in this study.

1.2 PROBLEM STATEMENT

The statement of the problem of this project is expressed as follow:

To model and simulate skyhook controller for active suspension system that can improved car performance on various condition of road profile. For an active suspension hydraulic actuators is apply to supply additional force to the system. In this case, the control systems need to be develop and the force that needs to be injected to the hydraulic need to be determined.

1.3 OBJECTIVES

The objectives of this project are as follows:

i. To develop hydraulic model.
ii. To develop force tracking controller.
iii. To develop skyhook controller to an active quarter car suspension using hydraulic actuator.

1.4 PROJECT SCOPE

This project is about modeling and simulation of skyhook controller for active suspension system. The quarter car modeling was based on passive suspension system. MATLAB software is being used in purpose to develop program for modeling and analyzing the system controller created. In this software, block diagram will be form based on the required equation using SIMULINK and then being analyzed by giving variable target value to get the desired force match with the various input from road profiles.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system and/or control bandwidth (Appleyard and Wellstead, 1995). A passive suspension system is a conventional suspension system consists of a non-controlled spring and shock-absorbing damper as shown in Figure 2.1. The semi-active suspension has the same elements but the damper has two or more selectable damping rate as shown in Figure 2.2. An active suspension is one in which the passive components are augmented by actuators that supply additional force as shown in Figure 2.3.

2.2 PASSIVE SUSPENSION SYSTEM

The commercial vehicles today use passive suspension system to control the dynamics of a vehicle's vertical motion as well as pitch and roll. Passive indicates that the suspension elements cannot supply energy to the suspension system. The passive suspension system controls the motion of the body and wheel by limiting their relative velocities to a rate that gives the desired ride characteristics. This is achieved by using some type of damping element placed between the body and the wheels of the vehicle, such as hydraulic shock absorber. Properties of the conventional shock absorber establish the tradeoff between minimizing the body vertical acceleration and maintaining good tire-road contact force. These parameters are coupled. That is, for a comfortable ride, it is desirable to limit the body acceleration by using a soft absorber, but this allows more variation in the tire-road contact force that in turn reduces the handling performance. Also, the suspension travel, commonly called the suspension displacement, limits allowable deflection, which in turn limits the amount of relative velocity of the absorber that can be permitted. By comparison, it is desirable to reduce the relative velocity to improve handling by designing a stiffer or higher rate shock absorber. This stiffness decreases the ride quality performance at the same time increases the body acceleration, detract what is considered being good ride characteristics.

An early design for automobile suspension systems focused on unconstrained optimizations for passive suspension system which indicate the desirability of low suspension stiffness, reduced unsprung mass, and an optimum damping ratio for the best controllability (Thompson, 1971). Thus the passive suspension systems, which approach optimal characteristics, had offered an attractive choice for a vehicle suspension system and had been widely used for car. However, the suspension spring and damper do not provide energy to the suspension system and control only the motion of the car body and wheel by limiting the suspension velocity according to the rate determined by the designer. Hence, the performance of a passive suspension system is variable subject to the road profiles.

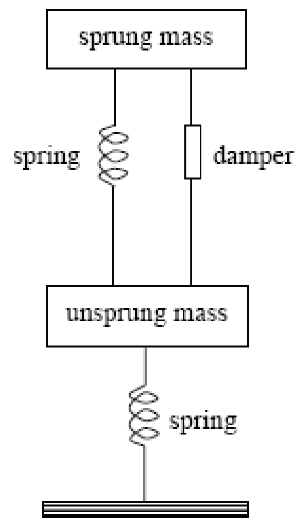


Figure 2.1: The passive suspension system

Source: Thompson (1971)

2.3 SEMI-ACTIVE SUSPENSION SYSTEM

In early semi-active suspension system, the regulating of the damper force can be achieved by utilizing the controlled dampers under closed loop control, and such is only capable of dissipating energy (Williams, 1994). Two types of dampers are used in the semi-active suspension namely the two state dampers and the continuous variable dampers.

The two state dampers switched rapidly between states under closed-loop control. In order to damp the body motion, it is necessary to apply a force that is proportional to the body velocity. Therefore, when the body velocity is in the same direction as the damper velocity, the damper is switched to the high state. When the body velocity is in the opposite direction to the damper velocity, it is switched to the low state as the damper is transmitting the input force rather than dissipating energy. The disadvantage of this system is that while it controls the body frequencies effectively, the rapid switching, particularly when there are high velocities across the dampers, generates high-frequency harmonics which makes the suspension feel harsh, and leads to the generation of unacceptable noise.

The continuous variable dampers have a characteristic that can be rapidly varied over a wide range. When the body velocity and damper velocity are in the same direction, the damper force is controlled to emulate the skyhook damper. When they are in the opposite directions, the damper is switched to its lower rate, this being the closest it can get to the ideal skyhook force. The disadvantage of the continuous variable damper is that it is difficult to find devices that are capable in generating a high force at low velocities and a low force at high velocities, and be able to move rapidly between the two.

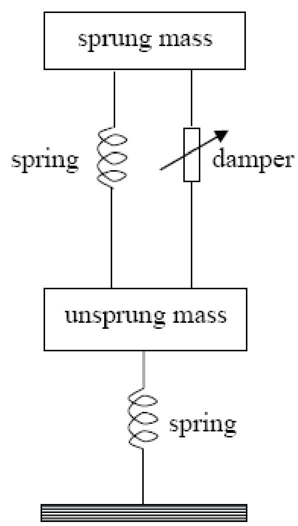
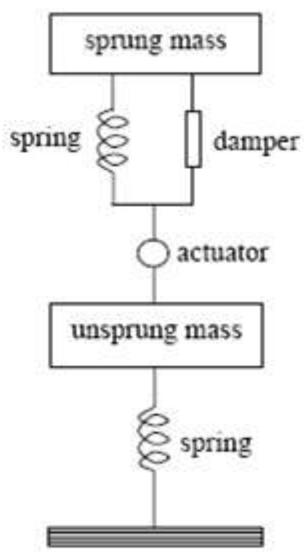


Figure 2.2: The semi-active suspension system

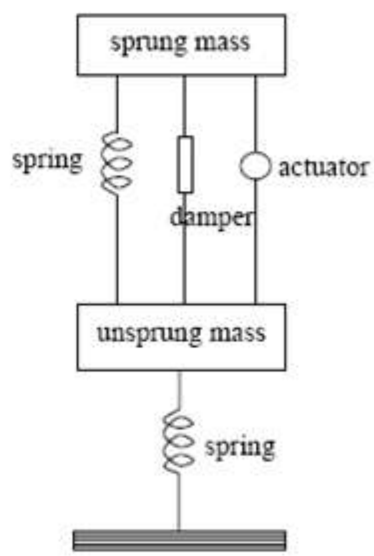
Source: Williams (1994)

2.4 ACTIVE SUSPENSION SYSTEM

Active suspensions differ from the conventional passive suspensions in their ability to inject energy into the system, as well as store and dissipate it. (Crolla, 1988) has divided the active suspensions into two categories; the low-bandwidth or soft active suspension and the high-bandwidth or stiff active suspension. Low bandwidth or soft active suspensions are characterized by an actuator that is in series with a damper and the spring as shown in Figure 2.3(a). Wheel hop motion is controlled passively by the damper, so that the active function of the suspension can be restricted to body motion. Therefore, such type of suspension can only improve the ride comfort. A high-bandwidth or stiff active suspension is characterized by an actuator placed in parallel with the damper and the spring as illustrated in Figure 2.3(b). Since the actuator connects the unsprung mass to the body, it can control both the wheel hop motion as well as the body motion. The high-bandwidth active suspension now can improve both the ride comfort and ride handling simultaneously. Therefore, almost all studies on the active suspension system utilized the high-bandwidth type. Various types of active suspension model are reported in the literature either modeled linearly (used most) or non-linear; examples are Macpherson strut suspension system (Al-Holou *et al.*, 1999, Hong *et al*., 2002).



(a) Low bandwidth



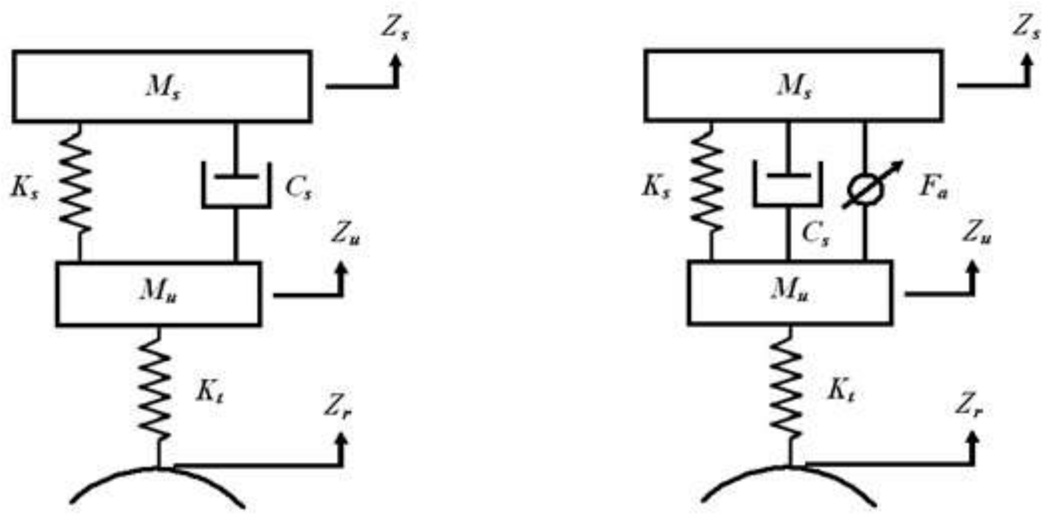
(b) High bandwidth

Figure 2.3: Types of active suspension system

Source: Crolla (1988)

2.5 QUARTER CAR MODEL

The vehicle model considered in this study is a quarter car models. The quarter car model for passive suspension system consists of one-fourth of the body mass, suspension components and one wheel as shown in Figure 2.4(a). The quarter car model for active suspension system, where the hydraulic actuator is installed in parallel with the spring, is shown in Figure 2.4(b).



(a) Passive suspension system (b) Active suspension system

Figure 2.4: The quarter car model

The assumptions of a quarter car modeling are as follows: the tire is modeled as a linear spring without damping, there is no rotational motion in wheel and body, the behavior of spring and damper are linear, the tire is always in contact with the road surface and effect of friction is neglected so that the residual structural damping is not considered into vehicle modeling. The equations of motion for the sprung and unsprung masses of the passive quarter car model are given by:

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) = 0$$
$$M_u\ddot{Z}_u + K_t(Z_u - Z_r) + K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) = 0 \qquad (2.1)$$

Whereas, the equations of motion for the sprung and unsprung masses of the active quarter-car model are given by:

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) + F_a = 0$$
$$M_u\ddot{Z}_u + K_t(Z_u - Z_r) + K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) - F_a = 0 \qquad (2.2)$$

Where,

M_s	=	sprung mass
K_s	=	spring stiffness
K_t	=	tire stiffness
M_u	=	unsprung mass
C_s	=	damping constant
Z_r	=	road profile
Z_s	=	sprung mass displacement
Z_u	=	unsprung mass displacement
F_a	=	actuator force

Due to the tire stiffness, vertical force acting on the contact point between tire and the road will be created when the tire hits a certain road profile. Then, the vertical force is transferred to the wheel resulting in vertical acceleration of the wheel. Part of the vertical force is damped out by the suspension elements, whereas, the rest is transferred to the vehicle body via the suspension elements. The vehicle body will move vertically in response to the vertical force of the suspension elements. The performance criteria of the suspension system to be investigated in this study are body acceleration ($\ddot{Z}_s$), body displacement (Z_s), suspension working space ($Z_u - Z_s$) and wheel displacement (Z_u).

Performance of the suspension system is characterized by the ability of the suspension system in reducing those four performance criteria effectively.

2.6 INNER LOOP CONTROLLER

The structure of force tracking control of hydraulic actuator is shown in Figure 2.5. The hydraulic actuator model take two input namely spool valve position and real time piston speed. Proportional Integral control is implemented which takes force tracking error as the input and delivers control voltage to drive the spool valve. The forcing functions are selected to represent real world situations which depending on the type of road disturbance, and may be represented by sinusoidal, saw-tooth, square, step functions and/or their combinations.

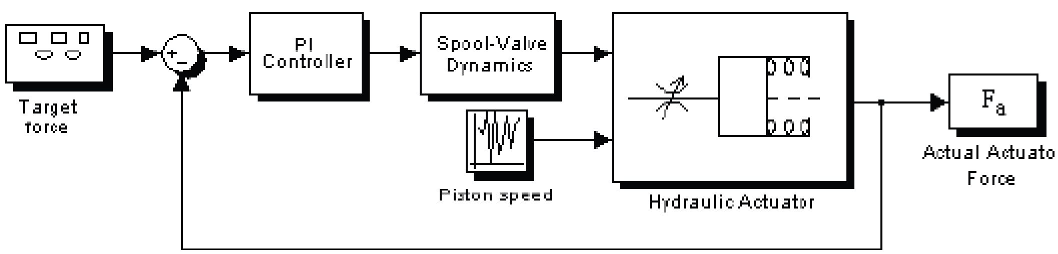


Figure 2.5: Force tracking control of hydraulic actuator

Source: Sam et al. (2005)

2.6.1 Hydraulic Actuator Model

A complete set of a hydraulic actuator consists of five main components namely electro hydraulic powered spool valve, piston cylinder, hydraulic pump, reservoir and piping system as shown in Figure 2.6. Power supply is needed to drive the hydraulic pump through AC motor and to control the spool valve position. The hydraulic pump will keep the supply pressure at the optimum level. The spool valve position will control the fluid to come in or out to the piston cylinder which determines the amount of force produced by the hydraulic actuator.

