ACTIVE SUSPENSION SYSTEM USING STABILITY AUGMENTATION SYSTEM FOR PASSENGER VEHICLE

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This report is submitted in fulfillment of the requirement for the degree of Bachelor of Mechanical Engineering (Automotive)

Faculty of Mechanical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

2017

DECLARATION

I declare that this project report entitled "Active Suspension using Stability Augmentation System for Passenger Vehicle" is the result of my own work except as cited in the references.

Signature : ....

Name : ....

Date : ....

APPROVAL

I hereby declare that I have read this project report and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering (Automotive).

Signature : ....

Name of Supervisor: MOHD HANIF HARUN

Date : 16/6/2017

DEDICATION

I dedicate this report to my loving parent, Mr Azmi Bin Mohd Yusop and Mrs Norhajira Binti Khalid support me to write this report and do the simulations. My deepest respect and thanks go to my family for all they have done for me in which I will never forget. Their patience and encouragement have been invaluable. I am deeply indebted to them. After this, I promise to be a good son. This is only the beginning of our family future success and prosperous. Not to forget, special thanks to brothers and sisters for giving me a real support, pray, and support me in ups and downs. Special thanks also to Prof. Madya Dr Azma Putra and JKPSM because have been guide us along we do final year project from the beginning until the end of study. Special thanks also to all lecturer that have been share their knowledge to us while us doing final year project. Thanks also to our supervisor that have guide us along we do final year project and share his experience with us. Thanks also to my group that under our supervisor have been help me and guide me. Thanks also to people that help me along this final year project officially or unofficially.

ABSTRACT

This study presents the use of active suspension system using stability augmentation system in reducing the effects of road surface to the vehicle ride comfort. A control oriented half vehicle simulation model was developed in Matlab and verify with CARSIM 8 software simulation. The vehicle model with active suspension and stability augmentation have been developed in Matlab/Simulink. An investigation of a suitable active control algorithm that can improved vehicle ride comfort was proposed in reducing unwanted vertical motion of passenger vehicle when passing a step or bump. The proposed algorithm known as stability augmentation system (SAS) that is able to reduce the effect of road disturbance, maintains load-levelling and load distribution during vehicle maneuvers in an ideal case for control algorithm parameters tuning is presented. Active suspension based on desired force from the controller and relative velocity between sprung mass and unsprung mass of the vehicle model and shown to be able to consistently provide the actual force to reduce the vibration of the vehicle body. The vehicle is tested at different speeds of 40km/h and 100km/h. Comparison of the simulation results demonstrates the improvement of the vehicle ride comfort.

ABSTRAK

Kajian ini membentangkan akan penggunaan sistem suspensi aktif dengan menggunakan sistem pembesaran kestabilan bagi mengurangkan kesan permukaan jalan terhadap keselesaan pemanduaan kenderaan. Simulasi model separuh kenderaan berorientasikan kawalan telah dicipta dan bagi mengesahkan dengan simulasi perisian CARSIM 8. Model kenderaan dengan penggantungan aktif dan kestabilan pembesaran telah dibangunkan dengan menggunakan Matlab/Simulink. Penyiasatan algoritma kawalan aktif sesuai kerana boleh meningkatkan keselesaan pemanduan kenderaan telah dicadangkan dalam mengurangkan pergerakan menegak yang tidak diingini kenderaan penumpang apabila melalui permukaan bertangga atau beralun. Algoritma yang dicadangkan adalah sistem pembesaran kestabilan yang mampu mengurangkan kesan gangguan permukaan jalan, mengekalkan badan kenderaan stabil dan pengagihan semasa pergerakan kenderaan berdasrkan permukaan jalan mengikut parameter kawalan algoritma telah di bentangkan. Suspensi aktif akan menghasilkan daya mengikut kuasa yang dikehendaki dari sistem kawalan dan halaju relatif antara badan kenderaan dan sistem suspensi dan terbukti mampu untuk mengurangkan getaran badan kenderaan. Kenderaan itu juga akan di uji pada kelajuan yang berbeza pada kelajuan 40km/j dan 100km/j. Perbandingan hasil keputusan simulasi menunjukkan peningkatan keselesaan pemanduan kenderaan.

ACKNOWLEDGEMENT

I would like to take this opportunity to express my sincere gratitude to my supervisor Mr. Mohd Hanif bin Harun for his invaluable advice, constructive criticisms, and constant encouragement. Thanks for giving me this opportunity to do final year project with him. He never hesitated to give me advice and guidance whenever I confronted problems. I am thankful for his patience and advice while leading me in this project.

Finally, I would like to thank a named Prof. Madya Dr. Mohd Azman bin Abdullah for teaching and giving his time to guide me during class session. He would share his knowledge in the field of vehicle control system with me and guide me to do simulation. I would like to thank my course mates for giving me their support, patience and encouragement. Finally, I would like to thank my family for their support.

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LIST OF ABBREVIATIONS

MR	Magnetorheological
ER	Electrorheological
PID	Proportional-Integral-Derivative
PI	Proportional Integral
PD	Proportional Derivative
LPV	Linear Parameter-Varying
FEBC	Frequency Estimation-Based Controller
SAS	Stability Augmentation System
DOF	Degree of Freedom

LIST OF SYMBOLS

m	-	Mass
k	-	Spring stiffness
c	-	Damping coefficient
l	-	Length
z	-	Displacement
$\dot{z}$	-	First derivative of z
$\ddot{z}$	-	Second derivative of z
M_b	-	Mass of vehicle body
M_{wr}	-	Mass of wheel rear
M_{wf}	-	Mass of wheel front
K_{sf}	-	Front suspension spring stiffness
K_{sr}	-	Rear suspension spring stiffness
K_{tf}	-	Tire spring stiffness front
K_{tr}	-	Tire spring stiffness rear
C_{sf}	-	Front suspension damping coefficient
C_{sr}	-	Rear suspension damping coefficient
I_θ	-	Pitch moment of inertia
l_f	-	Length of front vehicle from center gravity of the vehicle body
l_r	-	Length of rear vehicle from center gravity of the vehicle body

CHAPTER 1

INTRODUCTION

1.1 Background

Suspension system is an important part to support sprung mass and stabilize vehicle body from moving upward and downward, from uneven road surface. Spring will absorb unwanted forces from the road and reduce jolting, meanwhile damper prevent bouncing upward and downward of the vehicle. That is fundamental operation for passive suspension. Nowadays, there are three types of suspension system were built by the automotive industry. Which are passive, semi-active and active suspension systems. The producing of semi-active and active suspension is to improve the function of passive suspension and driving comfort on the uneven terrain. Besides that, passive suspension cannot control the spring stiffness and damper coefficient, in order to decrease the displacement of sprung mass vibration (Chen, et al., 2014).

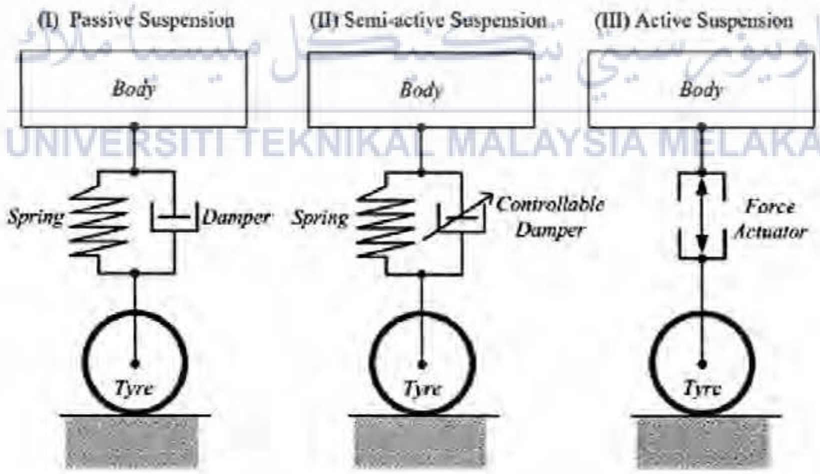

Figure 1.1 Type of suspension system (Chen, et al., 2012)

Passive suspension has a limitation to reduce the motion of the car, only certain ground surface. For example if the vehicle with passive suspension is driving on the bump surface continuously, the vibration of vehicle will not reduce and it will bounce until the tire with flat surface. By this phenomena of continuously bouncing, driver and passengers comfort and also ride quality of the vehicle body will be worse. Other than that, passive suspension cannot stabilize the vehicle body during the cornering. For an example, if the vehicle make cornering to the left, the suspension on the left side car will be moving downward. Passive suspension does not have control system to push the left side of the car moving upward, if the suspension system on the left and the right side are in the same position as the result stability of the car body during cornering will be improve.

The effect of the suspension system can be analyse by using simulation, experimental or both. In this project the effect of suspension system, to the ride comfort of the vehicle will be analyse using Matlab/Simulink. The parameter of sedan car has been taken as a model for the simulation. The simulation will be analyse on two type of suspension, one with passive suspension and the other is active suspension with stability augmentation controller. This project will use equation of motion for 4 degree of freedom half car pitch plane for sedan car with passive suspension. The advantage of magnetorheological damper can control the movement suspension by control damping coefficient according to the vertical force of the ground to the vehicle (Naik and Singru, 2009). The other benefit of semi-active suspension, it use less current to control the suspension distance in order to decrease vehicle motion from bounce upward and downward. Moreover, semi-active can be work as passive suspension if the control cannot be use or not function (Luo, et. al., 2010), the vehicle motion still can be reduce.

1.2 Problem Statement

Vibration from the vehicle body could make driver and passengers feel not comfort and loose stability of the vehicle body. If the vehicle stability does not control, while vehicle want to make cornering it will increase vehicle body roll. If the roll of the vehicle body increase, the probability to accident is high. Ground terrain also will affect the vehicle stability, because passive suspension only can absorb the vibration at certain ground surface, depend on the passive suspension parameter. If the vehicle move on the long bump surface, the vehicle body will be continue bounce and will stop until the wheel move on the flat road surface. Passive suspension cannot control the damper coefficient and/or spring stiffness to absorb vibration, this will effect the vehicle body motion(Campos, et al., 1999). Passive suspension does not have sense that can detect the vehicle motion in order to produce force to stabilize the motion.

Suspension system with stiff and harsh will affect the passenger comfort, while suspension system softer it will affect the road handling of the vehicle. This problem can be solve by using active suspension system, it can control the movement of the suspension. This system can reduce the vehicle body vibration and improve road handling. However, active suspension require high current to operate the system in order to stabilize vehicle body motion. It also use high cost for the maintenance, because the active suspension is fully control system.

If the system controller for the active suspension failure, the suspension work will be unstable (Luo et al., 2010). This is study is to improve ride comfort of vehicle during bouncing by using active suspension. Besides that, control strategies of active suspension with stability augmentation system controller will be developed to reduce the vibration of the vehicle body.

1.3 Objective

The objectives of this project are as follows:

i. To improve ride comfort of vehicle.
ii. To develop a control strategy for active suspension with stability augmentation system.

1.4 Scope of Project

The scopes of this project are:

i. The vehicle parameter of D-class sedan car half car model is selected.
ii. This study only focus on ride analysis of the car model.

1.5 Summary

Passive suspension cannot be adjusted or controlled the movement of the suspension to stabilize the vehicle body motion. Active suspension system with stability augmentation system have been used in this project to study the effect of the stability control on the suspension system. Since, active suspension system can control the vehicle body movement upward and downward or reduce the vehicle body displacement. So, the vibration of the vehicle motion can be reduce, as it can produce force to control vehicle body stability.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

This chapter presents, the advantage of active suspension, vehicle stability test simulation, type of semi-active damper, and followed suspension controller in active suspension technology. An overview of current works and research among the researchers is also presented.

2.2 Vehicle Suspension System

Suspension have been use in the vehicle system, to support the weight of the vehicle and also reduce vibration in the vehicle body. Suspension can be divide into two type, dependent and independent suspension system. Dependent suspension usually use in commercial vehicle, for example lorry and bus. Dependent suspension is common use in commercial vehicle because it can support high weight or load. The example of dependent suspension is leaf spring, the combination of flat plate can support large area and weight of vehicle load. The disadvantage of dependent, it will affect the other side tire if the tire rolling on the bump road surface. The advantage of independent suspension, the other tire will not bounce if the tire on the other driving drive on the bump surface. The main of the suspension system are spring, shock absorber, strut and tire of the vehicle. Besides that, there type of suspension system, that use in the vehicle suspension. The suspension system are passive, semi-active and active suspension system.

2.2.1 Passive Suspension System

Passive suspension are common use in the vehicle suspension system, because of simple and low cost of production than semi-active and active suspension system. However, passive suspension can control the movement of the suspension system according to the ground terrain (Chen, et al., 2014). Vehicle passive suspension with low hard suspension will reduce the vehicle passenger comfort. Besides that, if the vehicle use soft suspension it will give ride comfort but poor road handling (Ikenaga, et al, 2000). Passive suspension is the suspension that did not have any controller that can adjust the movement of the spring and shock absorber.

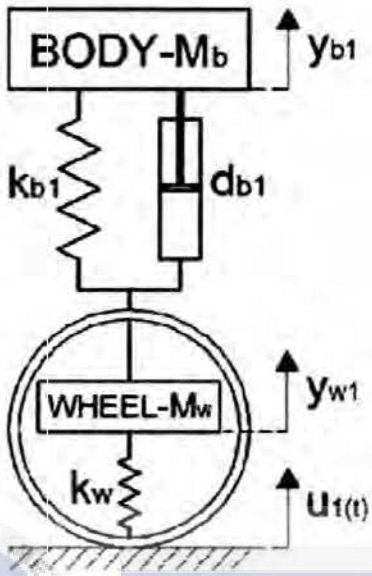

Figure 2.1 Schematic diagram for passive suspension (Paulides J. J. H. and Lomonova E. A., 2006)

2.2.2 Semi-active Suspension System

Suspension provide good handling and braking for safety driving and also reduce bump and vibration from the ground terrain. Semi-active suspension provide better vehicle stability than active and passive suspension. Semi-active suspension can act as passive suspension if the controller not function. But, active damper would be unstable if the controller cannot be use (Luo, et al., 2010). Besides that, (Chen, et al., 2014) wrote that semi-active can provide suspension control of spring stiffness and damping coefficient. Other than that, semi-active suspension time to react with the vehicle body vibration is less than 100 milliseconds. Semi-active suspension can provide harder and softer damping limit according to the ground terrain (Qazi, et al, 2013).

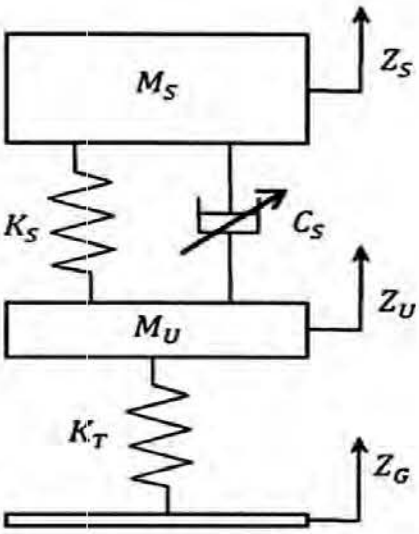

Figure 2.2 Schematic diagram for semi-active suspension (Raj A. and Rajamohan V., 2016)

2.2.3 Active Suspension System

Active suspension function is to isolate the vehicle body from wheel vibration by the ground terrain. Besides that, fully active suspension system are control by electronic system to control the vehicle body condition. Then, the vibration from the ground terrain will be directly control motion by electronic device of active suspension system. The suspension system is good to isolate the vehicle body motion. However, the installation and maintenance of the active suspension are quite expensive (Turnip, et al., 2008). Other than that, if the electronic control system in the suspension does not work, the active suspension will be unstable because of electronic system cannot control the movement of the suspension (Luo, et al., 2010).

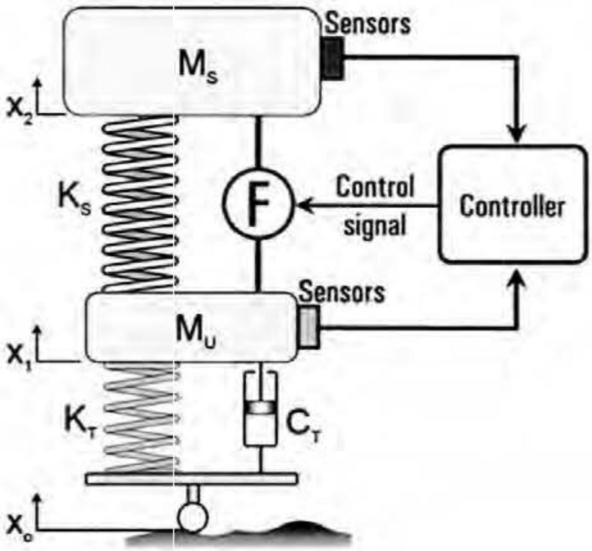

Figure 2.3 Schematic diagram for active suspension with controller (Yahia E., 2016)

2.3 Semi-active Damper in Vehicle Suspension System

There are two common semi-active damper that are usually use as damper for semi-active suspension system. One is magnetorheological (MR) damper and the other is electrorheological (ER) damper. Moreover, time taken for magnetorheological and electrorheological damper to respond with the vehicle vibration at 40ms, force can produce by the semi-active damper is large range and low cost (Vivas-Lopez, et al, 2015).

2.3.1 Magnetorheological Damper

Magnetorheological damper also semi-active damper that isolate vibration of suspension system and also building structure. Besides that, MR damper reaction depend on the current input. By the current input flow to the MR damper, it will adjust the suspension distance and isolate the vibration of the vehicle body (Mori, et al., 2007). Compare with electrorheological damper, use of electric current to operate the suspension system is less. Magnetorheological damper able to change the fluid in the damper become solid and semi-solid within millisecond when current flow to the MR damper and control the magnetic fields (Naik and Singru, 2009). Besides that, (Lozoya-Santos, et al, 2012) also said that, electric current that flow in the damper will produce magnetic field, which will modifies damping ratio by control oil viscosity in the damper. The liquid form will be solid according to the current supply to the suspension system.

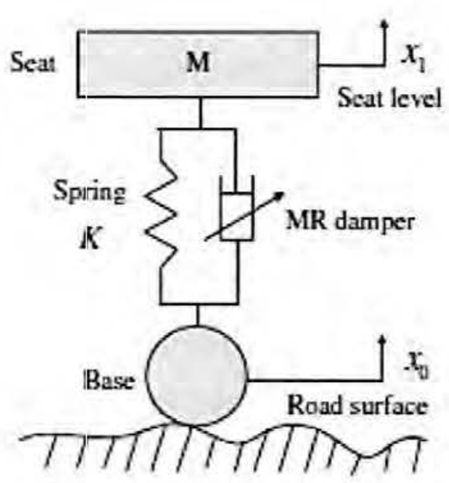

Figure 2.4 Suspension with MR damper (Mori, et al., 2007).

From Figure 2.5 magnetorheological damper operation are when current supply to the damper it will produce magnetic field, which is make iron particles change into linear chain parallel to the field. This will phenomena will solidify iron particles with respect to the vehicle vibration motion. Besides that, MR damper can control the force for suspension system continuously, good response and less power consumption (Kasemi, et al., 2012).

Figure 2.5 Transformation iron particle MR damper (Kasemi, et al., 2012)

2.3.2 Electrorheological Damper

Electrorheological (ER) damper also common semi-active damper that is use with the semi-active suspension system. In the electrorheological damper, flat plate will be forming by two electrodes with electrorheological fluid flow through in the damper (Holzmann, et al, 2006). By flow of the electric current, the viscosity of the electrorheological fluid will be increase and then will adjust the suspension expand according ground terrain. Electrorheological damper have been propose in the rear car suspension to improve the driving performance from bump and ground terrain. But, the fluid in the electrorheological damper are quite expensive (Choi and Kim, 2000).

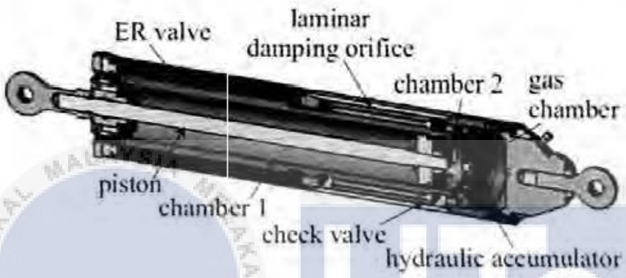

Figure 2.6 Cross section of ER damper (Holzmann, et al., 2006)

2.4 Suspension Controller

Semi-active suspension have several type of controller in the suspension to control the suspension movement in order to stabilize the vehicle body. Also to improve ride quality and passenger comfort. The controller reduce vertical force from ground or road disturbance to the vehicle body. While the current flow to the suspension system, the controller will stabilize the vehicle body according to road bump condition.

2.4.1 Proportional-Integral-Derivative (PID)

The control structure for the PID (Gaur, 2013) in the Figure 2.7, it have three separate parameters. The combination of three parameter will adjust the control element in the suspension system and reaction of PID control depend on rate of error change. The error input signal will be sent to PID controller, the derivative and integral will be compute from the error signal. After that, the output from PID will transfer to the system or process and produce new output. This output will be sent to the sensor again in order to obtain new error signal, the process will continue as close-loop system.

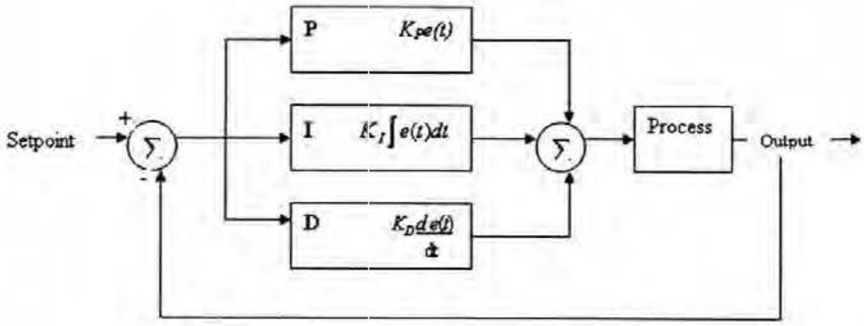

Figure 2.7 Control structure of PID (Gaur, 2013)

From simulation result of (Kasemi, et al., 2012) in the Figure 2.8, fuzzy-PID controller give better passenger comfort than PID and passive suspension system. Besides that, the peak reduce at 68% than PID controller at low frequency. And about 83% peak reduce from the passive suspension. From the result, the improvement are quite high than PID controller on the suspension system.

