Simulation and Modeling of Front Suspension System of Proton Gen-2 Model Car

By

Muhammad Bin Hasanudin

Dissertation submitted in partial fulfillment of

The requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

JUNE 2009

Universiti Teknologi PETRONAS

Bandar Seri Iskandar

31750 Tronoh

Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

Simulation and Modeling of Front Suspension System of Proton Gen-2 Model Car

by

Muhammad Bin Hasanudin

A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

Approved by,

____

(DR. VU TRIEU MINH)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
June 2009

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

____
MUHAMMAD BIN HASANUDIN

ABSTRACT

The main goal of this project entitled 'Simulation and Modeling of Front Suspension System of Proton Gen-2 Model Car' is to carry out a feasible study on the independent McPherson front suspension system installed to Proton Gen-2 Model car. The project starts with the thorough analysis on the system. Further analytical and literature studies will be conducted based on the simulation. Data gathered from the study will then being used to design a newly improved front suspension system. A 3D model of the enhanced model will be evaluated by the application of related computer aided engineering software. Finally, a feasibility study of the new system will be performed and data gathered for further revision. The focus of this project is to suggest the improvement on the existing front suspension system, with emphasize on the ride and handling quality being obtained, without jeopardizing safety and standard road regulations of the country.

ACKNOWLEDGEMENT

First and foremost, all praises and thanks to the almighty God, the source of my life and hope for giving me the wisdom to succeed in life.

I would like to thank my supervisor, Dr. Vu Trieu Minh for his great support and commitment in ensuring the smooth flowing process of this project. All advices and guidance given were highly appreciated.

I would also like to thank Ir. Dr. Masri Baharom for his supportive guidance on the application of ADAMS/Car software, and not least all lecturers and technicians who involved in completing this project.

This project shall not be completed without the endless support of my family. With that I would like to thank my parents, Mr. Hasanudin bin Abdul Rashid and Ms. Zainab bte Samat for their continuous love and support in term of moral and financial.

Lastly, I would like to express my gratitude to all who involved directly or indirectly prior to completing this project. Thank you very much.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND OF PROJECT

Suspension system is one of the crucial aspects of automotive vehicles. Many design of it which can suitably fit into a car, depending on various features it offers and so on. Conventional system using leaf spring has long being implemented. Until recently, automotive vehicles have been fitted with suspension systems consisting of a passive spring and damper in parallel. The desired suspension system performance was then achieved by suitable choice of stiffness and damping rates.

This inevitably leads to a difficult compromise between the conflicting requirements of ride quality and good handling. Current passive vehicles achieve very good ride and handling performance, but the best is always restricted by this compromise. The demand for a more comfort suspension system without jeopardizing the ride and handling qualities is high. Now with the availability of variable rate dampers and springs, there is an opportunity to improve further on the performance of such passive suspension systems, in terms of comfort ability and reliability wise.

Optimizing the design aspects of each configuration is a more complex problem. Suspension design is a compromise that requires evaluations of all the factors. In the 1930's, production cars started to use independent front suspension, but it wasn't until the 1960's that it was developed to provide good cornering power.

McPherson strut type of front suspension system became popular on production cars in the 70's because it offered a simple and inexpensive configuration that doesn't take up much space. It had since become the most widely used front suspension system. This is down to its flexibility, whereby its compact design requires only small amount of space for installation. It is particularly well-suited to front wheel drive production cars; one example is Proton Gen-2 model, because it allows room for the front drive-axle to pass through the front hub. Apart from that, most of the small to mid-size cars use this kind type of front suspension, because it is more economical compared to the others and gives a fairly good ride quality with the compact dimensions needed for front wheel drive cars.

Along with the system there comes a need to simulate and to model the suspension development techniques. With these more sophisticated suspensions, tuning the system on the test track is not feasible. Therefore, there is a requirement for modeling technique that allow complete non-linear simulation of vehicle behavior under various road and driving conditions, as well as the capability for controlling of the design and analysis. Thus the project 'Simulation and Modeling of Front Suspension System of Proton Gen-2 Model Car' is trying to determine possible factors that affect the performance of McPherson suspension system, and evaluating them by the implementation of necessary analysis via suitable Computer Aided Engineering (CAE) software. As part of the analysis, a virtual model will then be generated which can be further evaluated to determine its reliability. In the end it is hoped that the yielded results and data will be used further in introducing the enhancements to the existing suspension system applied to Proton Gen-2 model. This system hopefully will be able to improve the conventional one, so that it can be implemented and transferred into reality in the next future.

1.2 PROBLEM STATEMENT

Aim of this research is the front suspension system of Gen-2 Proton model being studied, in term of quality and safety. The study is motivated by the need for modeling the system for reference and analysis purposes. A thorough analysis conducted which defines the internal and external forces acting on the system is much appreciated. With this, it is hope that further enhancement on the existing model being developed as an improved suspension system. The feedback gained after the research was being carried out on the enhanced model, can be further evaluated. The new system which is going to be developed should not jeopardize the ride and handling quality of the car. Apart from that, it should be reliable in terms of manufacturability wise, affordable as well as cost effective to manufacture.

1.3 THE OBJECTIVES

The objective of this project is to complete a feasible study on the independent McPherson strut front suspension system of a Proton Gen-2 model car. To achieve this, implementation of CAE software namely ADAMS/Car, AutoCAD and MATLAB SIMULINK will be needed. It is hoped that the model which is going to be develop will be capable of providing the ride and handling comfort. The objectives of the project can be summed up as follow;

a) To carry out a systematic, comprehensive research and analysis on the working philosophy of an independent front McPherson suspension system.
b) To analyze the front suspension system based on the factors that affecting the performance of the system, by implementing the application of ADAMS/Car, MATLAB SIMULINK and so on.
c) To execute further analytical research to assess the reliability of the developed model.

CHAPTER 2

LITERATURE REVIEW

2.1 TWO DEGREES OF FREEDOM SYSTEM – QUARTER MODEL

In reality, a vehicle consists of a multiple spring-mass-damper system has six degrees of freedom (DOF). For the sake of simplifying assumption for the fundamental analysis on the system, the vehicle and suspension can be modeled in two dimensions. In actual thorough research and analysis of the suspension, entirely, there is no way we can disregard the lateral and transverse compliance, although the effective transverse and longitudinal stiffness of the suspension are much greater than the vertical stiffness. In veracity they may have a large impact on the overall dynamics of the vehicle.

There are two main sources of vibration excitations that influence how suspension system reacts. Both of them will have the effect on the ride comfort and handling. The excitation of the vehicle spring mass system is provided by the motion of the tire, which is the unsprung mass. This motion is primarily initiated by the unevenness of the road surface (noise). The second source is weight of load carried by the car. This includes mass of driver, other passengers as well as excessive luggage.

Figure 2.1 shows the displacement of the body (sprung mass, m_2) defined by x_2, whereas the displacement of the unsprung mass (m_1) is designated as x_1, as shown, and the Free Body Diagram (FBD) of the system.

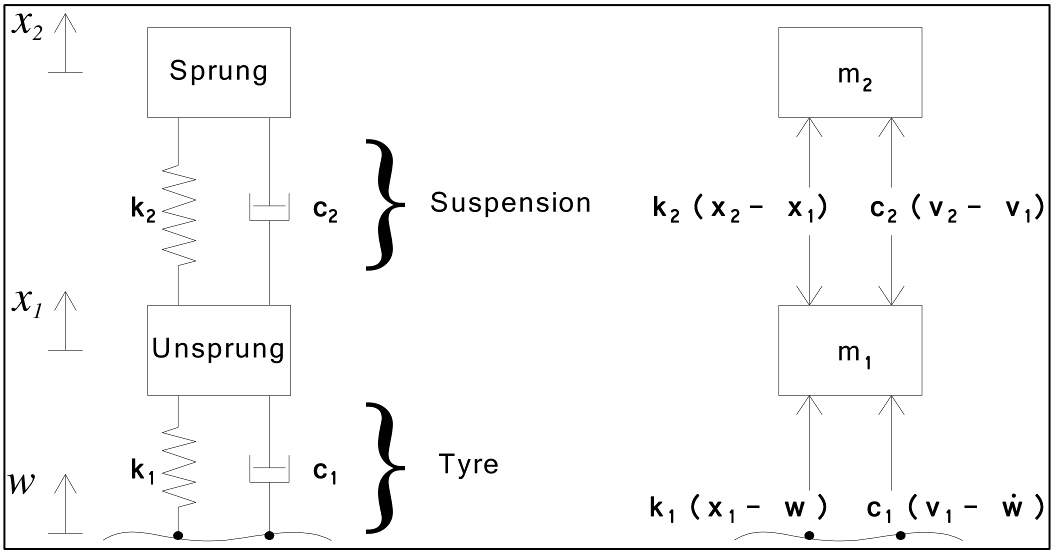


Figure 2.1-1: (Left) Vehicle excited by the motion of the masses due to road unevenness, and (Right) model of Free Body Diagram

<u>The sum of forces on unsprung mass, m_1</u>

$\sum F_1 = m_1\ddot{x}_1$

$k_1(x_1 - w) + c_1(\dot{x}_1 - \dot{w}) - k_2(x_2 - x_1) - c_2(\dot{x}_2 - \dot{x}_1) = m_1\ddot{x}_1$

$-c_2\dot{x}_2 + (c_1 + c_2)\dot{x}_1 - c_1\dot{w} - k_2x_2 + (k_1 + k_2)x_1 - k_1w = m_1\ddot{x}_1$

$\mathbf{m_1\ddot{x}_1 + c_2\dot{x}_2 - (c_1 + c_2)\dot{x}_1 + c_1\dot{w} + k_2x_2 - (k_1 + k_2)x_1 + k_1w = 0}$

<u>Similarly, the sum of forces on sprung mass, m_2</u>

$\sum F_2 = m_2\ddot{x}_2$

$k_2(x_2 - x_1) + c_2(\dot{x}_2 - \dot{x}_1) = m_2\ddot{x}_2$

$c_2\dot{x}_2 - c_2\dot{x}_1 + k_2x_2 - k_2x_1 = m_2\ddot{x}_2$

$\mathbf{m_2\ddot{x}_2 - c_2\dot{x}_2 + c_2\dot{x}_1 - k_2x_2 + k_2x_1 = 0}$

2.2 THREE DOF SYSTEM – QUARTER MODEL WITH DRIVER MASS

The force displacement characteristic of the spring have a direct effect on the level of disturbance transmitted from the road profile to the sprung mass, and thus to the passengers. When a tire hit a bump, the unsprung mass will follow the road profile and by doing so, exerts a certain force and displacement to the suspension spring. The spring response depends on its force-displacement characteristic, or in other words, its stiffness[2].

Figure 2.2 shows the displacement of the body (sprung mass, m_2) defined by x_2, whereas the displacement of the unsprung mass (m_1) is designated as x_1, as shown, and the Free Body Diagram (FBD) of the system. However, this time the addition of mass of driver, m_3 as well as the stiffness and damping characteristic of the seat, k_3 and c_3 is introduced. With that the investigation to reduce the effects of vibrations upon the driver due to road disturbance can be evaluated.

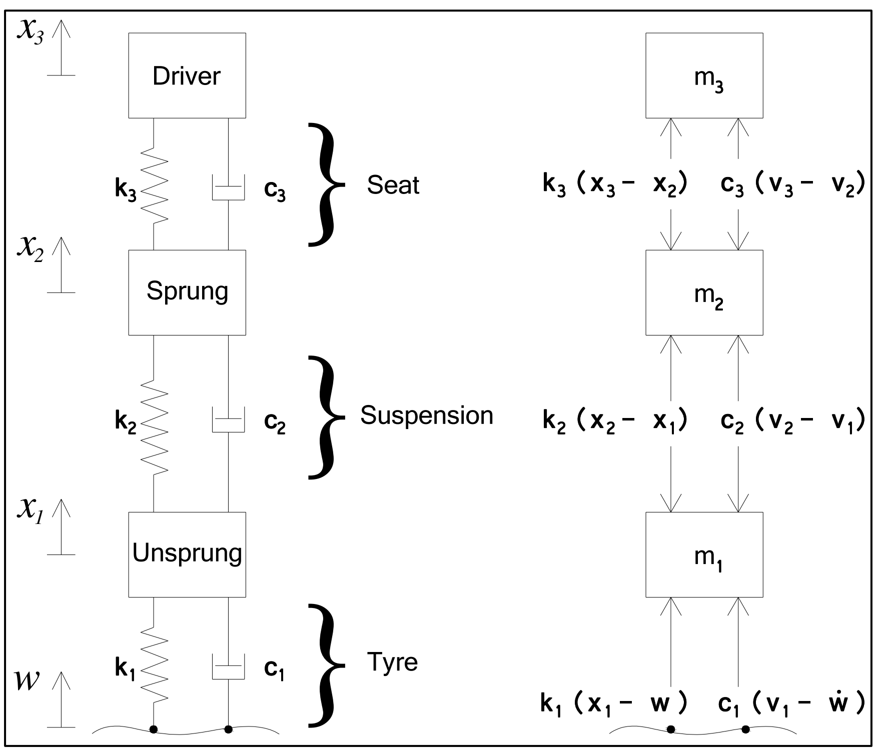


Figure 2.2-1: (Left) Vehicle excited by the motion of the masses due to road unevenness, and (Right) model of Free Body Diagram with mass of the driver

The sum of forces on unsprung mass, m_1

$\sum F_1 = m_1\ddot{x}_1$

$k_1(x_1 - w) + c_1(\dot{x}_1 - \dot{w}) - k_2(x_2 - x_1) - c_2(\dot{x}_2 - \dot{x}_1) = m_1\ddot{x}_1$

$-c_2\dot{x}_2 + (c_1 + c_2)\dot{x}_1 - c_1\dot{w} - k_2x_2 + (k_1 + k_2)x_1 - k_1w = m_1\ddot{x}_1$

Bring all terms on RHS to LHS,

$\mathbf{m_1\ddot{x}_1 + c_2\dot{x}_2 - (c_1 + c_2)\dot{x}_1 + c_1\dot{w} + k_2x_2 - (k_1 + k_2)x_1 + k_1w = 0}$

Similarly, the sum of forces on sprung mass, m_2

$\sum F_2 = m_2\ddot{x}_2$

$k_2(x_2 - x_1) + c_2(\dot{x}_2 - \dot{x}_1) - k_3(x_3 - x_2) - c_3(\dot{x}_3 - \dot{x}_2) = m_2\ddot{x}_2$

$-c_3\dot{x}_3 + (c_2 + c_3)\dot{x}_2 - c_2\dot{x}_1 - k_3x_3 + (k_2 + k_3)x_2 - k_2x_1 = m_2\ddot{x}_2$

Bring all terms on RHS to LHS,

$\mathbf{m_2\ddot{x}_2 + c_3\dot{x}_3 - (c_2 + c_3)\dot{x}_2 + c_2\dot{x}_1 + k_3x_3 - (k_2 + k_3)x_2 + k_2x_1 = 0}$

Similarly, the sum of forces on driver mass, m_3

$\sum F_3 = m_3\ddot{x}_3$

$k_3(x_3 - x_2) + c_3(\dot{x}_3 - \dot{x}_2) = m_3\ddot{x}_3$

$c_3\dot{x}_3 - c_3\dot{x}_2 + k_3x_3 - k_3x_2 = m_3\ddot{x}_3$

Bring all terms on RHS to LHS,

$\mathbf{m_3\ddot{x}_3 - c_3\dot{x}_3 + c_3\dot{x}_2 - k_3x_3 + k_3x_2 = 0}$

2.3 THREE DOF SYSTEM – ROLL AND PITCH MODEL

By using Lagrange Method, the equation of motion of vehicle track can be obtained.

