Design and Analysis of Suspension System for a Formula SAE Race Car

by

Fahmi bin Ali

Dissertation submitted in partial fulfilment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

DECEMBER 2008

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CERTIFICATION OF APPROVAL

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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,

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TRONOH, PERAK

JULY 2008

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.

FAHMI BIN ALI

ABSTRACT

The objective of this project is to further optimize the suspension system of a Formula SAE race car with steering system included. The suspension system is designed based on unequal length double wishbone suspension system. Several changes had been made for the new car with the usage of hybrid composite-spaced frame chassis and single cylinder engine. Thus, new design concepts has been introduced to suit the changes made for the vehicle which include the changes in mounting points, weight distribution, suspension kinematics plane, and steering geometry.

The scope of study consists of modeling the suspension and steering components by using computer aided software such as CATIA. In addition, the Finite Element Analysis (FEA) is performed by using CATIA which could give instantaneous yet accurate results. Dynamics analysis will compromise the usage of ADAMSCAR software which can simulate the whole suspension and steering system behavior according to the track layout which will make better understanding regarding the study. Although the fabrications of the actual product will not being carried out, the fabrication method will be inserted together in this study as reference for future planning.

Based on the designing and analysis performed, the calculated roll center height and static camber angle of the vehicle at the static position is -68mm from the ground and -0.5 degree respectively. In addition, the maximum lateral load transfer being transferred during cornering with radius of 7.5 meters is 91.82 N. The dynamics analysis performed in ADAMSCARS shows remarkable results in open loop step steer simulation. These results provide better understanding of the vehicle performance during the autocross events.

ACKNOWLEDGEMENT

Alhamdulillah. Thanks and gratitude to Allah S.W.T for the spirit, wisdom, knowledge and understanding for me in journey to complete my final project tasks and report. For the strength and health, He provides me that I am able to go through every single circumstances and problem during the development.

I would like to express my great thanks to my project supervisor, AP. Dr. Abdul Rashid Bin Abdul Aziz and Ir. Dr, Masri Bin Baharom for his advice, my co-supervisor, Mr Mohd. Syaifuddin Bin Mohd for his guidance and understanding throughout the development of this project. This project development would not have been possible without the assistance and guidance of certain individuals whose contributions have helped in its completion.

Also, to my beloved parents who always provide me with a lot of supports and prayers that strengthen me. Not forgotten, to my friends and peers for their unconditional support and love for me.

Last but not least, I would like to thank all persons who have contributed to this project in one way or another especially the UTP technicians.

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CHAPTER 1

INTRODUCTION

1.1 Background of Study

Small race cars competitions with a 1/3 scale of formula one race cars had been organized globally for a past 10 years. The Society of Automotive Engineers (SAE) is the organizer of all the events including the Formula SAE (FSAE) where the students congregate as a team in designing, developing, and race in the autocross event with the race car they built. The team consists of chassis, suspension and steering, power train, braking, and many more. Suspension and steering system plays a crucial role in FSAE race car, providing sufficient handling and stability of the vehicle and driver. Besides, general characteristics of the car are determined by the design of the suspension and steering systems and mechanisms. The suspension design is governed by the following regulations [10]:

a) Suspension travel of 25.4mm (Bound and Compression)
b) Minimum wheelbase of 1525mm
c) Track width difference of not more than 75%

The design stage of the suspension and steering system on FSAE race car consists of designing the on the whole positioning of components. The components comprise the spring and damper, uprights, control arm (wishbone), anti-roll bar, steering rack and pinion system, tie-rod, and other sub-parts exists in the system.

The Suspension and Steering system has been optimized from the previous design at this stage. Optimization includes identify flaws and improvements on several parts in the system, analysis on critical parts, and fabrication process or method. Some parts of the suspension components have been optimized by redesigning the parts to get minimum usage of metal billets (block). Specifically, components such as uprights have been redesigned for the ease of fabrication and still maintain the target of having a reliable and optimum design.

Ergonomics of driver inside the cabin is essential and vital in constructing a race car. Thus, the study of ergonomics had been conducted to get the optimum positions of the driver inside the cabin to increase the driver's stability and concentration during race day.

The analysis stage emphasizes on the applicability and reliability of the design of suspension and steering system to the autocross track layout. Kinematics and dynamics results had been done and the findings are been found impressive. Stress analysis had been done to several critical components and the results are remarkable. Finalizing the changes made, the scope of progress also covers the judgment that had been based upon for the new design implementation.

Discussion and brainstorming had been executed among other departments to get better understanding and clearer view of the constraints and limitations in designing. From the brainstorming, packaging of the system can be achieved smoothly without the needs of major changes in design.

1.2 Problem Statement

1.2.1 Problem Identification

The designing process of the suspension and steering system should include the consideration of the overall track layout. The FSAE competition track consists of various sections such as acceleration and deceleration, steady-state cornering, and transient cornering. Previous design indicates lack of analysis to the suspension kinematics and inadequate research of steering system related to the performance of the race car on FSAE competition track layout.

The usage of ADAMSCAR software aids in further understanding the kinematics and dynamics characteristics of the vehicle. Since previous vehicle did not perform the simulation on ADAMSCAR, there were no data exist to compare the overall performance of the car. In addition, performing simulation in ADAMSCAR was very difficult and need full understanding to interpret the findings.

From the tests that the vehicle will go through, the suspension is one of the most vital factors of the vehicle design. The design for the front and rear suspension, the selected shocks, and the materials that been chosen will determine how well the vehicle will perform in the aforementioned tasks. Since the new design of the vehicle will be using hybrid composite spaced-frame chassis, the suspension system need to be redesigned to fit the chassis without neglecting reliability, and performance of the vehicle.

In the 2007 Formula SAE UTP upright design, the manufacturing process involved is too detail and involves many steps that are sensitive. After considering the process involved and the facilities availability, an enhanced design that brings forth a simpler manufacturing process is needed. Some minor changes on the hard points and mounting points are made to suit the requirements and constraints agreed among other departments.

In terms of cost analysis and material savings, the new design illustrates the minimum material wastage and cost efficiency on the raw materials billets (block) used. The decision making is lead by the cost evaluation which requires lowest cost possible to construct a Formula SAE race car without neglecting the design feasibility and reliability.

1.2.2 Significance of the Project

The importance of this project will determine the overall performance of the vehicle. Suspension system acts as the handling mechanism of the car. The system provides supports and also stability.

1.3 Objective and Scope of Study

The main objective of this project/study is to design and develop a working suspension and steering system for hybrid composite-space frame chassis which is to be applied on a FSAE race car.

The design stage is narrowed down by the usage of CAD modeling software such as CATIA. With the aid from CATIA, the configuration of the system will be transparent and understandable during the packaging and assembly process.

The analysis stage involves Finite Element Analysis and kinematics and dynamics of the suspension and steering system. In implementing the Finite Element Analysis, better understanding of the stress point and possible failure point can be determined and further modifications can be made. This process also gives supplementary optimization on weight reduction and strength to mass ratio for better performance and maneuverability.

The suspension and steering kinematics can be simulated by using ADAMSCAR software. The software simulates the kinematics motion of the system design. From the displacements (angular) the simulation is able to display vehicle geometries that are needed by data processing and design optimization.

The process of studying the previous and other reference designs assist to understand the vehicle dynamics and types of parameters being used better. These parameters will be implemented to the design and affect the handling characteristics of the FSAE race car.

