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UNIVERSITI TEKNIKAL MALAYSIA MELAKA

ACTIVE SUSPENSION CONTROL USING MULTIORDER PID CONTROLLER

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ACTIVE SUSPENSION CONTROL USING MULTIORDER PID CONTROLLER

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A report submitted
in fulfillment of the requirements for the degree of
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2016

DECLARATION

I declare that this report entitle "Active Suspension Control using Multiorder PID Controller" is the result of my own research except as cited in the references. This report has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

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APPROVAL

I hereby declare that I have read this report and in my opinion this report is sufficient in terms of scope and quality as a partial fulfilment of Master of Mechanical Engineering (Automotive)

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14 NOV 2016

DEDICATION

To my beloved mother, father, my wife, son and daughter

ABSTRACT

This study investigates the propose controller for active suspension system to improve ride comfort of the vehicle. The main content of this study is the development and application of the multiorder Proportional-Integral-Derivative (MOPID) control scheme, and the investigation of the control system ability to provide improvement in vehicle ride comfort. The multiorder Proportional-Integral-Derivative (MOPID) are arranged in a separated control loops called the outer loop controller, which is used to amplify the disturbance from the road input and reduce unnecessary vehicle motions and the inner loop controller that is produced the signal to the active suspension system to produce the force by force actuator. The performance of the proposed controller is compared to the conventional PID controller and the passive suspension system. Simulation studies are shown in time domain simulation produce from Matlab/Simulink software. From the simulation, it shows that the proposed control scheme is able to provide improvement in terms of body vertical displacement and body vertical acceleration. The development of the multiorder PID is also easy to implement in practice based to its simple controller structure.

ABSTRAK

Kajian ini adalah berkenaan sistem kawalan yang dicadangkan untuk sistem suspensi aktif untuk meningkatkan keselesaan perjalanan kenderaan. Kandungan utama kajian ini ialah pembangunan dan penggunaan sistem kawalan berperingkat perkadaran-integrasi-terbitan (MOPID), dan penyelidikan pada keupayaan sistem kawalan untuk menyediakan peningkatan dalam keselesaan perjalanan kenderaan. sistem kawalan berperingkat perkadaran-integrasi-terbitan (MOPID) disusun dalam gelung kawalan berasingan dipanggil gelung pengawal luar, yang digunakan untuk mengawal gangguan dari input jalan raya dan mengurangkan pergerakan yang tidak perlu pada kenderaan dan pengawal gelung dalaman yang memproses isyarat kepada sistem suspensi aktif bagi menghasilkan daya melalui penggerak. Prestasi pengawal yang dicadangkan dibandingkan dengan pengawal perkadaran-integrasi-terbitan (PID) konvensional dan sistem penggantungan pasif. Kajian simulasi ditunjukkan melalui masa simulasi domain hasil daripada perisian Matlab/Simulink. Dari simulasi, ia menunjukkan bahawa skim kawalan yang dicadangkan mampu memberikan peningkatan dari segi anjakan menegak badan dan pecutan menegak badan. Pembangunan sistem kawalan berperingkat perkadaran-integrasi-terbitan (MOPID) ini juga mudah untuk dilaksanakan dalam amalan berdasarkan kepada struktur kawalan yang mudah.

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LIST OF SYMBOLS

C_s - Stiffness value of the passive damper

e - Errors

e_z - Body displacement errors

$e_{\dot{z}}$ - Body velocity errors

$e_{\ddot{z}}$ - Body acceleration errors

F_p - Augmented force from the actuator

K_I - Integral Constant

K_P - Proportional Constant

K_D - Derivative Constant

K_s - Stiffness value of the passive spring

K_T - Stiffness value of the tire

L - Stroke length of the piston

M - Mass of the pneumatic piston

M_u - Mass of the wheel axle (unsprung mass)

M_s - Mass of the vehicle body (sprung mass)

RMS - Root Mean Square

Z_u - Vertical displacement of the wheel axle

Z_s - Vertical displacement of the vehicle body

Z_r - Vertical displacement of road profile

$\dot{Z}_u$ - Vertical velocity of the wheel axle

$\dot{Z}_s$ - Vertical velocity of the vehicle body

$\dot{Z}_r$ - Vertical velocity of road profile

$\ddot{Z}_u$ - Vertical acceleration of the wheel axle

$\ddot{Z}_s$ - Vertical acceleration of the vehicle body

$r_{(t)}$ - Reference input signal

$e_{(t)}$ - Error Signal

$u_{(t)}$ - Input signal to the plant model

$d_{(t)}$ - Disturbance

$y_{(t)}$ - Output signal

DOF - Degree of Freedom

CHAPTER 1

INTRODUCTION

1.1 Introduction

Active suspension has attracted many researchers to study and develop a new way of solution to bring the most ideal vehicle suspension. Related research about active suspension are publish by Sarma and Kozin (1971), Sutton (1979), Margolis (1982), Yoshimura et al. (1986), Yamashita et al. (1990) and Lin and Lian (2011). All this research explains about the system in suspension of the vehicle that can be categories in 3 systems such as vehicle passive suspension system, vehicle semi-active suspension system and most advance vehicle active suspension system.

The vehicle passive suspension system, the system only contain by the spring that support the weight of the vehicle and the absorber that absorb the vibration and oscillation of the suspension movement. For the system in vehicle active suspension, we can see the spring that support the weight of the vehicle and the absorber to absorb the vertical movement of the vehicle are takeover by actuator whether in fully or by partial of the system. These actuator will operate to buildup force to counter the road surface movement by calculate and control the reaction from the movement of the vehicle suspension system.

Vehicle semi-active suspension systems are develop and be produce from the vehicle active suspension system by replace the use of actuator as force builder to the damper that can be controlled the movement of the vehicle and the spring to support thr weight of the vehicle. Semi-active suspension system is only capable of dissipating energy at a variable rate by adjusting the damping force (Williams, 1994). The adjustable damping characteristic can be accomplish by apply an adjustable damper with the appropriate control strategy. The simplicity of the control structure of semi active suspension yields that the controller can run on a low-cost system. Swevers et al. (2007). There are two different pattern of damper that been used in the vehicle semi-active suspension known as adjustable orifice damper and magnetorheological damper.

1.2 Problem Statement

The vehicle suspension system brings a major task in order to maintain vehicle driving performance and comfort. The studies for the front vehicle suspension system are very important to improve the vehicle performance in order to keep the vehicle in control and bring the comfort of driving. To enhance these criteria, many researchers come with the solution with introduction of active suspension component. The active suspension has a very wide function in order to absorb varies of driving condition and road profiles.

Vehicle suspension system is the structure on vehicle that connects the wheel and the body of the vehicle. The function of this system is to isolate the vehicle body with the road surface profile in order to maintain the ride comfort. Another function of the vehicle suspension is to keep the vehicle in control.

The primary function of the vehicle suspension can be achieve by using spring as a malleable component combine with damper as a force absorption component. An another function can be accomplish by manage the motion of the suspension system and wheel assembly using the mechanical link.

Vehicle suspension system needs to hold several type of vehicle dynamics aspect. For achieving optimum ride comfort, it needs to keep to minimum the vertical body acceleration of the suspension system and good dynamics performance needs an optimum road contact so it needs to contain the normal force between road and tyres in all conditions. It's all needs to working in vehicle body acceleration and vehicle body displacement. The disadvantages of the passive system are it needs to gain other performance but to sacrifice other performance area. To develop the vehicle suspension system that can give both ride comfort and handling performance, the reliance on vehicle passive suspension is irrelevant. This problem can be overcome by developing an active suspension system.

By combining the passive suspension component and force actuator that been control by a controller that sense by a sensor, the active suspension system work efficiently in order to reduce vertical body acceleration, reduce body vertical displacement and overall improve the ride comfort and handling performance of the vehicle.

Sensor in active suspension system play a major in detect the body displacement and body acceleration, the signal then is used by the controller to analyse the condition and measure what type of feedback to be send to the actuator. The force actuator receives the processed signal from the controller that contain how much force its need to control the active suspension system. This will form a closed loop system. The data from the closed loop system will be compared with the open loop system to measure the performance of active suspension system.

For this research, the aim is to design and analyse the proposed control system for the body vertical displacement and body vertical acceleration of proposed vehicle active suspension system. The control system will be implemented via a suspension active suspension system that consists of spring, damper and a force actuator. The forces between upper sprung mass and below unsprung are implement by a controller signal. The controller is design through simulation studies which compare between Proportional Integral Derivative (PID) Controller and multiorder Proportional Integral Derivative Controller (MOPID) in order to compare the best output. The result of the output signal is then compared with passive suspension system. Performance criteria to be evaluated in this research is the ability of the multiorder PID controller in reducing vertical sprung mass acceleration and suspension travel displacement to compare with passive suspension system and Proportional Integral Derivative (PID) active suspension system.

1.3 Research Background

They are many controllers that have been used in a research of a controller in active suspension system. The controllers may be divided into 3 groups of control strategies named linear controller, nonlinear controller and intelligent controller. The linear controller is mainly focus on optimal control theory such as Linear Quadratic Gaussian (LQG), Linear Quadratic Regulator (LQR) and Loop Transfer Recovery (LTR) that can reducing the performance indication but this type of controller have a less function of processing the variation of data and road profile (Sam *et al.* 2006). Based on the past study, the nonlinear controller use such as sliding mode control (SMC) and an adaptive controller have come out with the good result. However the main problem is this type of controller is unstable. The study on intelligent controller shows that this type of controller brings promising result but with the tradeoff of stability problem. For the study purpose, the stability characteristics usually been ignored in the design of the controllers. Even all these controllers are proof to be effective in controlling the active suspension; PID control method is selected to be used in these studies. PID controller based is chosen because the controller is easy to operate and sustain in controller hardware and already proven by many researchers.

To study the performance and behavior of vehicle active suspension using PID and multiorder PID controller, the MATLAB/Simulink software used. The result of the active suspension is then compared with the data from passive suspension system.

1.4 Objectives and Scope of Research

The objectives of this research are defined as follows:

1. To simulate and validate a model of two degree of freedom (2 DOF) quarter car suspension system.
2. To develop a PID and multiorder PID controller using Matlab/Simulink.
3. To study the performance of PID active suspension and multiorder PID active suspension signal.

The scope of this research covers the followings:

1. This study is focus on the controller's capabilities to reduce body vertical displacement and acceleration by using Matlab/Simulink simulation.
2. This study is using two degree of freedom (2 DOF) quarter car model. The tyre is show as a true spring without damper effect. The motion of rotational in body and is neglected. The movement of wheel and body are assumed perfectly vertical. The action of spring and damper are simulated as linear.
3. Only ride comfort analysis is performed.
4. The performance of the controller is investigated on the capability to attenuate the effects of road induced disturbances.
5. The simulation is carried out using Matlab/Simulink and the quarter car model is validated with the result from (Imaduddin, 2010).

1.5 Methodology

The step of methodology that been used in this research is described as the following stage of works:

1. Modelling and validation of two degree of freedom (2 DOF) active suspension quarter car model.

   This research started with the development of a quarter car suspension model to describe analytically the dynamic behaviour of a quarter car model in vertical direction. For this purpose, a 2 degree-of-freedom (DOF) quarter vehicle active suspension model is simulated.

2. Control design by simulation of active suspension and PID controller

   Simulation study on the performance of the active suspension system along with the PID controller for vehicle comfort was also been investigated. The main activity in this stage is to assess the capability of active suspension with PID controller to increase vehicle comfort compare to passive suspension system.

3. Control design by simulation of active suspension and multiorder PID controller

   As the capability of PID controller in usage as the controller of active suspension system has been established, this study proceeds with the enhanced controller design. Multiorder PID were designed and the performance were assess on a two degree of freedom (2 DOF) quarter car model as par as normal suspension usage. Performance evaluation of the control strategies were evaluate by the capability of the multiorder PID controller in order to increase ride comfort better than active suspension system controlled by PID controller.

4. Performance evaluation of active suspension controllers

   The concluding stage in this research was the simulation of the performance on the vehicle active suspension controller. All the simulation studies in normal suspension behaviour and evaluate between passive suspension behaviour, vehicle active suspension with the implementation of PID controller and active suspension with the implementation of multiorder PID controller. The simulation result will then be compared and performance between simulations will be evaluated.

1.6 Project Report Outline

This project report consists of five chapters. Chapter 1 is the introduction chapter. This chapter presents the problem statement, the research background, objectives and scopes of the study, methodology of research, and the outline of this thesis

Chapter 2 presents the literature review on the subjects regarding this thesis. In this chapter, the type of vehicle suspension system, the actuator types and the type of control strategies for active suspension system are described.

Chapter 3 presents the methodology of the studies on quarter car model suspension model and the controller design. In this chapter, the mathematical equation of 2DOF quarter car model is discussed and the validation parameter of quarter car test model is presented by using the data from (Imaduddin. 2010). Then the designs of proposed PID controller and multiorder PID controller are explained.

Chapter 4 describes the result and discussion of the validation of quarter car suspension model from the simulation and the data from (Imaduddin. 2010) and the performance comparison of the proposed controllers. In this chapter, the validation of the 2 DOF quarter car suspension model is compare with the data from (Imaduddin. 2010) and discussion is focus on the pattern of the validation graph. Then the simulation analysis on the frequency domain and time domain of the proposed PID controller and multiorder PID control structure are presented and a compared with passive suspension system.

Finally, Chapter 5 is the conclusion chapter. This chapter summarizes this entire study of works. The recommendations for future research works are also described.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

In the past decade, vehicle suspension system has been widely studied in order to reduce vibrations and add comfort in driving condition. Various types of suspension have been introduced in improving the quality of vehicle comfort. (Dongpu et al, 2011), (M.J Griffin 2007), (Kumar and Vijayarangan, 2007) and (Gaur and Jain, 2013). The performance of vehicle dynamic has been a major focus of previous researchers. There are many ways to quantify a vehicle dynamic performance. In the study of vehicle suspension systems, ride comfort and driving performance are two things that are often given attention.

Vehicle suspension system has several main functions. The first function is to reduce vibration for the comfort of the driver and passenger in the vehicle. The second function is to ensure performance of the drive system in the best condition. The third function is to ensure that the vehicle suspension system can support the weight of the vehicle in order to keep the minimum rattle in the vehicle. (Rajamani, 2006).

In ensuring the comfort of the driver and passenger in the vehicle, the vehicle suspension system must be successful in isolating vibration and road conditions and the vehicle body. Good vehicle suspension system is a system that divides the vibration under the vehicle with the upper body of the vehicle so that the resulting vibration from the road condition cannot be transfer to the passenger and driver. To ensure that the vehicle handling system is always in the best condition, the vehicle suspension system must be ensures that the wheels of the vehicle always grip the road while in a state of braking, cornering and driving on uneven road conditions.

In previous studies, the study of vibration on the human body and vibration in the vehicle body has ensured an increase in improvements in the design of the vehicle suspension system to ensure passenger comfort in vehicles is improved. (Pazooki et al, 2012b).

Besides that, the automotive suspension systems design is involves a number of trades off. (Hrovat, 1982), (Elmadany and Abduljabbar, 1989) and (Appleyard and Wellstead, 1995). The suspension system that provides ride comfort have low spring rates, low shock absorber rates and will produce an excessive vertical suspension movement. The suspension system that provides a good control drive has a high spring rate, high levels of shock absorbers and will give the small vertical movement of the suspension.

This number of trades off is rising because the concept of passive suspension system is based on the function of spring to reserve the energy and the ability of damper to deplete it. The design and parameter of passive suspension is generally fixed, these parameters have been at a rate appropriate to the road conditions and driving style. (Tamboli and Joshi, 1999). The recent advances in force actuator, types of sensors, and high performance microprocessors have yields to a more advance suspension systems than can provide better performance than passive suspension system. (Fischer and Isermann, 2004).

2.2 Classification of Vehicle Suspension System

The vehicle suspension system can be divided into three types, namely passive suspension systems, semi active suspension systems and active suspension system. Passive suspension system as shown in Figure 2.1 consists of a set of springs and shock absorbers in which the spring acts as energy absorption and the shock absorber work to absorb movement of spring back to its original condition. The type of suspension that contains a set of springs and adjustable shock absorbers is referred to as semi- active suspension system. This type of suspension as shown in Figure 2.2 has an ability to diversify the shock absorbers controlled by a particular controller. An active suspension is based on passive components but it consists of actuators that can supply additional force to the system.

