# DESIGN AND ANALYSIS OF THE SUSPENSION SYSTEM FOR SINGLE SEATED EDUCATIONAL RACING VEHICLE

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



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**2017**

# DECLARATION

I declare that this project report entitled "Design and Analysis of The Suspension System For Single Seated Educational Racing Vehicle" is the result of my own work except as cited in the references.

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Date : ________________________________

**SUPERVISOR'S DECLARATION**

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: DR. FUDHAIL BIN ABDUL MUNIR

Date : ______________________

# DEDICATION

I am hereby is dedicated this report to my beloved parents, Zaharuddin Bin Zakaria and Zaiton Binti Hamid who always keep support and encourage me when I needed them the most. Not to forget to my siblings, my friends and to my supervisor, Dr. Fudhail Bin Abdul Munir for supporting me throughout this project and their understanding in the way I am.

# ACKNOWLEDGEMENT

First of all, in this great opportunity, I would like to thank to Allah for providing me a great health and strength throughout this final year project. My warm thanks to whom had helped and guide me during my study period in Universiti Teknikal Malaysia Melaka (UTeM) and also whom had given me an advice and encourage me throughout my studies in UTeM.

I would like to express my deepest appreciation to my supervisor, Dr. Fudhail Bin Abdul Munir for giving me the opportunity to complete this final year project. It is a real eye opener on how to execute this type of project and to be able to learn from nothing to almost anything is simply thrilling and incomparable.

And I also would like to express millions of thanks to my parents, relatives, my friends and those who are involved in this project either directly or indirectly for helping me in completing this project.

Once again, a big gratitude to my supervisor for giving me the heads up on how to properly do the project and giving me step-by-step on how to even run the project. All those criticism and views from my presentations were hopefully used to improve myselves throughout the entire project and I hope to fulfil the specification and needs of this subject as I instructed to do so.

# ABSTRACT

The suspension system is one of the most important and basic systems in a vehicle. The purpose of the vehicle suspension system is to maximize the friction between the road surface and the tires to provide the stability of the steering and also a good handling of the vehicle. In this project, the suspension was designed by using a CATIA V5 software for the single seated racing vehicle. In case of the racing vehicle, the most suitable suspension system is a push rod suspension system because of the characteristic of the system is convenient with the racing vehicle. Furthermore, the designed suspension will be analyses with a Von Misses analysis and Translational Displacement analysis in a CATIA software. The parameters of the suspension are also considered in this project while running an analysis of the suspension by using MATLAB Simulink software to find a pitch body and roll dynamic of the vehicle. The objective of this projects is to design a low-cost suspension system and perform a stress analysis on the designed suspension system. Then, to analyses the behavior of the suspension in the pitch and roll dynamic of the vehicle.

# *ABSTRAK*

*Sistem suspensi adalah salah satu sistem yang paling penting dan asas sistem di dalam kenderaan. Tujuan sistem suspensi kenderaan ini adalah untuk memaksimumkan geseran antara permukaan jalan dan tayar untuk kestabilan stereng dan juga pengendalian kenderaan yang baik. Dalam projek ini, sistem suspensi telah direka dengan menggunakan perisian CATIA V5 untuk kegunaan kereta lumba. Untuk kereta lumba, sistem suspensi yang paling sesuai adalah sistem "pushrod" kerana ciri-ciri sistem ini sesuai dengan kenderaan yang laju. Tambahan pula, suspensi yang direka akan dianalisis dengan "Von Misses" analisis dan "Translational Displacement" analisis dalam perisian CATIA. Parameter suspensi juga diambil kira ketika menjalankan analisis suspensi dengan menggunakan perisian MATLAB Simulink untuk mencari "pitch body" dan "roll dynamic" kenderaan. Objektif projek ini adalah untuk mereka-bentuk sistem suspensi kos rendah dan membuat analisis tekanan pada sistem suspensi yang direka. Dan seterusnya, untuk menganalisis kelakuan suspensi itu dalam kondisi "pitch body dan "roll dynamic" kenderaan.*

# TABLE OF CONTENT


| CHAPTER | CONTENT | PAGE |
|---|---|---|
| | **DECLARATION** | ii |
| | **SUPERVISOR'S DECLARATION** | iii |
| | **DEDICATION** | iv |
| | **ACKNOWLEDGEMENT** | v |
| | **ABSTRACT** | vi |
| | ***ABSTRAK*** | vii |
| | **TABLE OF CONTENT** | viii |
| | **LIST OF FIGURES** | x |
| | **LIST OF TABLE** | xiii |
| | **LIST OF ABBREVIATIONS** | xiv |
| **CHAPTER 1** | **INTRODUCTION** | |
| | 1.1 Background | 1 |
| | 1.2 Problem Statement | 5 |
| | 1.3 Objective | 6 |
| | 1.4 Scope of Project | 6 |
| **CHAPTER 2** | **LITERATURE REVIEW** | |
| | 2.1 History Suspension System | 7 |
| | 2.2 Classification of The Suspension System | 9 |
| | 2.2.1 The Dependent Suspension System | 9 |
| | 2.2.2 The Independent Suspension System | 10 |
| | 2.3 Type of Suspension System | 14 |
| | 2.3.1 Passive Suspension System | 15 |
| | 2.3.2 Semi-Active Suspension System | 16 |
| | 2.3.3 Active Suspension System | 17 |

| | | |
|---|---|---|
| **CHAPTER 3** | **METHODOLOGY** | |
| | 3.1 Introduction | 19 |
| | 3.2 CATIA V5 Software | 21 |
| | 3.2.1 Component Design Process | 21 |
| | 3.2.1.1 Suspension Arms | 21 |
| | 3.2.1.2 Connecting Part | 24 |
| | 3.2.1.3 Rockers | 26 |
| | 3.2.1.4 Shock Absorber | 27 |
| | 3.2.2 Assembly design | 28 |
| | 3.3 MATLAB Software | 29 |
| | 3.3.1 Modelling Assumption | 29 |
| | 3.3.2 Vehicle Suspension System | 29 |
| | 3.3.3 Schematic Model of Vehicle | 30 |
| | 3.3.4 Mathematical Model of Vehicle | 31 |
| | 3.3.5 Parameter for Mathematical Model | 35 |
| | 3.3.6 MATLAB Simulation | 36 |
| | 3.3.7 Ride Model Simulink | 37 |
| **CHAPTER 4** | **RESULT AND ANALYSIS** | |
| | 4.1 Introduction | 44 |
| | 4.2 CATIA Software | 44 |
| | 4.3 MATLAB Software | 46 |
| | 4.3.1 Pitch Body | 46 |
| | 4.3.2 Roll Dynamic | 54 |
| **CHAPTER 5** | **CONCLUSION AND RECOMMENDATION** | |
| | 5.1 Conclusion | 63 |
| | 5.2 Recommendation | 64 |
| | **REFERENCES** | 65 |
| | **APPENDICES** | 70 |

# LIST OF FIGURES


| FIGURE | TITLE | PAGE |
|---|---|---|
| 1.1 | Double wishbone suspension | 4 |
| 1.2 | MacPherson strut suspension | 5 |
| 2.1 | Stout-Scarab | 8 |
| 2.2(a) | The dependent suspension system front view | 10 |
| 2.2(b) | The dependent suspension system side view | 10 |
| 2.3 | The independent suspension system | 10 |
| 2.4 | Macpherson strut suspension system | 11 |
| 2.5 | Double wishbone suspension system | 13 |
| 2.6 | Multi-link suspension system | 14 |
| 2.7 | Damping compromise for passive dampers | 15 |
| 3.1 | Flow chart of the project | 20 |
| 3.2 | Front lower arm suspension | 22 |
| 3.3 | Front upper arm suspension | 23 |
| 3.4 | Rear lower arm suspension | 23 |
| 3.5 | Rear upper arm suspension | 24 |
| 3.6 | Front connecting part | 25 |
| 3.7 | Rear connecting parts | 25 |
| 3.8 | Rockers | 26 |
| 3.9 | Shock absorber | 27 |
| 3.10 | Front assemble suspension | 28 |

| | | |
|---|---|---|
| 3.11 | Rear assemble suspension | 29 |
| 3.12 | Quarter car ride model | 30 |
| 3.13 | Overview of MATLAB software R2016a | 31 |
| 3.14 | Simulink library and browser | 32 |
| 3.15 | Pitch Simulink of the equation 1 | 38 |
| 3.16 | Pitch Simulink of the equation 2 | 38 |
| 3.17 | Pitch Simulink of the equation 3 | 39 |
| 3.18 | Pitch Simulink of the equation 4 | 39 |
| 3.19 | Subsystem of the Simulink for all four equations | 40 |
| 3.20 | Pitch Simulink of the equation 1 | 40 |
| 3.21 | Pitch Simulink of the equation 2 | 41 |
| 3.22 | Pitch Simulink of the equation 3 | 41 |
| 3.23 | Pitch Simulink of the equation 4 | 42 |
| 3.24 | Subsystem of the Simulink for all four equations | 42 |
| 3.25 | Simplified subsystem of the Simulink for all four equations | 43 |
| 4.1 | Von Mises analysis | 45 |
| 4.2 | Translational displacement analysis | 45 |
| 4.3 | Graph of Pitch against Time with Cs = 700 Ns/m | 47 |
| 4.4 | Graph of Pitch against Time with Cs = 900 Ns/m | 47 |
| 4.5 | Graph of Pitch against Time with Cs = 1200 Ns/m | 48 |
| 4.6 | Graph of Pitch against Time with Ks = 16200 N/m | 49 |
| 4.7 | Graph of Pitch against Time with Ks = 18500 N/m | 49 |
| 4.8 | Graph of Pitch against Time with Ks = 22300 N/m | 50 |
| 4.9 | Graph of Vertical acceleration against Time with Cs = 700 Ns/m | 51 |
| 4.10 | Graph of Vertical acceleration against Time with Cs = 900 Ns/m | 51 |

| | | |
|---|---|---|
| 4.11 | Graph of Vertical acceleration against Time with Cs = 1200 Ns/m | 52 |
| 4.12 | Graph of Vertical acceleration against Time with Ks = 16200 N/m | 53 |
| 4.13 | Graph of Vertical acceleration against Time with Ks = 18500 N/m | 53 |
| 4.14 | Graph of Vertical acceleration against Time with Ks = 22300 N/m | 54 |
| 4.15 | Graph of Roll against Time with Cs = 700 Ns/m | 55 |
| 4.16 | Graph of Roll against Time with Cs = 900 Ns/m | 55 |
| 4.17 | Graph of Roll against Time with Cs = 1200 Ns/m | 56 |
| 4.18 | Graph of Roll against Time with Ks = 16200 N/m | 57 |
| 4.19 | Graph of Roll against Time with Ks = 18500 N/m | 57 |
| 4.20 | Graph of Roll against Time with Ks = 22300 N/m | 58 |
| 4.21 | Graph of Vertical acceleration against Time with Cs = 700 Ns/m | 59 |
| 4.22 | Graph of Vertical acceleration against Time with Cs = 900 Ns/m | 59 |
| 4.23 | Graph of Vertical acceleration against Time with Cs = 1200 Ns/m | 60 |
| 4.24 | Graph of Vertical acceleration against Time with Ks = 16200 N/m | 61 |
| 4.25 | Graph of Vertical acceleration against Time with Ks = 18500 N/m | 61 |
| 4.26 | Graph of Vertical acceleration against Time with Ks = 22300 N/m | 62 |

# LIST OF TABLE


| TABLE | TITLE | PAGE |
|---|---|---|
| 3.1 | Parameters involved in vehicle model | 35 |

## LIST OF ABBEREVATIONS

| | |
|---|---|
| CATIA | Computer Aided Three-dimensional Interactive Application |
| MATLAB | Matrix Laboratory |
| CAD | Computer Aided Design |
| CAM | Computer Aided Manufacturing |
| CAE | Computer Aided Engineering |
| COG | Center Of Gravity |

# CHAPTER 1

# INTRODUCTION

## 1.1 BACKGROUND

Suspension system is among the most important and basic systems in a vehicle. The purpose of vehicle suspension system is to augment the friction between the surface of road and the tires to provide the stability steering and good handling of the vehicle [1]. The suspension also helps to support the vehicle weight, keeping the vehicle tires in contact with the road and maintaining the correct vehicle ride height [2]. In order to achieve the stability and vehicle ride comfort, there were three principles must be resolved which is road isolation, road handling and cornering. Numerous studies have been conducted to achieve the stability and ride comfort in the vehicle.



In addition, there are three elements that consist in a vehicle suspension system, which are wishbone, spring and shock absorber. The spring is like an elastic object that used to save a mechanical energy. They can twist, pulled or stretched by some of the force and then can return back to their original form when the force is released. A coil spring is made from a special wire of a single length, which is heated and wound on a former, to produce the required shape. The load carrying ability of the spring relies on upon the width of the wire, the general width of the spring, its shape, and also the range of the coils [3]. Shock absorbers are generally in charge of the undesirable structure-borne noise in cars. The part of the shock absorbers is to give a superior handling, comfort and the wellbeing of the car during travel by means of damping the relative movement between wheel and body of the car. [4]

The purposes of these elements are to filter and transmit the force exerted by the vehicle body to the road. The spring element is important as it carries the body mass and isolates the vehicle from uneven road surfaces that subsequently contributes to the drive comfort [5]. In addition, the damper system in the vehicle also contributes to safety as it absorbs the damping of the body wheel oscillations.

Generally, the suspension systems are categorized into two groups, dependent and independent system. A suspension connected to a rigid axle between the left and the right of the wheels is called a dependent suspension since the vertical movement of one wheel is delivered to the opposite wheel in these cases. The major disadvantage of this rigid steer able axle is their susceptibility to tramp-shimmy steering vibrations [6]. The independent suspension system allows the left and right wheel to move without affecting the other's motion. Nearly all the passenger cars and light trucks use an independent front suspension because of the advantages in providing room for the engine and also for the better resistance to steering induced vibrations. There are many forms and designs of independent suspensions. However, double wishbone and MacPherson strut suspensions are perhaps the simplest and most commonly used designs.

The double wishbone suspension as shown in Fig.1.1 is also known as the short long arm or double A-arm suspension. Each wishbone or arm has two mounting points that attached to the chassis and is connected to the knuckle by spherical joint. The damper and coil spring system is put between the chassis and both of the control arms to smoothen the vertical development. For this situation, the guiding connection is appended to the tie-rod by utilizing a spherical joint and the tie-rod is associated with the steering rack by a universal joint. The double wishbone is utilized as a part of an elite of the cars and SUVs because of its better kinematic reaction over different suspensions [7].

Various methods for the designing and modelling of the double wishbone suspensions are exist in the literature. The author has made use of the displacement matrices and the loop-closure constraints to synthesise and analyse these mechanisms by modelling the double wishbone suspension as a spatial RSSR-SS linkage [8]. Other reported methodologies, focused on the designing of the suspension system by optimising some particular suspensions performance indices, such as camber, caster, toe, and king-pin inclination [9-11]. However, in these works, the kinematic constraint equations are used solely in the formulating constrained optimisation problems. According to Wang et al. [12], the design of experiments module in the software Adams\View can be utilized to obtain the optimal values for the key design parameters of the suspension system by setting up the desired ranges for those design parameters.

Arikere et al. [11] are reported that the loop-closure equations for the double wishbone are derived by using lower A-arm angle as a surrogate input, while holding the steering input at a constant value. The kinematic analysis of the double wishbone is performed by using Euler angles that associated with the spherical joint that join the lower A-arm to the knuckle, as the surrogate input [13-14].

An American automotive engineer, Earle S. MacPherson has developed the MacPherson Strut suspension by using a strut configuration system [15]. Solid model of the suspension system is shown in Fig.1.2**.** One end of the strut is attached to the knuckle via the prismatic joint and the other end is connected to the chassis by revolute joint while the other end is connected to the chassis by spherical joint. The strut is also carries the spring and the damping elements. The steering link is attached to the tie-rod using the spherical joints, meanwhile the tie-rod is connected to the rack by universal joint. In spite of the fact that the MacPherson strut suspension system has less great kinematic characteristic contrasted with the double wishbone suspension [7]. It is sufficiently minimal to be good with the transversely mounted engines and is accordingly utilized broadly in front wheel drive cars.

A numerous technique for the kinematic analysis of MacPherson strut suspension system exist in the literature, a prominent one being [15]. Modelling the MacPherson strut suspension as a spatial mechanism, linearized approximate and non-linear position analysis is reported, based on the vector algebra. Later, a formulation in terms of the Cartesian coordinates of some defined a point in the links and also at the kinematic joints [16], whereas the constraint of the equations using Euler parameters is formed [17].

However, the focus of this project is on the double wishbone suspension design; it's a simple and widely used independent suspension system. This suspension assembly improves the drive comfort by making more stable ride while minimizes unnecessary vehicle movement during driving [18]. There two systems that usually used in the racing car such as Formula 1, Formula SAE, Formula Varsity, etc. The two systems are push rod and pull rod suspension system. Although, the system is quite the same but there is pro and cons for both systems.

Generally, racing cars tends to utilize push rod suspension system at the front of the car while pull rod system at the rear of the car [19]. However, a push rod suspension system will be design for both front and rear of the car. This is because a push rod system is more aerodynamically than the pull rod system and push rod system help the car maintains at the same level while pull rod system give a higher load on upper arm of suspension.



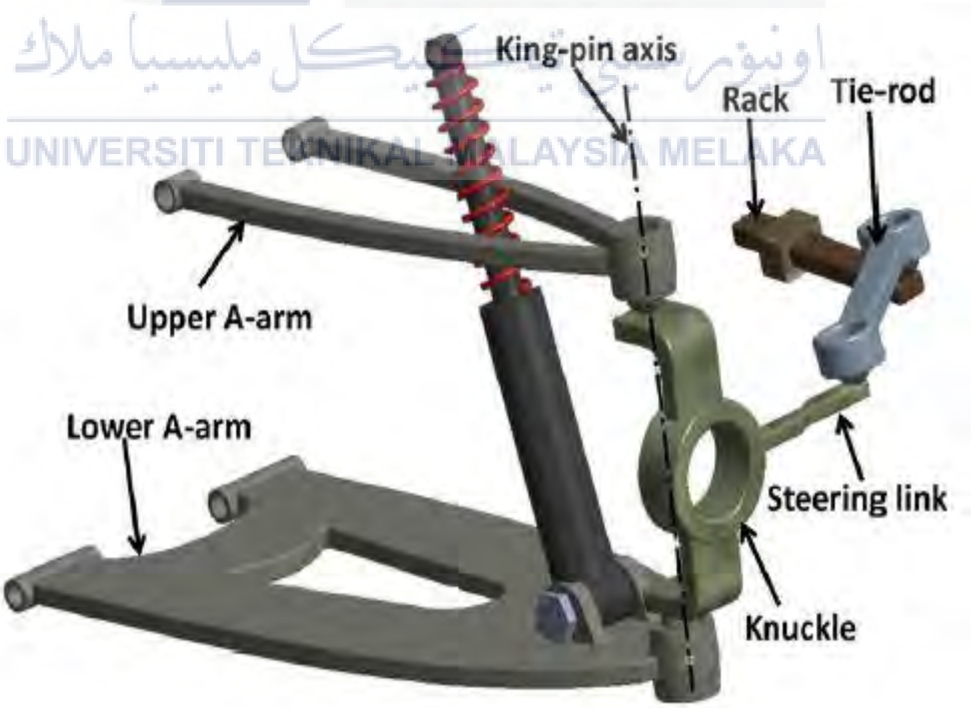


**Figure 1.1:** Double wishbone suspension [14]

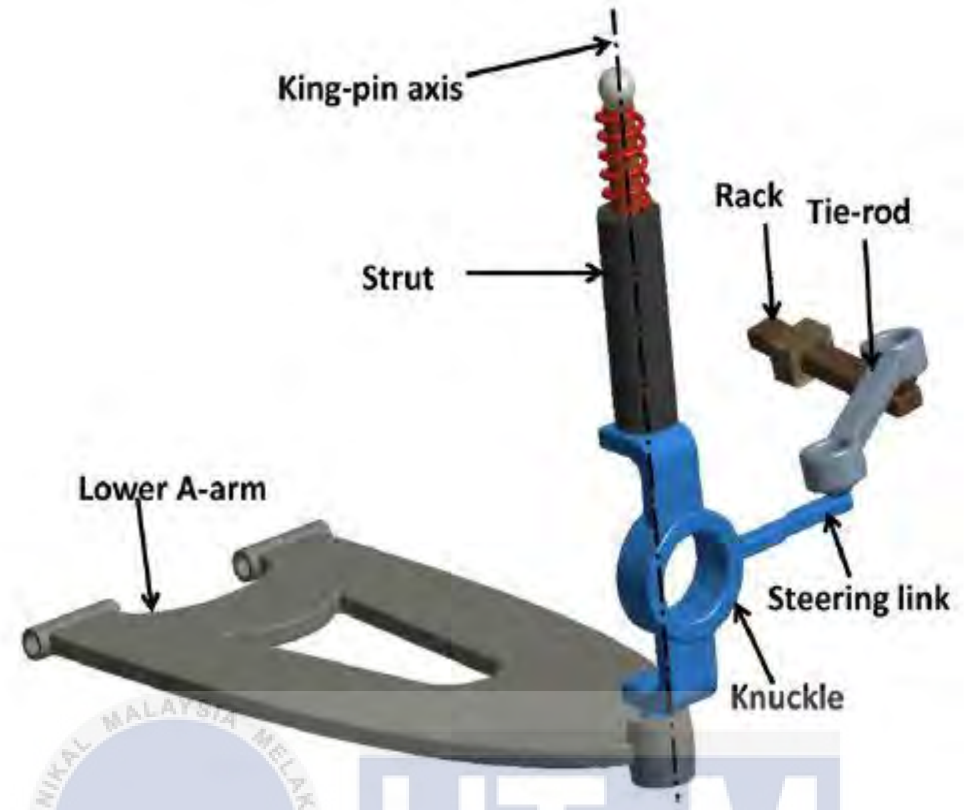


**Figure 1.2:** MacPherson strut suspension [14]



## 1.2 PROBLEM STATEMENT

The aim of this project is to design and analysis the suspension system for single seated racing car. This project focused on designing a double wishbone suspension system by using CATIA software for the single seated racing vehicle. The designing of the push rod suspension system will give the advantage for a racing car such as formula varsity or formula SAE. Hard cornering, braking or other road disturbance in the racing will affect the vehicle handling and stability. It is observed that by using this suspension system, the pitch and roll for the car is needed to be considered. So, to overcome this problem, the research had been made to design and analysis a low-cost push rod suspension system for a single seated racing vehicle.

## 1.3 OBJECTIVES

The objectives of this project are as follows:

1. To design a low-cost suspension system by using CATIA software for a single seated educational racing vehicle.
2. To perform a stress analysis on the designed suspension system.
3. To analyse the behaviour of the suspension in roll and pitch by using MATLAB Simulink software.

## 1.4 SCOPES OF PROJECT

The scopes of this project are as follows:

1. Focus on the design of the suspension system.
2. Stress analysis will be performed to determine the feasibility of the suspension system.
3. Analysis of the designed suspension in roll and pitch will be conducted in the second semester.

# CHAPTER 2

# LITERATURE RIVIEW

## 2.1 History Suspension System

The main suspension system has been intended for the light chariot of Ramses II around the year of 1296 B.C. Tragically, this suspension system was awful because of its shaky condition. Although, the Greeks have found numerous new standards with a specific end goal to take care of this issue, they were unable to find the best answer for good suspension system. Around then, one suspension system has been found that was truly agreeable for driving power and suspension. Nonetheless, there were issues for that design in which it decreases the paces and the rapid wear of the component and should be changed frequently. By that time, history saw a rapid evolution of suspension system design with a few issues have been found and distinguished. Regardless of the positive improvement of suspension system design, there still issues relating to the system, for example, mechanical noise coming from an iron chain suspension and also a portion of the passenger must be safe from the sea sickness.

A modern automobile suspension system has been introduced by a young man by the name of William Brush in 1906 [20]. This is because of the pile up at unpaved street with the speed of 30 mph which is included his sibling. The impression is that the car's right wheel began shimmy savagely and the whole car vibrated furiously. Brush has outlined a suspension system for the Brush Two-Seat Runabout car model. It highlighted a revolutionary suspension system that consolidated two developments that never assembled together. Front coil springs and devices to every wheel that damped spring bounced (shock absorbers) mounted on a flexible hickory axle.

Some European car producers had tried coil springs, with Gottlieb Daimler in Germany being the main type. In any case, most manufacturers stood quick with leaf springs. They were less expensive and by basically including leaves or changing the shape from full elliptic to the three quarter or half elliptic, the spring could be made to support varying weights.

Following a couple of years, General Motors, Chrysler, Hudson and others reintroduced coil spring front suspension. This time with every wheel sprung independently. In that year, most car began utilizing hydraulic shock absorbers and balloon (low pressure) tires. Coupling solid front axle with shock absorber and these tires truly exasperated front-end shimmy. Suspending every wheel separately diminished the impact of spring bounce [21].

Air suspension, which Lincoln ballyhooed for some model in 1984 was presented in 1990 by the Cowey Motor Works of Great Britain. It didn't function admirably on the grounds that it leaked. The primary practical air suspension was produced by Firestone in 1933 for a test car called the Stout-Scarab [22]. This was a rear engine vehicle that utilized four rubber treated roars as a part of place of conventional springs. Air was provided by small compressor connected to each bellow.

**Figure 2.1**: Stout-Scarab

## 2.2 Classification of the suspension system

The suspension system is always derived by some mechanical way. Generally speaking, the design of the suspension systems is classification in two groups [23-24]:

i. Dependent suspension system.
ii. Independent suspension system.

Each group can be functionally quite different and they are studied and discussed accordingly [25-26]. Recently, both suspension systems can be found on ordinary vehicles and commercial vehicles.