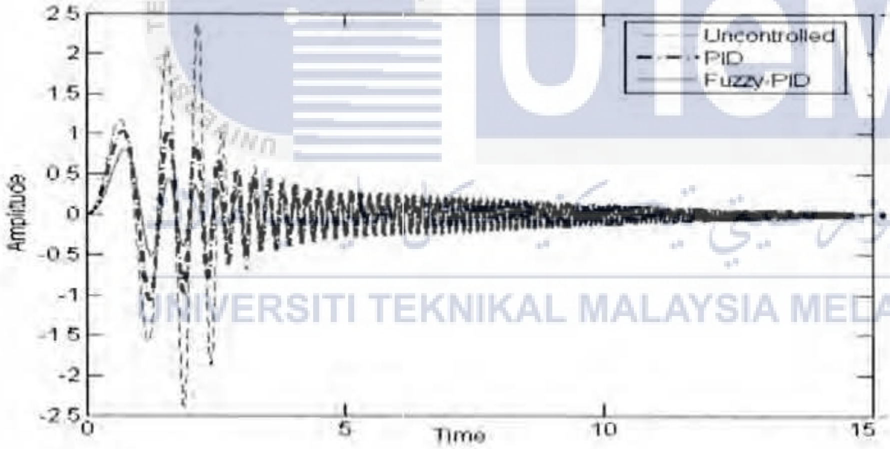

Figure 2.8 Controller response (Kasemi et al., 2012)

Other than that, (Hanafi, 2010) have make simulation on PID, PI and PD. By refer to controller parameter Table 2.1, quarter car suspension system with PID give better performance than other. PID also could reduce bounce effect to the vehicle while hit and after bump. PD controller cannot be used to reduce the vehicle motion according to the small parameter.

Table 2.1 Controller parameter (Hanafi, 2010)

Parameter	Type of controller		
	PID	PI	PD
K_p	0.1211	2.970	0.00001
T_i	0.0980	0.105	∞
T_d	0.1200	0	0.0000001

By referring to the Table 2.2, PID controller have been use with active suspension. PID controller have been reduce the percentage overshoot for car body acceleration, suspension travel and wheel travel. This result show that PID controller with active suspension can reduce the percentage of overshoot and improve ride quality.

Table 2.2 Reduction of percentage overshoot values for step road input (Ahmed S., et al., 2015)

No	Parameters	Passive	Active	Reduction $\left(\frac{passive-active}{passive}\right)$ 100 %
2	Car body acceleration	2 m/s^2	$0.55 m/s^2$	72.5 %
3	Suspension travel	0.052 m	0.0281 m	46 %
4	Wheel travel	0.0491m	0.0186 m	62.11 %

2.4.2 Linear Parameter-Varying (LPV)

Two type vehicle ride model, which are full-vehicle and quarter-vehicle with LPV controller have been used in the (Fleps-Dezasse and Brembeck, 2016) to compare the result with skyhook controller and passive suspension. From Figure 2.10, LPV controller for both full and quarter vehicle the optimum ride comfort are almost same and better than skyhook controller. Besides that, LPV controller also can provide more road holding about 5% than skyhook controller.

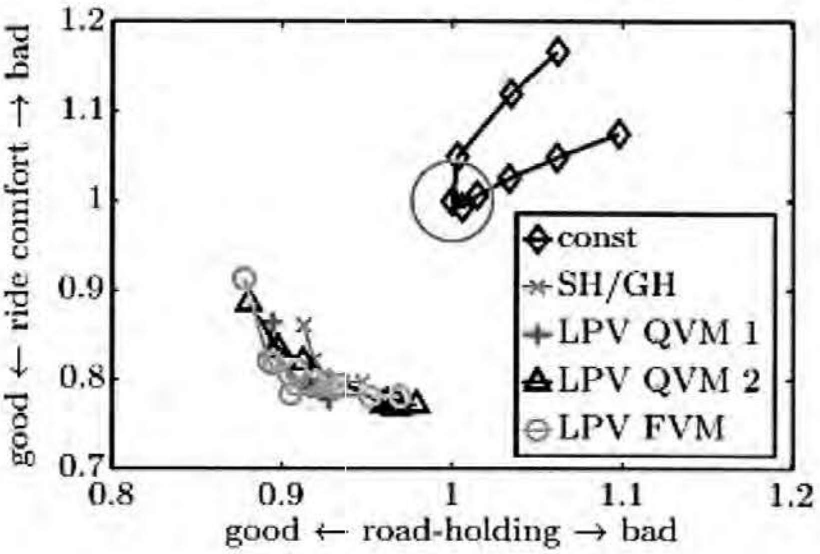

Figure 2.9 Result LPV and skyhook (Fleps-Dezasse and Brembeck, 2016)

Besides that, LPV compare with frequency estimation-based controller (FEBC) and passive suspension have been study by (Morales-menendez, et al., 2011), by refer to the Figure 2.11, the comfort performance improvement of LPV and FEB controller 0-10Hz than passive suspension system. The amplitude at 160mm and frequency of 0.2Hz for bump to evaluate the control system. By referring to the Figure 2.12, the LPV controller is soft deflection of the suspension and safe than FEB that produce harder damping. From the result also LPV controller amplitude is beyond FEB and passive suspension. However, FEB controller meet the comfort requirement, LPV and passive suspension are limit for uncomfortable condition.

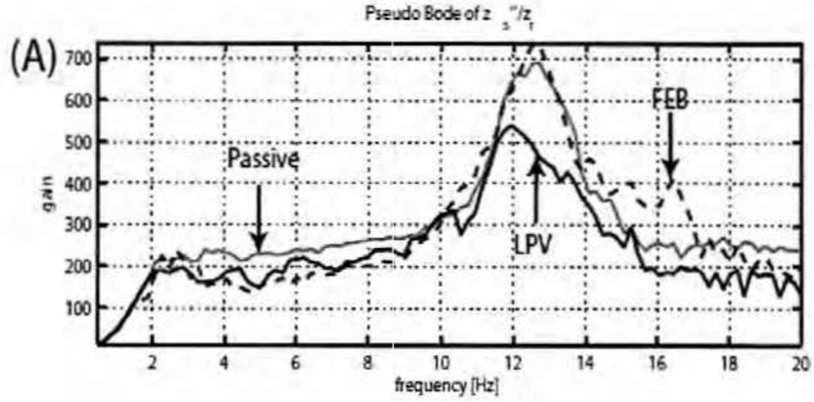

Figure 2.10 Suspension performance comparison (Morales-menendez, et al., 2011)

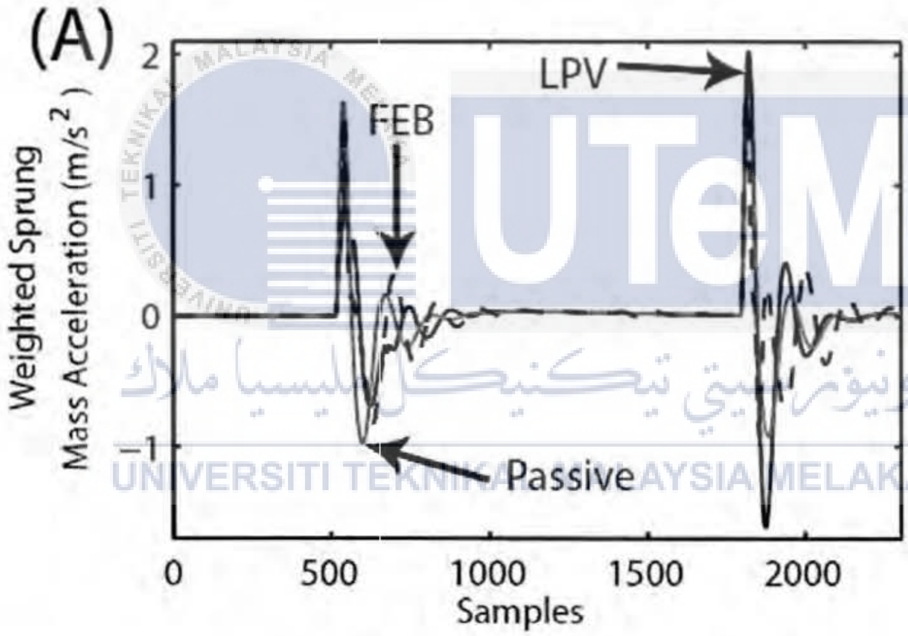

Figure 2.11 Sprung mass acceleration (Morales-menendez, et al., 2011)

2.4.3 Skyhook

Skyhook controller with MR damper have been study (Do, et al., 2010), from the simulation result it find that, skyhook controller will provide good damping rate at the second peak with hard MR damper. Mixed skyhook efficient as soft MR damper with low rate of damping. From the Figure 2.13 the most efficient controller and good damping rate is skyhook-add with MR damper.

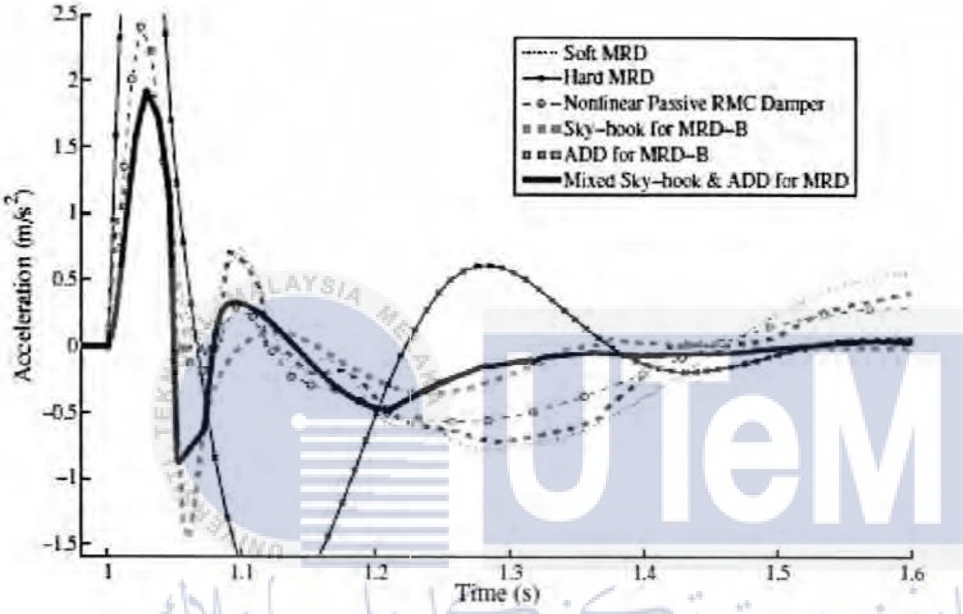

Figure 2.12 Acceleration at 0.01m step input (Do, et al., 2010)

Besides that, the passenger comfort have been improve about 17% by skyhook and ADD and soft MR damper can be achieve this improvement. The improvement passenger comfort, will be also improve the vehicle ride quality. The vehicle vibration can be reduce by using skyhook and ADD. By refer to the Figure 2.14, the performance of hard magnetorheological damper is worst performance for the vehicle suspension system.

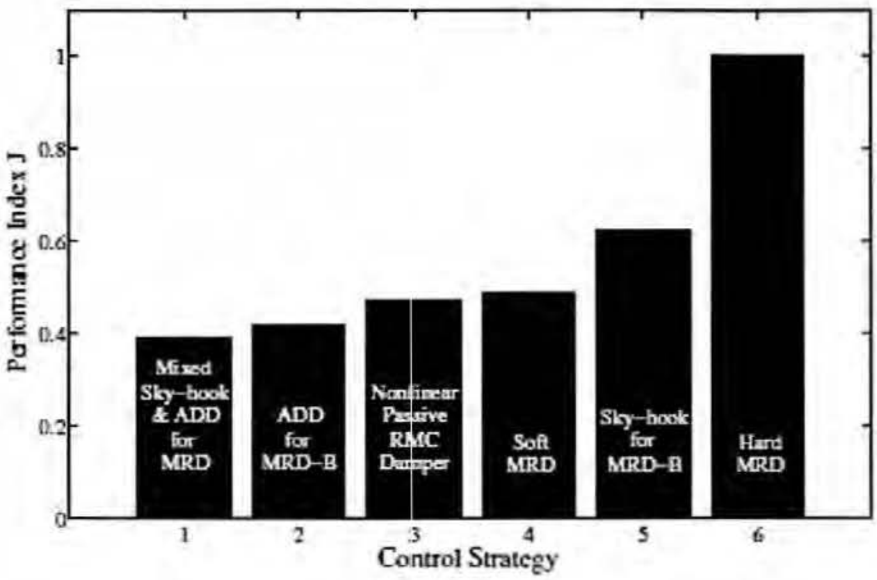

Figure 2.13 Performance comparison (Do, et al., 2010)

2.4.4 Stability Augmentation System (SAS)

Full vehicle of active suspension system (Ikenaga, et al, 2000) the controller system same as SAS of aircraft. The vehicle heave, pitch and roll acceleration have been improve by the active damper at low and high frequencies, by refer to the Figure 2.15, the active damper improve the passenger comfort than passive suspension. Besides that, the vibration of the vehicle have been reduce by using the active damper controller.

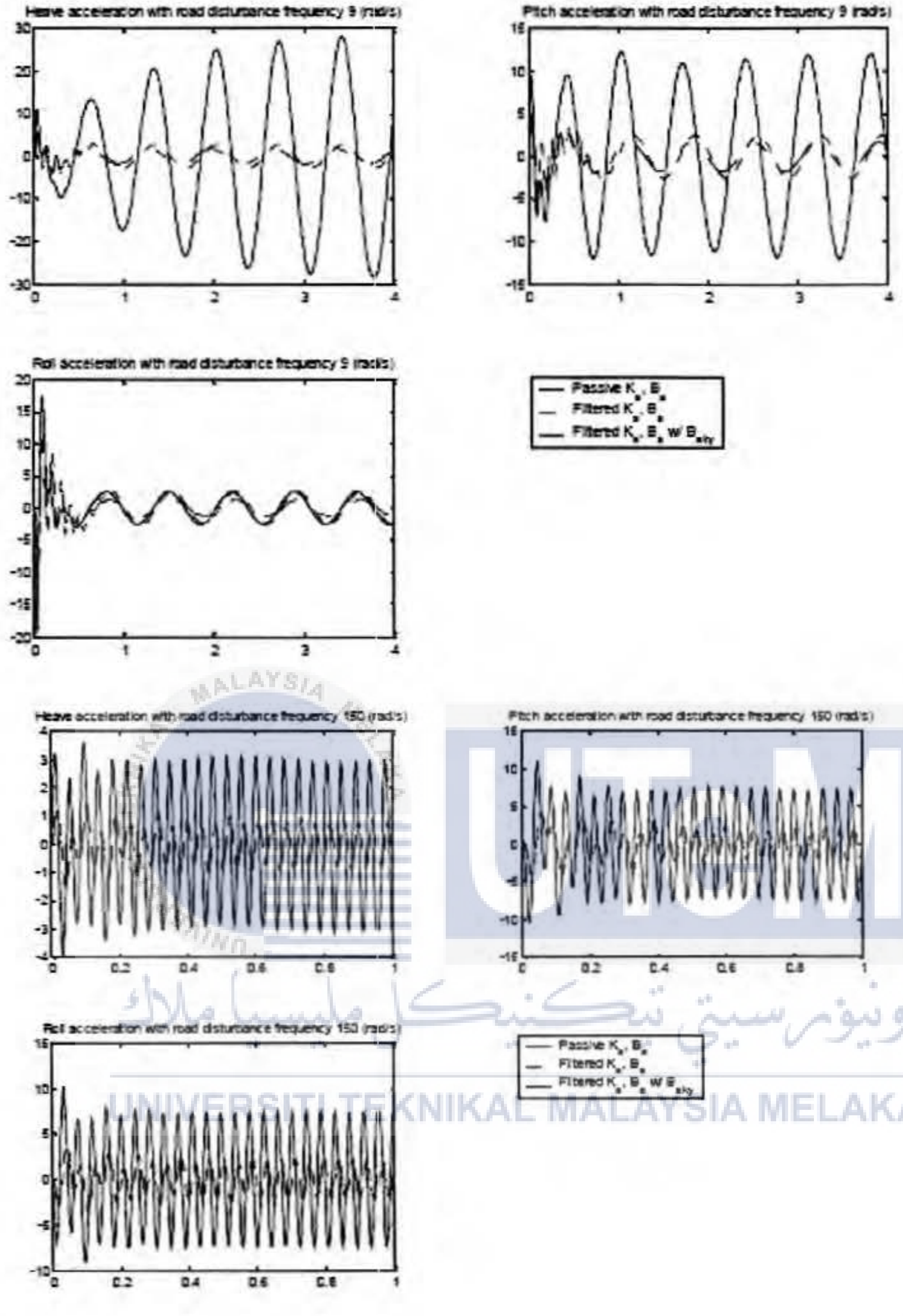

Figure 2.14 Low frequency and high frequency (Ikenaga, et al, 2000)

Besides that, half vehicle active suspension system by (Campos, et al., 1999) with controller system same as SAS used in aircraft. The vehicle heave and pitch acceleration have been improve by using active controller at low frequency. By refer to the Figure 2.16, the active damping with skyhook controller provide better vehicle isolation than the other controller. The amplitude of pitch acceleration with road disturbance by the active damping and skyhook is much reduction than passive suspension system. From the result, the heave acceleration with road disturbance, have been reduce by applying active damper and skyhook controller in the active suspension system.

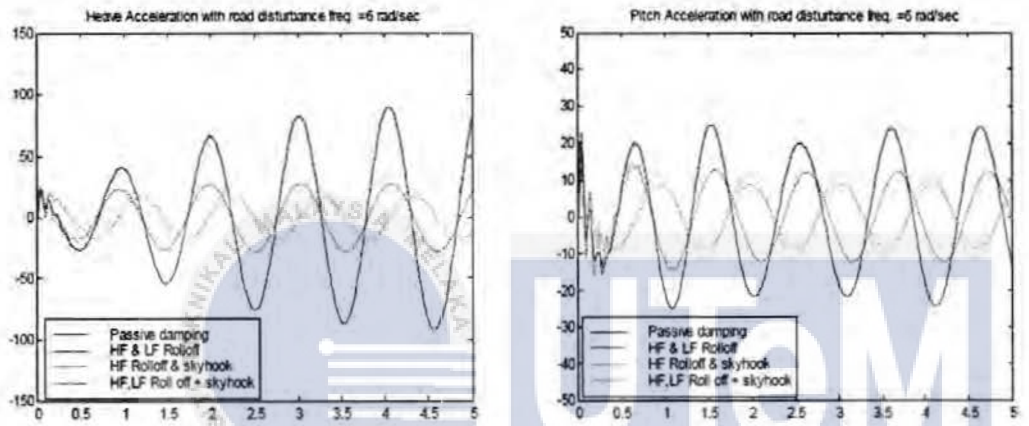

Figure 2.15 Simulation result for wr = 6 rad/s (Campos, et al., 1999)

The control structure for stability augmentation system with semi-active suspension have been develop by referring to Figure 2.17 ((Harun, et al., 2014). The controller in stability augmentation system, attitude and ride controller. Attitude is to control the heave and pitch motion and ride control is effect to the road disturbance rejection.

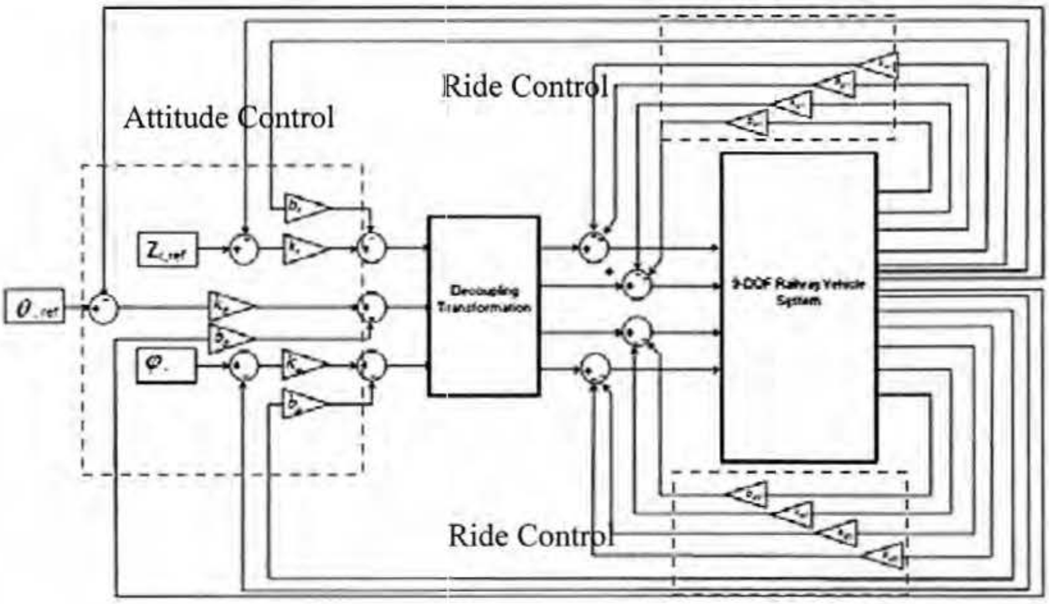

Figure 2.16 Control structure for stability augmentation system (Harun et al., 2014)

2.5 Summary

In this chapter, the vehicle suspension system have been reviewed, from conventional suspension or passive suspension, active and semi-active suspension. From the previous study there are many type of controller have been done on automotive suspension system. For example, proportional integral derivative (PID), linear parameter varying (LPV), skyhook and stability augmentation system. By using stability augmentation system with active suspension with ride control and attitude control it will reduce the vibration of the vehicle body. Active suspension system with stability augmentation system will be used in this study with half car ride model.

CHAPTER 3

METHODOLOGY

3.1 Introduction

Vehicle model can be study the pitching, rolling and also stability of the vehicle body by using Matlab, CARSIM and Adams software. Furthermore, vehicle dynamic model was developed by using mathematical equations in order to run the simulation in the Matlab software. The mathematical equations for the vehicle can divided into three main part, first is quarter car, half car and full car ride model. Quarter car ride model or two degree of freedom (2-DOF) is the combination of sprung mass and unsprung mass for quarter car suspension system either right, left or front and rear of the vehicle suspension system. Besides that, the combination of semi-active suspension with stability augmentation system was developed to control the vibration of the vehicle that effect the ride comfort to the driver and passenger while on the ground terrain. Road input has been use in this study which is step input and sine-wave test.

3.2 General Simulation Setup

In the beginning of this study, a fundamental and previous study on vehicle suspension has been reviewed. The type of suspension such as passive, semi-active or active suspension system and the controller used, such as proportional-integral-derivative (PID), skyhook or linier parameter-varying are also discussed. The mathematical equation are derive based on Newton's second law from the half car vehicle model. The vehicle model is then verified with the CARSIM model, to compare the response of the Matlab and CARSIM simulation models. If the result for Matlab and CARSIM are same or nearest it is accepted that the derivation of the vehicle model is accepted and can be continue with the next step.

After verify the simulation result, the passive suspension vehicle model have been combine with stability augmentation system (attitude and ride control). The stability augmentation system it will be used three different type of controller. The suspension system will be control with attitude controller, ride controller and combination of attitude and ride controller. Then, from the simulation result of active suspension with SAS controller, it will be analysed to identify controller that give the big impact to reduce vehicle body motion and ride comfort.