CHAPTER 2

LITERATURE REVIEW

2.1 Suspension Terminology

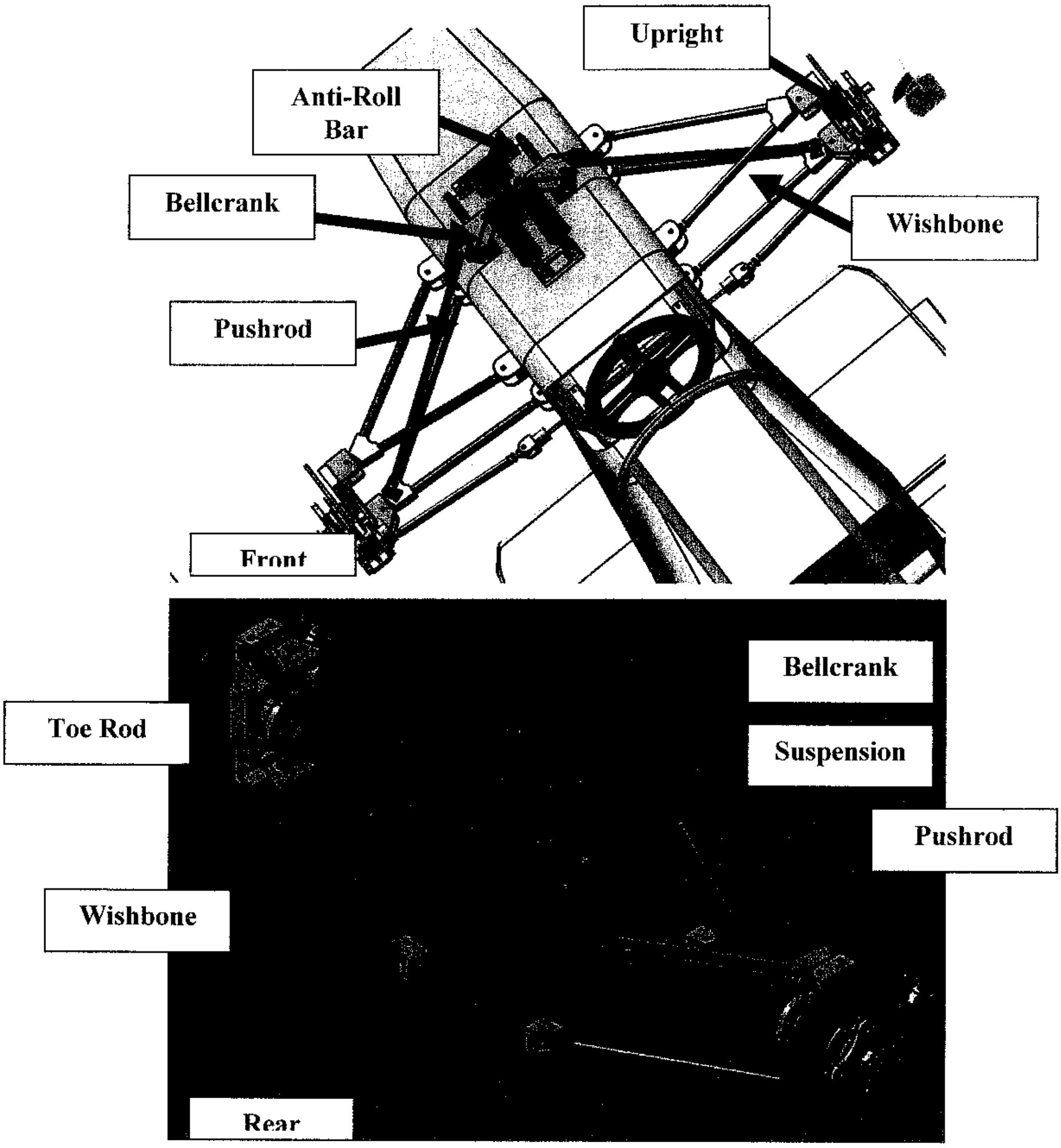


Figure 2.1: Suspension Terminology

2.2 Geometry Parameters

2.2.1 Camber Angle

Camber angle is regarded as the inclination of the wheel plane to the vertical [4]. Negative camber inclines the top of the tire toward the centerline of the vehicle as seen in Figure 2.2 and positive camber inclines the top of the tire away from the centerline.

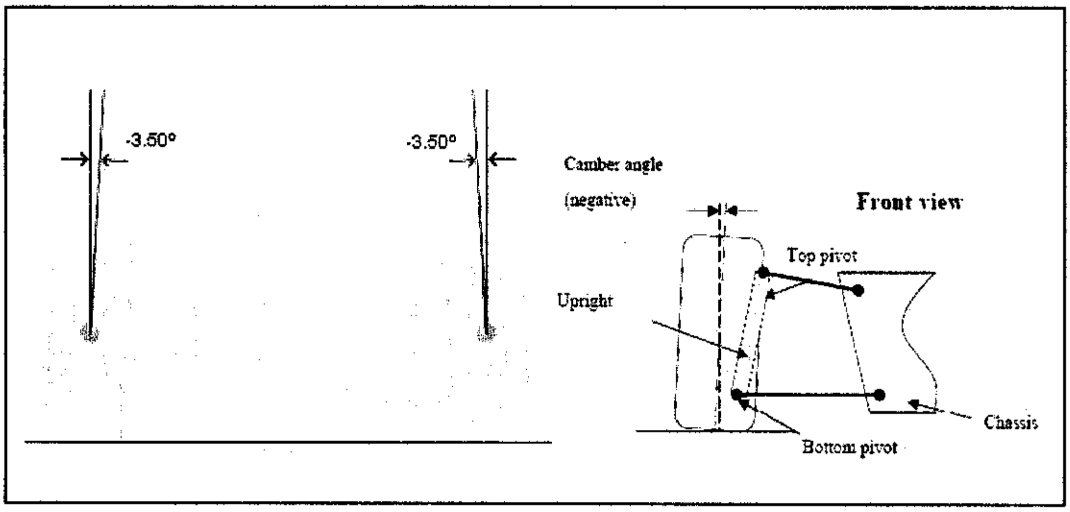


Figure 2.2: Camber Angle

A small amount of negative camber of up to 1.5 degrees it is recommended in order to induce camber thrust [3]. However, changes in camber should be kept at minimum during chassis roll in order to reduce the loss of camber thrust and the change in wheel track load distribution during cornering.

2.2.2 Rate of Camber Change

The rate of camber change is the change of camber angle per unit vertical displacement of the wheel centre relative to the sprung mass [4].

2.2.3 Caster Angle

Caster is the inclination of the kingpin axis in the Z-X plane, measured to the vertical center of the wheel, and is positive, clockwise.