### 2.2.1 The dependent suspension system

The dependent suspension system is otherwise called a solid axle suspension, when both wheel, left and right are mounted to the same solid axle as in Figure 2.2. For this situation, any development of any wheel will be transmitted to the opposite wheel making them to camber together. Solid drive axles are typically utilized on the rear axle of numerous passenger car, trucks and on the front axle in numerous four-wheel drive vehicles.

The benefit of the solid axles is viewed as the camber angle which is not influenced by rolling of the vehicle body. In this manner, produce a little camber in cornering with the exception of that which arises from slightly more compression of the tires on the outside of the turn. Besides, wheel alignment is readily kept up, which contribute to minimize the tire wear. While for the disadvantage of the solid steerable axles is their susceptibility in shimmy steering vibration, overwhelming mass, and etc. The most sorts of solid axles are Hotchkiss, Four link and De Dion.

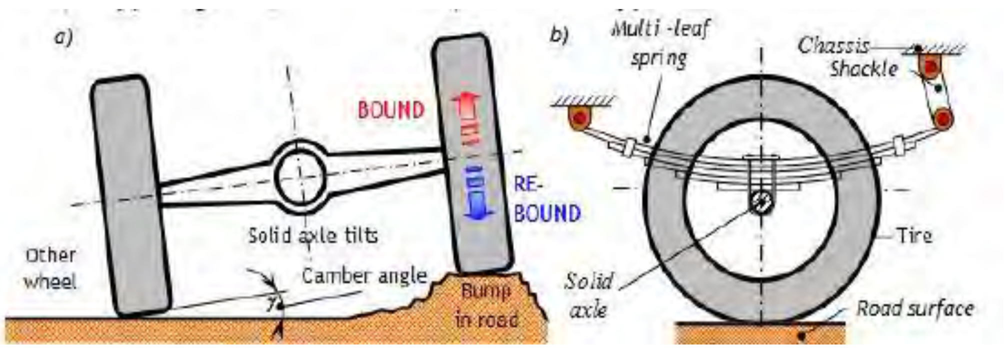


**Figure 2.2**: The dependent suspension system; **a)** front view **b)** side view [30]

### 2.2.2 The independent suspension system

The independent suspension system allows one wheel to move upward and downward with a minimum effect on the other wheel as in Figure 2.3. Mostly, the passenger cars and light truck use independent front suspension system because it provides much more space for installing vehicle engine than a dependent suspension system. Apart from that, independent suspension system likewise permits a great deal more displacement of wheel, better resistance in steering vibration like wobble and shimmy as well as offer a higher performance in passenger comfort. As the disadvantage of the independent suspension system, it can be viewed as the complexity of the design and manufacturing cost because of expanding of the quantity of parts.

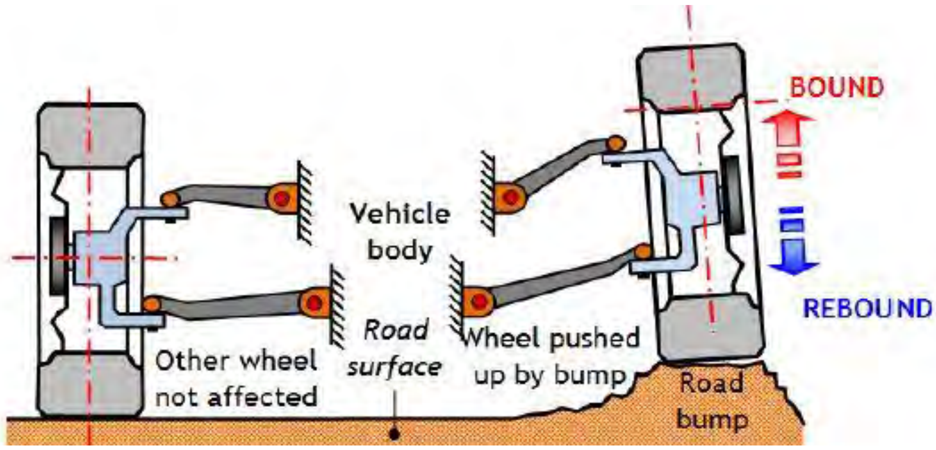


**Figure 2.3:** The independent suspension system [30]

Throughout the years, many types of independent suspension system have been attempted to developed such as MacPherson, double wishbone, multi-link, trailing arm and swing axle. There are a hefty portion of them have been disposed of for a different reason, with only a fundamental concept, Macpherson, double wishbone and multi-link suspension system have discovered application in many types of vehicles.

The MacPherson strut suspension system consists of one single control arm and a strut assembly which is spring and shock absorber which allows the tire and wheel to move upward and downward. The major components of the system are shown in Figure 2.4. It may be used on both the front and rear axle of the vehicle. This suspension system design allows reducing number of parts, lower unsprung mass as well as smooth driving comfort.

MacPherson strut suspension system is a shrewd trade off that grants sensible execution or customization such as double wishbones or multi-link suspension system. Also, this kind of suspension system requires sufficient vertical space and a solid top mount.



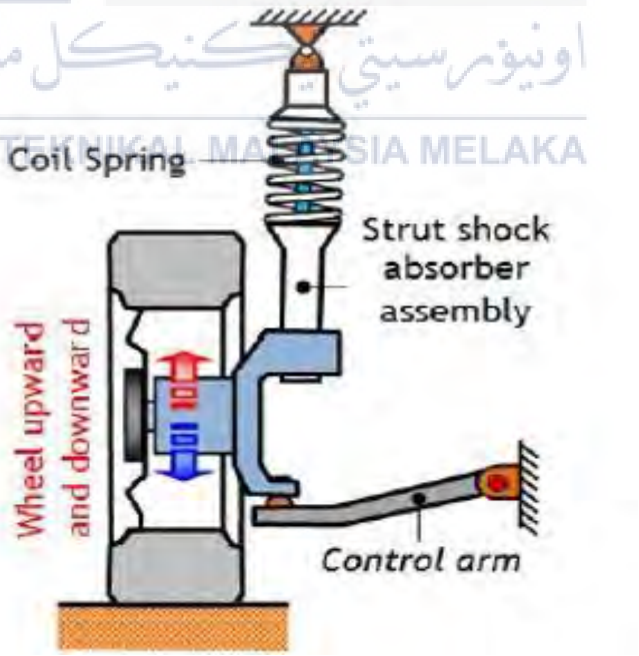


**Figure 2.4:** Macpherson strut suspension system [30]

The double wishbone suspension system is frequently called "A-arms" in the US and "Double Wishbone" in the UK. The double wishbone suspension system is typically connected to a luxury vehicles and a sport car in view of their design of versatile and kinematic parts allow to give an ideal compromise among handling and comfort [27]. The double wishbone utilized two lateral control arms to hold the wheel from tilting with the suspension activity (Figure 2.5). The upper and the lower control arms typically are with unequal length where the acronym SLA (short long arm) gets its name. During design of this type of suspension, it is required careful refinement of suspension geometry in order to get great performance [25-28].

There are a couple favourable circumstances of the double wishbone suspension system. For instance, the essential preferred standpoint is increment the negative camber as an eventual outcome of a vertical movement of the upper and lower arms. This legitimize to the stability performance for the vehicle as the tires on the outside keep up better contact with the road surface during cornering. The weakness of the double wishbone contrasted with the MacPherson strut suspension system gets to be in the complexity of the design, generation cost, and increment the amount of parts.

The proposed suspension system for the single seated racing vehicle is a double wishbone suspension system which include the pushrod suspension system. pushrod style suspension is used so that the spring and stun get together and bell crank can be mounted at any position that will better dampen the bigger wheel loads that are seen amid hustling. This design also lessens the car's drag by moving the spring inboard the casing. The pushrod suspension also gives simple customizability because of the difference motion ratios that can be changed by conforming the separations that the pushrod and spring and shock assembly are mounted from the pivot point of the bell crank itself. Changing these mounting areas effectively changes the ratio of wheel movement to spring compression, whereby changing this ratio different handling characteristic can be seen

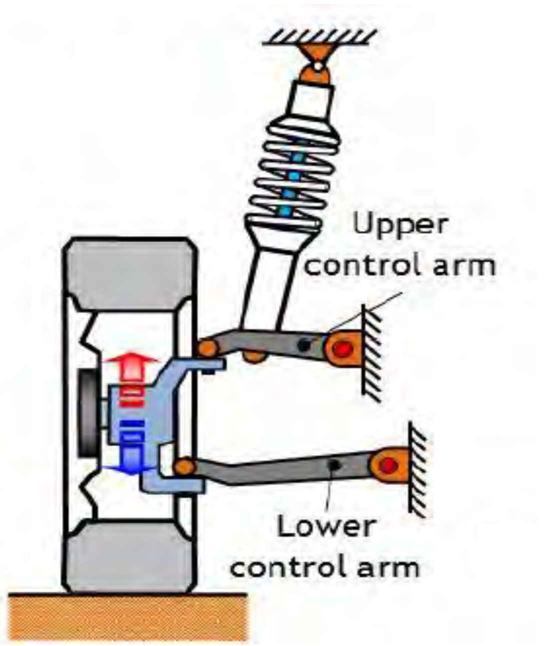


**Figure 2.5:** Double wishbone suspension system [30]



The multi-link suspension system belongs the group of the independent system. It is utilized on both the front and rear axles and is determined by refinement of the double wishbone. Utilize at least three lateral arms and at least one longitudinal arms, which don't require to being with equivalent length [29] (Figure 2.6). Recently, the multi-link suspension system appears to be the best independent system for vehicle since it offers the best compromises between comfort, stability and manoeuvrability [29]. Moreover, this suspension system permits vehicles to show signs of better performance compared with the other types of suspension system.

Most likely, multi-link would be great solution for outfit a terrain vehicle with a such system. The multi-link suspension system has advantage for the designer who empowers to change one parameter without affecting the whole assembly. This is a major difference compared with a double wishbone suspension system. By taking into a thought of every single good thing, the multi-link suspension system is more complex to the design and has higher manufactures cost. To show signs of better performance of the vehicle, the suspension geometry ought to be checked deliberately by adequate software.

As disadvantages of the multi-link suspension system, this is considered a basically fact that they are not allow enough the vertical movement of wheels in order to get a good performance and a characteristic of the terrain vehicles, it requires to be developed such a suspension system.

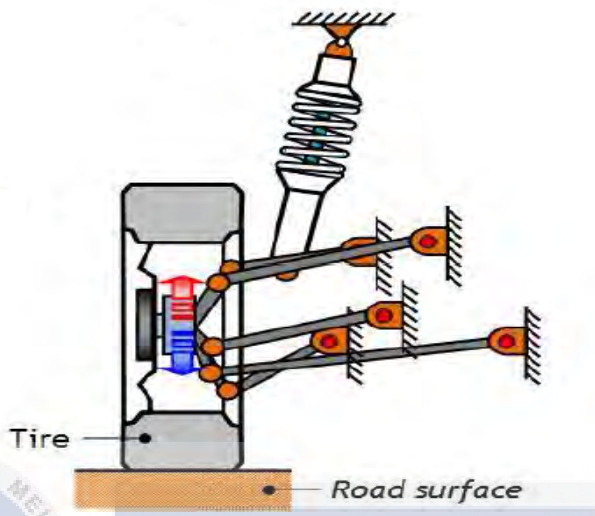


**Figure 2.6:** Multi-link suspension system [30]



## 2.3 Type of the suspension system

In a vehicle suspension system, there are different type of suspension system that can be connected to the particular vehicle. There are fundamentally three types of the suspension system, which is passive, semi-active and active suspension system. Every one of them have their own specialty in a suspension system for the vehicles.

### 2.3.1 Passive suspension system

Passive suspension system consists of spring and damper. Damper goes about as a vitality dissipating element while spring as an energy storing element. Since these two components don't add energy to the system this type of suspension system are called passive [31]. Passive suspension design is a trade-off between vehicle handling and ride comfort as shown in Figure 2.7. Parameters are by and large settled in passive suspension system, being chosen a specific level of compromise between a road handling, load carrying and load comfort [32].

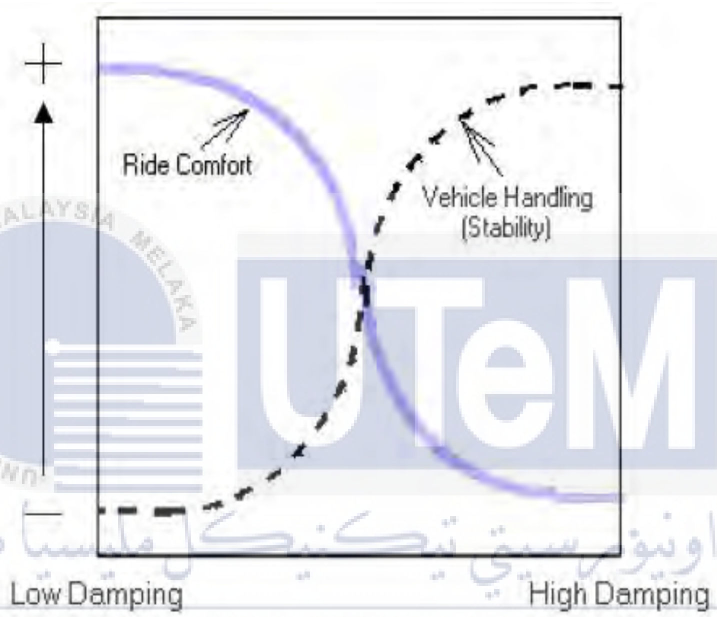


**Figure 2.7:** Damping compromise for passive dampers [33]

Problem in case of passive suspension system is that intensely designed damper will exchange a considerable measure of road input or throwing the car on unevenness of the road while in case of lightly designed damper suspension it will lessened the stability of the vehicle while alternating or changing the path or it might swing the car. Performance of the suspension system depends upon the road profiles [33]. In the passive suspension system, the ride comfort diminishes with the increasing in the spring stiffness. For an example, a good ride comfort for a soft spring and a terrible ride comfort is for a hard spring [34].

### 2.3.2 Semi-active suspension system

Semi-active suspensions were firstly introduced in early 1970s as an alternative to the costly, highly complicated and a power demanding active systems [35-36]. Similar work was performed by Rakheja and Sankar [37] and Alanoy and Sankar [38] in terms of active and semi-active isolators. A comparative study with passive system was carried out by Margolis [39] and Ahmad and Marjoram [40]. The most attractive feature of that work was that the control strategies were based on the only upon the measurement of the relative displacement and velocity.

A control scheme which is also known as skyhook damping based on the measurement of the absolute vertical velocity of the body of the car was proposed in the 1970s by Karnopp and is still employed in a number of variation [41]. Some authors are outlined a suspension in view of a biological, neuromuscular-like control system with a specific end goal to enhance the comfort [42]. Recently, Liu et al. [43], concentrated four different semi-active control systems in view of the skyhook and the balance control strategies.

In semi-active suspension system, the spring component is held yet the damper is replaced with a controllable damper which requires little measure of the external energy [44]. In semi-active suspension system, force actuator is utilized as a part of controller which control road irregularities and increase the comfort level. It works in closed looped control system. The advantage of the semi-active suspension system is improving the ride comfort than the active suspension system with the less amount of energy [31].



As far as the applications of the semi-active suspensions are concerned, they have been envisaged not only for the cars but also for other type of the vehicles. T. Ram Mohan Rao, et al. [31] and Rijumon, Murtaza, et al. [45] worked on skyhook control semi-active suspension system. It was observed that the skyhook control can accomplish more reduction resonant peak of the body mass than that of the passive suspension system. Dankan V. Gowda, et al [46] took a shot at suspension with PID controller. Omid Ghasemalizadeh, et al. [47] examined control philosophies with the application of the increasing speed driven damper and power driven damper. Amit A.

Hingane, et al. [48], worked on Bingham model for MR damper which is more to a fluid based.

### 2.3.3 Active suspension system

In an active suspension system, the passive force elements are replaced or assisted by a n active force elements. These elements are able to produce a force when required and act independent of the suspension condition [49]. Therefore, the trade-off between the ride comfort, suspension travel and the wheel load variations can be better resolved.

In addition, an active suspension system can be utilized in order to take out body roll during cornering [50]. As a result, the wheels can be arranged ideally as for the road both in case of encountering a bump and during the cornering [51]. The mentioned trade-off disappears and also an anti-roll bar is not required any longer. The complicated and space consuming suspension connection can be replaced with a compact and simple trailing arm suspension [52]. The compact suspension system allows for designing a smaller and lower car without affecting its interior spaces. This will lead to the lower air drag.



In case of the suspension system with the variable spring stiffness, this stiffness can be adjusted proportionally to the change of mass. As a consequence, the natural of the vertical frequency of the car’s body will not change and can be chosen at a frequency which is less uncomfortable for the human body. While in case of a passive suspension system, a significant portion of the available suspension travel is used to take care of static load variations and body roll caused by cornering. The active suspension system can take care of these variations by adjusting its stiffness. Therefore, a lower initial stiffness can be used which is favourable for the comfort level.

An active suspension introduces the possibility with alter the suspension setup to the type of driving circumstance and to individualize the handling of attributes and comfort level of the vehicle. The short trip to the supermarket may well be a bit bumpy but a long boring drive to the office should rather be more comfortable. Thanks to this, a possible passenger is able to serenely check his or her email, schedule of a meeting or a glance through the minutes of yesterday’s meeting without getting car sick.

# CHAPTER 3

# METHODOLOGY

## 3.1 Introduction

This chapter describes the methodology used in this project to obtain the data from the analysis that had been made at the designed suspension. The flow chart of the project is shown in Figure 3.1. To complete this project, two software had been used to make an analysis at the suspension which is CATIA V5 software and MATLAB/Simulink software.

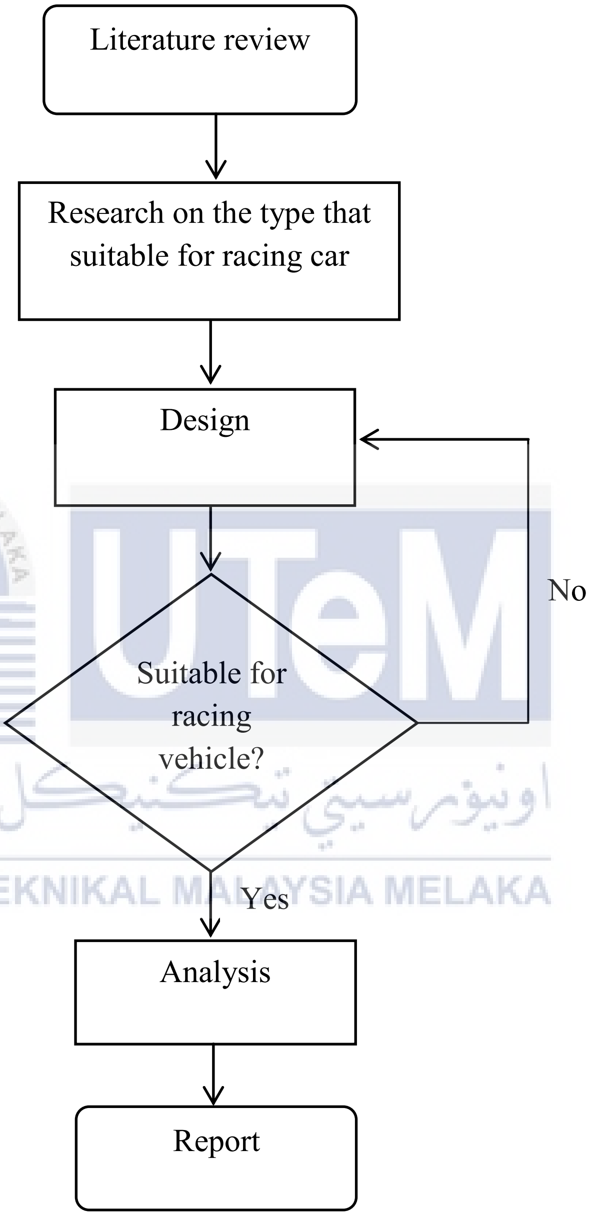


**Figure 3.1:** Flow chart of the project

## 3.2 CATIA V5 software

Computer Aided Three-dimensional Interactive Application (CATIA) is a multi-platform software suite for computer aided design (CAD), computer aided manufacturing (CAM), computer aided engineering (CAE) and 3D, developed by the French company. By using this software, the design of the suspension system managed to be done. The project design plan draws heavily on work and recommendations suggested by author's in the literature review with the final plan made up of the steps and ideas deemed most appropriate to the design of a single seated educational racing vehicle. This is providing an optimal pathway for the design and analysis made in this software.

### 3.2.1 Component design process

In this design process, the CATIA software had been used to make the part of the suspension system that consist of the lower arm, upper arm, connecting parts, rockers and shock absorber. This design is including both front and rear design of the suspension system.



#### 3.2.1.1 Suspension arms

The vehicle system is not just about the shock absorber and springs. These components are important, however there are other similarly vital suspension parts that complete the suspension system. Among them is the suspension arm which also called control arm, a triangular-shaped metal strut that connects the wheel to the vehicle's frame. Besides giving support to the vehicle's wheels, the control arm also performs a more essential role. The control arm allows the front wheels to change direction at whatever point the turn is make. Subsequently, this suspension part truly is something that can't drive appropriately without. There are basically three or four control arms in the vehicle, every unit conveying driving and brake torque. Because of its load-bearing and supportive functions, the control arm also commonly experiences early wear and tear.

For the control arm that had been design in this project, the rear and front suspension arm had a different design because of the design of the chassis. Apart from that, the power train system at the rear vehicle is one of the factor that there is different design between front and rear suspension. For this single seated racing vehicle, the pushrod suspension design had been chosen because of the benefits to the vehicle itself. All the designed part is shown in figures below:

