UNIVERSITI TEKNIKAL MALAYSIA MELAKA

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IMPROVEMENT OF FRONT WHEEL STEERING SYSTEM DUE TO SIDE WIND

This report is submitted in accordance with the requirement of the Universiti Teknikal Malaysia Melaka (UTeM) for the Bachelor of Mechanical Engineering Technology (Automotive Technology) with Honours

By

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DECLARATION

I hereby, declare that this thesis entitled “Improvement of front wheel steering system due to side wind” is the result of my own research except as cited in references.

Signature : ……………………………………………

Name : ……………………………………………

Date : ……………………………………………

APPROVAL

This report is submitted to the Faculty of Engineering Technology of UTeM as one of the requirements for the award of Bachelor's Degree Mechanical Engineering Technology (Automotive Technology) with Honours. The following are the members of supervisory committee:

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(Supervisor)

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(Co-Supervisor)

ABSTRAK

Satu sistem bantuan pemandu yang dikenali sebagai stereng aktif dinilai semasa menyimulasikan pelbagai jenis acara memandu. Sistem ini merupakan sistem yang penting di dalam kenderaan itu. Tujuan utama sistem stereng adalah untuk membolehkan pemandu mengawal kenderaan secara bebas. Sistem stereng mengekalkan tayar dengan kenalan jalan. Model kenderaan kereta disimulasi oleh MATLAB / Simulink dengan 14 DOF termasuk model memandu kenderaan dan juga model pengendalian.

Sudut stereng adalah input model kenderaan yang harus diberi perhatian. Sudut sistem stereng antara tayar apabila beralih kenderaan telah difokuskan. Simulasi dibuat pada keadaan jalan yang berbeza dan juga kesan gangguan angin sampingan. Nilai yang berbeza digunakan untuk simulasi untuk mengurangkan kesan peristiwa memandu. Oleh itu, simulasi ini meningkatkan prestasi untuk sistem stereng ini.

Tujuan untuk kerja ini adalah untuk memahami dan meningkatkan respon kenderaan dalam sistem stereng depan. Peristiwa memandu simulasi ini adalah perekatan jalan dan sisi gangguan angin sampingan. Semasa pemprosesan simulasi tersebut, bertambah baik stereng hadapan dapat diperolehi oleh kenderaan ini.

ABSTRACT

A steering aid system called active steering is evaluated by simulating different kinds of driving events. This system is an important system in the vehicle. The main purpose of the steering system is to allow the driver control the vehicle independently. The steering system maintains the tire to the road contact. A full car vehicle model is simulated in MATLAB/Simulink with 14 DOF of equations which include the drive vehicle model and also the handling model.

The steering angle is the input of the vehicle model that should be focused on. The angle of the steering system between the tires when turning the vehicle is taken in consideration. Simulations are made on different road conditions effect and also side wind disturbances. Different values are applied to the simulation to reduce the effect of the driving events. Therefore, these simulations results do provide a better improvement to the steering system.

The aim for this work is to understand and improve the response of the vehicle in the front steering system. Specific driving events in these simulations are the road adhesions and lateral side wind disturbances. Improved front steering is obtained for vehicle during the processing of the simulations.

DEDICATION

To my beloved parents

Tan Chin Hwee and Teoh Giek Chew

Raise me to become who I am

ACKNOWLEDGEMENT

I would like to express my gratitude to my supervisor, En. Hafiz bin Harun who guides me through this project. He gave me a lot of advice, ideas and confidence to complete this project. Besides, I would like to thank Faculty of Engineering Technology of University Teknikal Malaysia Melaka (UTeM) for helping me in preparing formal documentations and guidelines for my project report. Not to forget, my families for their support and blessing.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF ABBREVATIONS, SYMBOLS AND NOMENCLATURES