2.2.1 Passive Suspension System

Passive suspension system as Figure 2.1 is the most popular suspension system used in passenger and commercial vehicles. This type of suspension system is not too complex and requires minimum maintenance. The suspension system also does not require external forces in order to make this system function. The main components involved in passive suspension system are spring and shock absorber. Springs and shock absorbers is selected based on the load to be lifted, ride comfort and appropriate handling rates. In passive suspension system, type and specification shock absorbers play a major role in determining the needed driving patterns in addition to the spring rate in used.

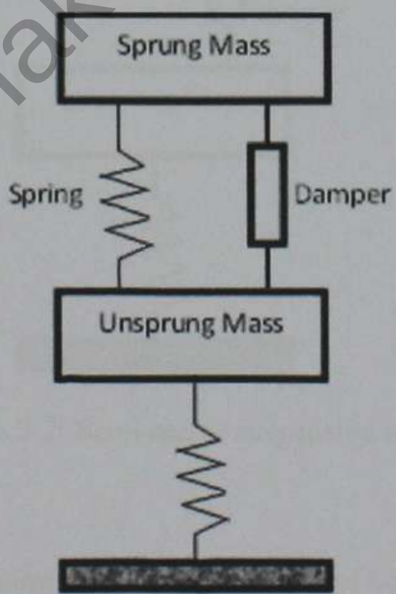

Figure 2.1: Passive suspension system

2.2.2 Semi Active Suspension System

Suspension system as shown in Figure 2.2 is only capable of varying the damper rate by adjusting the damping force dissipating energy is called semi active suspension system (Williams, 1994). To varying the damping shock, it can be achieve by utilizing an adjustable shock absorber with the design controller. In the semi active suspension system, there are two types of adjustable shock absorbers that are commonly used. The damper is magnetorheological shock absorber and adjustable orifice shock absorber.

Magnetorheological shock absorber is filling with the magnetorheological fluid in which it will change the nature of the magnetic field when electric current is passed by. The flow rate of the current is dependent on the design of the controller. (Yang et al, 2007); (Marin et al, 2004). This allows the shock absorbers to be controlled by controlling the magnetic field of MR fluid.

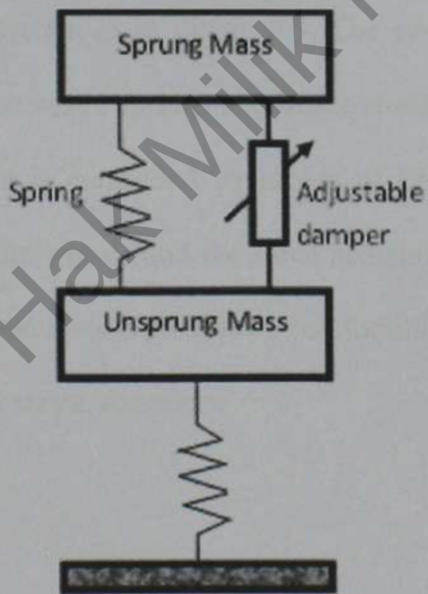

Figure 2.2: Semi active suspension systems

An adjustable orifice damper is the most useful of conventional semi active damper where it can vary diversify routes within the orifice tube shock absorbers. The variation of routes within the orifice tube shock absorbers can be controlled by control in on-off, by discreetly or continuously which resulting in on-off, by a fixed step or by continuous vary damping force characteristics (Fischer and Isermann, 2004).

2.2.3 Active Suspension System

The results of the current study on the system of suspension are leading to an active suspension system. This study increases because of its capabilities in an active suspension system that provides comfort of the ride and handling of the best. This occurs because the active suspension system consists of actuator force that works to reduce the movement of the vehicle body and reduce the acceleration in the vehicle body vertical movement. Crolla (1988) says that the active suspension system made up of two types.

Active suspension system made up of two types of active suspension system with soft and stiff active suspension system. These types of suspension are recognizing through the design of suspension system as in Figure 2.3. The system consists of springs, shock absorbers and actuator force where springs and shock absorbers are installed in series with force actuator. The system is controlled by means of the springs and shock absorbers control the movement of the wheels and the force actuator controls the movement of the vehicle body. In this way, the system provides a comfortable ride but the handling capacity depends on the springs and shock absorbers.

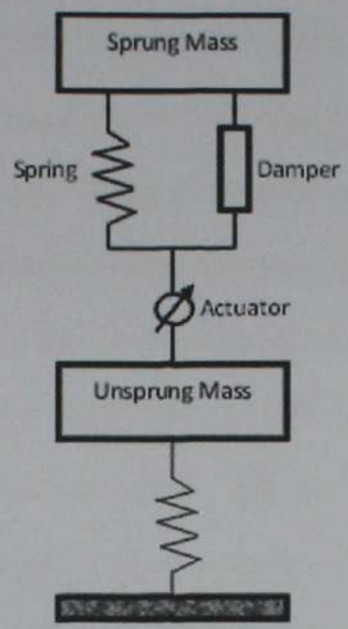

Figure 2.3: Soft active suspension system

Most researchers focus on the second type of suspension system as shown in Figure 2.4. This system is called stiff active suspension system. This system contains the same components as soft suspension system, but the arrangement design is different. Force actuator system is installed in parallel with the springs and shock absorbers. In this way it can control the movement of the vehicle body and wheel movements through the force actuator.

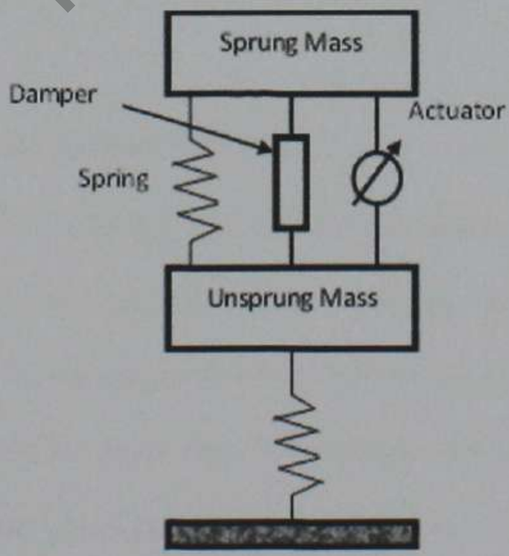

Figure 2.4: Stiff active suspension system

2.3 Actuator Selection in Active Suspension System

Most researchers have been focus in active suspension system in the recent years because of their abilities compared to the semi-active suspension system. For the force actuator in used in active suspension system, hydraulic systems are regularly used in both research and industrial field to realise the benefits of active suspension system. In a research, many researchers are employing hydraulic actuators with various control approach for validate their proposed system. The use of hydraulic actuator of active suspension system is more attractive than pneumatic actuator. This is because of the benefits of hydraulic actuator where they can provide huge forces with small cylinder compared to the pneumatic system. Even hydraulic system has a lot of advantages compare to hydraulic actuator, it has a major disadvantages that make it not favored to be used in the study application. The disadvantages of the hydraulic system are because of the expensiveness of the system. It also needs to consider the cost of the hydraulic fluid. Because of those disadvantages of the hydraulic system, the pneumatic system always favors to be used as the actuator in the active suspension study. This is because of the cheap cost to install in the vehicle, the no cost to pay for the air and the compressibility of the air that can convert to transmit the force.

2.4 Active Suspension Control Strategies

The vehicle active suspension system has been studies for modern control theory for a many decade. There are many published works that can be found as a resulted as many control strategies are proposed such as (Labaryade et al, 2004); (Kruczek et al, 2004); (Sergio et al, 2004). The papers that reviewed are been focus for control approaches and closely related with the use of active suspension system.

2.4.1 PID Controller

PID controller is a controller that is commonly used in the industry control system because of their simple structure, easy to implement and has a good performance in the wide range of system application (S.J. Bassi et al, 2011). PID controller is a controller that is commonly studied and used in the industry (Astrom and Hagglund, 2000).

PID controller is convenient to use in the application because the design is easy and simple but there is a drawback if the system is not linear. To solve this problem, this controller will be combined or modified with another controller to provide greater performance. In the study of active suspension system, PID controller has regularly been studied and used many times. Güçlü (2003) has study the dynamic behaviour of a non-linear 8 DOF vehicle suspension model equipped with active suspensions system and a passenger seat controlled by PID. Meanwhile, Kumar and Vijayarangan (2007) has been used PID controller to control an active suspension system for passenger vehicles to further improve ride comfort.

2.4.2 Linear Control

Linear control is a controller that has been studied and used in the control system . The controller normally uses the concept of optimal control in the active suspension system to control the force actuator. (Hrovat, 1997).

There are a number of approaches in linear control systems that are commonly used such as approach through linear quadratic regulator (LQR), linear quadratic Gaussian (LQG) and Loop Transfer Recovery (LTR). This approach is to minimize the linear quadratic cost function which is measured based on the current state and the inputs used. This approach has been studied by several researchers and tested in an active suspension system by Hrovat (1988); Tseng and Hrovat (1990); Esmailzadeh and Taghirad (1996); Sam et al. (2000) and Jialin Su et al. (2008).

2.4.3 Sliding Mode Control

Park and Kim (1998) have studied the sliding mode control system in an active suspension system on the model of a full range vehicle with multiple input and output and have been applied to the control system. In addition, an active suspension system of the quarter car model has also been studied. The systems that have been studied are using the vehicle stability system to control the entire four suspension system for full vehicle model. Active suspension model has been changed to the regular form for applying the sliding mode control system such as that introduced by Decarlo et al. (1998). Results of the study showed a slight increase compared with linear control systems.

2.4.4 Hybrid Control

In the present the intelligent based controller such as genetic algorithm based controllers, Bayesian control, neural network and fuzzy logic has been applied to the active suspension system as an example (Feng et al, 2003) which has been used as a generic algorithm with PID paired with Fuzzy logic control, while (Ting et al. 1995) using the sliding mode control with fuzzy logic control for use in an active suspension system through the quarter car model. From the study conducted, it was found sliding mode control along fuzzy logic control will produce outstanding results. The results shows that the system studied further enhance driving comfort and performance. The weaknesses of the system that can be identified is the movement of the suspension worsen compared with ordinary sliding mode.

2.5 Summary

From this chapter, a number of studies related that to advanced active suspension system have been summaries. The selection of actuator types been used and the specification of control strategies that been applied to active suspensions have been focusing in related section. From the literatures, the active suspension system could offer high performance to other types of suspension system in terms of ride and handling performance based on the types of actuator been chosen and the development of control strategies. In this studies, the design and implementation of control strategy of active suspension system is using PID and multiorder PID controller through numerical (simulation) studies are detailed out in the forthcoming chapters.

CHAPTER 3

METHODOLOGY

3.1 Introduction

To understand the point at issue related to suspension control, the simplified vehicle model will be used for analysis. The early section of this chapter is to study the mathematical model characteristics of the simplified vehicle that will affect dynamic components of the suspension. The quarter-car model of equation of motion will be used as the ground for all further analysis. The validation of the mathematical model of quarter car suspension will be perform at second section by associating with the experiment data of instrumented quarter car test rig by Imaduddin (2010) as the benchmark.

In this study for the active suspension system, the control system which will be used as the controller is proportional integral derivative controller (PID Controller) and multiorder proportional integral derivative controller (MOPID Controller). PID controller is chosen because the PID controller is efficient in many operations. It is also easy to maintain and embedded in a real system (Chuang et al, 2006). A PID controller attempts to calculate an error value between the set point and the control variable by minimize the error. The proposed system is a control loop feedback mechanism that makes use a difference of error to trigger the control response. The next section of this chapter will focus on a development of PID controller model and simulation study of the PID controller by applies a number of multiple loading conditions. From the simulation, the result will be analysed and compare with passive system to verify and set as a benchmark.

The final section of this chapter will studies on a control strategy of active suspension controller based on known as multiorder proportional integral derivative control (multiorder PID control). The control system used a closed feedback loop that employs a number of PID controllers to generate the control action to actuator. These control strategies are used to make a comparison studies in order to optimized ride comfort between PID controller and multiorder PID controller. The studies will focus on feedback of the controller by comparing the results of the movement of the vehicle body vertically and acceleration of the vehicle body vertically by relying on ride comfort. This section also highlights multiorder PID modelling and simulation with the input of different loading conditions. The output results from this the study will be analysed, compared and discuss with other PID controller and passive system for a verification and benchmarking.

3.2 Quarter Car Suspension Modelling

Well-known model for simulating one-dimensional vehicle suspension performance is by deriving a single wheel station or quarter car model. This model is setup using interconnections of springs, dampers and masses. The spring functions to support the sprung mass of the vehicle while the damper use in dispel the vibration energy and reduce the input from the road carry to the vehicle. The simulation using quarter car model is repeatedly used to solve problems related to suspension system (Hrovat, 1982), (Hrovat, 1990), (Abduljabbar and Elmadany, 1989), (Levitt and Mrad, 1994), (Ting et al. 1995), (Rao and Prahlad, 1997), (Roh and Park, 1999), (Yoshimura et al. 2001), (Hudha et al. 2005), (Lauwerys et al. 2005), (Huang and Chen, 2006) and (Türkay and Akçay, 2008). Quarter car model does not contain geometric effects of four wheels vehicle and does not represent the longitudinal interconnections. It also does not specify the problems related to handling but the quarter car model does enclose the element of real vehicle problem that related to vehicle suspension system. The quarter car model does show the better images of the problem of controlling vertical motions of wheel and body as well as suspension movement.

Two degree of freedom quarter car model contains the movement of unsprung section Z_u and the movement of sprung section Z_s. The road condition information is express by Z_r and the differential equations of quarter car Passive Suspension System Model according to Figure 3.1 are given by

$$M_u\ddot{Z}_u + K_t(Z_u - Z_r) + K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) = 0 \tag{3.1}$$

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) = 0 \tag{3.2}$$

where M_u represents the wheel mass or unsprung mass, M_s is the body mass or sprung mass, C_s is the rigidity of the damper, K_s is the rigidity of the spring and K_t is the rigidity of the wheel.

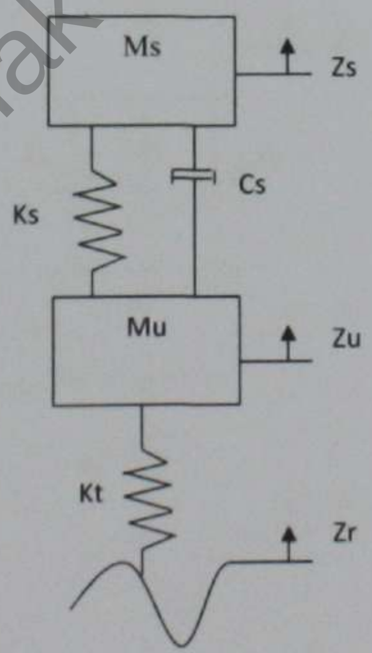

Figure 3.1: Quarter Car Passive Suspension System Model

The wheel is shown as a spring since the damping in the rolling wheel is typically very low and ignored in this analysis. It is assumed that the tire react as a point-contact follower that is always in contact with the road at all times. In this study, the effect of friction is ignored so that the damping structure is not taken into account in the review of the quarter car suspension model.

The quarter car active suspension model also consists of two vertical degrees of freedom. The different between the passive suspension and the active suspension is only at the additional force from the actuator F_p. The equations of active suspension model for 2 degrees of freedom (DOF) according to Figure 3.2 is

$$M_u\ddot{Z}_u + K_t(Z_u - Z_r) + K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) - F_p = 0 \qquad (3.3)$$

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) + F_p = 0 \qquad (3.4)$$

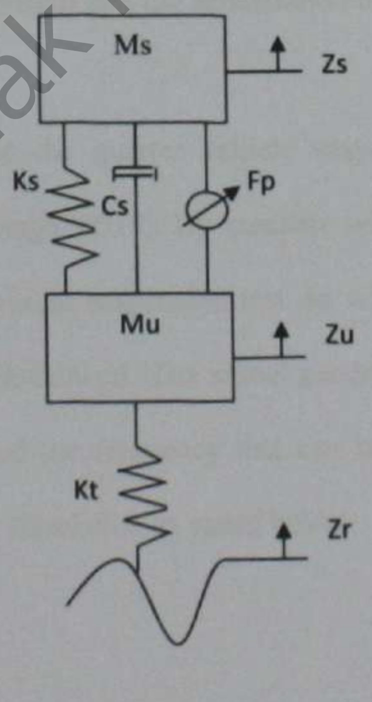

Figure 3.2: Quarter Car Active Suspension System Model

3.3 Quarter Car Suspension System Model Validation

Validation in this study of quarter car model is done by using the experiment data of Imaduddin (2010) as a benchmark. The trend of simulation data is compare by pattern comparison of graph result tendency of experiment data from Imaduddin (2010) using the same parameter conditions. Validation is described as the differentiation between the simulation graph trend and the experiment result. This means, the validation does not mean a graph similar to the graph of simulation and the experiment but a consistent pattern between simulation and experiment is important. The result of the test is also used to validate if the input parameter that been used is correct.