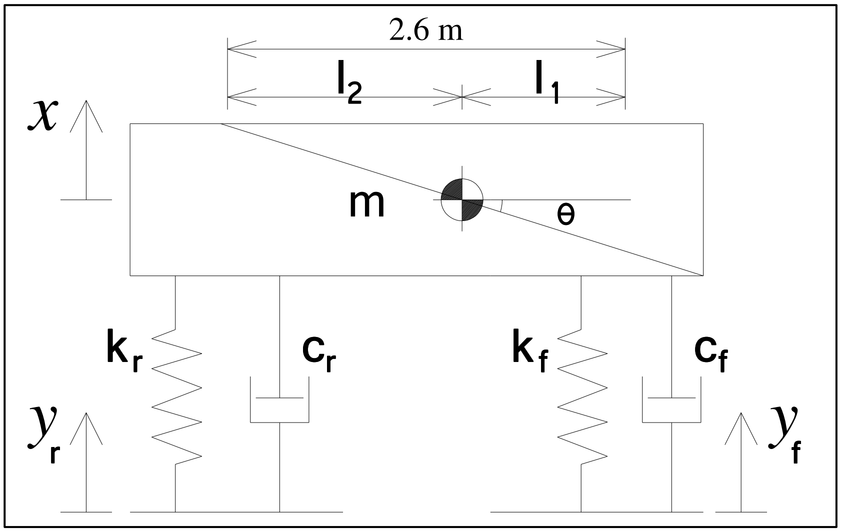


Figure 2.3-1: Three DOF system of a track

Main equation → $$\frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q}_i}\right) - \frac{\delta L}{\delta q_i} = Q_i \quad (1)$$

for $\boldsymbol{i} = \boldsymbol{x}$, $$\frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q}_x}\right) - \frac{\delta L}{\delta q_x} = Q_x \quad (2)$$

and for $\boldsymbol{i} = \boldsymbol{\theta}$, $$\frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q}_\theta}\right) - \frac{\delta L}{\delta q_\theta} = Q_\theta \quad (3)$$

where $Q_x = -c_f(\dot{x} - l_1\dot{\theta} - \dot{y}_f) - c_r(\dot{x} - l_2\dot{\theta} - \dot{y}_r)$

and $Q_\theta = -l_1 c_f(\dot{x} - l_1\dot{\theta} - \dot{y}_f) - l_2 c_r(\dot{x} - l_2\dot{\theta} - \dot{y}_r)$

With respect to the **main equation**, the first term can be solved as follow:

For $\boldsymbol{i = x}$,

$$\frac{\delta L}{\delta \dot{q_x}} = m\dot{x},\ \frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q_x}}\right) = m\ddot{x}, \tag{4}$$

and

For $\boldsymbol{i = \theta}$,

$$\frac{\delta L}{\delta \dot{q_\theta}} = J\dot{\theta},\ \frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q_\theta}}\right) = J\ddot{\theta}, \tag{5}$$

Following are the equations to solve for the second term, $\frac{\delta L}{\delta q_i}$ for each $i = x$ and $i = \theta$

$$L = T - U \tag{6}$$

where

$T =$ Kinetic Energy, $\quad T = \frac{1}{2}m\dot{x}^2 + \frac{1}{2}J\dot{\theta}^2$, and

$U =$ Potential Energy, $\quad U = \frac{1}{2}k_f\left(x - l_1\theta - y_f\right)^2 + \frac{1}{2}k_r(x - l_2\theta - y_r)^2$

Substitute T and U into the equation (6), we have

$$L = \left(\frac{1}{2}m\dot{x}^2 + \frac{1}{2}J\dot{\theta}^2\right) - \left(\frac{1}{2}k_f\left(x - l_1\theta - y_f\right)^2 + \frac{1}{2}k_r(x - l_2\theta - y_r)^2\right)$$

$$L = \frac{1}{2}m\dot{x}^2 + \frac{1}{2}J\dot{\theta}^2 \underbrace{- \frac{1}{2}k_f\left(x - l_1\theta - y_f\right)^2}_{\textbf{term A}} \underbrace{- \frac{1}{2}k_r(x - l_2\theta - y_r)^2}_{\textbf{term B}} \tag{7}$$

For $\boldsymbol{i} = \boldsymbol{x}$ and $\boldsymbol{i} = \boldsymbol{\theta}$,the first two terms in eq. (7) will be slash out since only term A and B has the unknown x in them. Those remaining terms will be solved by the application of the **Chain Rule**.

For $\boldsymbol{i} = \boldsymbol{x}$**, term A**,

$$L_{(A)} = -\frac{1}{2}k_f(u)^2, \qquad \text{where we assign} \qquad u = x - l_1\theta - y_f$$

$$\frac{\delta L_{(A)}}{\delta q_x} = \frac{\delta L_{(A)}}{\delta u} \cdot \frac{\delta u}{\delta x}$$

$$\frac{\delta u}{\delta x} = 1, \qquad \frac{\delta L_{(A)}}{\delta u} = -k_f(u)$$

$$\frac{\delta L_{(A)}}{\delta q_x} = -k_f\left(x - l_1\theta - y_f\right) \tag{8}$$

Similarly, for $\boldsymbol{i} = \boldsymbol{x}$**, term B**,

$$L_{(B)} = -\frac{1}{2}k_r(u)^2, \qquad \text{where we assign} \qquad u = x - l_2\theta - y_r$$

$$\frac{\delta L_{(B)}}{\delta q_x} = \frac{\delta L_{(B)}}{\delta u} \cdot \frac{\delta u}{\delta x}$$

$$\frac{\delta u}{\delta x} = 1, \qquad \frac{\delta L_{(B)}}{\delta u} = -k_r(u)$$

$$\frac{\delta L_{(B)}}{\delta q_x} = -k_r(x - l_2\theta - y_r) \tag{9}$$

Hence, for $\boldsymbol{i} = \boldsymbol{x}$, the equations (4), (8) and (9) are substituted into **main equation**, (2)

$$\frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q}_x}\right) - \frac{\delta L}{\delta q_x} = Q_x$$

$$\therefore \boldsymbol{m\ddot{x} + k_f\left(x - l_1\theta - y_f\right) + k_r(x - l_2\theta - y_r)}$$
$$\boldsymbol{= -c_f\left(\dot{x} - l_1\dot{\theta} - \dot{y}_f\right) - c_r\left(\dot{x} - l_2\dot{\theta} - \dot{y}_r\right)}$$

For $\boldsymbol{i = \theta}$, term A,

$$L_{(A)} = -\frac{1}{2}k_f(u)^2, \qquad \text{where we assign} \qquad u = x - l_1\theta - y_f$$

$$\frac{\delta L_{(A)}}{\delta q_\theta} = \frac{\delta L_{(A)}}{\delta u} \cdot \frac{\delta u}{\delta \theta}$$

$$\frac{\delta u}{\delta \theta} = -l_1, \quad \frac{\delta L_{(A)}}{\delta u} = -k_f(u)$$

$$\frac{\delta L_{(A)}}{\delta q_\theta} = l_1 k_f\left(x - l_1\theta - y_f\right) \tag{10}$$

Similarly, for **term B**,

$$L_{(B)} = -\frac{1}{2}k_r(u)^2, \qquad \text{where we assign} \qquad u = x - l_2\theta - y_r$$

$$\frac{\delta L_{(B)}}{\delta q_\theta} = \frac{\delta L_{(B)}}{\delta u} \cdot \frac{\delta u}{\delta \theta}$$

$$\frac{\delta u}{\delta \theta} = -l_2, \quad \frac{\delta L_{(B)}}{\delta u} = -k_r(u)$$

$$\frac{\delta L_{(B)}}{\delta q_\theta} = l_2 k_r(x - l_2\theta - y_r) \tag{11}$$

Hence, for $\boldsymbol{i = \theta}$, the equations (5), (10) and (11) are substituted into **main equation**, (3)

$$\frac{d}{dt}\left(\frac{\delta L}{\delta \dot{q}_\theta}\right) - \frac{\delta L}{\delta q_\theta} = Q_\theta$$

$$\therefore \boldsymbol{J\ddot{\theta} - l_1 k_f\left(x - l_1\theta - y_f\right) - l_2 k_r(x - l_2\theta - y_r)}$$
$$\boldsymbol{= -l_1 c_f\left(\dot{x} - l_1\dot{\theta} - \dot{y}_f\right) - l_2 c_r\left(\dot{x} - l_2\dot{\theta} - \dot{y}_r\right)}$$

2.4 INDEPENDENT FRONT SUSPENSION SYSTEM PARTS OVERVIEW

Coil spring is a rod made of steel which is wounded into a spiral. It supports the weight of the vehicle (sprung mass) while still allowing for suspension travel; permits the control arm and wheel to move up and down. Coil springs compress and expand to absorb the motion of the wheels. The stiffness of the springs affects how the sprung mass responds while the car is being driven. For a spring with low spring rate, it can tolerate bumps well and provide a considerably smooth and relatively comfortable ride, whilst for a suspension with a high spring rate, such as one being installed in the sport race car, is less forgiving on bumpy roads, but ironically increased the handling ability so that it can be driven aggressively even around corners, where such characteristic is much pleaded for this type of car.

The sprung mass is the mass of the vehicle that is supported by the springs and the suspension system, while the **unsprung mass** is the mass between the road and the suspension springs (tires, wheels, wheel bearings, knuckles, axle housing) which are not supported by the springs. Sprung mass should be kept relatively high in proportion to the unsprung mass. Unsprung mass should be kept low to improve ride smoothness. It is because a high unsprung mass (for example, heavy wheels and suspension components) would tend to transfer vibration to the passenger compartment.

Strut damper (or shock absorber) keeps the suspension from continuing to bounce after spring compression / extension. Its lower end is bolted to the knuckle while the other end is mounted to the vehicle chassis via a ball joint. The prime function of a damper is to dissipate the energy absorbed by the coil spring. It limits the spring oscillations (compression-extension movements) to smoothen the vehicle ride. Without it, after striking a dip or hump in the road, the vehicle will continue to bounce up and down at natural frequency of the spring until all of the energy originally put into it is used up. This would make the ride uncomfortable and unsafe; the passengers will experience an extremely bouncy ride, and the driver will find it very difficult with the car handling. Damper slows down and reduces the magnitude of vibratory motions by turning the kinetic energy of suspension movement into heat energy that can be dissipated through hydraulic fluid.

2.5 PARAMETER ADJUSTMENTS

The architecture and design of the suspension system depends on several influential parameters and also restricted to set of constraints existed. In order to improvise the performance and capability, the ‘what-if’ analysis will be executed by changing the values of the parameters so that the best results can be obtained, with respects to those constraints. As for an example, the position of the upper mounting point cannot be changed, as this will possibly affect the design of the subframe, wheel housing and fender, and consequently the front part of the car. This is something that cannot be done. Alternatively what we can do is to change the spring stiffness so that it can adhere to the excess amount of load being carried up by the car. The analysis of parameter adjustments will be conducted in three main points, namely spring characteristic, damping characteristic, and finally the hardpoints adjustment or geometrical positions (toe and camber angle).

2.5.1 Spring Characteristic

There are three common properties that can be manipulated in executing the analysis with respect to the characteristic of the spring. They are free length of spring, coil diameter D and wire diameter d. Any modification done to each of these will consequently results in the change of the spring stiffness (or spring rate) value k. In the case of executing analysis by application of ADAMS/Car software, the value of spring stiffness k is changed directly without paying extra attention to which properties is actually affected.

Springs have a significant effect on the feel of a car and its responsiveness to steering input. Increasing spring rate at a particular corner of the car transfers more weight to that corner making it slide more. If a car is understeering, by decreasing front spring stiffness or increasing rear spring stiffness will correct the condition. The opposite setting configuration (increase front spring stiffness or decrease rear spring stiffness) can be done in order to correct oversteer condition.

By changing the length of the spring, its spring stiffness will significantly change as well. Installing lowered (shorter) springs on car can provide a substantial improvement in handling. The center of gravity of the car will be lower, which will allow the four tires to stay more evenly planted to the ground. Lowering it too much will not allow for enough suspension travel. The car may bottom out on the suspension (suspension reaches full travel) or the chassis may actually hit the ground, leading to loss of traction and possible damage to the underside of car. This constraint will determine the boundary limits in adjusting the length of the spring.

The stiffness of the spring will determine the upper and lower limits of the amount of loads that the system can compensate. A stiffer spring (higher spring rate) will be sufficient to limit the suspension travel and prevent bottoming out. A too stiff spring will lead to the wheels being lifted off the ground instead of being absorbed by the suspension when the tire hits even a small bump. If the car bottoms out, it will experience a sudden loss in traction which will lead to a spinout (out of control).