The hydraulic actuators are governed by electro hydraulic servo valve allowing for the generation of forces between the sprung and unsprung masses. The electro hydraulic system consists of an actuator, a primary power spool valve and a secondary bypass valve. As seen in Figure 2.7, the hydraulic actuator cylinder lies in a follower configuration to a critically centered electro hydraulic power spool valve with matched and symmetric orifices. Positioning of the spool u_1 directs high pressure fluid flow to either one of the cylinder chambers and connects the other chamber to the pump reservoir. This flow creates a pressure difference P_L across the piston. This pressure difference multiplied by piston area A_p is what provides the actuator force F_A for the suspension system. The derivative of F_A is give by:

$$\dot{F}_A = A_p\alpha\left[C_{d1}wu_1\sqrt{\frac{P_S - sgn(u_1)P_L}{\rho}} - C_{d2}u_2sgn(P_L)\sqrt{\frac{2P_L}{\rho}} - C_{tm}P_L - A_p\left(\dot{Z}_s - \dot{Z}_u\right)\right] \tag{2.3}$$

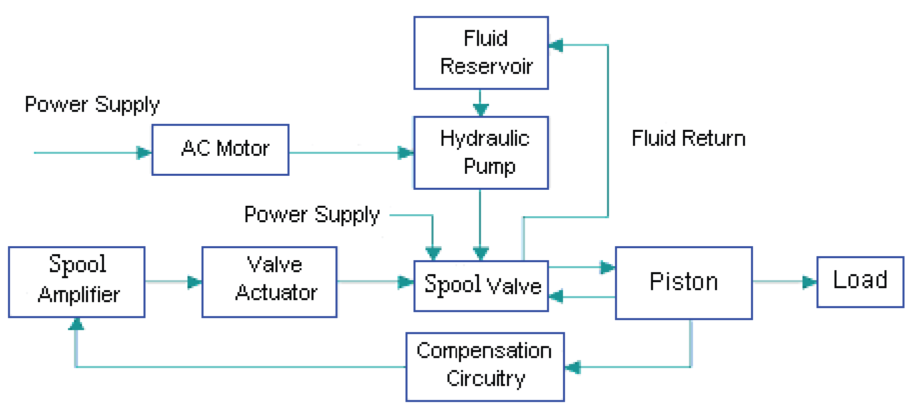


Figure 2.6: Diagram of a complete set of hydraulic actuator

Source: Donahue (2001)

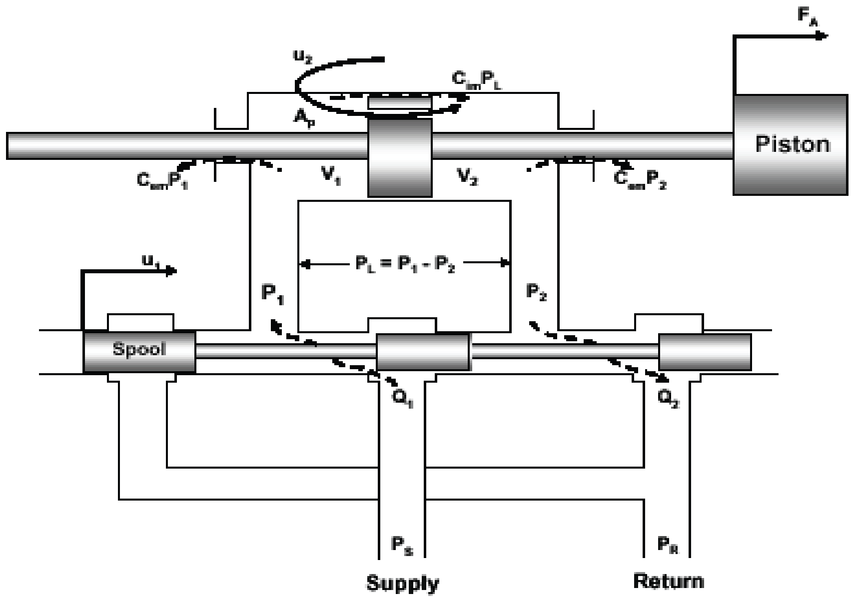


Figure 2.7: Physical schematic and variables for the hydraulic actuator

Source: Donahue (2001)

Dynamics for the hydraulic actuator valve are given as the followings: the change in force is proportional to the position of the spool with respect to center, the relative velocity of the piston, and the leakage through the piston seals. A second input u_2 may be used to bypass the piston component by connecting the piston chambers. The bypass valve u_2 could be used to reduce the energy consumed by the system. If the spool position u_1 is set to zero, the bypass valve and actuator will behave similar to a variable orifice damper. Spool valve positions u_1 and u_2 are controlled by a current-position feedback loop. The essential dynamics of the spool have been shown to resemble a first order system (Donahue, 2001) as the followings:

$$\tau\dot{u} + u = k_v \qquad (2.4)$$

The parameters of hydraulic actuator model are taken from (Donahue, 2001) as the followings:

A_p = 0.0044 m^2

α = 2.273e9 N/m^5

C_{d1} = 0.7

C_{d2} = 0.7

w = 0.008 m

P_s = 20684 kN/m^2

ρ = 3500 m/s^2

C_{tm} = 15e-12

τ = 0.001 sec^{-1}

2.7 OUTER LOOP CONTROLLER

The outer loop controller is used for disturbance rejection control to reduce unwanted vehicle's motions. The inputs of the outer loop controller are vehicle's states namely body velocity and wheel velocity, whereas the output of the outer loop controller is the target force that must be tracked by the hydraulic actuator. On the other hand, the inner loop controller is used for force tracking control of the hydraulic actuator in such a way that the force produced by the hydraulic actuator is as close as possible to the target force produced by the disturbance rejection control.

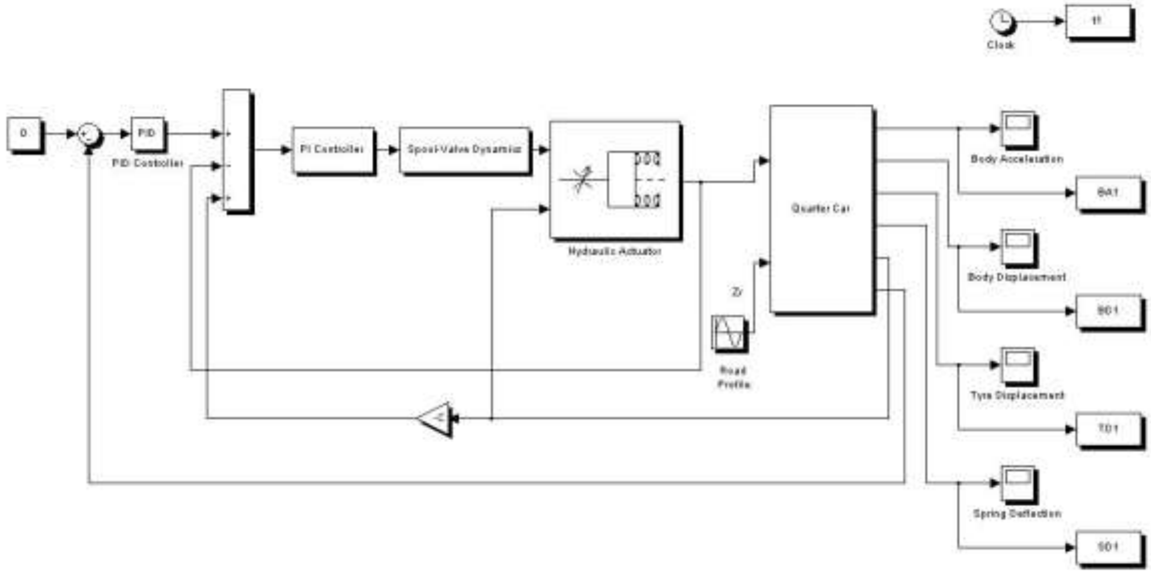


Figure 2.8: Controller structure of the active suspension system

2.7.1 Skyhook Controller

The skyhook control introduced by (Karnopp, 1995) to provide an effective solution in terms of the simplicity of the control algorithm. The fictitious force computed from the added skyhook damper is called the actuator force (F_{sky}). The force, F_{sky} of this element according to skyhook control law is:

$$F_{sky} = \begin{cases} -C_{sky}\dot{Z}_s & if \quad \dot{Z}_s(\dot{Z}_s - \dot{Z}_u) \geq 0 \\ 0 & if \quad \dot{Z}_s(\dot{Z}_s - \dot{Z}_u) < 0 \end{cases} \tag{2.5}$$

Where, C_{sky} is a constant value and determined to be approximately 3000 N/m s^{-1} in the experimental system taken from (M. Valásek and W. Kortüm, 2001).

The control strategy utilized a fictitious damper that is inserted between the sprung mass and the stationary sky as a way to suppress the vibration motion of the spring mass and as a tool to compute the desired skyhook force. The skyhook damper can reduce the resonant peak of the spring mass quite significantly and thus achieves a good ride quality. But, in order to improve both the ride quality and handling performance of a vehicle, both resonant peaks of the spring mass and the unsprung mass need to be reduced. It is known, however, that the skyhook damper alone cannot reduce both resonant peaks at the same time (Hong *et al*., 2002).

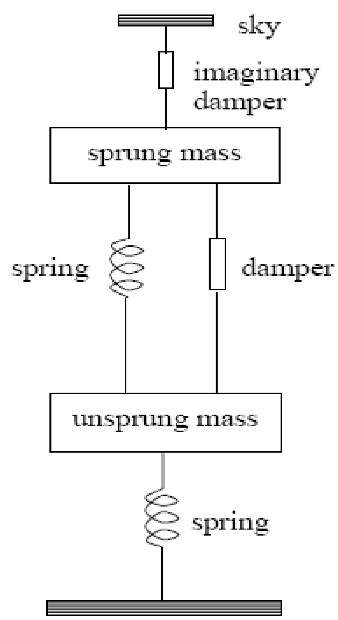


Figure 2.9: A skyhook damper

Source: Hong et al. (2002)

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

The development of an active suspension system for the vehicle is of great interest for both academic and industrial fields. The studies of active suspension system have been performed using various suspension models. In the quarter car model, the model takes into account the interaction between the quarter car body and the single wheel. Motion of the car is only in the vertical direction. Modeling of active suspension system in the early days considered that input to the active suspension system is a linear force. However due the development of new control theory, the force input to the active suspension system has been replaced by an input to control the actuator. Therefore, the dynamic of the active suspension now consist of the dynamic of the suspension and the dynamic of the hydraulic actuator.

This project involves modeling and simulation of a block diagram by using MATLAB. In designing the model, the equation and the value constant are about to be determined. Hence, this section will discuss on the modeling the block diagram based on the equation. There are three major components in the controller systems which are the quarter car, inner loop controller and outer loop controller.

3.2 FLOW CHART

In order to achieve the aim and the objective of the project, a methodology was constructed to have a proper guidance for a successful experimentation. A terminology of works and planning of the experiments conduct was shown in a flow chart to describe the detail of the project process. The flow chart is the best way to stay agile with the work in order to keep track of the work.

3.3 QUARTER CAR

The structure of the quarter is consists of one-fourth of the body mass, suspension components and one wheel. For an active suspension system, the hydraulic actuator was installed in parallel with the spring as shown in Figure 2.4(b). The modeling of the quarter car will be built in SIMULINK based on passive type suspension system referring to Equation 2.1 and Equation 2.2.

3.4 INNER LOOP CONTROLLER

The inner loop controller or force tracking controller structure consists of hydraulic actuator, dynamic spool valve, and PI controller as show in Figure 2.5. The hydraulic actuator takes two inputs namely the spool valve position and the real time piston speed. The modeling structure of force tracking controller is built in SIMULINK by referring to the Equation 2.3 and Equation 2.4. Proportional Integral control is then implemented which takes force tracking error as the input and delivers control voltage to drive the spool valve. The forcing functions are selected to represent real world situations which depending on the type of road disturbance, and may be represented by sinusoidal, saw tooth, square, step functions and/or their combinations.

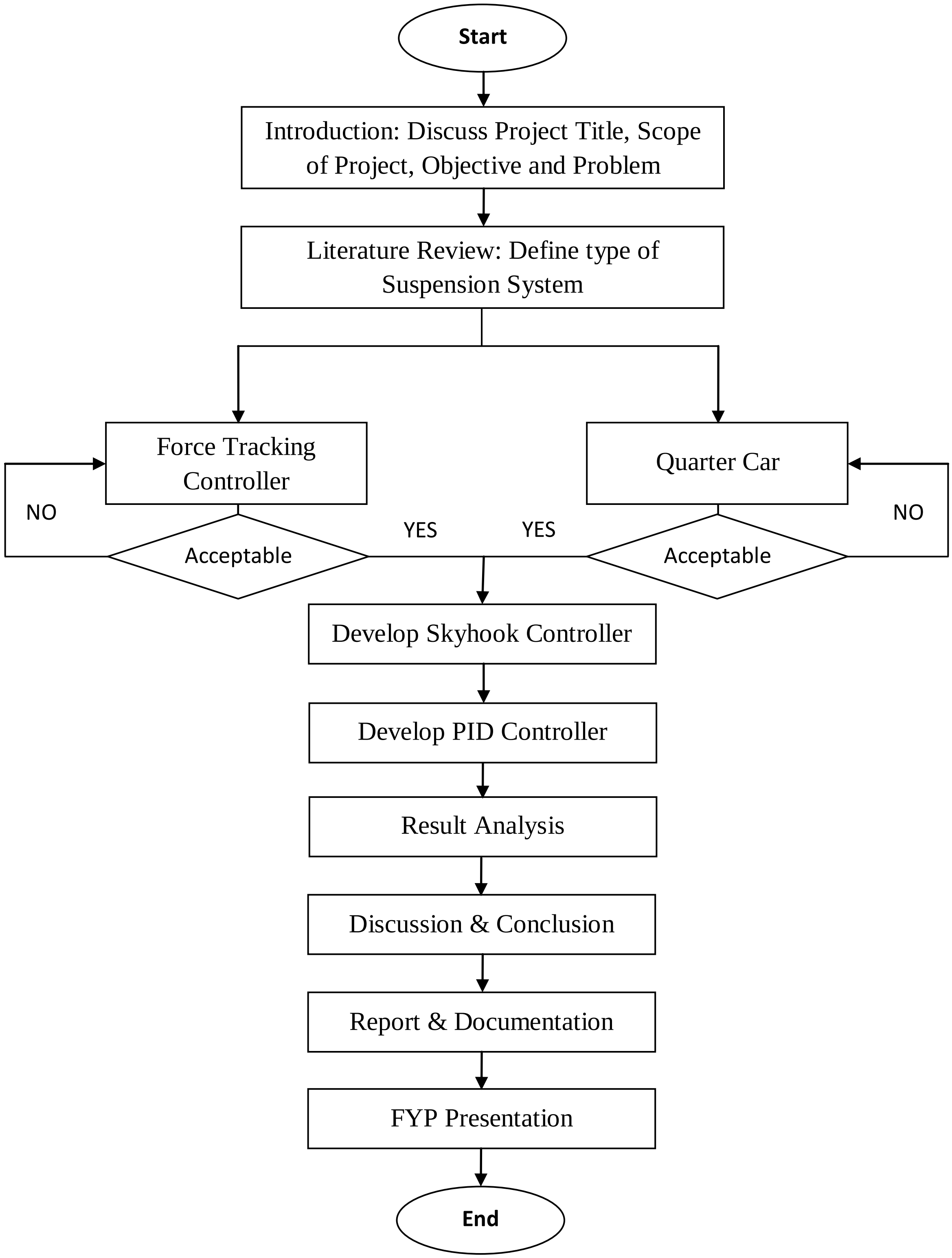


Figure 3.1: Flow chart

3.5 OUTER LOOP CONTROLLER

After complete modeling the inner loop and quarter car structure, both are then connected and the skyhook controller is implemented to the system to complete the outer loop controller. The actuator force (F_{sky}) of this element according to skyhook control law, can be obtain by referring to Equation 2.5. The outer loop functions as disturbance rejection control to reduce unwanted vehicle's motions.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 SIMULATIONS

This section contains the results of simulations studies in both inner loop and outer controllers. The parameters of inner loop controller must be optimized separately until the hydraulic actuator is able to provide the actual target force as close as possible with the predefined target force. Then, the inner loop controller is integrated with the outer loop controller. In this configuration, the inner loop controller is used to track the optimum target force produced by the outer loop controller. The performance of the inner loop controller is characterized by its ability in tracking the target force with small amount of force tracking error. Whereas, the performance of the outer loop controller is characterized by the four performance criteria namely body acceleration, body displacement, tire displacement, and suspension deflection.

4.2 SIMULATION PERFORMANCE OF FORCE TRACKING CONTROLLER

The force tracking error of the hydraulic actuator model shown in Figure 4.1 using Proportional Integral controller for sinusoidal, square, saw-tooth and step functions of the target force are shown in Figures 4.2 till Figure 4.5 respectively. This is to check the controllability of the force tracking controller for a class of continuous and discontinuous functions. In this simulation study, the parameter of proportional gain *P* is set to 1.00 and for Integral gain *I* is set to 1.00. From these figures, it can be seen clearly that the hydraulic actuator model tracks the desired force well.