3.3.1 Quarter Car Model Validation Procedures

The validation of quarter vehicle model dynamic response characteristics such as vehicle vertical body displacement and vehicle vertical body acceleration, are perform thru a test involving sinusoid profile. The goal of this research is to improve driving comfort, the deflection of the suspension system and the acceleration of the wheel is not taken into account in this study.

The procedure to validate the quarter vehicle suspension model in difference frequencies was proposed by Ikenaga (2010). By compare with the experiment data from Imaduddin (2010), the quarter vehicle suspension test rig will generated sinusoidal road profile by using a slider-crank mechanism. The signal generated by this mechanism will produce the continuous signal and the frequency that can be adjusted. Parameter of the quarter vehicle suspension model simulation as stated below:

$M_u = 30$ kg

$M_s = 150$ kg

$C_s = 1{,}000$ Nsec/m

$K_s = 37{,}500$ N/m

$K_t = 100{,}000$ N/m

In order to use the experiment data from Imaduddin (2010). The parameter of simulation will be simulated using the parameter of Imaduddin (2010). These parameters been optimized to provide the same pattern as experiment data. The procedures of optimization as follows: Firstly the experiment data graph will be taken and the optimized parameter result from the simulation will be compared. Secondly, the parameter will be optimized in simulation of Matlab/Simulink until the result brings the same pattern of experiment data. Finally the result will be compared and discussed. The validated model of quarter car suspension will then be used to develop the proposed controller to control the actuator of active suspension in the next chapter.The discussions are shown in the next chapter.

3.4 Active Suspension PID Controller Design

A typically use structure of PID control system as shown in Figure 3.3. Error signal is used to generate signals for the proportional, integral and derivative thus generates signals for controlling quarter car suspension model.

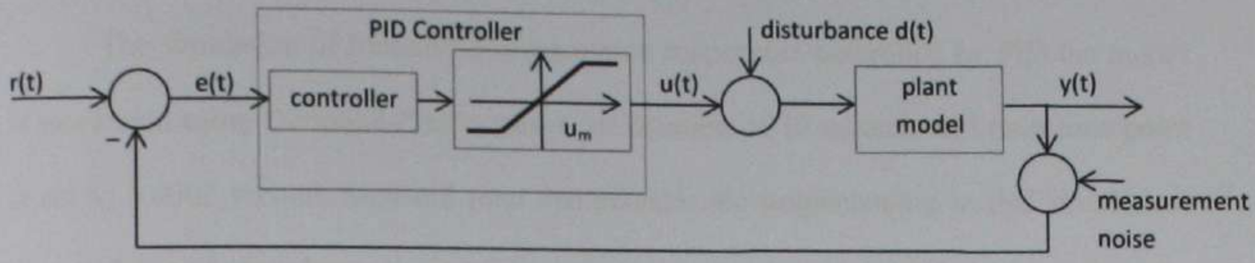

Figure 3.3: Overall control structure of PID controller

A PID controller is separate into a three system that consists of proportional compensation, integral compensation and derivative compensation. The main purpose of the proportional compensator is to produce the gain by comparing to the system output and input that is proportional to the error reading. The purpose of integral compensation is to introduce the integral of the error signal and then multiplied by a gain. The area under the curve of error signals will bring effect to the output signal. This will improve the steady-state error of overall closed-loop controller system. The derivative compensation will introduce the error signal of derivative multiplied by a gain. The main purpose of derivative compensation is to improve the transient response of the overall closed-loop controlled system. For a PID controller, the mathematical model of the system is:

$$u(t) = k_p e(t) + k_i \int_0^t e(\tau)d\tau + k_d \frac{de}{dt}, \qquad (3.5)$$

The input signal to the plant model is $u(t)$, $e(t)$ is the error signal designated as $e(t) = r(t) - y(t)$, and $r(t)$ is named as the input signal reference.

3.4.1 Active Suspension PID Controller Simulation Parameters

The simulation of Matlab/Simulink active suspension controlled by PID the model is monitored using Dormand-Prince solver for duration of 10 seconds and each time point is set to 0.0002 second. Sinusoid road disturbances are implementing in this simulation. Sinusoidal road disturbances are used to show the actual road condition. To review the performance of active suspension system that uses a PID control, frequency suspension system is divided into 3 areas. The first area is the natural frequency at the bottom of the vehicle body, the second is the natural frequency between the vehicle body and wheel and the third area is the natural frequency above the wheel of the vehicle. For suspension system considered in these studies and the suspension in common light passenger vehicle, the value of the vehicle body natural frequency is around 2 Hertz, while the vehicle wheel natural frequency is around 10 Hertz. The frequencies of 0.5 Hertz, 5 Hertz, and 15 Hertz is selected as sinusoid road profiles to use in this simulation for represent the frequency of road disturbance in natural frequency at the bottom of the vehicle body, the natural frequency between the vehicle body and wheel and the natural frequency above the wheel of the vehicle. (Hudha et al. 2005).

Quarter car model parameter is same with the validated model in the early section. The parameters of PID controller are list in Table 3.1.

Table 3.1: The parameters of the PID controller

Parameter	Value
K_P	17155
K_I	-515
K_D	375

3.5 Active Suspension Multiorder PID Controller Design

A multiorder PID controller structures is shown in Figure 3.4. The control structure of multiorder PID control system utilized two different control loops. The loops named inner loops and outer loops. The vehicle controller is called outer loop controller, it is used to avoid the interruption of input of sinusoidal shape and to reduce irrelevant vehicle movements. Controller for the actuator referred to as internal loop controller so that the power generated from actuators is controlled so that target force produced is as close as possible. For the outer loop controller, the input is a condition of the vehicle, which vehicle body movement, velocity and acceleration of the vehicle body vehicle body in which the output of the outer loop controller is an input of the actuator.

The named for the variable of multiorder PID controller for the zero-order variables are the vertical movement of the vehicle body, first-order variable is the vertical velocity of the vehicle body and the second-order variable is the vertical acceleration of the vehicle body. All the required values for the variables are placed in zero to prioritize driving comfort.

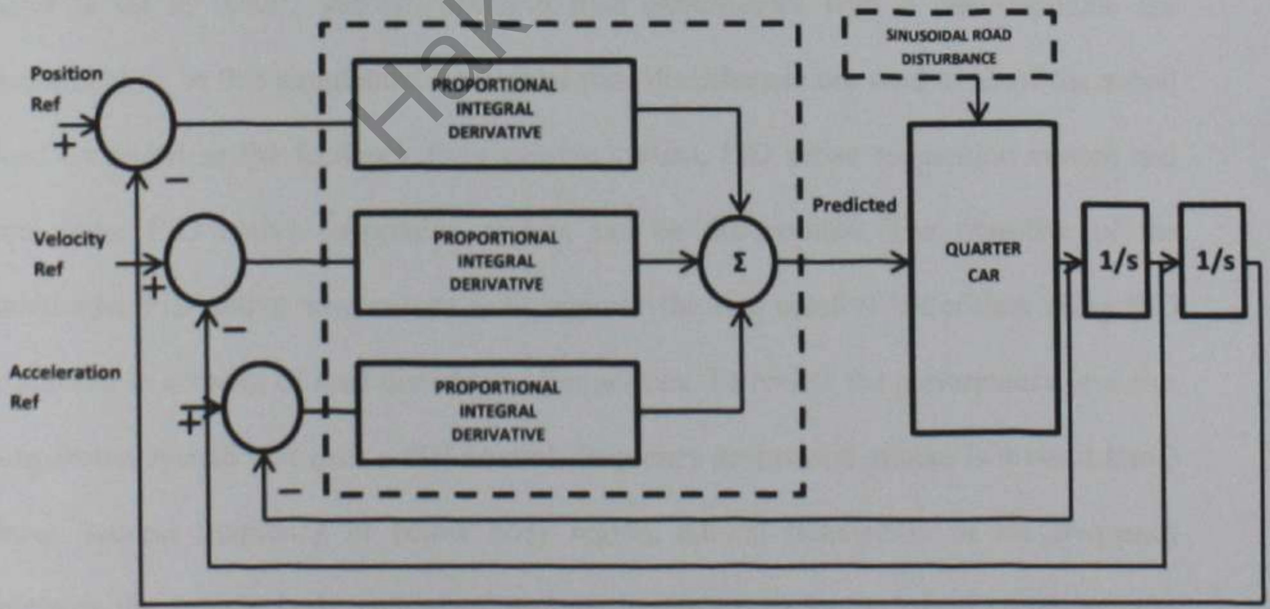

Figure 3.4: Structure of multiorder PID control

From Figure 3.4, the output value for predicted force to active suspension quarter car model is given by

$$\sum F = e_z\left[K_P + \frac{K_I}{s} + K_D s\right]_z + e_{\dot{z}}\left[K_P + \frac{K_I}{s} + K_D s\right]_{\dot{z}} + e_{\ddot{z}}\left[K_P + \frac{K_I}{s} + K_D s\right]_{\ddot{z}} \qquad (3.6)$$

with $e_z, e_{\dot{z}}$ and $e_{\ddot{z}}$ is the error signal of body displacement, body velocity and body acceleration. K_P, K_I and K_D are the proportional constant, the integral constant and derivative constant of the PID controller.

The main purpose for the choice for this controller is to enhance the ride comfort of the vehicle while compare the performance with PID controller.

3.5.1 Active Suspension Multiorder PID Controller Simulation Parameters

The simulation of Matlab/Simulink active suspension controlled by multiorder PID model is monitored using Dormand-Prince solver for duration of 10 seconds and each time point is set to 0.0002 second. Sinusoid road disturbances with 4 cm amplitude are implementing in this simulation. Sinusoidal road disturbances are used to show the actual road condition so the feedback from passive system, PID active suspension system and multiorder PID active suspension system can be differentiate. The objective of the multiorder PID active suspensions is to improve the ride comfort better than using PID controller in a varies of road disturbance frequencies. To review the performance of active suspension system that uses a PID control, frequency suspension system is divided into 3 areas, natural frequency of below body region, natural frequencies of the frequency between the vehicle body and wheel and the frequency above the wheel of the vehicle. Suspension considered in these studies and the suspension in common light passenger vehicle, the value of the vehicle body natural frequency is around 2 Hertz, while the vehicle wheel natural frequency is around 10 Hertz. The frequencies of 0.5 Hertz, 5 Hertz, and 15 Hertz is selected as sinusoid road profiles to use in this simulation for represent the frequency of road disturbance in natural frequency at the bottom of the vehicle body, the frequency between the vehicle body and wheel and the frequency above the wheel of the vehicle. These performance evaluations are using the same methods as PID Controller simulation referred from Hudha et al. (2005).

For this study, the parameters used are based on the same parameters as the quarter car model that has been validated at the beginning of this chapter. The parameters of PID controller are list in Table 3.2.

Table 3.2: The parameters of multiorder PID controller

Parameter	Value		
	Displacement	**Velocity**	**Acceleration**
K_P	14671	3026	811
K_I	84163	15466	3984
K_D	1000	50	10

3.6 Performance Evaluation of PID and Multiorder PID Controller

The performance evaluation of the PID and multiorder PID controller will be shown in frequency domain analysis graph and time domain analysis graph. In frequency domain analysis, bode plot analysis is perform. Bode plot of frequency domain response compare between controller active system compared with passive system for body vertical displacement and body vertical acceleration. For PID controller the comparison is made between passive suspension system and PID control active suspension system and for multiorder PID controller suspension system, the comparison is shown between passive system, PID control active suspension and multiorder PID control active suspension system. The responses of body natural frequency of suspension system are shown in the next chapter.

In time domain analysis, the time domain response is splitting into natural frequency of below body region, natural frequency at the bottom of the vehicle body, the frequency between the vehicle body and wheel and the frequency above the wheel of the vehicle. Vehicle body natural frequency is around 2 Hertz, while the vehicle wheel natural frequency is around 10 Hertz. The frequencies of 0.5 Hertz, 5 Hertz, and 15 Hertz are selected as sinusoid road profiles to use for performance evaluated in terms of Root-Mean-Square (RMS). The performance will be compared with the passive suspension system, PID control active suspension system and multiorder PID control active suspension system.

Root Mean Square can be described for a steadily fluctuating function as integral of the squares root of the rapid values during a cycle of the waveform. For a set of n values $\chi_1, \chi_2, \dots \chi_n$, the Root Mean Square is given by:

$$x_{RMS} = \sqrt{\frac{1}{n}\left(x_1^2 + x_2^2 + \dots + x_n^2\right)} \tag{3.7}$$

The Root Mean Square for a steadily fluctuating function or waveform *f(t)* defined over the interval $T1 \leq t \leq T2$ is given by:

$$f_{RMS} = \sqrt{\frac{1}{T_2 - T_1} \int_{T_1}^{T_2} [f(t)]^2 \, dt} \tag{3.8}$$

When two sinusoidal road profile is compare, one set from the simulation of passive suspension and one set from the active suspension simulation controller function, the RMS variation of the two graph is the present of amount of how far on regular the error from 0. Accordingly, the RMS of the variation is the measure of the error. Therefore this data can be used to compare the performance of the design controller in active suspension system. The graph and discussion will be shown on the next chapter.

CHAPTER 4

RESULT AND DISCUSSION

4.1 Introduction

Based on the methodology of previous chapter, the study of optimization parameter for the quarter car suspension model has been simulated. The result of simulation graph will then be compared with the result from experiment and simulation from Imaduddin (2010). The comparison of the result will be discussed later in this chapter. Quarter car model that has been validated to be used to assess the effectiveness of controls in an active suspension system

In this first chapter, the performance of PID controller and multiorder PID controller will be discussed in term of frequency domain simulation. In this section the Bode diagram is simulated to show the feedback of the active controller in term of frequency domain.

The last section of this section will discussed the result of the performance comparison of the active suspension controller between passive system, PID controller and multiorder PID controller in term of time domain simulation. The performance will be discussed between the performance graphs of sinusoidal in terms of RMS.

4.2 Quarter Car Suspension Model Result of Validations

The validation of the vertical movement of the Car Body shown in Figures 4.1 to 4.5. The Result of Validation figures of Car Body vertical displacement shown into five different frequencies of 0.94 Hertz, 1.18 Hertz, 1.42 Hertz, 1.66 Hertz, and 1.89 Hertz.

From outcome of the results obtained shows the trends between experimental data and simulation result are identical and come with small error. There are misalignments at the early of the simulation and after 2 second the graph become stable and identical. Overall reading of Car Body vertical displacement follows the same pattern of experiment data and simulation data from Imaduddin (2010).