2.5.2 Damping Characteristic

Dampers are the most useful tool for adjusting corner entry and corner exit handling characteristics of a street car. They are easily accessible on most cars, and aftermarket shocks are usually adjustable in many ways. Shocks are necessary to dampen, or gradually reduce, the bouncing of the springs on the car. Without them, the suspension would oscillate up and down for an extended period of time after running over a bump or experiencing weight transfer.

Pre adjustment on the damping value is adequate to improve the car handling. A decreased value of front compression or increased value of rear rebound will improve handling when the car experiencing understeer on corner entry. An increased value of front compression or decreased value of rear rebound, on the other hand will improve handling for oversteered car.

2.5.3 Camber

Camber is the angle of the wheel relative to vertical, as viewed from the front of the car. A wheel has a negative camber when it leans in towards the chassis and a positive one if it leans away from the car. Independent suspensions are designed so that the camber varies as the wheel moves up and down relative to the chassis.

The cornering force that a tire can develop is highly dependent on its angle relative to the road surface, and so wheel camber has a major effect on the road holding of a car. A tire's performance is at optimum when its contact patch area is perpendicular to road surface because it provides the maximum road grip for the tire. It is preferred to have a zero camber angle in a straight line driving. In a real situation however, common practice is to have a slightly negative camber angle, so that it can compensate the positive camber gained during cornering. The usual setting value of camber angle for road cars will be -1 to -2 °.

Camber angle alters the handling qualities of a particular suspension design; in particular, negative camber improves grip when cornering. This is because it places the tire at a more optimal angle to the road, transmitting the forces through the vertical plane of the tire, rather than through a shear force across it. Another reason for negative camber is that a rubber tire tends to roll on itself while cornering. If the tire had zero camber, the inside edge of the contact patch would begin to lift off of the ground, thereby reducing the area of the contact patch. By applying negative camber, this effect is reduced, thereby maximizing the contact patch area. Note that this is only true for the outside tire during the turn; the inside tire would benefit most from positive camber. On the other hand, for maximum straight-line acceleration, the greatest traction will be attained when the camber angle is zero and the tread is flat on the road.

2.5.4 Toe

Toe is the symmetric angle that each wheel makes with the longitudinal axis of the vehicle, as a function of static geometry, and kinematic and compliant effects. Positive toe, or toe in, is the front of the wheel pointing in towards the centerline of the vehicle. Negative toe, or toe out, is the front of the wheel pointing away from the centerline of the vehicle. Toe can be measured in linear units, at the front of the tire, or as an angular deflection.

CHAPTER 3

METHODOLOGY

3.1 IMPLEMENTATION OF ENGINEERING ANALYSIS SOFTWARE

Advantages of working with CAE software are (1) the analysis and the changes of the design can be executed much faster and at a lower cost than physical prototype testing requires, (2) work in a secure virtual environment, whereby all the data can be saved and kept digitally, and (3) the initial design can be further improved in terms of its quality and reliability by exploring numerous design variations, for this case by changing the kinematic parameter of the suspension system in order to optimize the performance.

3.2 MATLAB AND SIMULINK – ANALYTICAL ANALYSIS

SIMULINK is an element from MATLAB software whereby it is useful to model a simple mechanical system and observe its behavior under various conditions. A model of a car suspension can be represented by schematic diagrams provided. Model parameters, such as the stiffness of the spring, the mass of the body, or the force profile can be varied according to any desired values. The resulting changes to the velocity and position of the body can be viewed graphically as a result.

By the application of SIMULINK, several important analyses on the behavior of the suspension model under a given force can be carried out. The underlying concept of executing the analysis is just the same as ADAMS/Car, whereby in SIMULINK, spring and damping characteristics can be manipulated, and then the results can be compared to obtain the optimum configuration of the suspension. After the suspension diagram had been build, the analysis can be run. The initial or baseline setting then be modified, for instance the spring stiffness and the analysis will be carried out once again to obtain the results. Values of the parameter will be kept adjusted until the desired optimization can be achieved. Shown below is the example of the diagram which represents a simple basic model of a car suspension, together with the output of modeled and simulated second-order under-damped dynamic system.

One distinction between SIMULINK and ADAMS/Car is that the mathematical equations and formulas, for instance in a shape of transfer functions which involved in the suspension system can be applied and manipulated, which is something that is unavailable in ADAMS/Car. That is why by modeling suspension in SIMULINK will broaden the understanding on the response of the system when subjected to loadings or forces. Several spring characteristics which can be manipulated in the analysis are the spring stiffness and the damper viscosity. Other parameters are the input signal, step size, and several others. As the results, SIMULINK will provide the mass displacement and velocity as a function of time, and these results will be shown in the graphical wise.

3.3 ADAMS/CAR – KINEMATIC ANALYSIS

ADAMS/Car is a motion simulation solution for analyzing the complex behavior of mechanical assemblies. Virtual prototypes and optimize designs for performance, safety, and comfort can be tested without having to build and test numerous physical prototypes. ADAMS/Car is used in the study to model the suspension system, to simulate its behavior and to carry out sufficient analysis on the model. Key parameters such as the spring and damping characteristic can easily be changed to explore "what-if" studies. Adequate analysis for example tire test rig and steady state cornering can be performed to determine its effectiveness, and the behavior of the suspension model can be animated. Finally the dynamic results can be plotted by graph.

With ADAMS/Car, deep understanding on the behavior of suspension can be evaluated by executing thorough analysis on it. Once the kinematics of the assembly had been assessed, further modification can be carried out to determine how the suspension will act due to the changes. For instance, we would like to see what happened when the geometry is modified. Change in the inclination of kingpin axis will results in decreasing of the scrub radius. Change in diameter coil of the spring will certainly affect the capability of the suspension in absorbing shock caused by the roads irregularities. These are some of the expected results that can be obtained from the analysis using ADAMS/Car. Sets of results compiled from any particular analysis can be compared among them to determine if the modifications were successful, and finally to find the best setting for the suspension. With that it is hoped the optimization of suspension model can be achieved.

The methodology of implementing the study is simplified by three ways; **flow diagram**, **the flow chart** and the **Gantt chart**.

3.4 FLOW DIAGRAM

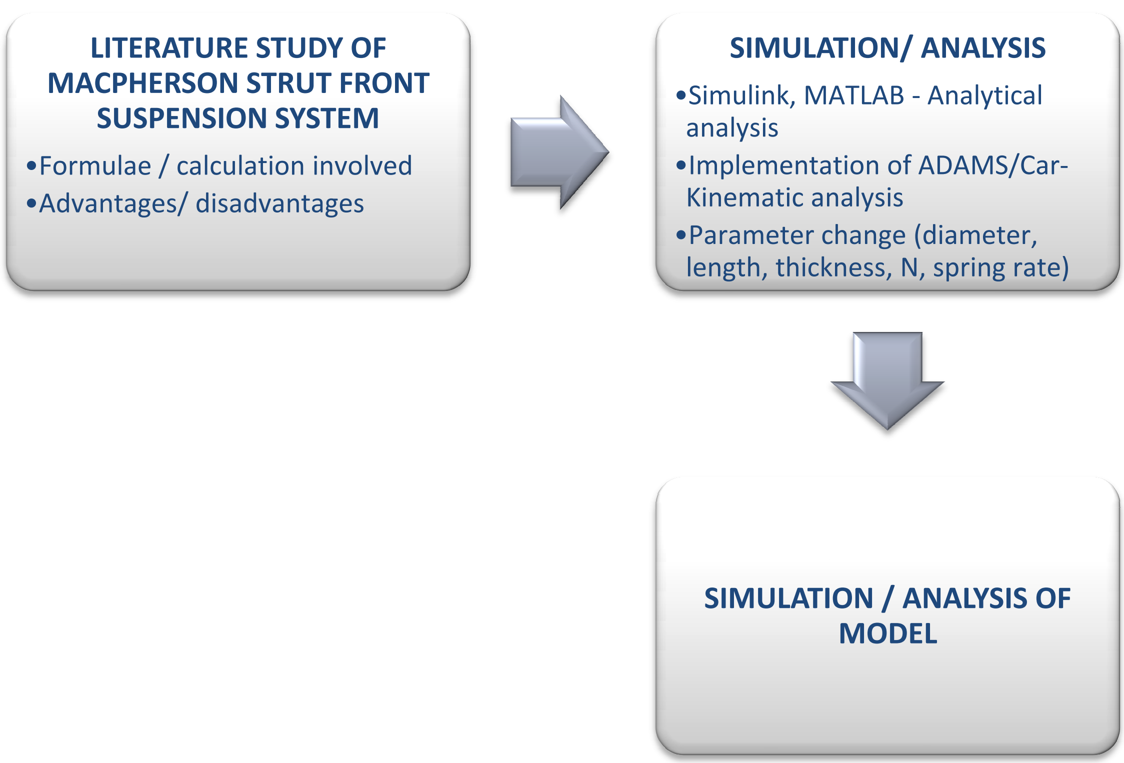


Figure 3.4-1: Flow diagram

3.5 FLOW CHART

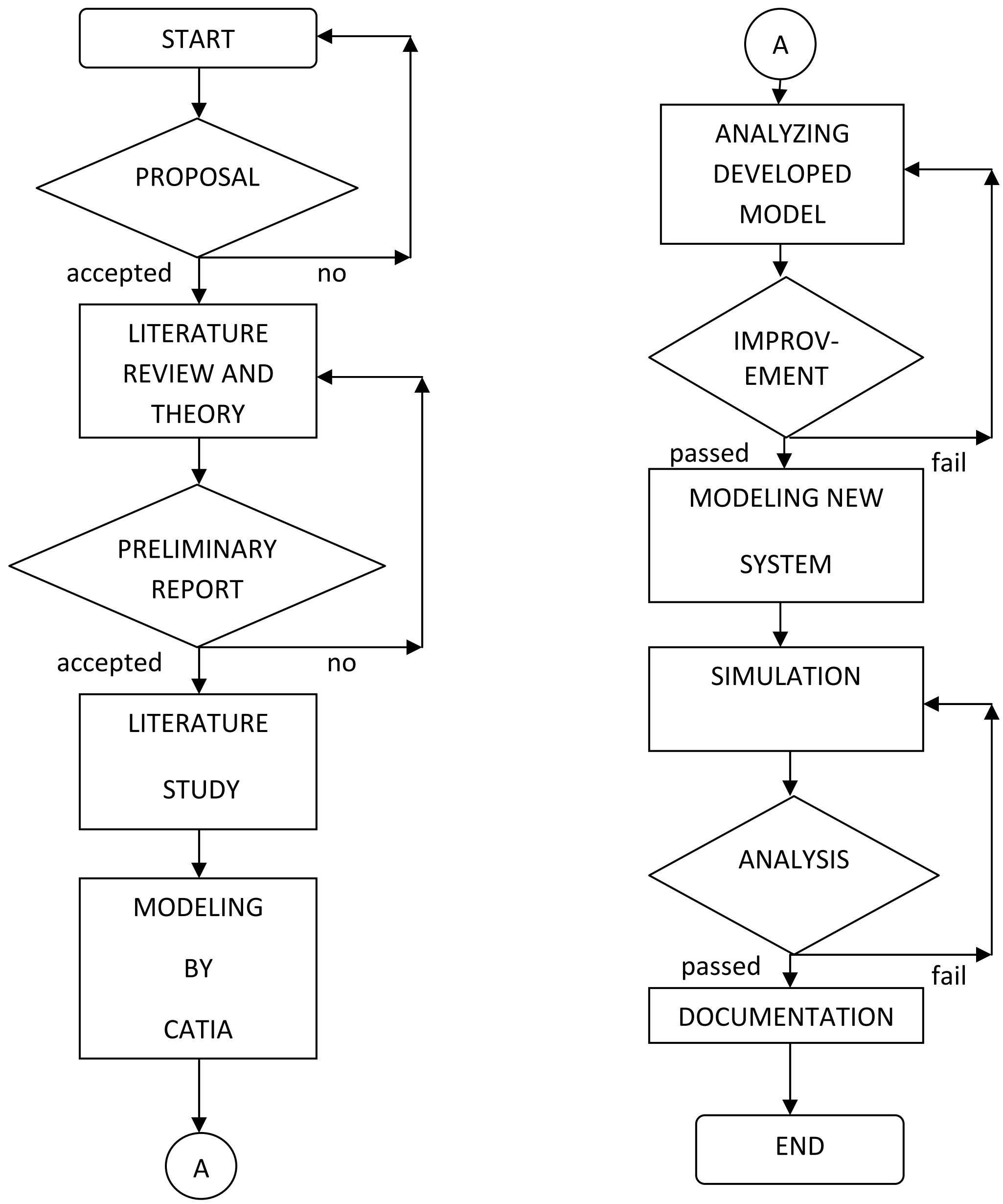


Figure 3.5-1: Flow chart

3.6 GANTT CHART

Table 3.6a: Gantt chart

No	Detail / Week	1	2	3	4	5	6	7	8	9	10		11	12	13	14
1	Selection / Proposal of Project Topic											Mid Semester Break (Hair Raya Aidilfitri Holiday)				
2	Literature Review and Theory															
3	Submission of Preliminary Report				●											
4	Literature Study of Front Suspension System															
5	Submission of Progress Report								●							
6	Seminar								●							
7	Initial modeling by SIMULINK															
8	Analyzing Developed Model															
9	Submission of Interim Report Final Draft														●	
10	Oral Presentation															●

Process ●Suggested Milestone

No	Detail /Week	15	16	17	18	19	20	21		22	23	24	25	26	27	28
11	Improvement of The Existing Model								Mid Semester Break							
12	Submission of Progress Report 1				●											
13	SIMULINK model of 2 and 3 DOF															
14	Submission of Progress report 2									●						
15	Seminar									●						
16	Initial work on ADAMS/Car															
17	Poster Exhibition											●				
18	Analysis executed															
19	Submission of Dissertation (Soft Bound)													●		
20	Oral Presentation														●	
21	Submission Project Dissertation (Hard Bound)															●

Process ●Suggested Milestone

3.7 MATLAB SIMULINK – SYSTEM INPUT OF QUARTER CAR MODEL

The system is modeled by application of SIMULINK software. SIMULINK is an element from MATLAB software whereby it is useful to model a simple mechanical system and observe its behavior under various conditions. A model of a car suspension can be represented by schematic diagrams provided. Model parameters, such as the stiffness of the spring, the mass of the body, or the force profile can be varied according to any desired values. Initial parameters for this quarter car model being setup as following:

3.7.1 Parameter for Mass, Stiffness and Damper

Table 3.7a: Parameter values for tire and suspension of the quarter car model

Variable	Attribute	Symbol	Value	Unit
Tire[1]	Stiffness	k_1	500,000	N/m
	Damper	c_1	15,020	Ns/m
Suspension[2]	Stiffness	k_2	16,000	N/m
	Damper	c_2	579	Ns/m

Both sprung and unsprung masses are assumed to be 2500 kg and 320 kg respectively[1].