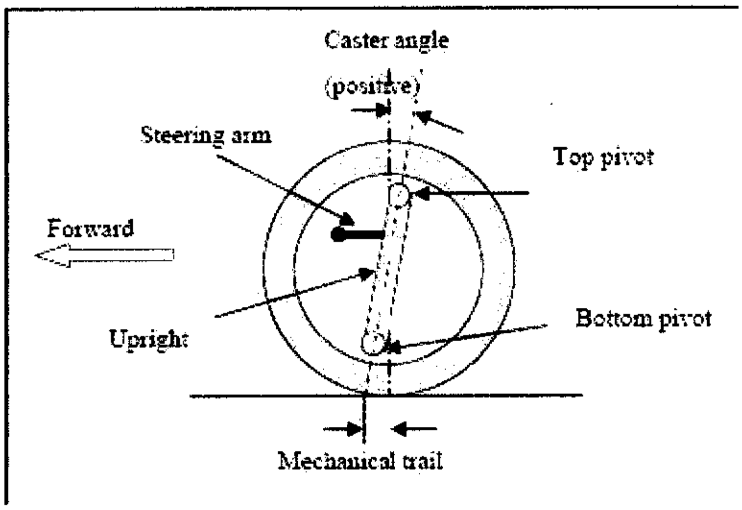


Figure 2.3: Caster Angle

Positive caster induces a self correcting force that provides straight line stability, but increases steering effort. Caster ranges from approximately 2 degrees in racing vehicles and up to 7 degrees in sedans [3].

2.2.4 Kingpin Inclination

The angle in front elevation between the steering axis and the vertical is regarded as kingpin inclination [4]. It is also known as steering axis inclination (SAI) and can be seen in Figure 2.4. It is used to reduce the distance measured at the ground between steering axis and tire's centre of pressure in order to reduce the torque about the steering axis during forward motion. A right kingpin inclination will reduce the steering effort and will provide the driver with a good 'road feel"

2.2.5 Kingpin Offset

Kingpin offset measured at the ground is the horizontal distance in front elevation between the point where the steering axis intersects the ground and the centre of tire contact [4]. Kingpin offset it is also known as scrub radius. It is positive when the centre of tire contact is outboard of the steering axis intersection point on the ground. Kingpin offset is usually measured at static conditions (zero degree camber).

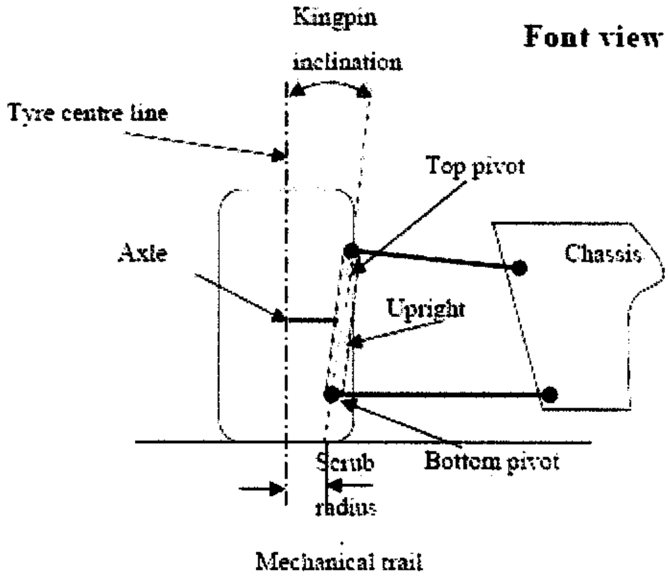


Figure 2.4: Kingpin Inclination

2.2.6 Pitching

Pitching can be defined as rotation of the car around y-axis. The weight of the car causes it during longitudinal acceleration. Pitching is exhibited in two forms:

1. Dive- when load is transferred from the rear axle to the front.
2. Squat- when load is transferred from the front axle to the rear.

Dive can be identified when the front of the car is lower than the rear during braking and vice versa for squat when accelerating (See Figure 2.5).

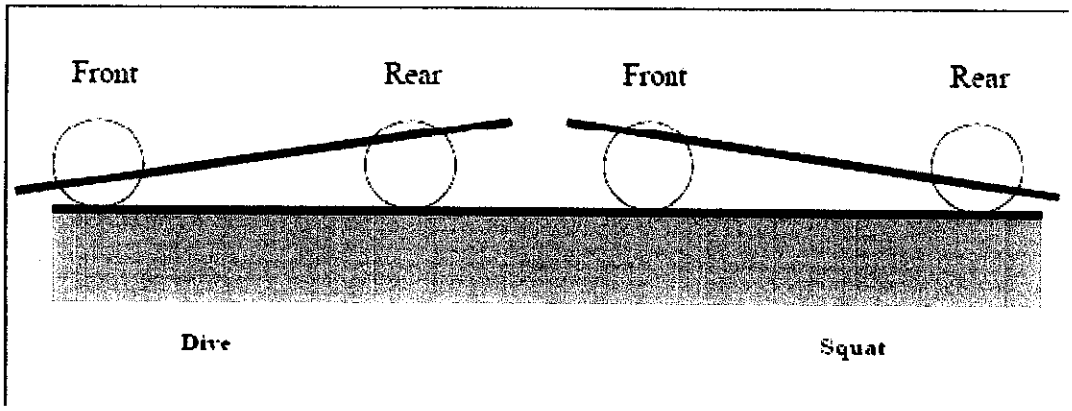


Figure 2.5: Dive and Squat Definition

2.2.7 Static Toe Angle

Static toe angle is measured in degrees and is the angle between a longitudinal axis of the vehicle and the line of intersection of the wheel plane and the road surface. The wheel is "toed-in" if the forward position of the wheel is turned toward a central longitudinal axis of the vehicle, and "towed-out" if turned away [4]. Static toe-in or toe-out of a pair of wheels is measured in millimeters and represents the difference in the transverse distance between the wheel planes taken at the extreme rear and front points of the tire treads. When the distance at the rear is greater, the wheel is "toed-in" by this amount; and where smaller, the wheels are "toed-out" [4] as illustrated in Figure 2.6.

It is necessary to set the static toe such way to prevent the tires to become toe out during maximum bump and roll in order to prevent the outboard tire to steer the vehicle to the outside of the turn when cornering. Toe-in produces a constant lateral force inward toward the vehicle centerline during forward motion that will enhance the straight line stability.

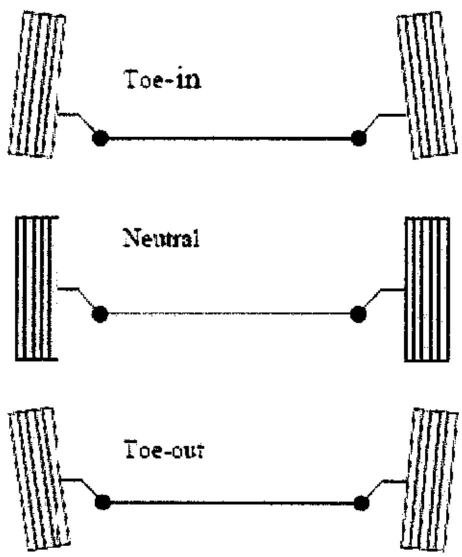


Figure 2.6: Toe configuration

2.3 Suspension Kinematics Parameters

2.3.1 Instantaneous Center

Instantaneous Center (IC) position is defined by drawing lines extending the wishbones in the direction that they converge until they meet again. The point at which they meet is the IC (see Figure 2.7). It is defined as 'instant' as it migrates with suspension deflection. The IC is a 'projected imaginary point that is effectively the pivot point of the linkage at that instant'.