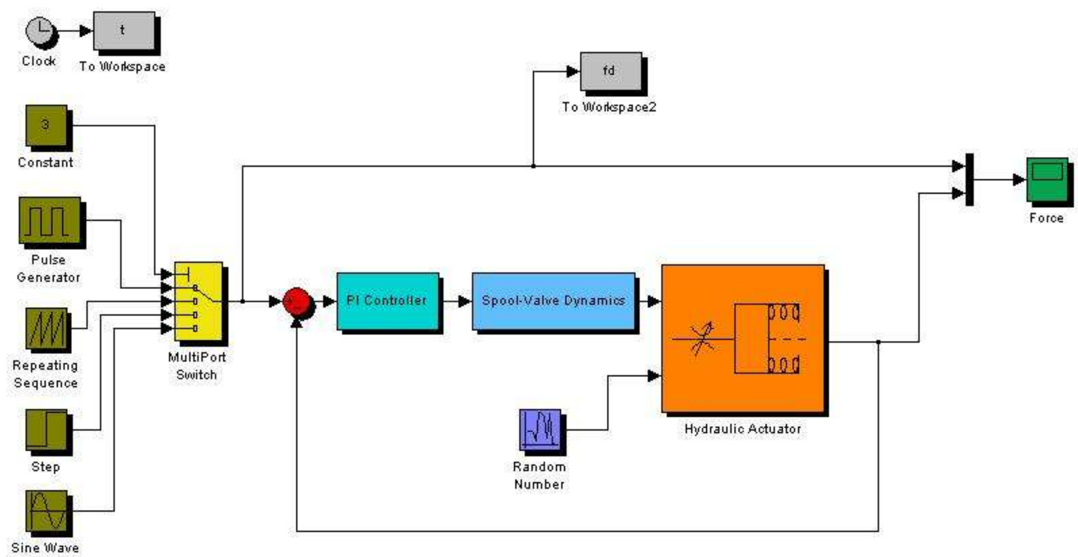
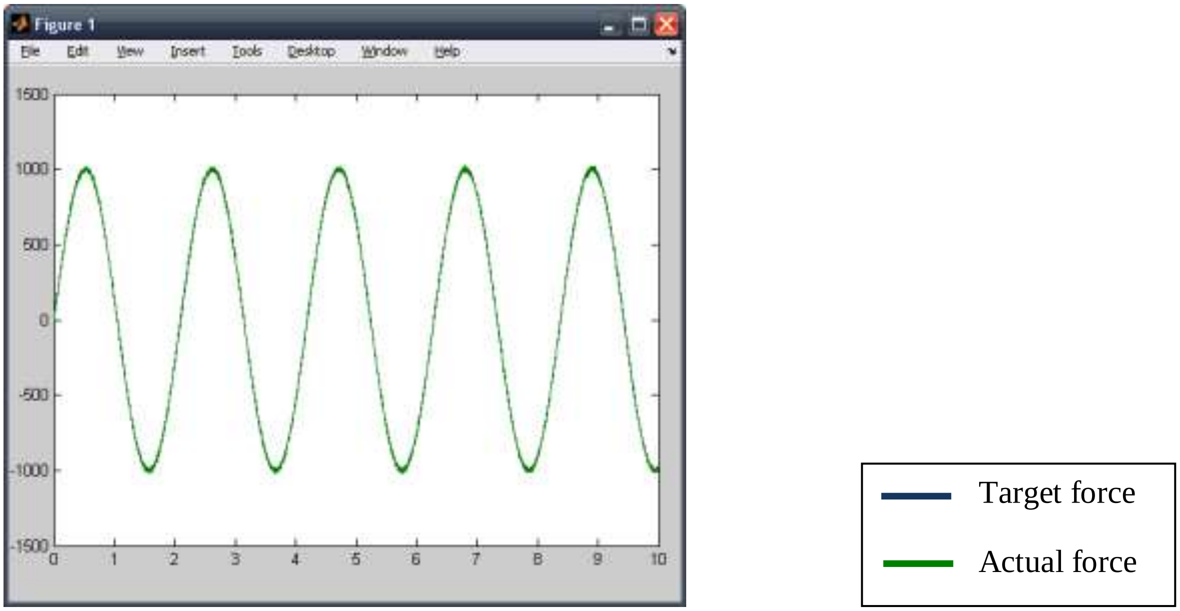
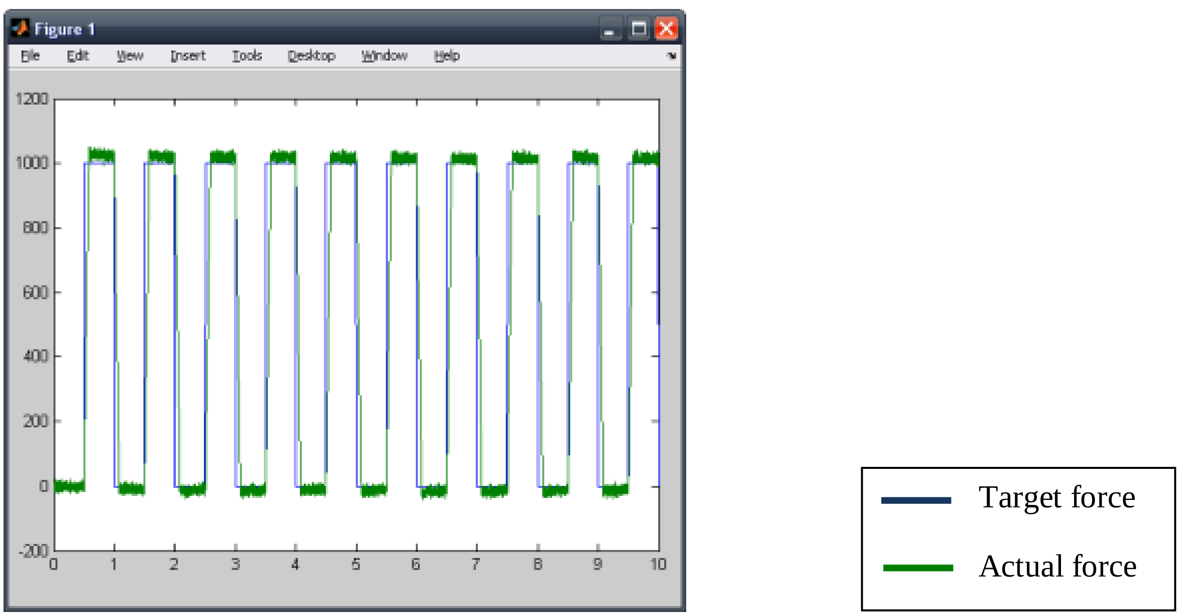


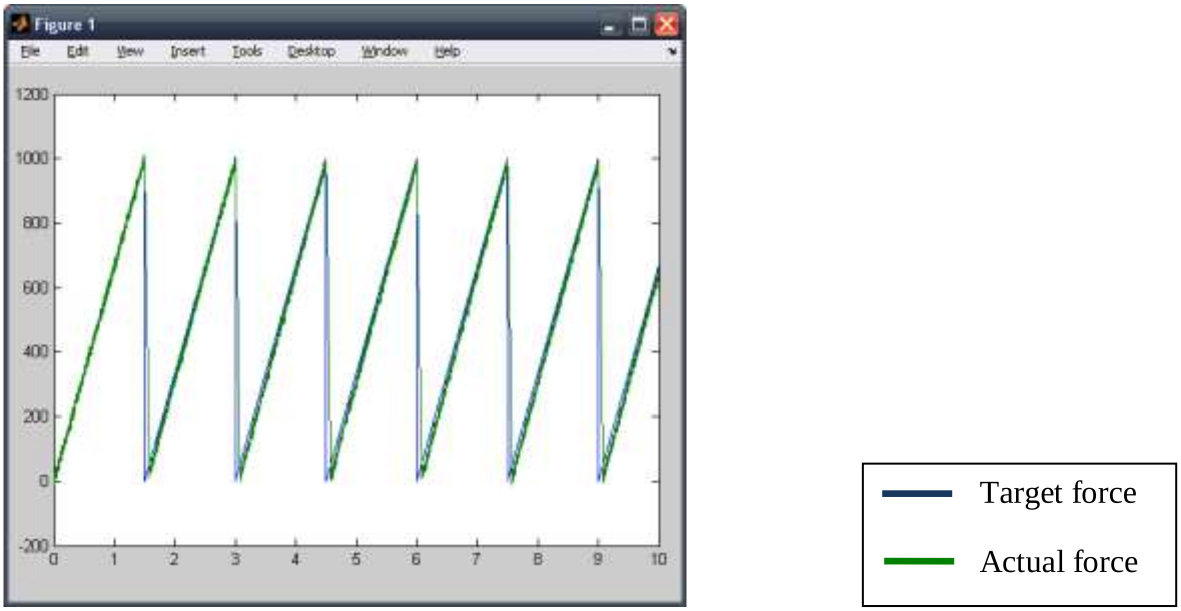
Figure 4.1: Force tracking control for hydraulic actuator



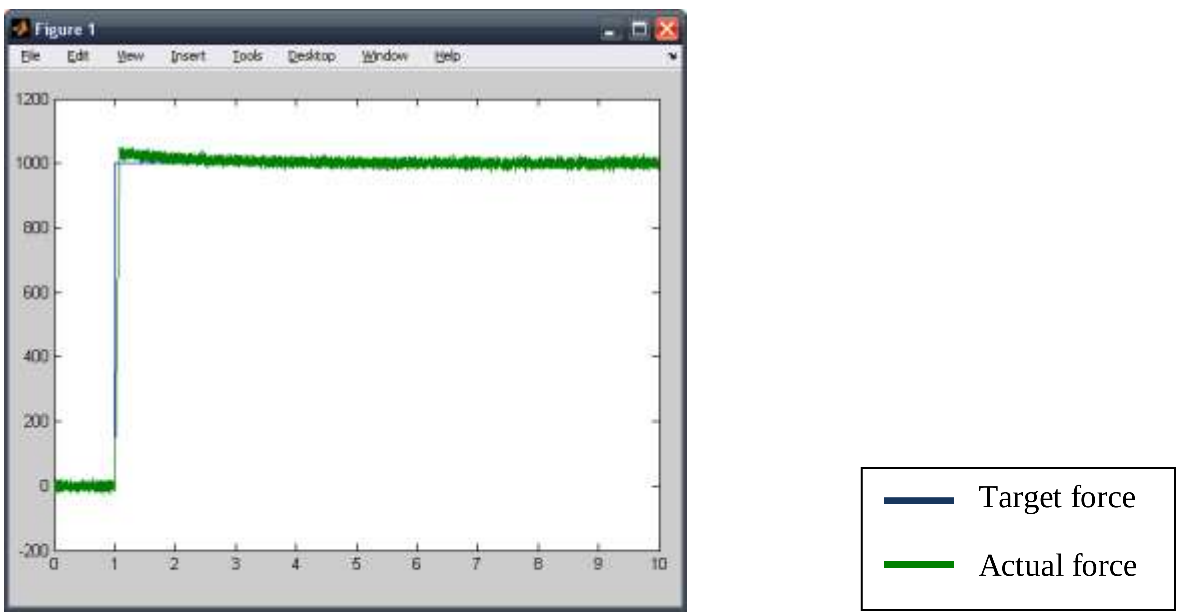
(a) Sinusoidal function



(b) Square function



(c) Saw-tooth function



(d) Step function

Figure 4.2: Force tracking performance of the target force

4.3 SIMULATION PERFORMANCE OF QUARTER CAR MODEL

The quarter car model of passive suspension system is using sine wave function as a road profile, $Z_{r,}$, by assigning the parameter for amplitude and frequency as shown in Figure 4.6. This is to find the best response for all four parameters required, which are the body acceleration, body displacement, tire displacement, and spring deflection as shown in Figure 4.7 till Figure 4.18. In this simulation study, the amplitude is set to 0.10 and for frequency is set to be 2, 3 and 4 rad/sec as reference.

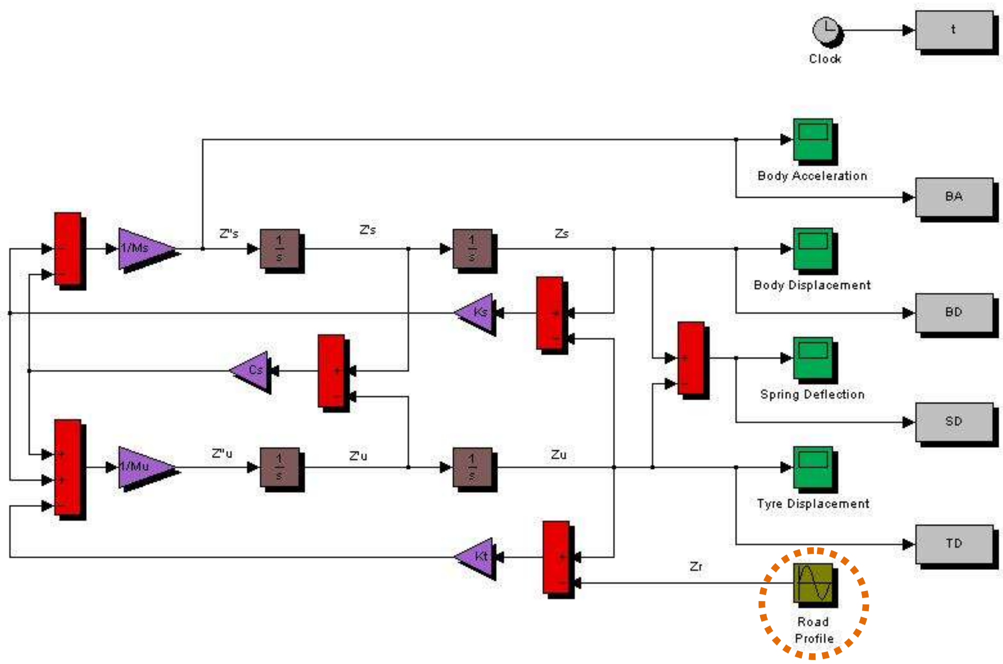
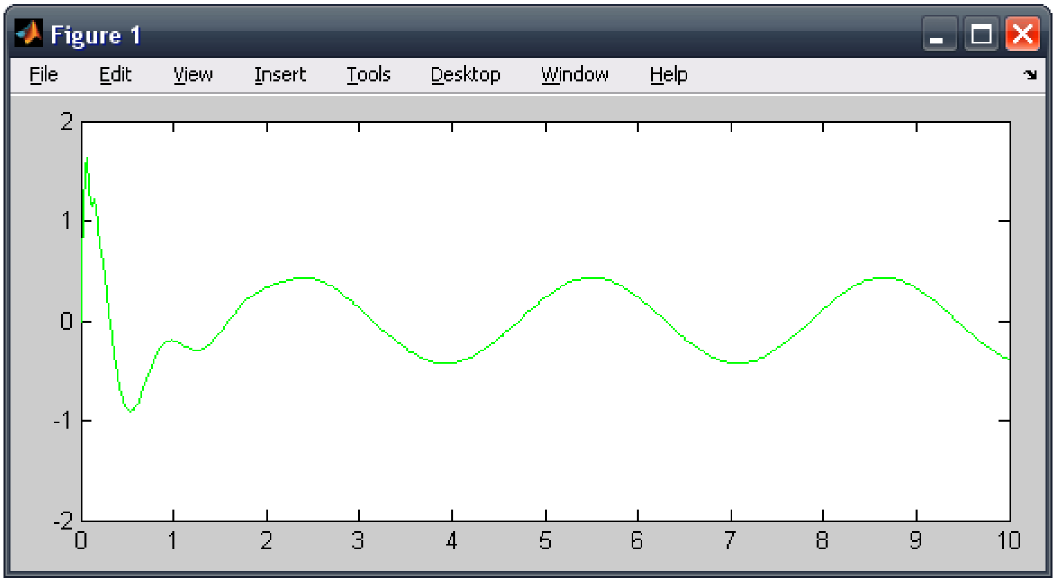
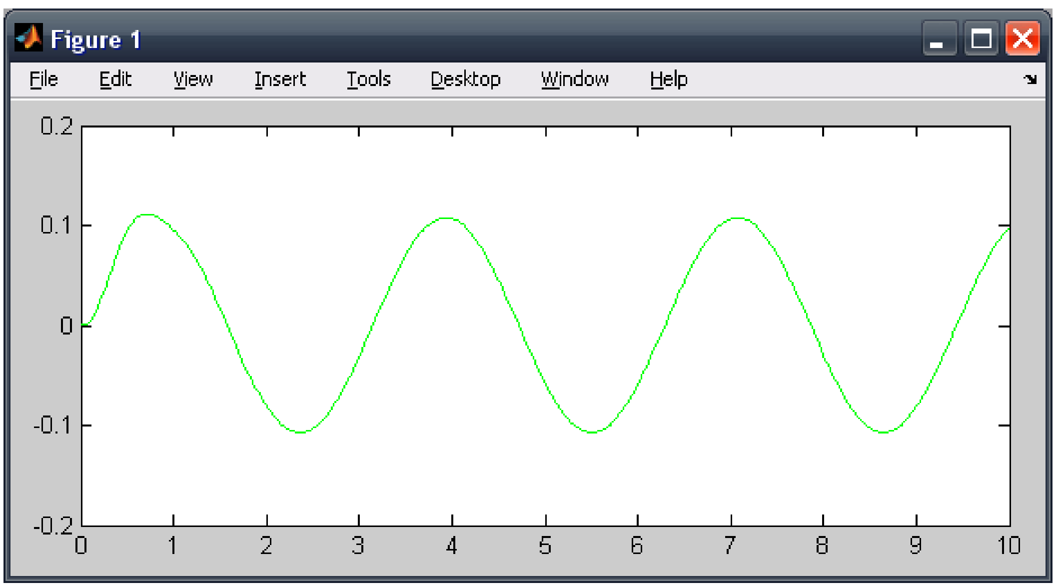