As for driver,

Table 3.7b: Mass for driver and stiffness and damper of seat

Variable	Attribute	Symbol	Value	Unit
Driver	Mass	m_3	80	kg
	Stiffness	k_3	8,000	N/m
	Damper	c_3	3,000	Ns/m

3.7.2 Input Source of Road

One aspect that will determine how the system reacts due to its change is the road noise. When a car is driving, all four tires are subjected to change of surface of the road. Each individual tire might be subjected to different height of road to each other, contributing to pitch or roll or even both, which simultaneously jeopardizing ride comfort as well as car handling.

For example, figure 3.7-1 shows a sinusoidal wave as the road input. The front tire experienced different height displacement to rear tire. This is due to wavelength of the wave is a bit smaller compared to wheelbase of the car, even though its amplitude is constant throughout the wave. Hence, displacement of front tire to constant road surface without any variation Δw_{frt} is not same with displacement of rear one Δw_{rr}. When car is subjected to this kind of road noise, hence this will lead to pitch.

$$\Delta w_{frt} \neq \Delta w_{rr}$$

where w is the constant road surface without any variation (road variation = 0).

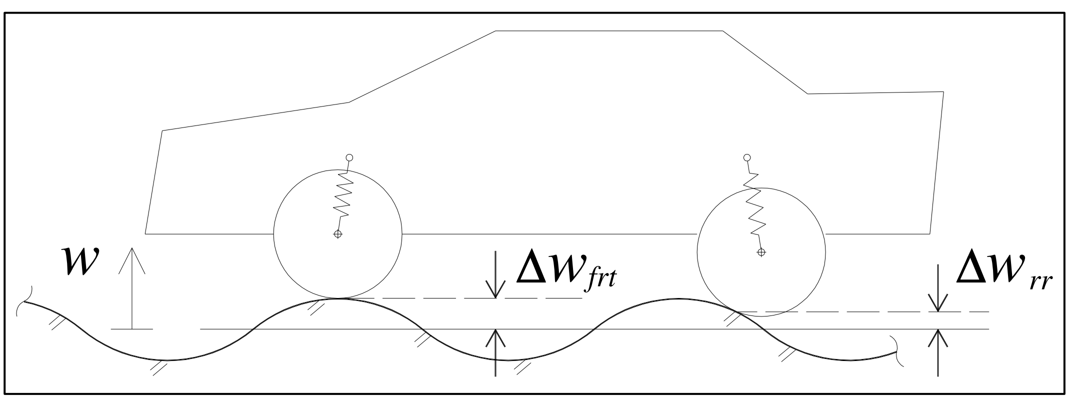
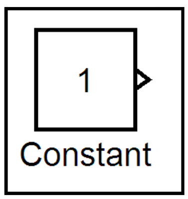
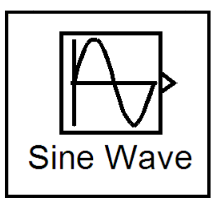
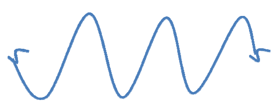
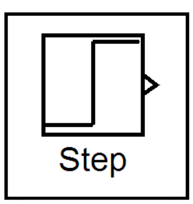
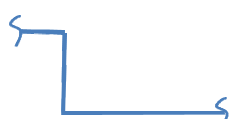
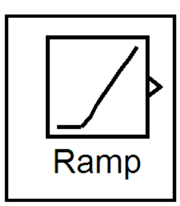
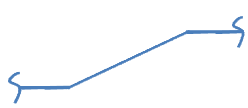


Figure 3.7-1: Sinusoidal wave affecting the amount of deflection of suspension

Up to this stage, the study is focusing only to quarter car model, meaning to investigate the outcome from the system when only one single tire is subjected to different type of road noise. Four common road inputs are (1) constant (replicates smooth, normal road surface condition), (2) sine wave (wavy or bumpy road with similar repeated pattern), (3) step (replicates potholes on bad condition road i.e. under construction), and (4) ramp input for hill slope- alike. Table 3 shows the specification of each type of input being used in simulation.

Table 3.7c: Various type of input of road surface

Input	Example of road condition	Specification
	 Normal	Constant value: 0
	 Wavy	Amplitude: 0.5 Frequency: π rad/sec Phase: 0 rad Sample time: 0
	 Pothole	Step time: 5 Initial value: 0 Final value: -2 Sample time: 0
	 Hill slope	Slope: 1 Start time: 5 Initial output: 0

3.8 SIMULINK DIAGRAM – 2DOF QUARTER MODEL

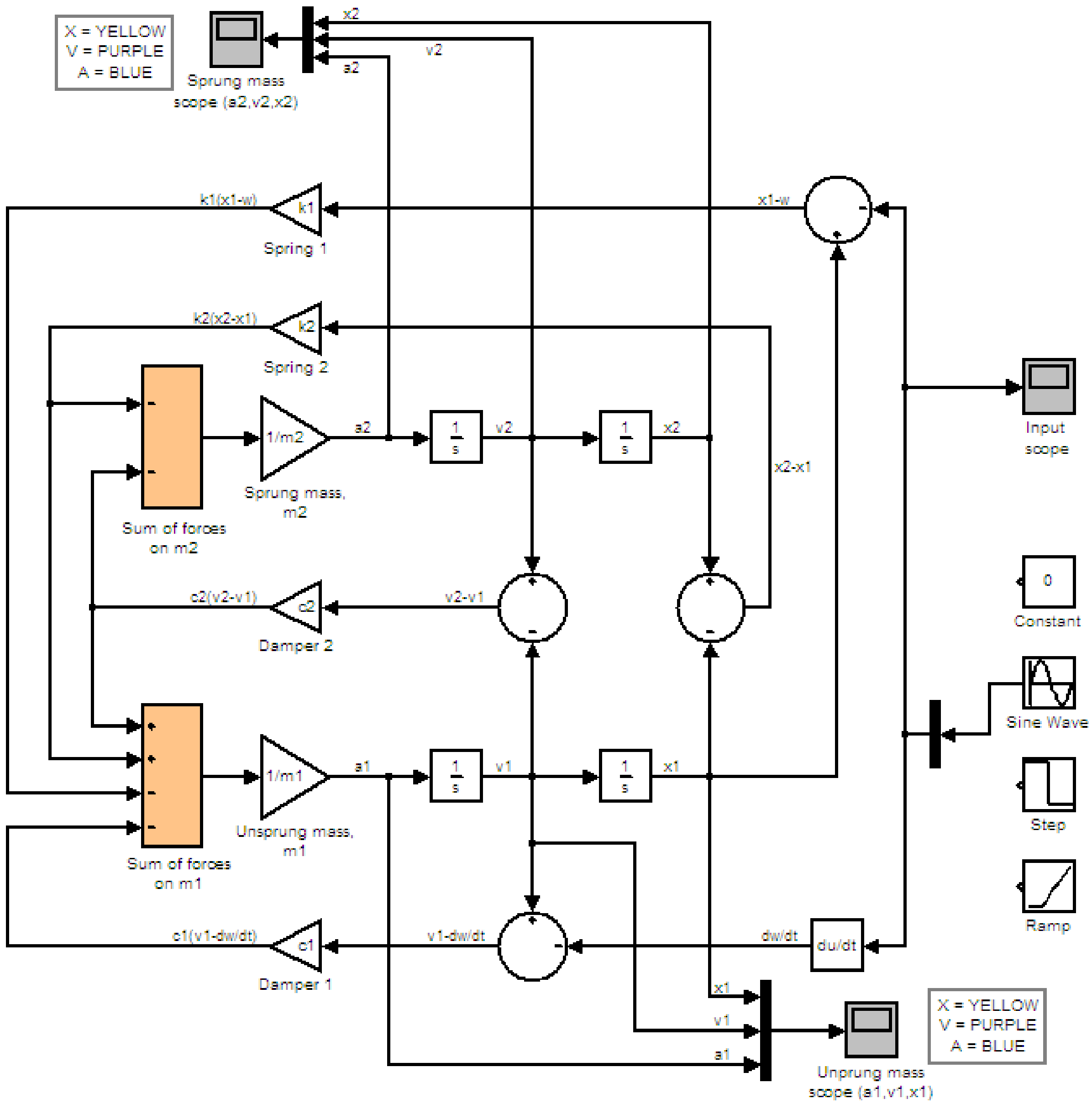


Figure 3.8-1: SIMULINK diagram of 2 Degree of Freedom for quarter car model

3.9 SIMULINK DIAGRAM – 3DOF WITH DRIVER MASS

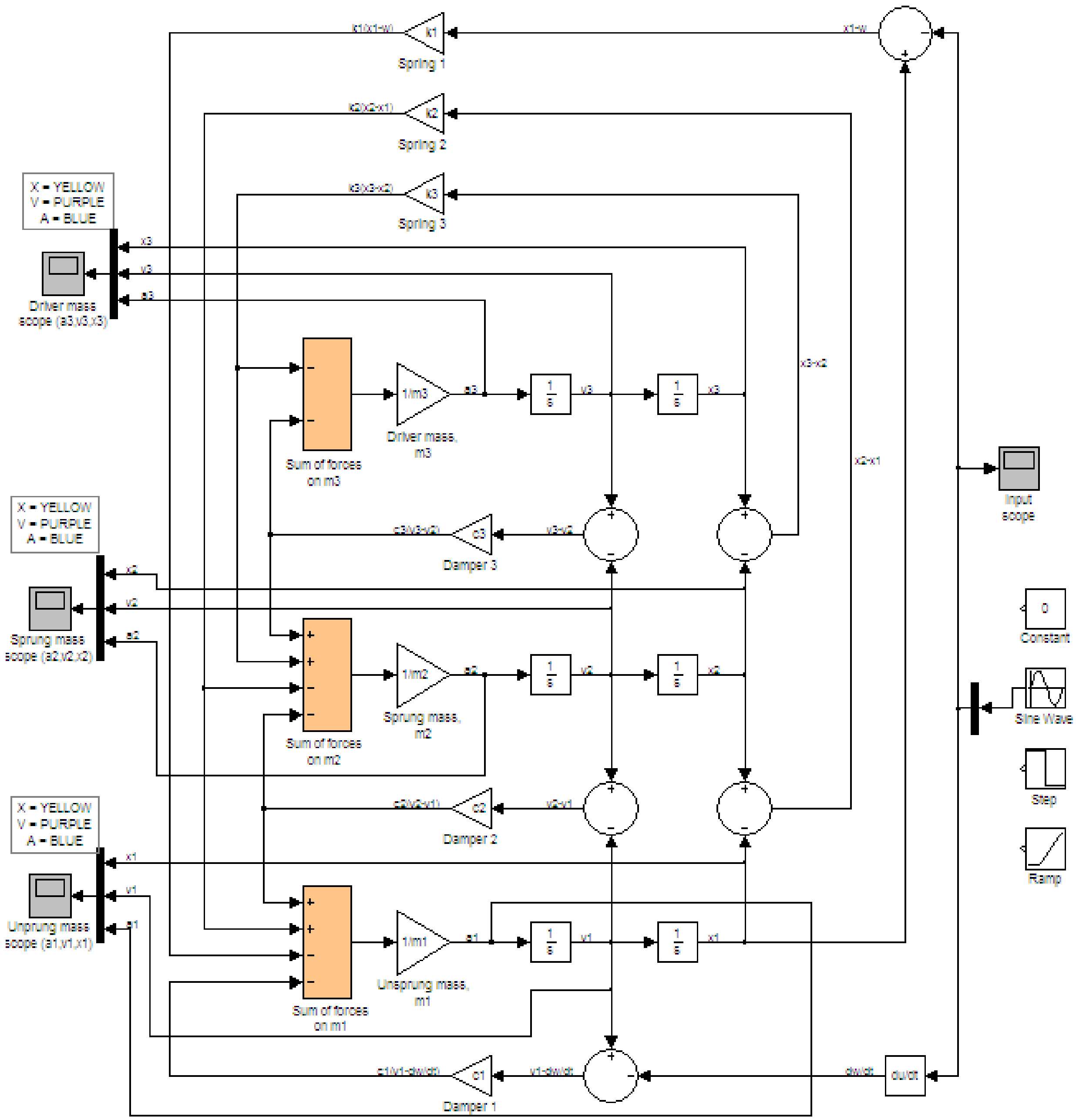


Figure 3.9-1: SIMULINK diagram of 3 Degree of Freedom for quarter car model with driver mass

3.10 ADAMS/CAR – ANALYSIS

Two analyses being conducted are Power off Straight Line analysis and Single Lane Change analysis. Throughout analysis, we only change the parameters of front and keep constant the rear suspension. The Double Wishbone rear suspension is preset to have characteristics of spring stiffness, k of 30,000N/m and damping coefficient of 3,000Ns/m.

All analyses being conducted should replicate as close as possible to the real model of Gen-2 car. In order to achieve that, all important and critical parameters should be following exact model. The parameters are shown in the table below:

Gen- 2 1.6 L Model

Total mass	: 1175kg
Wheelbase	: 2600mm
Front track	: 1475mm
Rear track	: 1470mm

3.10.1 Power off Straight Line (POSL)

POSL is used in the analysis of manipulating spring stiffness k, damping coefficient c and toe angle. The POSL doesn't require any steering input; the model is driven in a straight lane without any cornering. The model is driven at constant speed by external source without getting the power from the engine to drive its movement, hence the term 'power off'. In this case, the objective of the analysis is to obtain the output when a single parameter is manipulated, while keeping the rest constant. For example with respect to analysis of varying spring stiffness k, the ones that need to be maintained are damping coefficient c, toe and camber angle.