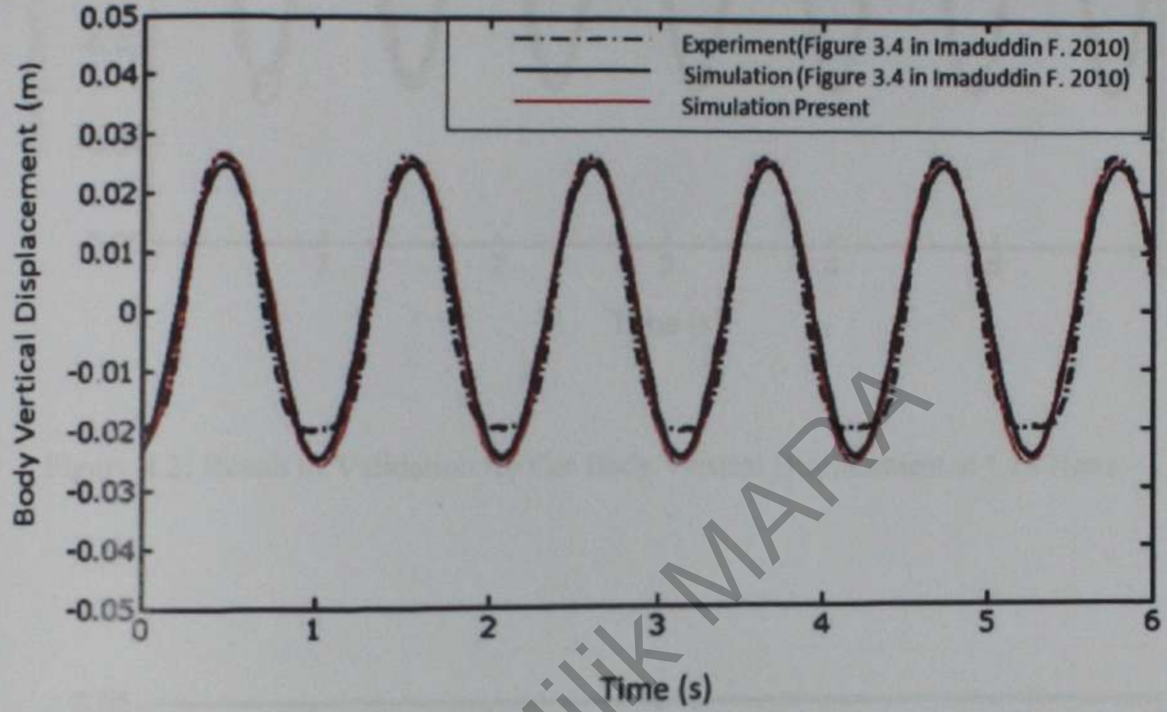

Figure 4.1: Result of Validation for Car Body Vertical Displacement at 0.94 Hertz

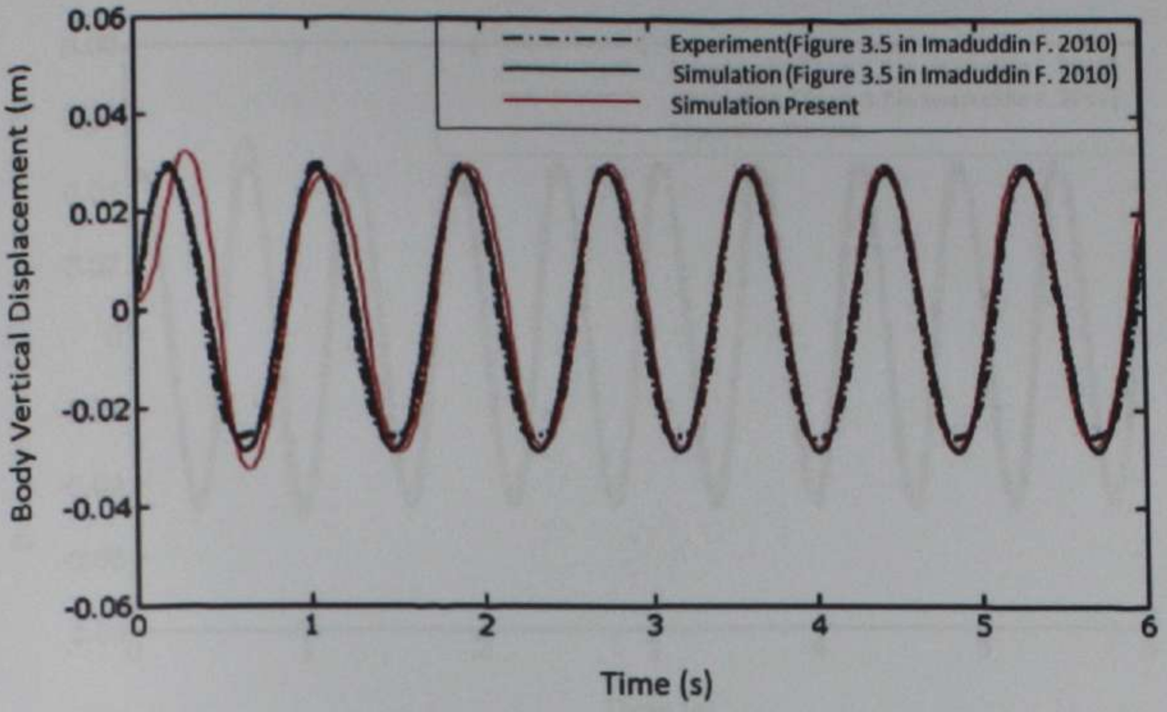

Figure 4.2: Result of Validation for Car Body Vertical Displacement at 1.18 Hertz

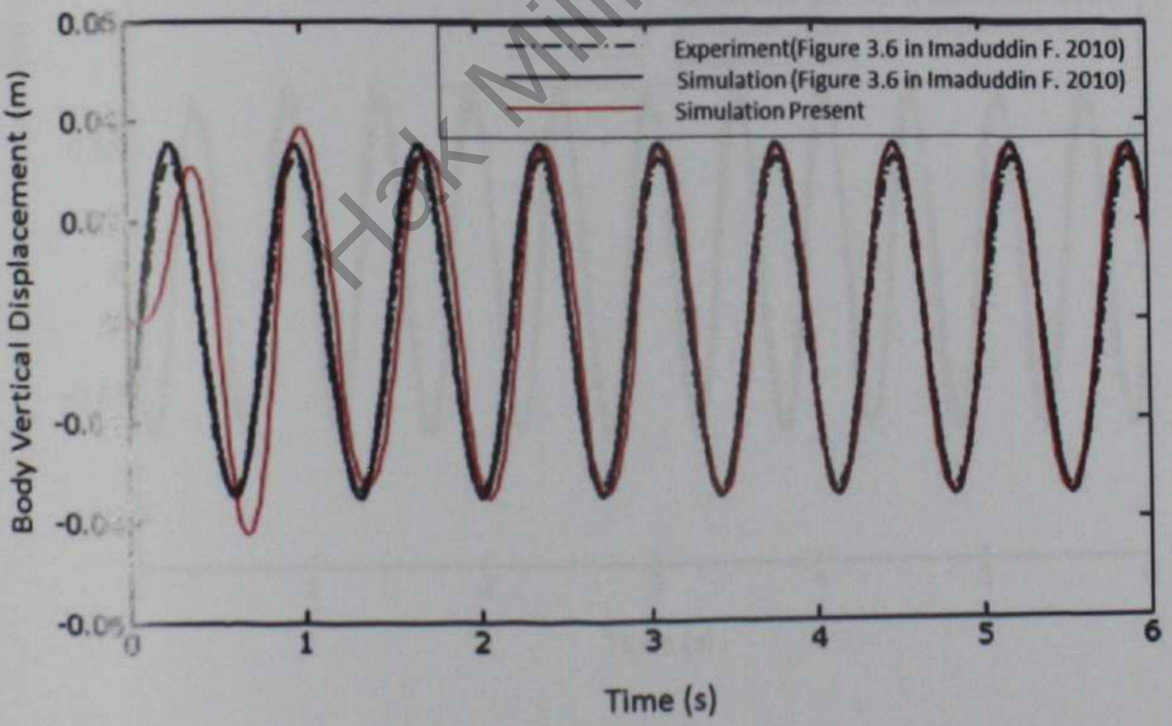

Figure 4.3: Result of Validation for Car Body Vertical Displacement at 1.42 Hertz

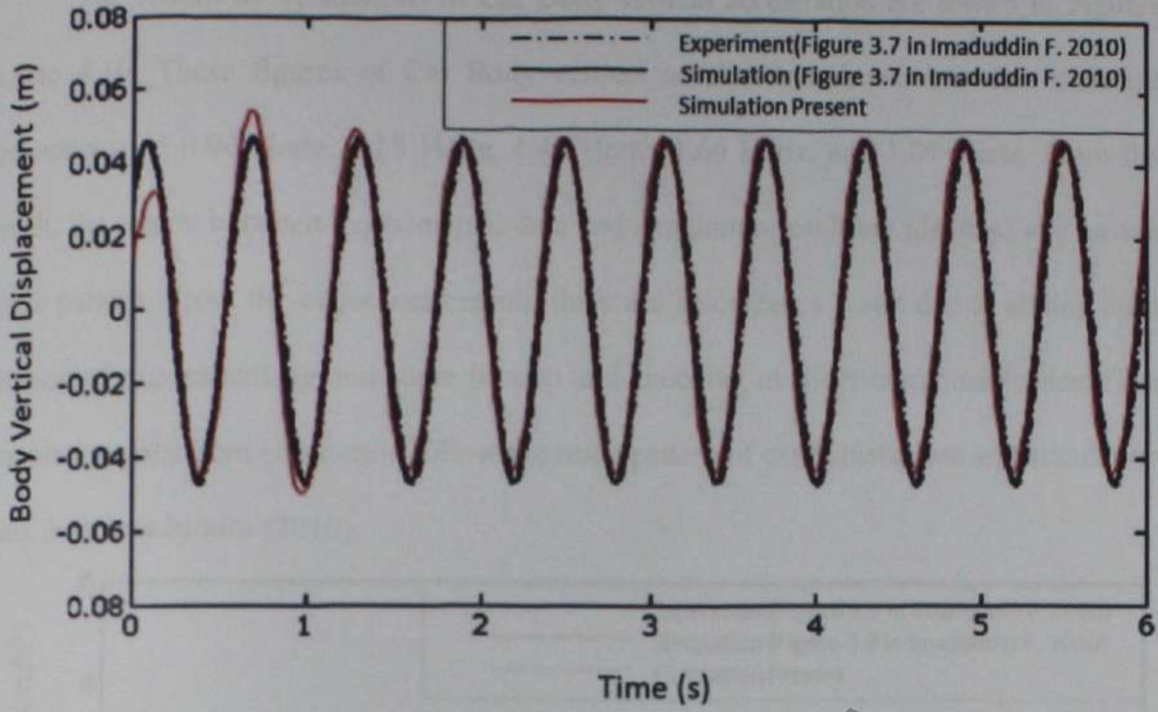

Figure 4.4: Result of Validation for Car Body Vertical Displacement at 1.66 Hertz

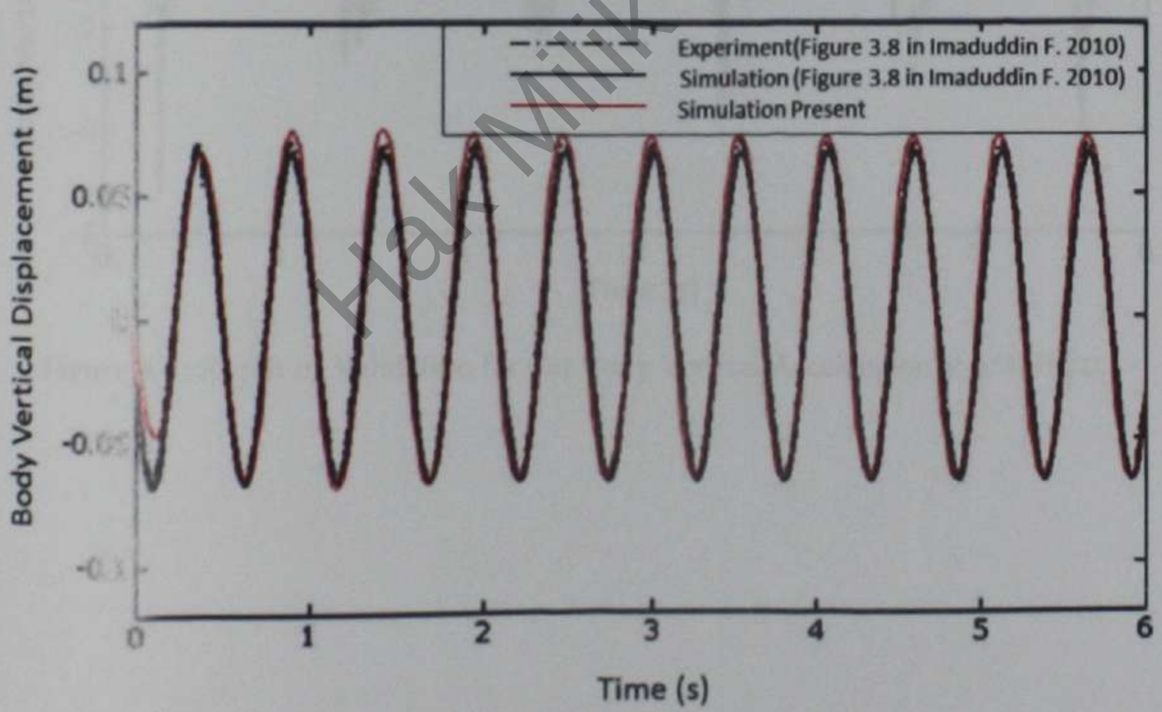

Figure 4.5: Result of Validation for Car Body Vertical Displacement at 1.89 Hertz

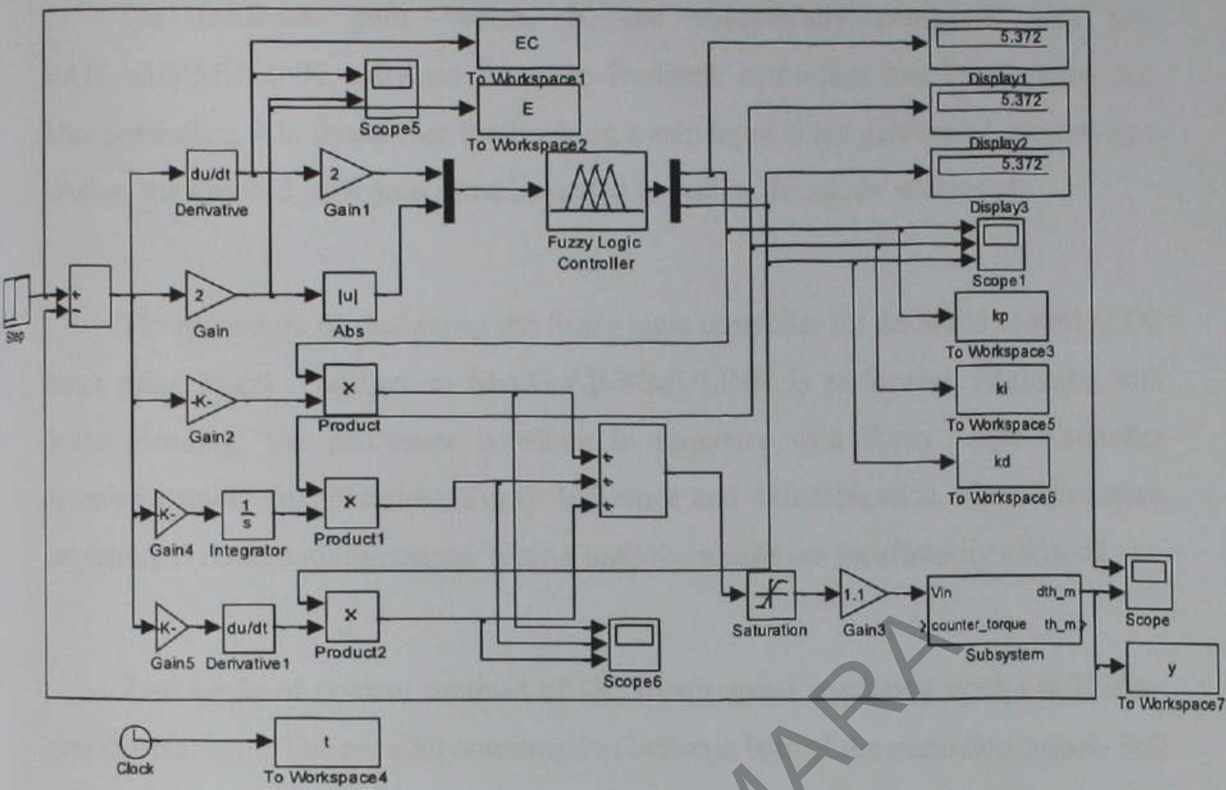

Figure 3.22 Simulation of Fuzzy PID control System Model

3.6 Summary

The PID controller design method with Ziegler-Nichols tuning (type 1) have been widely used in the process control systems where the plant dynamics are not precisely known. It is a time-domain method and very useful. The S-shaped curve had been obtained in order to get important parameter delay time, L and time constant, T. In designing a system using the PID approach, formula was suggested by Ziegler-Nichols that will give stable operation and fine tuning still required until an acceptable results is achieved. The simulation diagram for PID controller had also been successfully constructed.

Meanwhile, state feedback technique is chosen by which all desired poles can be selected at the start of the design process. The solving of Lyapunov Equation procedures are followed by considering proper matrix solutions.

The Result of Validations of Car Body vertical acceleration are shown in Figures 4.6 to 4.10. These figures of Car Body vertical acceleration shown into five different frequencies of 0.94 Hertz, 1.18 Hertz, 1.42 Hertz, 1.66 Hertz, and 1.89 Hertz. From the result, the trends between experimental data and simulation result are identical and have a same pattern. From the experiment result, there are inconstancy result due to sliding bush friction of experiment rig and some friction and knocking in slider-crank mechanism. The simulation data from simulation follow the same pattern of experiment data and simulation data from Imaduddin (2010).