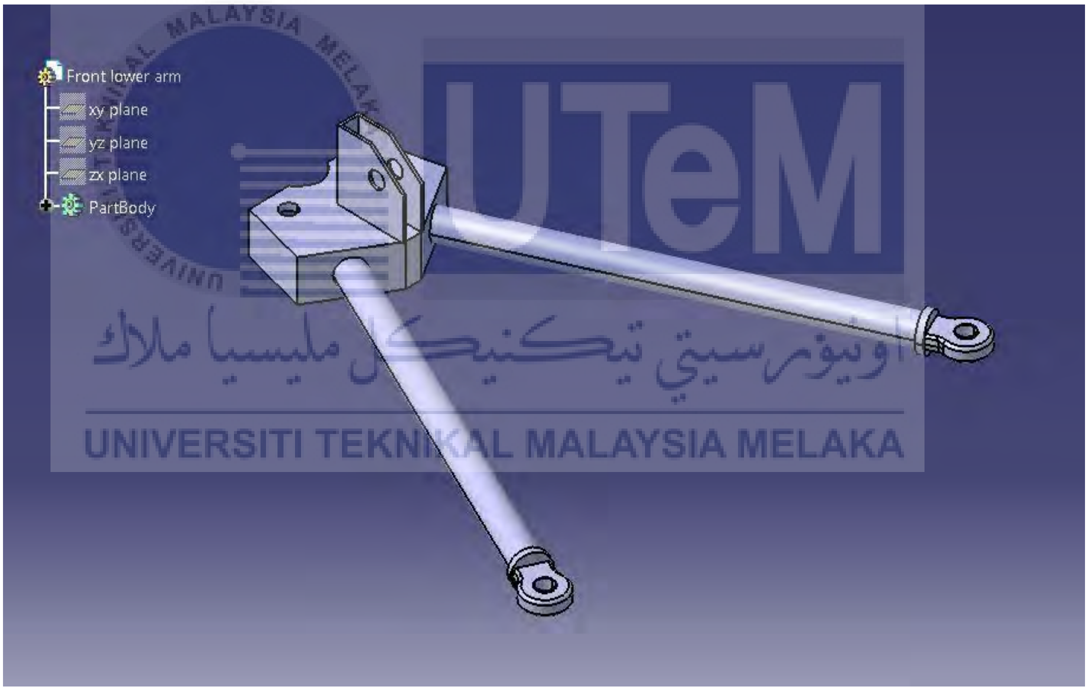


**Figure 3.2:** Front lower arm suspension

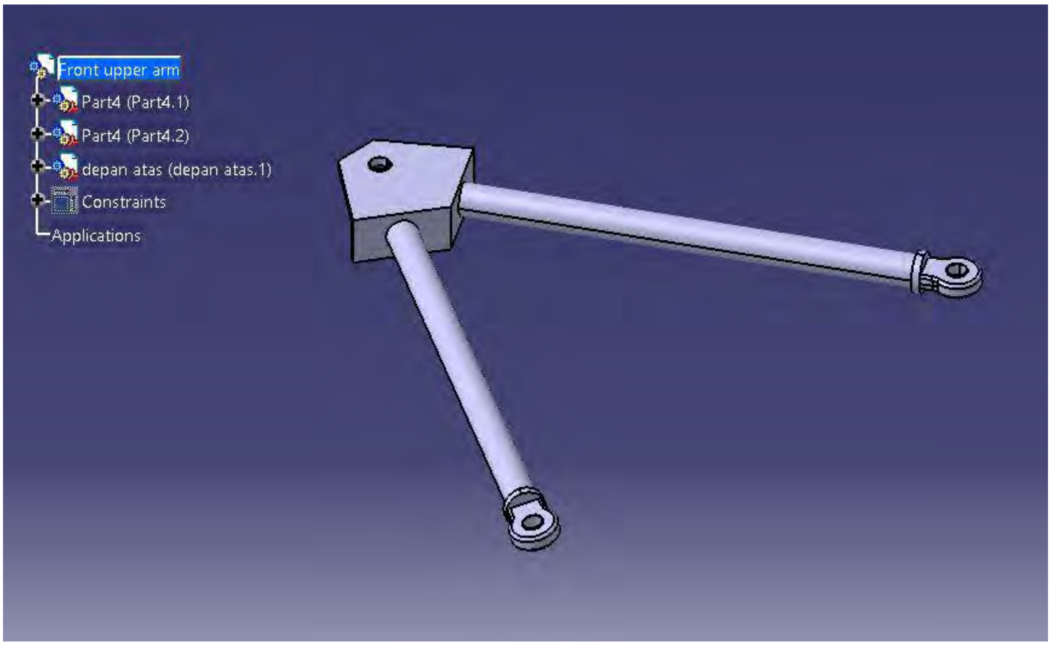


**Figure 3.3:** Front upper arm suspension

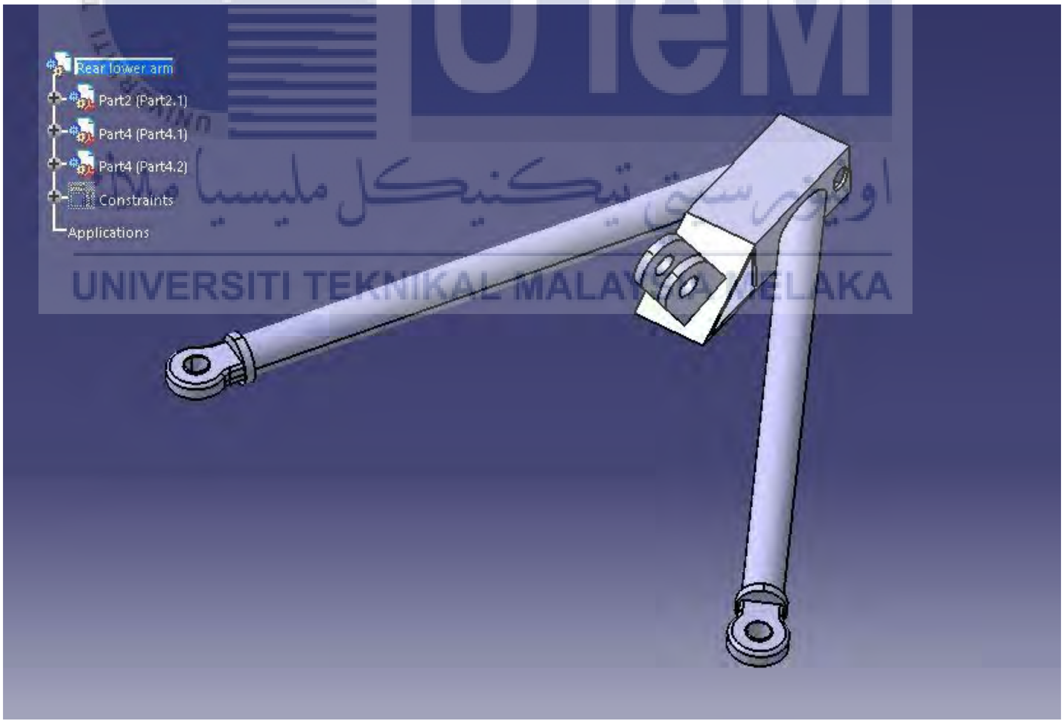


**Figure 3.4:** Rear lower arm suspension

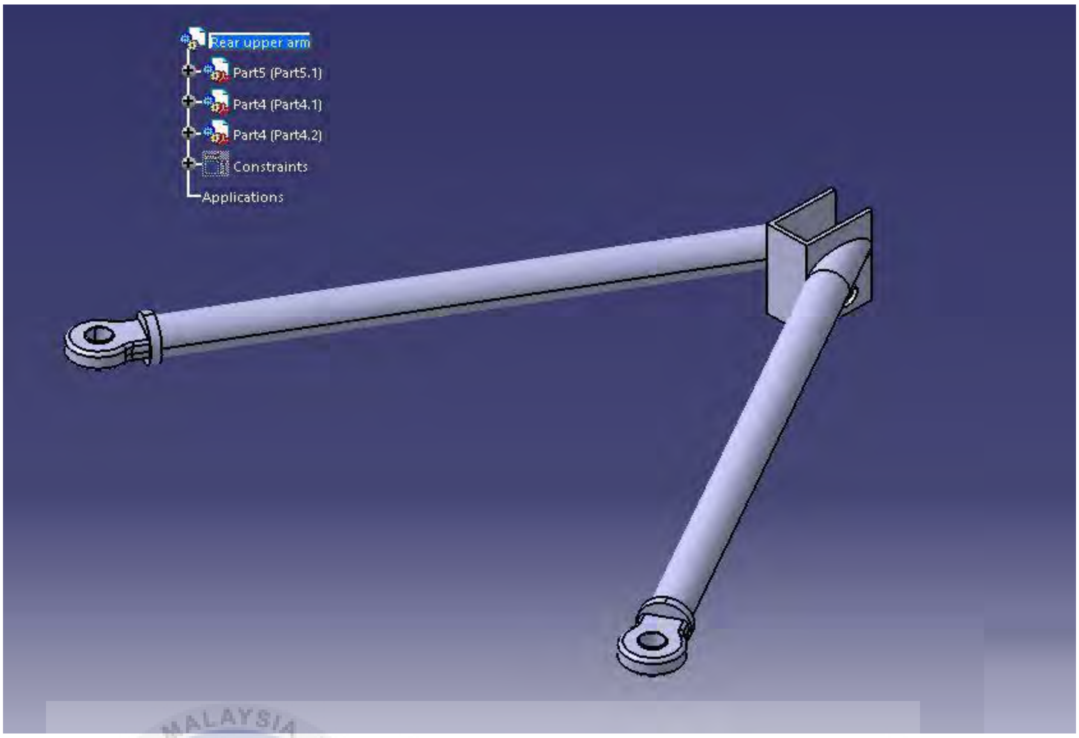


**Figure 3.5:** Rear upper arm suspension



#### 3.2.1.2 Connecting part

For the chosen suspension system designed in this project, the connecting part need to use so that the control arm can be attached directly to the suspension or shock absorber. This part also important in the design because of the vibration from the tire to the control arm will be absorb by the shock absorber and the driver of this car will feel less vibration from the car. Figure 3.6 and Figure 3.7 shows the front and rear connecting part between the control arm and the shock absorber.

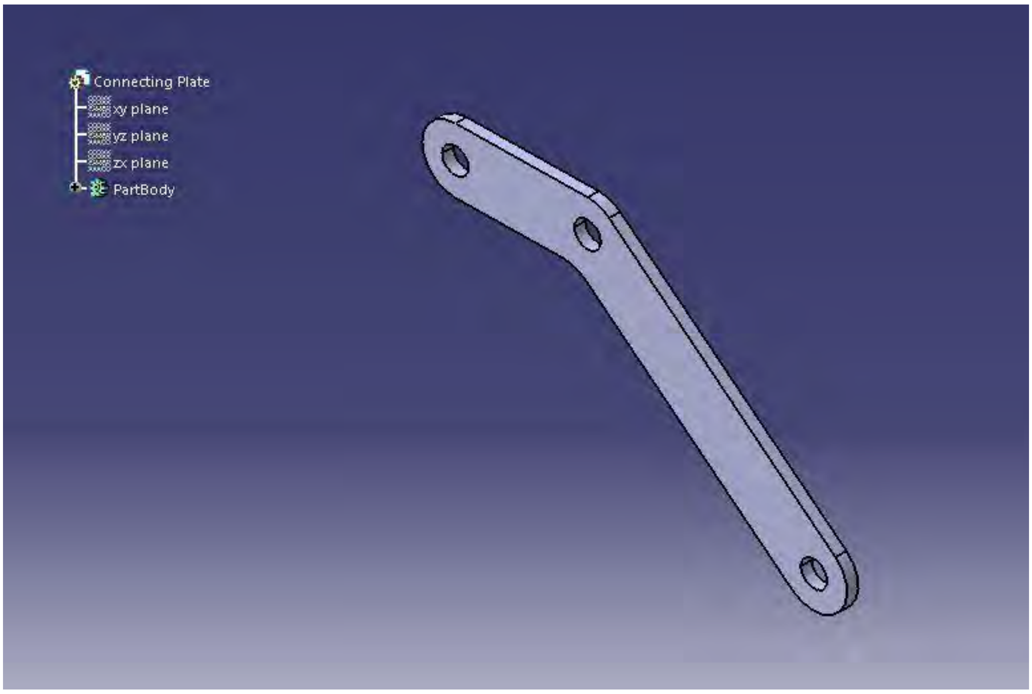


**Figure 3.6:** Front connecting part



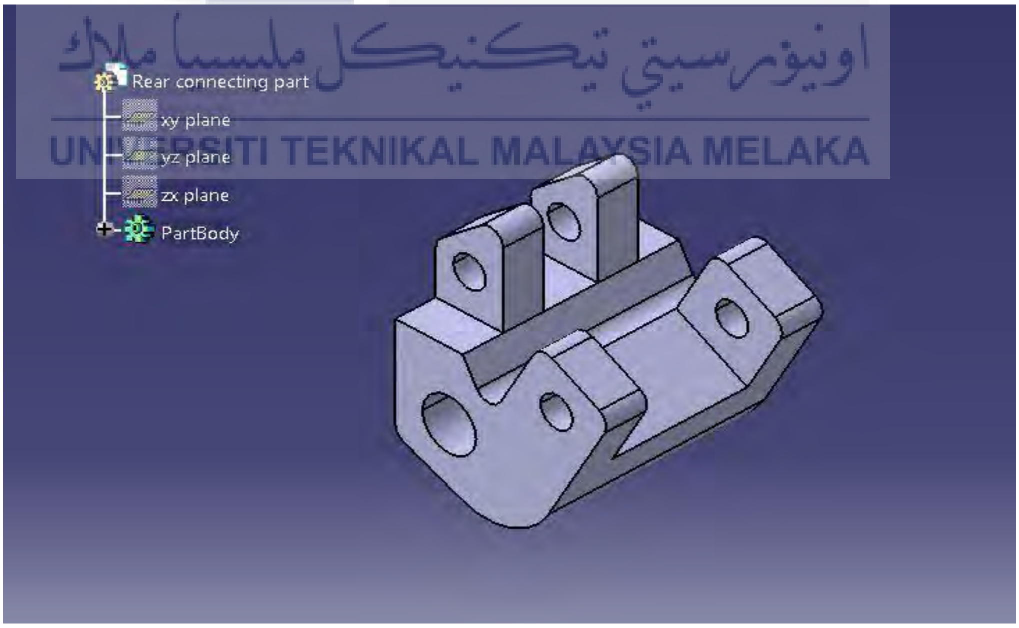


**Figure 3.7:** Rear connecting parts

### 3.2.1.3 Rockers

The rockers for the suspension system are shown in the Figure 3.8. The components are intended to be constructed from aluminium alloy and incorporate two thin plates separated by two hollow circular spacers that allow bolts to pass through them so that the rocker can be clamped together. The two plates feature holes machined in them to accommodate the fasteners needed to secure the push rods and shock absorbers and to allow the spacer bolts to pass through. Although not represented in the figures below, the rocker will also require some bearing support around its pivot axis to improve the smoothness of suspension actuation and to ensure that the pivot shaft does not wear excessively. As for manufacture, the aluminium plates would be best profile cut and drilled to achieve the holes while the spacers would be machined.




**Figure 3.8:** Rockers

### 3.2.1.4 Shock absorber

Shock absorber as shown in Figure 3.9 are hydraulic (oil) pump like system that control the effect and bounce back development of the vehicle's springs and suspension. Along with smoothening out knocks and vibrations, the key part of the shock absorber is to guarantee that the vehicle's tires stay in contact with the road surface at all circumstances, which guarantees the most secure control and braking reaction from the car. Basically, shock absorber does two things. Apart from controlling the development of springs and suspension, shock absorber also keeps the tires in contact with the ground at all circumstances. At rest or in movement, the base surface of the tires is the main part of the vehicle in contact with the road. At any time that a tire's contact with the ground is broken or lessened, the capacity to drive, control and brake is severely affected.



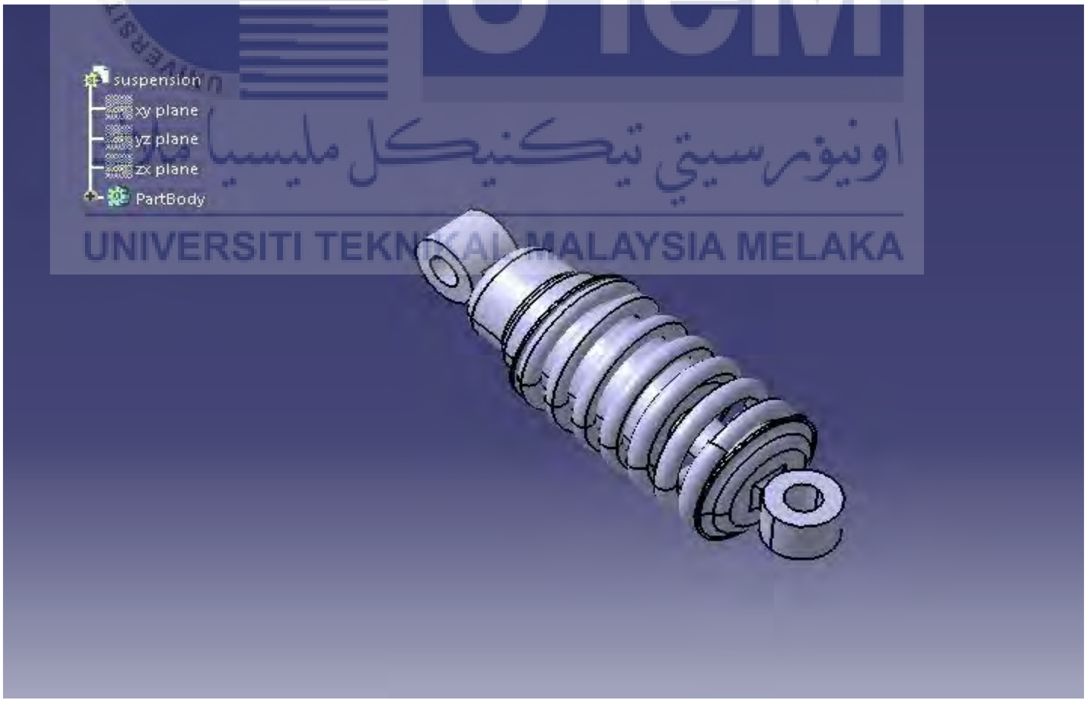


**Figure 3.9:** Shock absorber

### 3.2.2 Assembly design

Assembly design is the parts of the suspension that had been draw by using CATIA software are compiled together to become a product which is suspension system. The two figures below show both front and rear assembled design which is consist all the part that had been mention earlier. During the assembly part which is also using a CATIA software, all the suspension part need to be constrain so that it will be in the same axis and can be assemble perfectly.

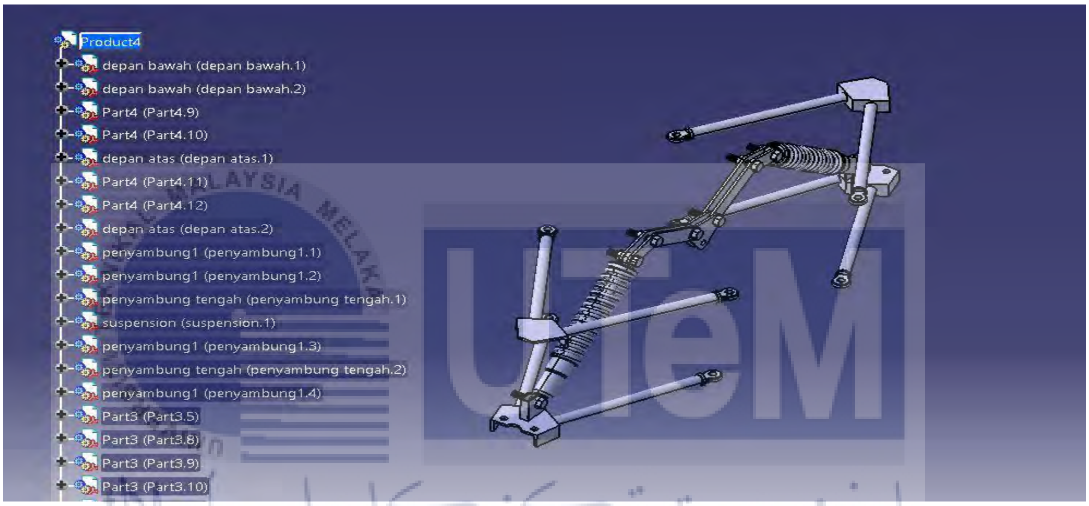


**Figure 3.10:** Front assemble suspension



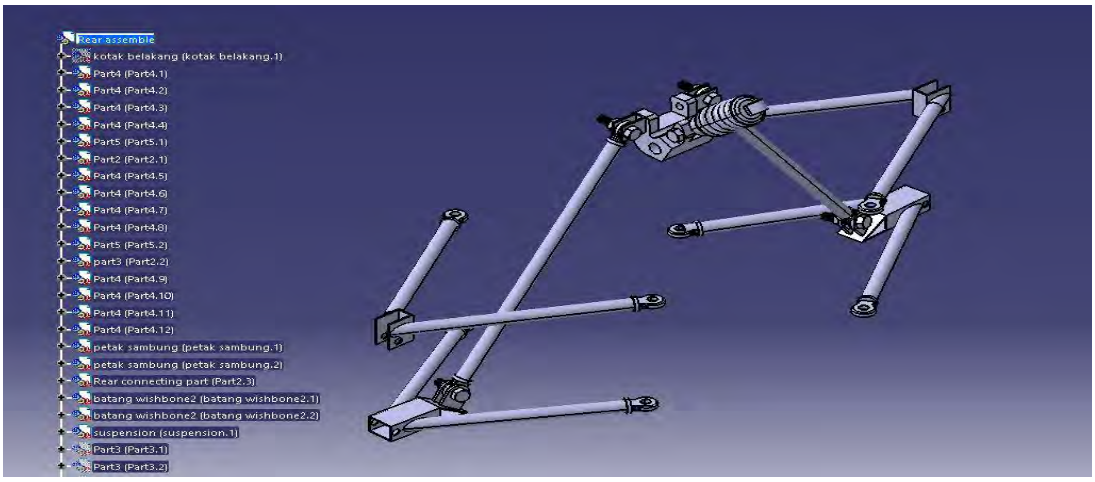


**Figure 3.11:** Rear assemble suspension

### 3.3 MATLAB/SIMULINK Software

In this chapter, the project implementation involving the step from modelling process until the final result of analysis will be explained. The modelling of half-car for passive suspension system were design. To develop the modelling, the MATLAB R2016a software will be used. A few simulation analyses will be carry out on this model.

#### 3.3.1 Modelling Assumption

Before developing the mathematical model, some assumption was done in order to verify the result. The vehicle body is lumped into a single mass which is referred as a sprung mass. The wheel was assumed as an unsprung mass. The tire was assumed to having contact with the ground all the time. The wheels also were assumed to be perpendicular to the vehicle body at all time and the effect of aerodynamics and cross wind effect are neglect.



#### 3.3.2 Vehicle Suspension System

Primary vehicle suspension is the term used to designate those suspension components that connecting to the axle and wheel assemblies of a vehicle to the frame of the vehicle. This is in contrast to the secondary suspensions, which are the elements connecting the other components to the frame or body of a vehicle such as engine mounts, seat suspensions, and cab mounts. There are two essential sorts of elements in suspension systems. These elements are springs and dampers. The part of the spring in a vehicle's suspension system is to endorse the static weight of the vehicle body. The part of the damper is to disperse vibration vitality and control the contribution from the road that is transmitted to the vehicle.

### 3.3.3 Schematic Model of Vehicle

This is the beginning of the step to design a suspension modelling of vehicle after review, analyse and understand the literature review as stated in chapter two above. It will involve establishing the design requirement and equation of motion for the single seated racing vehicle. This report only will be cover pitch and roll dynamic of the vehicle. The purpose of this phase is to make clearly about the physical setup of half-car model of the vehicle with its suspension. In order to be useful, the mathematical model must be sufficiently complex to accurately include the dynamics of the vehicle, yet be reasonably simple to manipulate. In order to examine the pitch and roll of the suspension for a vehicle, the simplest model that can be used is a four degree-of-freedom model.

Firstly, the sprung mass usually will be a reference for the mass of the vehicle body meanwhile the unsprung mass will be a reference for the mass of the wheel of the vehicle. The formula has been studied from the vibration characteristics of the vehicle to get the equations of motion based on Newton's second law for each mass. The response can be determined by solving the equation of motion after the excitation of the system is known. By using the half-car model as shown in Figure 3.16, a study can be made to review acceleration of the vehicle body and the motion of the wheel.



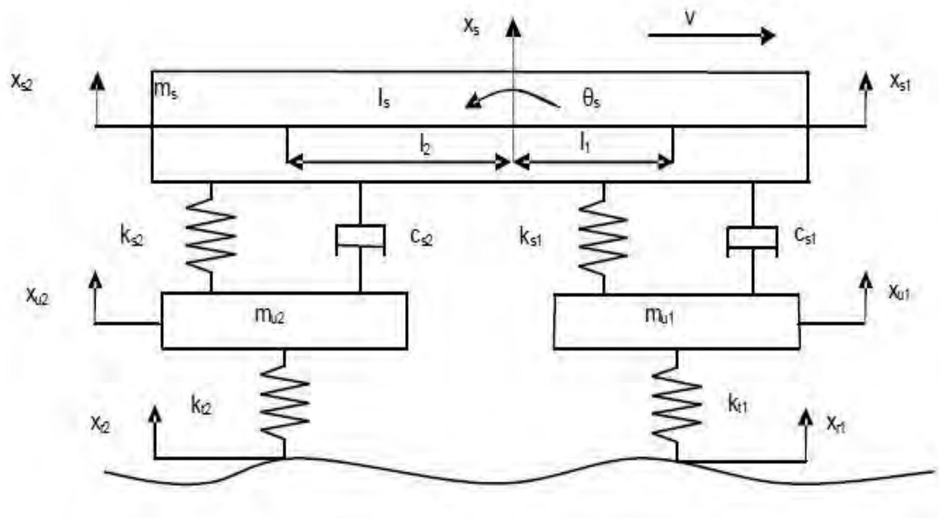


**Figure 3.12:** Half-car ride model [53]

### 3.3.4 Mathematical model of vehicle

The equation of motion of 4DOF system is given as:

**<u>Pitch</u>**

$Z_{br} = Z_b + l_r \sin\theta$ and $\dot{Z}_{br} = \dot{Z}_b + l_r \sin\dot{\theta}$

$Z_{bf} = Z_b - l_f \sin\theta$ and $\dot{Z}_{bf} = \dot{Z}_b - l_f \sin\dot{\theta}$

Since the value of $\theta$ is very small. So that, the value of $\sin\theta$ can be neglect and from $\theta$ only. So,

<u>Equation 1</u>

$Z_{br} = Z_b + l_r\theta$ and $\dot{Z}_{br} = \dot{Z}_b + l_r\dot{\theta}$

$Z_{bf} = Z_b - l_f\theta$ and $\dot{Z}_{bf} = \dot{Z}_b - l_f\dot{\theta}$

1) The vertical motion of the vehicle (Sprung Mass)

<u>Equation 2</u>

$$\ddot{Z}_b = \frac{1}{m_b}\left[ -c_{sr}(\dot{Z}_{br} - \dot{Z}_{wr}) - k_{sr}(Z_{br} - Z_{wr}) - c_{sf}(\dot{Z}_{bf} - \dot{Z}_{wf}) - k_{sf}(Z_{bf} - Z_{wf}) \right]$$

Substitute equation (1) into equation (2),

<u>Equation 3</u>

$$\ddot{Z}_b = \frac{1}{m_b}\left[ -c_{sr}(\dot{Z}_b + l_r\dot{\theta} - \dot{Z}_{wr}) - k_{sr}(Z_b + l_r\theta - Z_{wr}) - c_{sf}(\dot{Z}_b - l_f\dot{\theta} - \dot{Z}_{wf}) - k_{sf}(Z_b - l_f\theta - Z_{wf}) \right]$$

2) The rotational motion of vehicle body (Pitch)

<u>Equation 4</u>

$$\ddot{\theta} = \frac{1}{I_\theta}\{(-l_f\ [\ -c_{sf}(\dot{Z}_{bf} - \dot{Z}_{wf}) - k_{sf}(Z_{bf} - Z_{wf})] - [-l_r\ (c_{sr}(\dot{Z}_{br} - \dot{Z}_{wr}) - k_{sr}(Z_{br} - Z_{wr})]\}$$

Substitute equation (1) into equation (4),

Equation 5

$$\ddot{\theta} = \frac{1}{I_\theta}\{(-l_f\,[\,-c_{sf}(\dot{Z}_b - l_f\dot{\theta} - \dot{Z}_{wf}) - k_{sf}(Z_b - l_f\theta - Z_{wf})] - [-l_r\ (c_{sr}(\dot{Z}_b + l_r\dot{\theta} - \dot{Z}_{wr}) - k_{sr}(Z_b + l_r\theta - Z_{wr})]\}$$

3) The vertical motion of rear wheel (Tyre)

Equation 6

$$\ddot{Z}_{wr} = \frac{1}{m_{wr}}[\,-c_{sr}(\dot{Z}_{wr} - \dot{Z}_{br}) - k_{sr}(Z_{wr} - Z_{br}) - k_{tr}(Z_{wf} - Z_{rr})]$$

Substitute equation (1) into equation (6),

Equation 7

$$\ddot{Z}_{wr} = \frac{1}{m_{wr}}[\,-c_{sr}(\dot{Z}_{wr} - l_r\dot{\theta} - \dot{Z}_b) - k_{sr}(Z_{wr} + l_r\theta - Z_b) - k_{tr}(Z_{wf} - Z_{rr})\ ]$$



4) The vertical motion of front wheel (Tyre)

Equation 8

$$\ddot{Z}_{wf} = \frac{1}{m_{wf}}[\,-c_{sf}(\dot{Z}_{wf} - \dot{Z}_{bf}) - k_{sf}(Z_{wf} - Z_{bf}) - k_{tf}(Z_{wf} - Z_{rf})\ ]$$

Substitute equation (1) into equation (8),

Equation 9

$$\ddot{Z}_{wf} = \frac{1}{m_{wf}}[\,-c_{sf}(\dot{Z}_{wf} + l_f\dot{\theta} + \dot{Z}_b) - k_{sf}(Z_{wf} + l_f\theta - Z_b) - k_{tf}(Z_{wf} - Z_{rf})\ ]$$