Figure 4.3: Quarter car model of passive suspension system

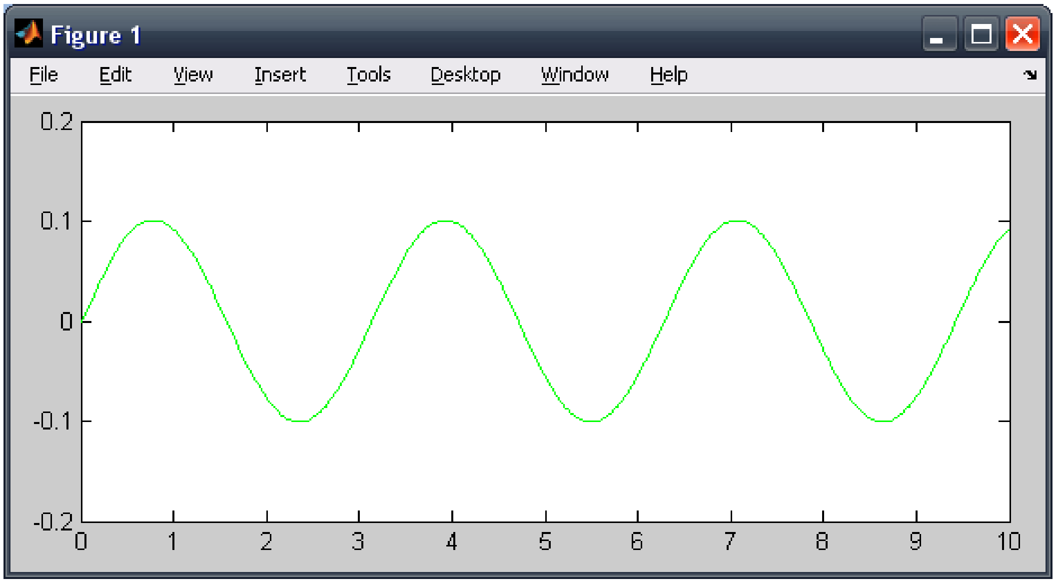
4.3.1 Frequency, ω = 2 rad/sec



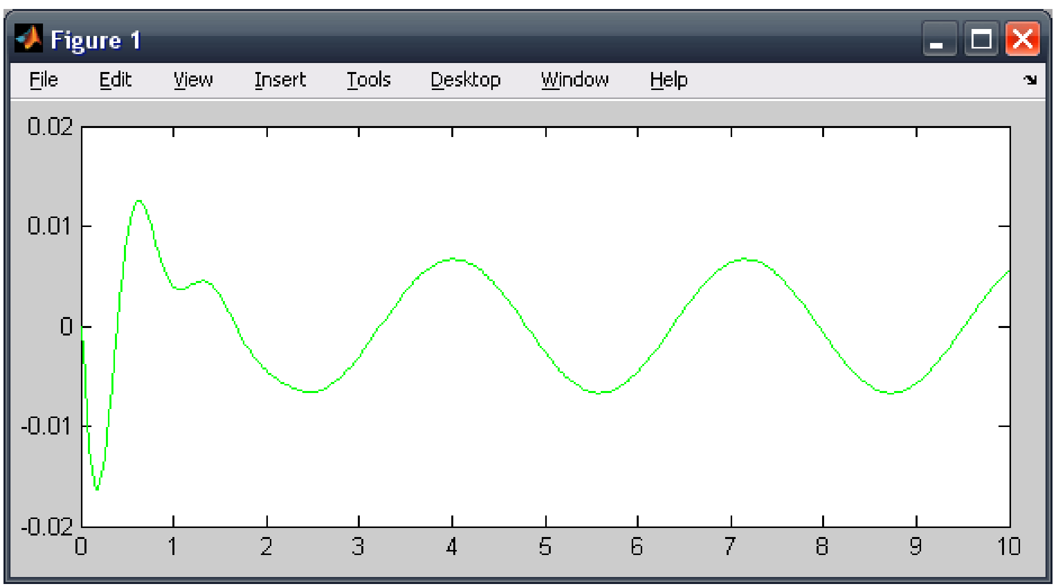
(a) Body acceleration



(b) Body displacement



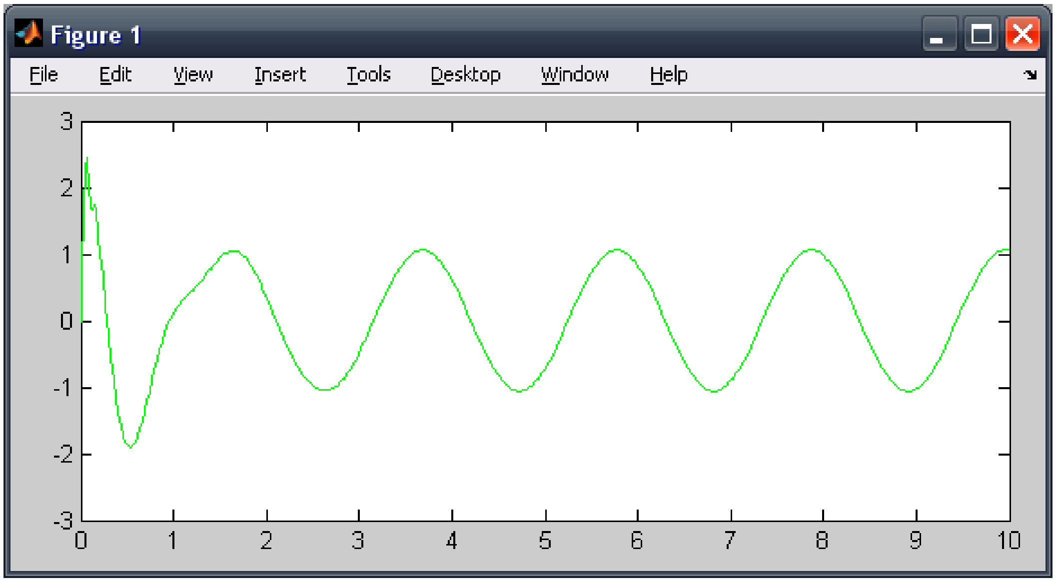
(c) Tire displacement



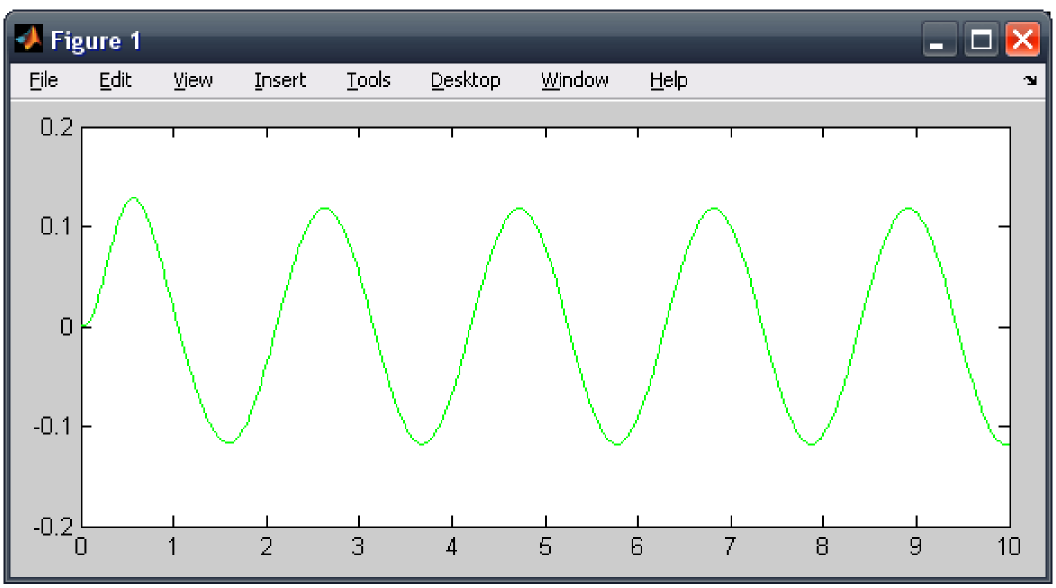
(d) Spring deflection

Figure 4.4: Quarter car model performance at frequency, $\omega = 2$ rad/sec

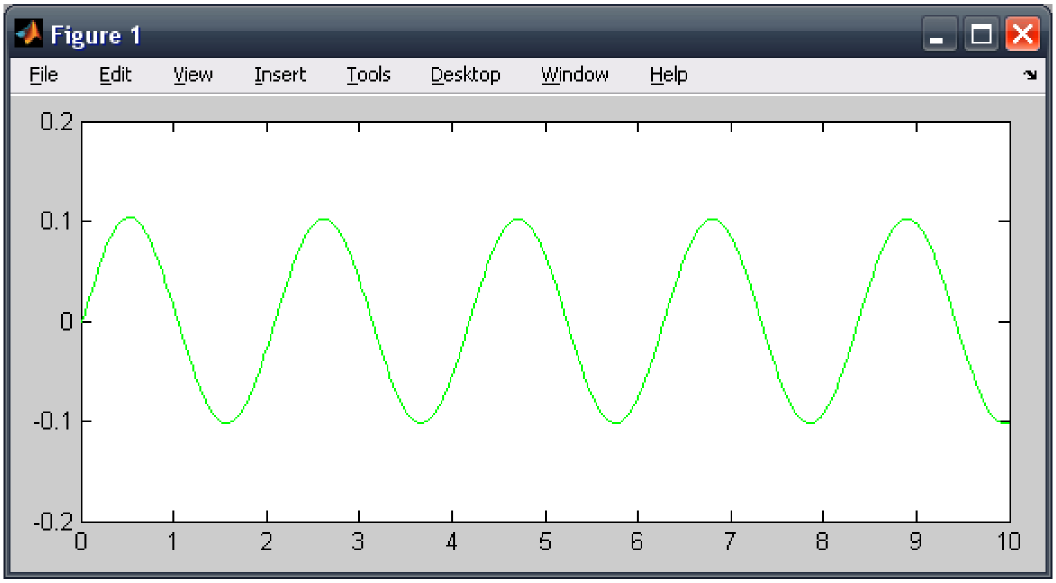
4.3.2 Frequency, ω = 3 rad/sec



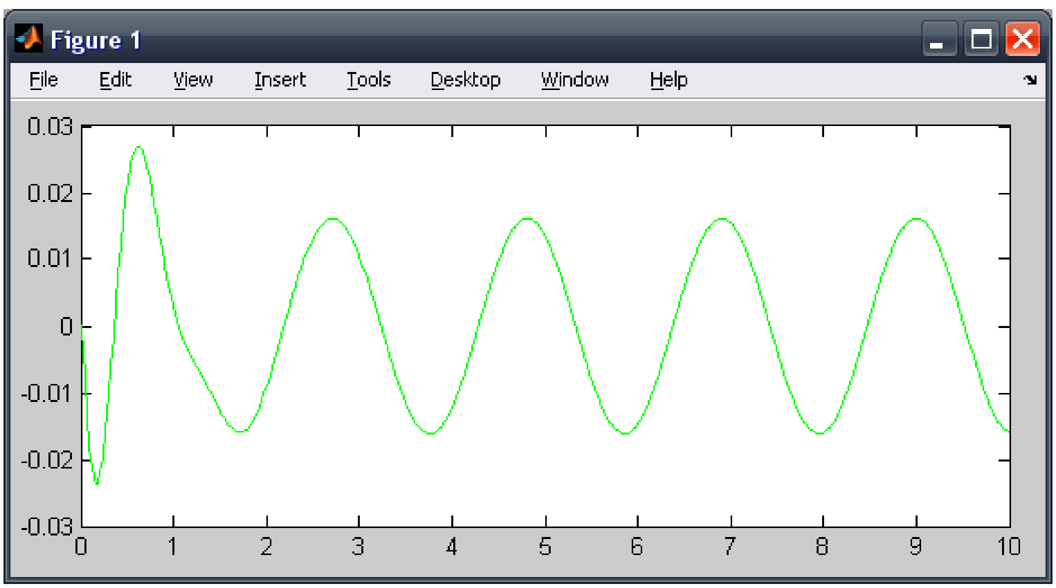
(a) Body acceleration



(b) Body displacement



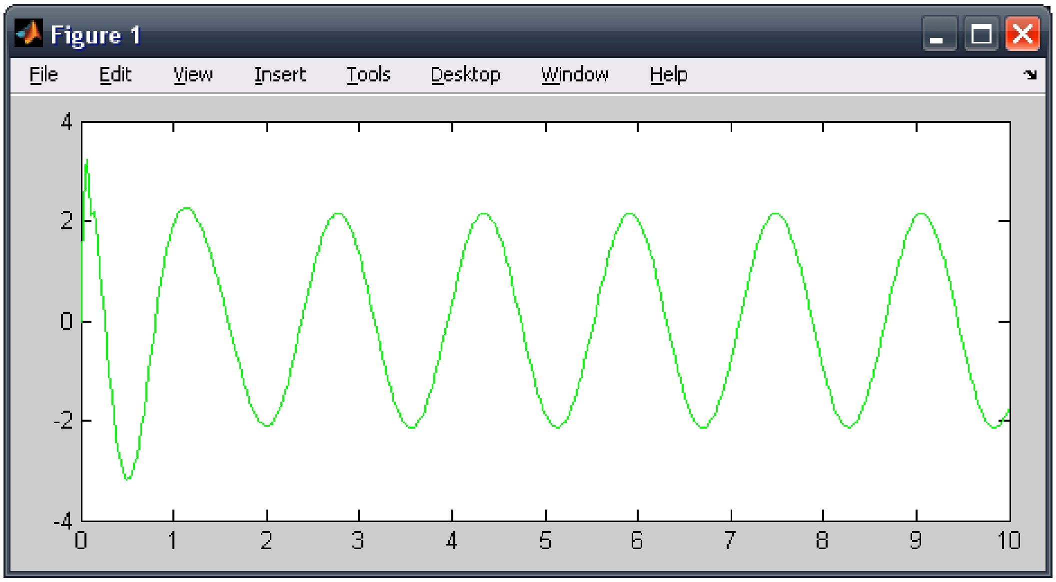
(c) Tire displacement



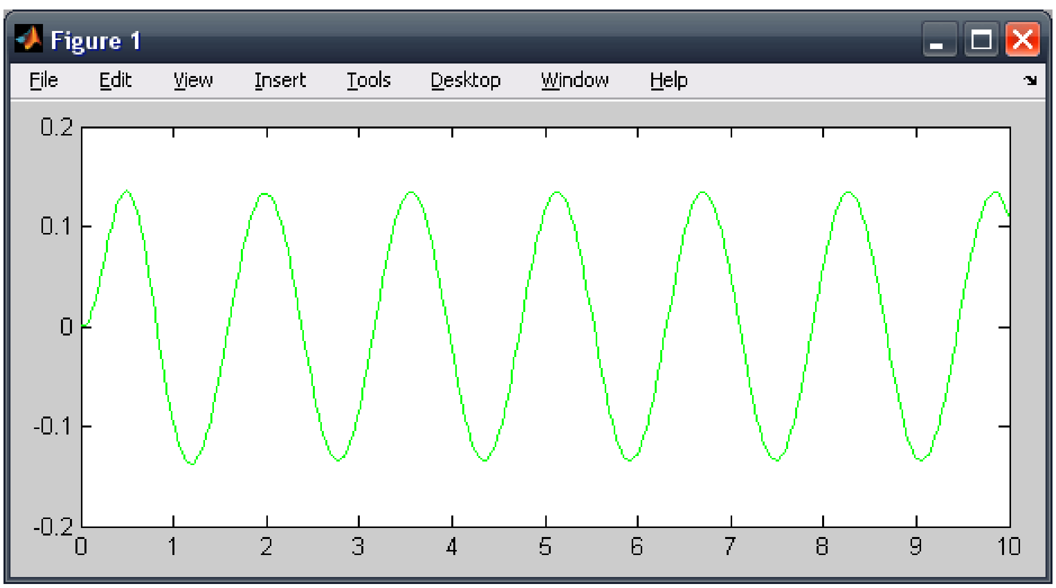
(d) Spring deflection

Figure 4.5: Quarter car model performance at frequency, ω = 3 rad/sec

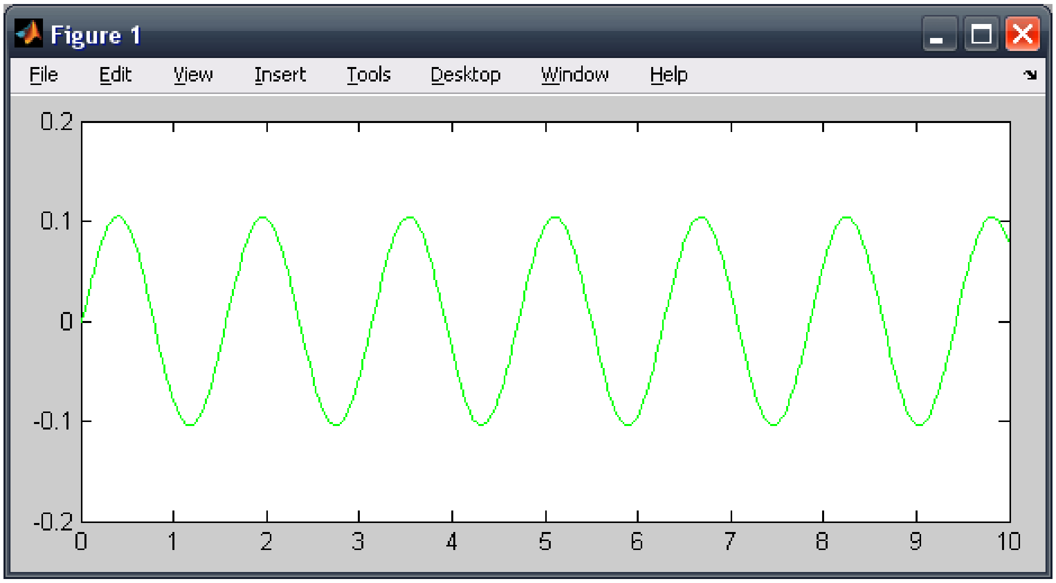
4.3.3 Frequency, ω = 4 rad/sec



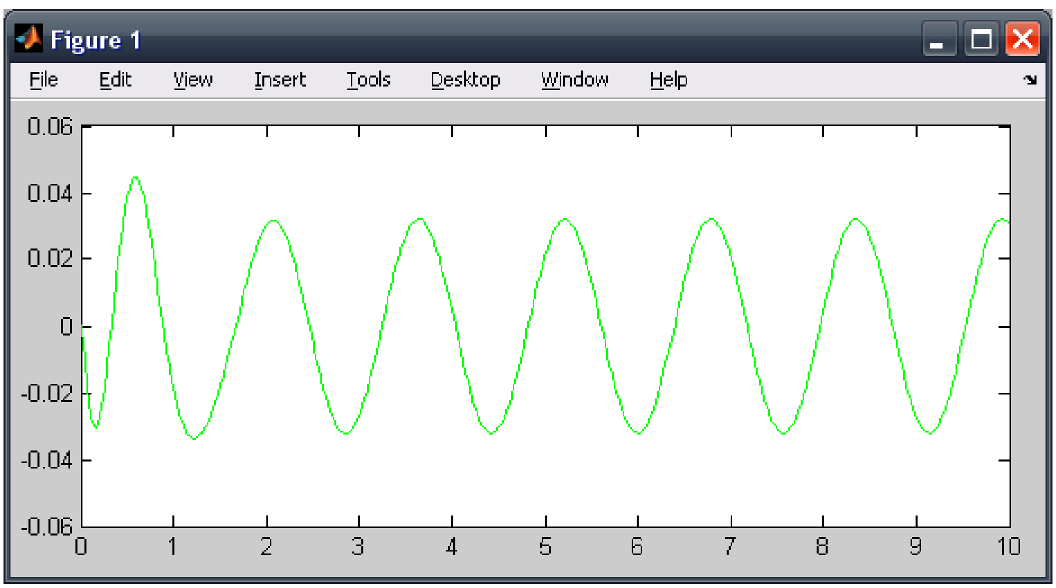
(a) Body acceleration



(b) Body displacement



(c) Tire displacement



(d) Spring deflection

Figure 4.6: Quarter car model performance at frequency, ω = 4 rad/sec

4.4 SIMULATION PERFORMANCE OF QUARTER CAR MODEL WITH SKYHOOK CONTROLLER

The force tracking controller of inner loop is then attached to the quarter car model and make it the active suspension system. Here the skyhook controller, *C,* is then attached to the outer loop system as shown in Figure 4.19. By using tune by feel method, the value of *C* is to be determined, and the three different *C* parameter values is shown in Figure 4.20 till 4.31. This is to find the best response due to the body acceleration, body displacement, tire displacement, and spring deflection compare to the quarter car model of passive