AFS	-	Active Front Steering
CG	-	Centre Gravity
DOF	-	Degree of Freedom
HSRI	-	Human Services Research Institute
ISO	-	International Organization for Standardization
MATLAB	-	Matrix Laboratory
RPM	-	Rotation per Minute
RAM	-	Random Access Memory
ROM	-	Read-only Memory
SBW	-	Steer-by-Wire
VSC	-	Vehicle Stability Control

F_{fl}	Suspension force at the front left corner	$Z_{u,fl}$	Front left unsprung masses displacement
F_{fr}	Suspension force at the front right corner	$Z_{u,rr}$	Rear right unsprung masses displacement
F_{rl}	Suspension force at the rear left corner	$Z_{u,rl}$	Rear left unsprung masses displacement
F_{rr}	Suspension force at the rear right corner	$\dot{Z}_{u,fr}$	Front right unsprung masses velocity
M_s	Sprung mass weight	$\dot{Z}_{u,fl}$	Front left unsprung masses velociy
$\ddot{Z}$	Sprung mass acceleration at the body centre of gravity	$\dot{Z}_{u,rr}$	Rear right unsprung masses velocity
F_{pfl}; F_{pfr}; F_{prl}; F_{prr}	Pneumatic actuator forces at front left, front right, rear right corners, respectively.	$\dot{Z}_{u,rl}$	Rear left unsprung masses velocity

$K_{s,fl}$	Front left suspension spring stiffness	a	Distance between front of vehile and C.G. of sprung mass
$K_{s,fr}$	Front right suspension spring stiffness	b	Distance between rear of vehicle and C.G. of sprung mass
$K_{s,rr}$	Rear right suspension spring stiffness	θ	Pitch angle at body centre of gravity
$K_{s,rl}$	Rear left suspension spring stiffness	φ	Roll angle at body centre of gravity
$C_{s,fr}$	Front right suspension damping	$Z_{s,fl}$	Front left sprung mass displacement
$C_{s,fl}$	Front left suspension damping	$Z_{s,fr}$	Front right sprung mass displacement
$C_{s,rr}$	Rear right suspension damping	$Z_{s,rl}$	Rear left sprung mass displacement
$C_{s,rl}$	Rear left suspension damping	$Z_{s,rl}$	Rear left sprung mass displacement
$Z_{u,fr}$	Front right unsprung masses displacement	$Z_{s,rr}$	Rear right sprung mass displacement
$\dot{\theta}$	Pitch rate at body centre of gravity	$\ddot{Z}_{u,fr}$	Front right unsprung masses acceleration
Z_s	Sprung mass displacement at body centre of gravity	$\ddot{Z}_{u,fl}$	Front left unsprung masses acceleration
$\dot{Z}_s$	Sprung mass velocity at body centre of gravity	$\ddot{Z}_{u,rr}$	Rear right unsprung masses acceleration
$K_{s,f}$ $(K_{s,fl} = K_{s,fr})$	Spring stiffness of front suspension	$\ddot{Z}_{u,rl}$	Rear left unsprung masses acceleration
$K_{s,r}$ $(K_{s,rl} = K_{s,rr})$	Spring stiffness of rear suspension	$Z_{r,rl} = Z_{r,fl} = Z_{r,rr} = Z_{r,rl}$	Road profiles at front left, front right, rear right and rear left tires respectively
$C_{s,f} = C_{s,fl} = C_{s,fr})$	Damping constant of front suspension	δ	Steering angle

$C_{s,r} = C_{s,rl} = C_{s,rr}$	Damping constant of rear suspension	r	Yaw rate
$\ddot{\theta}$	Pitch acceleration at body centre of gravity	m_t	Total vehicle mass
$\ddot{\varphi}$	Roll acceleration at body centre of gravity	v_x and v_y	Longitudinal and lateral vehicle velocities
I_{xx}	Roll axis moment of inertia	α_f	Front side slip angle
I_{yy}	Pitch axis moment of inertia	α_r	Rear side slip angle
w	Wheel base of sprung mass	l_f	Distance between the front tire to the body centre
ω_r	Angular velocities of the rear tires	l_r	Distance between the rear tire to the body centre
ω_f	Angular velocities of the front tires	r_w	Wheel radius
J_z	The moment of inertia around the z-axis	c	The height of the sprung mass centre of gravity to the ground
j_{sx}	The moments of inertia of the sprung mass around the x-axes	j_{sy}	The moment of inertia of the sprung mass around the y-axes