**Roll Dynamic**

$Z_{bl} = Z_b + 0.5t \sin \emptyset$ and $\dot{Z}_{bl} = \dot{Z}_b + 0.5t \sin \dot{\emptyset}$

$Z_{br} = Z_b - 0.5t \sin \emptyset$ and $\dot{Z}_{br} = \dot{Z}_b - 0.5t \sin \dot{\emptyset}$

Since the value of $\emptyset$ is very small. So that, the value of $\sin \emptyset$ can be neglect and form $\emptyset$ only. So,

Equation 1

$Z_{bl} = Z_b + 0.5t\emptyset$ and $\dot{Z}_{bl} = \dot{Z}_b + 0.5t\dot{\emptyset}$

$Z_{br} = Z_b - 0.5t\emptyset$ and $\dot{Z}_{bl} = \dot{Z}_b - 0.5t\dot{\emptyset}$

1) The vertical motion of the vehicle (Sprung Mass)

Equation 2

$$m_b\ddot{Z}_b - k_{sr}(Z_{wr} - Z_{br}) - c_{sr}(\dot{Z}_{wr} - \dot{Z}_{br}) - k_{sl}(Z_{wl} - Z_{bl}) - c_{sl}(\dot{Z}_{wl} - \dot{Z}_{bl}) = 0$$

Substitute equation (1) into equation (2),

Equation 3

$$m_b\ddot{Z}_b = k_{sr}(Z_{wr} - Z_b - 0.5t\emptyset) + c_{sr}(\dot{Z}_{wr} - \dot{Z}_b - 0.5t\dot{\emptyset}) + k_{sl}(Z_{wl} - Z_b + 0.5t\emptyset) + c_{sl}(\dot{Z}_{wl} - \dot{Z}_b + 0.5t\dot{\emptyset})$$

2) The rotational motion of vehicle body (Pitch)

Equation 4

$$I_\emptyset\ddot{\emptyset} + 0.5t\,[\,c_{sr}(\dot{Z}_{wr} - \dot{Z}_{br}) + k_{sr}(Z_{wr} - Z_{br})] - 0.5t\,[(c_{sl}(\dot{Z}_{wl} - \dot{Z}_{bl}) + k_{sl}(Z_{wl} - Z_{bl})] = 0$$

Substitute equation (1) into equation (4), Equation 5

$$I_\emptyset \ddot{\emptyset} = 0.5t\ [\ c_{sl}(\dot{Z}_{wl} - 0.5t\dot{\emptyset} - \dot{Z}_b) + k_{sl}(Z_{wl} - 0.5t\emptyset - Z_b)] -$$
$$0.5t\ [c_{sr}(\dot{Z}_{wr} + 0.5t\dot{\emptyset} - \dot{Z}_b) + k_{sr}(Z_{wr} - 0.5t\emptyset - Z_b)]$$

3) The vertical motion of rear wheel (Tyre)

Equation 6

$$m_{wr}\ddot{Z}_{wr} + c_{sr}(\dot{Z}_{wr} - \dot{Z}_{br}) + k_{sr}(Z_{wr} - Z_{br}) + k_{tr}(Z_{wr} - Z_{rr}) = 0$$

Substitute equation (1) into equation (6),

Equation 7

$$m_{wr}\ddot{Z}_{wr} = -c_{sr}(\dot{Z}_{wr} - 0.5t\dot{\emptyset} - \dot{Z}_b) - k_{sr}(Z_{wr} - 0.5t\emptyset - Z_b) + k_{tr}(Z_{wr} - Z_{rr})$$

4) The vertical motion of front wheel (Tyre)

Equation 8

$$m_{wl}\ddot{Z}_{wl} + c_{sl}(\dot{Z}_{wl} - \dot{Z}_{bl}) + k_{sl}(Z_{wl} - Z_{bl}) + k_{tr}(Z_{wl} - Z_{rl}) = 0$$

Substitute equation (1) into equation (8),



Equation 9

$$m_{wl}\ddot{Z}_{wl} = -c_{sl}(\dot{Z}_{wl} + 0.5t\dot{\theta} - \dot{Z}_b) - k_{sl}(Z_{wl} + 0.5t\theta - Z_b) + k_{tr}(Z_{wl} - Z_{rl})$$

This equation will be used to make an analysis by using the MATLAB/SIMULINK software. All variable will be involving during an analysis but the result is only for searching the pitch vs time and roll dynamic vs time.

### 3.3.5 Parameter for Mathematical model

All the parameter of the vehicle has been shown in the Table 3.1. This is parameter for single seated racing vehicle, but there are several parameters are from the assumption.

**Table 3.1:** Parameters involved in vehicle model

| Parameter | Symbol | Value |
|---|---|---|
| Mass of Vehicle Body (Kg) | $M_b$ | 350 |
| Mass of Front Wheel (Kg) | $M_{wf}$ | 10 |
| Mass of Rear Wheel (Kg) | $M_{wr}$ | 10 |
| Front Suspension Stiffness (N/m) | K1<br>K2<br>K3 | 16200<br>18500<br>22300 |
| Rear Suspension Stiffness (N/m) | K1<br>K2<br>K3 | 16200<br>18500<br>22300 |
| Front Tire Stiffness (N/m) | Kt | 191000 |
| Rear Tire Stiffness (N/m) | Kt | 191000 |
| Front Suspension Damping Coefficient (Ns/m) | C1<br>C2<br>C3 | 700<br>900<br>1200 |
| Rear Suspension Damping Coefficient (Ns/m) | C1<br>C2<br>C3 | 700<br>900<br>1200 |
| Front Tyre to COG | $l_f$ | 0.65 |
| Rear Tyre to COG | $l_r$ | 0.85 |
| Wheel Track | $l_w$ | 1.15 |

### 3.3.6 MATLAB Simulation

In implementing the MATLAB software, "Simulink" simulation will be used to design the mathematical model for half-car ride model. From the Simulink, the Simulink Library Browser can be open which allows user to use the block such as gain, integrator, scope, step response etc. Based on Figure 3.16, the Simulink are available to used and design the mathematical model for ride model. Figure 3.17 shown the overview of the MATLAB software which are have a workspace, editor and command window and the Simulink library and browser are shown in Figure 3.18.

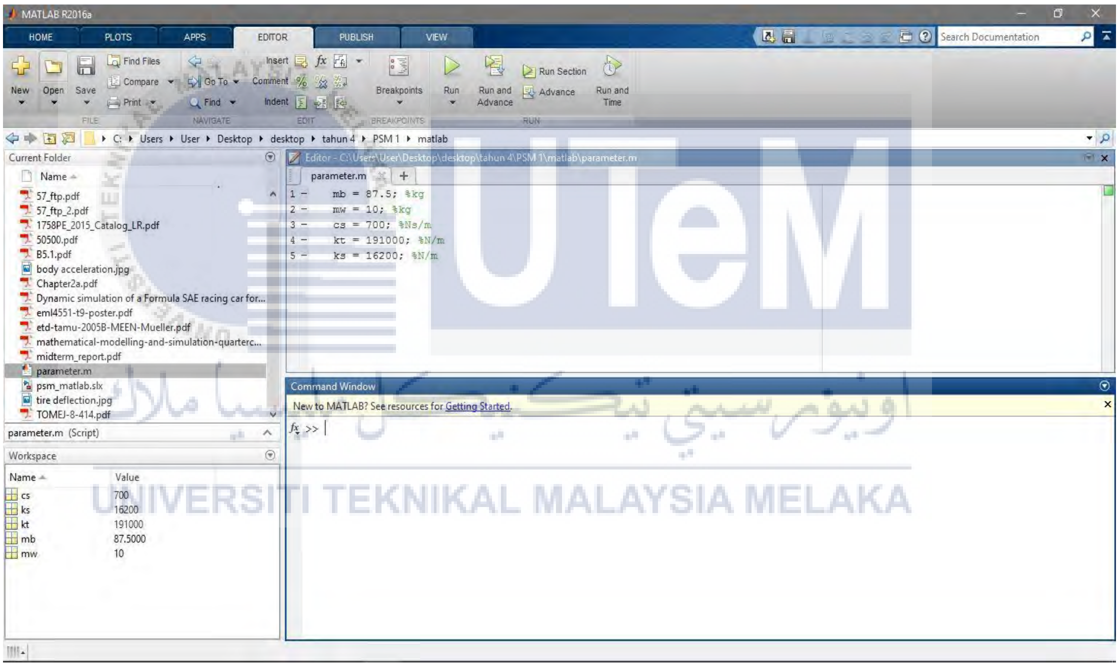




**Figure 3.13:** Overview of MATLAB software R2016a

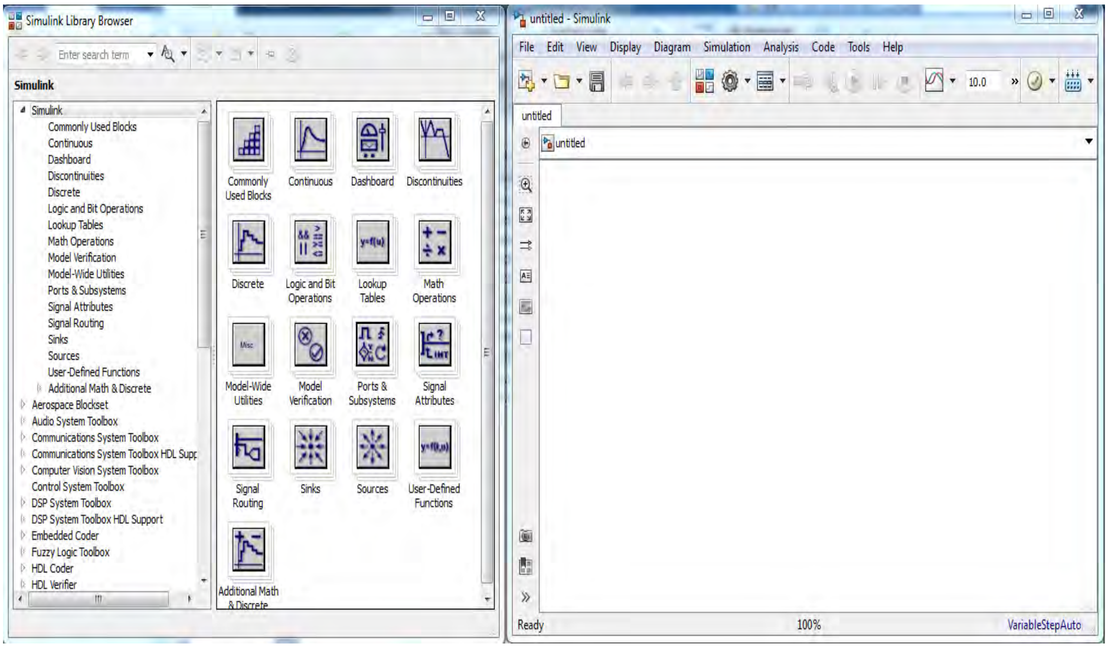


**Figure 3.14:** Simulink library and browser

### 3.3.7 Ride Model Simulink

The ride model will develop in MATLAB Simulink based the equation that had been mentioned above. The design will represent different input and output. From this ride model, we can analyse the effect on the both suspension and wheel in order to improve the comfort of the vehicle. To improve the comfort of the vehicle, stiffness of spring and damping coefficient is an important role. So that, the parameter of spring stiffness and damping coefficient for the system need to change until the analysis on comfort will be achieve. The model for both pitch and roll that had been made by using MATLAB Simulink are as figure below.