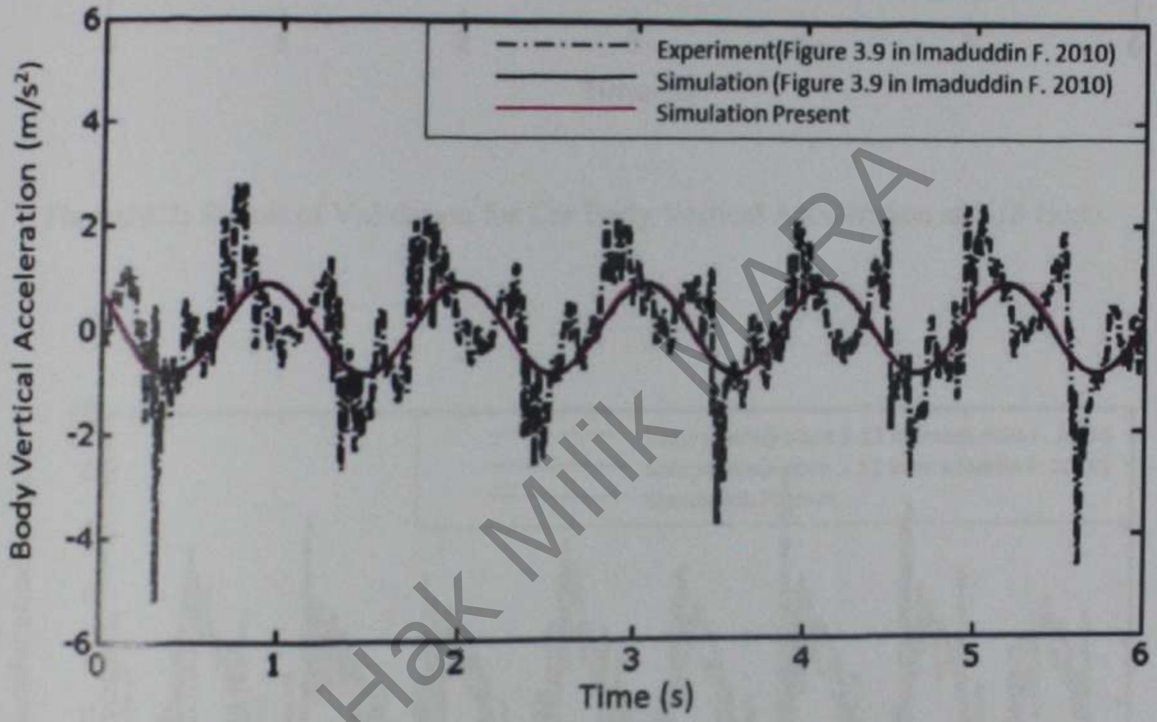

Figure 4.6: Result of Validation for Car Body Vertical Acceleration at 0.94 Hertz

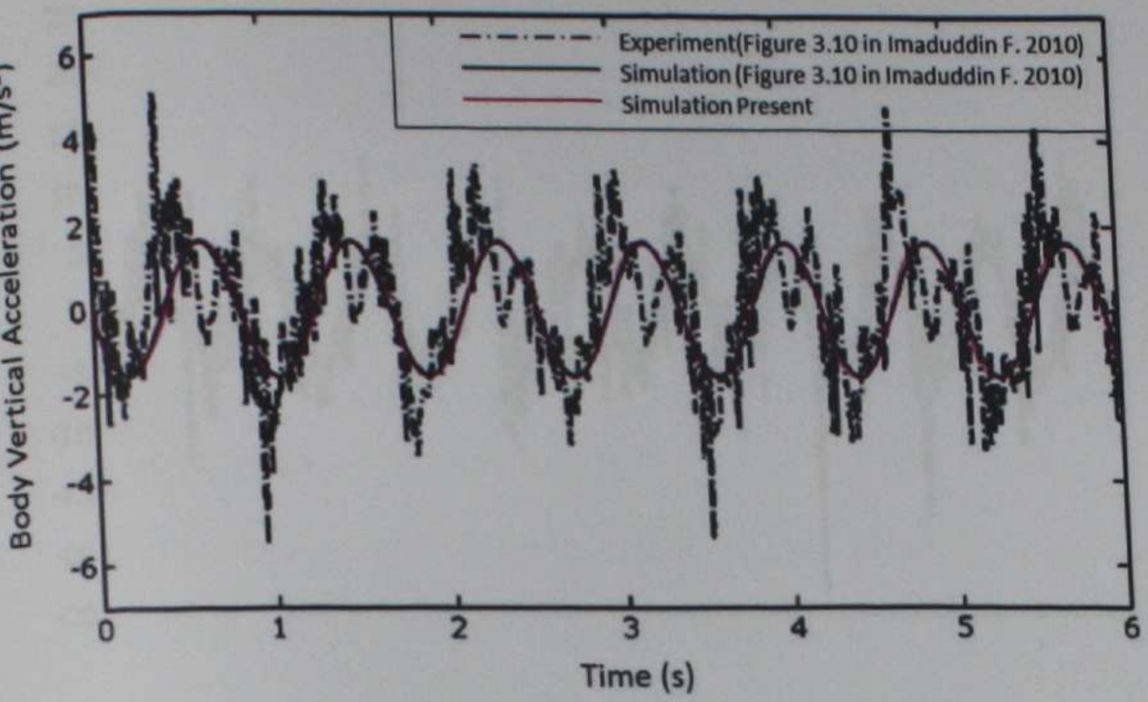

Figure 4.7: Result of Validation for Car Body Vertical Acceleration at 1.18 Hertz

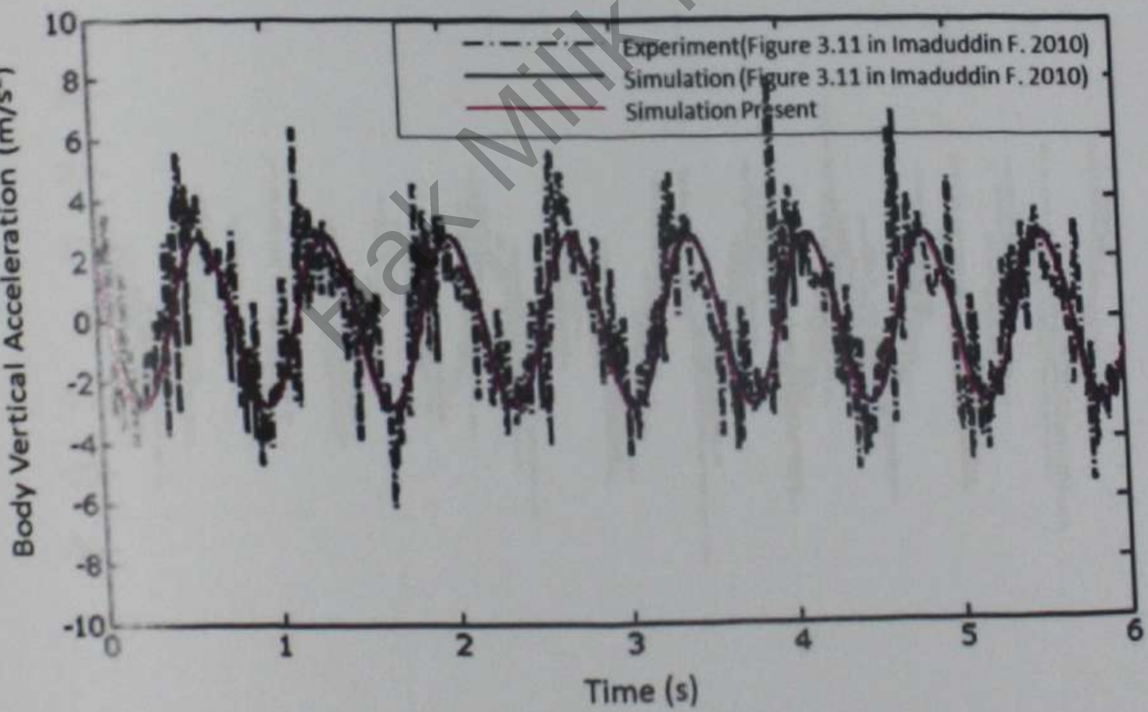

Figure 4.8: Result of Validation for Car Body Vertical Acceleration at 1.42 Hertz

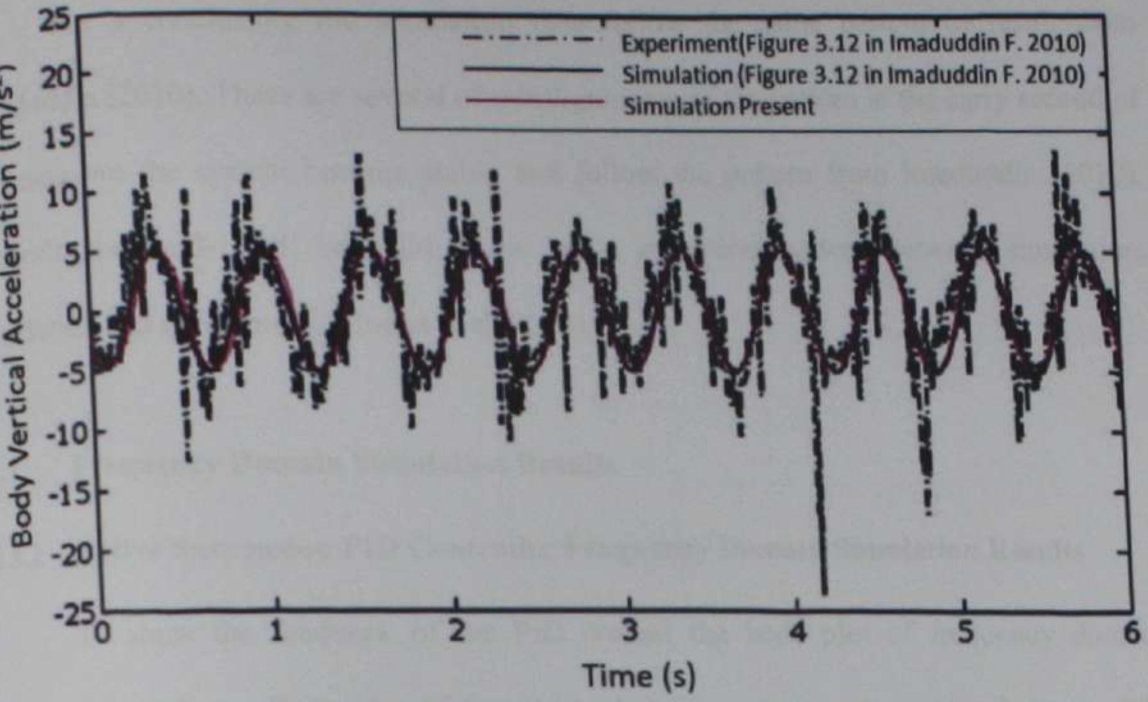

Figure 4.9: Result of Validation for Car Body Vertical Acceleration at 1.66 Hertz

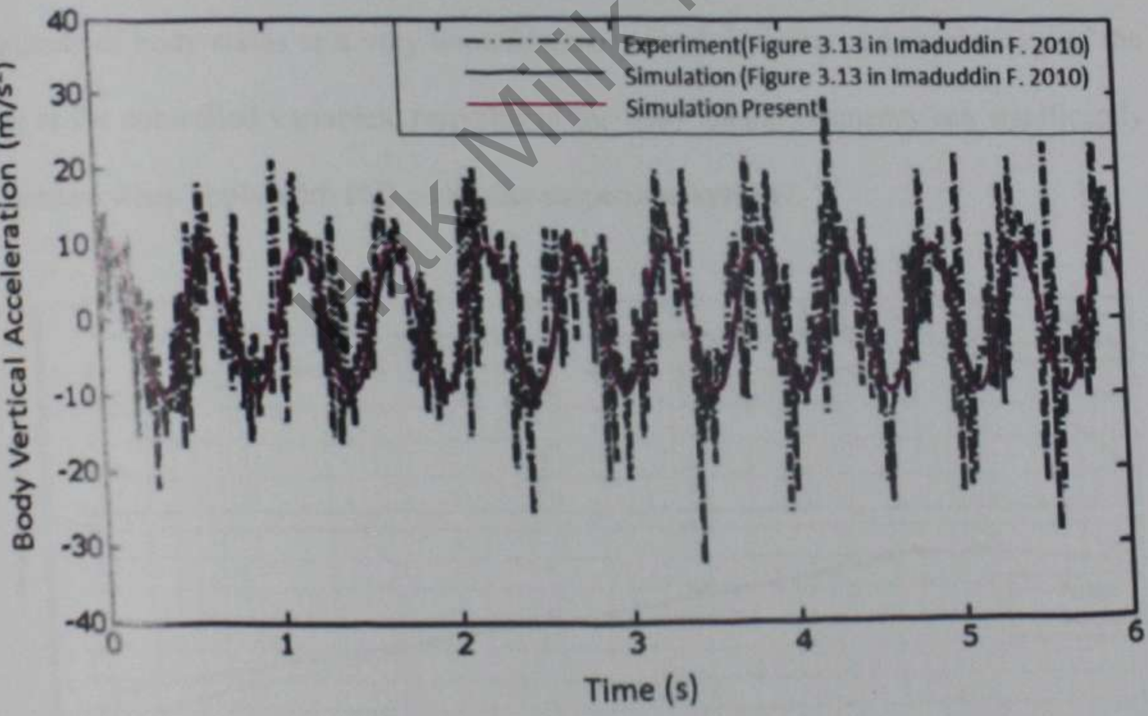

Figure 4.10: Result of Validation for Car Body Vertical Acceleration at 1.89 Hertz

CHAPTER 4

SIMULATION RESULTS AND DISCUSSION

4.1 Introduction

This chapter contains all of the results of the simulation mentioned in the previous chapters. For every controller simulation, there will be simulation to obtain improved motor speed regulation and transient response based on graph presented. In order to meet these objectives, simulation works using MATLAB with SIMULINK® is performed and the responses of the speed under various system parameters are illustrated. The graph represents the output speed of the DC motor measured in rad/s with refer to the time, t for stability analysis.

4.2 Simulation Using Open Loop of the Separately Excited Linear DC Motor

In this section, the simulation is carried out without the controller. The DC motor System simulated with and without Coulomb friction, F_C.

Figure 4.1 shows the simulation of the open loop nonlinear DC motor model. The simulation consists only the DC motor model with step input signal. Figure 4.2 shows the subsystem nonlinear model of the DC model that included all the important parameters as in equation (2.6) and equation (2.9).

For a conclusion, the simulation data follow the same pattern of result from Imaduddin (2010). There are several of misalignments of the pattern at the early second of the data but the system become stable and follow the pattern from Imaduddin (2010). Overall, the model will be valid if the has a consistent pattern between simulation, validation and experiment. (Hudha et al, 2009).

4.3 Frequency Domain Simulation Results

4.3.1 Active Suspension PID Controller Frequency Domain Simulation Results

To show the feedback of the PID control the bode plot of frequency domain analysis is perform. Bode plot of frequency domain response comparison between PID controller active system compared with passive system for movement of the vehicle body vertically and acceleration of the vehicle body vertically is shown in Figure 4.2. Based on the Figure 4.11, the system of an active suspension has successfully scale down the amplitude of body states at a very wide frequency band. Velocity and acceleration of the body as the controlled variables, response in the body natural frequency can significantly be reduced when apply with PID controller suspension system.