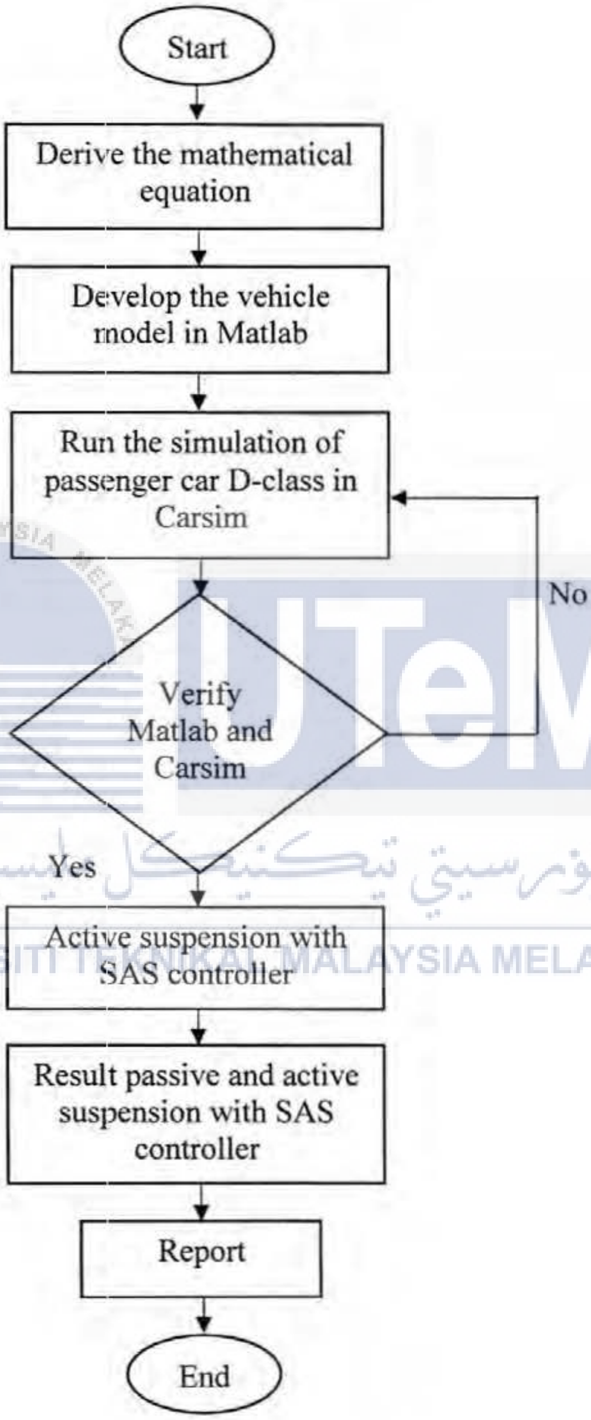

Figure 3.1 Flow chart of the methodology

3.3 Modelling Assumptions

Mathematical equations is important in developed the vehicle model before running the vehicle model with it parameters. There are several assumptions were made in developing the vehicle model, due to limitation of the software.

i. The vehicle mass is lumped into a single mass as sprung mass.

ii. Aerodynamics lift and drag forces, as well as tyre rolling resistance on the road surface are negligible.

iii. The pitch axis for ride model is at the center of gravity.

iv. The vehicle remain on the ground at all times, which is the front and rear tyre never lose contact with the ground while maneuvering.

v. The deflection in pitch are small and it was simplified with small angle approximation.

vi. The tyre for the vehicle model is simplified as linear spring constant without damping.

3.4 Equation of Motion Development

3.4.1 Ride Model

The ride model for half car has been shown in Figure 3.2, mathematical equations will be carry out from the half car vehicle ride model sprung mass and unsprung mass information. A half-vehicle suspension system is represent as four degree-of-freedom system for half car ride model. The system include the sprung mass and unsprung mass, which is rear and front suspension and wheel. The suspension system between vehicle body and wheel are modelled as linear viscous spring and damper element. Then, tire are modelled as spring without damping. The deflection of pitch are small and simplified with small angle approximation.

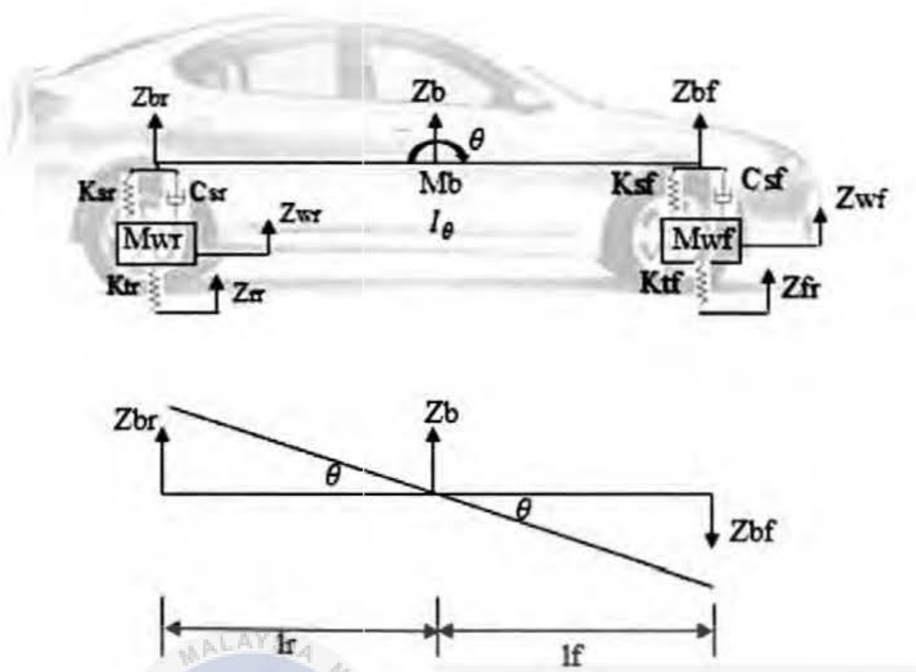

Figure 3.2 A 4-DOF vehicle ride model

From Figure 3.2 the vertical displacement and the vertical velocity for the sprung mass are given by;

Vertical displacement

Front wheel

$$Z_{bf} = Z_b - l_f sin\theta = Z_b - l_f\theta \tag{3.1}$$

Rear wheel

$$Z_{br} = Z_b + l_r sin\theta = Z_b + l_r\theta \tag{3.2}$$

Vertical acceleration

Front wheel

$$\dot{Z}_{bf} = \dot{Z}_b - l_f sin\dot{\theta} = \dot{Z}_b - l_f\dot{\theta} \tag{3.3}$$

Rear wheel

$$\dot{Z}_{br} = \dot{Z}_b + l_r sin\dot{\theta} = \dot{Z}_b + l_r\dot{\theta} \tag{3.4}$$

By using Newton's second law and using the static equilibrium position. The equations of motion can be;

The equation of motion for heave is

$$M_b\ddot{Z}_b + C_{sr}(\dot{Z}_{br} - \dot{Z}_{wr}) + K_{sr}(Z_{br} - Z_{wr}) + C_{sf}(\dot{Z}_{bf} - \dot{Z}_{wf}) + K_{sf}(Z_{bf} - Z_{wf}) - f_f - f_r = 0$$

$$\ddot{Z}_b = \frac{1}{M_b}[- C_{sr}(\dot{Z}_{br} - \dot{Z}_{wr}) - K_{sr}(Z_{br} - Z_{wr}) - C_{sf}(\dot{Z}_{bf} - \dot{Z}_{wf}) - K_{sf}(Z_{bf} - Z_{wf}) + f_f + f_r] \tag{3.5}$$

Substitute the value of Z_{br}, Z_{bf}, $\dot{Z}_{br}$ and $\dot{Z}_{bf}$ into the equation (3.5).

$$\ddot{Z}_b = \frac{1}{M_b}[- C_{sr}(\dot{Z}_b + l_r\dot{\theta} - \dot{Z}_{wr}) - K_{sr}(Z_b + l_r\theta - Z_{wr}) - C_{sf}(\dot{Z}_b - l_f\dot{\theta} - \dot{Z}_{wf}) - K_{sf}(Z_b - l_f\theta - Z_{wf}) + f_f + f_r] \tag{3.6}$$

The equation of motion for pitch is

$$I_\theta\ddot{\theta} = l_f[C_{sf}(\dot{Z}_b - l_f\dot{\theta} - \dot{Z}_{wf}) + K_{sf}(Z_b - l_f\theta - Z_{wf})] - l_r[C_{sr}(\dot{Z}_b + l_r\dot{\theta} - \dot{Z}_{wr}) - K_{sr}(Z_b + l_r\theta - Z_{wr})] - l_f f_f + l_r f_r = 0$$

$$\ddot{\theta} = \frac{1}{I_\theta}[l_f[C_{sf}(\dot{Z}_b - l_f\dot{\theta} - \dot{Z}_{wf}) + K_{sf}(Z_b - l_f\theta - Z_{wf})] - l_r[C_{sr}(\dot{Z}_b + l_r\dot{\theta} - \dot{Z}_{wr}) + K_{sr}(Z_b + l_r\theta - Z_{wr})] - l_f f_f + l_r f_r] \tag{3.7}$$

By applying Newton's second law on the front and rear wheel unsprung mass, the equation of motion can also be formulated as

Front wheel

$$M_{wf}\ddot{Z}_{wf} - C_{sf}(\dot{Z}_{bf} - \dot{Z}_{wf}) - K_{sf}(Z_{bf} - Z_{wf}) + K_{tf}(Z_{wf} - Z_{rf}) + f_f = 0$$

$$\ddot{Z}_{wf} = \frac{1}{M_{wf}}[C_{sf}(\dot{Z}_{bf} - \dot{Z}_{wf}) + K_{sf}(Z_{bf} - Z_{wf}) - K_{tf}(Z_{wf} - Z_{rf}) - f_f] \tag{3.8}$$

Substitute the value of Z_{bf} and $\dot{Z}_{bf}$ into the equation (3.8)

$$\ddot{Z}_{wf} = \frac{1}{M_{wf}}[C_{sf}(\dot{Z}_b - l_f\dot{\theta} - \dot{Z}_{wf}) + K_{sf}(Z_b - l_f\theta - Z_{wf}) - K_{tf}(Z_{wf} - Z_{rf}) - f_f] \tag{3.9}$$

Rear wheel

$$M_{wr}\ddot{Z}_{wr} - C_{sr}(\dot{Z}_{br} - \dot{Z}_{wr}) - K_{sr}(Z_{br} - Z_{wr}) - K_{tr}(Z_{wr} - Z_{rr}) + f_r = 0$$

$$\ddot{Z}_{wr} = \frac{1}{M_{wr}}[C_{sr}(\dot{Z}_{br} - \dot{Z}_{wr}) + K_{sr}(Z_{br} - Z_{wr}) + K_{tr}(Z_{wr} - Z_{rr}) - f_r] \tag{3.10}$$

Substitute the value of Z_{br} and $\dot{Z}_{br}$ and into the equation (3.10).

$$\ddot{Z}_{wr} = \frac{1}{M_{wr}}[C_{sr}(\dot{Z}_b + l_r\dot{\theta} - \dot{Z}_{wr}) + K_{sr}(Z_b + l_r\theta - Z_{wr}) + K_{tr}(Z_{wr} - Z_{rr}) - f_r] \tag{3.11}$$

From the mathematical equation passive suspension of half car ride model, Matlab/Simulink was developed to verify the result of Matlab and CARSIM. Furthermore, verify between Matlab and CARSIM, to make sure the mathematical equations for half car ride is right and accepted to use in this study. Besides that, after verify between CARSIM and Matlab have been made, the passive suspension half car model will be test with two different input first step input test and sine-wave test.

Then, vehicle suspension model will be test at three different speeds at 40km/h and 100 km/h. Besides that, there are two road input that are used in this study which is step input and sine-wave road input. Then, for the sine-wave road input it will run at frequency of 1Hz. From the Figure 3.3, it is the passive suspension system for half vehicle model.

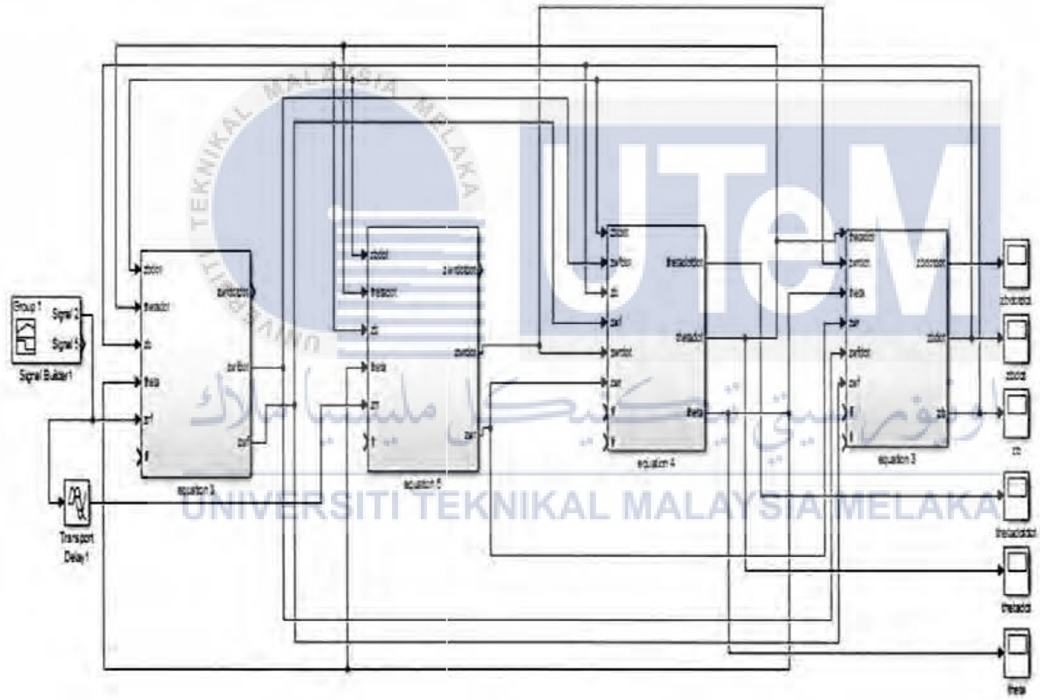

Figure 3.3 The half vehicle model subsystem Matlab/Simulink

3.5 Validation of the Passive Vehicle Model

3.5.1 Vehicle Parameters

The vehicle parameters used in this simulation shown in Table 3.1 are based on vehicle suspension system parameter in Campos et al., (1999). This parameter will be used throughout the study involving passive and active suspension system.

Table 3.1 Passenger car parameters

Parameter	Numerical Value
M_b	1500kg
M_{wr}	59kg
M_{wf}	59kg
K_{sf}	35000N/m
K_{sr}	38000N/m
K_{tf}	190000N/m
K_{tr}	190000N/m
C_{sf}	$1000\text{N}/ms^{-1}$
C_{sr}	$1100\text{N}/ms^{-1}$
I_θ	$2160kg.m^2$
l_f	$1.4m$
l_r	$1.7m$

3.5.2 Matlab Model Description and Simulation

From the Figure 3.4 it shows the subsystem developed from equation of motion in Matlab/Simulink environment. There are 4 sub-system for the half car model, from vehicle model there is subsystems of body pitch motion, heave motion, vertical motion for front and rear wheel of the vehicle.

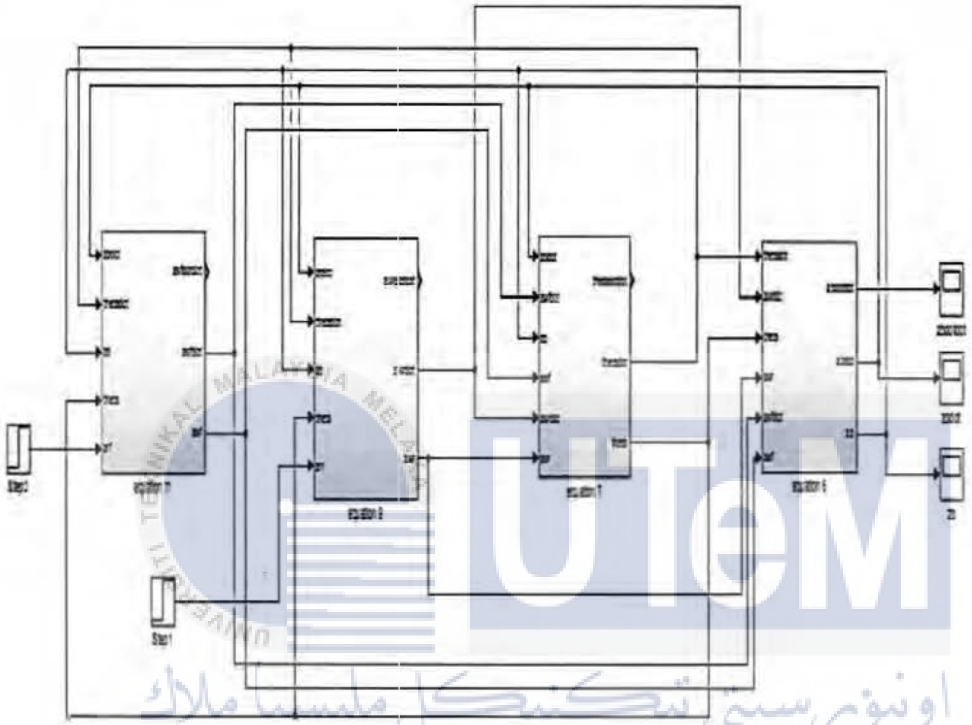

Figure 3.2 Simulation model of half vehicle model Matlab/Simulink environment

3.5.3 Verification of Half Vehicle Model in Matlab with CARSIM Software

From the four subsystem of the vehicle model of heave motion, body pitch motion, vertical motion of front and rear wheel, compress it become one subsystem. The input for the subsystem are ground elevation of front right and rear right of vehicle from the CARSIM data simulation. Then, the output for the subsystem are vertical acceleration and pitch angle of the vehicle body from the CARSIM. The result from vehicle model Matlab/Simulink will be compared with CARSIM result. The model verification was performed for a period of 20 seconds in CARSIM.

CARSIM 8 is the version for full vehicle simulation, and D-class sedan has been selected for the study as shown in Figure 3.5. After the vehicle selected, the simulation model is running on bounce sine sweep test. There are two results generated from the simulation which are the video for vehicle simulation and the other is plotting graph for the ground elevation, pitch angle and vertical acceleration of vehicle body. The results of ground elevation, vertical acceleration and pitch angle of vehicle body are shown in Figure 3.6, 3.7 and 3.8. The dynamics behavior of a vehicle obtained from CARSIM will be used as the benchmark in verification the Matlab model.