**Pitch model**

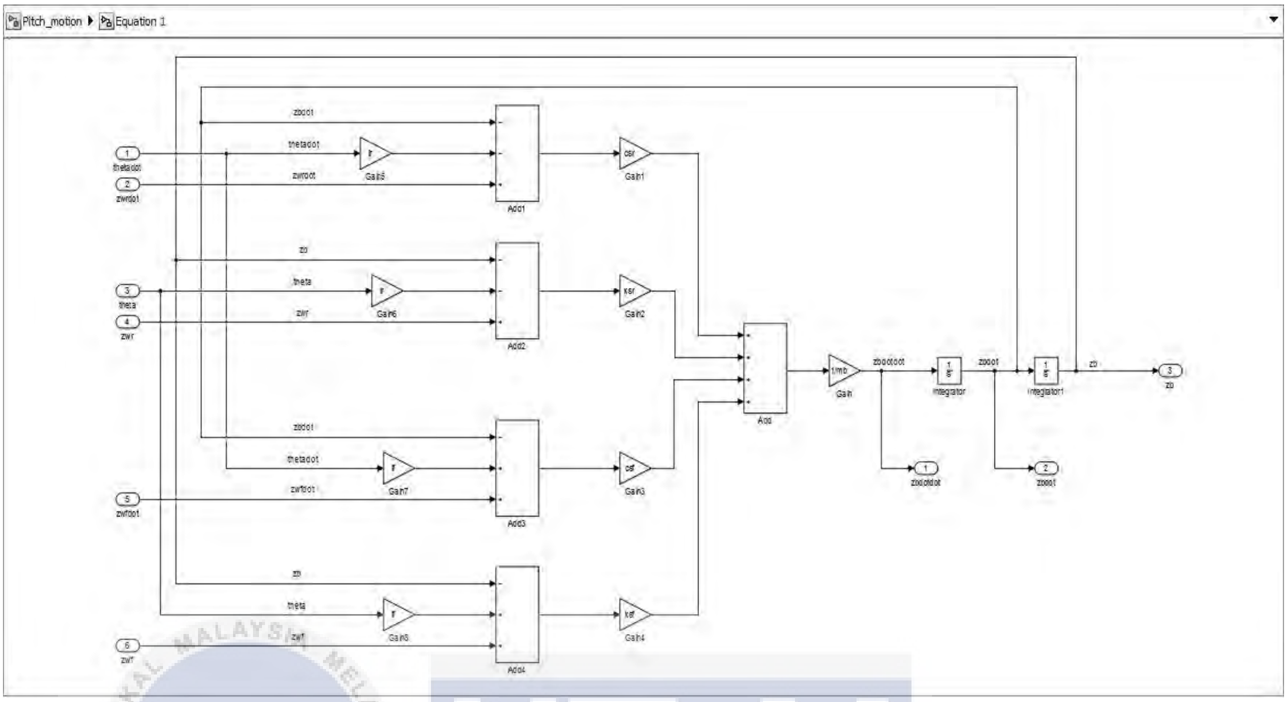


**Figure 3.15:** Pitch Simulink of the equation 1



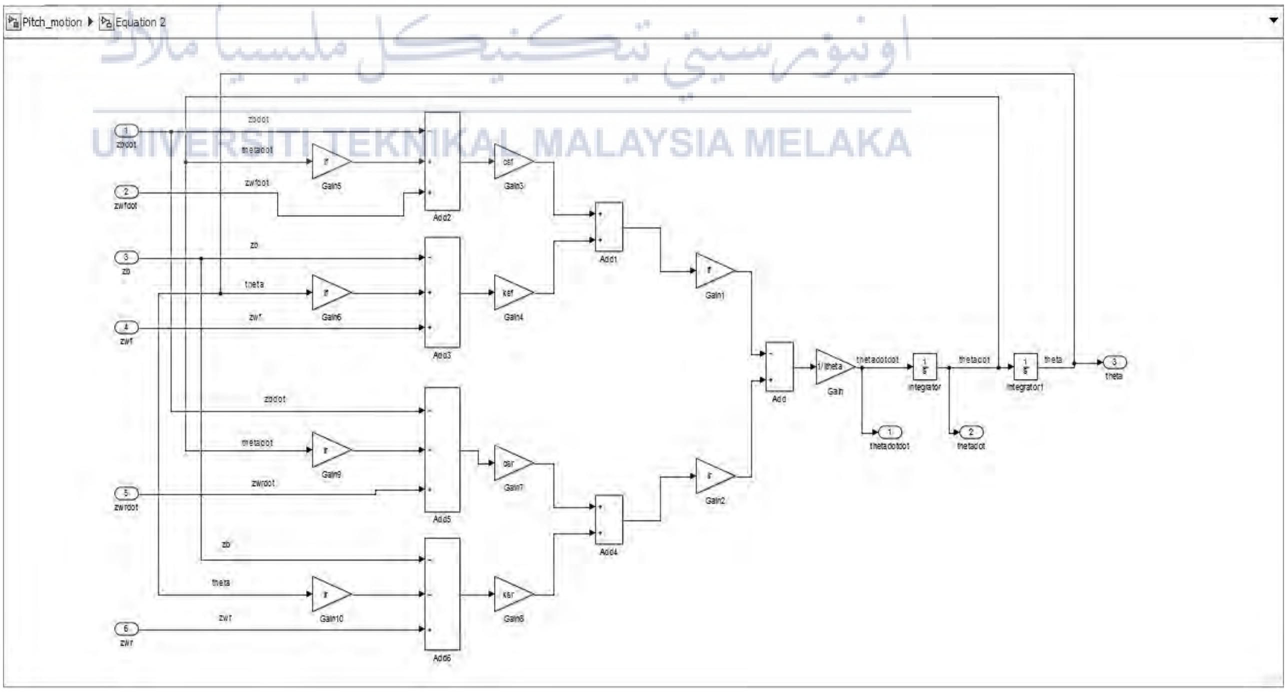


**Figure 3.16:** Pitch Simulink of the equation 2

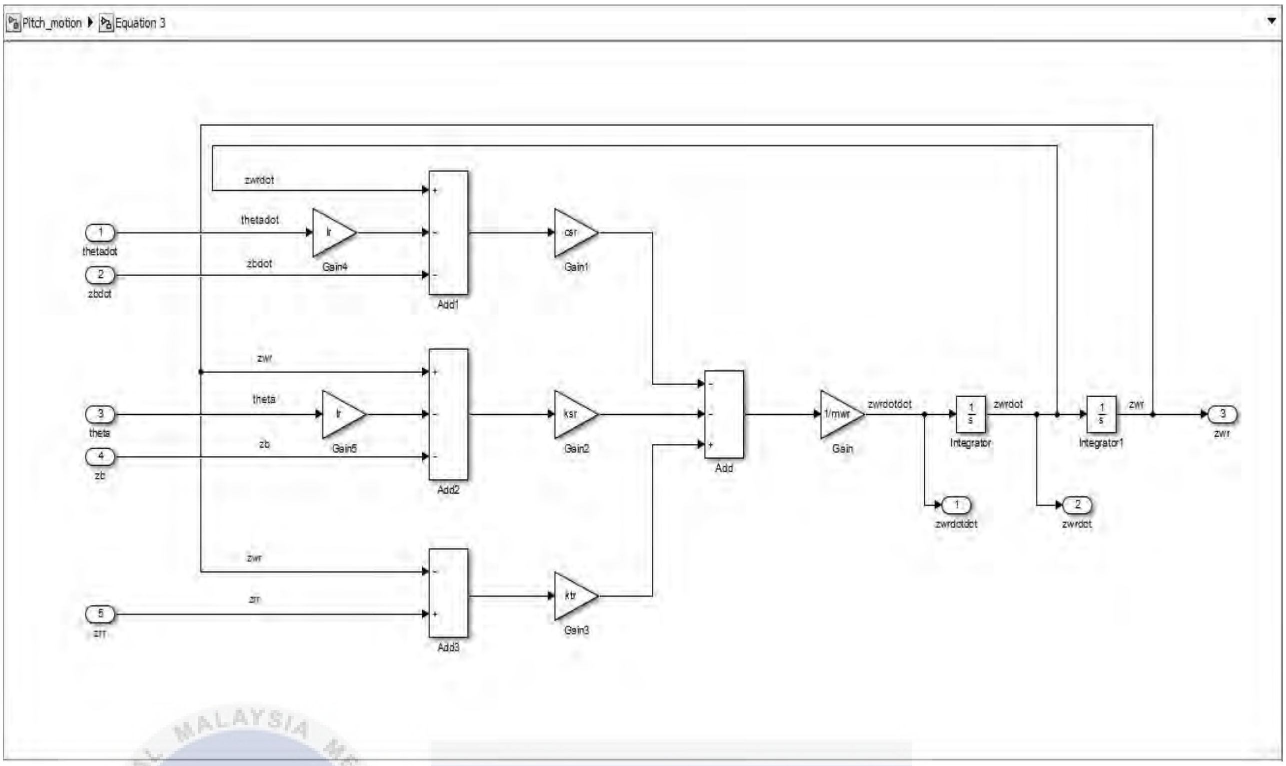


**Figure 3.17**: Pitch Simulink of the equation 3



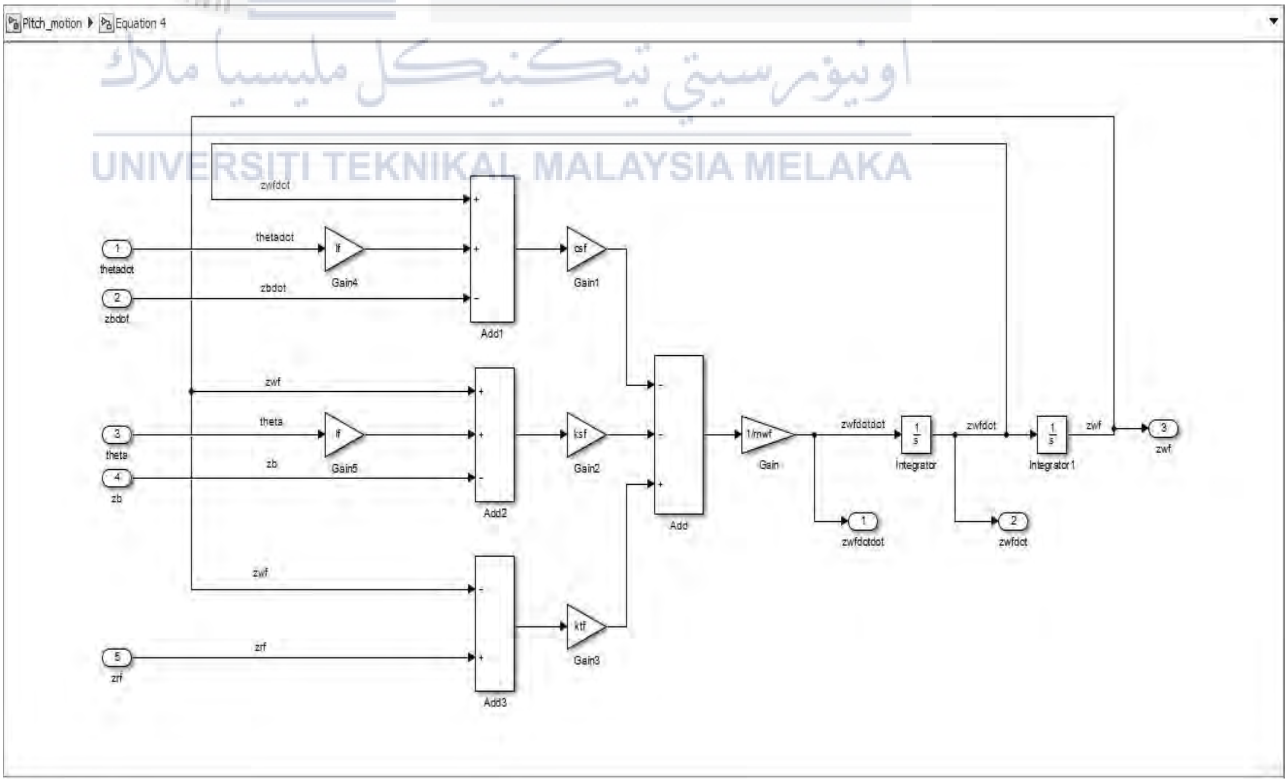


**Figure 3.18**: Pitch Simulink of the equation 4

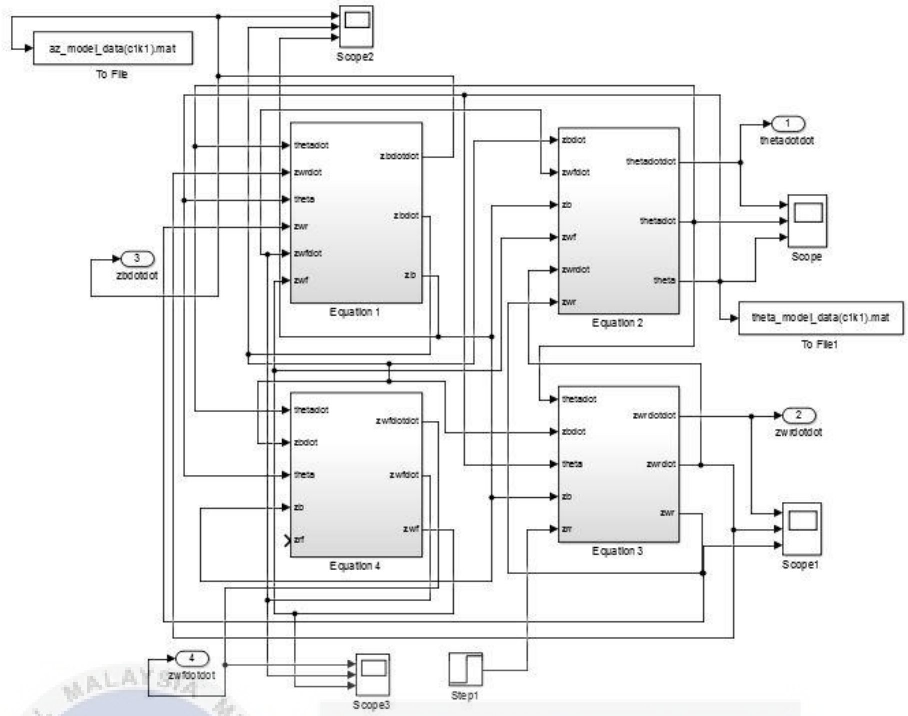


**Figure 3.19**: Subsystem of the Simulink for all four equations



**Roll model**

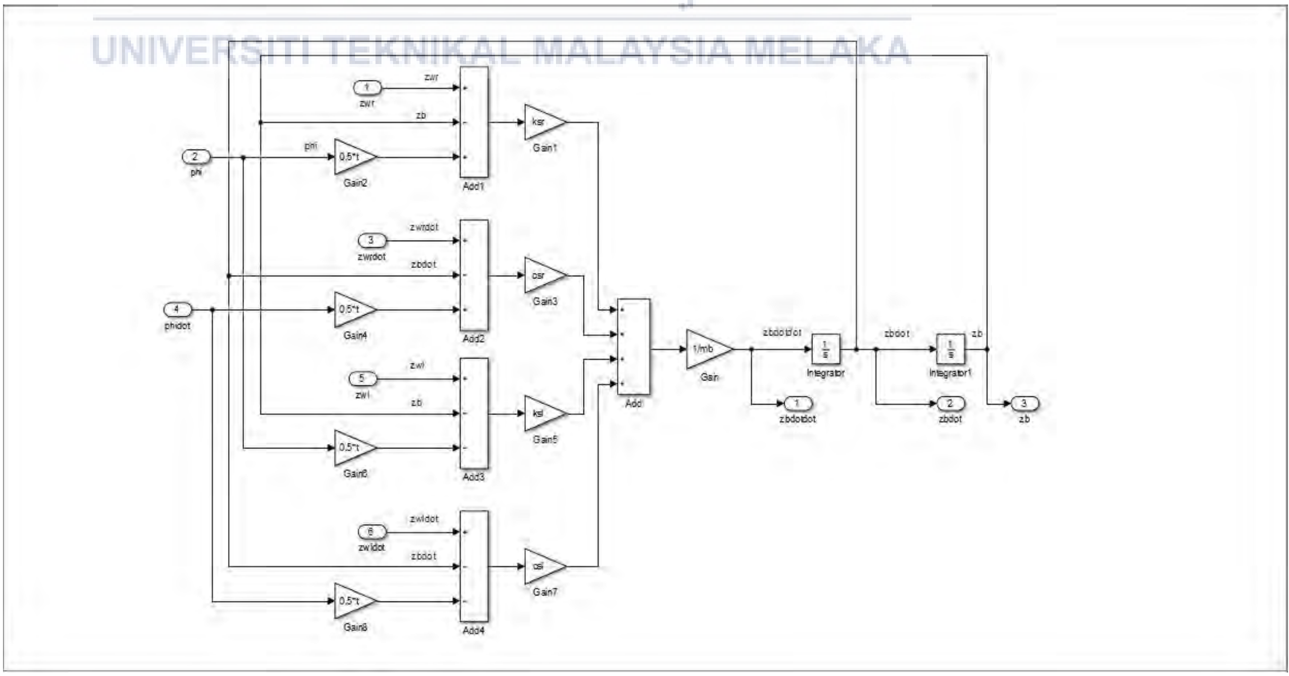


**Figure 3.20**: Roll Simulink of the equation 1

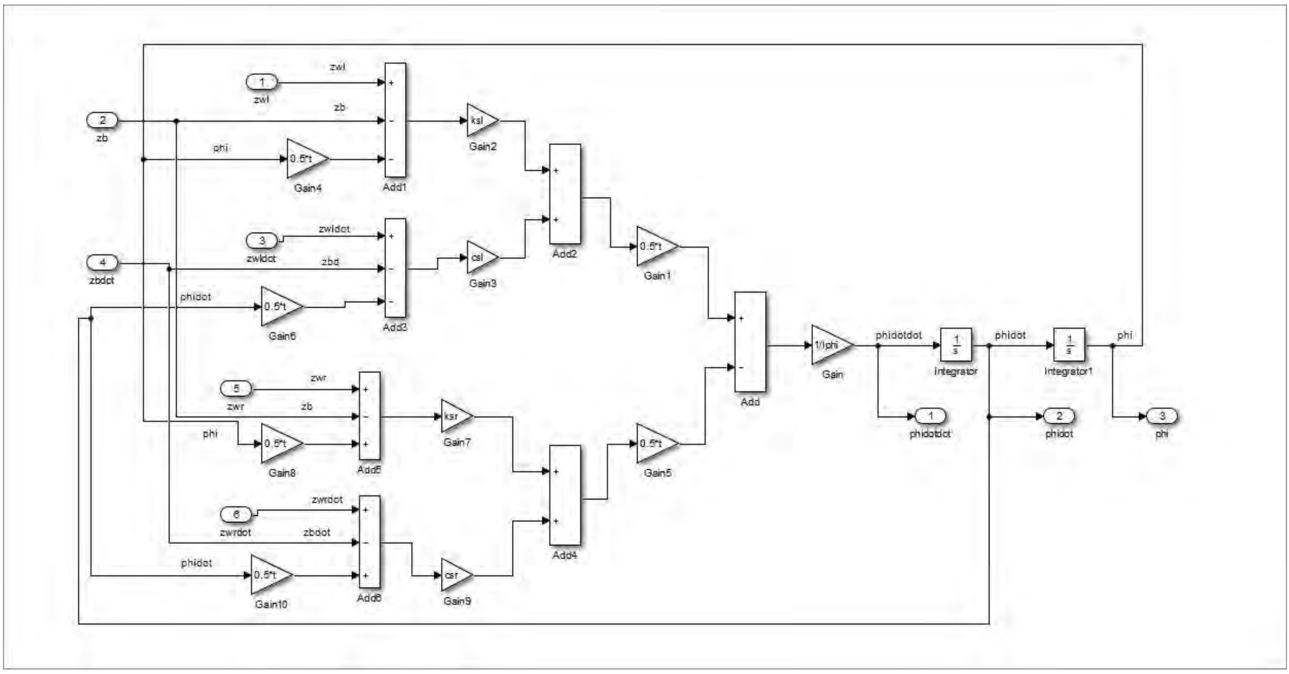


**Figure 3.21**: Roll Simulink of the equation 2



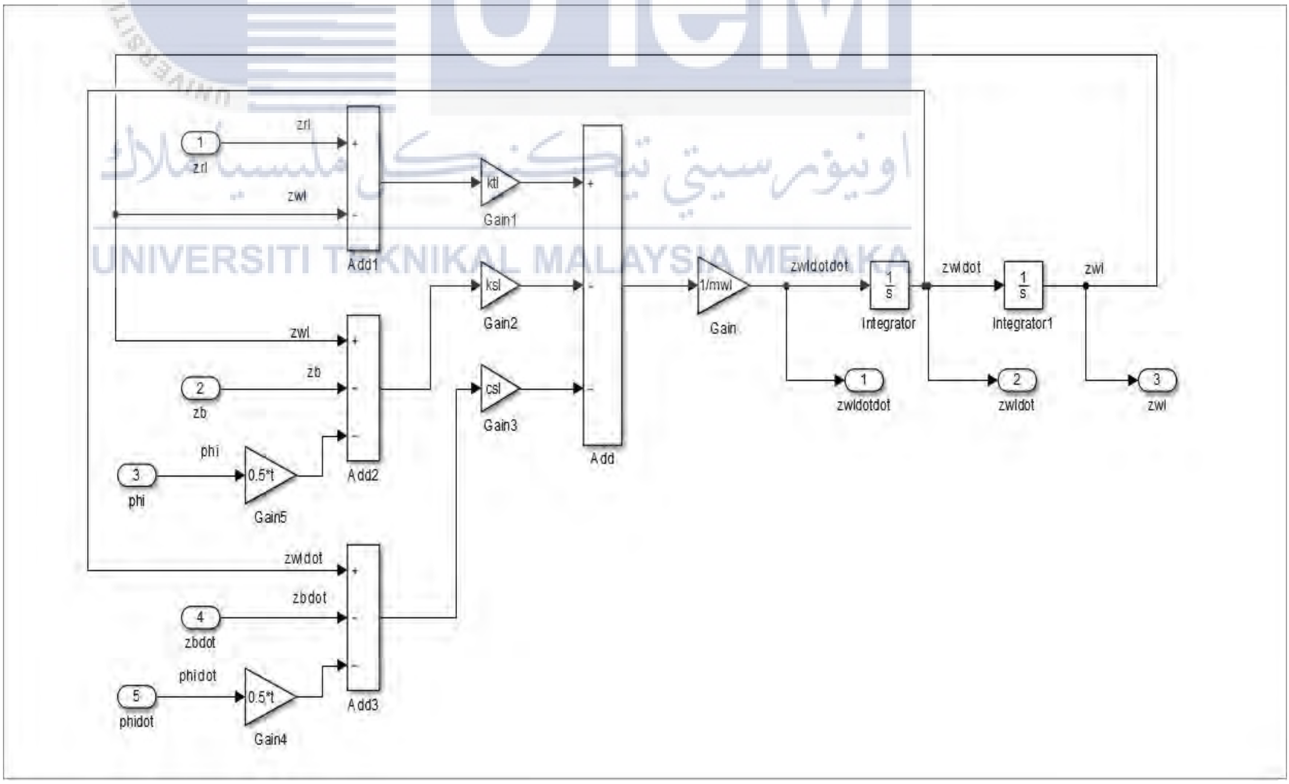


**Figure 3.22**: Roll Simulink of the equation 3

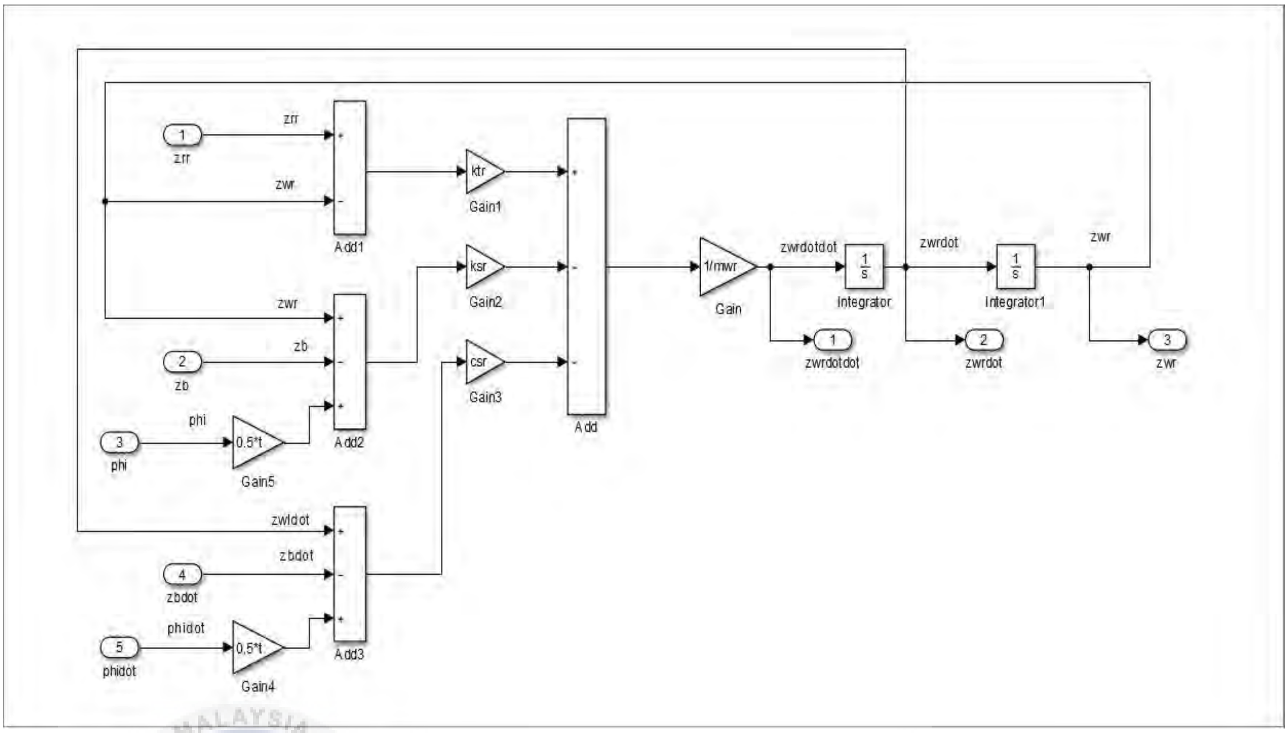


**Figure 3.23**: Roll Simulink of the equation 4



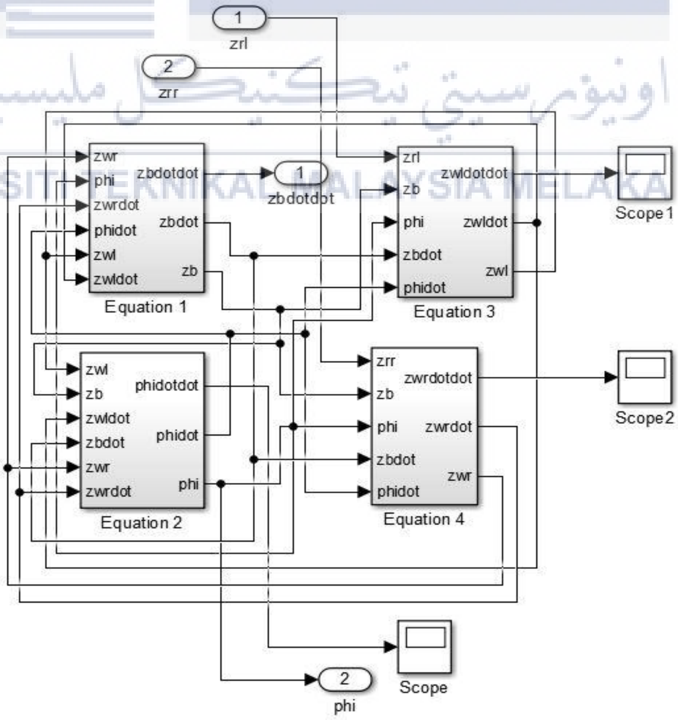


**Figure 3.24**: Subsystem of the Simulink for all four equations

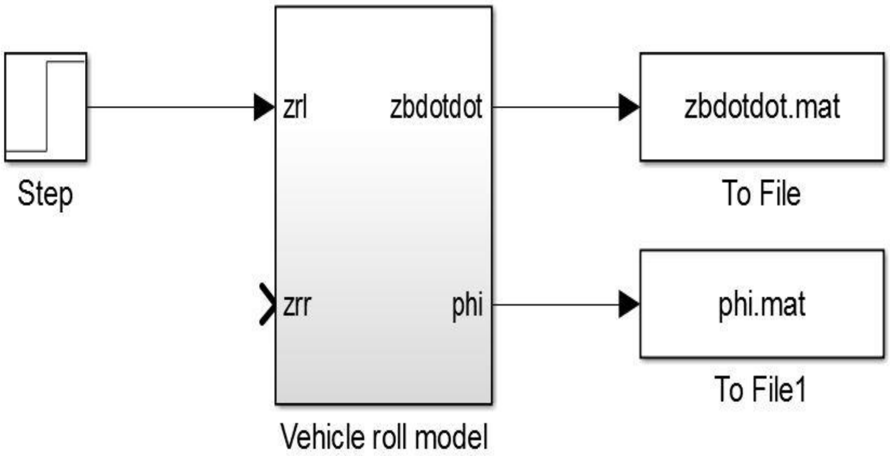


**Figure 3.25**: Simplified subsystem of the Simulink for all four equations

# CHAPTER 4

# RESULT AND ANALYSIS

## 4.1 Introduction

This chapter describes the analysis of the results obtained from the analytical and simulations studies. Among the item that had discussed in the analysis, the result included a several phases, which starts from. the design phase using CATIA software up to the analysis of the material and also the result of three-dimensional (3-D) prototype design. In addition, dynamic numerical simulation had been performed by using MATLAB to determine the relationship between the pitch body, and roll dynamics with respect of the time.

## 4.2 CATIA Software

By using this software, there are two types of analysis that have been performed, that is Von Mises stress analysis and the translational displacement of the designed product. This analysis is conducted by applying s total force of 1000N on the designed suspension system. For the Von Mises analysis, the purpose of the analysis is to obtain the stress of the product. This stress determination is important to ensure that the product is strong enough. . The results obtained are depicted in Figure 4.1. Meanwhile, Fig. 4.2 shows the results of the translational displacement.

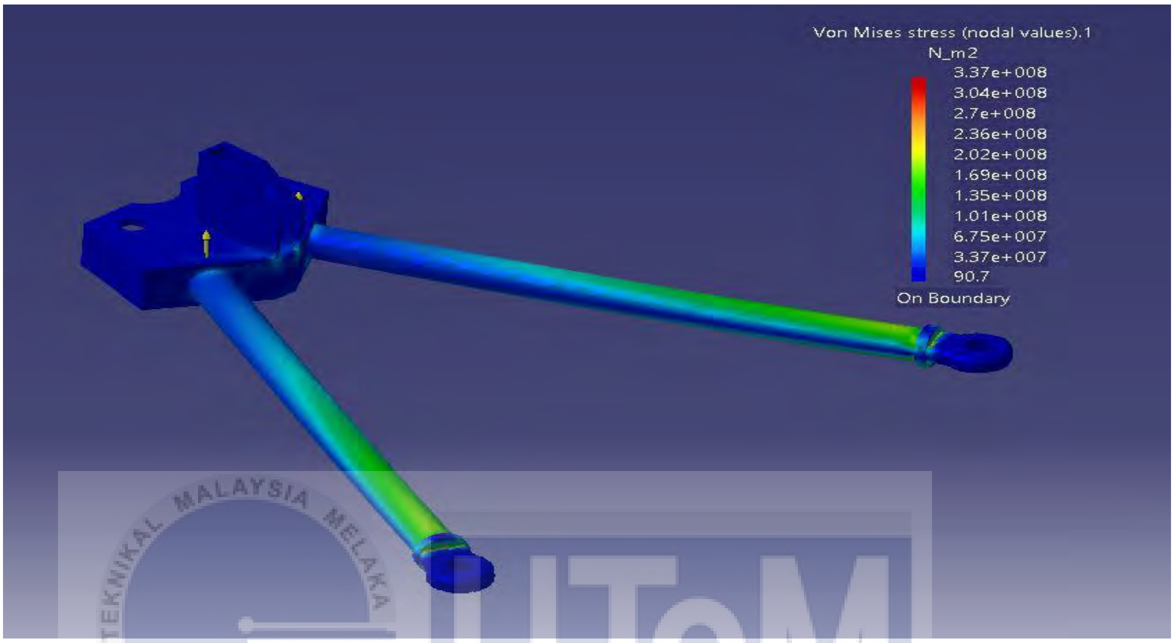


**Figure 4.1:** Von Mises analysis



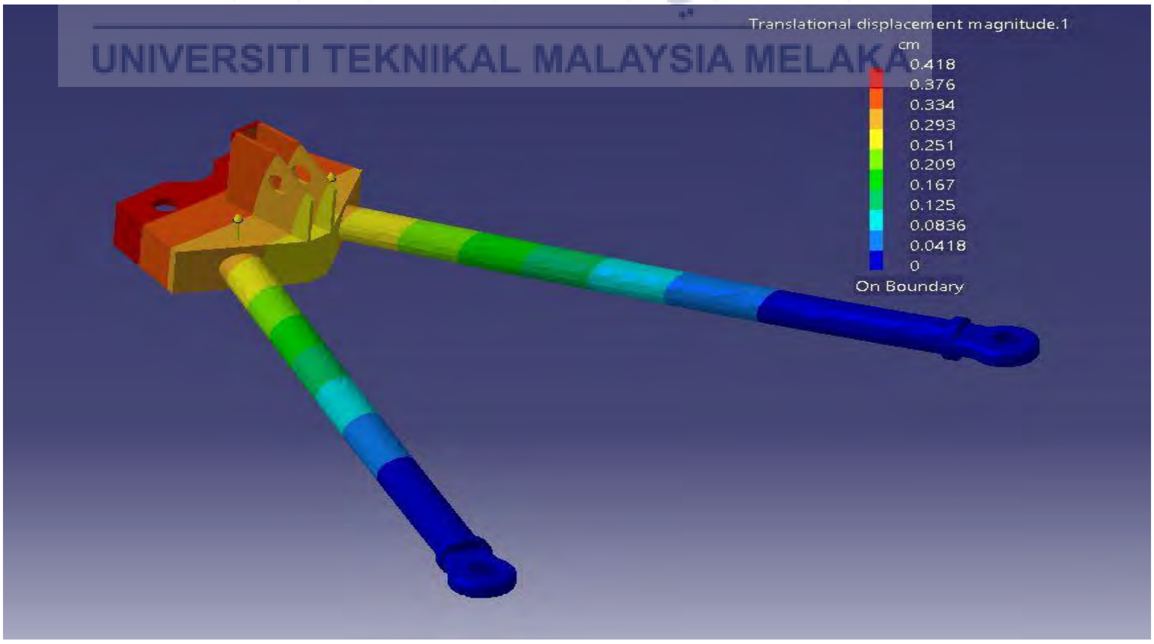


**Figure 4.2:** Translational displacement analysis

As seen in Fig. 4.1, the result suggests that the highest stress on the suspension arm is 0.337 MPa, that occurs at the edge of the arm. This maximum stress occurs as the arm is clamped that had to the chassis of the vehicle. On the other hand, the lowest stress is 90.7 Pa, which occurs at the area where the force is applied. Apart from that, the analysis of displacement on the product shows that the maximum displacement is 4.18 mm at the place where the force is applied.

### 4.3 MATLAB Software

Analysis was carried out to determine the relationship between the pitch and the roll dynamics of the vehicle with respect to the time. To obtain the result of the analysis, the coding must be clearly put in the command prompt so that the graph of the result will appear. The coding for the command prompt of pitch and roll dynamic is as follows:

#### 4.3.1 Pitch Body



From the Figure 4.3 until Figure 4.5 shows that the result of pitch body effect with different pattern of graph for three different value of spring stiffness which are K1 = 16200 N/m, K2 = 18500 N/m and K3 = 22300 N/m and with three different constant value of damping (C1 = 700 Ns/m, C2 = 900 Ns/m, C3 = 1200 Ns/m). Then, from the Figure 4.6 to Figure 4.8, it shows that the result of the pitch body effect with different value of damping and three different value of constant spring stiffness.

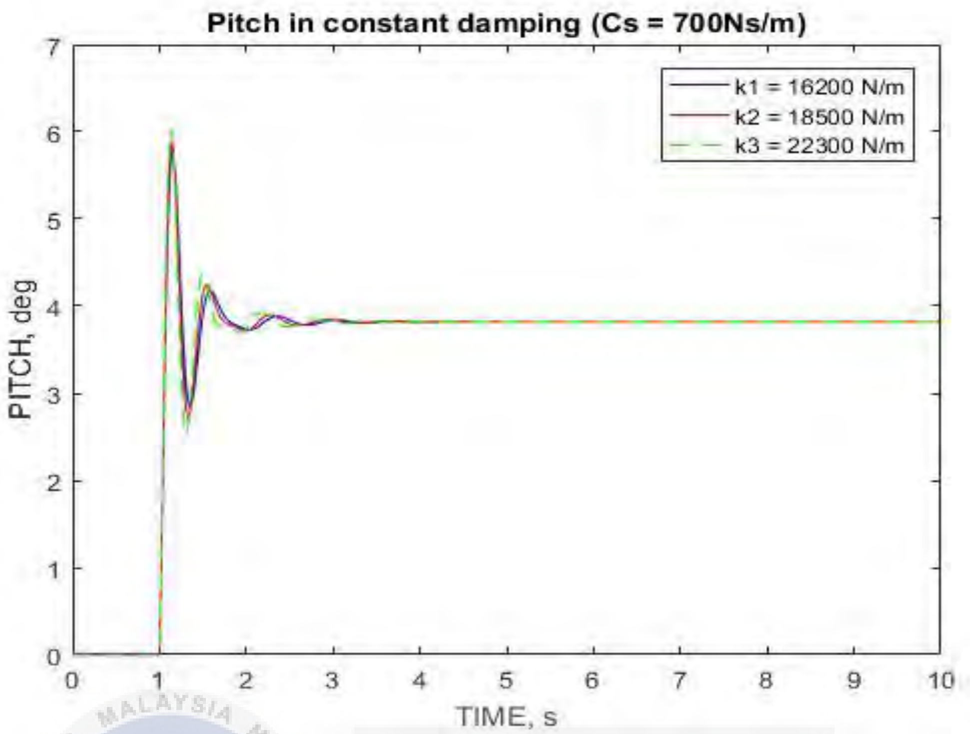


**Figure 4.3:** Graph of Pitch against Time with Cs = 700 Ns/m



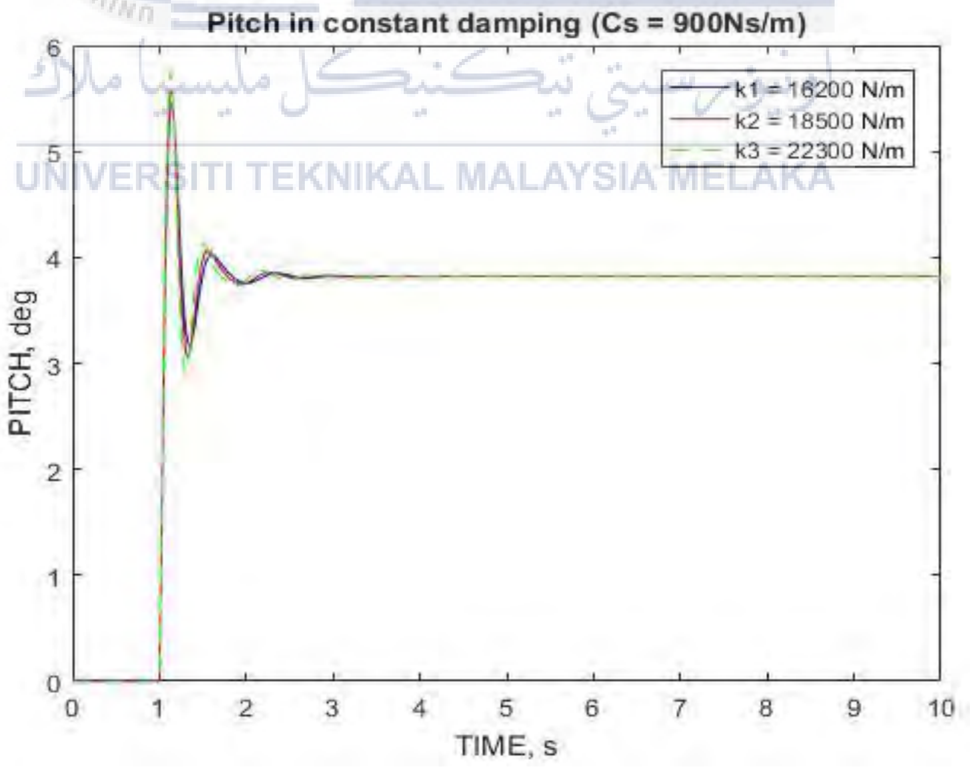


**Figure 4.4:** Graph of Pitch against Time with Cs = 900 Ns/m

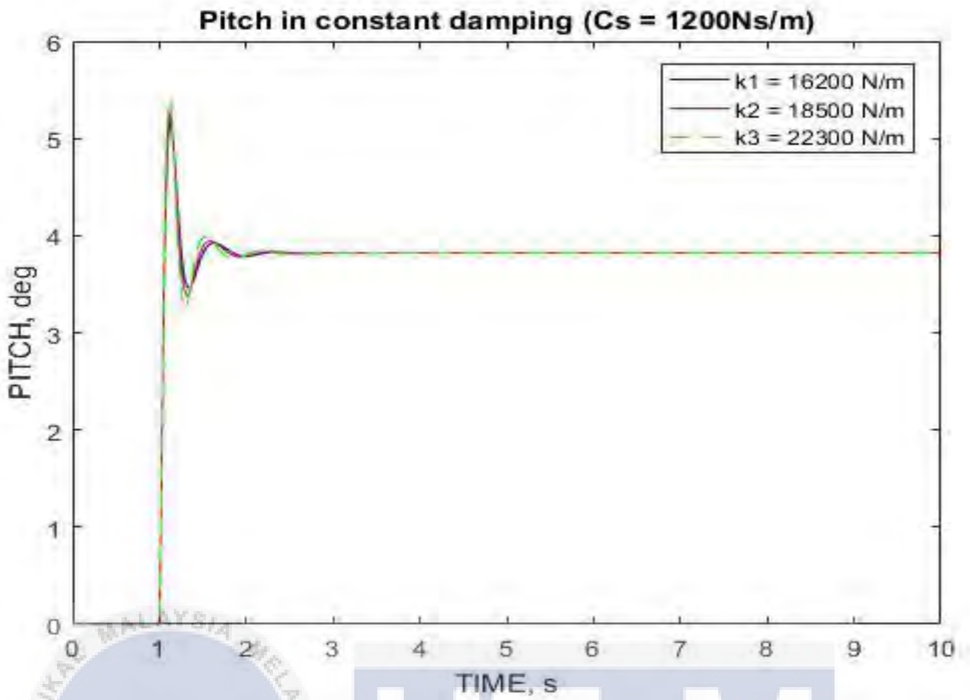


**Figure 4.5:** Graph of Pitch against Time with Cs = 1200 Ns/m

Based on the figure above, it can be observed that the overshoot and settling time of the pitch body are effect on each of the value of spring stiffness on the constant damping value. The overshoot will happen when the tire of the vehicle hit the bump and will cause the impact to entire vehicle body. The amplitude is decreasing with respect to time which means the overshoot decreasing with settling time. In addition, it seen that the amplitude overshoot of pitch body effect will be increase when the spring stiffness is stiffer. When the overshoot is higher, it will affect the settling time of vehicle body to become longer for the vehicle to steady. The lowest the amplitude response overshoot of the pitch body effect, the vehicle will be more comfort when driving. Hence, to improve the comfort of the vehicle, it needed to select the optimal spring stiffness and damping value that suitable with the single seated racing vehicle.