CHAPTER 1

INTRODUCTION

1.1 Background

In an age of advanced technology, automobile was a life's necessities for everyone. It also acts as a transportation to travel for a distance no matter a short or long journey. Therefore, automobile had become more and more convenience nowadays. In this competitive field, vehicle must be improved in comfort and safety. By doing so, development of vehicles should be done to ensure new technology is implanted into the driving system to enhance driver pleasure.

A vehicle consists of many systems to maintain each of the vehicle parts function well. The steering system acts as one of the important system in the vehicle. For commonly known, steering system used to drive the vehicle to different directions which control by the driver alone. Then, the steering system also provided the driver a road feel feedback when the vehicle drove through an uneven surface or facing the wind disturbances. The main purpose of the steering system is to make contact between the tire and the road surfaces. Therefore, concentrations were mainly focus on the tire life of the vehicle to achieve better angle of the steering system. Furthermore, the driver can turned their vehicle with minimum effort by the help of the steering system which is not so easy to control.

In steering arrangement, the most comfortable position of the hand-operated steering wheel is placed in front of the driver with the steering column that contained universal joints. This was to allow the vehicle to deviate from a straight line. Different kinds of vehicle consist of different kinds of arrangement.

1.2 Problem Statement

The problem in this project is the side wind disturbances. This problem could contribute to vehicle stability. Therefore, this project focused on the improvement of the front steering system. This study wills focused and highlighted these problems that become challenges to complete the task. Therefore, simulations between the Matlab model and Carsim actual model which both consists of 14 DOF equations are made to improve the front steering system.

1.3 Objectives

The main objective of this project is to focus on the design and development of mathematical equations, apply them to the MATLAB Simulink and make some modification on controller design.

1.4 Scope of the study

Scopes of the study are listed as follows:

i) Identifying the effects of the vehicle riding.
ii) Developed mathematical equation with 14 Degree Of Freedom vehicle model.
iii) Produce a Simulink diagram by using the MATLAB software and compare with Carsim.
iv) Applied active front steering system in the front system and compare to conventional system.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

For further process of this research project, literature review made an important key of it. Literature survey in various fields, such as active steering system, steering control, handling, vehicle modelling and parameters estimation is carried out to establish the state of the art.

2.2 Nonlinear Dynamics

First, the dynamic behaviour of a nonlinear seven degrees-of-freedom ground vehicle model is examined. The nonlinearity occurs due to suspension dampers and springs. The disturbances from the road are assumed to be sinusoid and the time delay between the disturbances is also considered. Numerical results show that the responses of the vehicle model could be chaotic. Using bifurcation phenomenon, the chaotic motion is detected and confirmed with the Poincare' maps. The results can be applicable in dynamic design of a vehicle. [Hajikarami et al, 2009]

As we know, vehicle dynamics is concerned with controllability and stability of an automobile. Therefore, it was suitable to be design as a ground vehicle. There were a lot of mathematical models and equations for vehicle and the dynamic response of them has been examined in large number of previous investigation. In general, these models can be classified into three types which are the quarter car model, half car model and also full car model.

Because of high nonlinearities of differential equations in the full car model, the dynamic response of the system is studied numerically using the forth order Runge-Kuntta algorithm provided by MATLAB.

The frequency responses diagrams are obtained by plotting the amplitude of the oscillating system versus the frequency of the excitation. This are used to analyse the dynamics of a system. Bifurcation is the phenomenon of sudden change in the motion as a parameter is varied. Therefore, the parameter values at which they occur are called bifurcation points. Poincare' maps were one of the instruments to analyse the dynamic system.