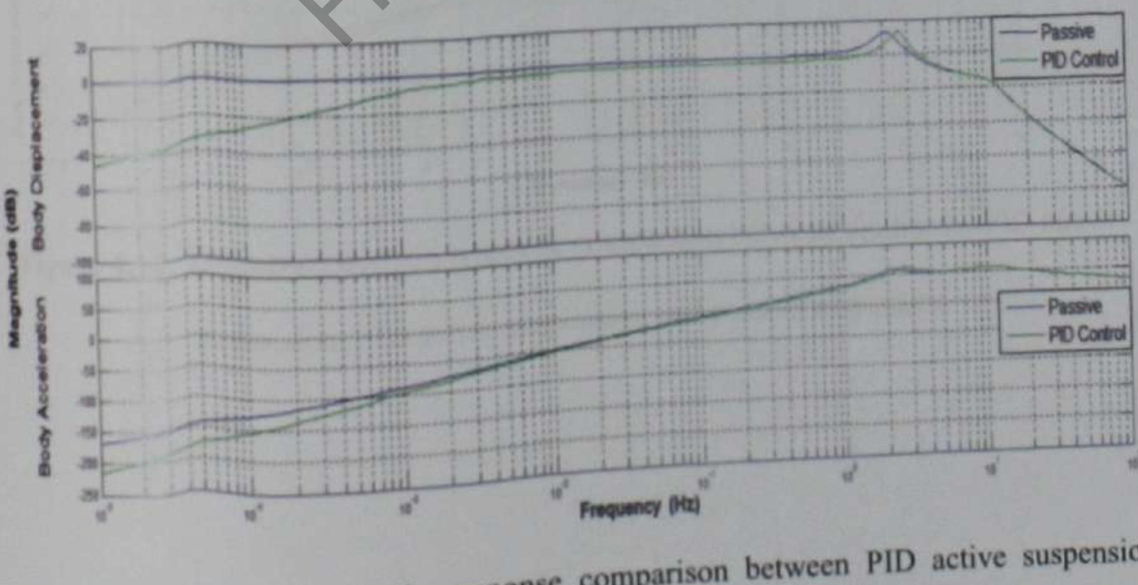

Figure 4.11: Frequency domain response comparison between PID active suspension systems and passive suspension

4.3.2 Active Suspension multiorder PID Controller Frequency Domain Simulation Results

To show the feedback of the multiorder PID control the bode plot of frequency domain analysis is perform. Bode plot of frequency domain response comparison between multiorder PID controller active system compared with passive system and PID controller active system for movement of the vehicle body vertically and acceleration of the vehicle body vertically are shown in Figure 4.12. From the figure, the multiorder PID active suspension system has successfully scale down the amplitude of body states at a very wide frequency band. Velocity and acceleration of the body as the controlled variables, response in the body natural frequency can be reduced more when apply with multiorder PID controller suspension system.

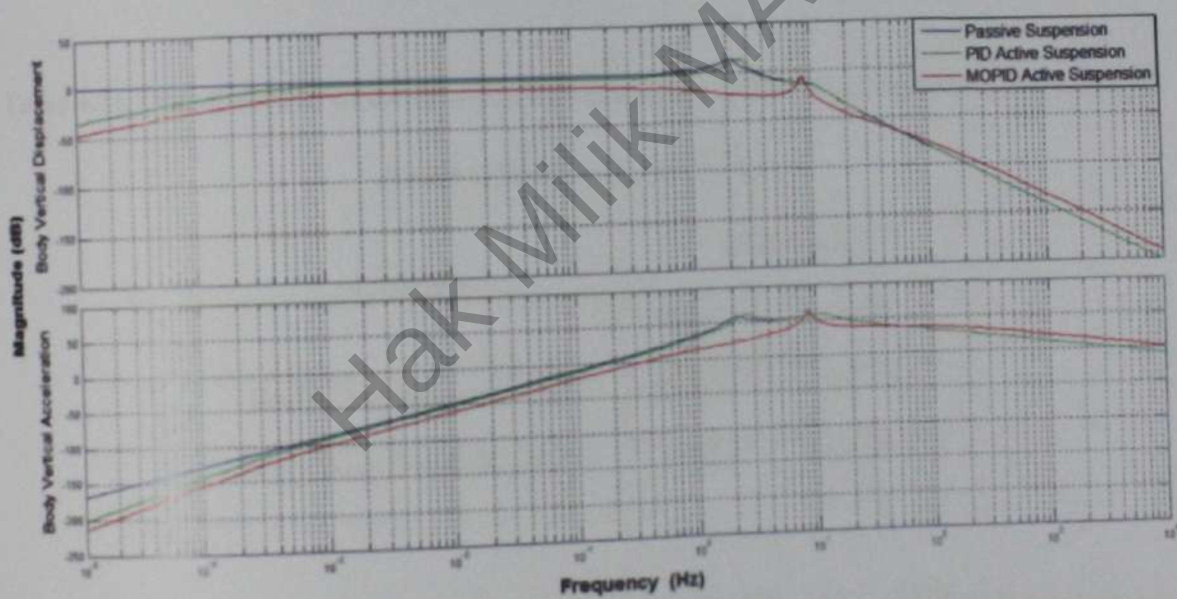

Figure 4.12: Frequency domain response comparison between multiorder PID active suspension, PID active suspension and passive suspension

4.4 Time Domain Simulation Results

4.4.1 Active Suspension PID Controller Time Domain Simulation Results

To identify performance improvement for PID control system in an active suspension system, time domain analysis is used. The time domain response is divided into 3 areas of natural frequency, natural frequency at the bottom of the vehicle body, the natural frequency between the vehicle body and wheel and the natural frequency above the wheel of the vehicle. The value of the Car Body natural frequency is around 2 Hertz, while the vehicle wheel natural frequency is around 10 Hertz. The frequencies of 0.5 Hertz, 5 Hertz, and 15 Hertz are selected as sinusoid road profiles to use for performance analysis of active suspension system in terms of Root Mean Square (RMS). The result of performance analysis as listed in Table 4.1, 4.2 and 4.3 as follows:

Table 4.1: Time domain response comparison between PID active suspension systems and passive suspension at 0.5 Hertz

Criteria	RMS value	
	Passive	**PID**
Body Displacement (m)	0.0424	0.0255
Body Acceleration (m/s^2)	0.4184	0.2515

Table 4.2: Time domain response comparison between PID active suspension systems and passive suspension at 5 Hertz

Criteria	RMS value	
	Passive	**PID**
Body Displacement (m)	0.0071	0.0055
Body Acceleration (m/s^2)	7.033	4.864

Table 4.3: Time domain response comparison between PID active suspension systems and passive suspension at 15 Hertz

Criteria	RMS value	
	Passive	**PID**
Body Displacement (m)	0.0018	0.00063
Body Acceleration (m/s^2)	15.68	6.116

The PID controller has reduced the RMS values of movement of the vehicle body vertically and acceleration of the vehicle body vertically. Tables 4.1, 4.2 and 4.3 shows the significantly reduced of RMS values of movement of the vehicle body vertically and acceleration of the vehicle body vertically is compare with passive system. Results from the study result, the introduction of an active suspension system in passenger vehicles can improve ride comfort.

The movement of the vehicle body vertically and acceleration of the vehicle body vertically from input of the 0.5 Hertz road disruption as in a Figure 4.13 and 4.14 which shown the natural frequency at the bottom of the vehicle body. By refer to the figure, the movement of the vehicle body vertically and acceleration of the vehicle body vertically after the PID active suspension system was apply are better compared to conventional passive suspension system. From the Figure 4.13, the Sinusoid input at the amplitude of 0.04 m is reduced to 0.03 m. The 25% reduce of body displacement after the Sinusoid input will reflex that the introduction of PID active suspension will improve the performance of the ride comfort in vehicle suspension. From the Figure 4.14, body vertical acceleration value of PID active suspension is decrease compared to passive suspension. This reducing value after the introduction of PID active suspension system will improve vehicle ride comfort compared with passive suspension system.

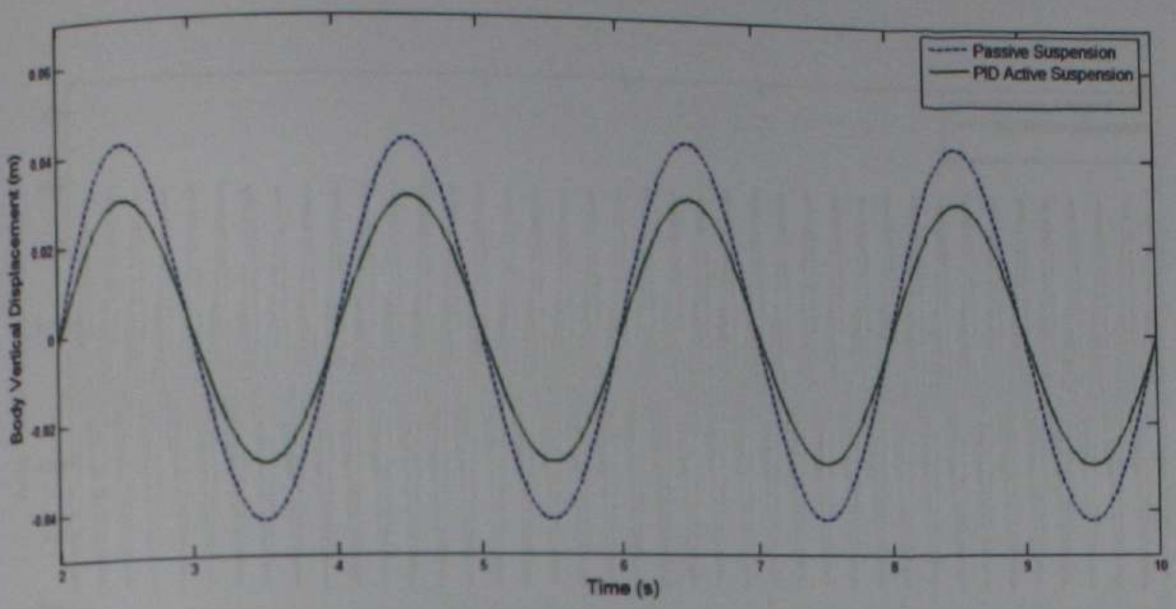

Figure 4.13: Body vertical displacement of 0.5 Hertz sinusoid road profile

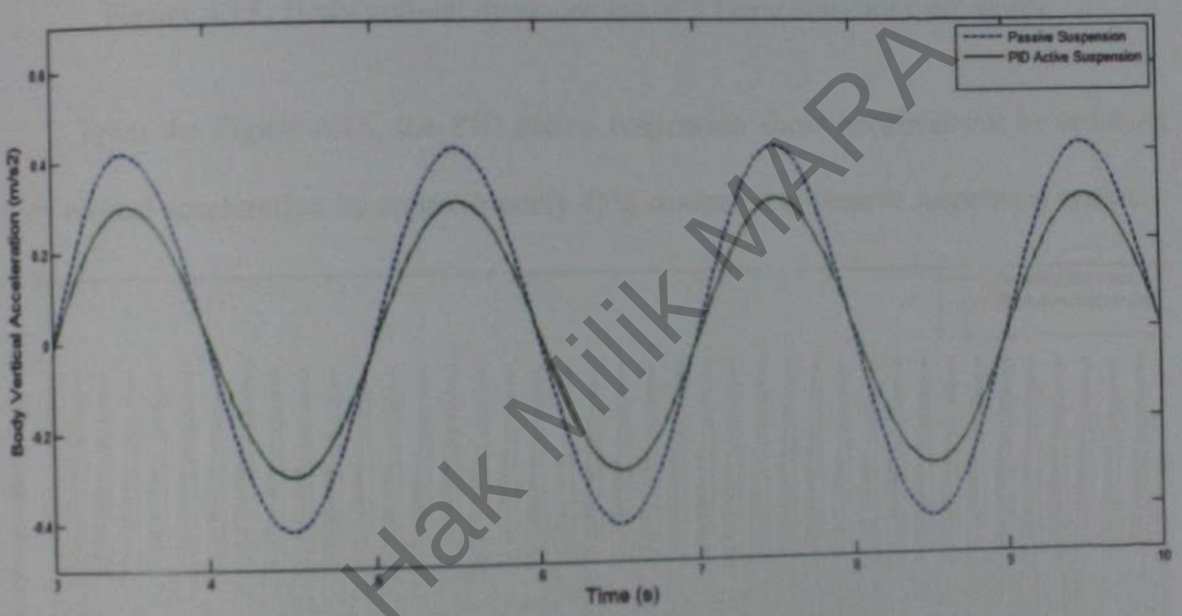

Figure 4.14: Body vertical acceleration of 0.5 Hertz sinusoid road profile

For the frequency in the region between body and wheel, natural frequencies at 5 Hertz is used to show the advancement of movement of the vehicle body vertically and acceleration of the vehicle body vertically. Based on Figure 4.15, which represents the figure of body vertical displacement, the active system shows significant improvement in reducing the displacement value by approximately 66% compared to passive suspension system.

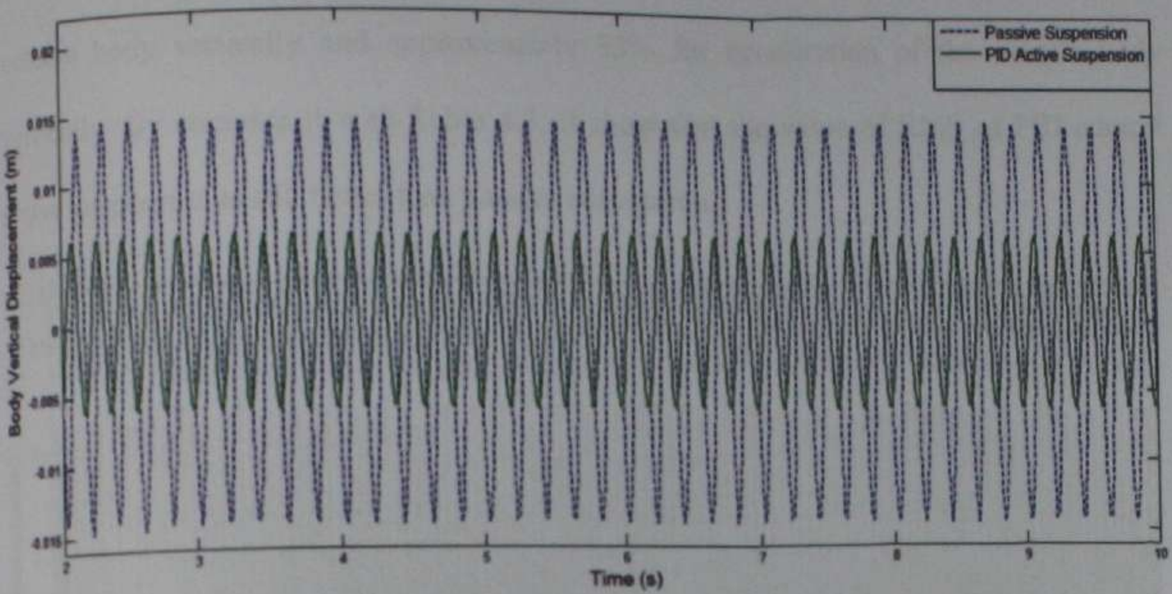

Figure 4.15: Body vertical displacement of 5 Hertz sinusoid road profile

From the Figure 4.16, the PID active suspension show improvement by reducing body vertical acceleration by approximately 45% compared to passive suspension system.

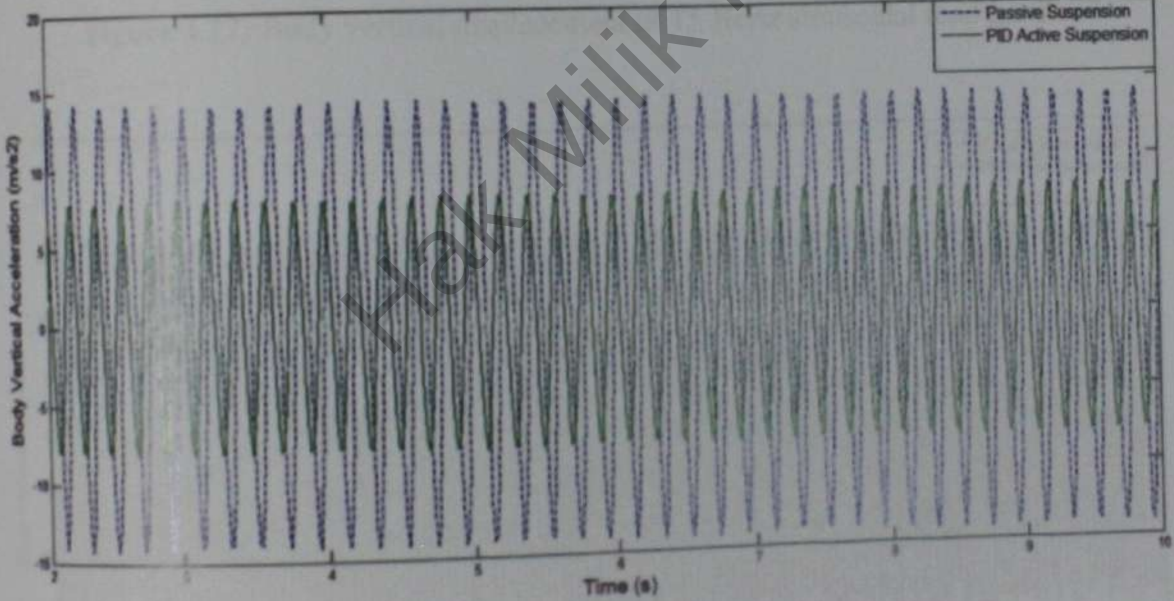

Figure 4.16: Body vertical acceleration of 5 Hertz sinusoid road profile

For the region above wheel natural frequency, the frequency of 15 Hertz is used. Enhancement for the movement of the vehicle body vertically and acceleration of the vehicle body vertically for are shown in Figures 4.17 and 4.18. At this region, the improvement at 15 Hertz show that the reduction nearly 50% of the movement of the vehicle body vertically and approximately 33% for acceleration of the vehicle body vertically. By compare it with Table 4.3, it show that the value of RMS of PID control active suspension is still better than passive suspension.