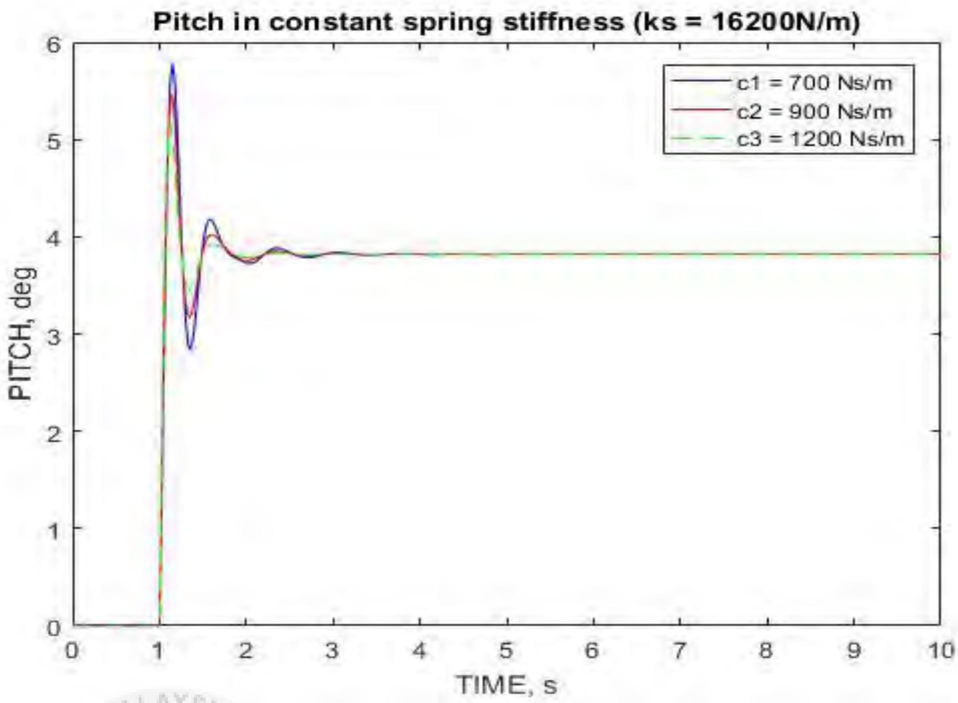


**Figure 4.6:** Graph of Pitch against Time with Ks = 16200 N/m



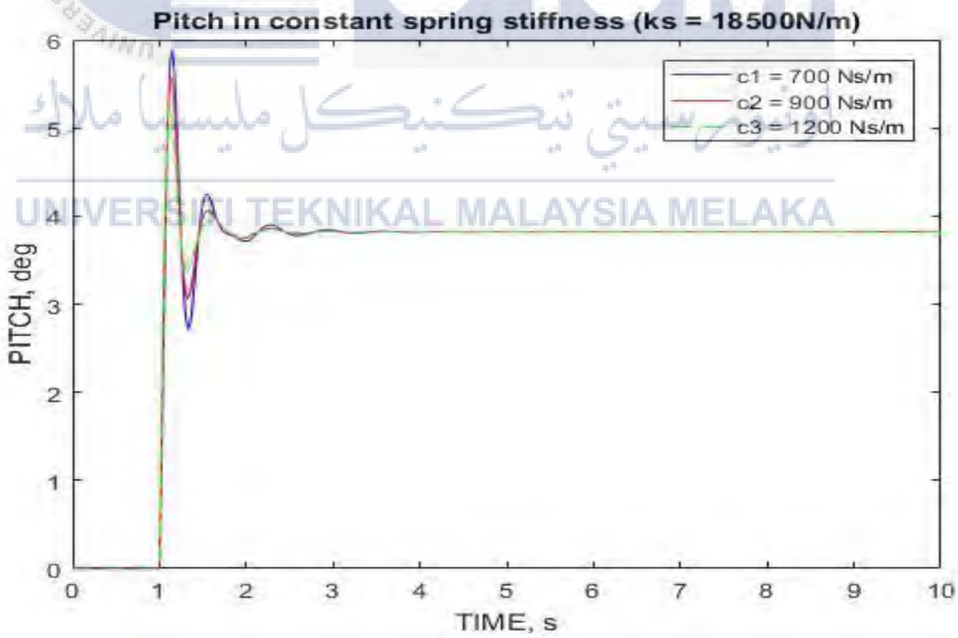


**Figure 4.7:** Graph of Pitch against Time with Ks = 18500 N/m

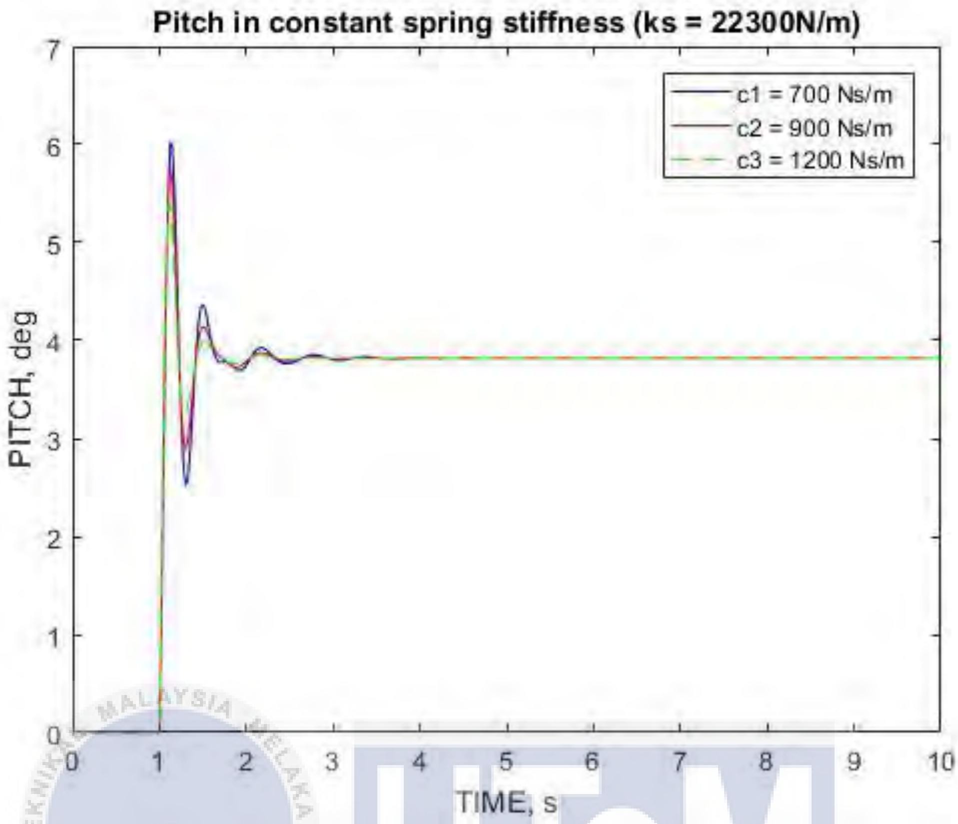


**Figure 4.8:** Graph of Pitch against Time with Ks = 22300 N/m

Based on the result above, it shows that the overshoot and settling time of the pitch body are effect on each of the value of spring stiffness and damping. Furthermore, it seen that the overshoot is higher when the settling time is in the early stage, which is undesirable for good ride and handling of the vehicle. So, the higher the value of damping, the overshoot of pitch body become smaller with shortest settling time that able to reduce the bounce from the vehicle body. It can be noted that when no damping is provided the amplitude of overshoot is increased with time which means passenger will be getting lots of vibration and it can be damage to the vehicles with a lot of wear and tear in the system. Hence, the highest the value of damping will provide the vehicle more comfort during driving through an uneven road.

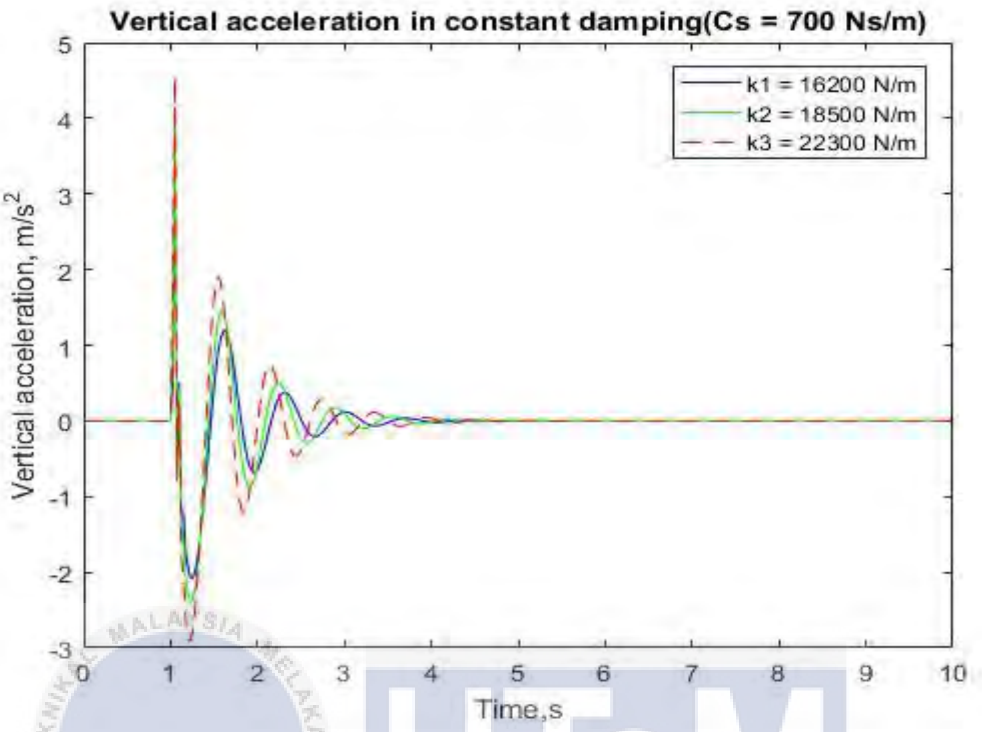


**Figure 4.9:** Graph of Vertical acceleration against Time with Cs = 700 Ns/m



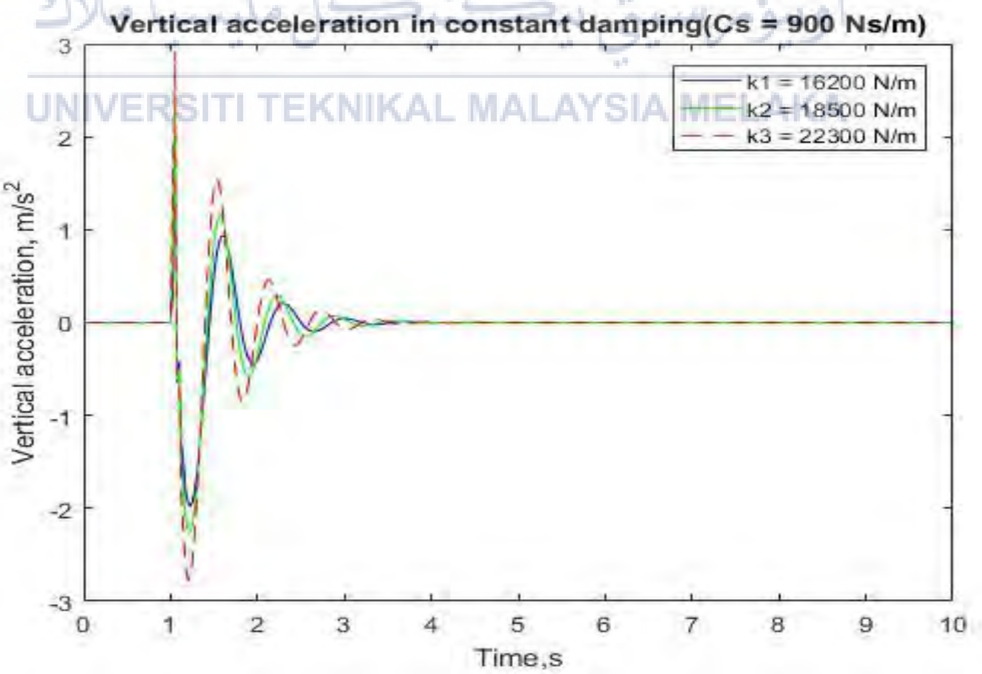


**Figure 4.10:** Graph of Vertical acceleration against Time with Cs = 900 Ns/m

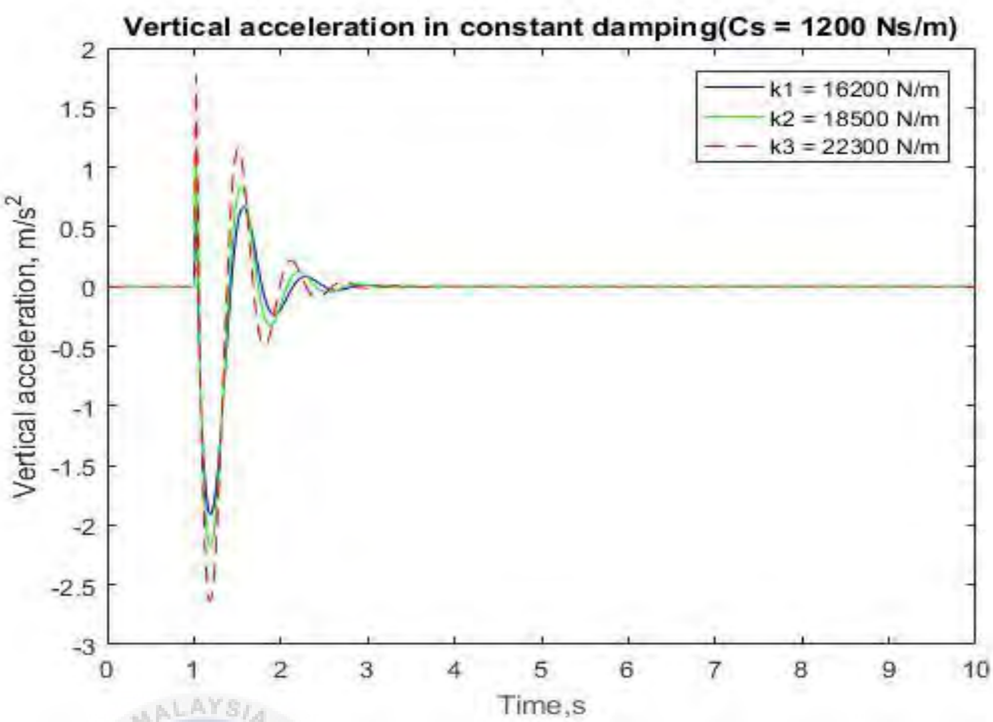


**Figure 4.11:** Graph of Vertical acceleration against Time with Cs = 1200 Ns/m

Based on the Figure 4.9 until Figure 4.11, it can be noticed that the effect of three different value of spring stiffness and damping due to the vertical acceleration of the vehicle body. Basically, vertical acceleration is the movement of centre of gravity on the vehicle body when hit the bump or driving through an uneven road. The vertical acceleration also the important factor to improve the comfort of the vehicle. For this simulation, the vehicle body also will experience the overshoot. Based on the result above, it shows that the higher value of spring stiffness, it will affect the overshoot on vertical body acceleration of vehicle body. When the overshoot on vertical body acceleration is higher, it will reduce the comfort of the vehicle and able to damage the system.

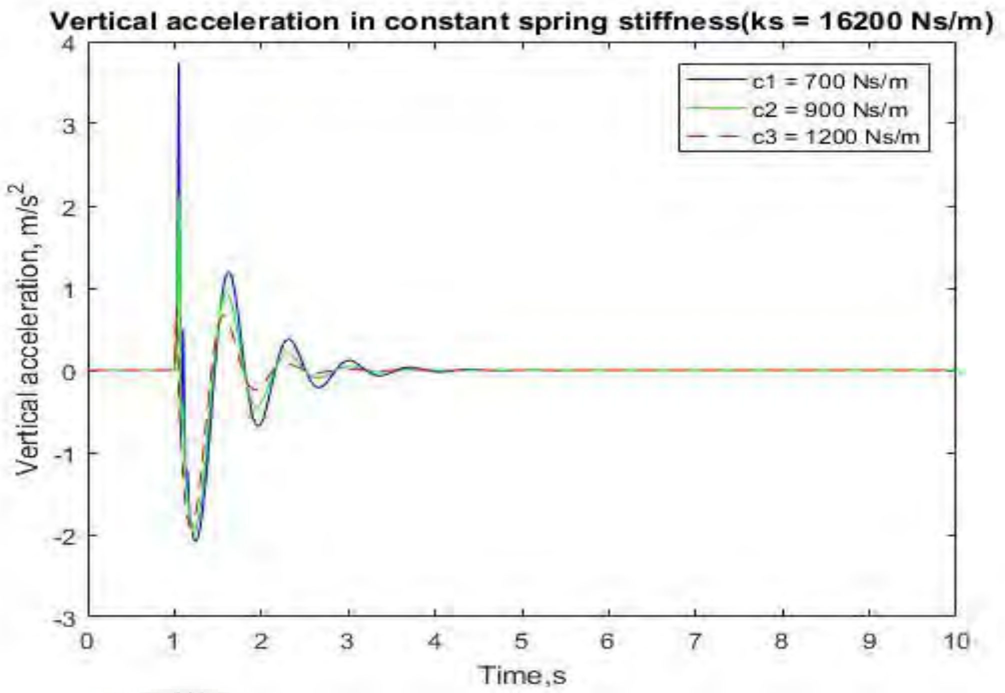


**Figure 4.12:** Graph of Vertical acceleration against Time with Ks = 16200 N/m



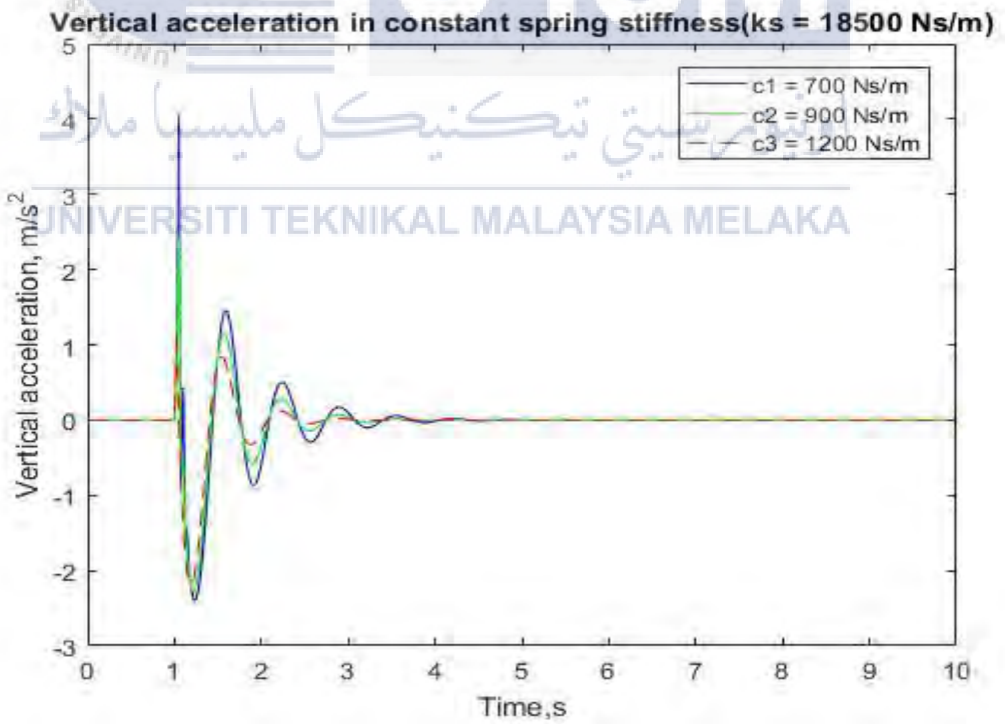


**Figure 4.13:** Graph of Vertical acceleration against Time with Ks = 18500 N/m

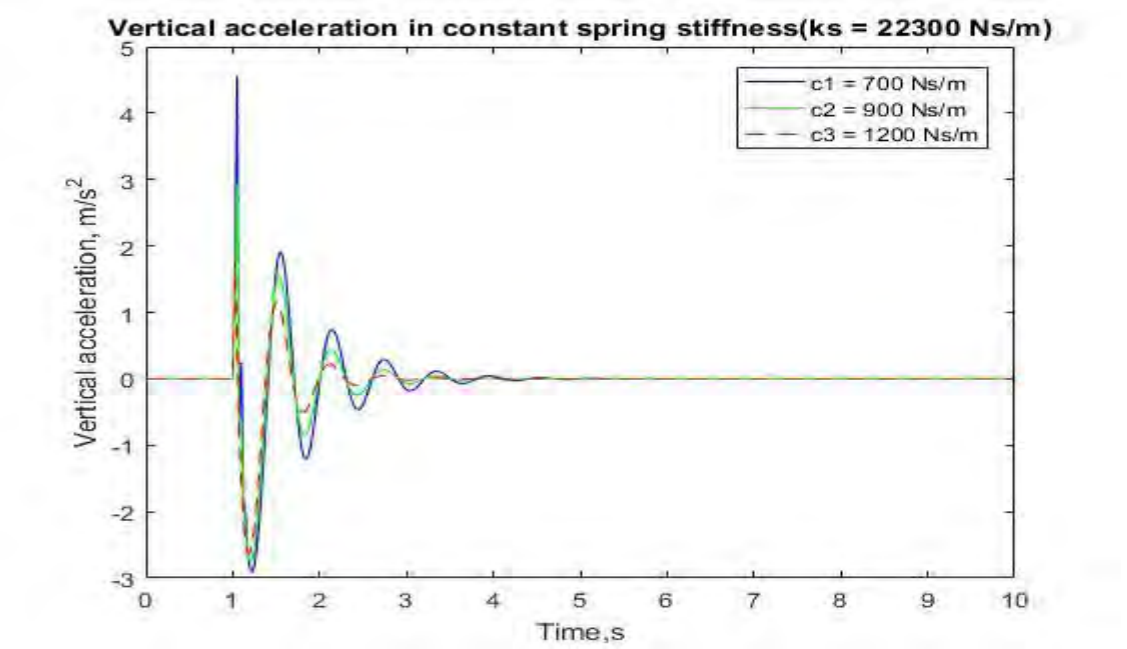


**Figure 4.14:** Graph of Vertical acceleration against Time with Ks = 22300 N/m

From Figure 4.12 until Figure 4.14 above, it can be observed that the effect of vertical body acceleration based on the three different value of damping with constant spring stiffness. Damper is the important component of the suspension system to absorb the force from the road exerted on the vehicle body. So, to improve the comfort of the vehicle, it need to determine the optimal damping value that are able to reduce the overshoot value of vertical body acceleration and shorten the settling time on vehicle body.