In the interval of 4.96< f <5.47 Hz, there are two unstable regions caused by the increasing in frequency, f=3.08Hz as a huge jump and a small jump which was f=3.18Hz is observed from the figure 2.1. f= 4.76 Hz and f= 3.07 Hz is observed for the decreasing frequency. Besides, 4.99< f <5.41 Hz and 2.97< f < 3.55 Hz was the unstable region.

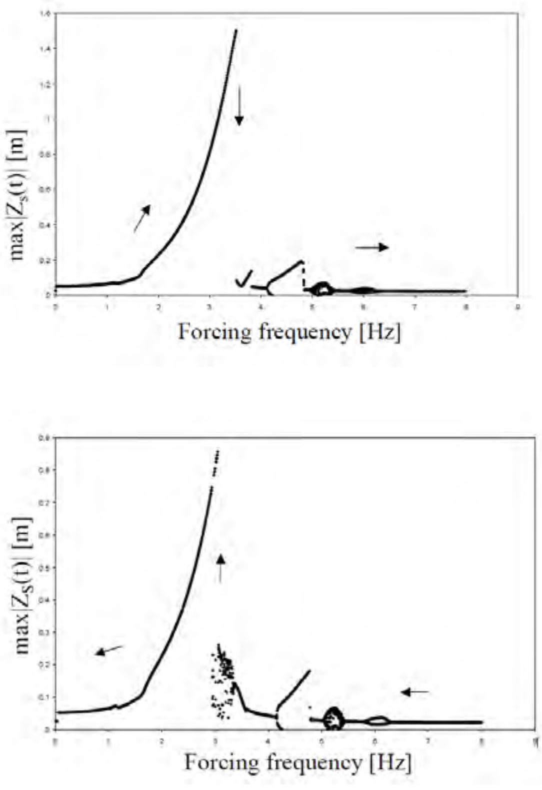

Figure 2.1: Frequency Response Diagrams when the Forcing Frequency f is slowly Increased and Decreased

As conclusion from the frequency response diagrams, the number of jumps and unstable regions of the response were focused. The result of increasing frequency is different from those for decreasing frequency. This was an ordinary phenomenon for nonlinear problems but the phenomenon was not observed in the study of two or four DOF vehicle model.

2.3 Dynamic Model & Tire Model

Most of reference models for chassis controls usually have low level degree of freedom like a bicycle model. However, these models include some different value in real vehicle motion and have a difficulty to adapt new technology. In addition, it is not good for real time that very high degree of freedom like multi-body dynamic analysis programs because of their long solving time. So, I developed adaptive full vehicle dynamic model that has 14 degree of freedom with theoretical equations and experimental data. [Lee et al, 2008]

This paper studies the technology for a thesis of a vehicle dynamics model based on a full car model with 14 DOF (Degree of Freedom). MATLAB was done on the 14 DOF full vehicle models which consist of horizontal direction and vertical direction. 14 DOF model consider of whole direction of the full car vehicle model. The time used in this software was less compared to commercial program. Therefore, a more accurate result can be obtained.

The degree of freedom for a full vehicle model consists of 3 horizontal, 7 vertical, and 4 tire model. The procedure and steps were arranged in the modular architecture of vehicle model. First, the tire model (wheel rotations) has the longitudinal force at each wheel which relate with the vertical dynamic (vertical, pitching, rolling motions) and horizontal dynamic (longitudinal/lateral, yawing, steering motions). The driver model had a steering wheel angle and acceleration and deceleration torque to the tire model. With this modular vehicle model, clearer procedure and process can be followed.