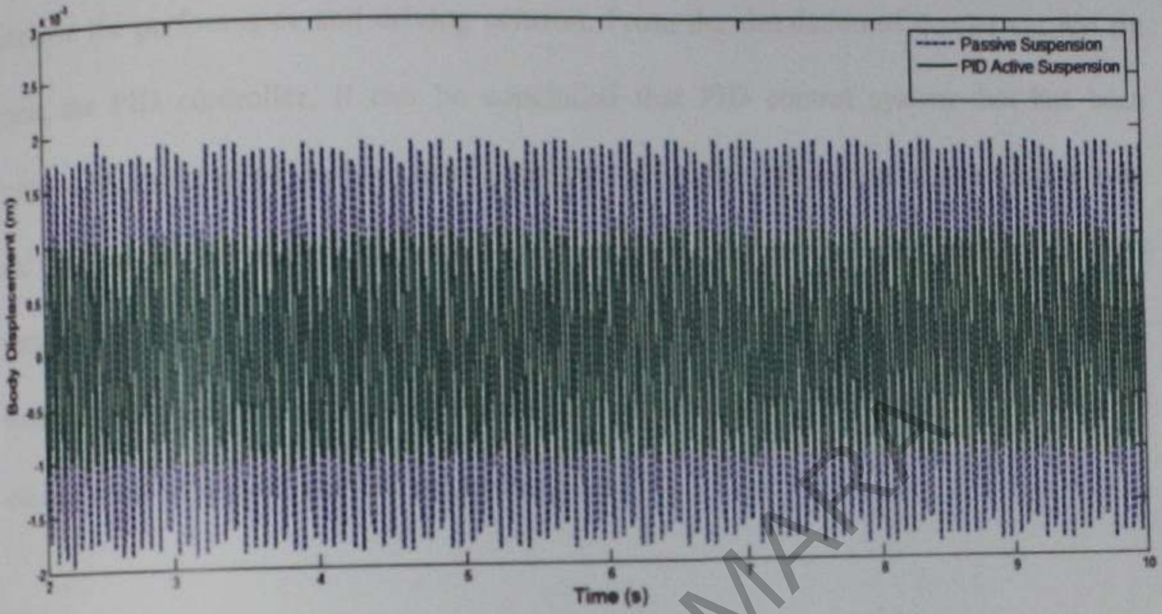

Figure 4.17: Body vertical displacement of 15 Hertz sinusoidal road profile

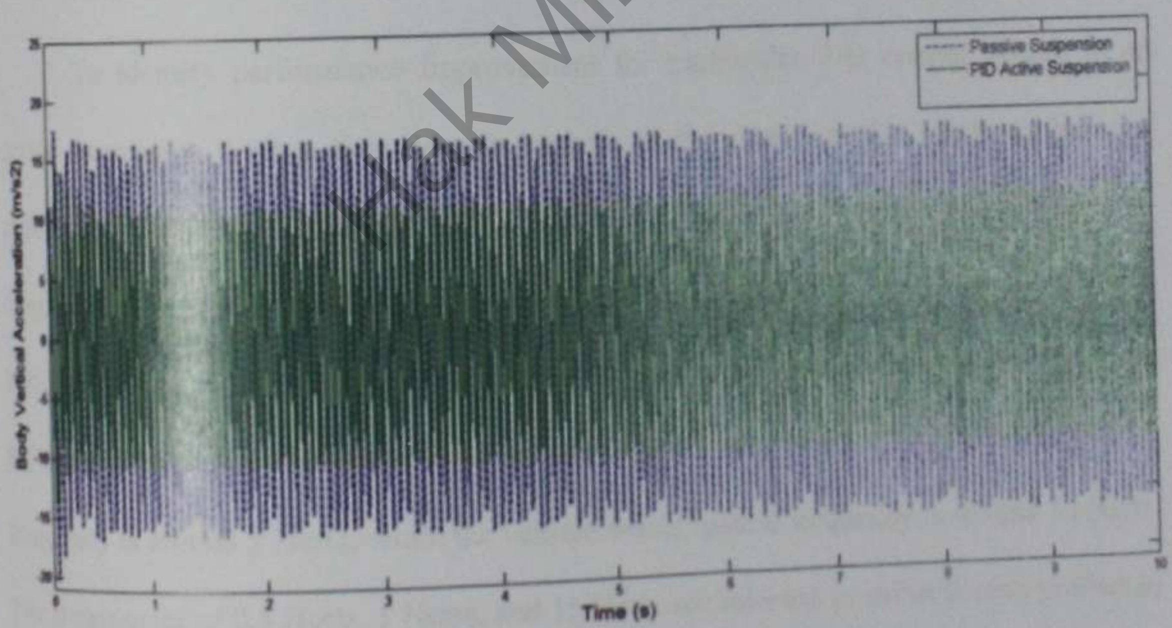

Figure 4.18: Body vertical acceleration of 15 Hertz sinusoidal road profile

A simulation of PID control for active suspension has been studies to investigate the performance of the proposed controller. These studies focus on the improvement of ride performance and ride comfort in the suspension system. Results of the study will focus on the vertical movement of the vehicle body and vertical acceleration of the vehicle body only. From the simulation results, the PID controller shows better performance compared to the conventional passive suspension system. The control system that been introduced to improve the performance and driving comfort. From the simulation of quarter car test rig using the PID controller, it can be concluded that PID control system that has been introduced has the good performance in control the vertical movement of the vehicle body and vertical acceleration of the vehicle body criteria. For the next section, the PID controller system will be compared to Multiorder PID controller system and passive system. It is expected that the Multiorder PID controller will improve the ride performance and ride comfort better than PID controller rides.

4.4.2 Active Suspension multiorder PID Controller Time Domain Simulation Results

To identify performance improvement for multiorder PID control system in an active suspension system, time domain analysis of the system is also compared between multiorder PID active controller, PID active controller and passive system. The time domain response is divided into 3 areas of natural frequency, natural frequency at the bottom of the vehicle body, the natural frequency between the vehicle body and wheel and the natural frequency above the wheel of the vehicle. The value of the Car Body natural frequency is around 2 Hertz, while the vehicle wheel natural frequency is around 10 Hertz. The frequencies of 0.5 Hertz, 5 Hertz, and 15 Hertz are selected as sinusoid road profiles to use for performance analysis of active suspension system in terms of Root Mean Square (RMS). The result of performance analysis as listed in Table 4.4, 4.5 and 4.6 as follows:

Table 4.4: Time domain response comparison between various systems for 0.5 Hertz

Criteria	RMS value		
	Passive	PID	Multiorder PID
Body Displacement (m)	0.0424	0.0255	0.0156
Body Acceleration (m/s^2)	0.4184	0.2515	0.2157

Table 4.5: Time domain response comparison between various systems for 5 Hertz

Criteria	RMS value		
	Passive	PID	Multiorder PID
Body Displacement (m)	0.0071	0.0055	0.0016
Body Acceleration (m/s^2)	7.033	4.864	1.623

Table 4.6: Time domain response comparison between various systems for 15 Hertz

Criteria	RMS value		
	Passive	PID	Multiorder PID
Body Displacement (m)	0.0018	0.00063	0.0001
Body Acceleration (m/s^2)	15.68	6.116	1.164

From the table above, the multiorder PID controller has reduced the RMS values of movement of the vehicle body vertically and acceleration of the vehicle body vertically better than PID control and passive system. Tables 4.4, 4.5 and 4.6 shows the significantly reduced of RMS values of body displacement and body acceleration compared with PID controller and passive system. Results from the study result, the introduction of multiorder PID in active suspension system in passenger vehicles can improve ride comfort more than PID active system.

Movement of the vehicle body vertically and acceleration of the vehicle body vertically feedback from the input of the 0.5 Hertz road disruption is shown in a Figure

4.19 and 4.20 which shown the natural frequency at the bottom of the vehicle body. From this figure, it shown that movement of the vehicle body vertically and acceleration of the vehicle body vertically when apply with multiorder PID active suspension vehicle system are better compared to PID active suspension and conventional passive suspension system. From the Figure 4.19, the Sinusoid input at the amplitude of 0.04 m reduced to approximately 0.01 m. The 75% reduce of body displacement after the Sinusoid input comes out that the multiorder PID controller on active suspension will improve the performance of the vehicle suspension. From the Figure 4.20, body vertical acceleration value of multiorder PID active suspension value is reduce by a large percentage compared to PID active suspension and passive suspension. This reducing value after the introduction of multiorder PID active suspension system will improve vehicle ride comfort compared with passive suspension system.

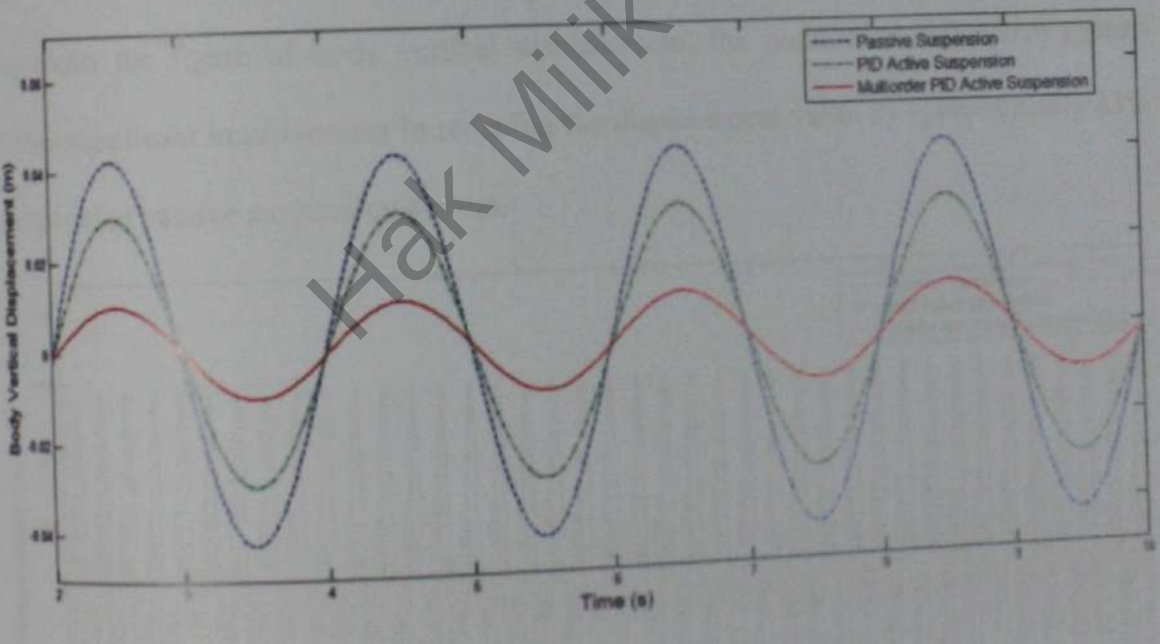

Figure 4.19: Body vertical displacement of 0.5 Hertz sinusoid road profile

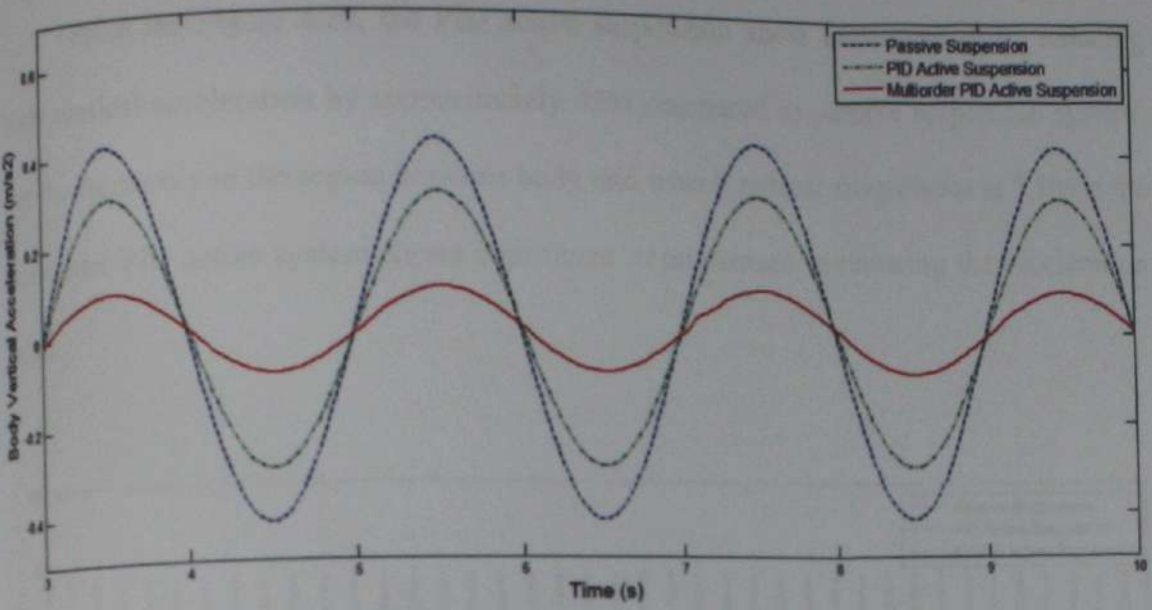

Figure 4.20: Body vertical acceleration of 0.5 Hertz sinusoid road profile

For the frequency in the region between body and wheel, natural frequencies at 5 Hertz are used to show the enhancement of movement of the vehicle body vertically and acceleration of the vehicle body vertically response. Based on Figure 4.21, which represents the figure of body vertical displacement, the multiorder PID active system shows significant improvement in reducing the displacement value by approximately 83% compared to passive suspension system.

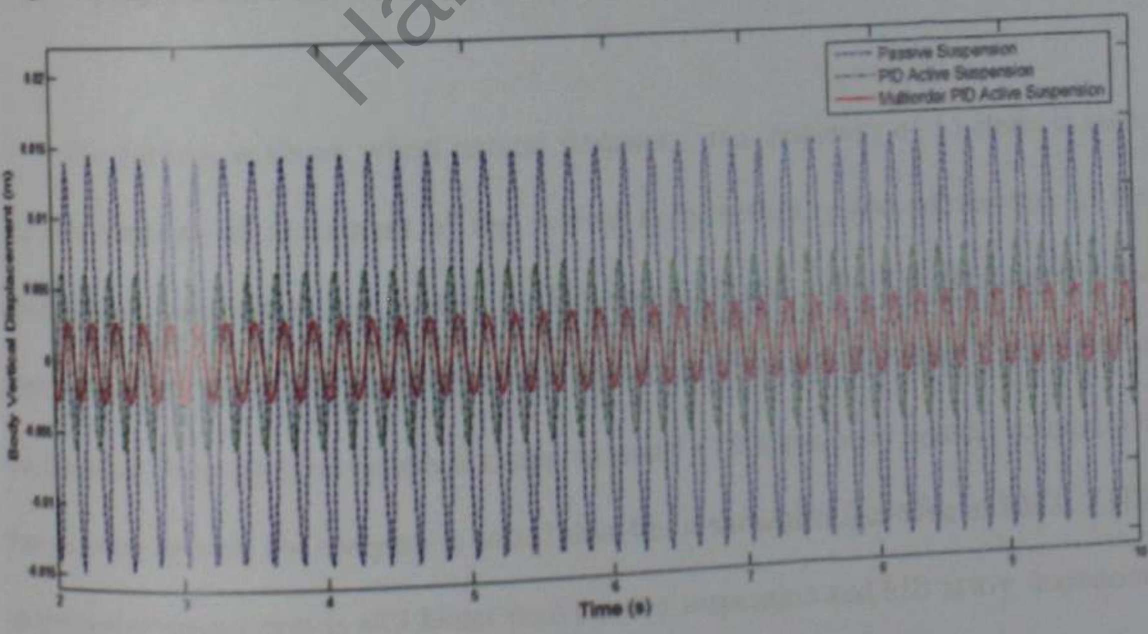

Figure 4.21: Body vertical displacement of 5 Hertz sinusoid road profile

From the Figure 4.22, the PID active suspension show improvement by reducing body vertical acceleration by approximately 45% compared to passive suspension system. For the frequency in the region between body and wheel, natural frequencies at 5 Hertz the multiorder PID active system shows significant improvement in reducing the acceleration value.