### 4.3.2 Roll Dynamic

The result of roll dynamic effect is shown from the Figure 4.15 until Figure 4.17 with different pattern of graph for three different value of spring stiffness which are K1 = 16200 N/m, K2 = 18500 N/m and K3 = 22300 N/m and with three different constant value of damping (C1 = 700 Ns/m, C2 = 900 Ns/m, C3 = 1200 Ns/m). Then, from the Figure 4.18 to Figure 4.20, it shows that the result of the roll dynamic effect with the respect to time in a different value of damping and three different value of constant spring stiffness.

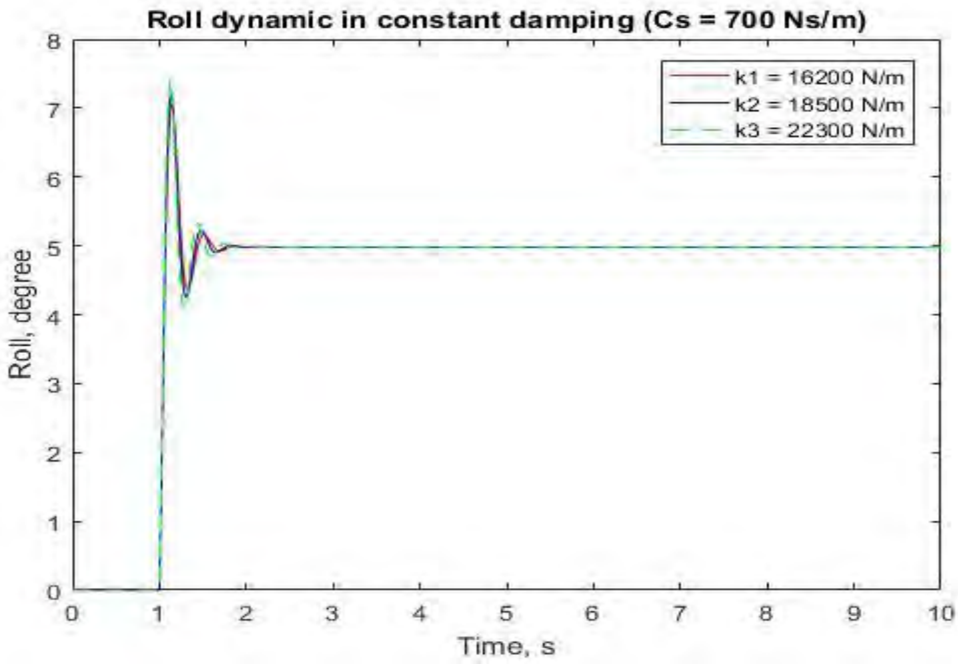


**Figure 4.15:** Graph of Roll against Time with Cs = 700 Ns/m

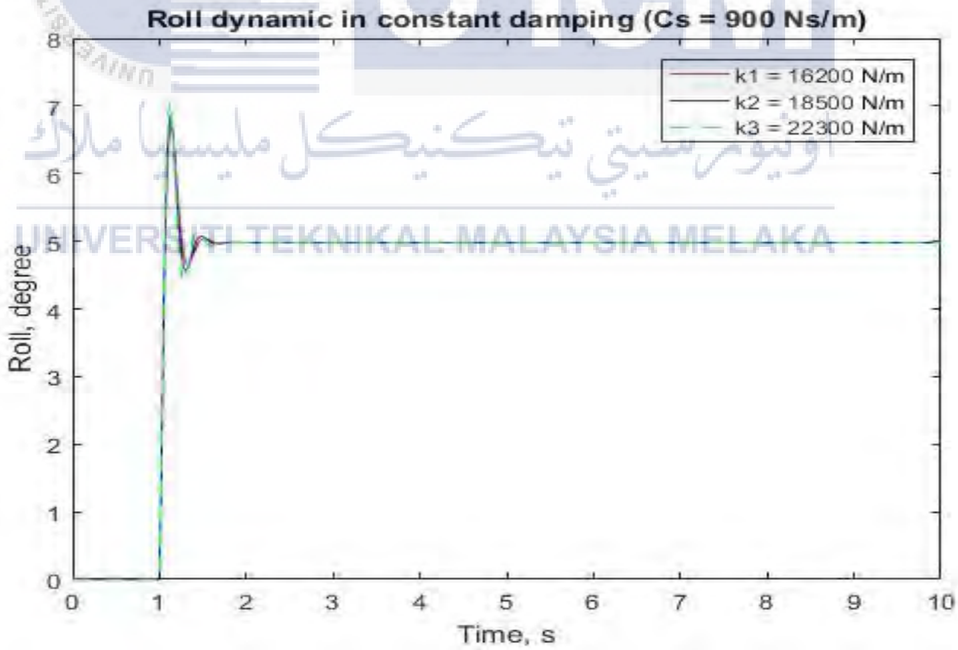




**Figure 4.16:** Graph of Roll against Time with Cs = 900 Ns/m

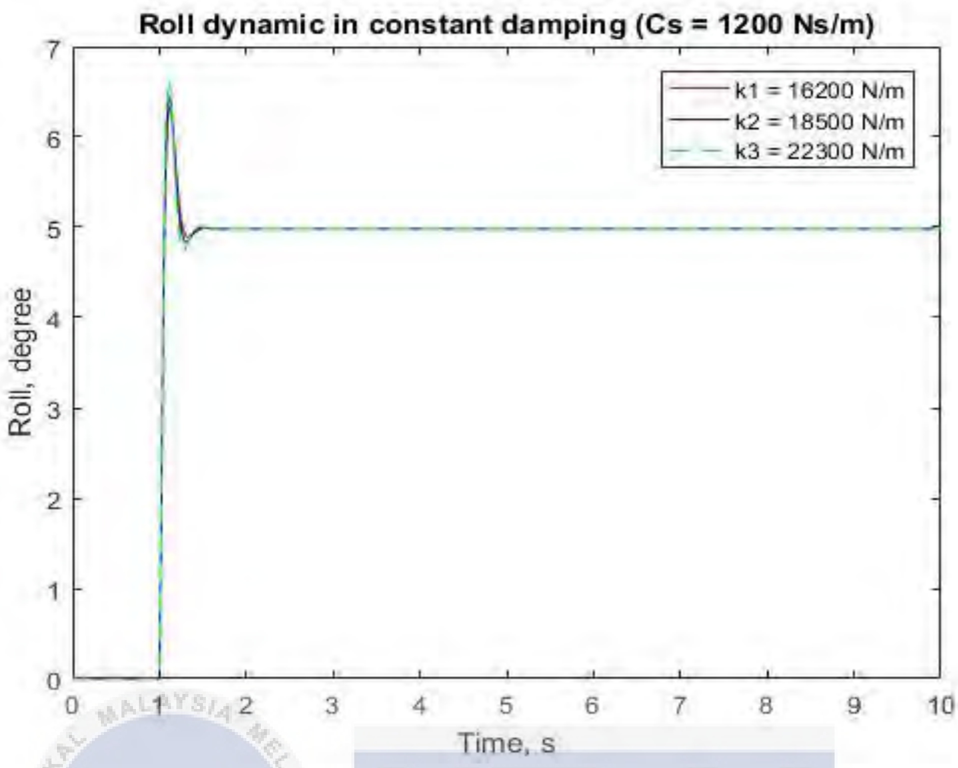


**Figure 4.17:** Graph of Roll against Time with Cs = 1200 Ns/m

Based on the figure above, it can be observed that the overshoot and settling time of the roll dynamic of the vehicle are effect on each of the value of spring stiffness on the constant damping value. The amplitude is decreasing with respect to time which means the overshoot decreasing with settling time. Furthermore, it seen that the amplitude overshoot of roll dynamic effect will be increase when the spring stiffness is stiffer. When the overshoot is higher, it will affect the settling time of vehicle body to become longer for the vehicle to steady. The lowest the amplitude response overshoot of the roll dynamic effect, the vehicle will be more comfort. Hence, to improve the comfort of the vehicle, it needed to select the optimal spring stiffness and damping value that suitable with the single seated racing vehicle.

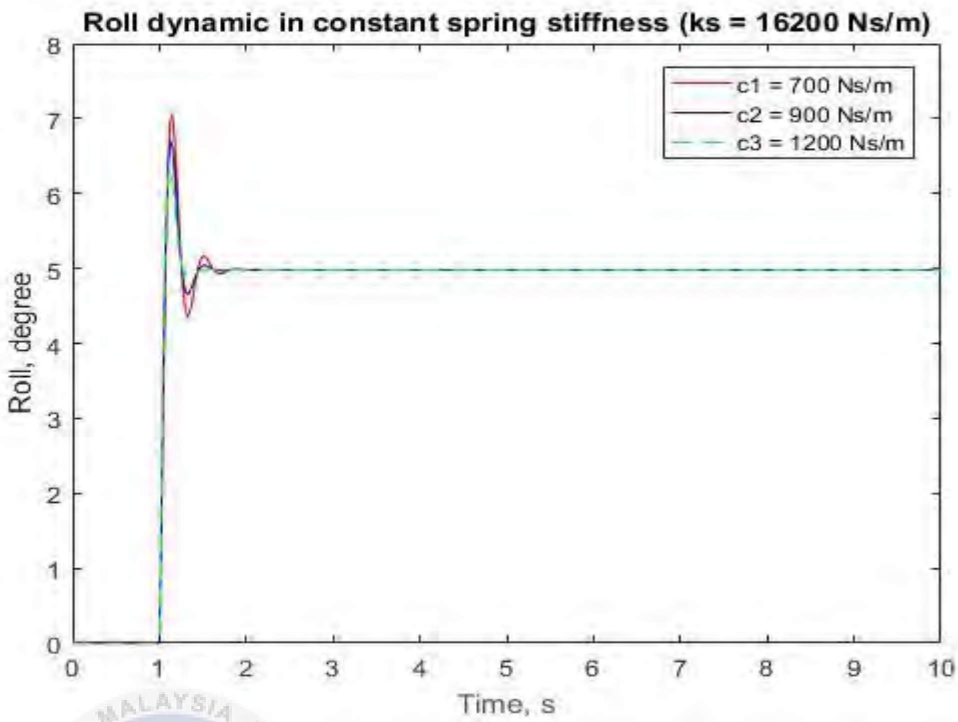


**Figure 4.18:** Graph of Roll against Time with Ks = 16200 N/m

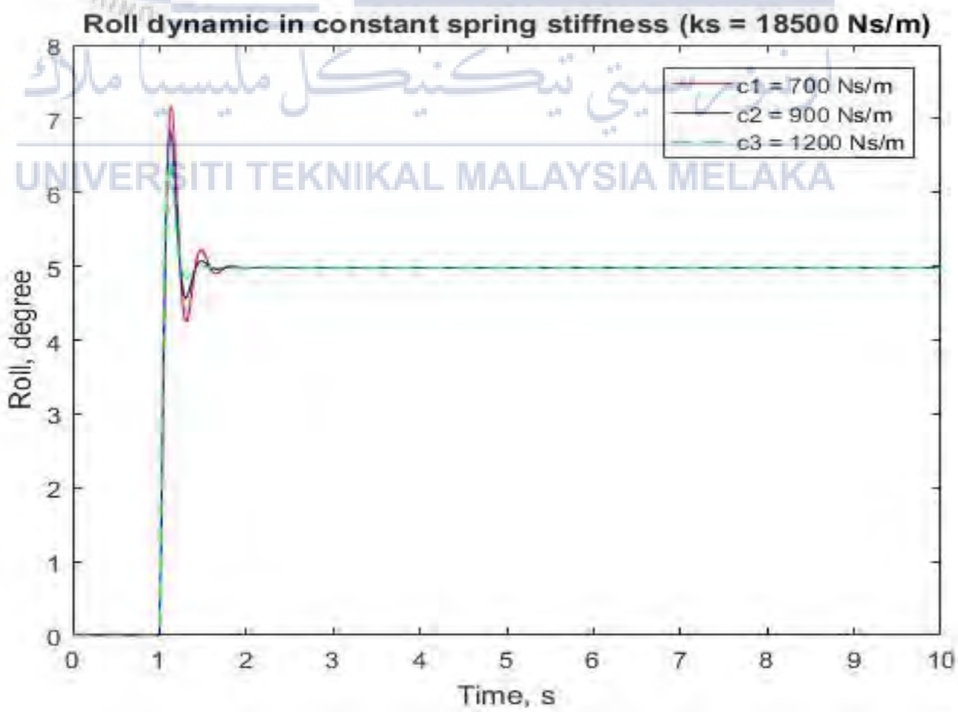


**Figure 4.19:** Graph of Roll against Time with Ks = 18500 N/m

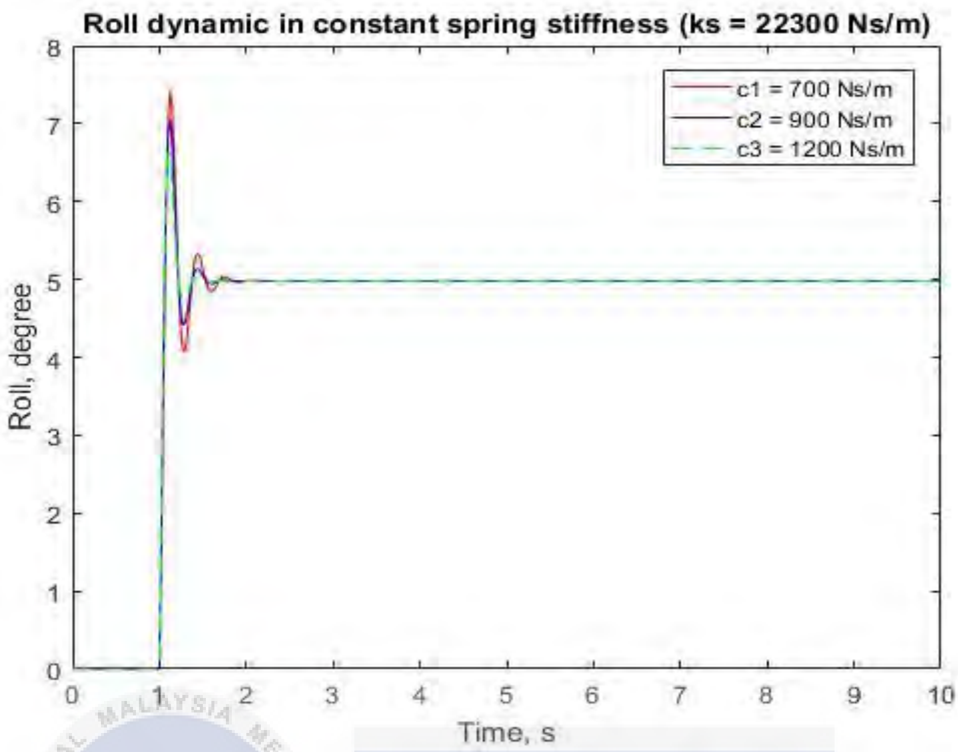


**Figure 4.20:** Graph of Roll against Time with Ks = 22300 N/m

Based on the result above, it shows that the overshoot and settling time of the roll dynamic are effect on each of the value of spring stiffness and damping. Apart from that, it seen that the overshoot is higher when the settling time is in the early stage, which is undesirable for good ride and handling of the vehicle. So, the higher the value of damping, the overshoot of roll dynamic become smaller with shortest settling time that able to reduce the bounce from the vehicle body. Hence, the highest the value of damping will provide the vehicle more comfort during driving through an uneven road.

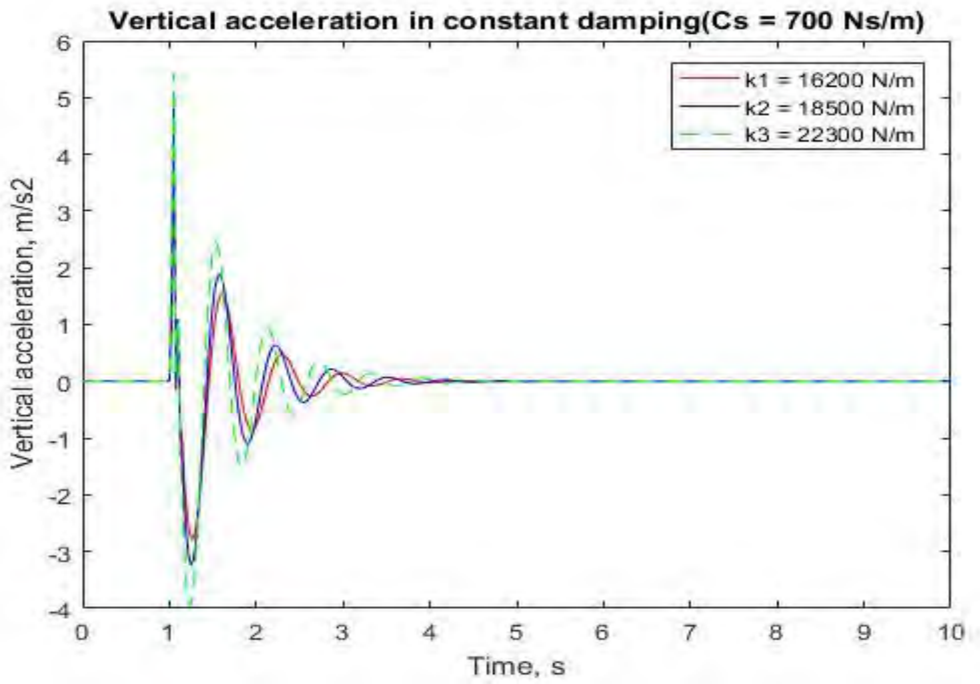


**Figure 4.21:** Graph of Vertical acceleration against Time with Cs = 700 Ns/m



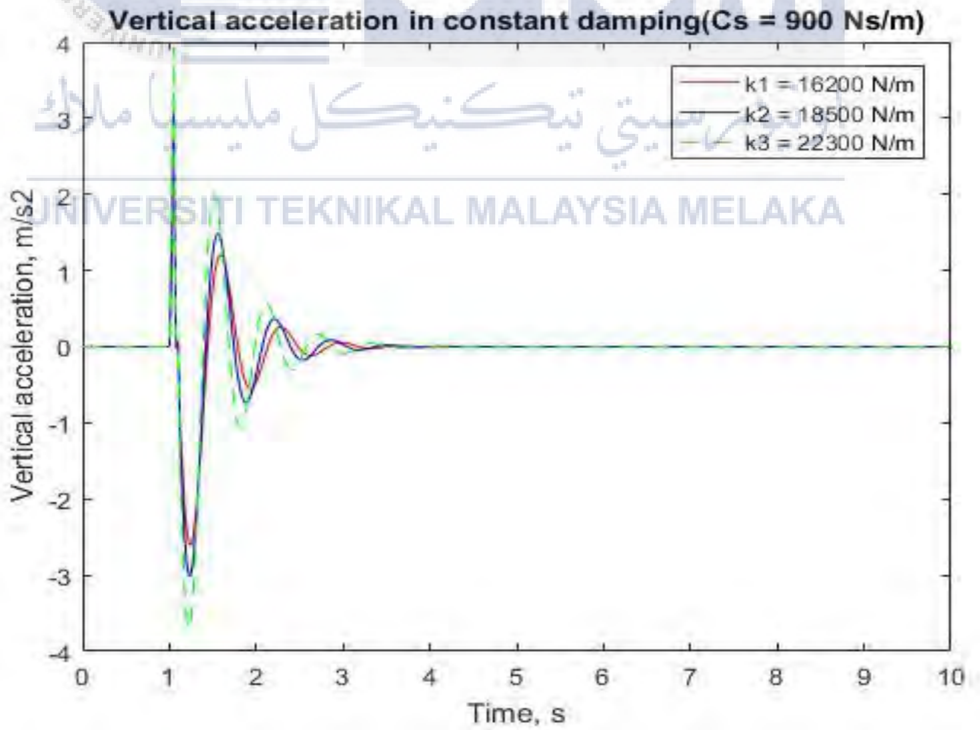


**Figure 4.22:** Graph of Vertical acceleration against Time with Cs = 900 Ns/m

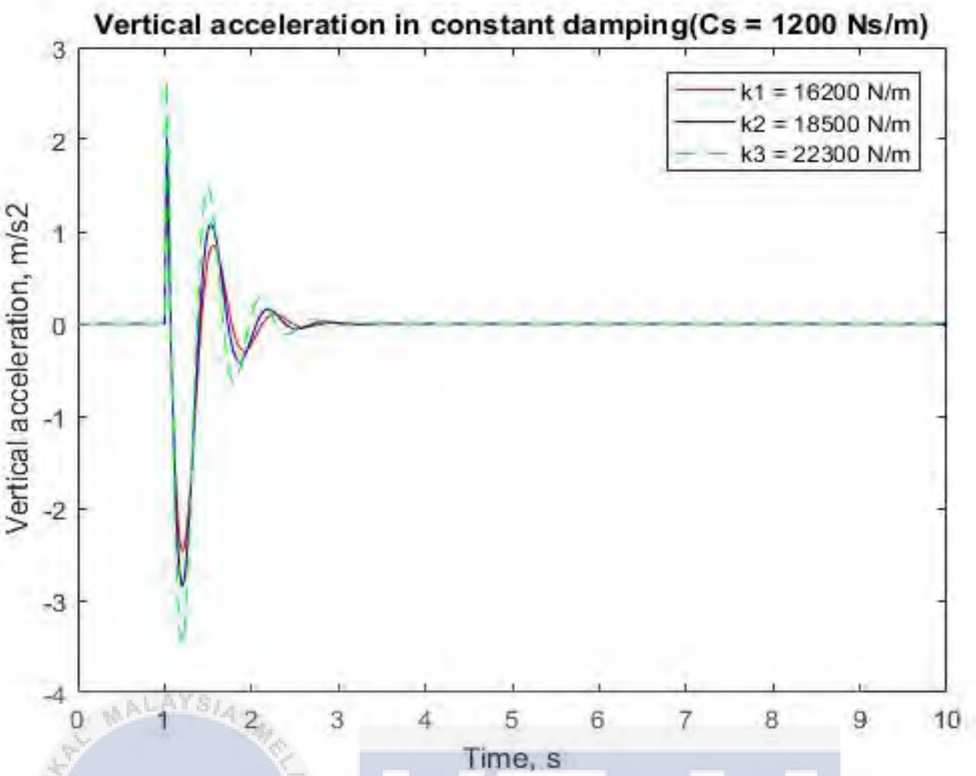


**Figure 4.23:** Graph of Vertical acceleration against Time with Cs = 1200 Ns/m

Based on the Figure 4.21 until Figure 4.23, it shows that the effect of three different value of spring stiffness and damping due to the vertical acceleration of the vehicle body. Basically, vertical acceleration is the movement of centre of gravity on the vehicle body when hit the bump or driving through an uneven road. The vertical acceleration also the important factor to improve the comfort of the vehicle. For this simulation, the vehicle body also will experience the overshoot. Based on the result above, it shows that the higher value of spring stiffness, it will affect the overshoot on vertical body acceleration of vehicle body. When the overshoot on vertical body acceleration is higher, it will reduce the comfort of the vehicle and able to damage the system.