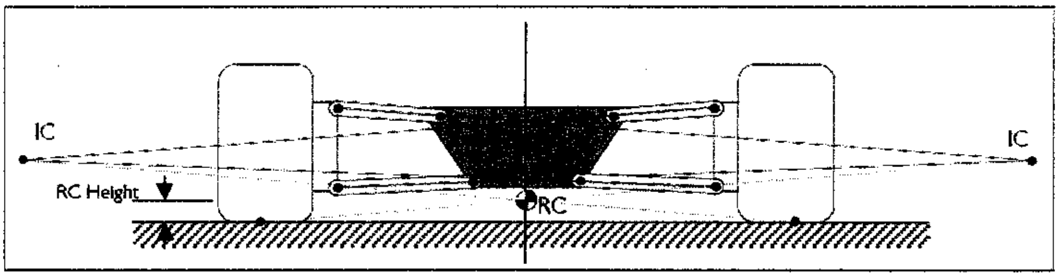


Figure 2.7: Instantaneous Center and Roll Center

2.3.2 Roll Center

The Roll Center (RC) is effectively the instantaneous center of rotation of body roll at that axle. Its position is determined by projecting a line from the corresponding wheel contact patch to that wheels instantaneous center. The point of intersection of the lines is the roll center (see Figure 2.7). As with the instantaneous center, the roll center is a point that is affected by suspension displacement.

2.3.3 Roll Resistance Arm

The roll resistance arm is the lever arm formed between the thread's centre of pressure and the vehicle centre line. This moment arm creates a roll resisting torque when acted on by the reaction forces generated at the tire contact patch by the spring and anti-roll bars [3].

2.3.4 Rollover Arm

Rollover moment arm is the summation of three components that results from three force and roll moment arm pairs. These are lateral acceleration of the sprung mass acting on the arm formed by the CG and roll centre, vertical accelerated sprung mass acting on the arm formed by the lateral displacement of CG and vehicle's centre line and finally the jacking forces acting on the arm formed by lateral displacement of the roll centre from the centre line during roll.

2.3.5 Jacking

The tire reaction forces generated when the vehicle is accelerated during cornering are transmitted to the vehicle through the suspension links. In suspension that place the roll centre above the ground, the upward tire reaction force generated by the outside tire is greater than the downward tire reaction force generated by the inside tire. Summing these forces the resultant will be positive upward acting through its roll centre. This upward jacking force lifts or "Jacks" the sprung mass upward when cornering.

CHAPTER 3

METHODOLOGY AND PROJECT WORK

3.1 Overall Design Flow

The methodology of the overall design starts with theory understanding and parameters acquisition as shown in Chart 1 below.

The flow then goes through kinematics and dynamics analysis of the system itself. The analysis consists of equations derivation and iterations and comparison between the analytical method and modeling simulation.

CAD drawing comes on after all specifications are met. The CAD drawing will be aided by using CATIA V5 R14. The selection of this software is based on its usefulness on modeling and analysis. The packaging of overall system should be easy by using this software.

From the modeling, FEA analysis will be done in order to match good quality of materials for the system. All processes will go through critical review and improvements.

Full vehicle simulation will be conducted after design had been finalized in order to see the characteristic of the car during steady-state cornering, transient cornering, and accelerating. The simulation is performed by using ADAMSCAR.

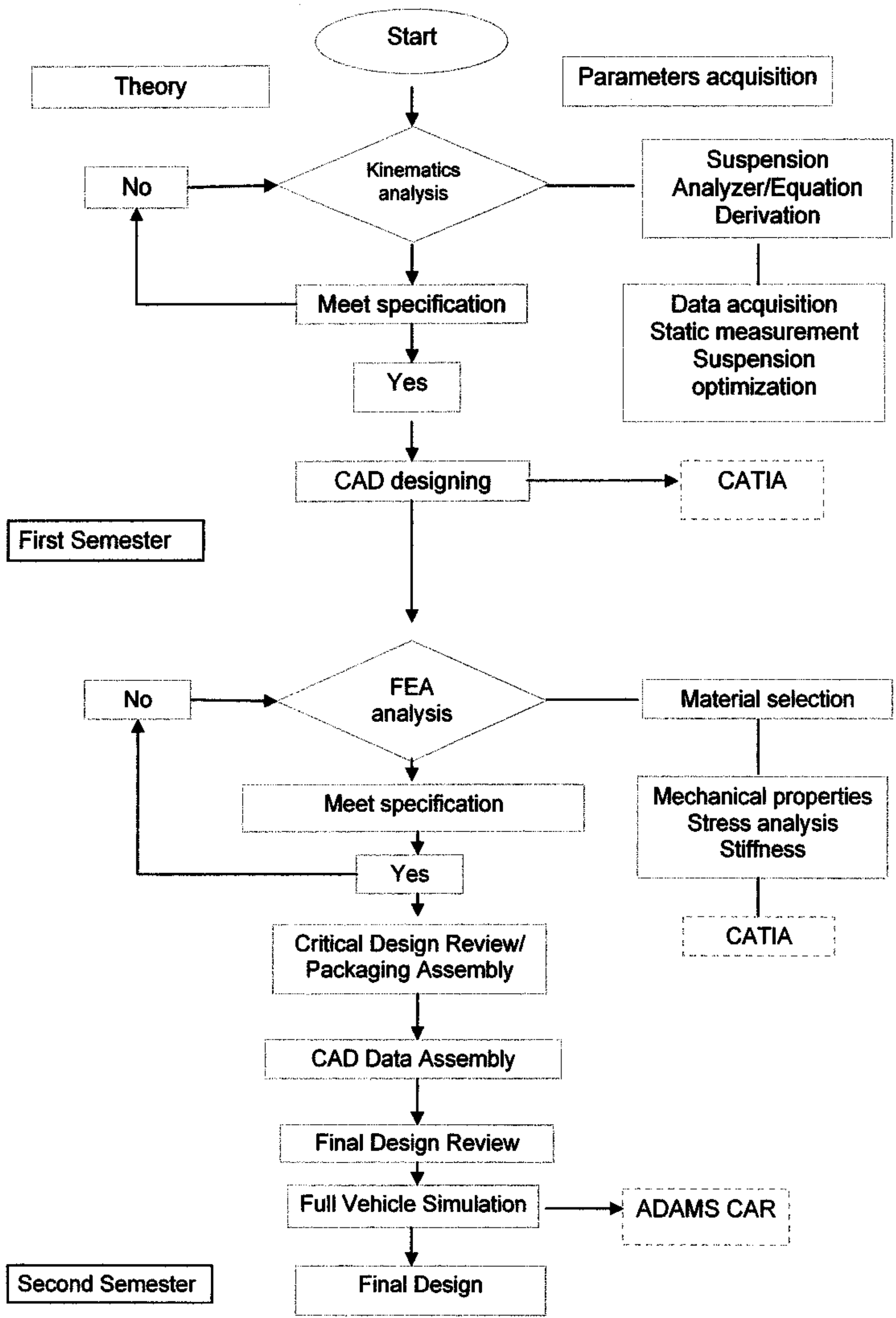


Figure 3.1: Overall Design Flow

3.2 Suspension Design Selection

Suspension types	Degree of Geometry Change		Suspension placement		Component sizing		Degree of Geometry Setup		Manufacturing/ fabricating		Design Simplicity		Total Points
MacPherson Struts	Geometry is not independent. Setup of different geometries affects related geometry		Outboard, needs larger suspension size due to 1:1 installation ratio		Least components but at the expense of size		Inability in design to have independent Geometry setup		Accuracy and tolerance difficult to compensate		Simple design and assembly		10
Equal length Double A-Arm	Design of equal length links results of excessive positive camber gain		Inboard, freedom to size suspension to installation ratio		Medium components, size of components dependant of suspension		Design freedom to vary geometry except to anti-dive and anti-squat mechanism		Wide variety of Man. Process and fabrication		Detail design and for in a confined space		20
Unequal length Double A-Arm	Unequal length links provides desirable negative camber gain		Inboard, freedom to size suspension to installation ratio		Medium components, size of components dependant of suspension		Design freedom to vary all aspects of suspension geometry		Wide variety of Man. Process and fabrication		Detail design and for in a confined space		24
Beam Axle	Geometry is not independent. Setup of different geometries affects related geometry		Inboard, but needs larger suspension size due to larger installation ratio		Least components but at the expense of size		Inability in design to have independent Geometry setup		Wide variety of Man. Process and fabrication		Simple design and assembly		15