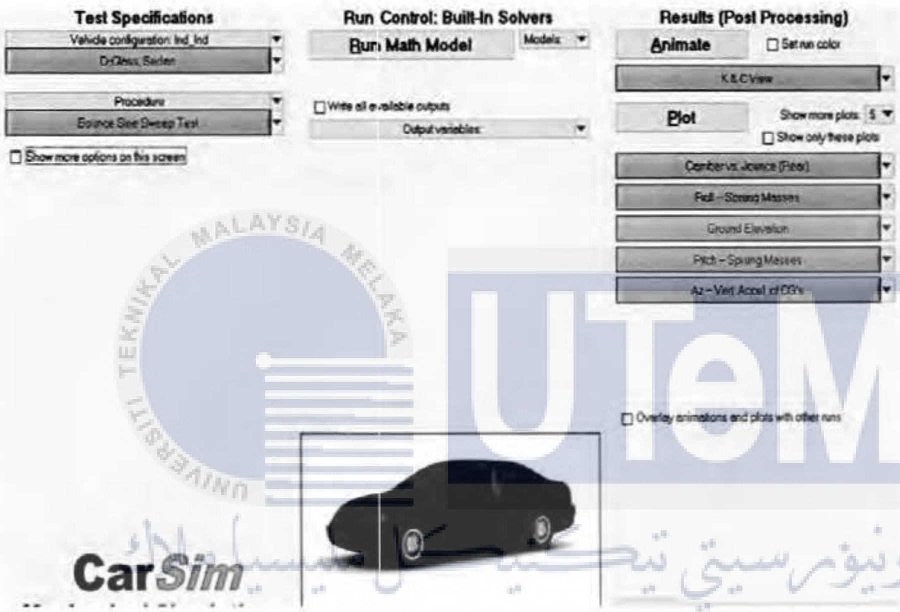

Figure 3.3 Simulation model of D-Class sedan car CARSIM

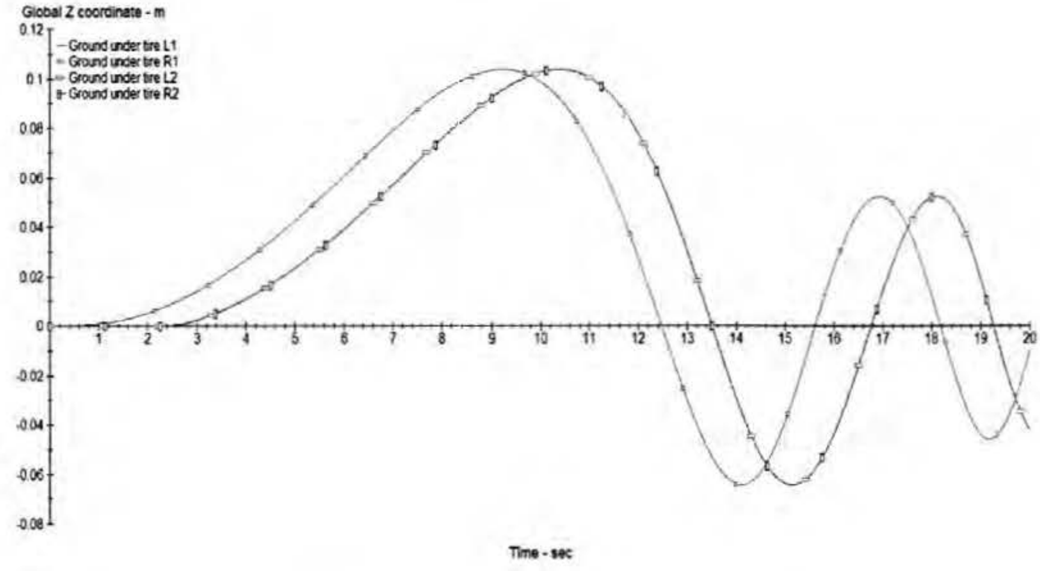

Figure 3.4 Simulation CARSIM ground elevation

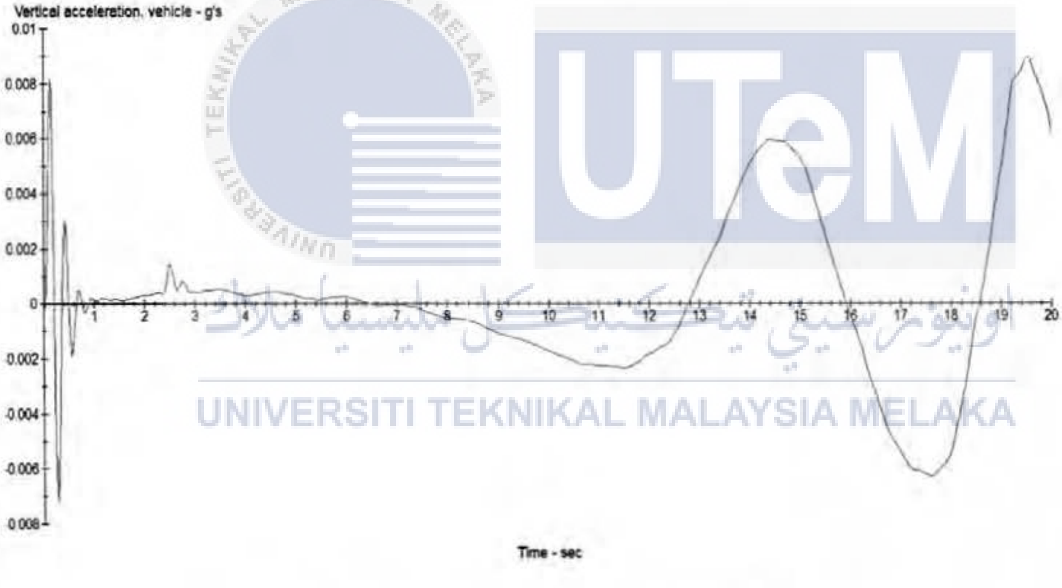

Figure 3.5 Simulation CARSIM vertical body acceleration

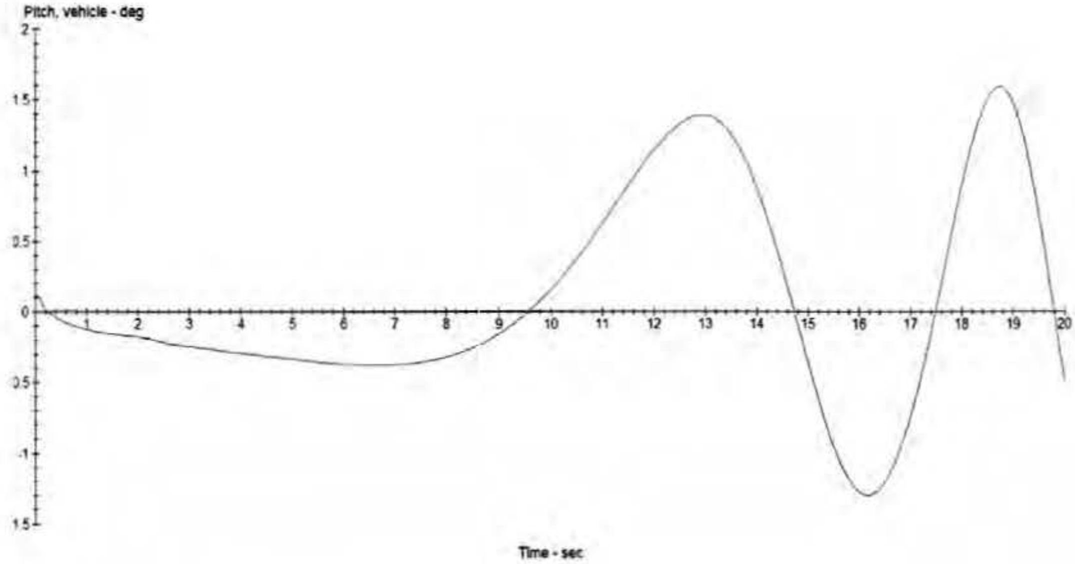

Figure 3.6 Simulation CARSIM pitch angle vehicle body

3.5.3.1 Model Verification of Vehicle Dynamics

The verification between CARSIM and Matlab/Simulink are to ensure that the subsystems for the vehicle model in the Matlab/Simulink is correct and can be used in this study. If the result are not with CARSIM, vehicle model need to be checked. Figure 3.8, shows the subsystem for half car model, it has two inputs and outputs for the subsystem.

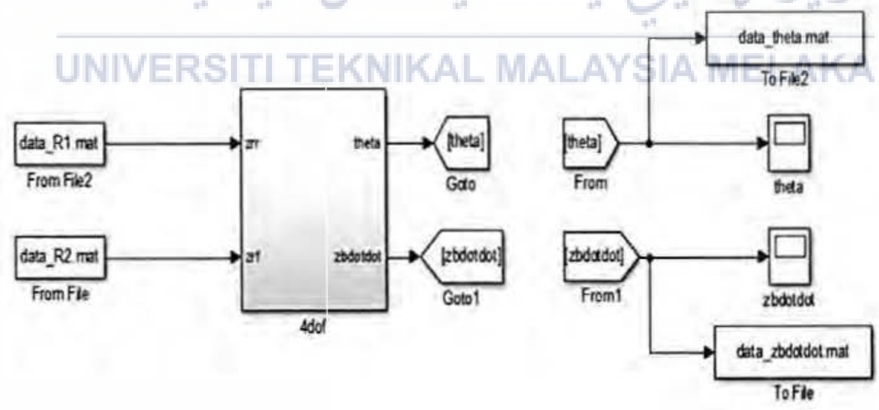

Figure 3.7 Simulation subsystem Matlab/Simulink verify with CARSIM

From the simulation in Matlab/Simulink, the result for pitch angle and vertical acceleration of vehicle body are shown Figure 3.9 and 3.10. The graph between CARSIM and Matlab are quite same, which are the blue line for Matlab and red line for CARSIM. The pattern of the line is almost the same and has a small different at certain time. By referring to the Figure 3.9, the first amplitude for CARSIM is around 1.391 degree and the first amplitude for Matlab is around 1.273 degree. The different between first amplitude for CARSIM and Matlab about 0.118 degree. Then, for the second amplitude is 1.591 degree for CARSIM graph and 1.501 degree for Matlab pitch angle. The different between Matlab and CARSIM second amplitude is 0.09 degree. Which is the different between Matlab and CARSIM become smaller at time between $17s$ and $20s$.

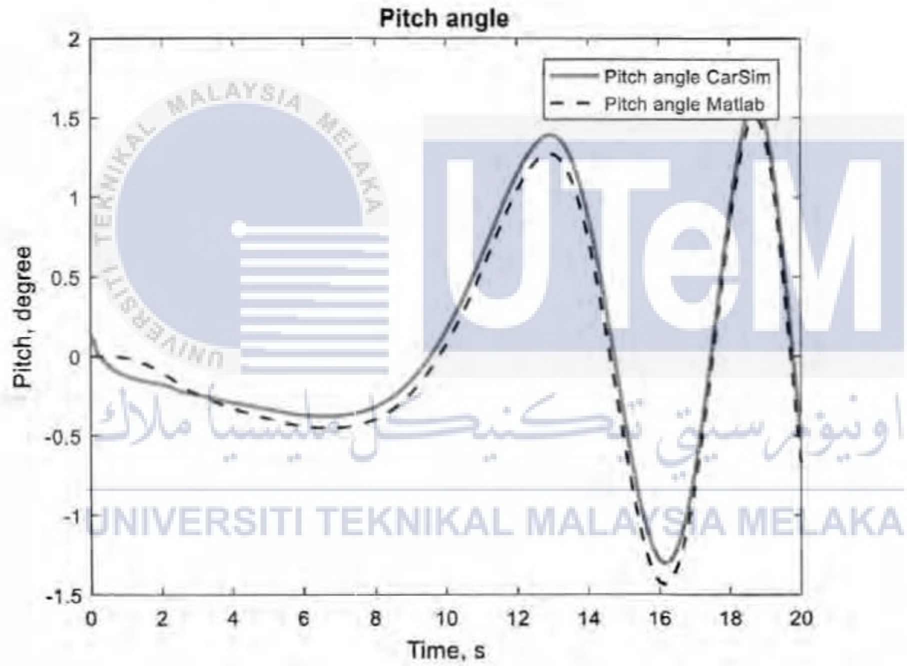

Figure 3.8 Verification of pitch angle

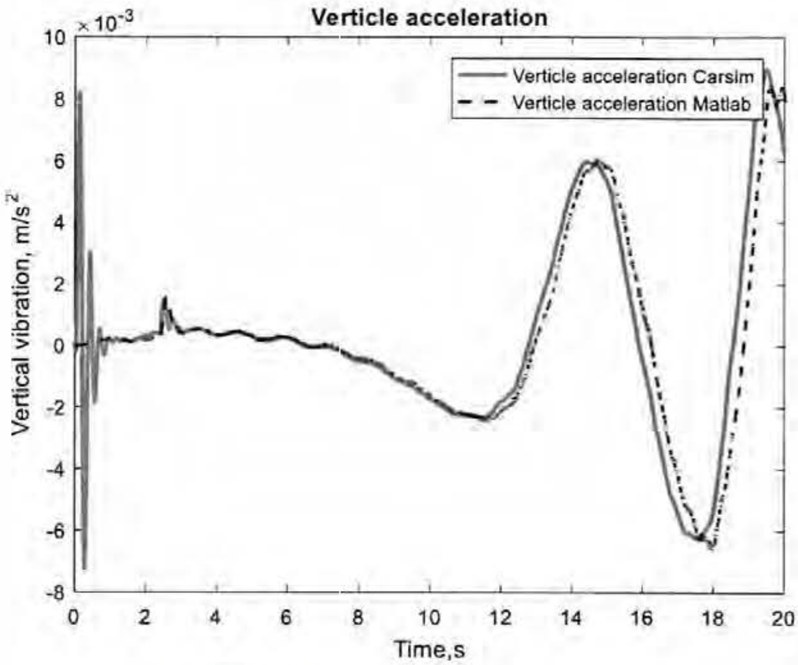

Figure 3.9 Verification of vertical acceleration vehicle body

Besides that, for Figure 3.10, the graph trend between CARSIM and Matlab is almost same at time $2s$ to $20s$. The first peak for CARSIM is at 0.001459 m/s^2 and for Matlab is at 0.001568 m/s^2. The different between this two peaks is 0.000109 m/s^2, which the peak for Matlab much higher than CARSIM. Then, the second amplitude of CARSIM is at 0.005944 m/s^2 and for Matlab is at 0.005754 m/s^2. The different between this two peaks is about 0.00019 m/s^2. The different between Matlab and CARSIM graph trend still same even there is small different amplitude at certain time.

Since, the graph trend for pitch angle and vertical acceleration for CARSIM and Matlab is almost same, the vehicle model subsystem in Matlab build from the mathematical equations is accepted. Then, the vehicle model subsystem in Matlab can be used to make subsystem for active suspension.

3.6 Active Suspension Vehicle Model

Active suspension system have use in this study, to study the impact of active suspension system to the ride comfort of the vehicle. Active suspension is different with passive suspension system, which is passive cannot control the unwanted movement of the vehicle body. But, active suspension it will produce force that will control the movement of the vehicle and reject vertical force that produce while the tire move on the road or ground terrain. Active suspension is controlled using stability augmentation system (SAS) controller in controlling an attitude and ride quality of the vehicle.

Figure 3.12 is the subsystem and vehicle model for active suspension system. The outer loop are attitude controller and the inner loop are ride controller for stability augmentation system. Outer loop controller or vehicle control is used to compute the ideal target force to cancel out unwanted vibratory motion of a vehicle. Inner loop control or actuator control is used to control the actuator in such a way that the target force.

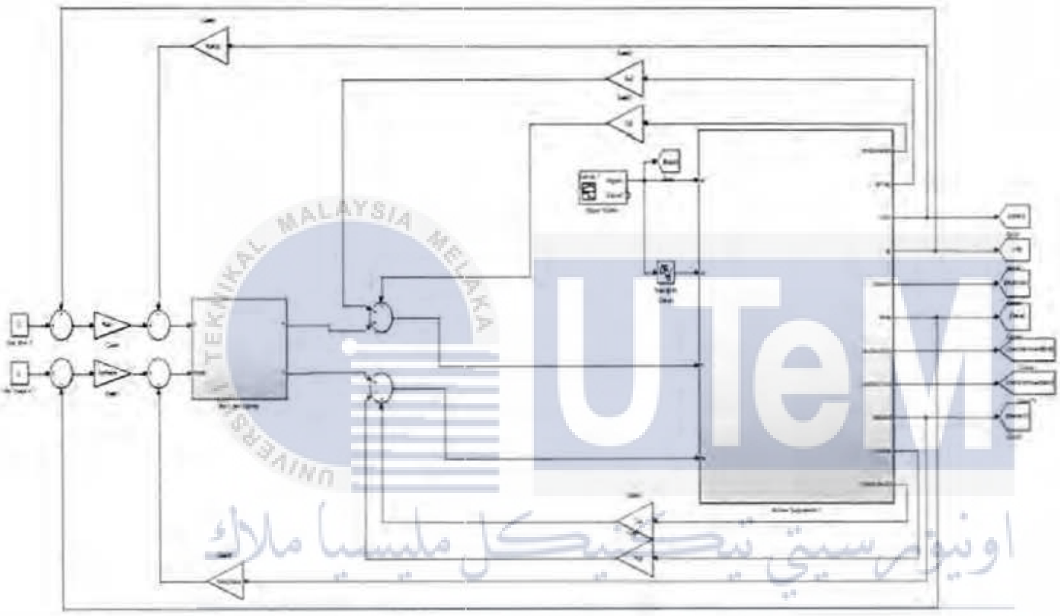

Figure 3.10 Active suspension system with stability augmentation system

3.6.1 Input Decoupling Transformation

From the heave control and pitch control equation, transform it into matrix form as $a = cb$ then rearrange it become $ac^{-1} = b$. For the $c^{-1} = c^T(cc^T)^{-1}$, transpose the c matric first before continue with the other step. Finally, get the equation for f_f and f_r, then transfer it into MATLAB for Simulink the semi-active suspension. Figure 4.2 show the Simulink block diagram for the input decoupling transformation. For the front suspension and rear suspension, pitch and heave equivalent forces can be decoupled by equation (3.12) and (3.13).

$$f_z = f_f + f_r = heave\ control \tag{3.12}$$

$$f_\theta = -af_f + bf_r = pitch\ control \tag{3.13}$$

Then, from the heave and pitch force in equation (3.12) and (3.13) express it, in matrix form.

$$\begin{bmatrix} f_z \\ f_\theta \end{bmatrix} = \begin{bmatrix} 1 & 1 \\ -a & b \end{bmatrix} \begin{bmatrix} f_f \\ f_r \end{bmatrix} \tag{3.14}$$

a c b

From the equation (3.14), transfer the matrix c to the left with matrix a. Then, equation (3.15) is the formula for the inverse of matrix c.

$$c^{-1} = c^T(cc^T)^{-1} \tag{3.15}$$

$$c = \begin{bmatrix} 1 & 1 \\ -a & b \end{bmatrix} \tag{3.16}$$

Then, transfer matrix c to the left, to be inverse of matrix c. Make transpose of matrix c as in equation (3.17).

$$c^T = \begin{bmatrix} 1 & -a \\ 1 & b \end{bmatrix} \tag{3.17}$$

Then, multiply matrix c with matrix c transpose (c^T), as in the equation (3.18).

$$cc^T = \begin{bmatrix} 1 & 1 \\ -a & b \end{bmatrix} \begin{bmatrix} 1 & -a \\ 1 & b \end{bmatrix} \tag{3.18}$$

$$cc^T = \begin{bmatrix} 1 \times 1 + 1 \times 1 & 1 \times -a + 1 \times b \\ -a \times 1 + b \times 1 & -a \times -a + b \times b \end{bmatrix} \tag{3.19}$$

$$cc^T = \begin{bmatrix} 2 & -a + b \\ -a + b & a^2 + b^2 \end{bmatrix} \tag{3.20}$$

After, get the matrix for cc^T, inverse the matrix from equation (3.20) to get the equation (3.21).

$$(cc^T)^{-1} = \begin{bmatrix} 2 & -a + b \\ -a + b & a^2 + b^2 \end{bmatrix}^{-1} \tag{3.21}$$

$$(cc^T)^{-1} = \frac{1}{(a^2+b^2)(2)-(-a+b)(-a+b)} \begin{bmatrix} a^2 + b^2 & -a + b \\ -a + b & 2 \end{bmatrix} \tag{3.22}$$

Then, equation (3.22) is the inverse of cc^T.

$$(cc^T)^{-1} = \frac{1}{a^2+b^2+2ab} \begin{bmatrix} a^2 + b^2 & a - b \\ a - b & 2 \end{bmatrix} \tag{3.23}$$

$$c^T(cc^T)^{-1} = \begin{bmatrix} 1 & -a \\ 1 & b \end{bmatrix} \cdot \frac{1}{(a^2+b^2)(2)-(-a+b)(-a+b)} \begin{bmatrix} a^2 + b^2 & -a + b \\ -a + b & 2 \end{bmatrix} \tag{3.24}$$

Equation (3.25) is the matrix for $c^T(cc^T)^{-1}$.

$$c^T(cc^T)^{-1} = \begin{bmatrix} \frac{b}{a+b} & \frac{-1}{a+b} \\ \frac{a}{a+b} & \frac{1}{a+b} \end{bmatrix} \tag{3.25}$$

$$c^{-1} = c^T(cc^T)^{-1} = \frac{1}{a+b}\begin{bmatrix} b & -1 \\ a & 1 \end{bmatrix} \tag{3.26}$$

Then, the equation (3.27) is the input of decoupling for 4DOF suspension for pitch.

$$\begin{bmatrix} f_f \\ f_r \end{bmatrix} = \frac{1}{a+b}\begin{bmatrix} b & -1 \\ a & 1 \end{bmatrix}\begin{bmatrix} f_z \\ f_\theta \end{bmatrix} \tag{3.27}$$

From equation (3.28) and (3.29) transfer it into Simulink with stability augmentation system.

$$f_f = \frac{1}{a+b}[bf_z - f_\theta] \tag{3.28}$$

$$f_r = \frac{1}{a+b}[af_z + f_\theta] \tag{3.29}$$

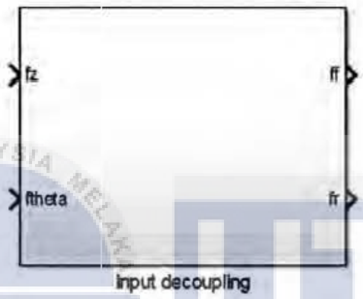

Figure 3.11 Block diagram for the input decoupling transformation.

Input decoupling use in the Simulink is to adjust the height of the suspension or force produce by the suspension in order to stabilize the vehicle during cornering or on the bump road surface. It will process the controller input before goes to half car subsystem model.

3.6.2 Road Input for Simulation

Passive and active suspension will be run on two type of road input, one is step input and the other one is sine wave input. Since, the vehicle will be run on two different at $40km/h$ and $100km/h$. Front tire will be hit the road input first and then the rear tire will hit the at certain delay time. Then, to find time for rear tire hit the road input, it can be calculated by using velocity equation. For the sample calculation of time delay for rear tire at speed $40km/h$ can be referring at appendices.

3.7 Summary

A half vehicle model 4-DOF has been developed and mathematical equations has been derived based on Newton's second law. The verification of half vehicle mathematical model with CARSIM model were implemented. It can be said that the model is valid due to the trend of the model response is closely similar magnitude of CARSIM and Matlab graph. It can be concluded that the half car model developed is realistic to use for further investigation.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Introduction

This chapter discusses the results of passive and active suspensions simulation. For the active suspension with stability augmentation system, it was controlled with three different controllers, which are attitude control, ride control and combination of attitude and ride controls. The vehicle are tested on two different road surfaces which are step and bumpy road. For each road condition vehicle are sets at a different speed limit of 40km/h and 100km/h. The simulation results will be evaluated and discussed the vertical acceleration and vertical displacement of the vehicle body.

4.2 Passive Suspension

The simulation will be run in different speed at 40km/h and 100km/h and also road input for simulation are step input and sine wave. The results of passive suspension will be compared with active suspension system to reduce the vibration and improve the ride comfort.

4.2.1 Step Input Road Condition

The height for the step input is at 0.05m, and the vehicle hit this step road surface at the speed of 40km/h and 100km/h.

4.2.1.1 Vehicle Speed at 40km/h

From Figure 4.1 and Figure 4.2, it shows the simulation result of vertical body acceleration and vertical body displacement for passive suspension. Figure 4.1, it show that when the vehicle is driven with speed of 40km/h on the uneven road surface, the vertical body acceleration is around 2.274m/s^2 at 1.037 sec. Besides that, the vertical body displacement will move upward about 0.08844m at time 1.53 sec.

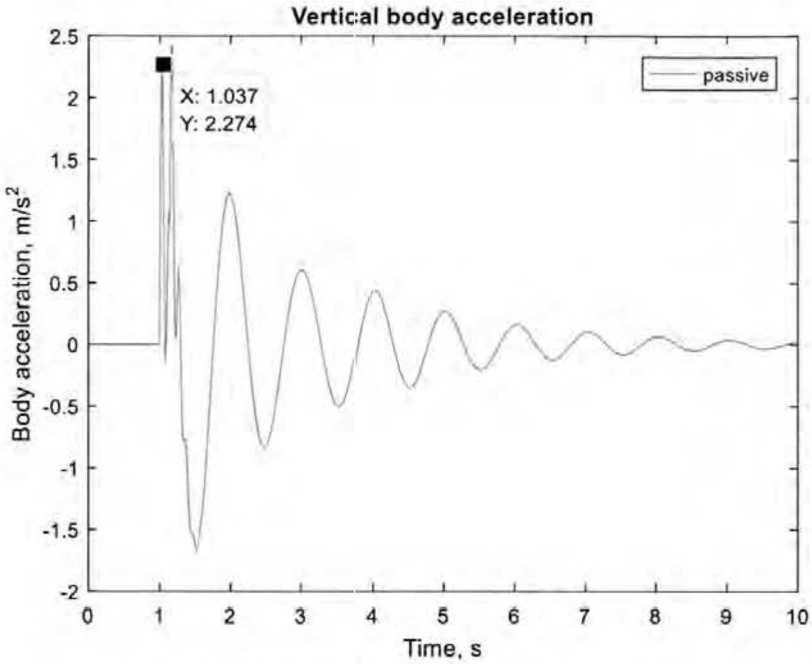

Figure 4.1 Vertical body acceleration step input at 40km/h

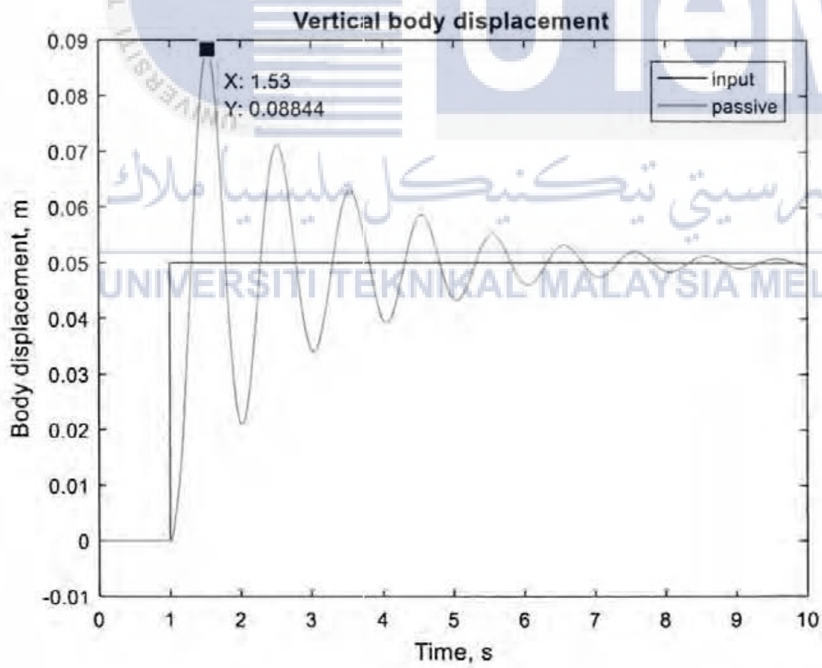

Figure 4.2 Vertical body displacement step input at 40km/h

4.2.1.2 Vehicle Speed at 100km/h

Figure 4.3 and Figure 4.4, it shows the simulation results of vertical body acceleration and vertical body displacement for passive suspension system. From Figure 4.3, the vertical body acceleration is around 2.272m/s^2 at time 1.037 sec when vehicle driven at speed 100km/h with step input height 0.05m. Besides that, from Figure 4.4 the vehicle body moved upward about 0.09003m at time 1.504 sec, which is effect the ride comfort of the vehicle.