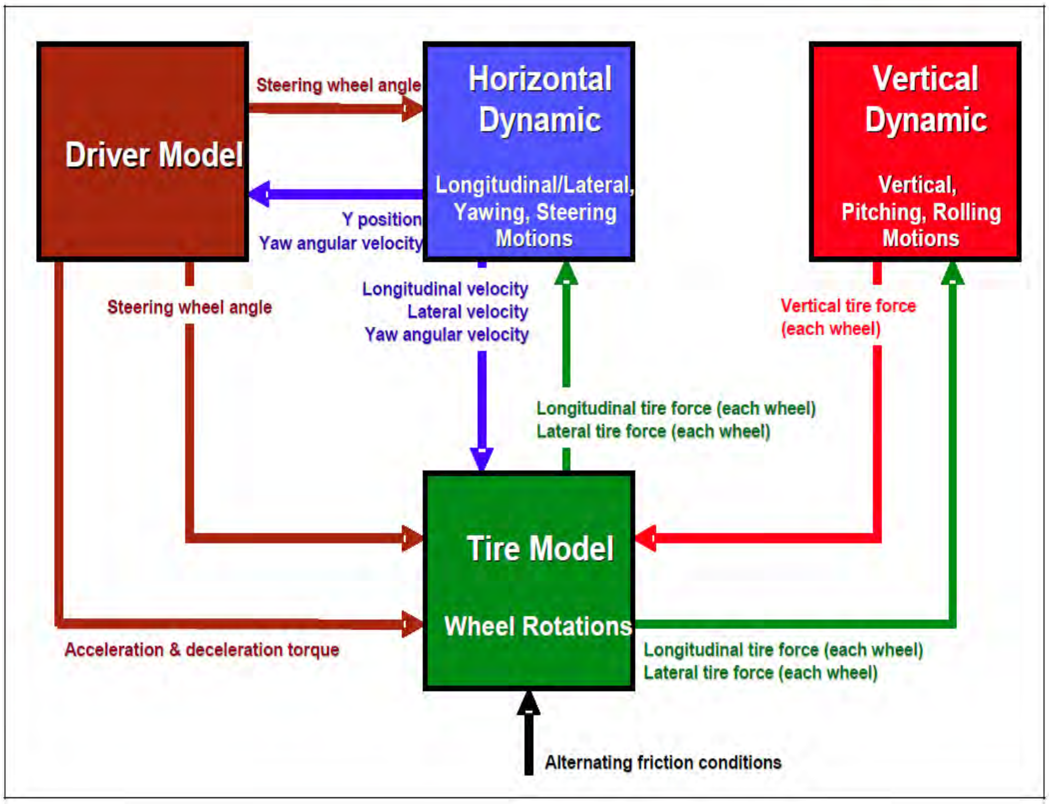

Figure 2.2: Modular Architecture of Vehicle Model

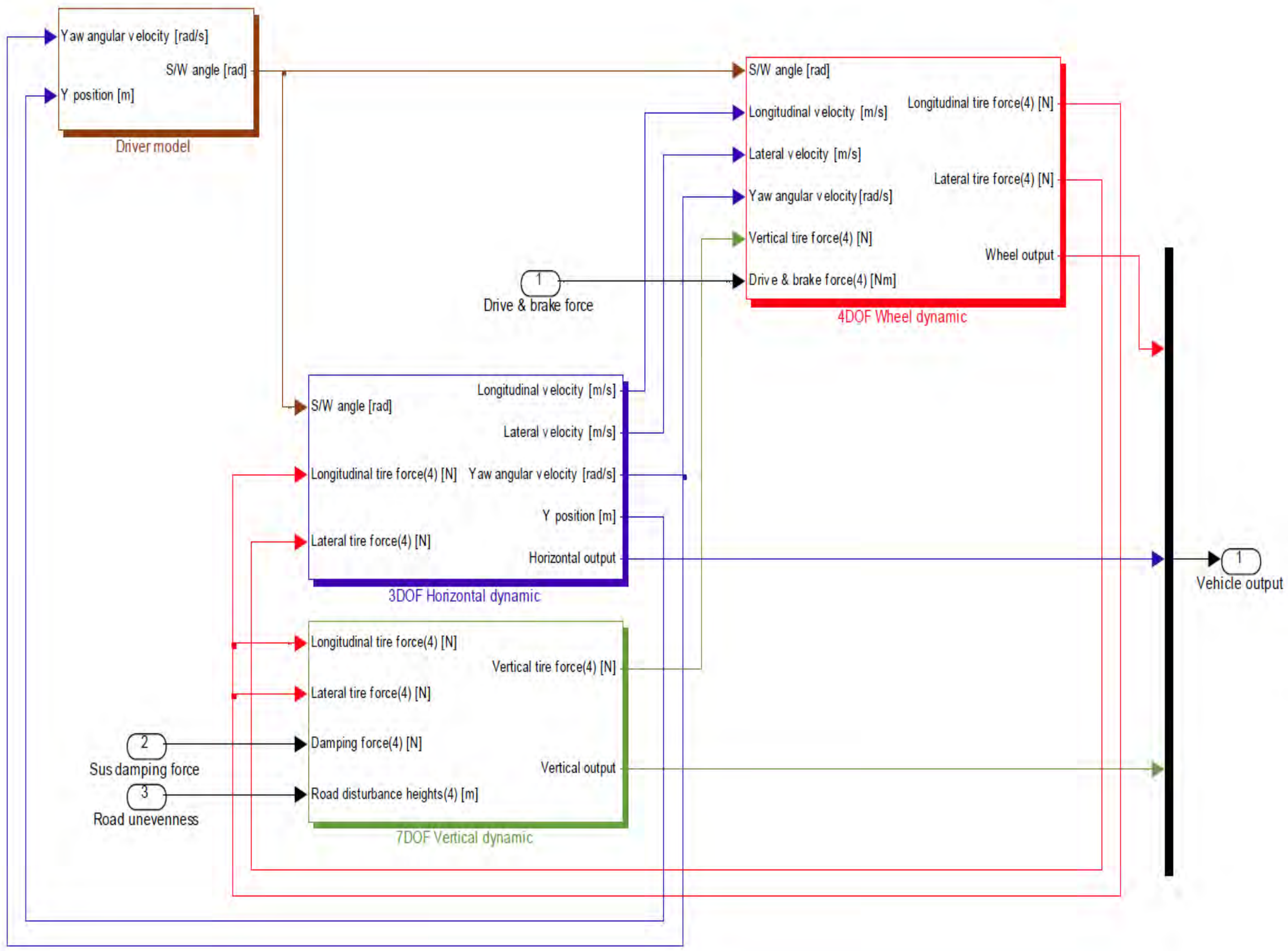

Figure 2.3: Modular Vehicle Model in MATLAB/Simulink

As conclusion, the result of 14 DOF model shows the accuracy of lateral acceleration and yaw rate in single lane change. This model has difference of 17% error in roll angle. Therefore, an accuracy of 85-98% is obtained according to the results. From this simulation, one of the errors that were not reflected on the vehicle model is the road conditions. By considering effect of suspension's kinematics and compliance, a more accurate model can be achieved with a better road conditions.

2.4 Suspension System in MATLAB

Suspension system design is a challenging task for the automobile designers in view of multiple control parameters, complex objectives and stochastic disturbances. For vehicle, it is always challenging to maintain simultaneously a high standard of ride comfort, vehicle handling under all driving conditions. The objective of this paper is to develop a MATLAB/SIMULINK model of a full car to analyse the ride comfort and matrix are to be developed and validation of Simulink model with analytical solution of state space matrix is to be done elaborately on this paper. [Mitra et al, 2008]

In this research, mathematical modelling was taken in measure using seven degree of freedom model of the full car for passive system. There were mathematical equation formed which include sprung mass and also unsprung mass of a full model vehicle. Next, the equation was tested with MATLAB/SIMULINK to obtain the result from the graphs. From the result obtained, vehicle body acceleration was high at velocity of 10kmph. Therefore, the effect of bump of same amplitude had no effect on pitch angle and roll angle of the vehicle.