Table 3.1: Suspension Configuration Selection

Designing the suspension system involves various criteria which govern all the build up configuration of the system. The degree of geometry change is depends on the degree of freedom the design selection offers. The degree of geometry setup is indeed in terms of camber, toe, caster, anti dive and squat. Suspension placement involves the suspension sizing and Center of Gravity (CG) placement. Inboard placement offers better compact design and better CG placement. Manufacturing process involves the ease of fabricating and keeping tolerances of the design. Unequal length Double A-Arm suspension is chosen based on the total points awarded to every criterion that suits the characteristics of the vehicle. Radar chart below explains graphically.

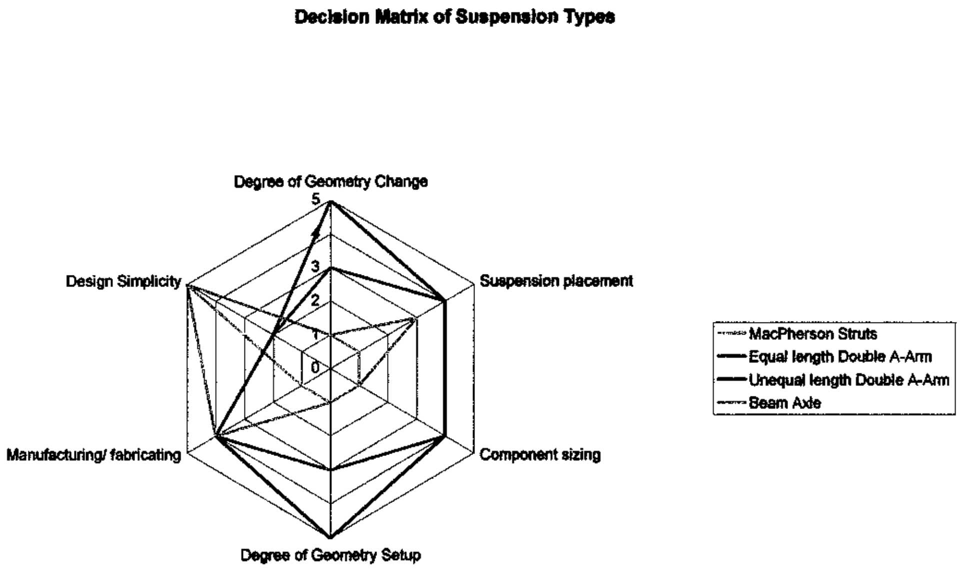


Figure 3.2: Decision Matrix on Suspension Selection.

3.3 Kinematics Analysis

The first step on kinematics analysis is to derive equations based on Maurice Olley's Derivation (Suspension and Handling, 1937). The equations then are to be transferred to Microsoft Excel to iterate the desirable values. The results are then being compared with findings from Suspension Analyzer V2.0.

The derivations begin by assuming that an Unequal Length or short long arm (SLA) suspension mechanism may be approximated in front view by a planar four-bar linkage [8] as shown in Figure 3.3 below.

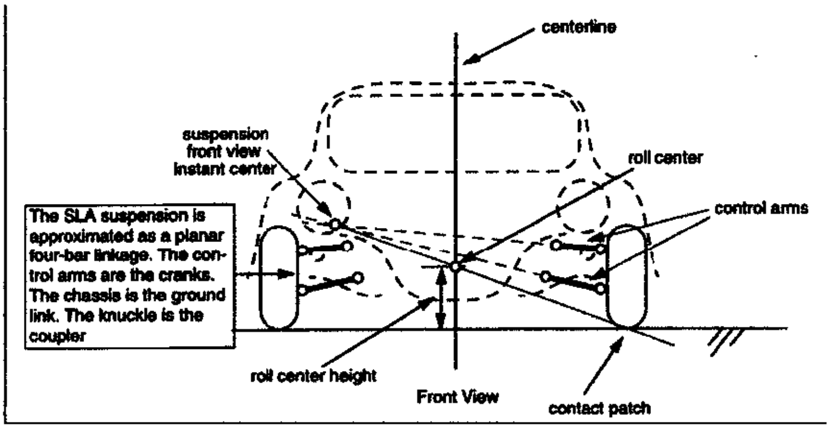


Figure 3.3: Front View of Suspension

It then proceeds with a parabolic approximation to the circular arc $x = y^2/2R$ to mathematically relate small motions of the outer ball-joints to jounce-rebound as shown in Figure 3.4(a). When the arms are not initially vertical but has an initial lift "a" in Figure 3.4(b), the expression for "x" becomes:

$$\frac{y^2}{2R} + \frac{ya}{R}$$

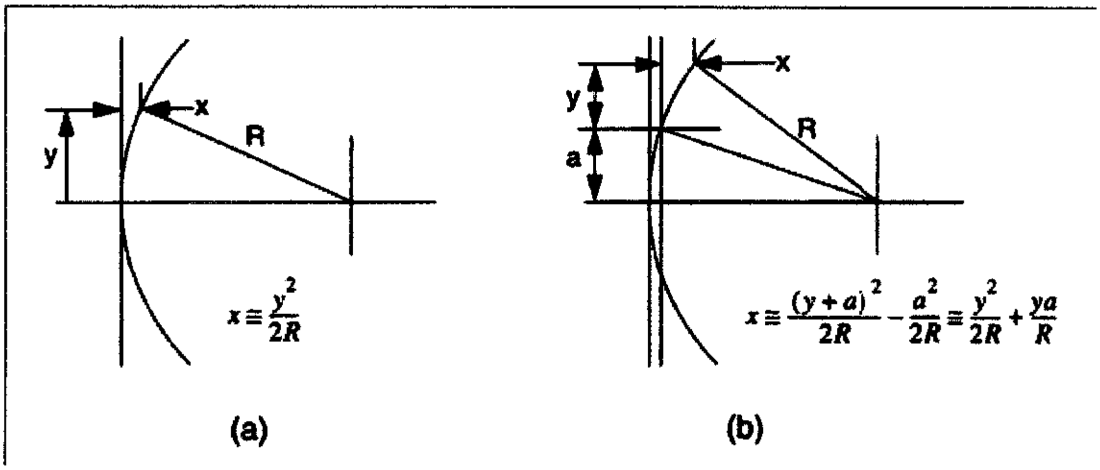
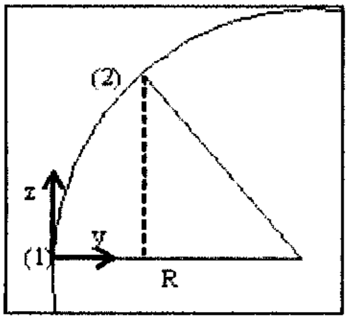


Figure 3.4: Approximating parabolic arc.