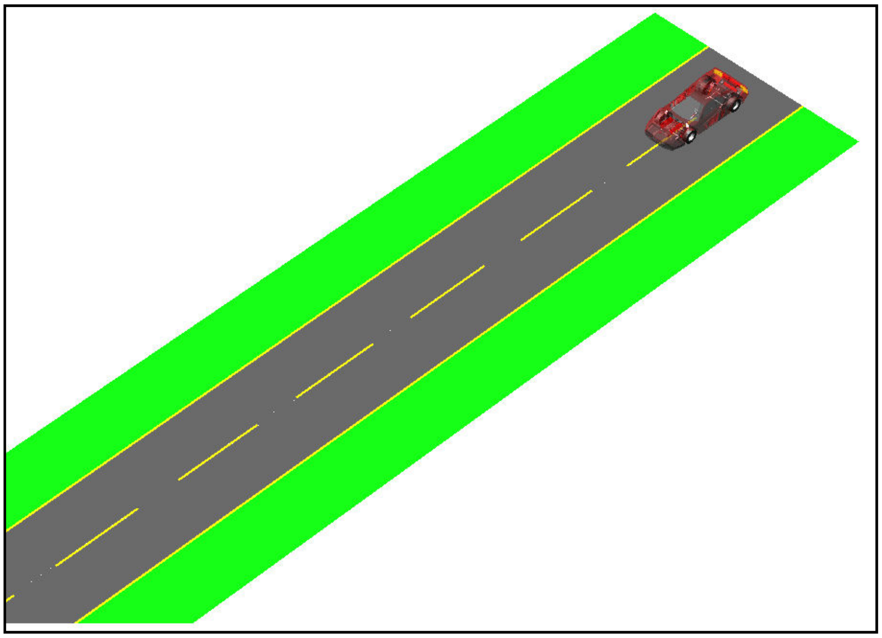


Figure 3.10-1: Power off Straight Line Analysis (POSL)

3.10.2 Single Lane Change (SLC)

During a single lane-change analysis, the steering input goes through a complete sinusoidal cycle over the specified length of time. The steering input can be length, which is a motion applied to the rack of the steering subsystem, angle, which is angular displacements applied to the steering wheel, force applied to the rack, or torque applied to the steering wheel. In this case, throughout analyses the input used is amount of angle being applied, which is a constant value of 45 degree.

3.11 ADAMS/CAR – VARYING PARAMETER VALUES

The major objectives of the analyses are to observe the expected outcomes as the value of particular parameter is changed according to initial values taken from references. For every analysis being carried out, the varying parameter will be divided into two parts; exaggerated values and optimized. Exaggerated values are to set what is the range of parameter values to be tested (upper and lower limits of parameter values). It is common that the values are multiplied by the order of 10 times greater or smaller than the reference value. The resulting outcome graph should posses the desired pattern as pre expected. In order to obtain more accurate analysis outcome after desired pattern had been determined, then the optimized values should follow next.

For example, in case of varying spring stiffness, the exaggerated k values are set to be 1,000N/m and 100,000N/m, which is four times smaller and greater with respect to initial value, 25,000N/m given by reference. From resulting graph, it is obvious that vertical acceleration vs. time of 1,000N/m is very much following the pattern of initial value to be compared to 100,000N/m value. Hence, the optimized values would be in the range of 1,000N/m till 25,000N/m.

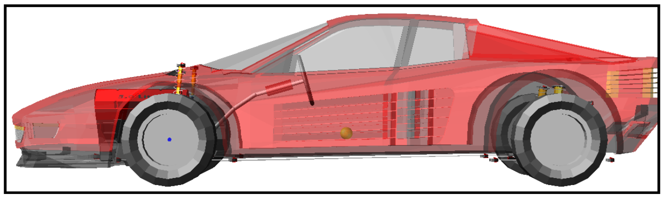


Figure 3.11-1: Vehicle model from side view

3.12 ADAMS/CAR –RECORDED RESULT PLOT

The results of carried out analysis can be observed in many types of plots, depending on what to be evaluated. In all case of analyses, the result of interest is the chassis displacement vs. time and the chassis acceleration vs. time. The latter one is very crucial in analyses since acceleration of chassis will determine the effect of any parameter change to passenger's body. It can be assumed that with minimal chassis acceleration, the lower the 'sensation' effect experienced by passengers. Hence, all analyses being carried out are designated to achieve the smallest acceleration of vehicle as possible with time in order to ensure the comfort is guaranteed, without jeopardizing the handling capability.

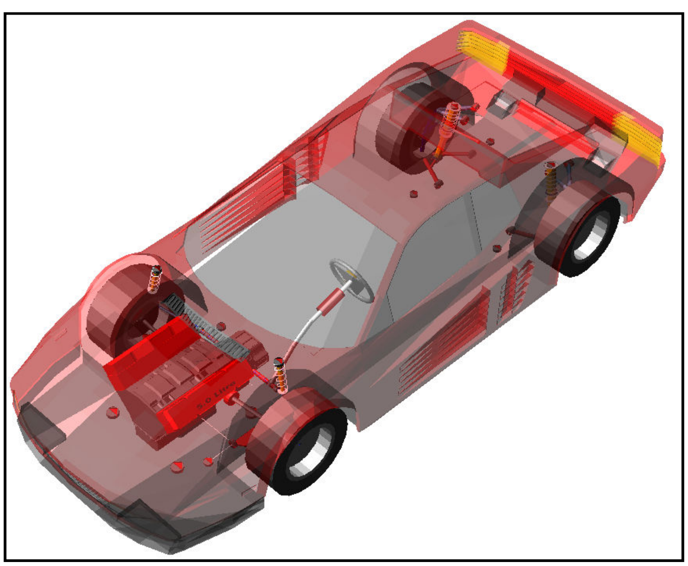


Figure 3.12-1: Vehicle model from standard isometric view

CHAPTER 4

RESULTS AND DISCUSSION

4.1 SIMULINK – OUTCOME OF SYSTEM ON 2 DOF FOR VARIETY INPUTS

<u>For Constant input</u>

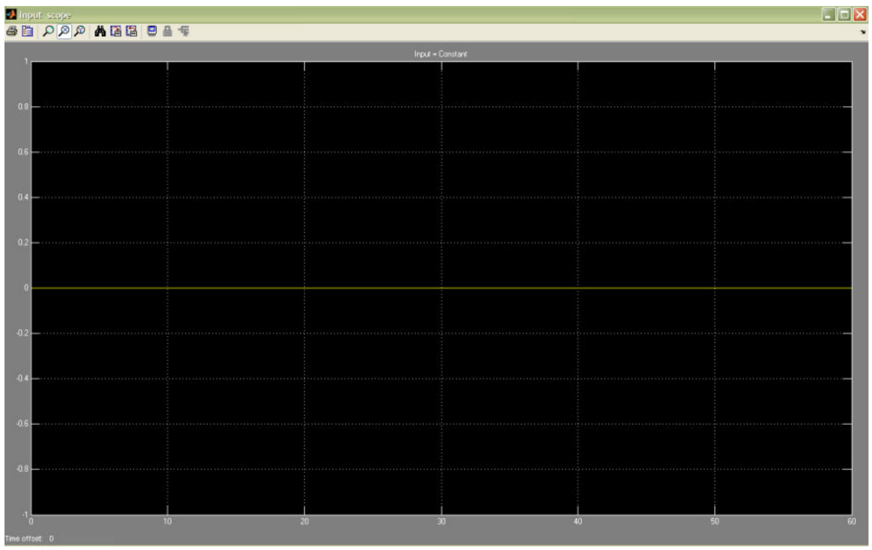


Figure 4.1-1: Constant input scope

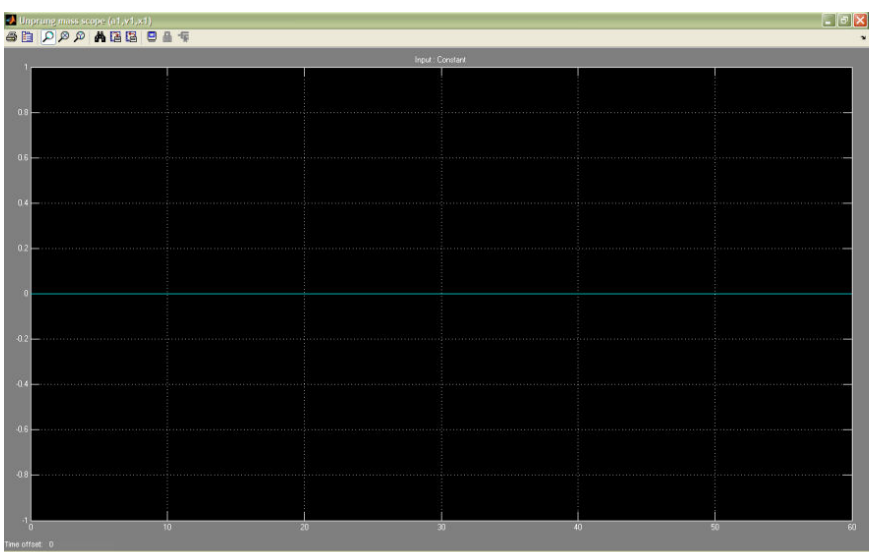


Figure 4.1-2: Acceleration (a1), velocity (v1) and displacement (x1) graph of unsprung mass (m1) scope for constant input

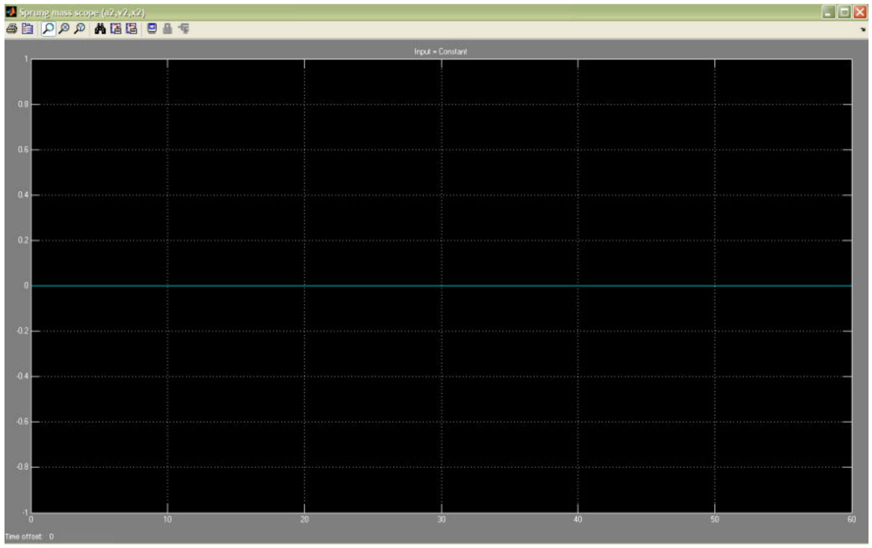


Figure 4.1-3: Acceleration (a2), velocity (v2) and displacement (x2) graph of sprung mass (m2) scope for constant input

For Sinusoidal input

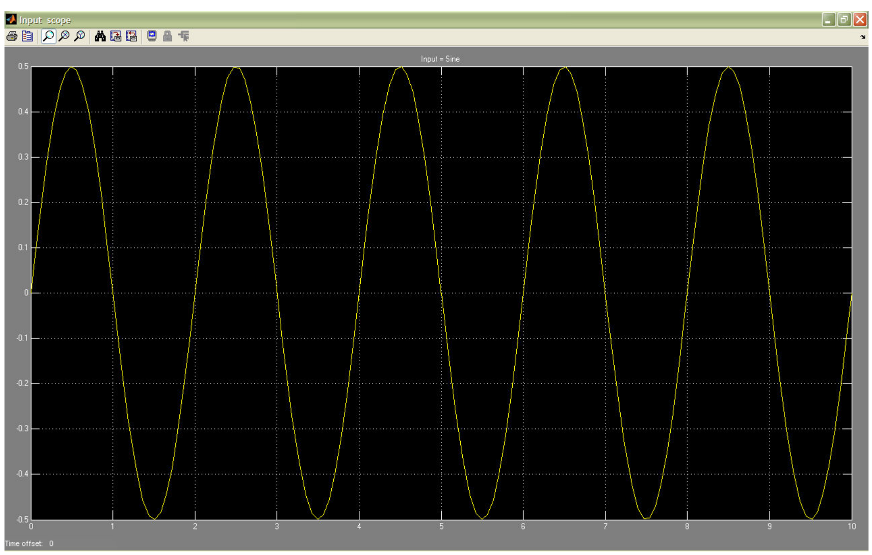


Figure 4.1-4: Sinusoidal input scope

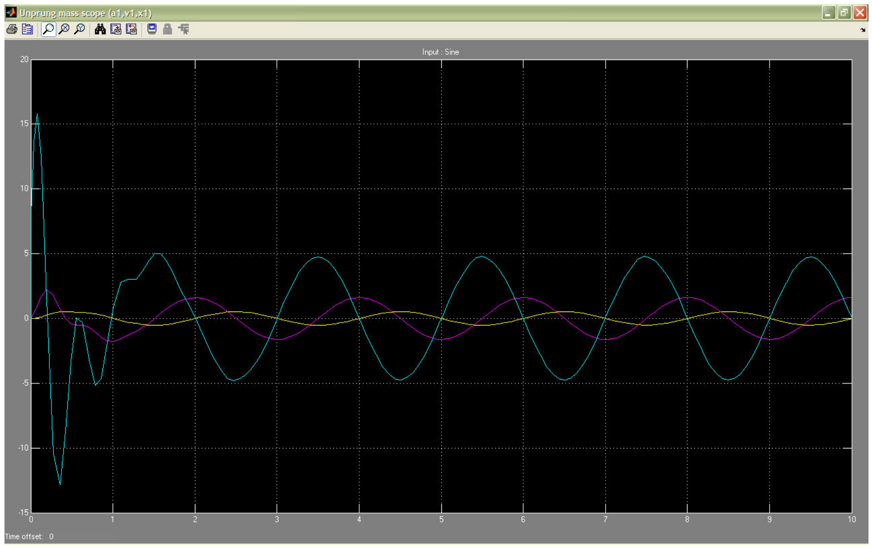


Figure 4.1-5: Acceleration (a1), velocity (v1) and displacement (x1) graph of unsprung mass (m1) scope for sinusoidal input