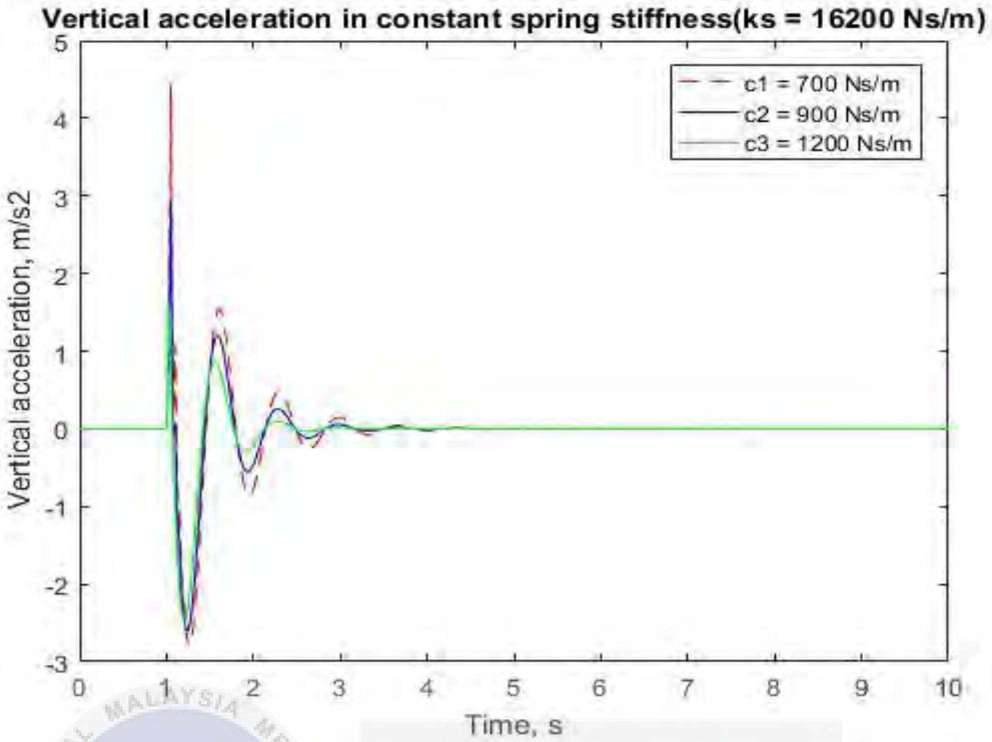


**Figure 4.24:** Graph of Vertical acceleration against Time with Ks = 16200 N/m


**Figure 4.25:** Graph of Vertical acceleration against Time with Ks = 18500 N/m

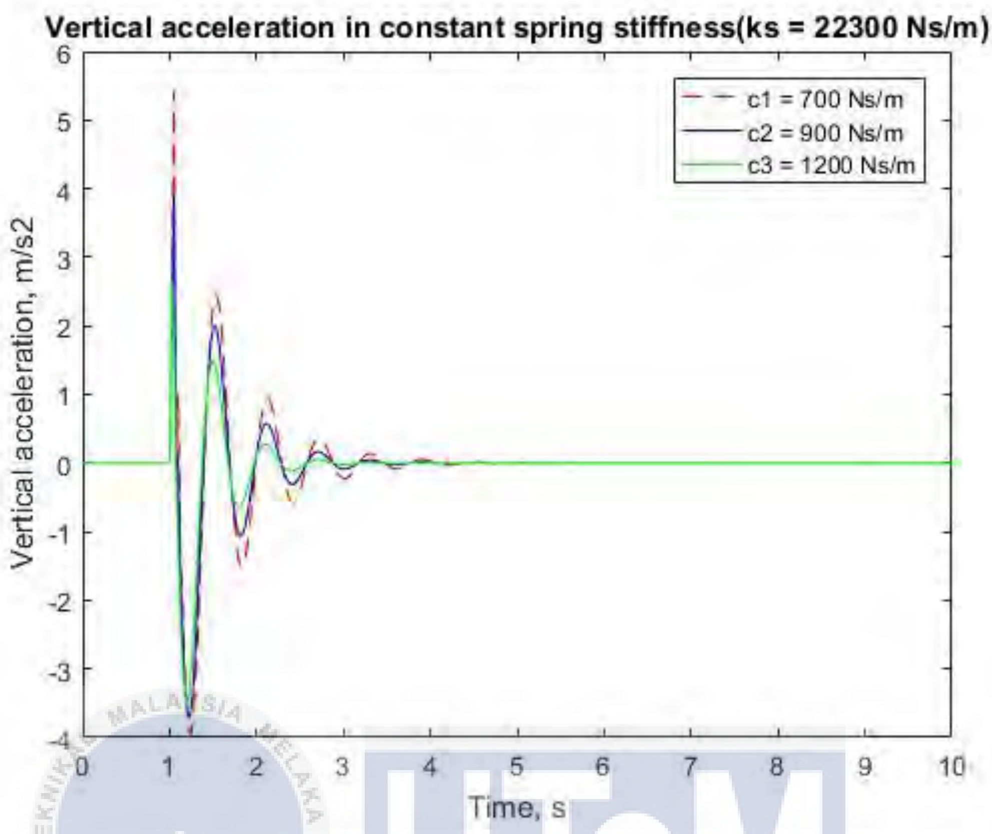


**Figure 4.26:** Graph of Vertical acceleration against Time with Ks = 22300 N/m

From Figure 4.24 until Figure 4.26 above, it can be observed that the effect of vertical body acceleration based on the three different value of damping with constant spring stiffness. Damper is the important component of the suspension system to absorb the force from the road exerted on the vehicle body. So, to improve the comfort of the vehicle, it need to determine the optimal damping value that can reduce the overshoot value of vertical body acceleration and shorten the settling time on vehicle body.

# CHAPTER 5

# CONCLUSION AND RECOMMENDATION



## 5.1 CONCLUSION

Once the designed of the suspension system and its components have been completed, the engineering analysis are performed on the parts. . Two types of methods of analysis are employed which is Finite Element Analysis using CATIA software and numerical coding using Matlab Simulink software.

By using CATIA software, there are two analysis that have been performed which is Von Misses stress and translational displacement analysis. The result from the Von Misses analysis shows that the highest stress when the load of 1000N is applied to the product is 0.337 MPa, while the lowest stress obtained is 90.7 Pa.

The second analysis is to define the dynamic movement displacement of the suspension system when the load is applied. The result suggests that the maximum displacement of the suspension is 4.18 mm when the load is applied on the suspension. This result suggests that if the suspension is over bended more than 4.18 mm, the arm of the suspension might be damaged.

MATLAB Simulink software is the second method to make an analysis to the designed suspension system. There are also two types of analysis had been conducted which is to find the pitch body and roll dynamic of the vehicle by using the designed suspension. There are several parameters which is three different parameters for the damping value, Cs, and spring stiffness, Ks, that are used in this analysis to find the optimum parameter for the single seated racing vehicle. Based on the result, the overshoot can be seen within the settling time for the vehicle return to the normal state. It clearly analyses that the higher the overshoot, the less comfort of the vehicle and the optimum parameter for the racing vehicle can be selected.

## 5.2 RECOMMENDATION

After design and analysis of this project, it is recommended that the shock absorber is placed at each of the tire side because it can provide more comfort in the vehicle compare to a single shock absorber at both front and rear of the vehicle. It is also recommended to make an analysis by using an ANSYS software instead of only using CATIA software. This is due to the ANSYS software is more accurate in making an analysis compare to CATIA software. In addition, for further study on suspension system on a racing vehicle, the 4-DOF ride model are recommended to replace with the 7-DOF ride model full car model. This full car model is required to validate first to make sure that this full car model representing real application model simulation. Last but not least, the parameter that had been using to compare the pitch body and the roll dynamic should be added instead of three different parameters to find a the most optimum parameter for the single seated racing vehicle.

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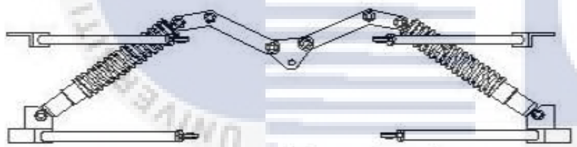
# APPENDICES

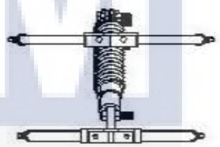
## Gantt Chart of the project

| Activity | Semester 1 Session 16/17 | | | | | | | | | | | | | | | Semester 2 Session 16/17 | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Week | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| PSM 1 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 |
| Project Planning | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Chapter 1: Introduction | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Chapter 2: Literature Review | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Chapter 3: Methodology | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Chapter 4: Result | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Chapter 5: Conclusion and Recommendation | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Design | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Analysis | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Progress Report Submission | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Draft Report Submission (PSM 1) | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Final Report Submission | | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |

Top view



Front view



Side view

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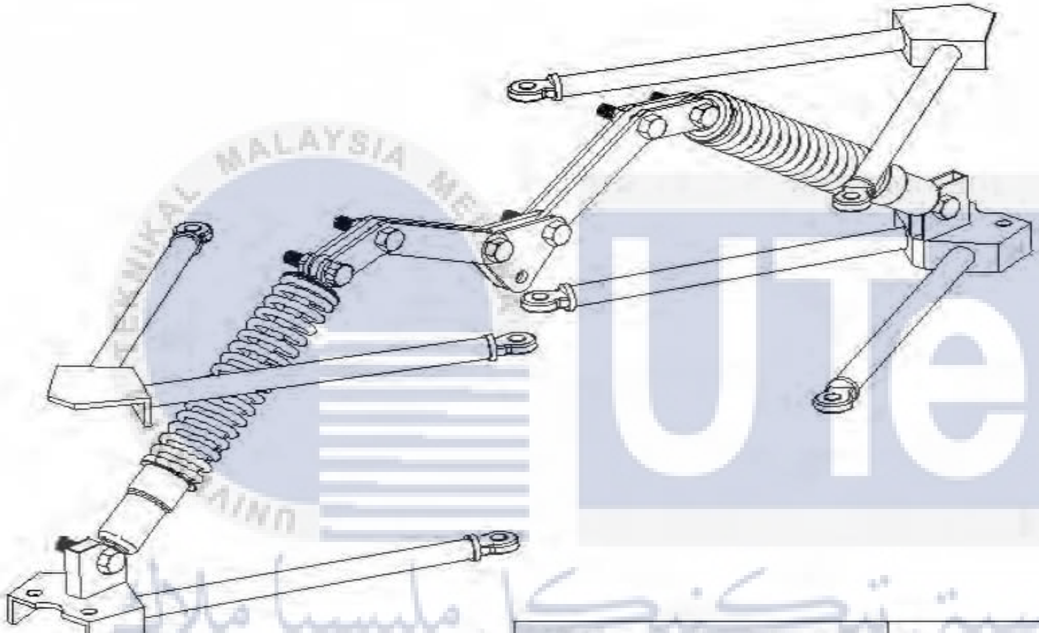
| DESIGNED BY Zulhafizi | | SCALE 1:1 | SHEET 1/1 |



Isometric view

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| DRAWN BY Zulhafizi | DATE 19/12/2016 | DRAWING TITLE Front Suspension | |
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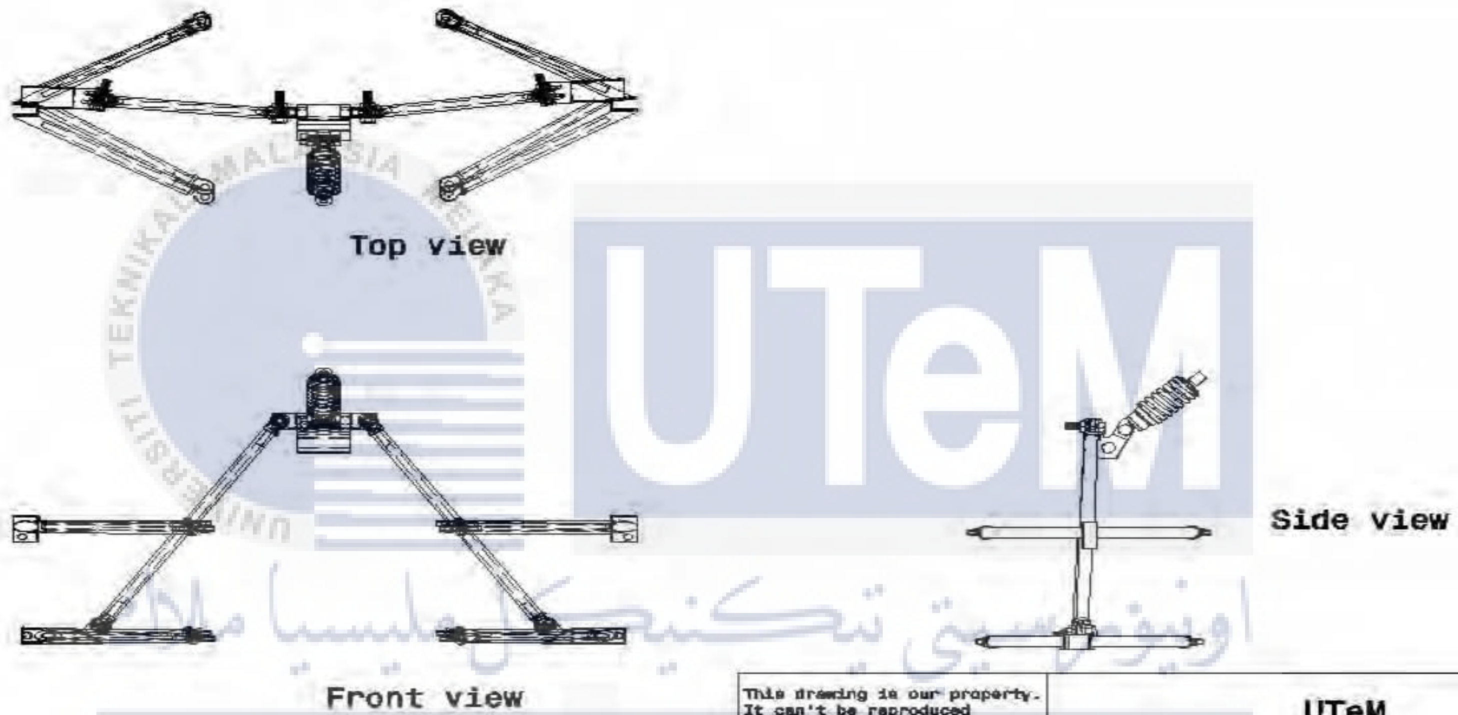




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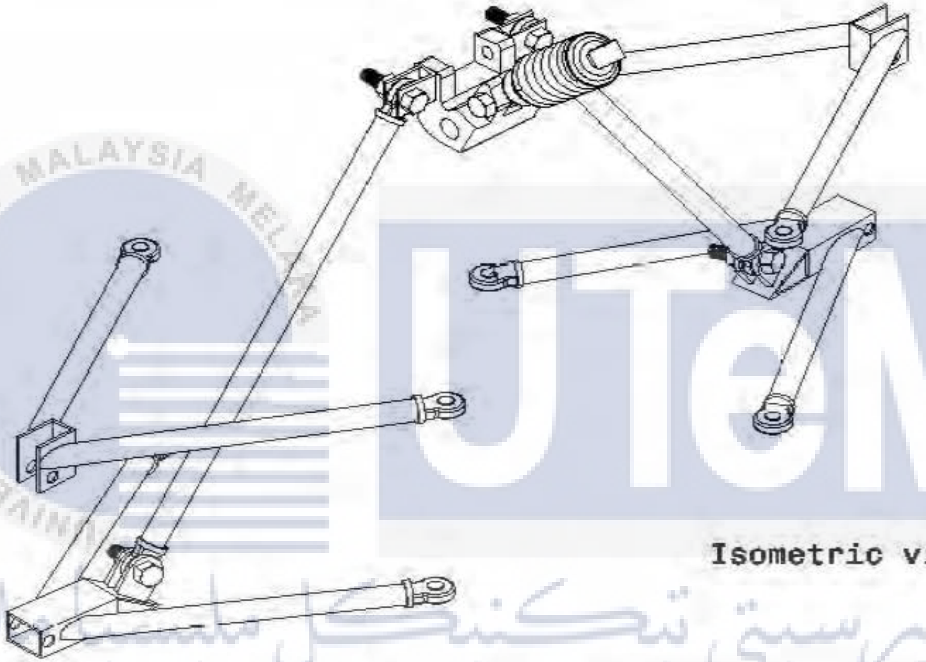

Isometric view

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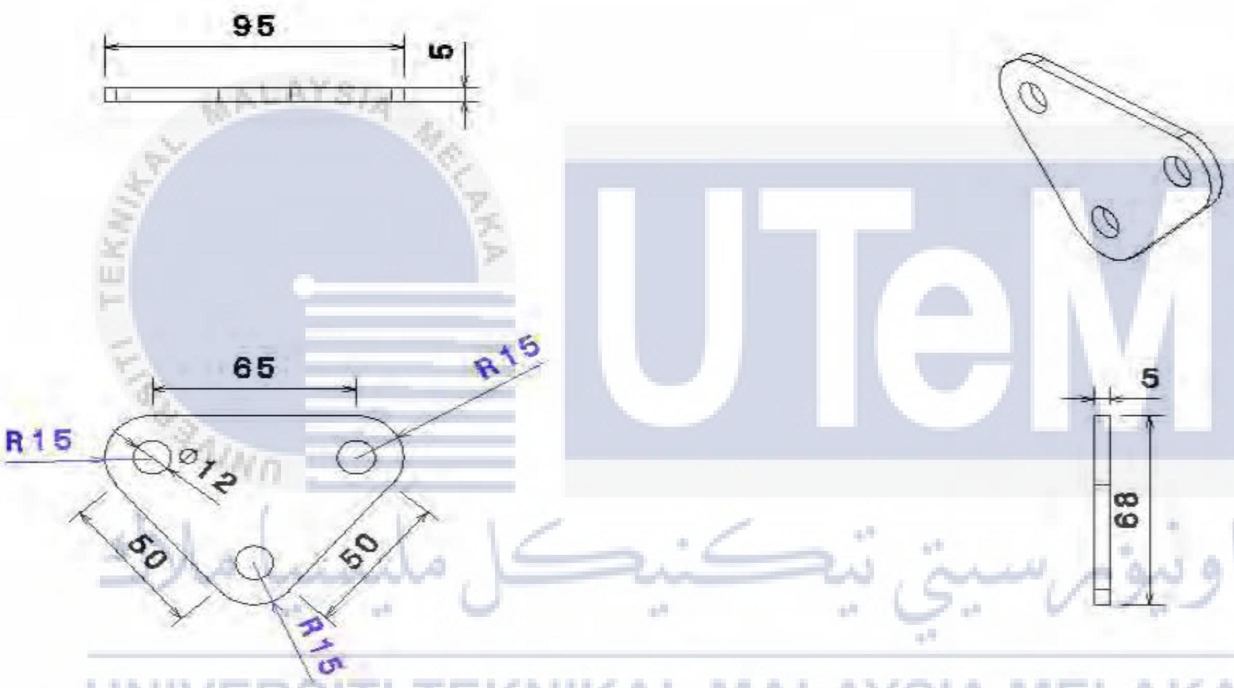


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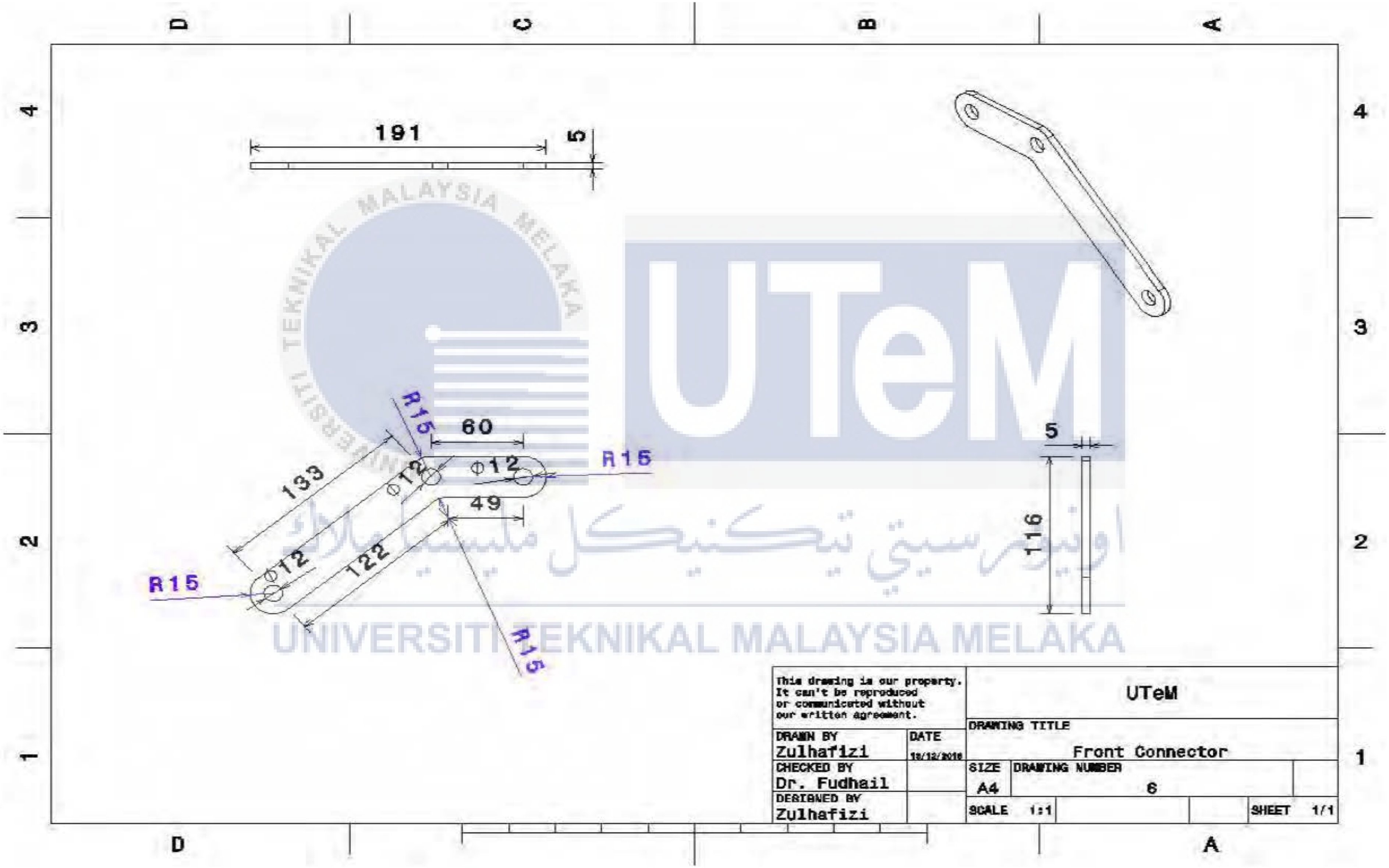
191
5
60
R15
R15
ϕ12
ϕ12
133
49
ϕ12
122
R15
R15
5
116
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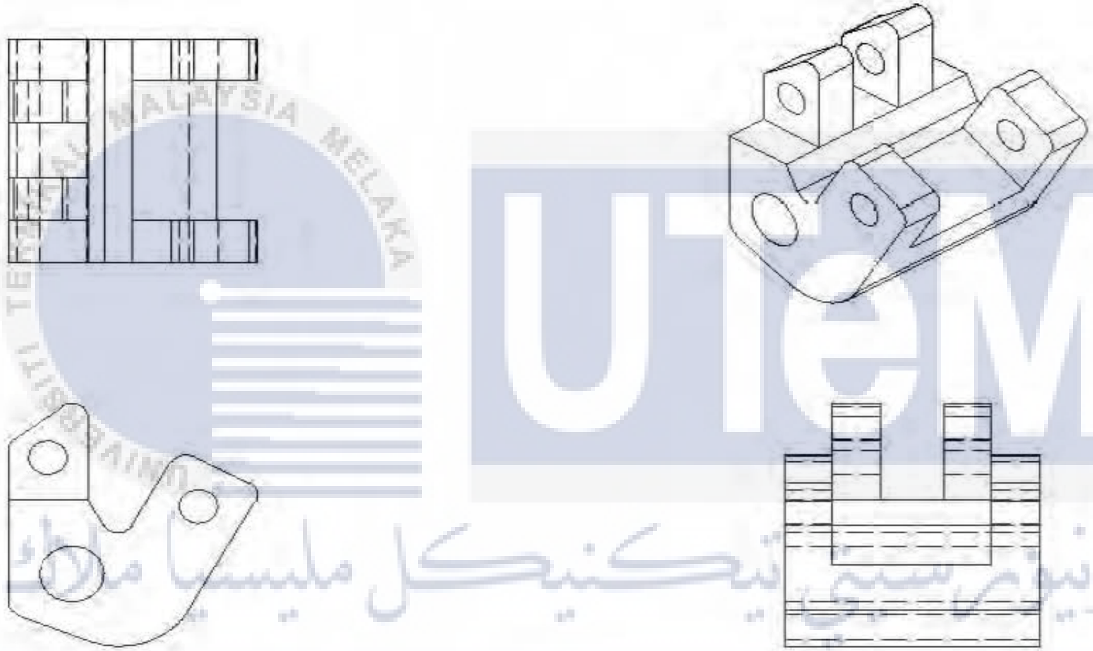
UTeM
DRAWING TITLE
Front Connector
DRAWN BY
Zulhafizi
DATE
19/12/2016
CHECKED BY
Dr. Fudhail
DESIGNED BY
Zulhafizi
SIZE
A4
DRAWING NUMBER
6
SCALE 1:1
SHEET 1/1



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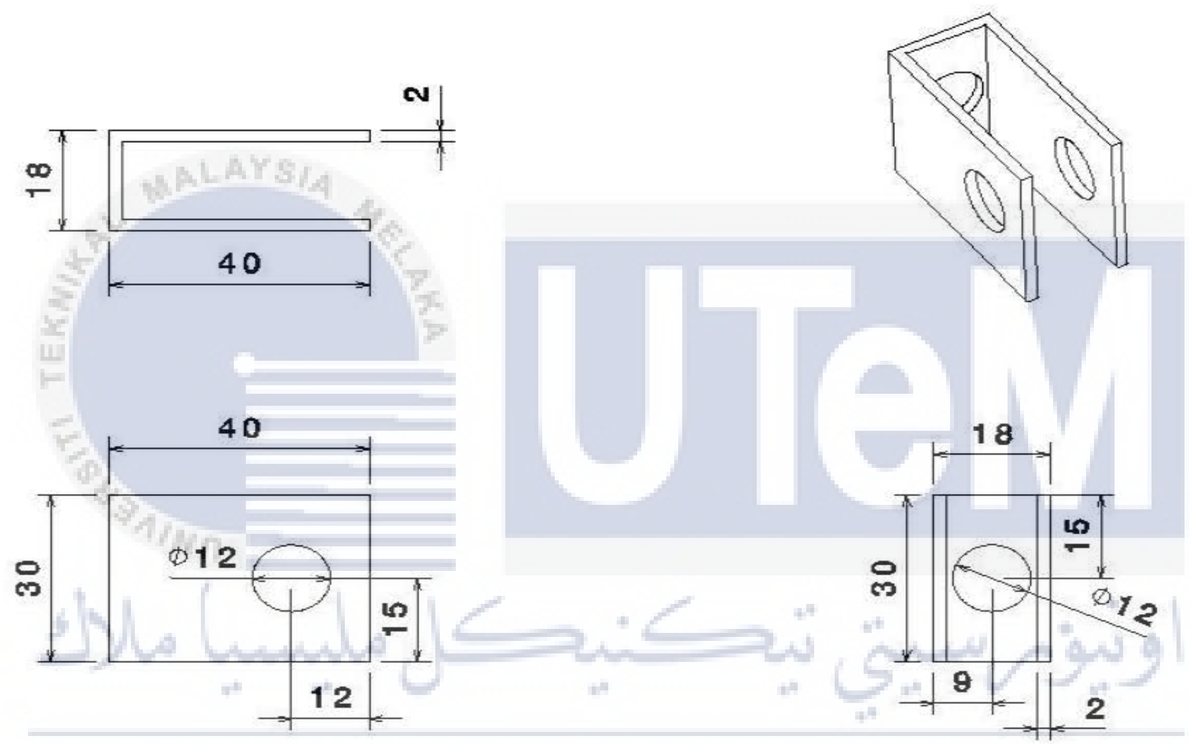




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