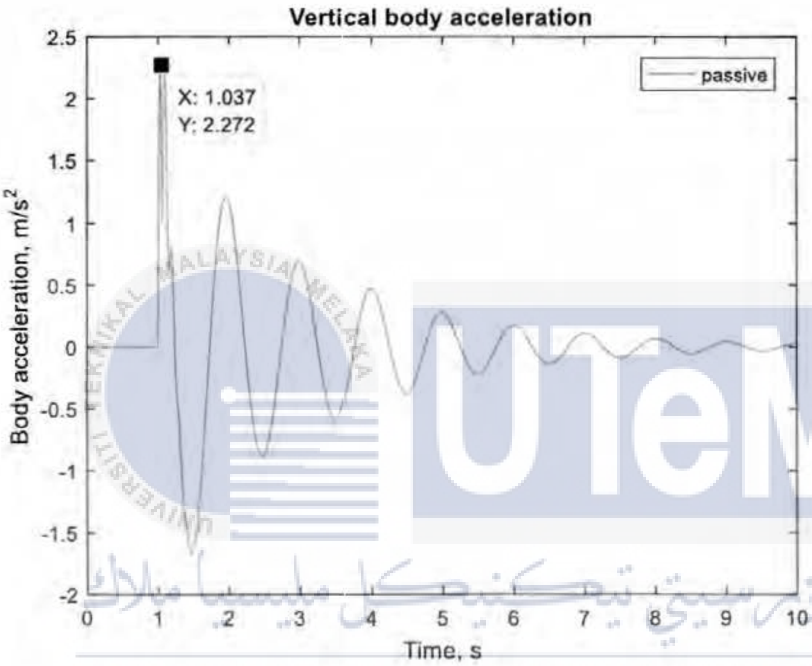

Figure 4.3 Vertical body acceleration step input at 100km/h

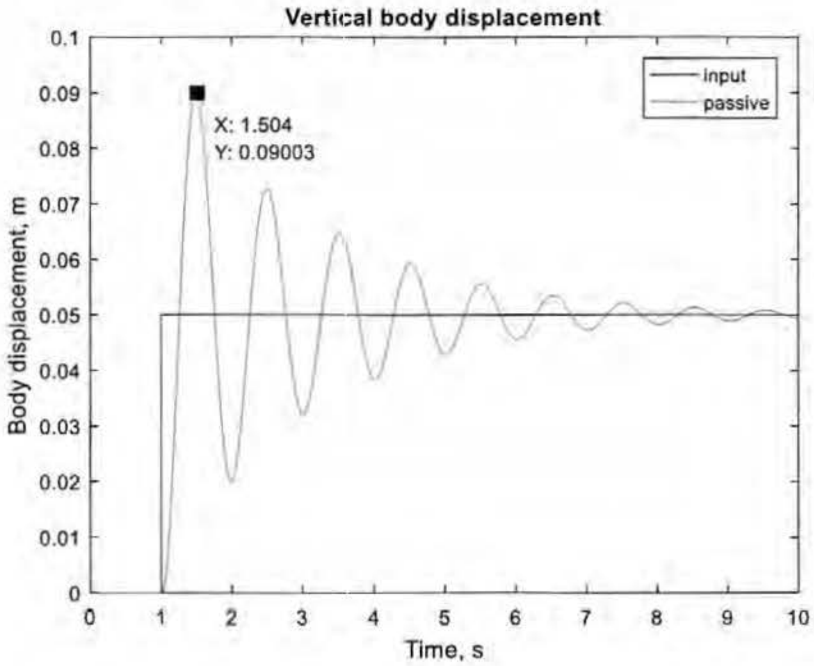

Figure 4.4 Vertical body displacement step input at 100km/h

4.2.2 Sine Wave Road Condition

The height for the sine wave input is at 0.05m and frequency equal to 1Hz. The vehicle hit this bump at the speed of 40km/h and 100km/h.

4.2.2.1 Vehicle Speed at 40km/h

By referring to Figure 4.5 and Figure 4.6, it shows the simulation results of vertical body acceleration and vertical body displacement for passive suspension system. From Figure 4.5, the vertical body acceleration is around 0.7437m/s^2 at time 0.2222 sec when the vehicle driven at speed 40km/h with sine wave input height is 0.05m. The vehicle body will moved upward about 0.07393m at time 0.7025 sec, this will make the vehicle body moving upward and downward and will reduce the vehicle ride comfort.

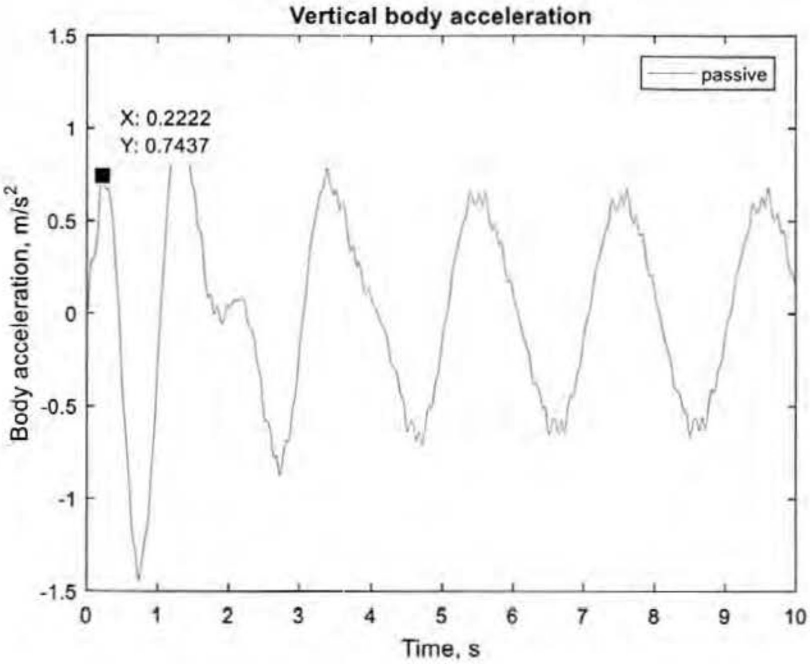

Figure 4.5 Vertical body acceleration sine wave input at 40km/h

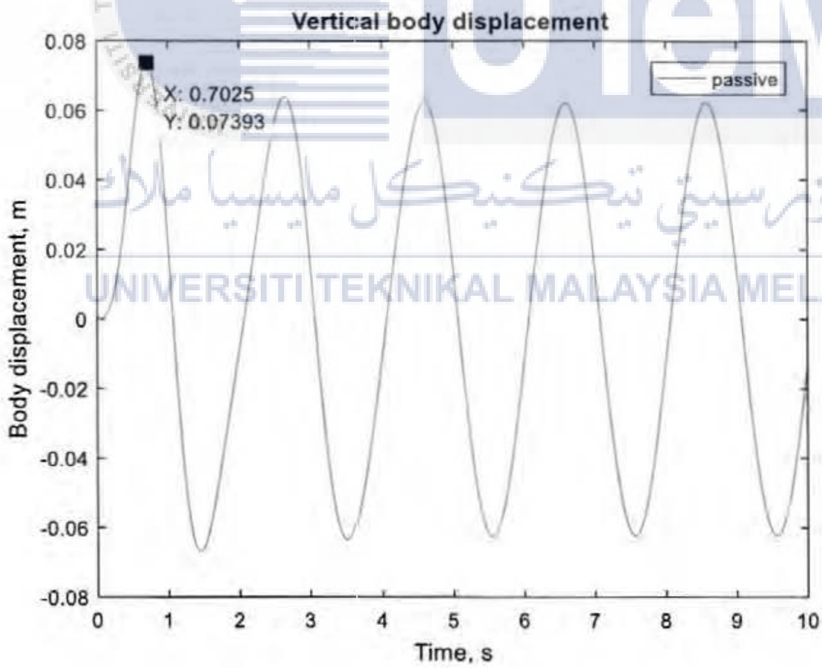

Figure 4.6 Vertical body displacement sine wave input at 40km/h

4.2.2.2 Vehicle Speed at 100km/h

Figure 4.7 is the vertical body acceleration for passive suspension with speed of 100km/h, the first amplitude for the sine wave input is 0.8478m/s^2, while Figure 4.8 is the vertical displacement is around 0.07562m at time equal to 0.6667s. The vehicle body moved upward about 0.02562m and continue moving upward and downward.

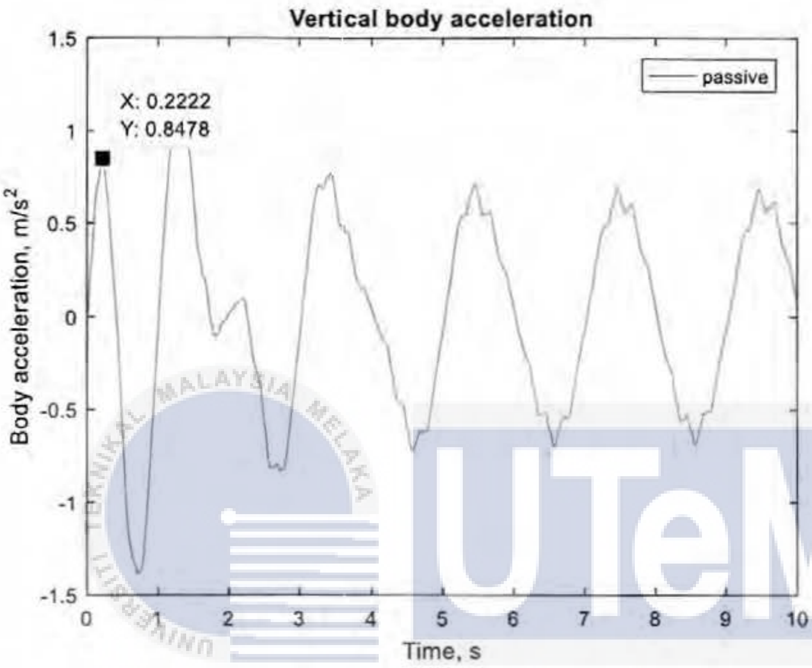

Figure 4.7 Vertical body acceleration sine wave input at 100km/h

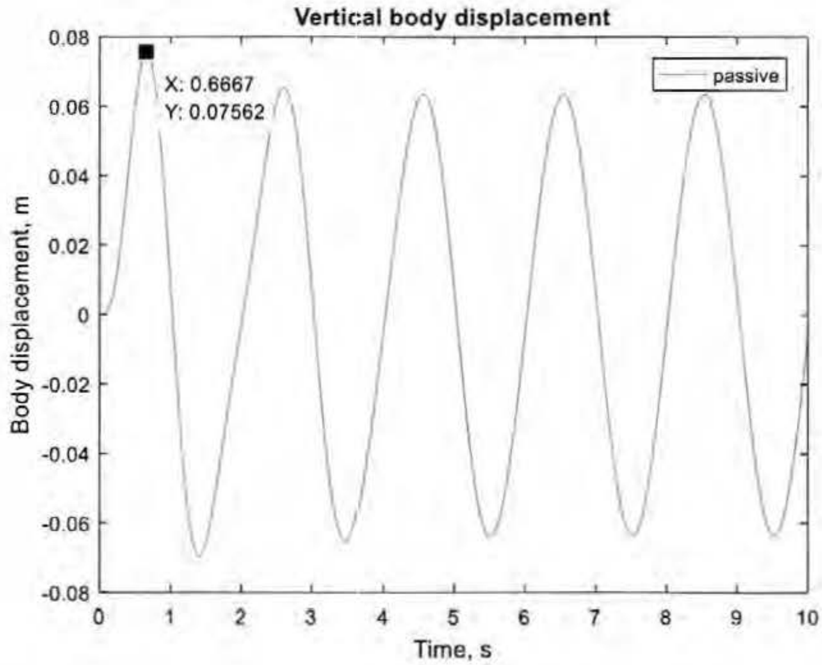

Figure 4.8 Vertical body displacement sine wave input at 100km/h

4.3 Active Suspension with Stability Augmentation System

For active suspension, it will be controlled by stability augmentation system, to reduce the vertical vehicle acceleration, vertical body displacement and also improve the ride comfort. The stability augmentation system consists of attitude and ride controllers, and the simulation will be performed in three condition which are attitude, ride and combination of attitude and ride.

4.3.1 Step Input Road Condition

This part will discuss on the result from attitude controller, ride controller and combination of attitude and ride controller. At vehicle speed of 40km/h and 100km/h, and the step input height is 0.05m and time for the simulation is during 10s.

4.3.1.1 Vehicle Speed at 40km/h

Active suspension system has been control using stability augmentation system. The controller has been tuned to get the optimum peak that can reduce vibration on the vehicle body. Table 4.1 is the fine-tuned numerical gain values of the controller parameters for the attitude control, ride control and attitude combine with ride control. The result from each controllers are compared. For attitude and ride controller, it begin by controlling attitude control parameter and follow by tuning the parameter for ride controller. The graph sensitivity for controller parameter can be refer in appendix.

Table 4.1 Numerical value of controller parameter step input at 40km/h

Controller	Controller parameter							
	K_{sz}	$K_{s\theta}$	K_{skyz}	$K_{sky\theta}$	K_{sf}	C_{sf}	K_{sr}	C_{sr}
Attitude	130	700	11000	10	-	-	-	-
Ride	-	-	-	-	10	10	6000	600
Attitude + ride	130	700	11000	10	10	80	900	20

Table 4.2, shows the simulation results of vertical body acceleration and vertical body displacement for passive, attitude control, ride control and combination of ride and attitude controllers. The combination of attitude and ride controllers simulation result is better than ride controller and attitude controller. It is because combination of attitude and ride controllers able to reduce the peak amplitude of the vehicle acceleration from 2.274m/s^2 to 2.055m/s^2 and about 9.63% reduction than other controllers. Ride controller only able to reduce the vertical body acceleration and vertical body displacement about 1.5% and 1.65%, a slight reduction compared to other controllers.

Table 4.2 Peak value for vehicle speed step input at 40km/h

Peak value	Body responses	
	Acceleration (m/s^2)	Displacement (m)
Passive	2.274	0.08844
SAS (ride)	2.24	0.08698
Reduction (%)	1.5	1.65
SAS (attitude)	2.09	0.05052
Reduction (%)	8.09	42.88
SAS (attitude+ride)	2.055	0.05006
Reduction (%)	9.63	43.4

From Figure 4.9 and 4.10, it shows the graph trend simulation results of vertical body acceleration and vertical body displacement for passive and stability augmentation system controllers. The attitude control and combination of attitude and ride control, graph pattern are quite similar. Furthermore, attitude control able to reduce the peak amplitude and improve the settling time for the vehicle body. By referring Figure 4.10, vertical body displacement is around 0.05006m peak amplitude reduction. That is better compare to passive suspension, which peak amplitude around 0.08844m. From the results, stability augmentation system, combination of ride and attitude controllers with active suspension able to improve the ride comfort when the vehicle driven at speed 40km/h with step input height is 0.05m.

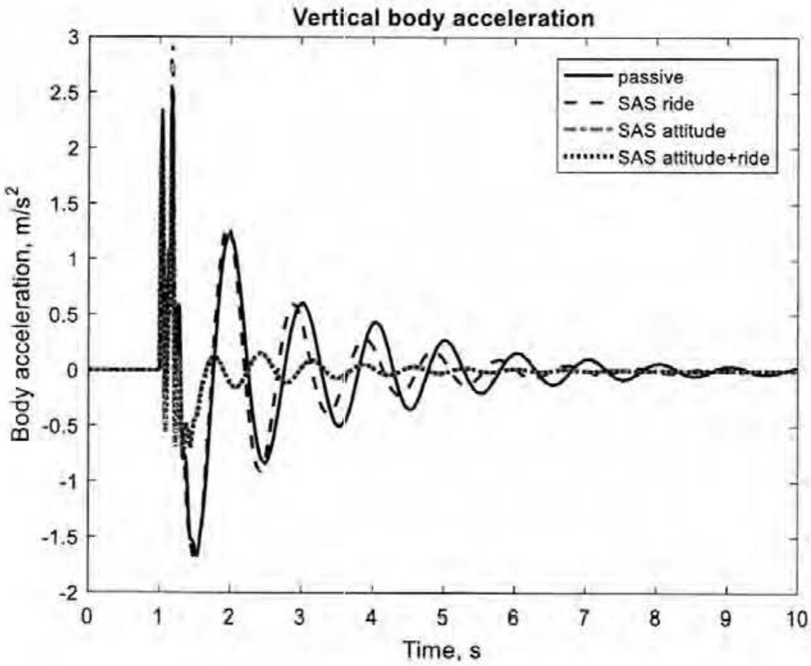

Figure 4.9 Vertical body acceleration step input at speed 40km/h

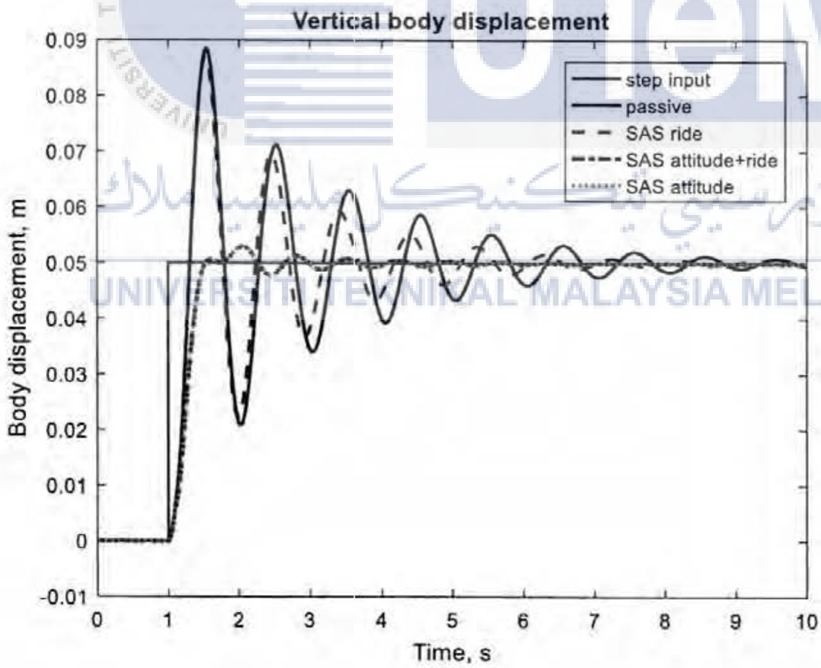

Figure 4.10 Vertical body displacement step input at speed 40km/h

4.3.1.2 Vehicle Speed at 100km/h

The Simulink model is composed three types of controllers, which are half car with ride controller, attitude controller and combination of attitude and ride controllers. The vehicle model will is set at speed 100km/h. The step input function will be connected to both of the input port front and rear tyre of subsystem. The step function have been set at 1 sec with 0.05m height. Table 4.3 is the controller parameter, that have been tuned for each stability augmentation system controllers to achieve the optimum peak amplitude reduction from passive suspension.

Table 4.3 Numerical value of controller parameter step input at 100km/h

Controller	Controller parameter							
	K_{sz}	$K_{s\theta}$	K_{skyz}	$K_{sky\theta}$	K_{sf}	C_{sf}	K_{sr}	C_{sr}
Attitude	130	10000	11000	100	-	-	-	-
Ride	-	-	-	-	10	10	20	700
Attitude + ride	130	10000	11000	100	10	10	3000	10

Table 4.4, it shows that the simulation results of vertical body acceleration and vertical body displacement for ride controller, attitude controllers, combination of attitude and ride controllers and passive suspension. The combination of attitude and ride controllers simulation result is better than ride controller and attitude controller. It is because the combination of attitude and ride controllers is able to reduce the vertical body acceleration and vertical body displacement, around 7.61% and 44.03% compared to other controllers.

Table 4.4 Peak value for vehicle speed step input at 100km/h

Peak value	Body responses	
	Acceleration (m/s^2)	Displacement (m)
Passive	2.272	0.09003
SAS (ride)	2.179	0.08742
Reduction (%)	4.09	2.90
SAS (attitude)	2.098	0.05064
Reduction (%)	7.66	43.75
SAS (attitude+ride)	2.099	0.05039
Reduction (%)	7.61	44.03

From Figure 4.11 and 4.12, it shows that the simulation result of vertical body acceleration and vertical body displacement for stability augmentation system controllers and passive suspension. The attitude control simulation result is better than ride control simulation results. It is because attitude control is able to reduce the peak amplitude of vertical acceleration and vertical displacement the reduction is around 7.66% and 43.75% than ride controller around 4.09% and 2.90%. The combination of ride and attitude controllers will improve the settling time for vertical body displacement, where at 6 sec the vehicle will not continue moving upward and downward. The combination of attitude and ride controller is able to improve the ride comfort of the driver and passengers while vehicle travel on step input height is 0.05m at speed 100km/h.