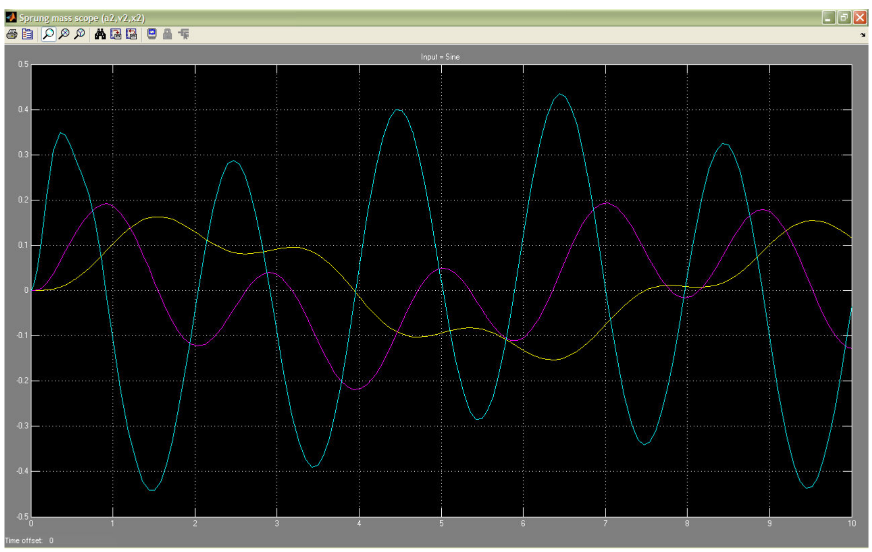


Figure 4.1-6: Acceleration (a2), velocity (v2) and displacement (x2) graph of sprung mass (m2) scope for sinusoidal input

For Step input

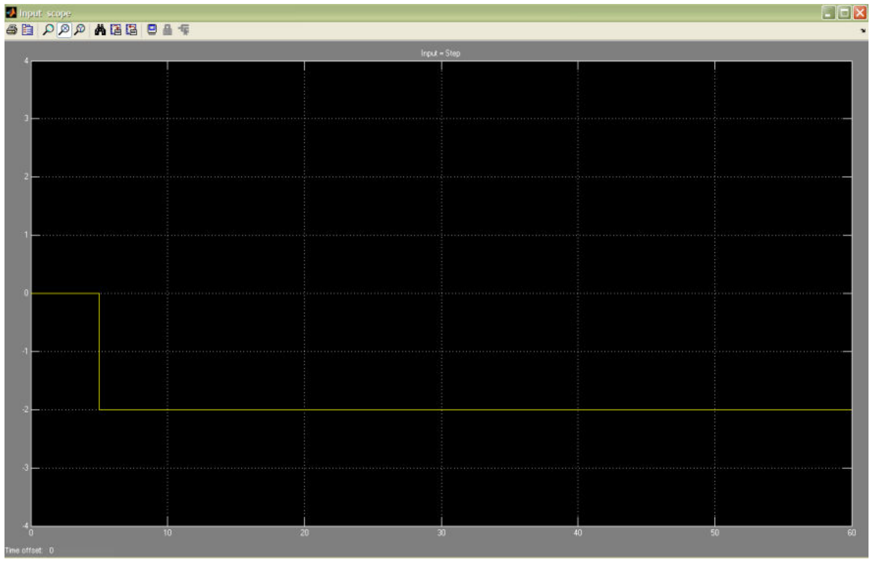


Figure 4.1-7: Step input scope

Figure 4.1-8: Acceleration (a1), velocity (v1) and displacement (x1) graph of unsprung mass (m1) scope for step input

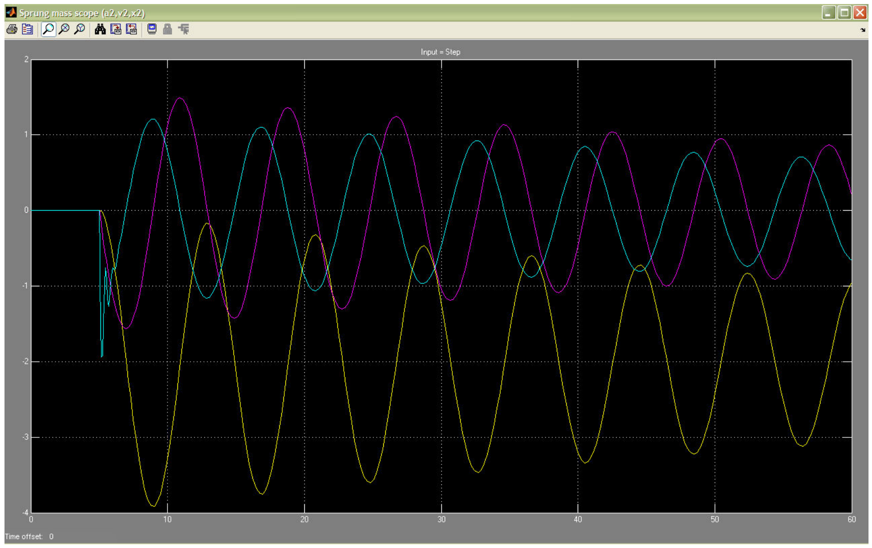


Figure 4.1-9: Acceleration (a2), velocity (v2) and displacement (x2) graph of sprung mass (m2) scope for step input

<u>For Ramp input</u>

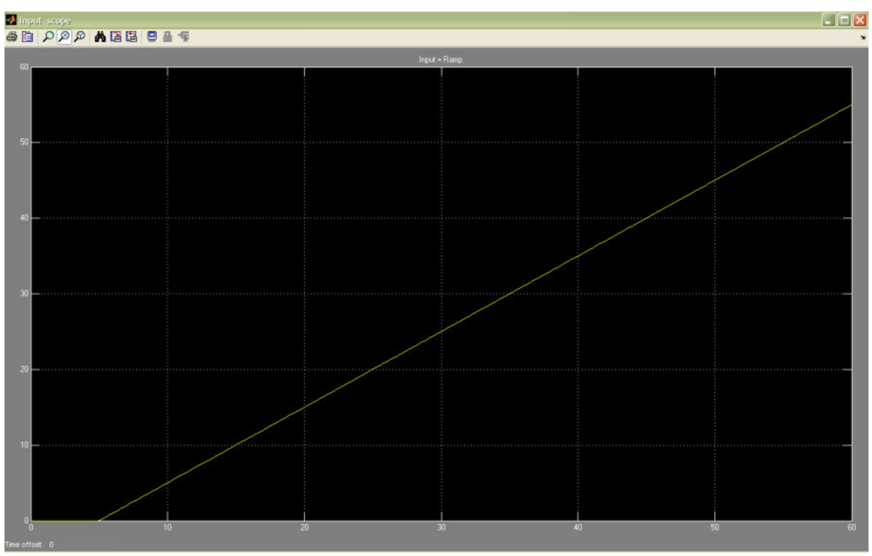


Figure 4.1-10: Ramp input scope

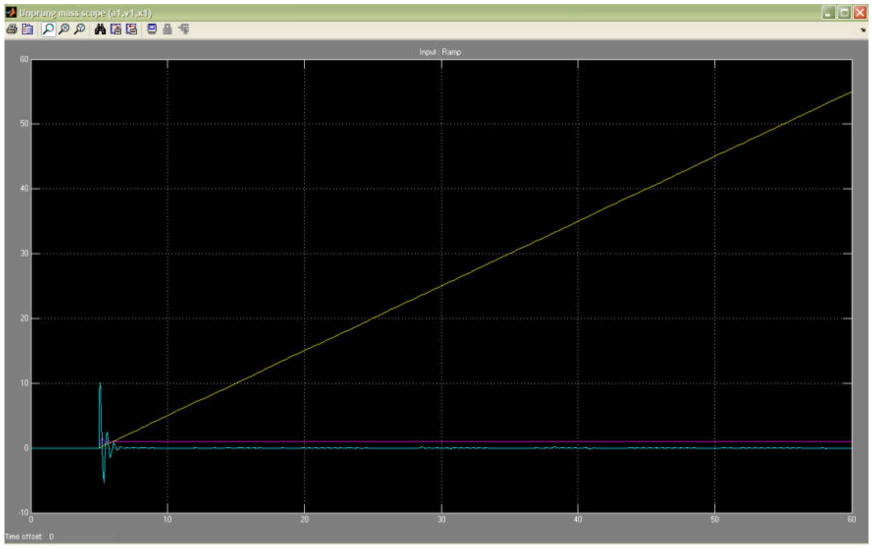


Figure 4.1-11: Acceleration (a1), velocity (v1) and displacement (x1) graph of unsprung mass (m1) scope for ramp input

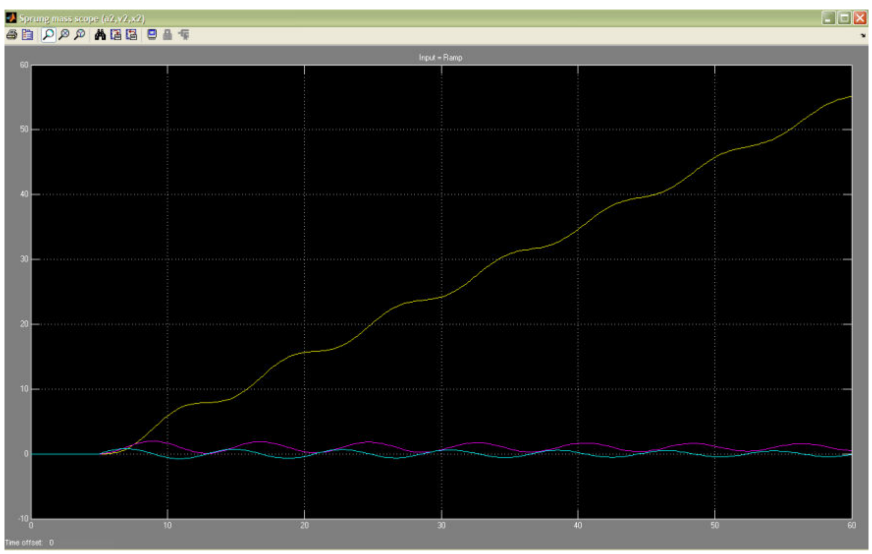


Figure 4.1-12: Acceleration (a2), velocity (v2) and displacement (x2) graph of sprung mass (m2) scope for ramp input

4.2 SIMULINK – OUTCOME OF SYSTEM ON 3 DOF FOR SINE INPUT

Rotation of tire = π rad/sec (means it will require 2 seconds to complete 1 cycle)

Max. amplitude, A = 0.5 m, Phase angle = 0°

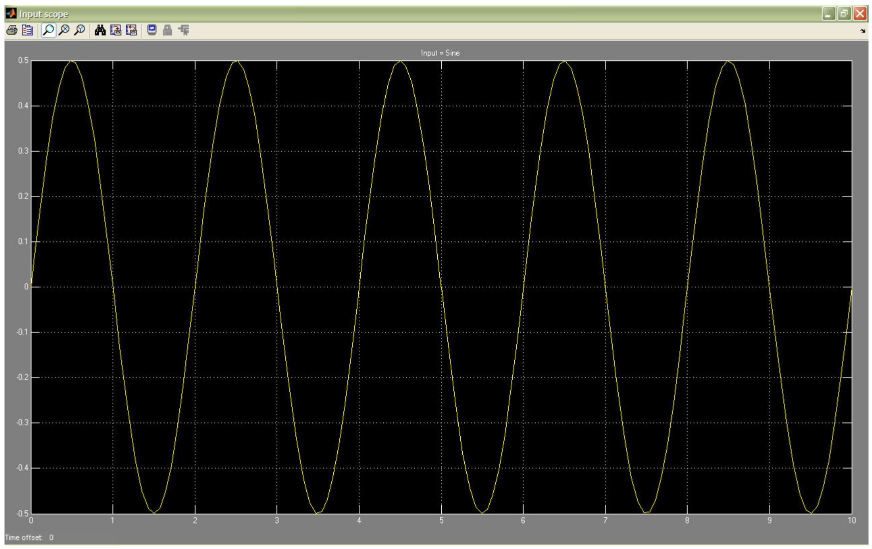


Figure 4.2-1: Sine input scope for 3 DOF (with driver)

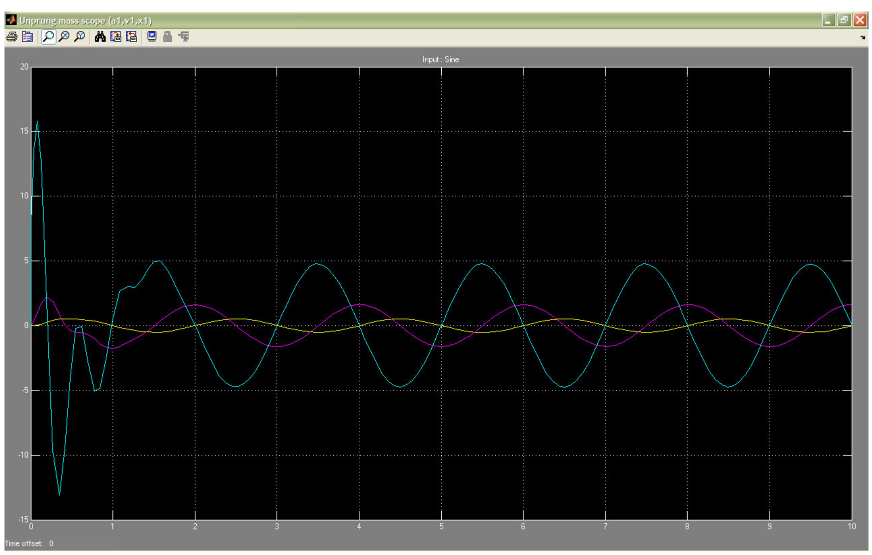


Figure 4.2-2: Acceleration (a1), velocity (v1) and displacement (x1) graph of unsprung mass (m1) scope for sine input for 3 DOF (with driver)