suspension system. From the Figure 4.32 till Figure 4.35, the best value of *C* chosen to act with these four parameters would be -200. Here, even the values of *C* satisfied all the parameters, the step time response for *C* = -200 shows a better results compared to the others.

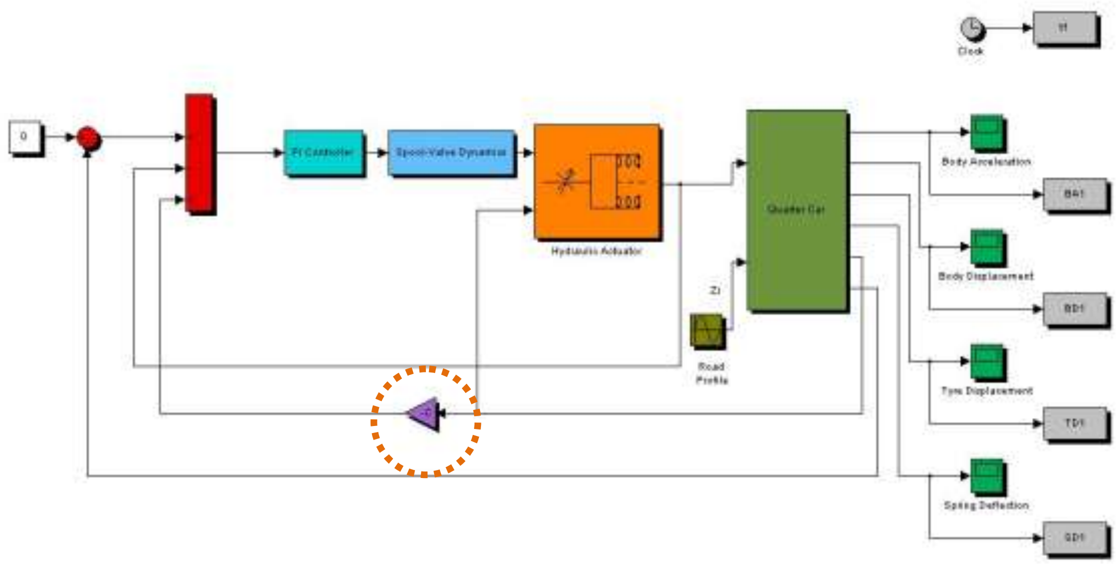
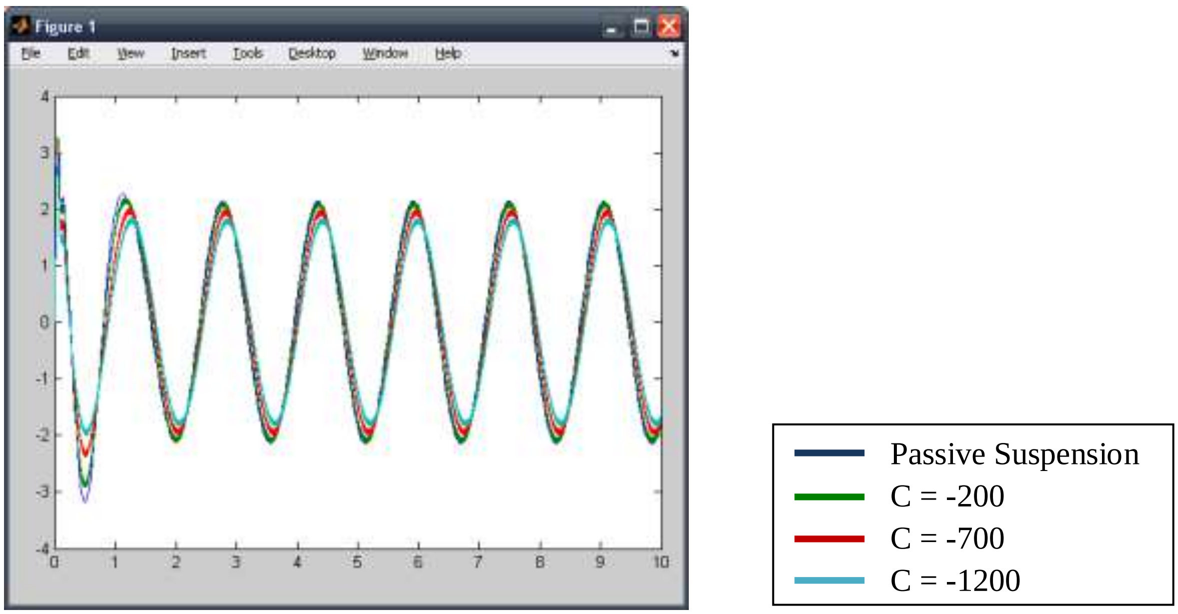
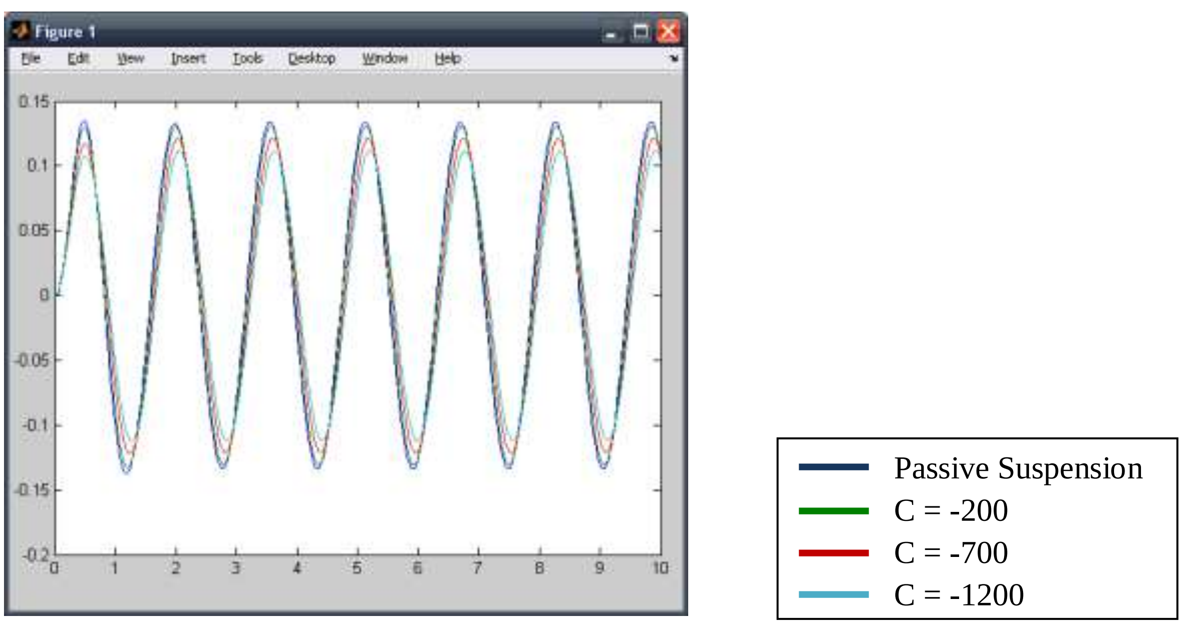


Figure 4.7: Quarter car model with skyhook controller

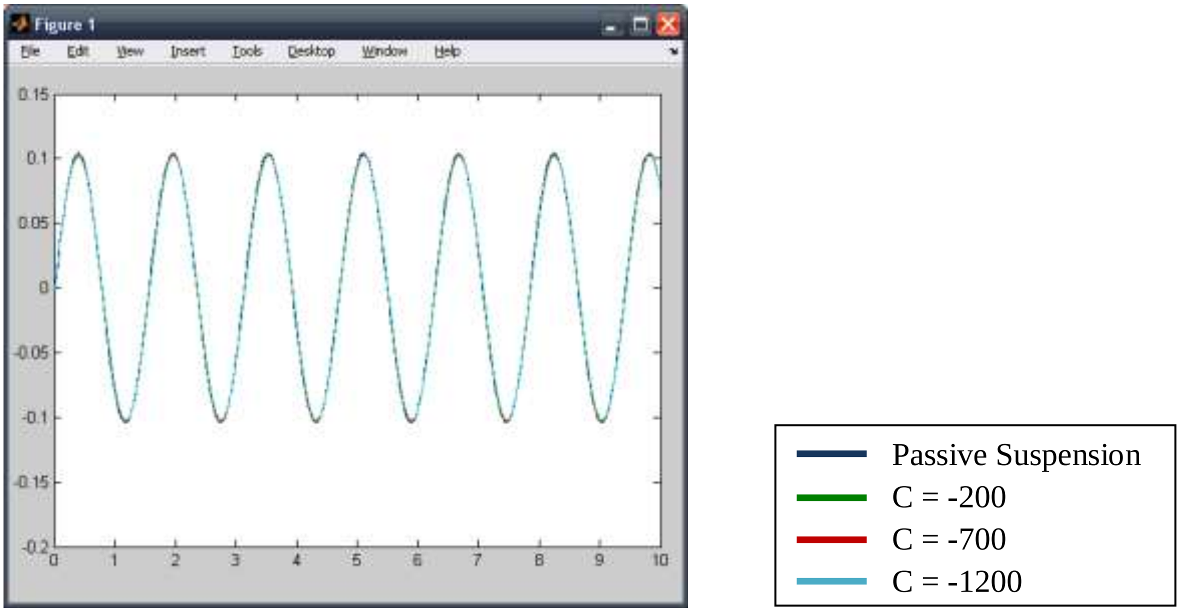
4.4.1 Comparing Skyhook Controller with Passive Suspension System of Quarter Car Model



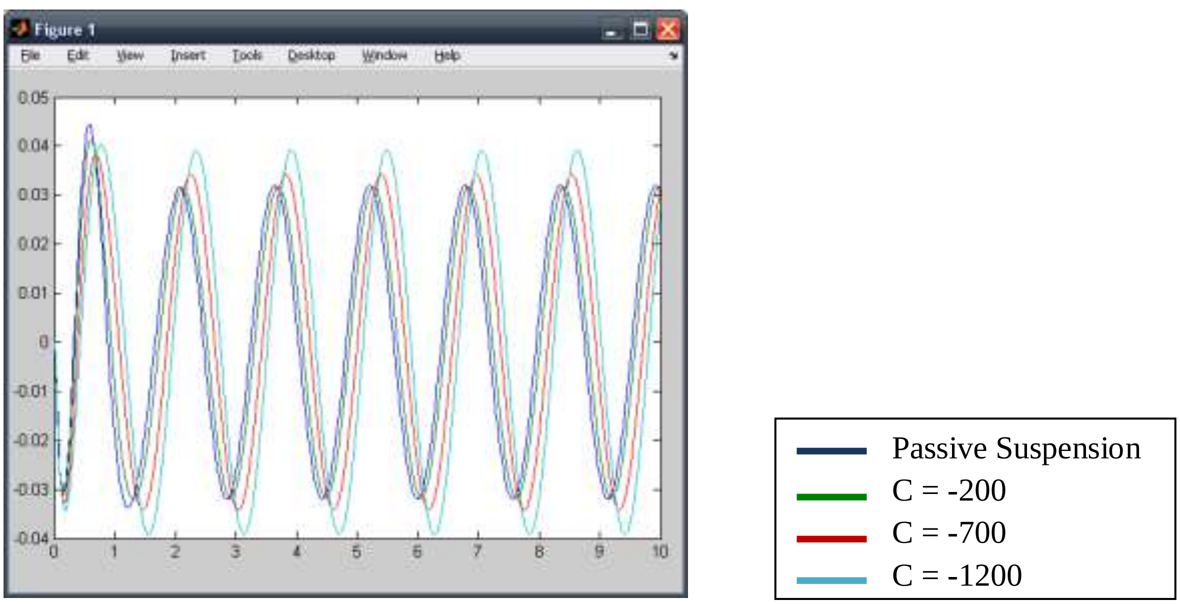
(a) Body acceleration



(b) Body displacement



(c) Tire displacement



(d) Spring deflection

Figure 4.8: Active suspension with skyhook control system vs. passive suspension system at frequency, ω = 4 rad/sec

4.5 PID CONTROLLER

Here, the final step where PID Controller, is then attached to the outer loop system as shown in Figure 4.36. By try an error, the value of *P*, *I*, and *D* is to be determined. The value of *P* is adjusted first, followed by the value of *I* and *D*. However, in this case, only the value of *P* and *I* are being adjusted since the *D* value does not influence the results as shown in Figure 4.37 till Figure 4.44. The final best response due to the body acceleration, body displacement, tire displacement, and spring deflection compare to the quarter car model of passive suspension system is obtain as shown in Figure 4.45 till figure 4.48. After performing the analysis at frequency = 2, 3, and 4 rad/sec and *C* = -200, the best value of *P* and *I* chosen to act with these four parameters would be 1100 and 300 respectively.