The fixed parameters mentioned in various research papers are taken for the simulation study. Suspension spring stiffness were considering 55000 N/m, 25000 N/m and damping coefficient 4000 N-s/m, 1000 N-s/m respectively. The combination of stiffness and damping coefficient were studied in simulation.

Table 2.1: Fixed Parameters of Full Car Model

M_s = 1200Kg	$K_{wr1} = K_{wl1}$ = 30000 N/m	a = b = 1.5 m
$M_{wr1} = M_{wl1}$ = 60Kg	$K_{wr2} = K_{wl2}$ = 30000 N/m	C = d = 1m
$M_{wr2} = M_{wl2}$ = 60Kg	I_{xx} = 4000 Kg-m^2	I_{yy} = 950 Kg-m^2

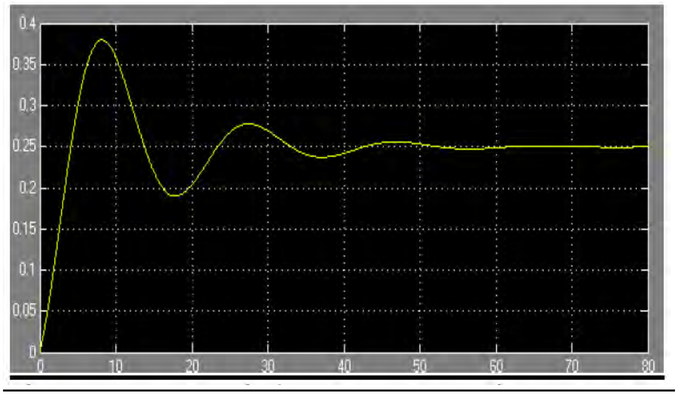

Figure 2.4: Sprung Mass Displacement (Z_{CG}) vs. Time in Simulink Model

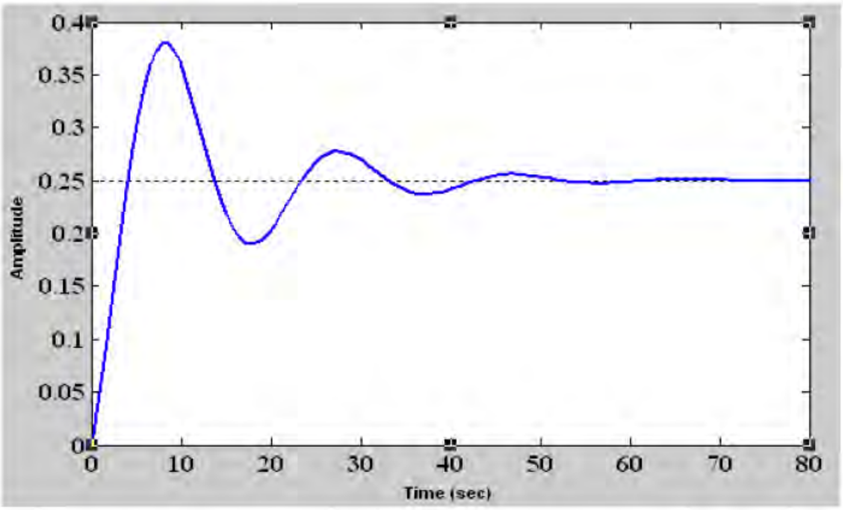

Figure 2.5: Sprung Mass Displacement (Z_{CG}) vs. Time using Analytical solutions (State Space)

This validated simulation model was ride comfort and road holding for different standard road profiles. As results, the speed range of 5 to 10kmph must be an optimum speed to cross the bump without affecting the Human tolerance zone of 0.315 m/s^2 to 0.625 m/s^2 as per ISO standard. There was no effect on pitch angle and roll angle when crossing the bump. The damping coefficient as 25000 N/m and 4000 N-s/m in the result of stiffness may provide better comfort to the driver. MATLAB simulation was done as shown on both of the graph above. The maximum amplitude for both graph is approximately 0.38.