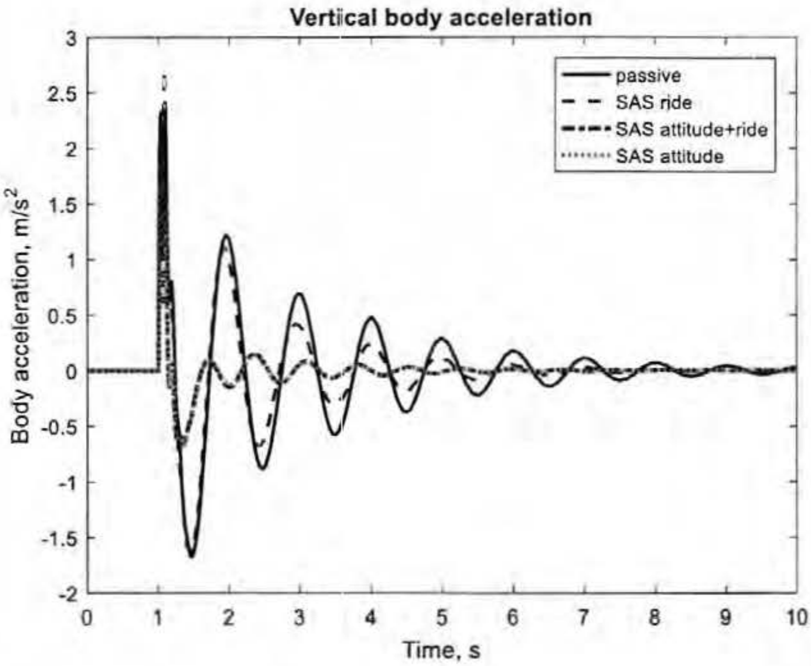

Figure 4.11 Vertical body acceleration step input at speed 100km/h

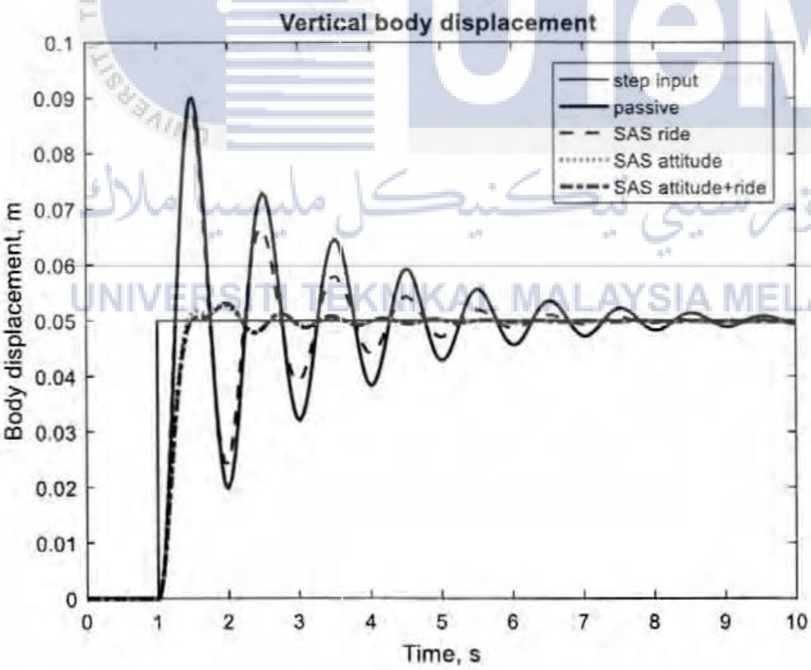

Figure 4.12 Vertical body displacement step input at 100km/h

4.3.2 Sine Wave Input Road Condition

Passenger vehicle have been run simulation with sine wave input, it will be run at different speeds at 40km/h and 100km/h. By running the simulation it will show, the optimum controllers can tune the parameter of the active suspension with stability augmentation system in order to reduce vertical body acceleration and vertical body displacement.

4.3.2.1 Vehicle Speed at 40km/h

The active suspension with stability augmentation system with attitude, ride and combination of attitude with ride controller have been used to reduce vertical body acceleration and vertical body displacement. Then, Table 4.5 is the numerical value for parameter of the controller while the vehicle passenger speed at 40km/h.

Table 4.5 Numerical value of controller parameter sine wave input at 40km/h

Controller	Controller parameter							
	K_{sz}	$K_{s\theta}$	K_{skyz}	$K_{sky\theta}$	K_{sf}	C_{sf}	K_{sr}	C_{sr}
Attitude	2000	1800	40000	50	-	-	-	-
Ride	-	-	-	-	10	7000	50	10
Attitude + ride	2000	1800	40000	50	20	20	10	20

Result of active suspension with stability augmentation system have been tabulated in Table 4.6. From Table 4.6, the vertical body acceleration for passive is equal to 0.7437m/s^2 and the vertical body displacement is equal to 0.07393m at the first peak.

Table 4.6 Peak value for vehicle speed sine wave input at 40km/h

Peak value	Body responses	
	Acceleration (m/s^2)	Displacement (m)
Passive	0.7437	0.07393
SAS (ride)	0.584	0..06373
Reduction (%)	21.47	13.80
SAS (attitude)	0.2495	0.02478
Reduction (%)	66.45	66.48
SAS (attitude+ride)	0.2489	0.02477
Reduction (%)	66.53	65.50

By refer to Figure 4.13 the vertical body acceleration amplitude are not stable, which the amplitude is moving upward and downward rapidly at the certain time. From Figure 4.14 the graph trend for attitude control and attitude with ride control are quite similar which amplitude for attitude controller around 0.02478m and attitude with ride control is 0.02477m. The difference between this two controllers is small compare to ride controller. Ride controller reduction is too small, reduction of displacement is only 13.80% and the reduction for acceleration of vehicle body is only 21.47%.

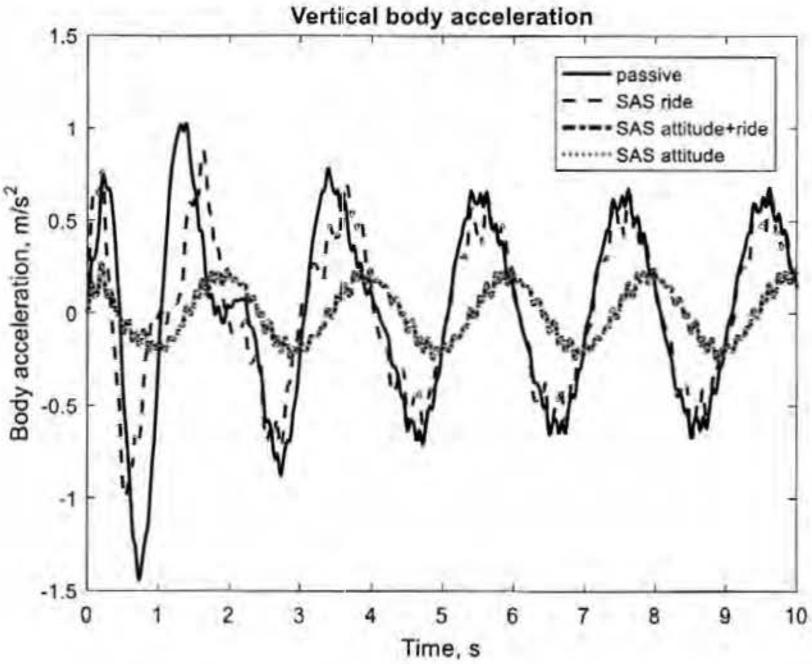

Figure 4.13 Vertical body acceleration sine wave input at speed 40km/h

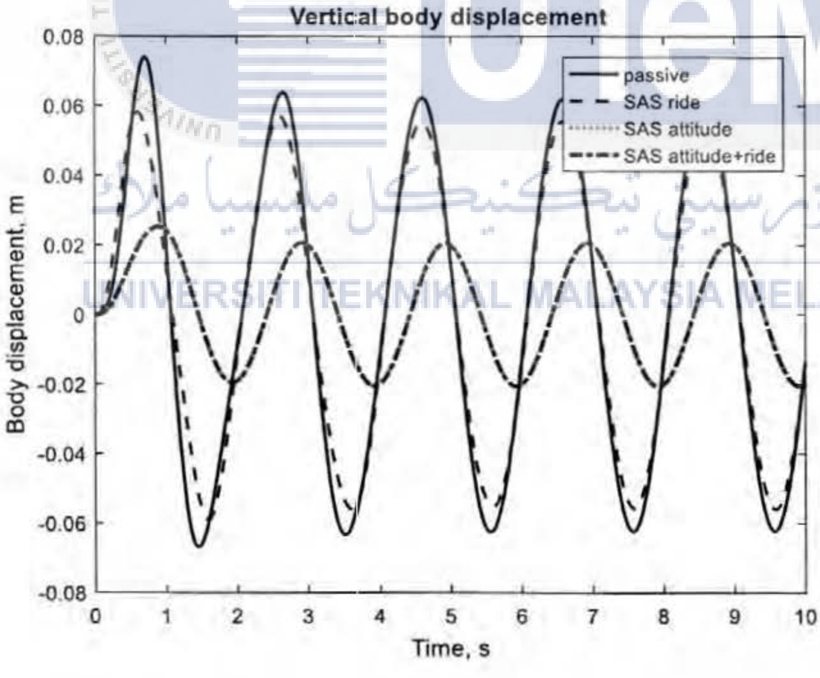

Figure 4.14 Vertical body displacement sine wave input at speed 40km/h

4.3.2.2 Vehicle Speed at 100km/h

This part will discuss the result of vehicle passive and active suspension by using stability augmentation system. Vehicle suspension model will run at speed 100km/h with sine wave input height 0.05m. From Table 4.7, it is the numerical value controller parameter for each controllers. The controllers have been tune until it achieve the optimum reduction amplitude of the vertical body acceleration and vertical body displacement.

Table 4.7 Numerical value of controller parameter sine wave input at 100km/h

Controller	Controller parameter							
	K_{sz}	$K_{s\theta}$	K_{skyz}	$K_{sky\theta}$	K_{sf}	C_{sf}	K_{sr}	C_{sr}
Attitude	7000	2000	40000	200	-	-	-	-
Ride	-	-	-	-	20	3000	10	10
Attitude + ride	7000	2000	40000	200	20	10	20	10

By refer to Table 4.8, the peak value for vertical body acceleration passive suspension is around $0.8478 m/s^2$ and vertical body displacement is 0.07562m. This will affect the ride comfort of driver and passengers during driving on sine wave input at speed 100km/h, it will make vehicle moving upward and downward rapidly.

Table 4.8 Peak value for vehicle speed sine wave input at 100km/h

Peak value	Body responses	
	Acceleration (m/s^2)	Displacement (m)
Passive	0.8478	0.07562
SAS (ride)	0.7774	0.06718
Reduction (%)	8.30	11.16
SAS (attitude)	0.2681	0.02523
Reduction (%)	68.38	66.64
SAS (attitude+ride)	0.2648	0.02522
Reduction (%)	68.77	66.65

Then, from Figure 4.15, the vertical body acceleration have been reduced by using SAS with attitude and ride control for active suspension system. The reduction from the passive suspension is height, around 68.77% have been reduce. After that, the vertical body displacement have been reduced by using attitude with ride control than passive suspension in the Figure4.16. The percentage reduction of displacement is equal to 66.65%. Besides that, by refer to Figure 4.15 and 4.16 the used of active suspension with attitude will give the big impact to the vehicle ride comfort, which is vertical acceleration and displacement have been reduce more. Compare to ride control with active suspension, the reduction of vertical body acceleration is only 8.30% and vehicle body displacement only 11.16%. However, combination of ride and attitude control with active suspension will get better result of vertical acceleration and body displacement reduction.

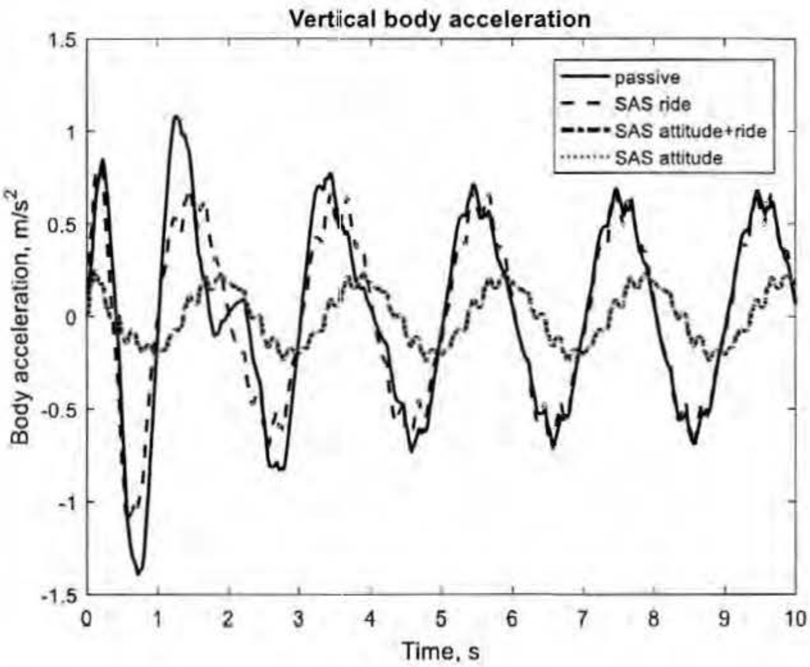

Figure 4.15 Vertical body acceleration sine wave input at speed 100km/h

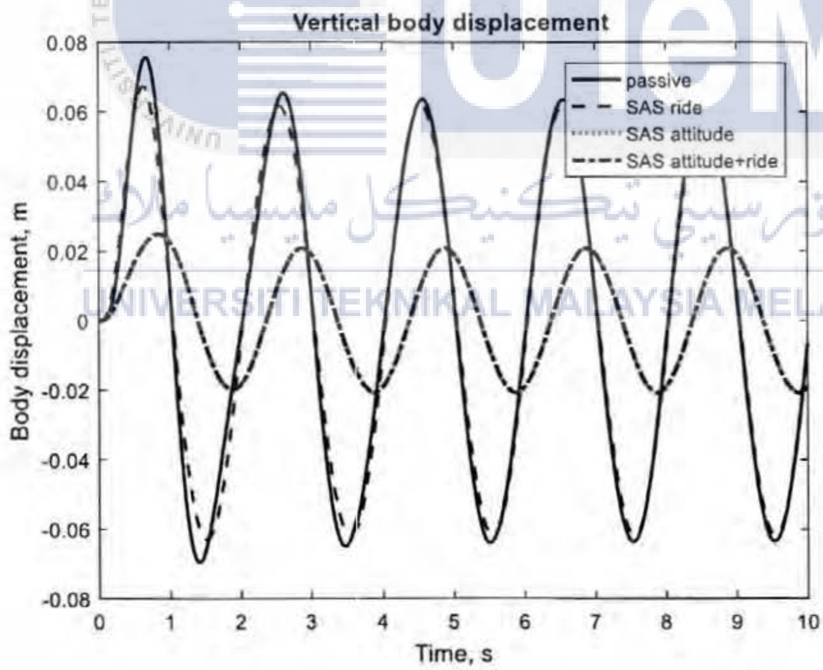

Figure 4.16 Vertical body displacement sine wave input at speed 100km/h

4.4 Summary

From the simulation results, it can be seen that active suspension with stability augmentation system by using attitude and ride control is able to reduce both the amplitude and settling time of unwanted body motions in the form of body displacement and body acceleration as compared with attitude controller, ride controller and passive suspension system.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS FOR FUTURE RESEARCH

5.1 Introduction

In this study, the use of active suspension with stability augmentation system to improve the vertical motion, ride comfort and stability of vehicle was explored. It begin by derive a 4-DOF mathematical equations for half car ride model. After, the vehicle model has been built by using mathematical equations, it has been verified with CARSIM. Sensitivity study for the tuning control parameter was conducted for each controller.

5.2 Conclusion

From the results and discussion presented in this study, it can be concluded that a half vehicle model and control strategy for active suspension with stability augmentation system have been developed. The results of half vehicle model verification with CARSIM showed that the graph trend are similar but different in magnitude at certain time. By referring to Table 4.2, active suspension with stability augmentation system control by attitude and ride control can reduce the peak of the vehicle acceleration.

Besides that, from the Table 4.4, stability augmentation system control by attitude and ride has been reduced the peak of vertical body acceleration. From simulation it shows that active suspension with stability augmentation system with attitude and ride controller will reduce the amplitude of the vertical body acceleration body displacement more than attitude and ride controller. Furthermore, between attitude and ride controllers, attitude will reduce the amplitude and improve settling time and ride comfort rather than ride attitude. Active suspension system has better performance capabilities over passive suspension system.

5.3 Recommendation for Future Research

From the simulation result, it shown that active suspension with stability augmentation system can reduce the vertical body acceleration and displacement at speed 40km/h and 100km/h with road input step and sine wave. It is required to conduct an experiment on vehicle with active suspension using stability augmentation system, in order to validate the simulation results and experimental results. For future work, need to build full car ride model that will study the roll motion, pitch motion by using stability augmentation system.

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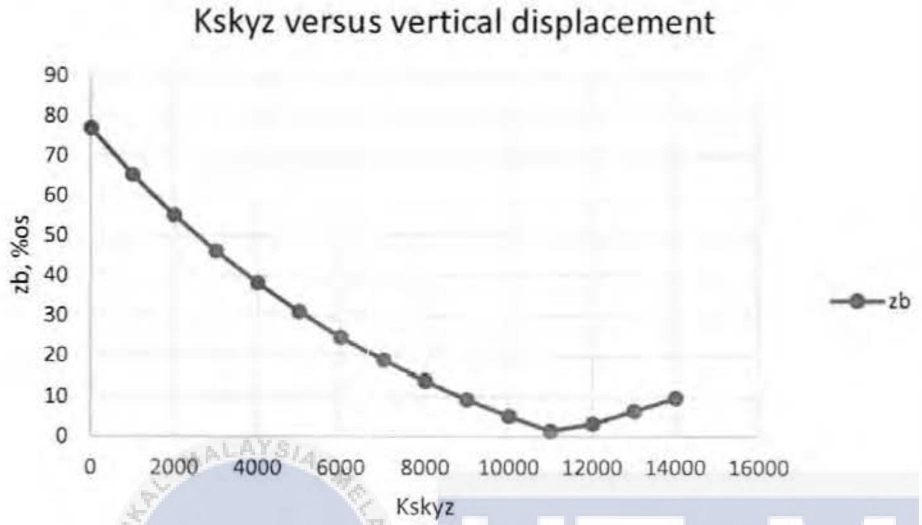
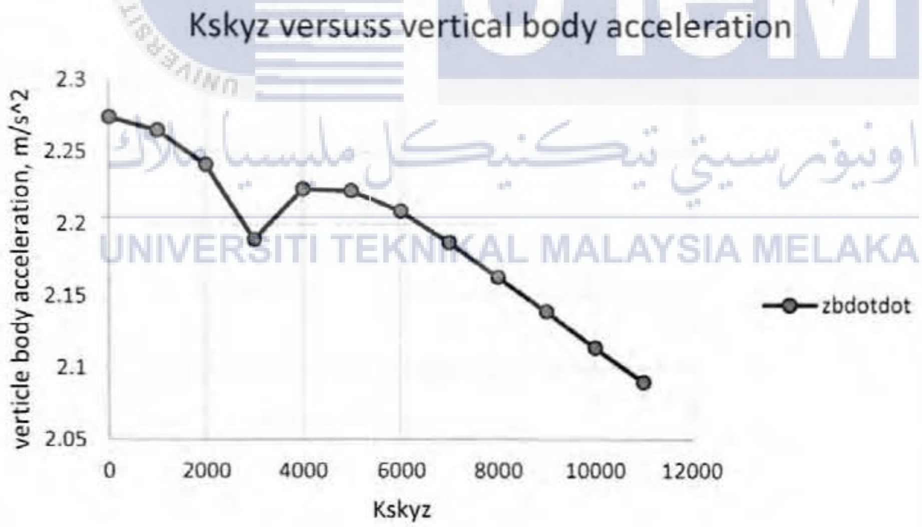
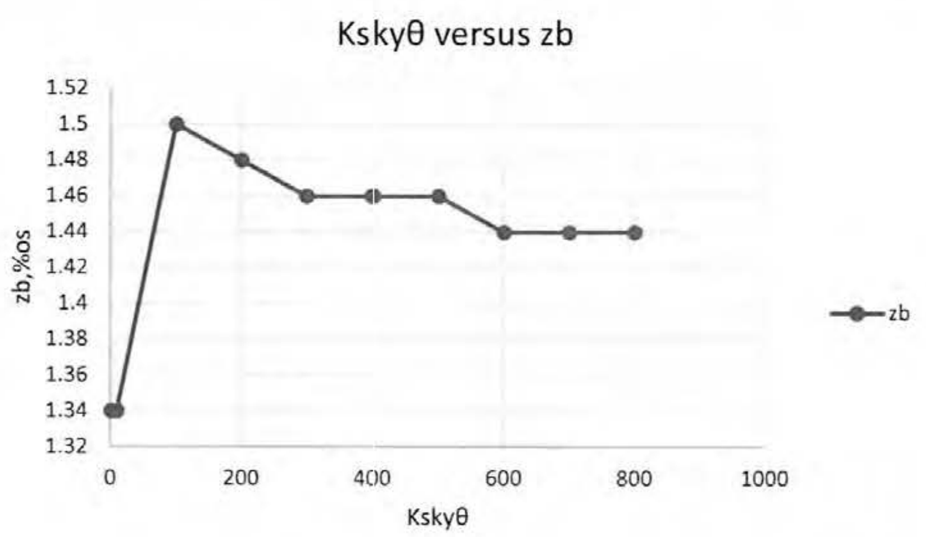
APPENDICES

A Controller parameter step input at 40km/h

- SAS attitude and attitude with ride controller



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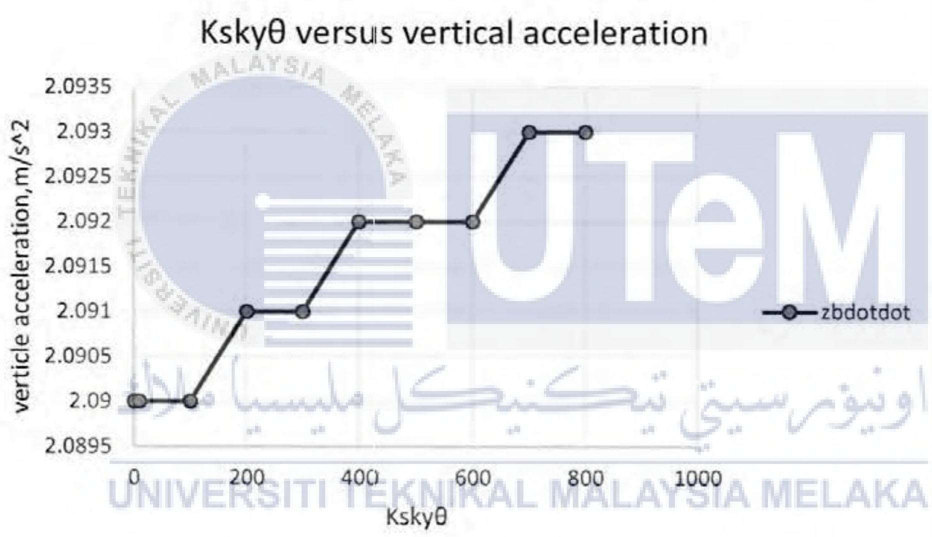
800
1000
Kskyθ
zb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2.092
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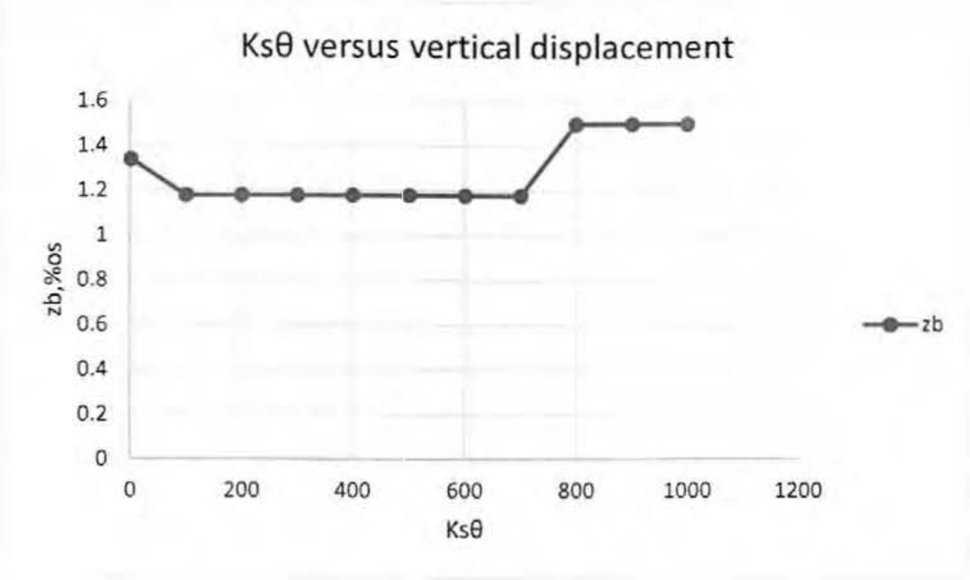
200
400
600
800
1000
Kskyθ
zbdotdot