Similarly, by simple circle geometry, considering movement of point (1) to point (2) in Figure 11 below, it dictates that:



$$z^2 + (y-R)^2 = R^2$$
$$\Rightarrow z^2 + y^2 - 2yR + R^2 = R^2$$
$$\Rightarrow 2yR - y^2 = z^2$$

Considering small movement of y, so that y^2 becomes negligible

$$2yR \approx z^2$$
$$y \approx z^2/2R$$

Again, similarly with initial lift "a", the expression for y is

$$y \approx (z^2/2R) - (za/R)$$

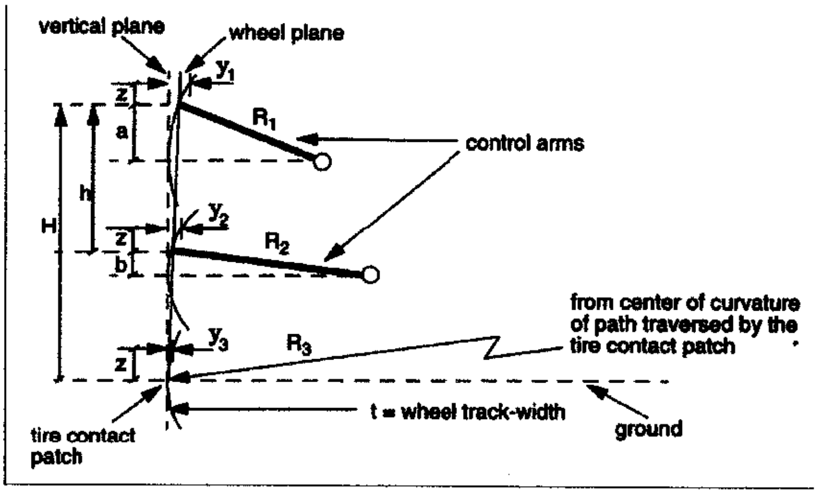


Figure 3.5: Curvature

Referring to Figure 3.5, it proceeds with: $y_1 \approx (z^2/2R) - (za/R)$ (Eq 1)

$$y_2 \approx (z^2/2R) - (zb/R) \quad \text{(Eq 2)}$$

The equation of the line through the outer joints of the two control arms gives the following expression for the displacement of the tire patch as a function of the control arm displacements. Using similar triangles, it can be deduced a relation between y1, y2, and y3.

$$\Rightarrow y_3 = \frac{H}{h} y_2 - \frac{(H-h)}{h} y_1$$

(Eq 3)

Furthermore, camber equation yielded, by assuming a very small angle:

$$\gamma \approx (y_1 - y_2)/h \qquad \text{(Eq 4)}$$

Substituting from Eqs. (1) and (2) into Eq. (3) and (4):

$$\Rightarrow y_3 = \frac{z^2}{2h}\left[\frac{H}{R_2} - \frac{(H-h)}{R_1}\right] + \frac{z}{h}\left[\frac{(H-h)a}{R_1} - \frac{Hb}{R_2}\right]$$

$$\Rightarrow \gamma = \frac{z^2}{2h}\left(\frac{1}{R_1} - \frac{1}{R_2}\right) + \frac{z}{h}\left(\frac{b}{R_2} - \frac{a}{R_1}\right) \qquad \text{(Eq 5)}$$

Differentiating Eq. (5) with respect to z yields rate of tread change or scrub:

$$\Rightarrow \frac{dy_3}{dz} = \frac{z}{h}\left[\frac{H}{R_2} - \frac{(H-h)}{R_1}\right] + \frac{1}{h}\left[\frac{(H-h)a}{R_1} - \frac{Hb}{R_2}\right] \qquad \text{(Eq 6)}$$

The height of the roll center is related to the rate of tread change by the following expression:

Height of roll center = rate of tread change X (track width / 2)

$$\Rightarrow h_{RC} = \frac{t}{2h}\left[\frac{Hb}{R_2} - \frac{(H-h)a}{R_1}\right] \qquad \text{(Eq 7)}$$

From Eq.(5), differentiation with respect to z yields:

$$\Rightarrow \frac{d\gamma}{dz} = \frac{z}{h}\left(\frac{1}{R_1} - \frac{1}{R_2}\right) + \frac{1}{h}\left(\frac{b}{R_2} - \frac{a}{R_1}\right) \qquad \text{(Eq 8)}$$

Introducing suspension constants derived by Maurice Olley ("Suspension and Handling", 1937):

$$P_1 = \frac{1}{R_1} - \frac{1}{R_2}$$

$$Q_1 = \frac{a}{R_1} - \frac{b}{R_2}$$

$$P_2 = \frac{H}{R_2} - \frac{(H-h)}{R_1}$$

$$Q_2 = \frac{Hb}{R_2} - \frac{(H-h)a}{R_1}$$

Rewriting the results in terms of these constants,

$$y = \frac{P_1 z^2 / 2 - Q_1 z}{h}$$

$$\frac{dy}{dz} = \frac{P_1 z - Q_1}{h}$$

$$y_3 = \frac{1}{h}\left[P_2 z^2 / 2 - Q_2 z\right]$$

$$\frac{dy_3}{dz} = \frac{1}{h}\left[P_2 z - Q_2\right] \tag{Eq 9}$$

$$\Rightarrow h_{RC} = \frac{-t}{2h}\left[P_2 z - Q_2\right] \tag{Eq 10}$$

It appears that, it is a little bit more complicated when the uprights are assembled in the diagram so that, some offsets are seen from the centerline of the tires. Refer Figure 3.6 below:

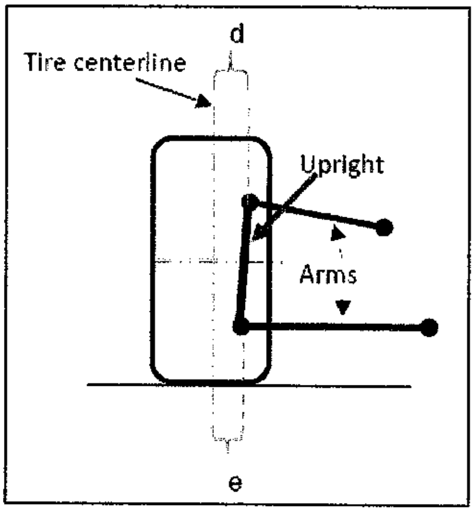


Figure 3.6: Upright location with offsets d and e.