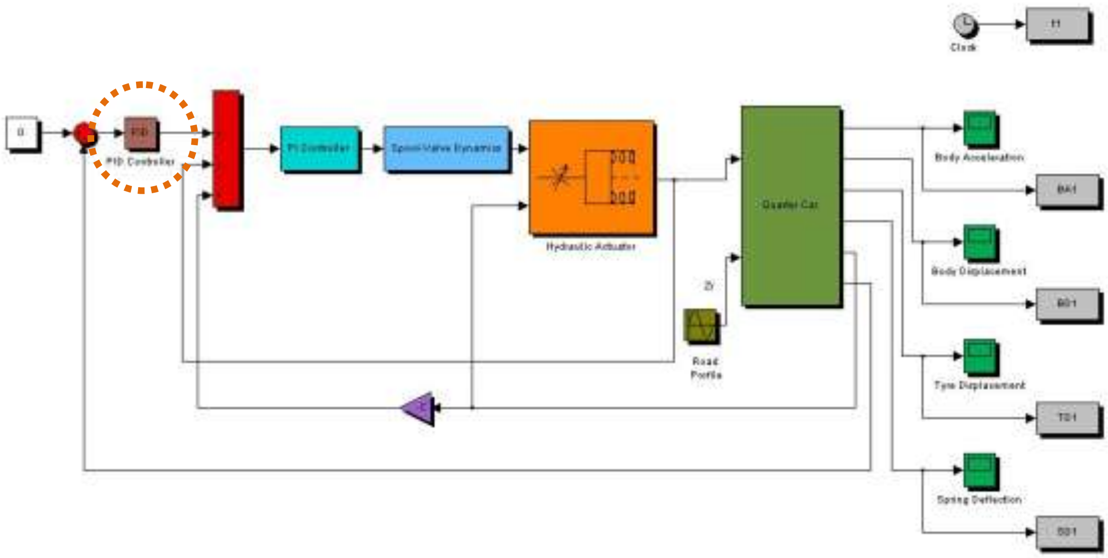
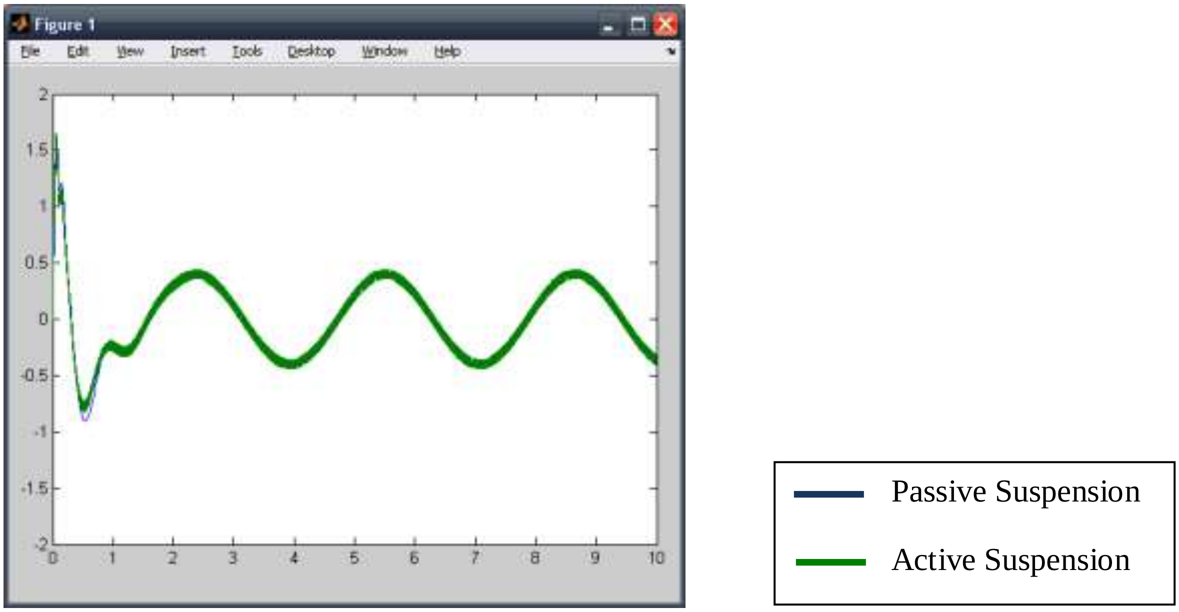
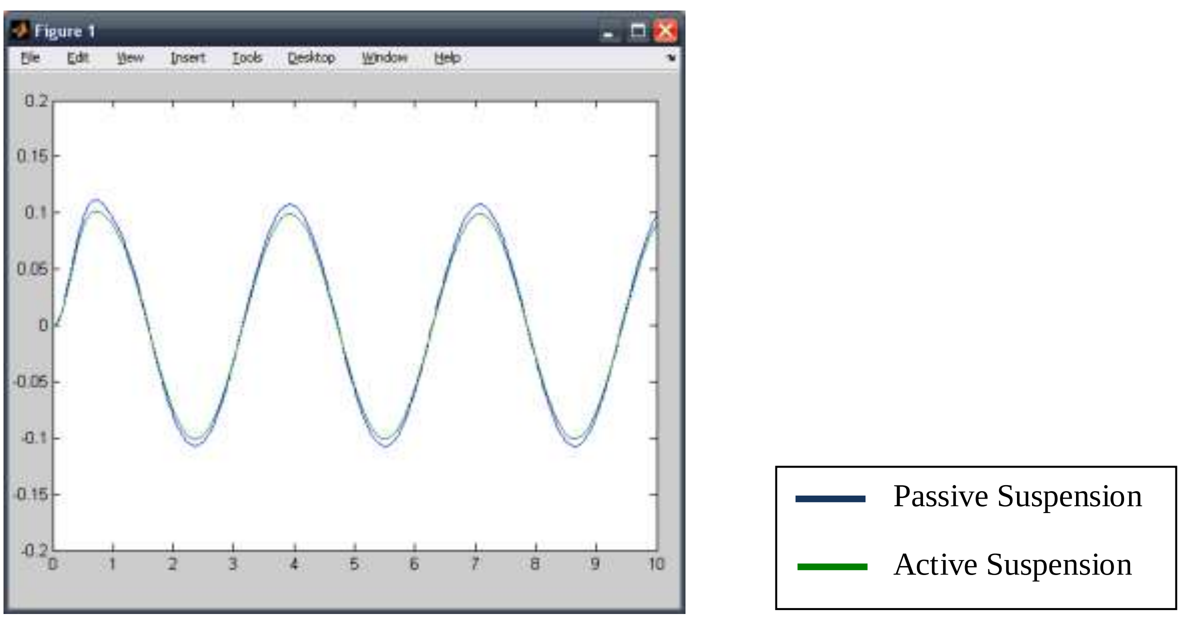


Figure 4.9: Quarter car model of active suspension system

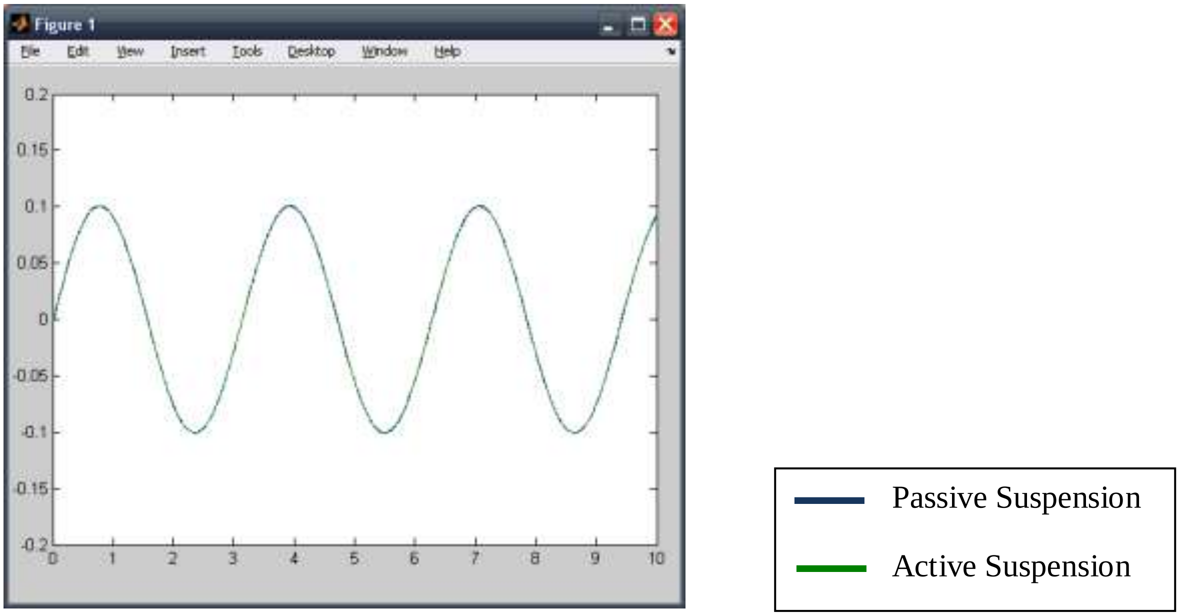
4.5.1 Frequency, ω = 2 rad/sec



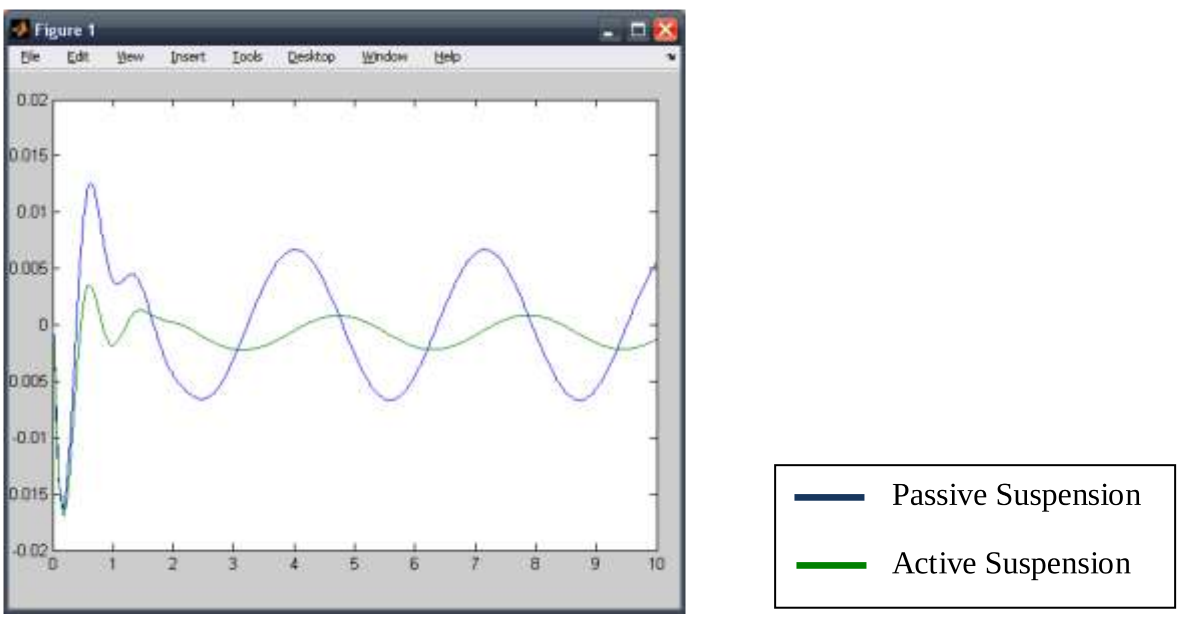
(a) Body acceleration



(b) Body displacement



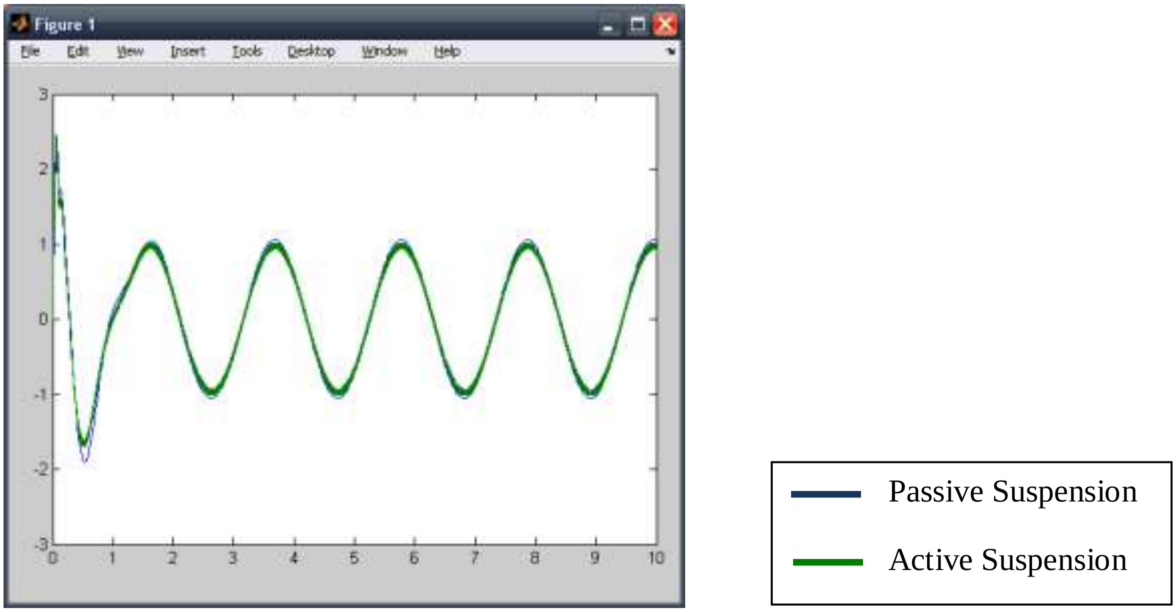
(c) Tire displacement



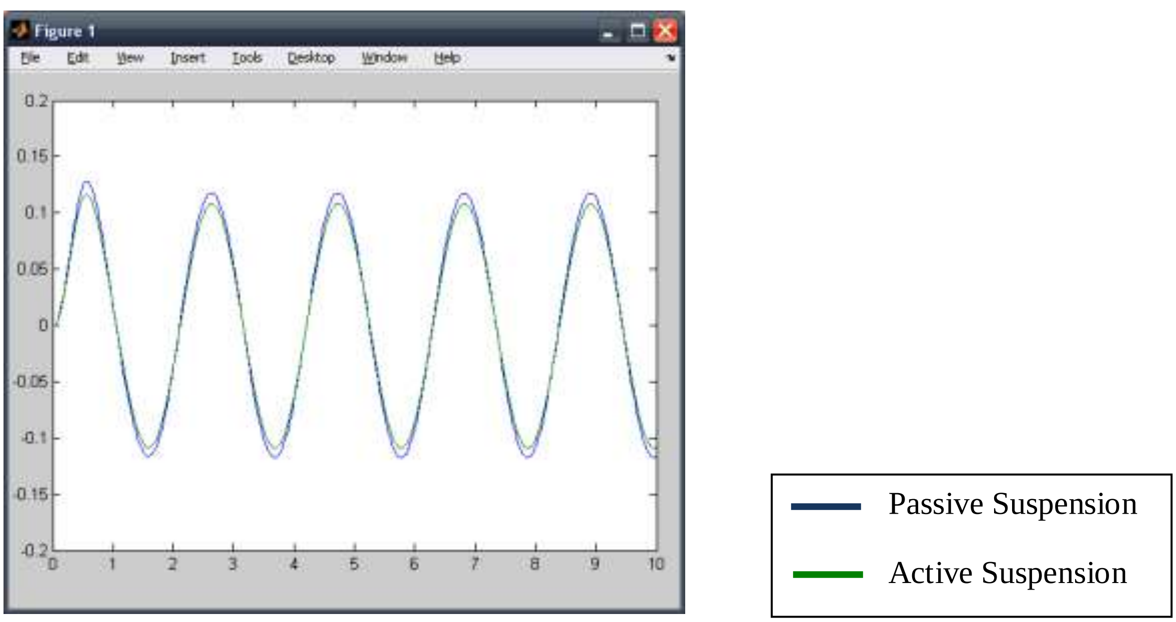
(d) Spring deflection

Figure 4.10: Active suspension system vs. passive suspension system at frequency, ω = 2 rad/sec