Ksθ versus vertical displacement
zb,%os
Ksθ
zb
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1
1.2
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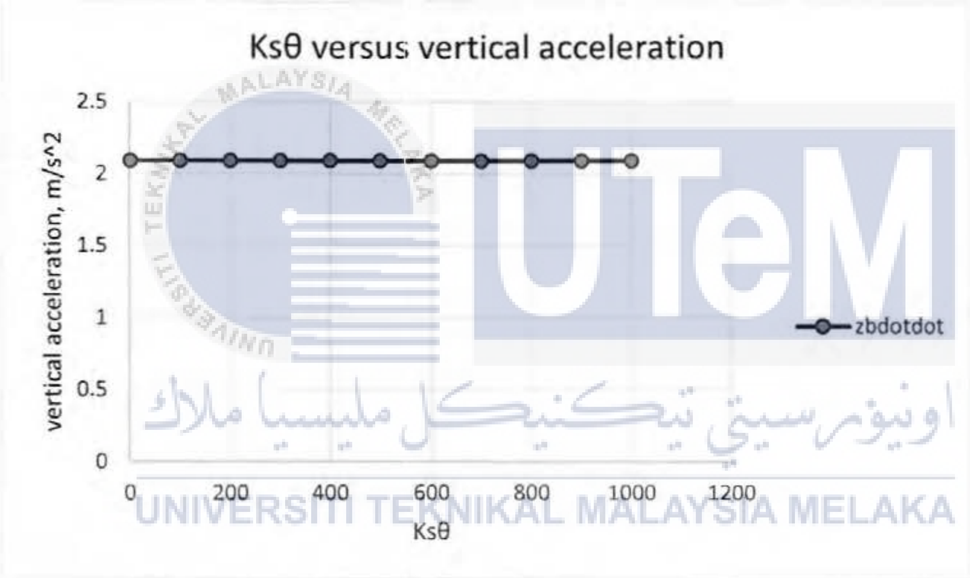

Ksθ versus vertical acceleration
vertical acceleration, m/s^2
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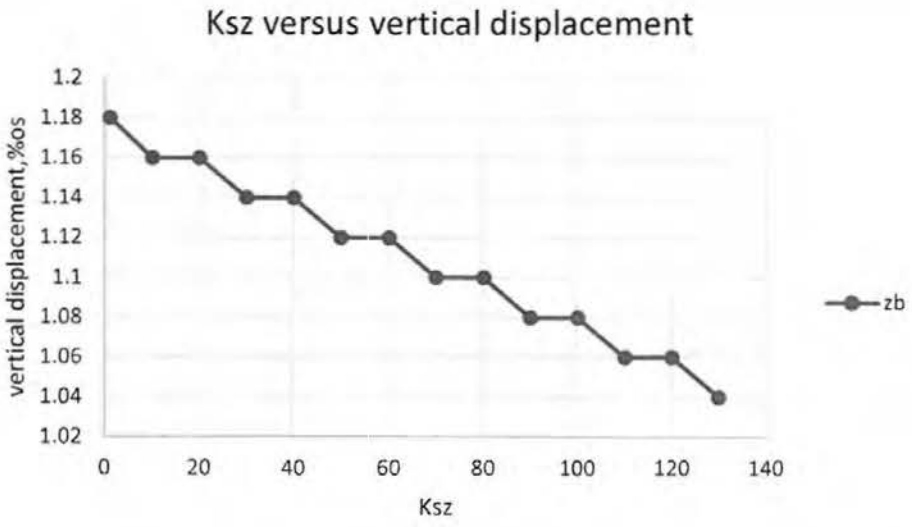
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Ksz
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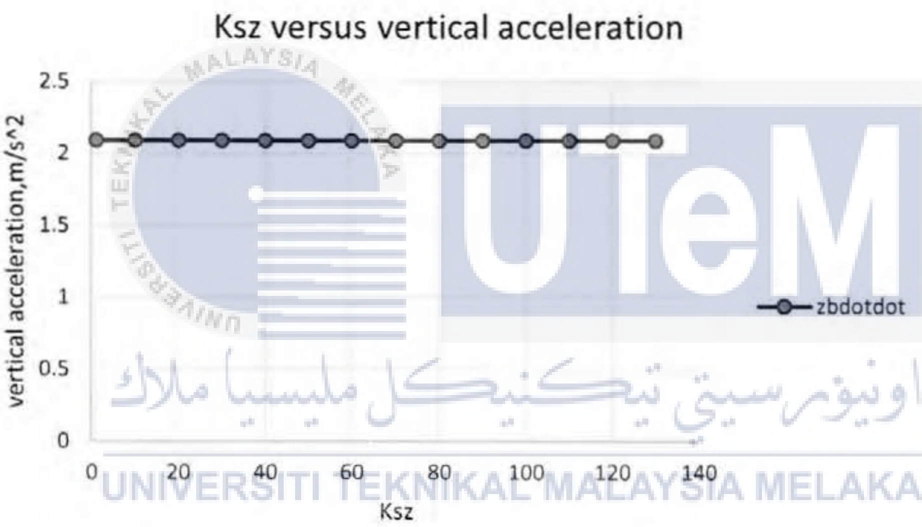
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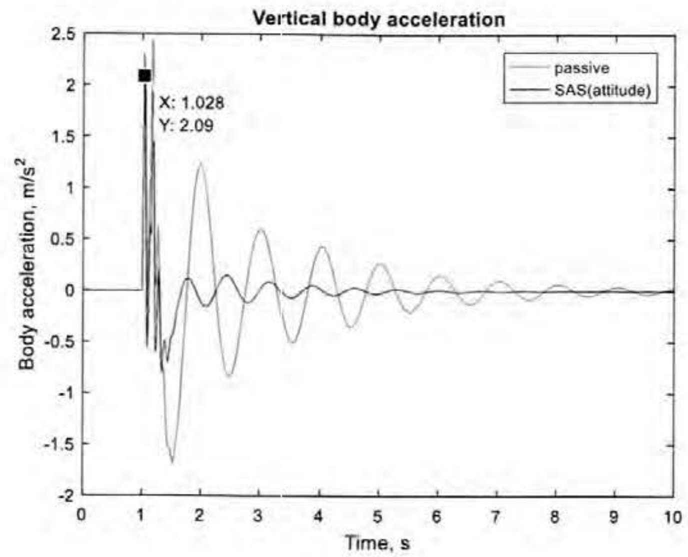
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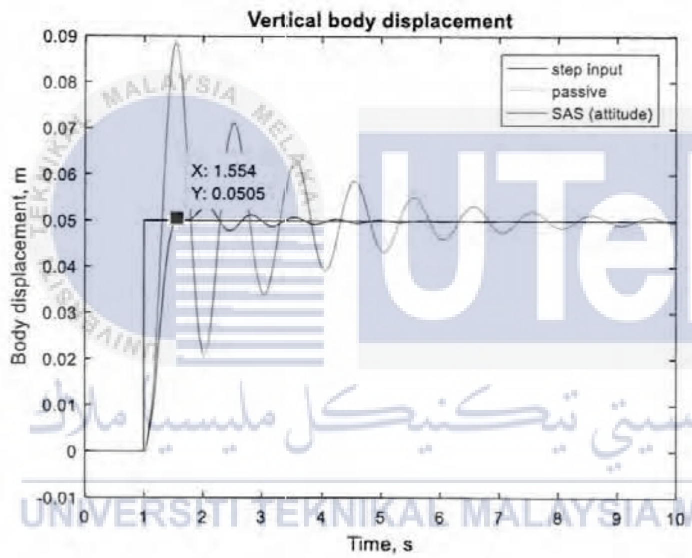
Y: 2.09
Body acceleration, m/s²
Time, s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step input
passive
SAS (attitude)
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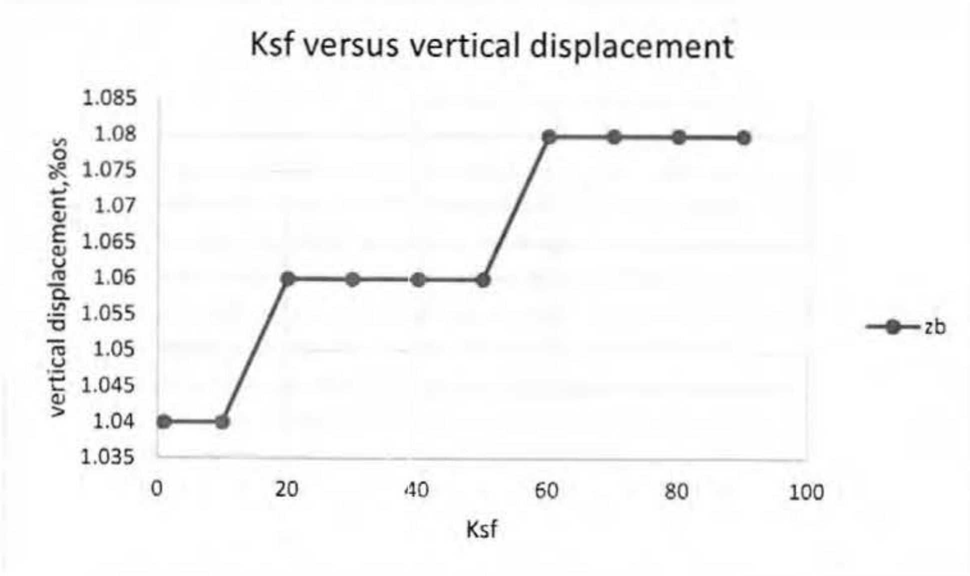
Y: 0.0505
Body displacement, m
Time, s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Ksf
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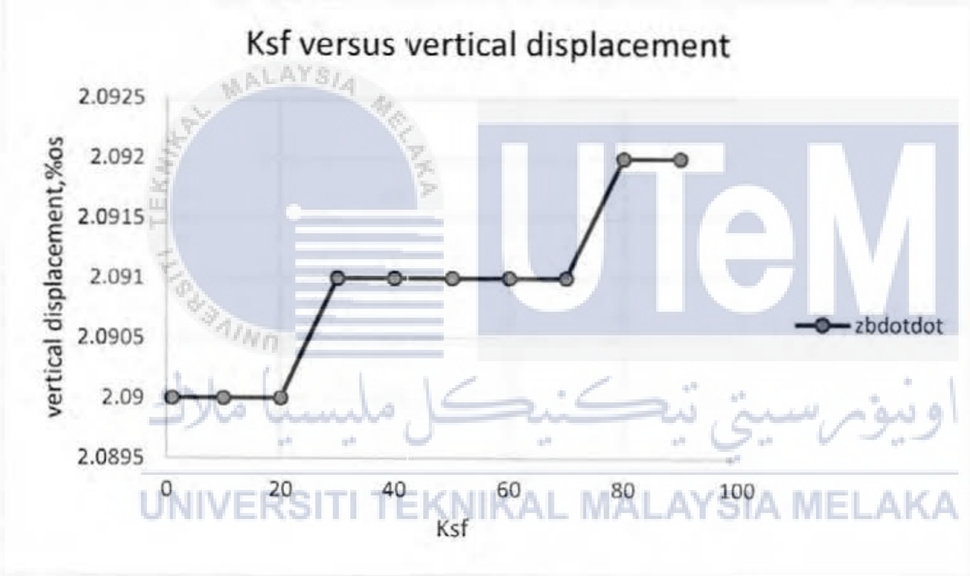
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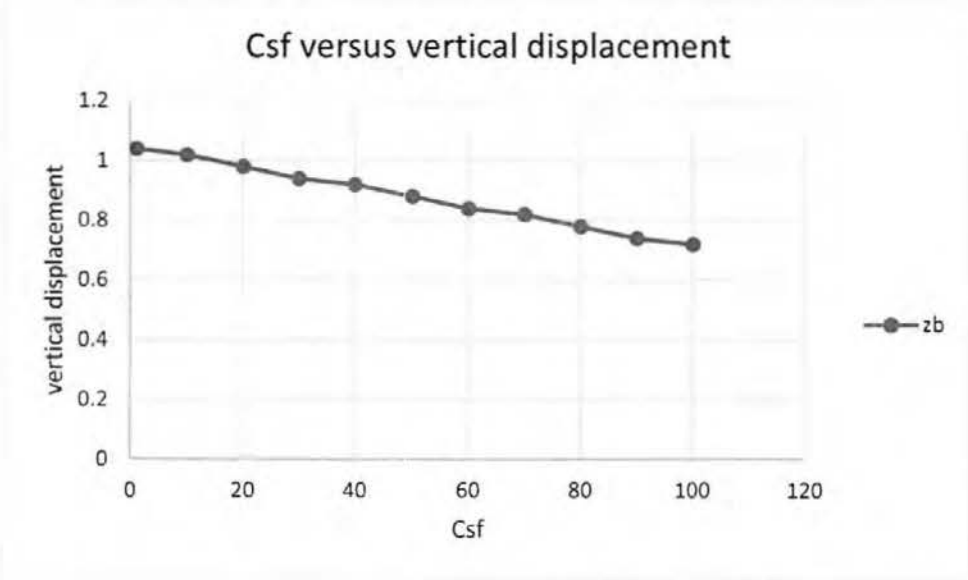
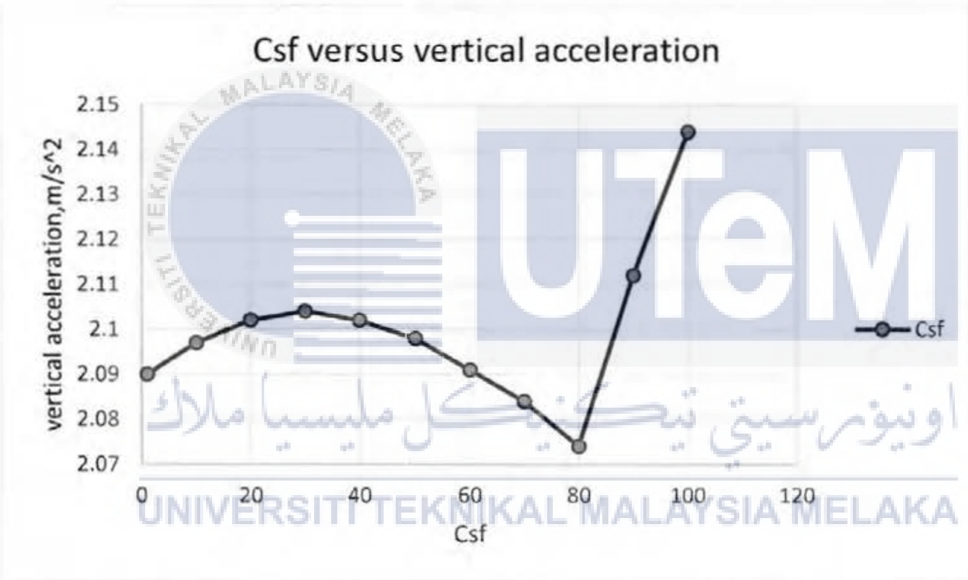
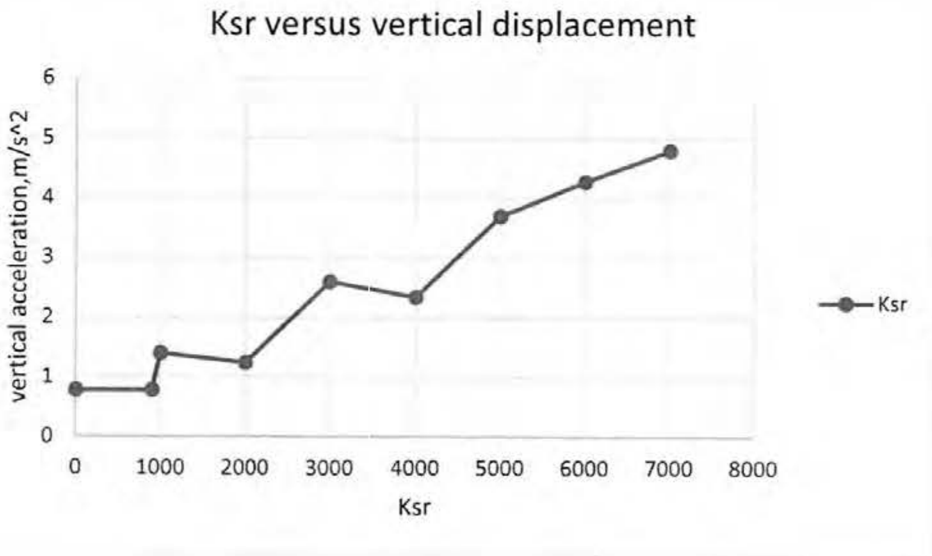
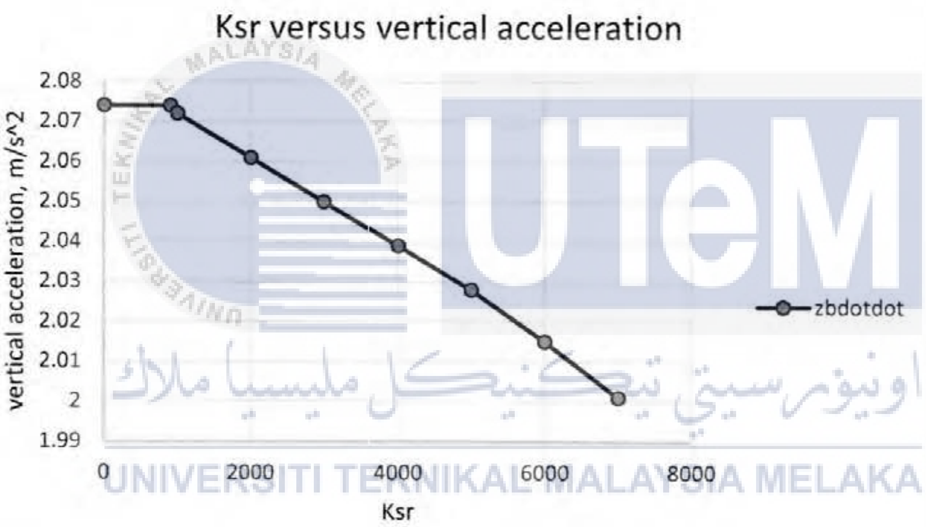
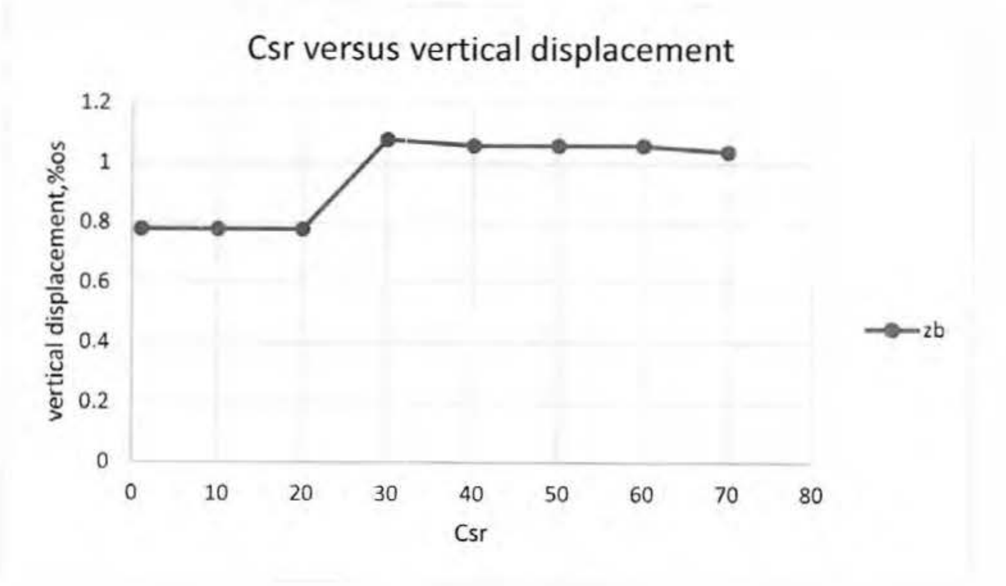
Ksf
zbdotdot

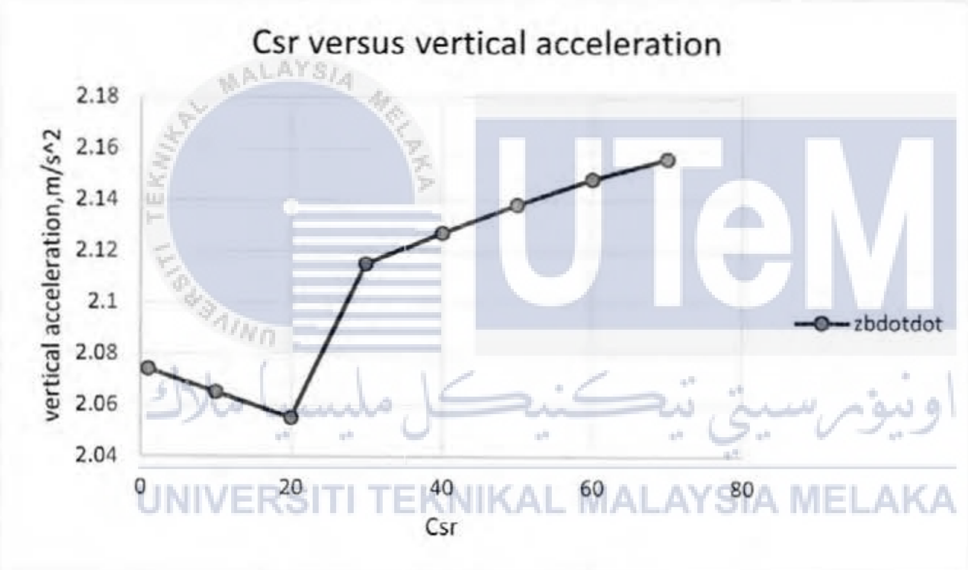

Csf versus vertical displacement
vertical displacement
Csf
zb
1.2
1
0.8
0.6
0.4
0.2
0
0
20
40
60
80
100
120