The re-derivation of the equations is similar but a bit tedious and involves other Olley's constants:

$$U_1 = \frac{d}{R_1} - \frac{e}{R_2}$$
$$V_1 = \frac{ad}{R_1} - \frac{be}{R_2}$$
$$U_2 = \frac{He}{R_2} - \frac{(H-h)d}{R_1}$$
$$V_2 = \frac{Hbe}{R_2} - \frac{(H-h)ad}{R_1}$$

Using these constant as well as the previous constants, the corrected equations have been arrived below:

$$\left.\begin{aligned}
\gamma &= \frac{P_1 z^2/2 - Q_1 z}{h + V_1 + U_1 z} \\
\frac{d\gamma}{dz} &= \frac{P_1 z - Q_1 - \gamma U_1}{h + V_1 + U_1 z} \\
y_3 &= \frac{1}{h}\left[P_2 z^2/2 - Q_2 z - \gamma(U_2 z + V_2)\right] \\
\frac{dy_3}{dz} &= \frac{1}{h}\left[P_2 z - Q_2 - U_2\gamma - \frac{d\gamma}{dz}(U_2 z + V_2)\right] \\
h_{RC} &= -\frac{t}{2h}\left[P_2 z - Q_2 - U_2\gamma - \frac{d\gamma}{dz}(U_2 z + V_2)\right]
\end{aligned}\right| \qquad \text{(Eq 11)}$$

3.4 Lateral Load Transfer

There are four (4) steps involved in computing the lateral load transfer (LLT) which includes load transfer due to the roll moment, sprung mass inertia, unsprung mass inertia, and the determination of total load transfer.

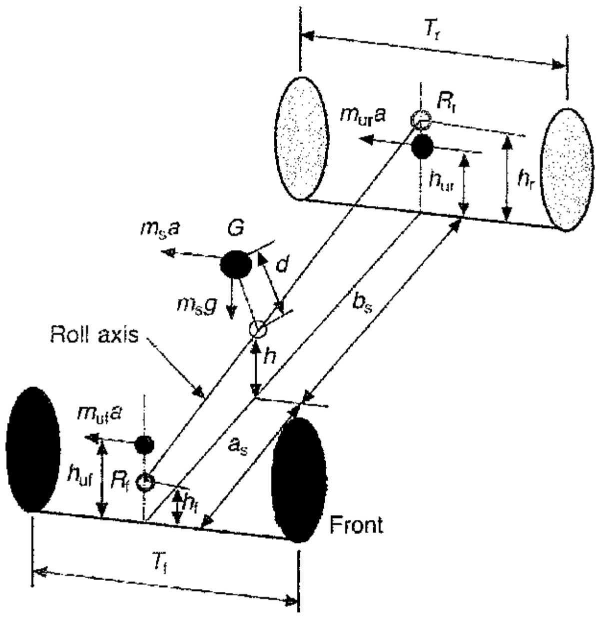


Figure 3.7: LLT nomenclature.

For Figure 3.7 shown above, replace the 2 forces at *G* with the same forces at *A* plus a moment (roll moment) about the roll axis.

$$M_s = m_s ad \cos\ \phi + m_s\ gd \sin\ \phi \approx m_s\ ad + m_s g\phi$$

Ø is treated as a small angle. *Ms* is reacted by a roll moment *Mø* (at the suspension spring and anti roll bars) and distributed to the front and rear suspensions.

$$M_\phi = k_s\phi$$

Where k_s = total roll stiffness (function of chassis torsional rigidity, suspension and tires). From the above equations,

$$\phi = \frac{m_s ad}{k_s - m_s gd}$$

$M\phi$ can be splitted into components $M\phi_f$ and $M\phi_r$ at the front and rear axles, such that:

$$M_{\phi} = M_{\phi f} + M_{\phi r} = k_{sf}\phi + k_{sr}\phi$$

Where k_{sf} and k_{sr} are the roll stiffness components at the front and rear axles.

$$(k_{sf} + k_{sr} = k_s)$$

The front load transfer due to the roll moment is then:

$$F_{fsM} = \frac{k_{sf}\phi}{T_f} = \frac{k_{sf} m_s ad}{T_f (k_{sf} + k_{sr} - m_s gd)}$$

Similarly, the rear load transfer due to roll moment is:

$$F_{rsM} = \frac{k_{sr}\phi}{T_r} = \frac{k_{sr} m_s ad}{T_r (k_{sf} + k_{sr} - m_s gd)}$$

Where T_f and T_r are the front and the rear track widths.

The sprung mass is distributed to he roll centers at the front and rear axles. The respective masses at front and rear are:

$$m_{sf} = \frac{m_s b_s}{L} \text{ and } m_{sr} = \frac{m_s a_s}{L}$$

The centrifugal force at A is distributed to the respective roll centers as follows:

$$F_{fs} = m_{sf} a \text{ and } F_{rs} = m_{sr} a$$

The corresponding load transfers are:

$$F_{fsF} = \frac{m_{sf} a h_f}{T_f} \text{ and } F_{rsF} = \frac{m_{sr} a h_r}{T_r}$$

The respective load transfers at the front and rear axles due to unsprung mass inertia forces are:

$$F_{fuF} = \frac{m_{uf} a h_{uf}}{T_f} \text{ and } F_{ruF} = \frac{m_{ur} a h_{ur}}{T_r}$$

The load transfers for the front and rear wheels are:

$$F_f = F_{fsM} + F_{fsF} + F_{fuF}$$

$$F_r = F_{rsM} + F_{rsF} + F_{ruF}$$

3.5 Steering Design Selection

3.5.1 Worm and Sector Steering

The manual worm and sector assembly uses a steering shaft with a three-turn worm gear supported and straddled by ball bearing assemblies. In operation, a turn of the steering wheel causes the worm gear to rotate the sector and the pitman arm shaft and the movement is transmitted through the steering train to the wheel spindles.

3.5.2 Manual Rack and Pinion Steering

A typical rack and pinion steering gear assembly consists of a pinion shaft and bearing assembly, rack gear, gear housing, two tie rod assemblies, an adjuster assembly, dust boots and boot clamps, and grommet mountings and bolts. When the steering wheel is turned, this manual movement is relayed to the steering shaft and shaft joint, and then to the pinion shaft. Since the pinion teeth mesh with the teeth on the rack gear, the rotary motion is changed to transverse movement of the rack gear. The tie rods and tie rod ends then transmit this movement to the steering knuckles and wheels.

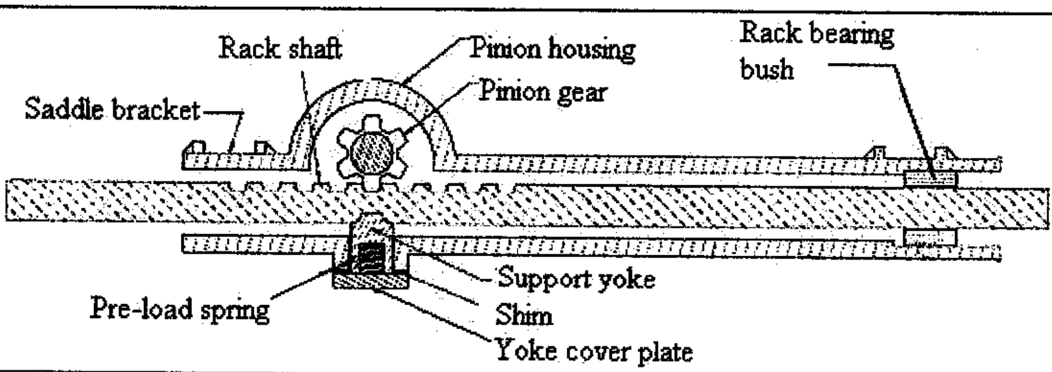


Figure 3.8: Rack and Pinion steering [11].