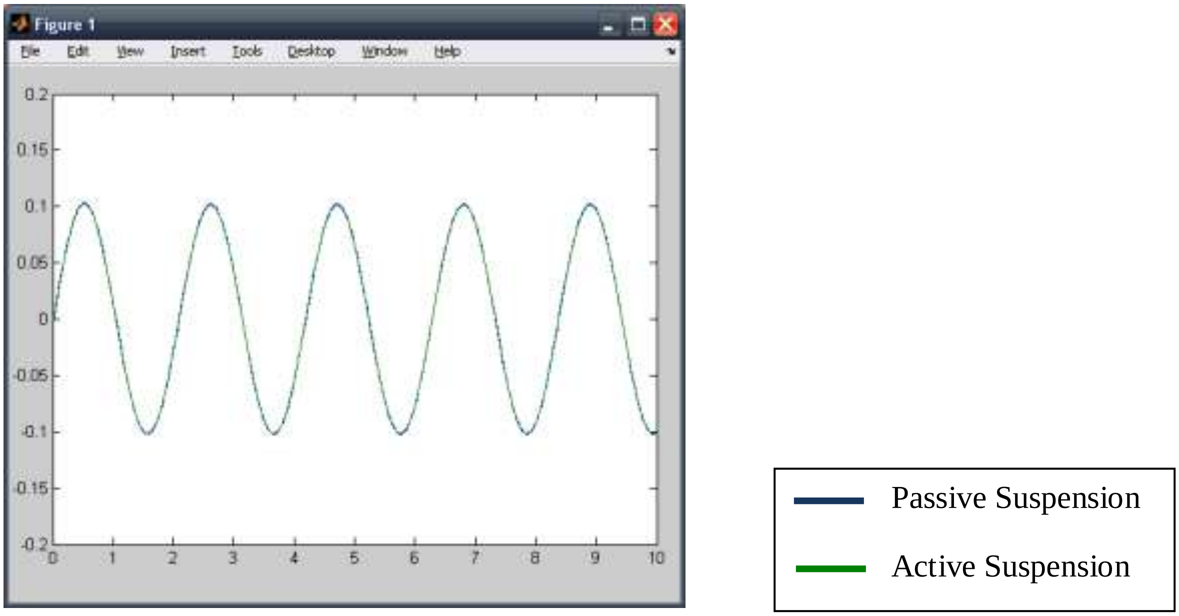
4.5.2 Frequency, ω = 3 rad/sec



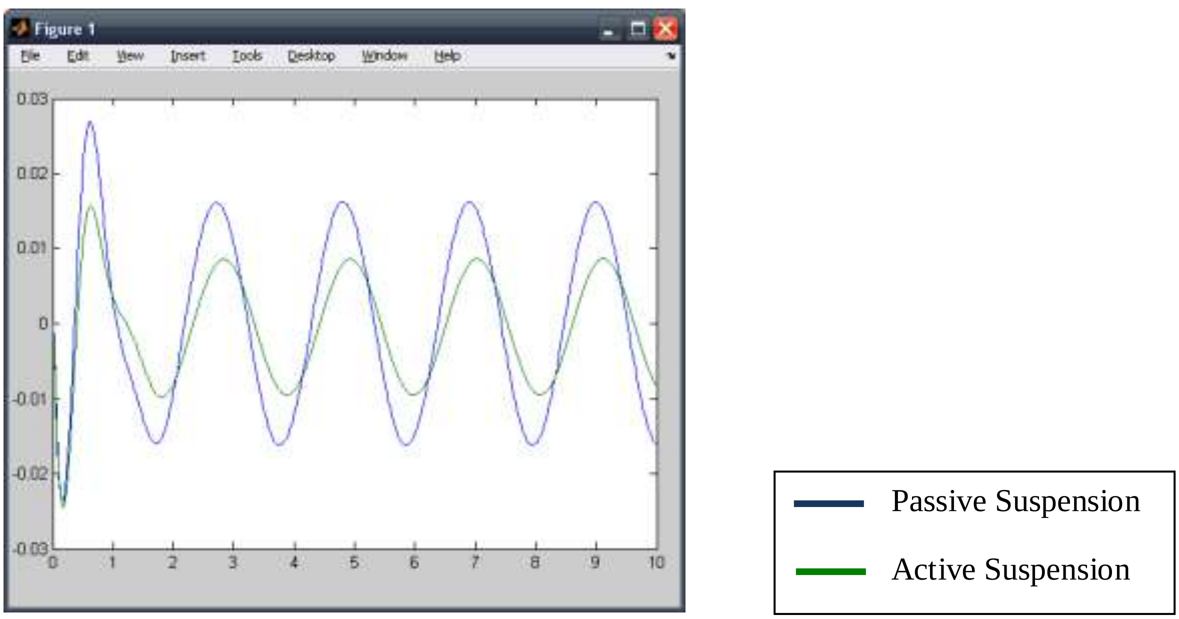
(a) Body acceleration



(b) Body displacement



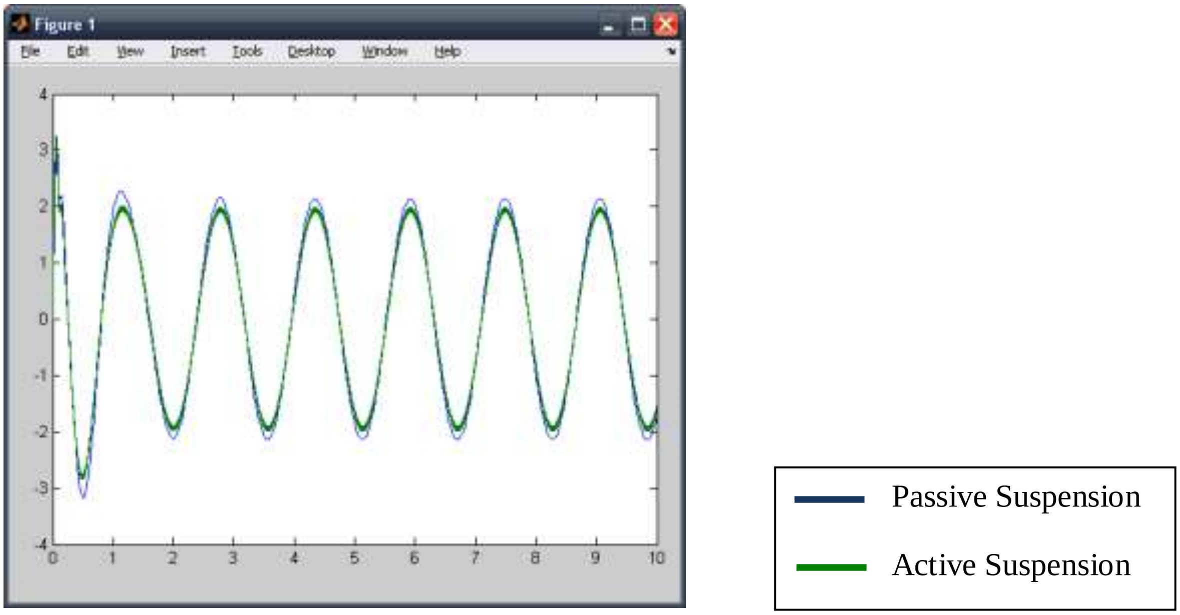
(c) Tire displacement



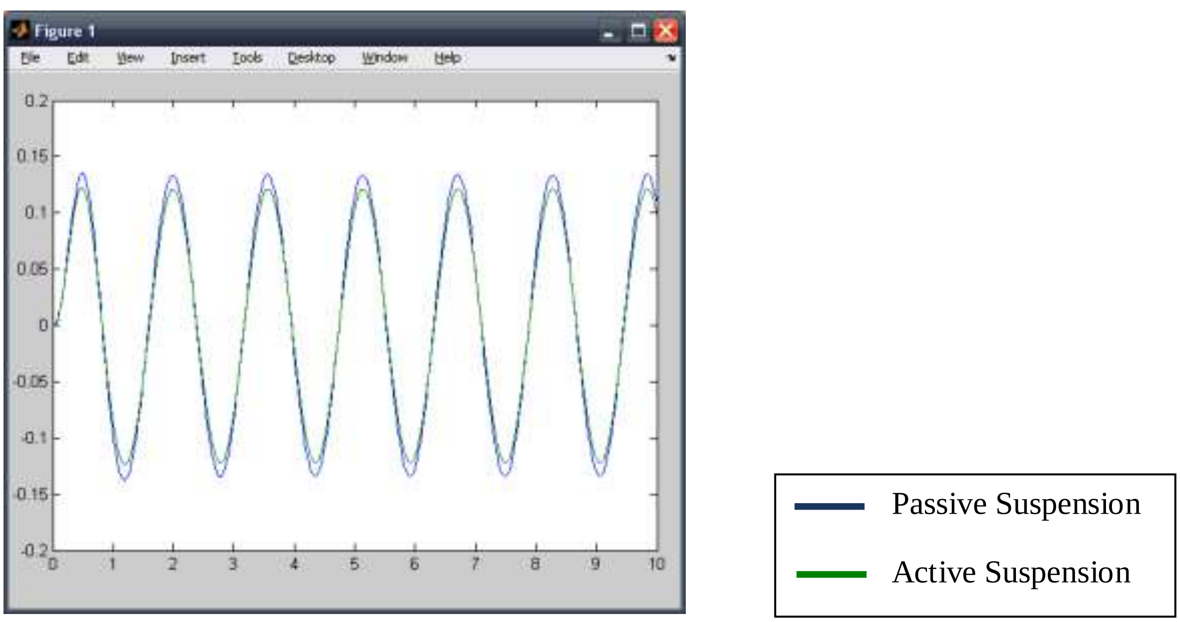
(d) Spring deflection

Figure 4.11: Active suspension system vs. passive suspension system at frequency, ω = 3 rad/sec

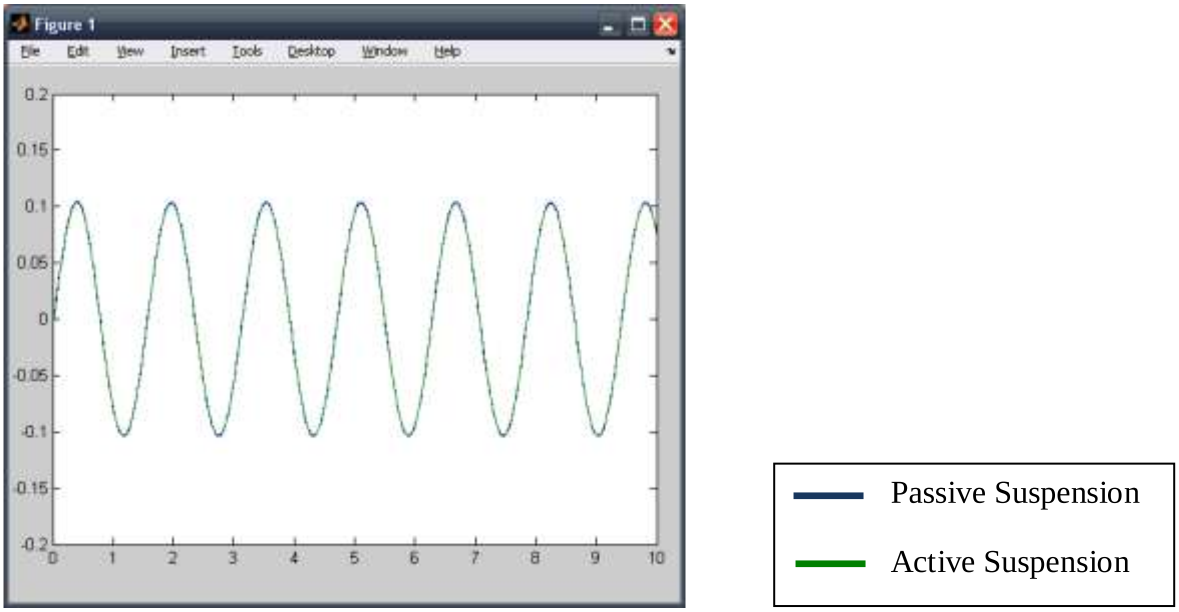
4.5.3 Frequency, ω = 4 rad/sec



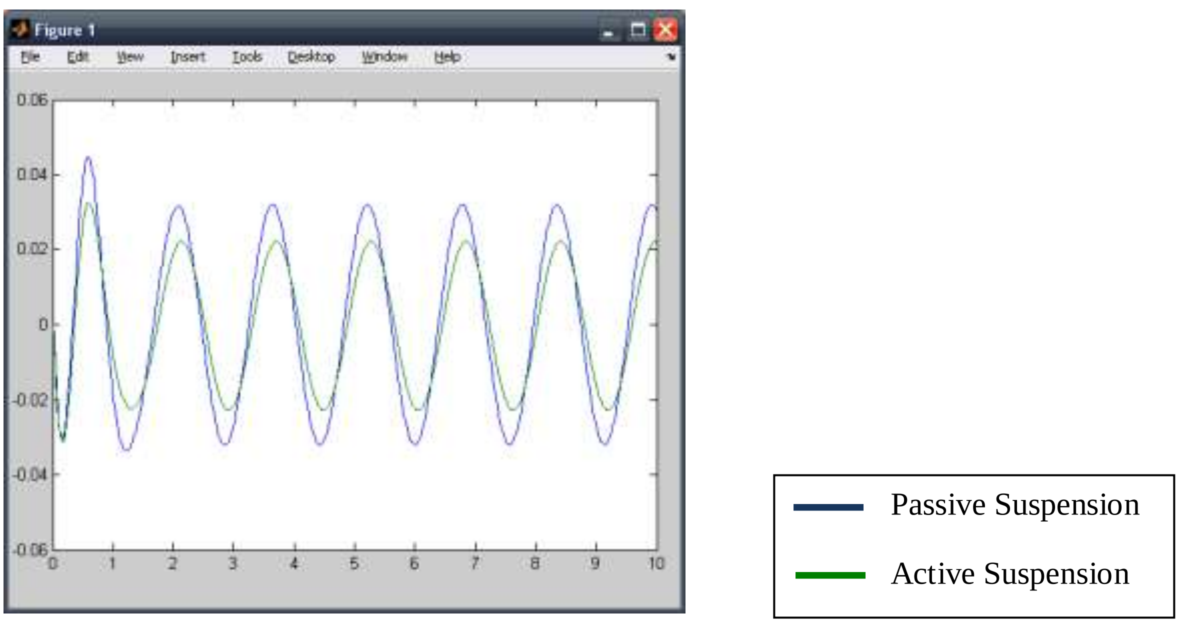
(a) Body acceleration



(b) Body displacement



(c) Tire displacement



(d) Spring deflection

Figure 4.12: Active suspension system vs. passive suspension system at frequency, ω = 4 rad/sec

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

The study presents a skyhook controller for an active suspension system based on variable structure control theory, which is capable of satisfying all the design requirements within the actuators limitation. The Proportional-Integral-Derivative (PID) controller is presented and used to reject the effects of road induced disturbances on the quarter car model. The performance characteristics of the active suspension system are evaluated and then compared with the passive suspension system. The result shows that all three objectives are successfully achieved.