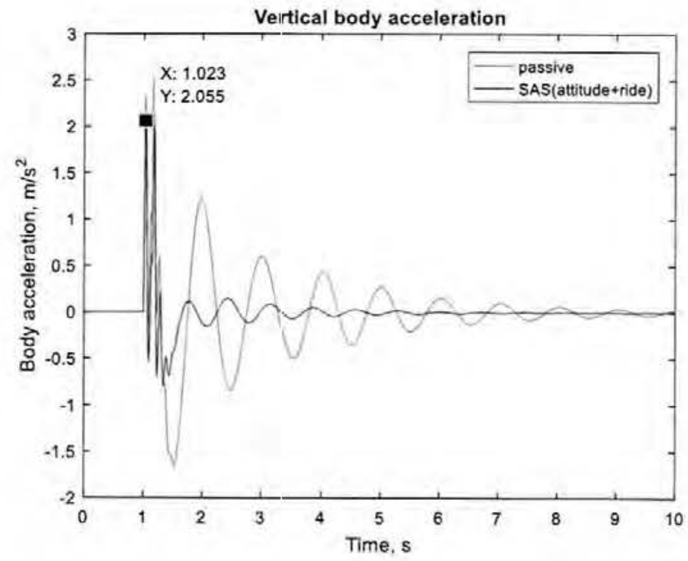

Csf versus vertical acceleration
vertical acceleration,m/s^2
Csf
Csf
2.15
2.14
2.13
2.12
2.11
2.1
2.09
2.08
2.07
0
20
40
60
80
100
120

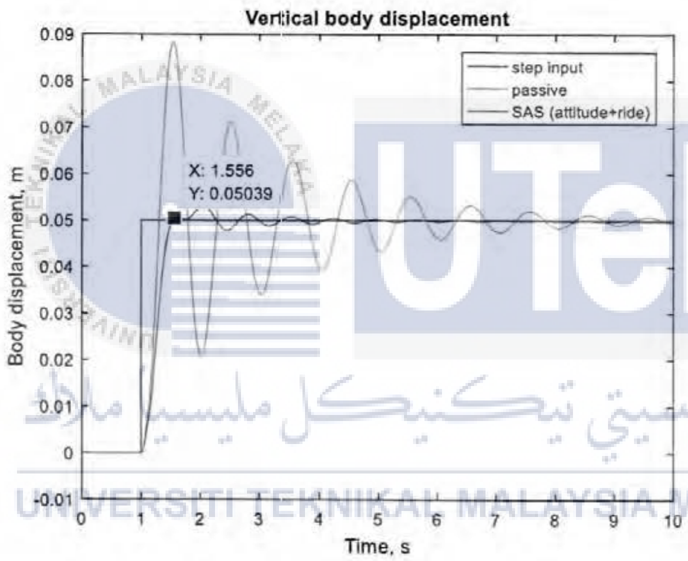

Ksr versus vertical displacement
vertical acceleration,m/s^2
6
5
4
3
2
1
0
0
1000
2000
3000
4000
5000
6000
7000
8000
Ksr
Ksr


Ksr versus vertical acceleration
vertical acceleration, m/s^2
2.08
2.07
2.06
2.05
2.04
2.03
2.02
2.01
2
1.99
0
2000
4000
6000
8000
Ksr
zbdotdot

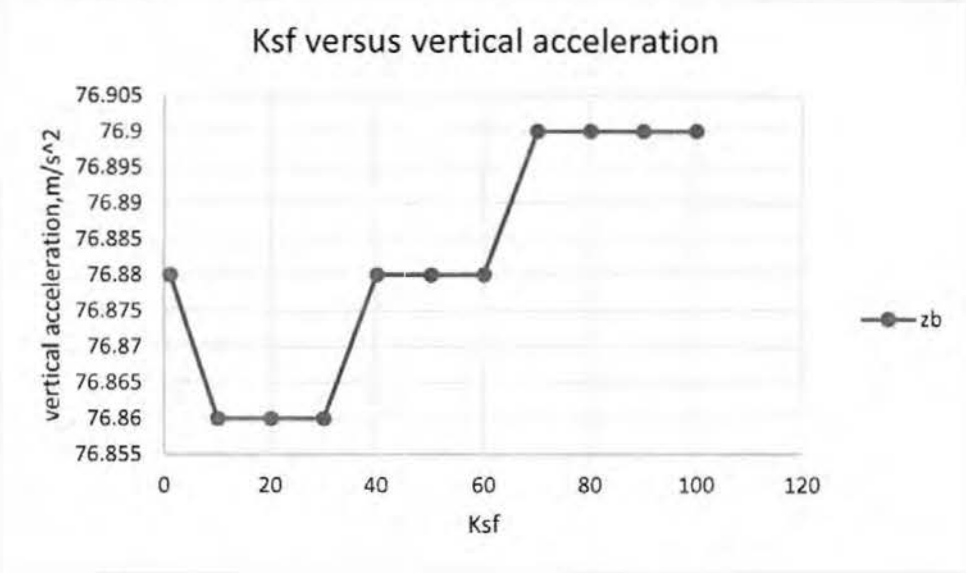

Csr versus vertical displacement
vertical displacement,%os
1.2
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
50
60
70
80
Csr
zb

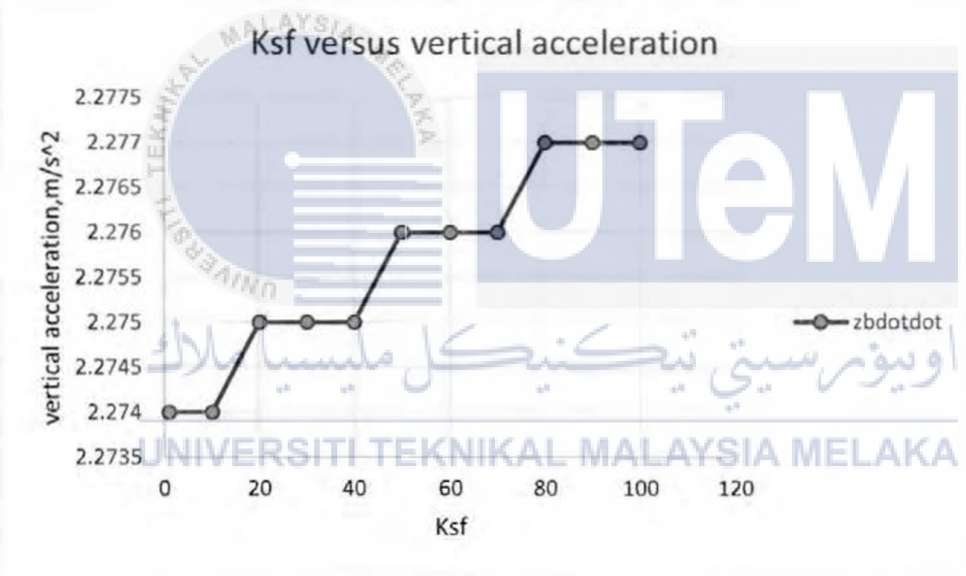

Csr versus vertical acceleration
vertical acceleration,m/s^2
2.18
2.16
2.14
2.12
2.1
2.08
2.06
2.04
0
20
40
60
80
Csr
zbdotdot

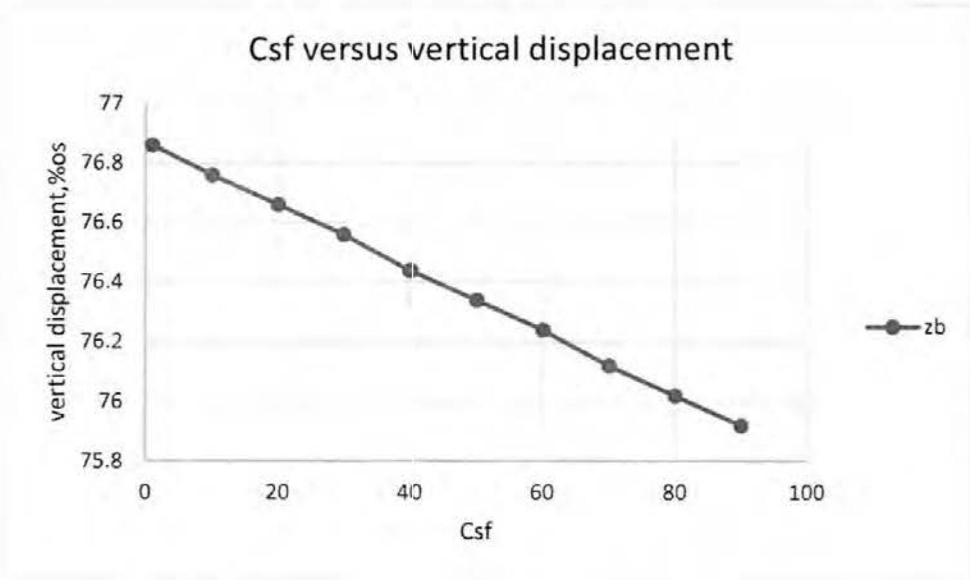

Vertical body acceleration
X: 1.023
Y: 2.055
passive
SAS(attitude+ride)
Body acceleration, m/s^2
Time, s

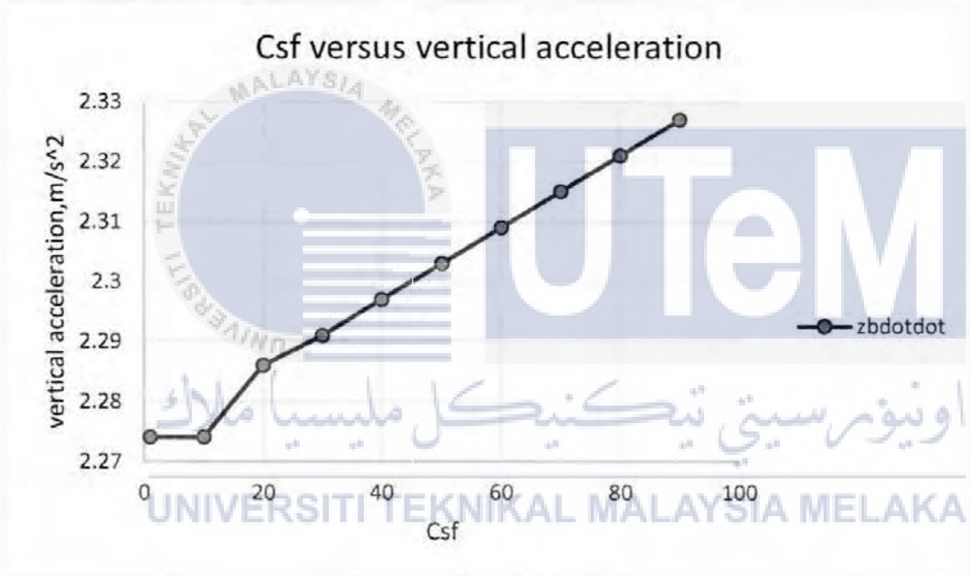

Vertical body displacement
step input
passive
SAS (attitude+ride)
X: 1.556
Y: 0.05039
Body displacement, m
Time, s

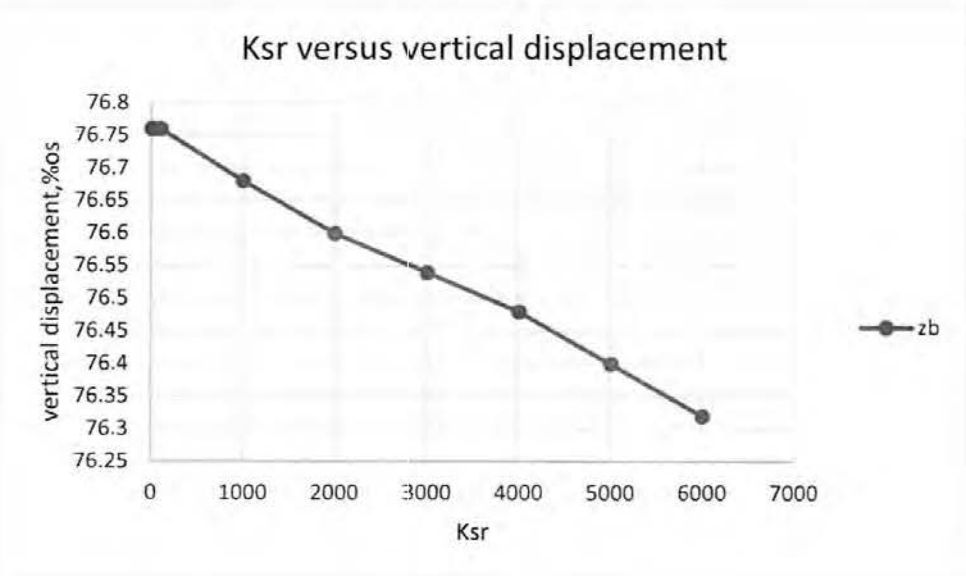
➢ SAS ride controller

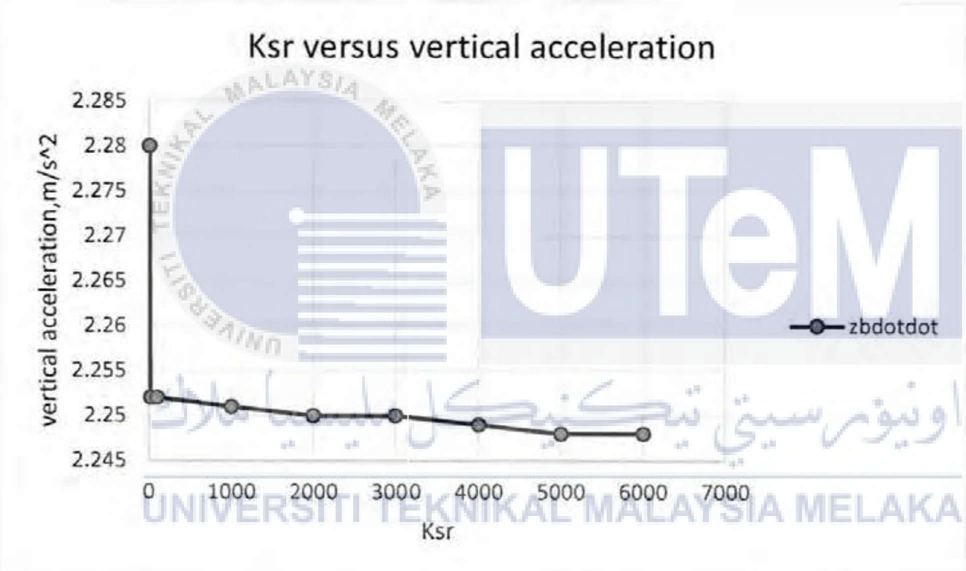


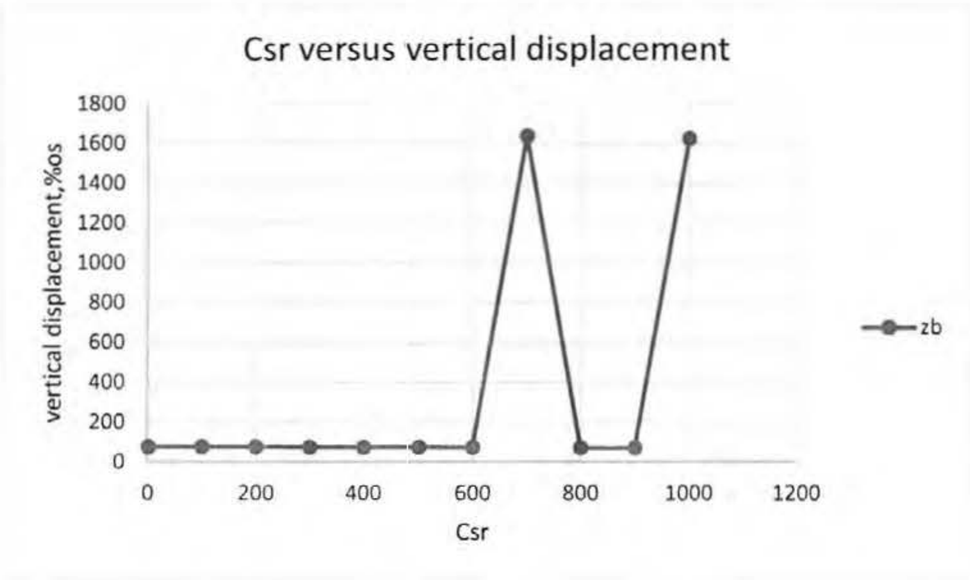


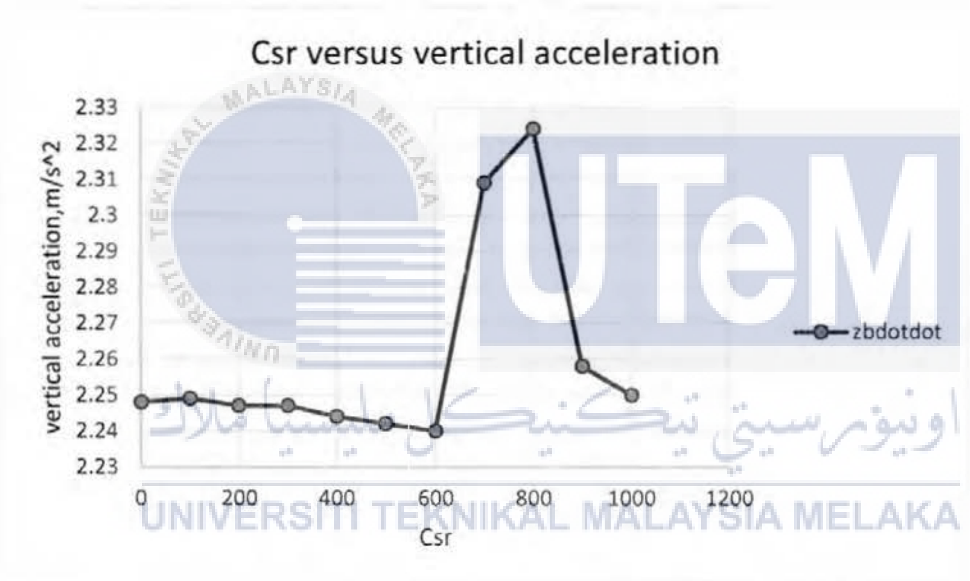

Csf versus vertical displacement
vertical displacement,%os
77
76.8
76.6
76.4
76.2
76
75.8
0
20
40
60
80
100
Csf
zb

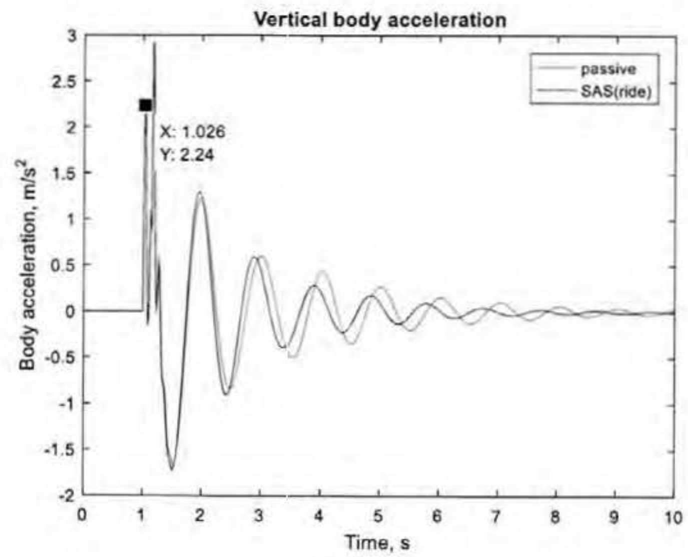

Csf versus vertical acceleration
vertical acceleration,m/s^2
2.33
2.32
2.31
2.3
2.29
2.28
2.27
0
20
40
60
80
100
Csf
zbdotdot

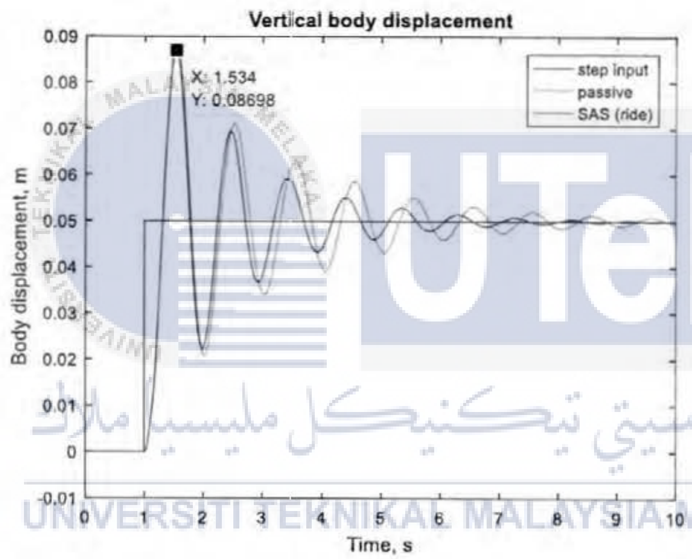

Ksr versus vertical displacement
vertical displacement,%os
76.8
76.75
76.7
76.65
76.6
76.55
76.5
76.45
76.4
76.35
76.3
76.25
0
1000
2000
3000
4000
5000
6000
7000
Ksr
zb


Ksr versus vertical acceleration
vertical acceleration,m/s^2
2.285
2.28
2.275
2.27
2.265
2.26
2.255
2.25
2.245
0
1000
2000
3000
4000
5000
6000
7000
Ksr
zbdotdot


Csr versus vertical displacement
vertical displacement,%os
1800
1600
1400
1200
1000
800
600
400
200
0
0
200
400
600
800
1000
1200
Csr
zb


Csr versus vertical acceleration
vertical acceleration,m/s^2
2.33
2.32
2.31
2.3
2.29
2.28
2.27
2.26
2.25
2.24
2.23
0
200
400
600
800
1000
1200
Csr
zbdotdot


Vertical body acceleration
passive
SAS(ride)
X: 1.026
Y: 2.24
Body acceleration, m/s^2
Time, s


Vertical body displacement
step input
passive
SAS (ride)
X: 1.534
Y: 0.08698
Body displacement, m
Time, s

B Sample calculation for transport delay

$Velocity, v = \frac{distance}{time}$

Then, take distance for the vehicle wheel base, which is equal to $3.1m$. Convert the vehicle speed to m/s, equal to $11.11m/s$.

$11.11m/s = \frac{3.1m}{time}$

$time = 0.279s$

By using transport delay block, set the time delay for rear tire equal to $0.279s$.