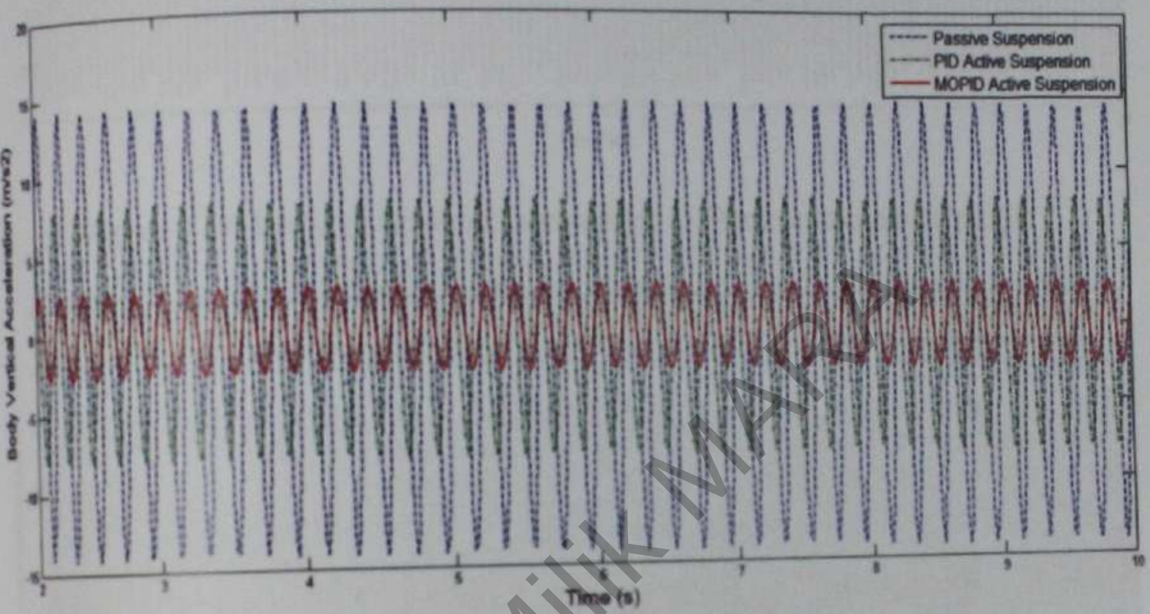

Figure 4.22: Body vertical acceleration of 5 Hertz sinusoid road profile

For the region above wheel natural frequency, the frequency of 15 Hertz is used. The enhancement of movement of the vehicle body vertically and acceleration of the vehicle body vertically are shown in Figures 4.23 and 4.24. At this region, the improvement at 15 Hertz show that the reduction nearly 85% of the body vertical displacement from passive system and approximately 80% for body vertical acceleration from passive system. By compare it with Table 4.6, it show that the value of RMS of PID control active suspension is still better than passive suspension and PID active suspension controller.

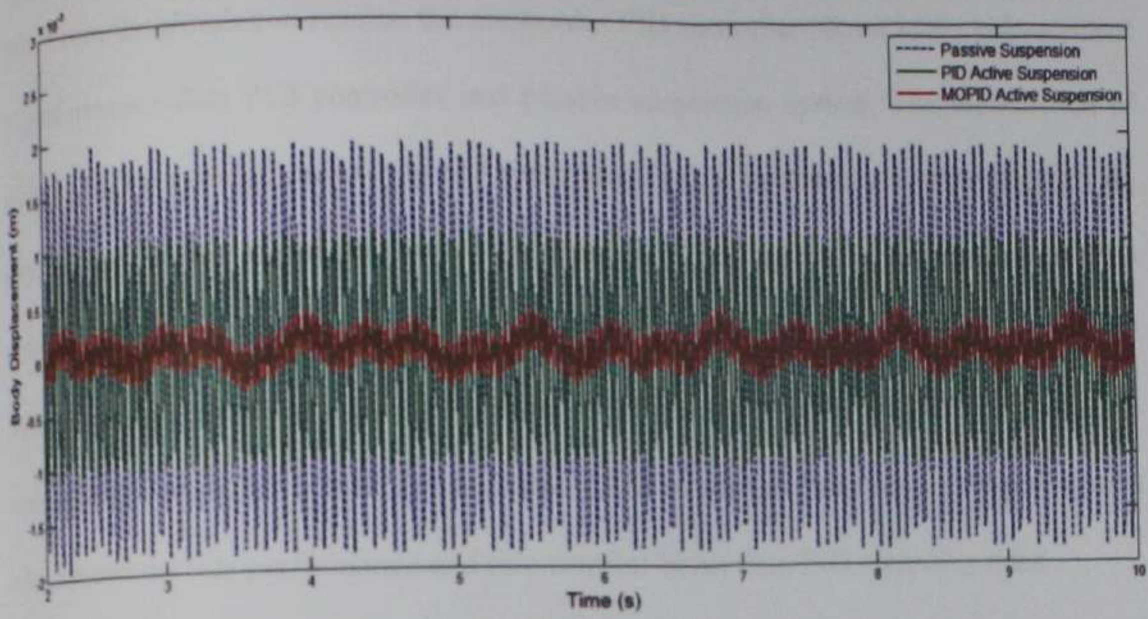

Figure 4.23: Body vertical displacement of 15 Hertz sinusoid road profile

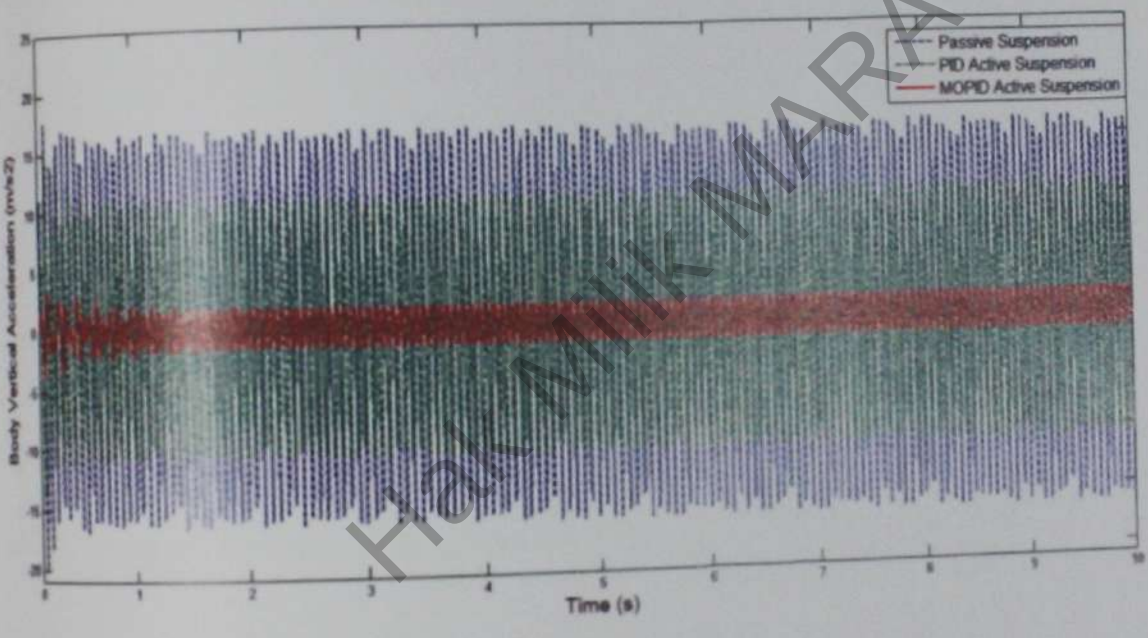

Figure 4.24: Body vertical acceleration of 15 Hertz sinusoid road profile

A simulation of multiorder PID control for active suspension has been studies to investigate the performance improvement between multiorder PID controller, PID controller and passive system. These studies focus on the improvement of ride performance and ride comfort in the suspension system. Therefore, the result will focus on movement of the vehicle body vertically and acceleration of the vehicle body vertically only. From the simulation results, the multiorder PID controller shows better enhancement of performance than PID controller and passive suspension system. The introduction of multiple PID controller, outer loop and inner loop is able to enhance comfort and driving performance. Based on simulation of quarter car test rig using the multiorder PID controller, it can be said that the introduced multiorder PID controller system has the good performance in control the movement of the vehicle body vertically and acceleration of the vehicle body vertically criteria. The multiorder PID controller on active suspension system will improve the ride performance and ride comfort better than PID controller rides.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Introduction

In these studies, the introduction of PID active suspension and multiorder PID active suspension controller against passive suspension system in order to improve ride comfort of a vehicle has been analyse. The analysis begins with the development of two degree of freedom (2DOF) quarter car model in Matlab/Simulink software. This model is validated with a study by Imaduddin. F (2010) that running an experimental method of active suspension system using two degree of freedom (2DOF) quarter car model. To improve ride comfort, a proportional integral derivative (PID) controller has been proposed. To enhance the performance of the proposed controller, an outer loop controller called multiple proportional integral derivative (MOPID) controllers also has been proposed. Finally, to evaluate the performance of the proposed system, the comparison of simulation results were analysis and presented.

5.2 Conclusions

In this study, multiorder PID controller and PID controller was propose in active suspension system controller in to minimize the impact of the movement of the vehicle body. The aim of this study is to improve ride comfort of the vehicle.

A two degree of freedom quarter car model for passenger vehicle was developed. The model was verified with Matlab/Simulink software to monitor the performance and capability of vehicle models used to simulate an active suspension system controller. By monitoring the result, it shows that the vehicle model in used will produce favorable results. Then the quarter car vehicle model is compared to result of experiment from Imaduddin. F (2010) for validation with experimental quarter car test rig.

The PID control and multiorder PID control have been proposed and successfully implemented in the active suspension system. The implementation is successfully done in simulation at chapter 4 and chapter 5 and by using a validated quarter car model.

PID controller shows the improvement in reducing body displacement between passive suspension system and multiorder PID controller show the better performance than the PID controller. The proposed PID control scheme is also able to improve ride comfort. Meanwhile, the multiorder PID control shows significant enhancement to the PID control. The multiorder PID control can enhance the performance of the PID control at the all region especially at body vertical acceleration.

5.3 Summary of Research Contributions

There are three contributions output from this study. Firstly, the PID controller was presented to reduce the unwanted vertical vehicle motion to enhance ride comfort. The control structure has been designed and tested in comparison with a conventional passive suspension system. From the results obtained, it can be said that the control techniques that have been introduced have been effective in controlling the vertical movement of the vertical acceleration of the vehicle body and the vehicle body.

The second contribution of this study is the modelling and development of a validated quarter car suspension system model. This quarter car model has been validated with experiment result from Imaduddin. F (2010). The two degree of freedom (2DOF) quarter car suspension system model is used for the purpose of actual quarter car simulation analysis.

The third contribution from this study is the development of the multiorder PID active suspension controller. This controller was proposed based on the fact that outer loop of the controller can be divided into three orders, the body vertical displacement, the body vertical velocity and the body vertical acceleration. The multiorder PID controller has been shown in the simulation that is able in improving the body vertical displacement and acceleration in active suspension system. The performance of the multiorder PID controller in improving body vertical displacement and acceleration in simulation is also better than the performance of PID controller. It is concluded that, the controller is verified able to enhance the vehicle comfort and ride quality significantly as the improvement in body vertical acceleration and body vertical displacement in the simulation responses.

5.4 Recommendation for Future Works

For the continuation of this study, there are several recommendations that can be proceeding. The first recommendations are the experimental evaluation of hydraulic actuated active suspension to perform in quarter car test rig. The test rig should be design to generate the frequency experimentally up to 15 Hz (above wheel natural frequency). Therefore, the development of a quarter car test rig is necessary for validation of proposed controller. The quarter car test rig is also capable to generate various types of disturbances. Further observation for other types of actuator for active suspension like a pneumatic or stepper motor system is also still interesting.

For the second recommendation is the proposed controller should be applied in active suspension system test rig. It is used for validation of data between multiorder PID controller, PID controller and passive system. Result from simulation is not strong enough to claim that the proposed controller is able to control the active suspension. It still needs to take the consideration of mechanical friction, resistance and signal noise in order to get the actual set of data.

The third recommendation is the proposed controller should involve the several types of other controller. In order to reduce the cost of active suspension system, the number of sensors used in the system should be considered. Other studies have been successfully reducing the cost of active suspension application by reducing the number of sensor. Reducing the number of sensors is also affecting the algorithm and number of states in controller. That is the reason to find the most efficient and effective controller performance.

For the final recommendation, the simulation and experiment should involve higher degree of freedom. It should handle 4 DOF or 7 DOF in order to study several aspects of vehicle dynamics such as vehicle roll or vehicle pitch.

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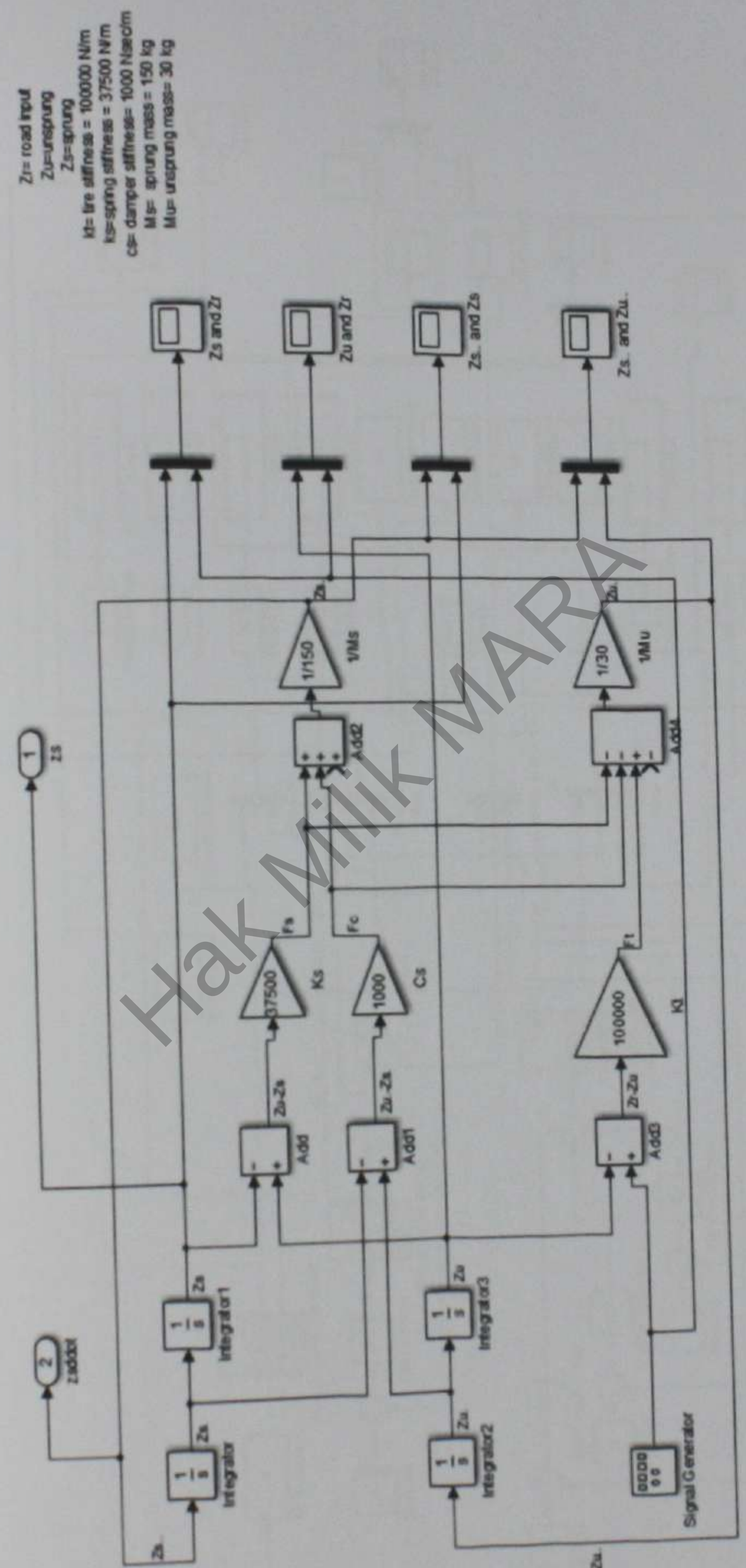
QUARTER CAR MODEL MATLAB-SIMULINK



ACTIVE SUSPENSION MODEL MATLAB-SIMULINK