Force tracking performance of the hydraulic actuator model was investigated. The model development of the hydraulic actuator model was based on the previous paper study (Donahue, 2001). Proportional Integral control and spool valve dynamics was implemented for force tracking control of the hydraulic actuator. The results of the study show that the hydraulic actuator is able to provide the actual force close to the target force with acceptable force tracking error for sinusoidal, square, saw-tooth and step functions of target forces.

The use of the skyhook controller along with the Proportional-Integral-Derivative (PID) controller is effective in controlling a vehicle compared to the counterparts. From the simulation results, it can be seen that the proposed controller shows significant improvement in reducing both magnitude and settling time of the body acceleration, body displacement, tire displacement, and suspension deflection. The proposed controller is capable of satisfying all the requirements for active suspension design.

5.2 RECOMMENDATION FOR THE FUTURE RESEARCH

For future work, it is suggested that the quarter car model use for simulation in this thesis to be improved by using more complex type of suspension such as Macpherson suspension system. In this study, the constants in the skyhook controller, *C* and PID controller are determined using trial and error approach. Thus, it is recommended that specific and more reliable method can be used. In addition, the proposed control strategy could also be validated with the experimental study.

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Hong, K. S., Jeon, D. S., Yoo, W. S., Sunwoo, H., Shin, S. Y., Kim, C. M., and Park, B. S. 1999. A new model and an optimal pole-placement control of the Macpherson suspension system. *7th Mediterranean Conference on Control and Automation (MED99)*. Haifa, 28-30 June. 01–1331: 267–276.

Hong, K. S., Jeon, D. S., and Sohn, H. C. 1999. A new modeling of the macpherson suspension system and its optimal pole placement control. Proceeding of the 7th Mediterranean *Conference on Control and Automation (MED99).*Haifa, 28-30 June. pp. 559- 579.

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APPENDIX A1

GANTT CHART/PROJECT SCHEDULE FOR FYP I

Month	**January**				**February**				**March**				**April**			
Subject/Week	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**
1. Introduction																
• Propose Project Title																
• Objective of Project																
• Discussion of Project Scope																
• Identify Problem Statement																
2. Modeling and Simulation (MATLAB)																
• Develop Hydraulic Actuator Model																
• Develop Force Tracking Controller																
3. Literature Review																
4. Methodology																
5. Presentation Preparation																
6. Submission Proposal and Logbook																
7. Presentation FYP I																

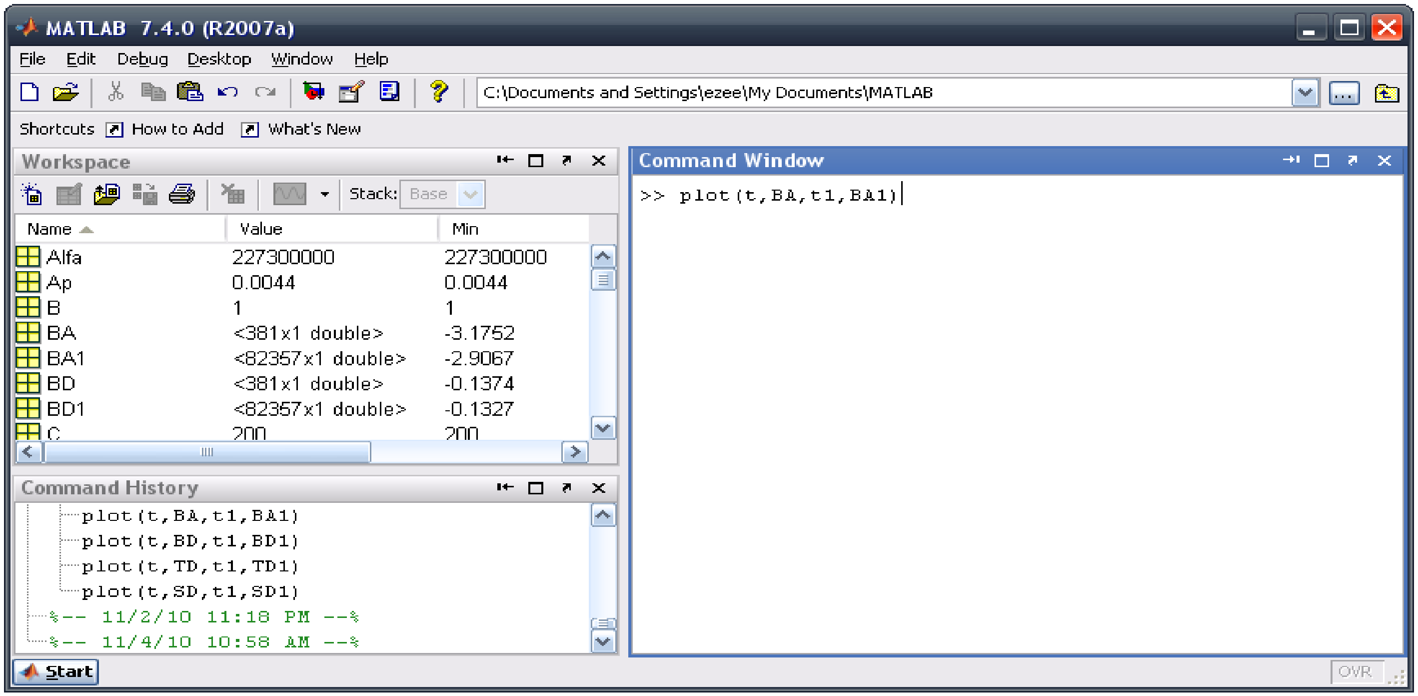
APPENDIX A2

GANTT CHART/PROJECT SCHEDULE FOR FYP II

Month	**July**		**August**				**September**				**October**				**November**				**December**		
Subject/Week	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**
1. Modeling and Simulation (MATLAB)																					
• Develop a Quarter Car Model of Passive Suspension System																					
• Develop a Skyhook and PID Controller to an Active Quarter Car Suspension System																					
2. Experimentation & Analyzing																					
3. Result & Discussion																					
• Data Collection																					
• Interpret Results																					
4. Conclusion																					
5. Presentation Preparation																					
6. Presentation FYP II																					
7. Submission Draft & Final Report																					

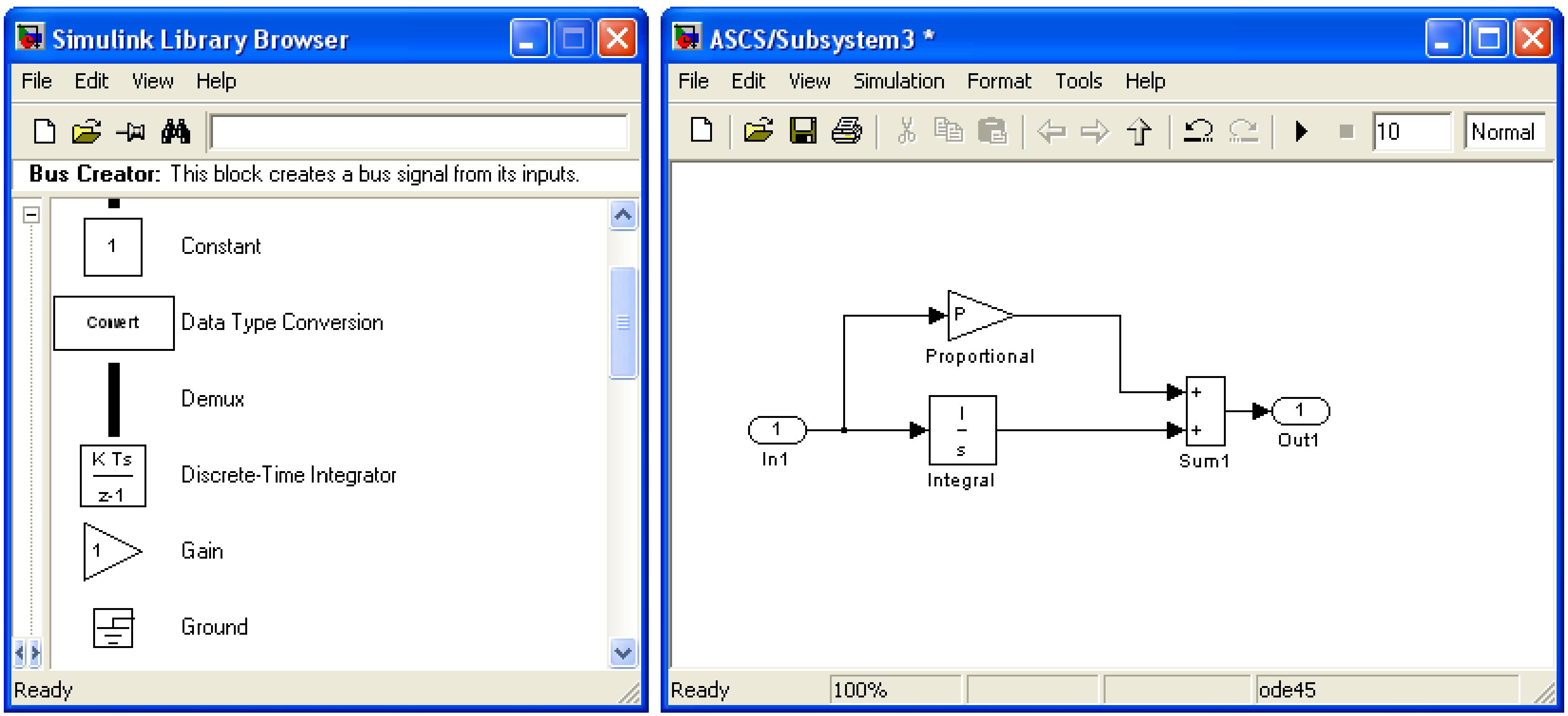
APPENDIX B1

MATLAB WORKSPACE



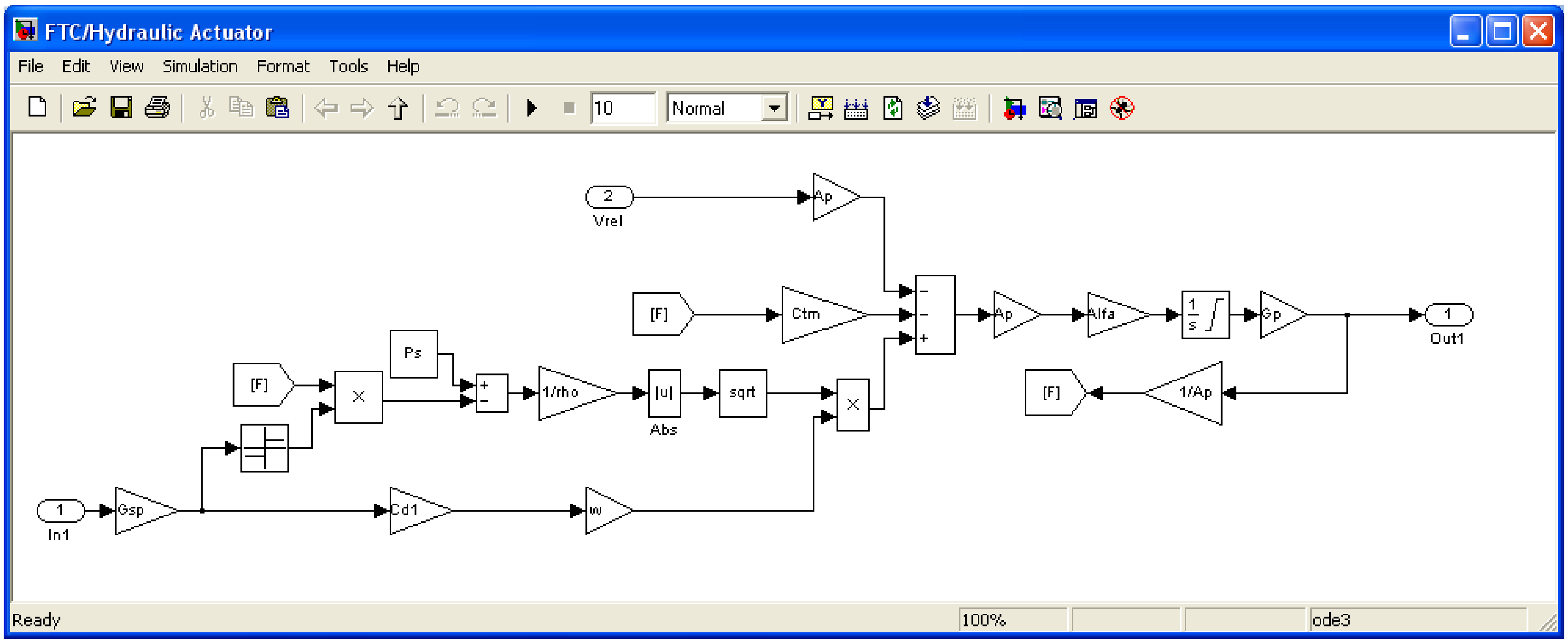
APPENDIX B2

SIMULINK LIBRARY BROWSER AND MODEL WORKSPACE



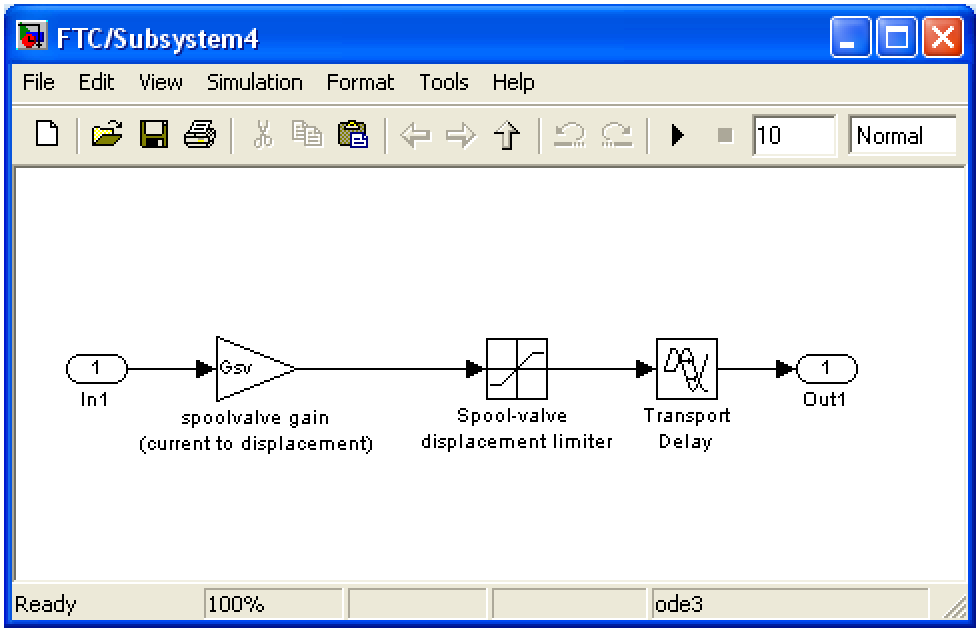
APPENDIX C1

HYDRAULIC ACTUATOR MODEL



APPENDIX C2

SPOOL VALVE DYNAMICS MODEL



APPENDIX C3

PI CONTROLLER MODEL