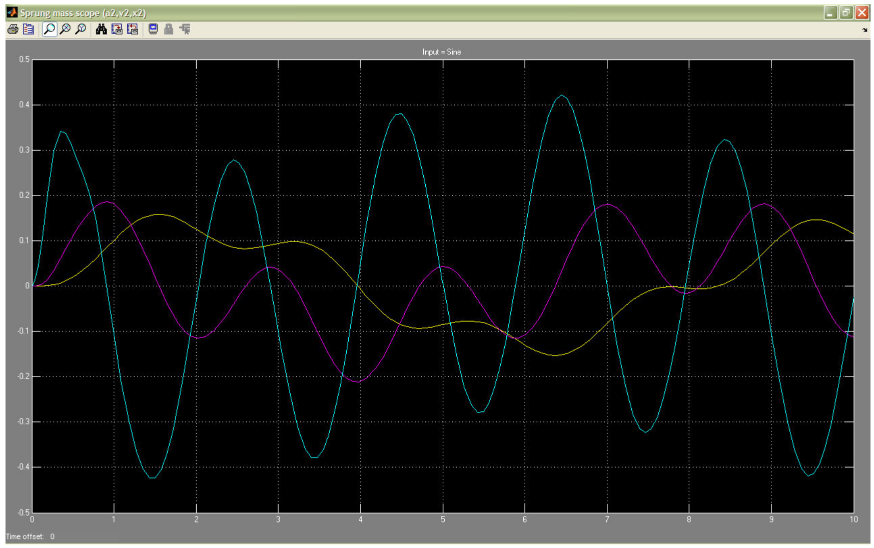


Figure 4.2-3: Acceleration (a2), velocity (v2) and displacement (x2) graph of sprung mass (m2) scope for sine input for 3 DOF (with driver)

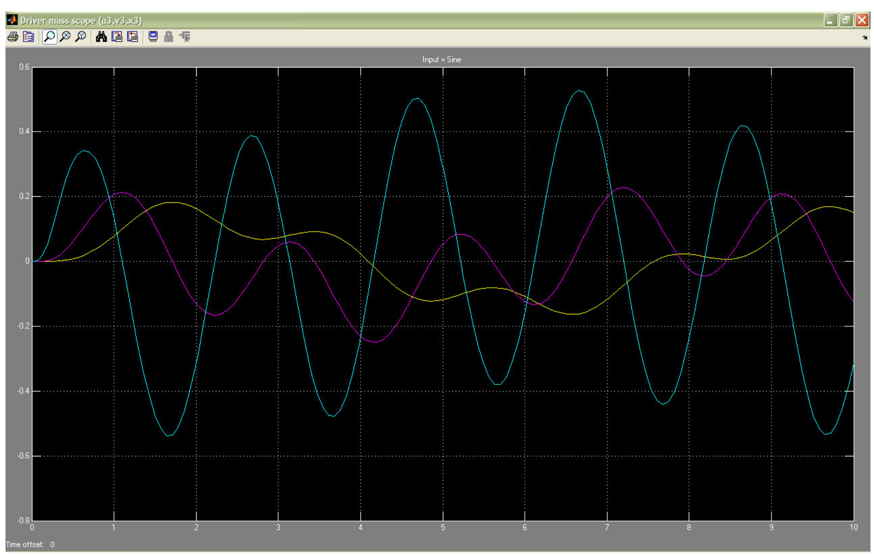


Figure 4.2-4: Acceleration (a3), velocity (v3) and displacement (x3) graph of driver mass (m3) scope for sine input for 3 DOF (with driver)

4.3 ADAMS/CAR – SPRING STIFFNESS K AND DAMPING COEFFICIENT C

4.3.1 Spring Stiffness, k Value

Constant preset for Double Wishbone rear suspension throughout all analysis:

DwB_rr_std	slope:	30	3000N/100mm	=	30000N/m

Constant preset for McPherson front suspension throughout all analysis:

Free length: 205.7mm

Property file: use hardpoints

Limits: -100,100

Manipulation preset of McPherson front suspension:

Exaggerated values tested:

McP_frt_std	**slope:**	**25**	**2500N/100mm**	**=**	**25,000N/m**
McP_frt_k1	slope:	1	100N/100mm	=	1,000N/m
McP_frt_k100	slope:	100	10000N/100mm	=	100,000N/m

Optimized values tested:

McP_frt_k10	slope:	10	1000N/100mm	=	10,000N/m
McP_frt_k30	slope:	30	3000N/100mm	=	30,000N/m

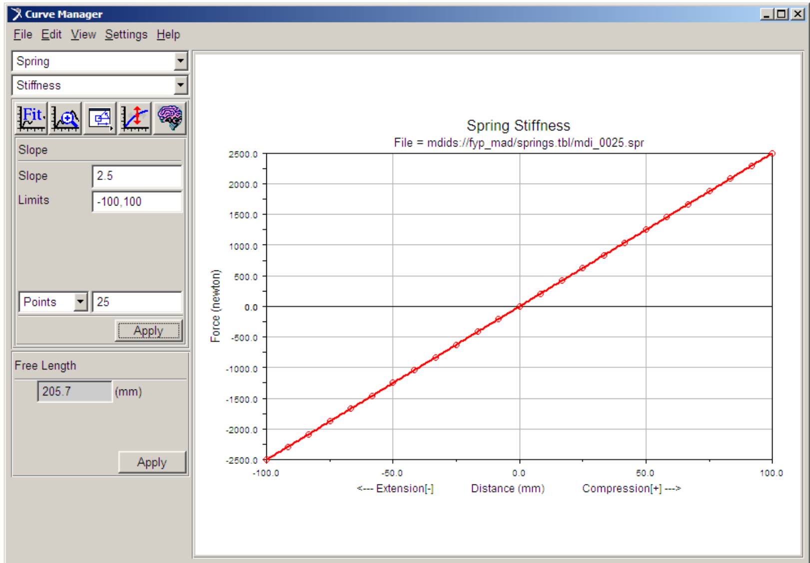


Figure 4.3-1: Example of spring stiffness adjustment

4.3.2 Spring Stiffness, k Plot Result

Analysis: Power Off Straight Line Analysis (posl)

Exaggerated value (k = 1, 100)

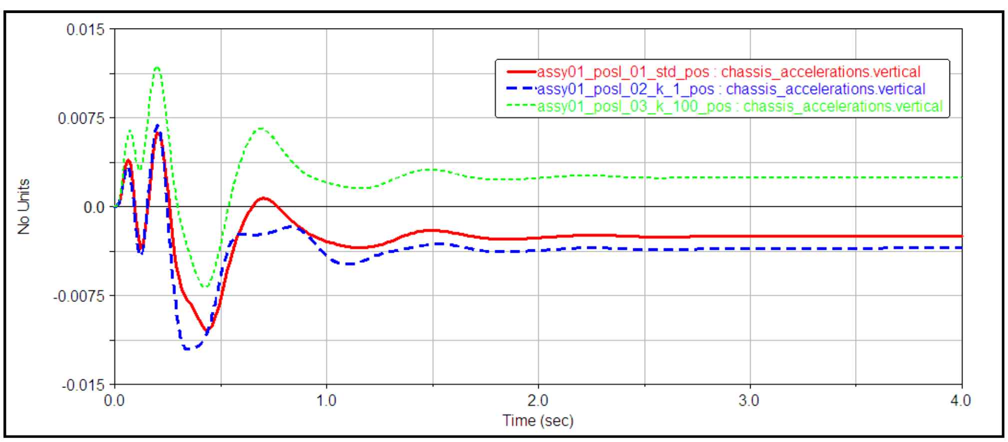


Figure 4.3-2: Vertical chassis acceleration vs. time for k value analysis

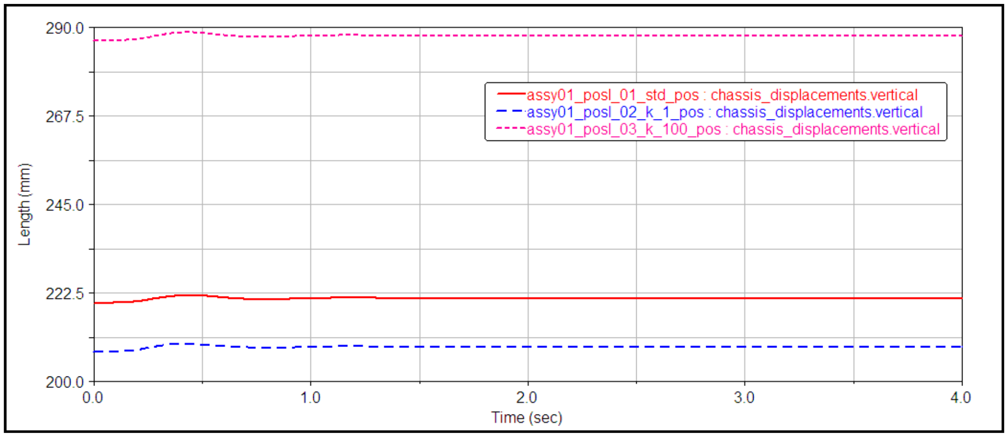


Figure 4.3-3: Vertical chassis displacement vs. time for k value analysis

Optimized value (k = 10, 30)

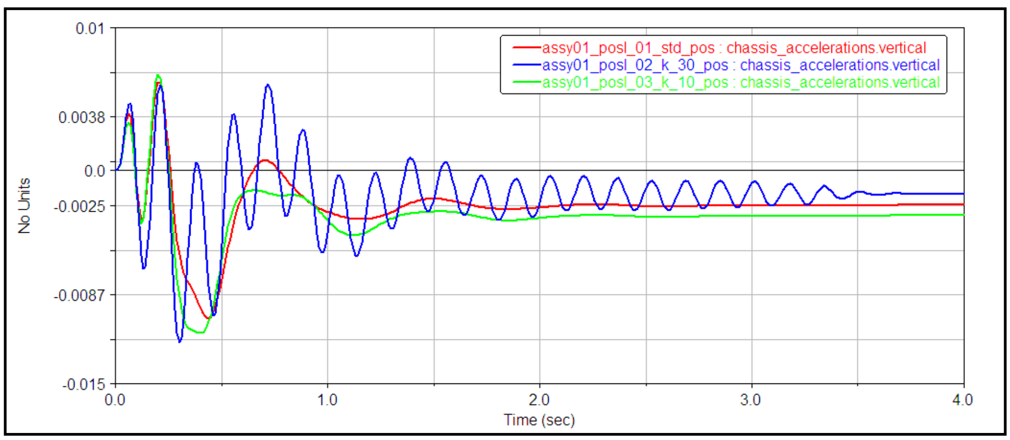


Figure 4.3-4: Vertical chassis acceleration vs. time for k value analysis (optimized)

4.3.3 Damping Coefficient, c Value

Constant preset for Double Wishbone rear suspension throughout all analysis:

DwB_rr_std slope: 3 300N/100mm/s = 3000Ns/m

Constant preset for McPherson front suspension throughout all analysis:

Free length: 205.7mm Limits: -100,100

Manipulation preset of McPherson front suspension:

Exaggerated values tested:

McP_frt_std	**slope:**	**2.5**	**250N/100mm/s**	**=**	**2,500Ns/m**
McP_frt_c_pnt1	slope:	0.1	10N/100mm/s	=	100N/m
McP_frt_c10	slope:	10	1000N/100mm/s	=	10,000Ns/m

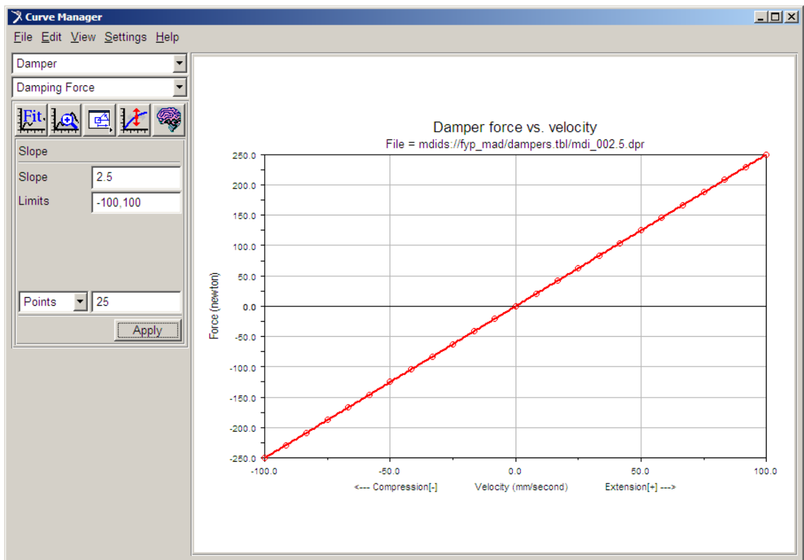


Figure 4.3-5: Example of damping coefficient adjustment

4.3.4 Damping Coefficient, c Plot Result

Analysis: Power Off Straight Line Analysis (posl)

Exaggerated value (c = 0.1, 10)

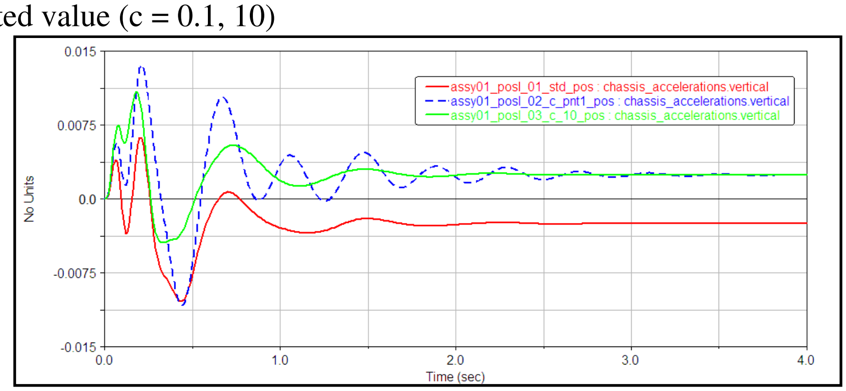


Figure 4.3-6: Vertical chassis acceleration vs. time for c value analysis

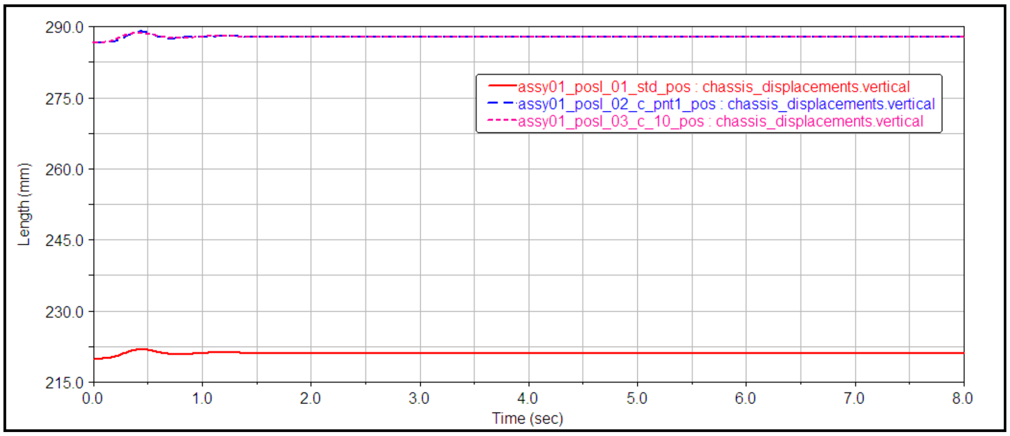


Figure 4.3-7: Vertical chassis displacement vs. time for c value analysis

4.4 ADAMS/CAR – TOE AND CAMBER ANGLE

4.4.1 Toe Angle

Analysis: Power Off Straight Line Analysis (posl)

Exaggerated value (angle = 5, -5, -10)

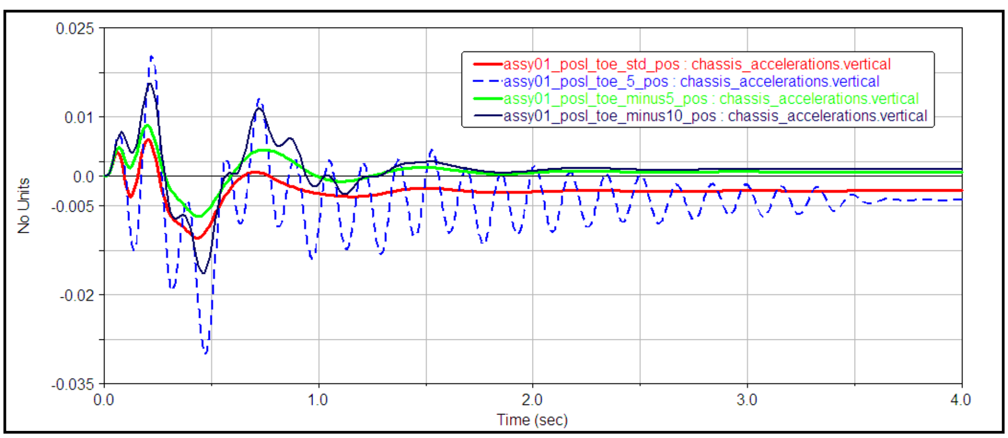


Figure 4.4-1: Vertical chassis acceleration vs. time for toe angle value analysis

Optimized value (angle = -1.5, -2, -5)

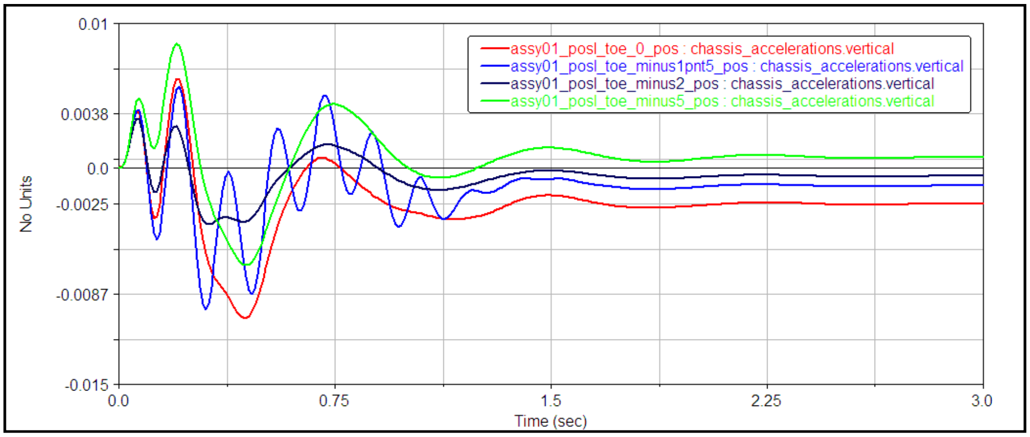


Figure 4.4-2: Vertical chassis acceleration vs. time for toe angle value analysis (optimized)

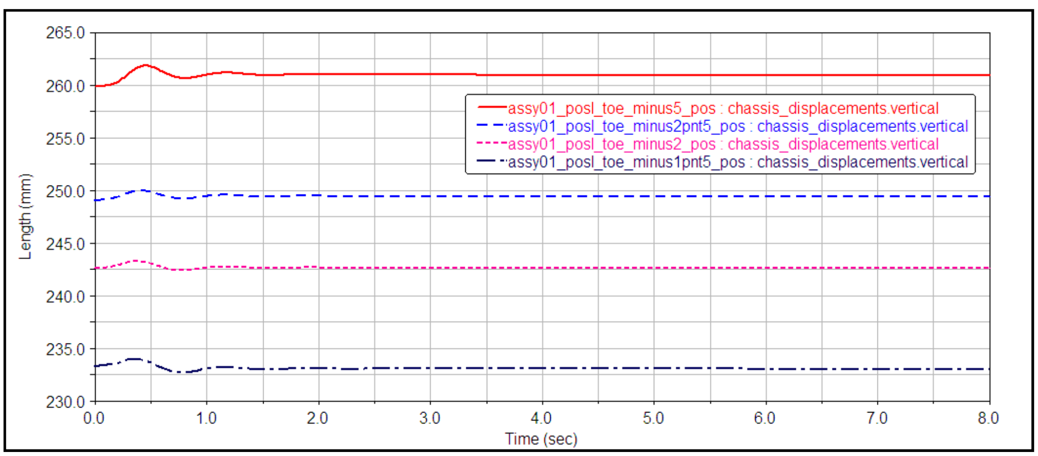


Figure 4.4-3: Vertical chassis displacement vs. time for toe angle value analysis (optimized)

4.4.2 Camber Angle

Analysis: Single Lane Change Analysis (slc)

Exaggerated value (angle = -5, 5, 10, 15)

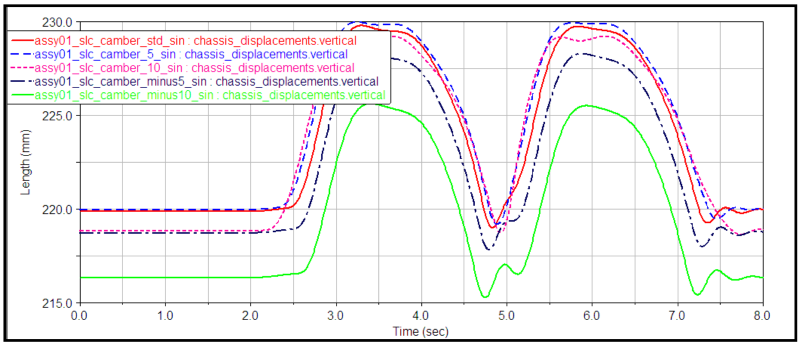


Figure 4.4-4: Vertical chassis displacement vs. time for camber angle value analysis

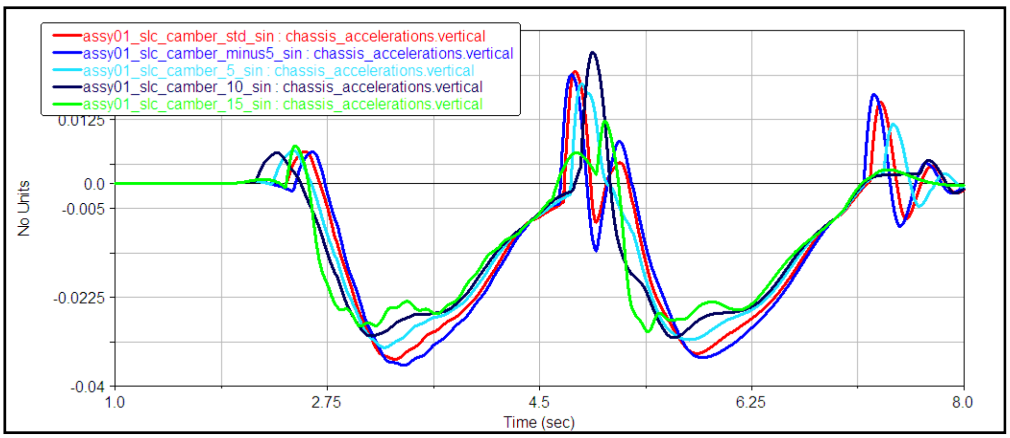


Figure 4.4-5: Vertical chassis acceleration vs. time for camber angle value analysis

Optimized value (angle = 7, 9, 10)

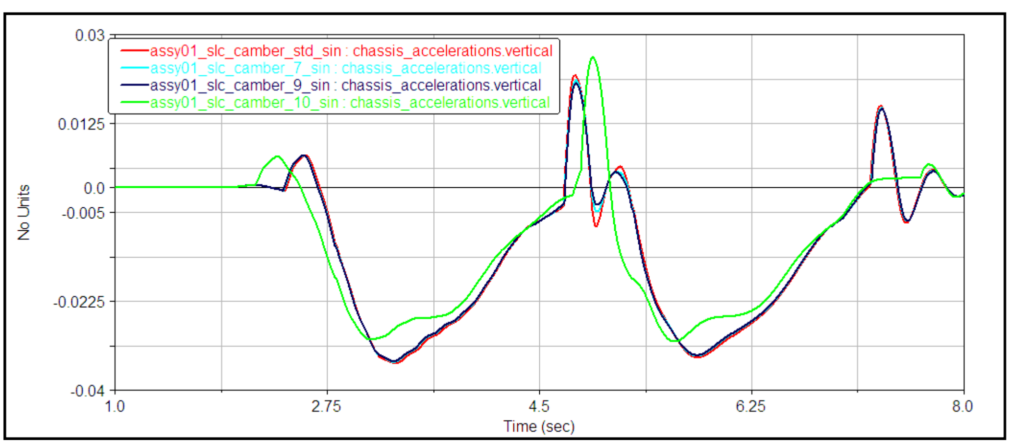


Figure 4.4-6: Vertical chassis acceleration vs. time for camber angle value analysis (optimized)

4.5 DISCUSSION

4.5.1 Spring Stiffness k (reference = 25,000N/m)

From the obtained plot of chassis accelerations vs. time as result, it is clear that the range of test values would be between 1,000N/m until 25,000N/m. After several analyses being carried out, it is found that the optimum values of k in this case are from 10,000N/m until the reference value (25,000N/m). All values within that range of values are identified to have a close behavior (such as the amplitude and plot smoothness) with the reference value. Additional analysis by introducing value of 30,000N/m is found to give a relatively bad result compared to the reference, whereby it produces much greater amplitude value as well as higher frequency, which will cause serious injury to passenger and perhaps serious damage to the suspension system as well.

4.5.2 Damping coefficient c (reference = 2,500Ns/m)

The upper and lower limits of testing parameters are 10,000Ns/m and 100Ns/m respectively, multiplication of 4 times greater and smaller compared to reference value. It is found that the optimized values are within the range of 2,500Ns/m until 10,000Ns/m. A decreasing value away from reference would be very lean in absorbing the effect of wobbling vehicle, as proven by higher frequency obtained from the plot. As a result, the vehicle tends to accelerate vertically at very high frequency and amplitude, which is not good.

4.5.3 Toe Angle (reference = 0 angle degree)

Exaggerated values for toe angle analyses are 5, -5 and -10 degree of angle. Negative value indicates the tires is aligned outer of vehicle body individually towards the front end of vehicle, as if the tires configuration were looked like ‘diffused’ if viewed at driver seating position, while the opposite is true for the opposite condition (positive toe angle). From results obtained it can be seen that as the negative sign values are approaching reference value, the plots are likely to follow the behavior of standard value. Hence, the optimized values are set to be in the range of -5 until 0 degree.

4.5.4 Camber Angle

For the case of camber angle, it is not recommended to carry out POSL analysis. This is because camber angle is associated with cornering event of vehicle, hence it would not be proper to executed POSL analysis since it doesn't involve any cornering to take place (steering angle = 0). The analysis is replaced with SLC instead, which required vehicle cornering.

From the obtained plot results, it can be observed the behaviors of outputs plots are as follows:

Table 4.5a: Observed behavior of camber angle plots for SLC analysis

No.	Parameter value in angle degree (color)	Characteristic
1	Reference, 0 (red)	- As benchmark.
2	-5 (blue)	- Tend to follow reference, but with larger amplitude. - Less good than reference.
3	5 (sea blue / turquoise)	- Smoother curve shape than reference. - Smaller amplitude than reference. - Better than reference.
4	10 (black)	- Smaller amplitude than reference. - transition from one peak (or valley) to another not too smooth.
5	15 (green)	- Amplitude much smaller than reference. - Transition less smooth, not good.

CHAPTER 5

CONCLUSION

SIMULINK is flexible and versatile software in performing lumped parameter modeling in this suspension system. With respect to the suspension system, it can be concluded that the driver does experience vibration during normal operations under passive suspension system, but may not be as much as with respect to the sprung mass or unsprung mass, but significant enough to cause an effect on driver's health.

It is hope that from the analysis which had been conducted by this research, further understanding of a McPherson front suspension system installed for Gen-2 Proton model can be developed. By conducting the feasible analytical and kinematic analysis of the simulation, the behavior of the system can be observed for further reference and analysis purposes.

REFERENCES

[1] Vu Trieu Minh, July 2008, *Vehicle Suspension and Driving Stability*, Universiti Teknologi PETRONAS, Tronoh, Perak

[2] Mohamed Rehan, M.S., Yusuf, M. and Abd. Rahman, R., *Parametric Studies on Tramcar Suspension System*, Thesis, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, Johor

[3] Antony, K., Jan 2008, <http://www.mathworks.com/matlabcentral/fileexchange/18410>

[4] William Harris, <http://auto.howstuffworks.com/car-suspension.htm>

[5] Ball, Jeffrey K. 2004, Automotive Engineering Fundamentals, Warrendale, PA, SAE International

[6] James E. Duffy, 2004, Modern Automotive Technology, The Goodheart-Willcox Company, Tinley Park, Illinois

[7] William F., Douglas L. Milliken, 1995, Race Car Vehicle Dynamics, Warrendale, PA, SAE International