From all manual steering systems the most suitable system is rack and pinion steering due to the simple construction, has a high mechanical efficiency, and reduced space requirement. Table 3.2 below shows the steering selection based on various criterions.

[illegible]	component sizing	points	manufacturing /fabricating	points	design simplicity	points	Installation space	points	reliability	points	total points
[illegible]	Medium components, size of components dependant of gears	3	Complicated and detailed manufacturing process of worm gears	2	Detail design for various diameter of worm gear	2	Consumes portion of spaces for steering system	3	High reliability and mechanical efficiency	4	14
[illegible]	Small components, size of components dependant of pinion	4	Wide variety of manufacturing process and fabrication	3	Simple design for pinion and rack	4	Reduced space requirements, least interfere with other components	4	High reliability and mechanical efficiency	4	19

Table 3.2: Steering configuration selection

Manual Rack and Pinion Steering is chosen based on total points that governed by various criterions stated. Radar graph below explains visually.

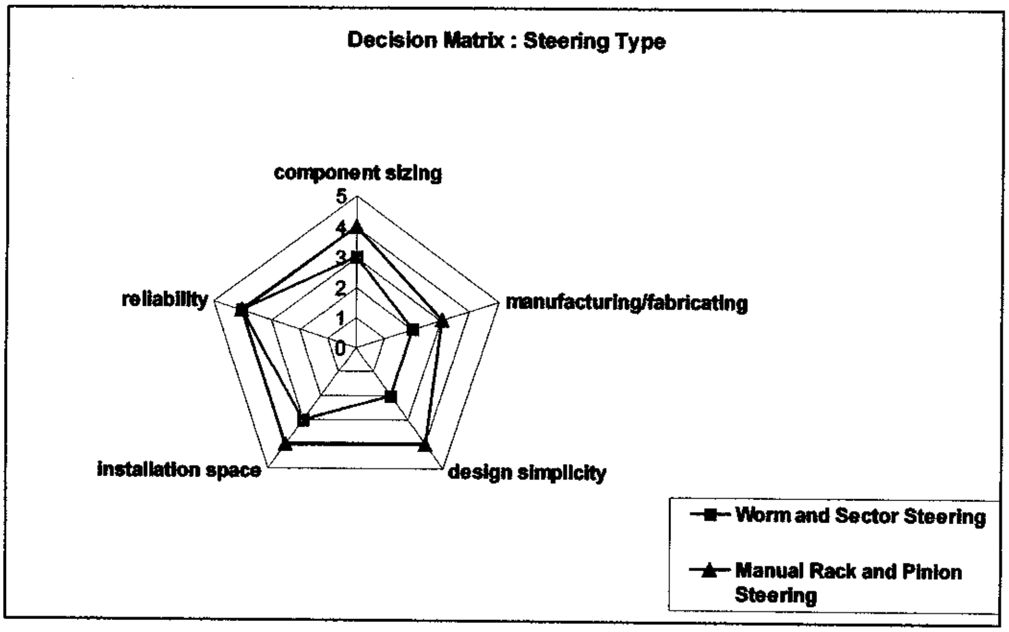


Figure 3.9: Decision Matrix on steering selection

3.6 ADAMSCAR Simulation

ADAMSCAR which is a specific environment for automotive application was used. It has built in simulations for common vehicle dynamics tests such as constant radius turning, lane changes, and steering input among others. One great advantage of using ADAMSCAR is the flexibility that parametric models can give to the analysis. Changes in design can easily be evaluated by modifying the parameters like geometry or mass without building a new model.

In using ADAMSCAR as a simulation platform for full vehicle analysis, the first step is to setup the hard points according to the CAD design. This is to ensure that the simulation will give the results according to the vehicle specifications. The hard points can be modified from the templates of the component inside ADAMSCAR.

Analysis and simulation can only be performed with assemblies in ADAMSCAR. There are two types of assemblies which are suspension assembly and full vehicle assembly. The first type of assembly is used to perform suspension analysis and it must contain at least a suspension system. A full vehicle assembly is needed to perform vehicle dynamics analysis.

In conducting the full vehicle analysis, there are two different maneuvers were mainly used to evaluate the steady state and transient behavior of the race car; a quasi-static constant radius cornering test and an open loop step steering input test. A constant radius cornering test is used to evaluate the steady state characteristics of a vehicle. Transient maneuvers such as step steering input can help analyze the response and corner entry behavior.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Design Optimization

4.1.1 Previous Upright Design

The previous 2007 upright design uses Aluminum 7075. The featured design gives detailed profiles which require advanced machining tools. Further discussion about flaws and disadvantages are as below:

- Pockets require advance machining process such as usage of five-axis CNC machine.
- Material wastage: various sections need to be shaved in order to get the final product. (See Figure 4.1 and 4.2 section A)
- Bulky design: several profiles can be eliminated to reduce weights
- Rigidity of steering arm mounting points is doubtful since stress concentration can occur on the edges and sharp corners.

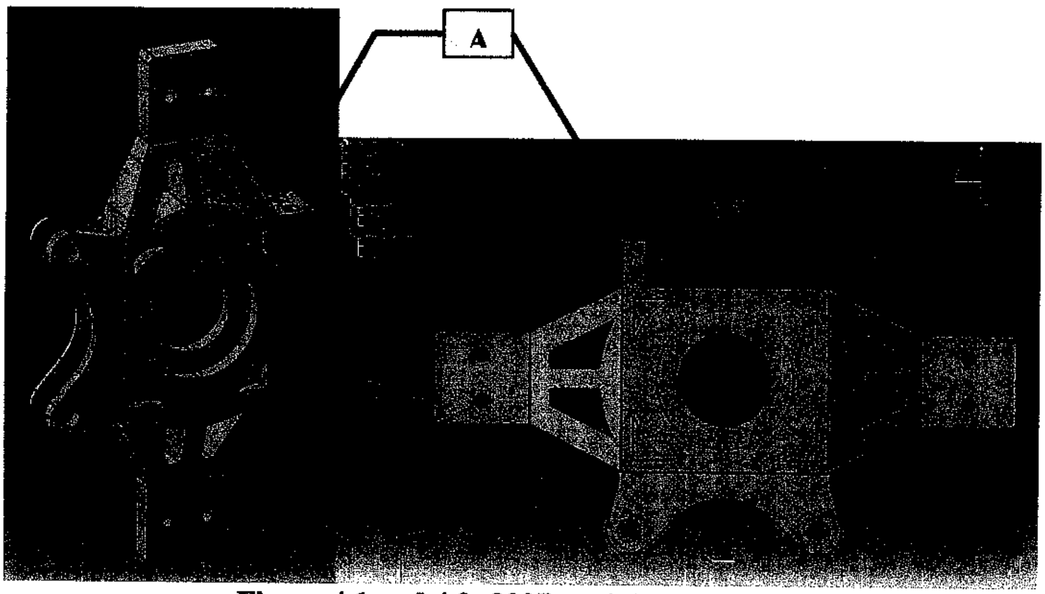


Figure 4.1 and 4.2: 2007 upright design [10].

4.1.2 New Upright Design

The basis of new upright design concentrates on reducing material wastage, strong and rigid structure, ease of manufacturing, and reducing stress concentration points on edges and corners. Figure 4.3 below indicates the components of new upright.

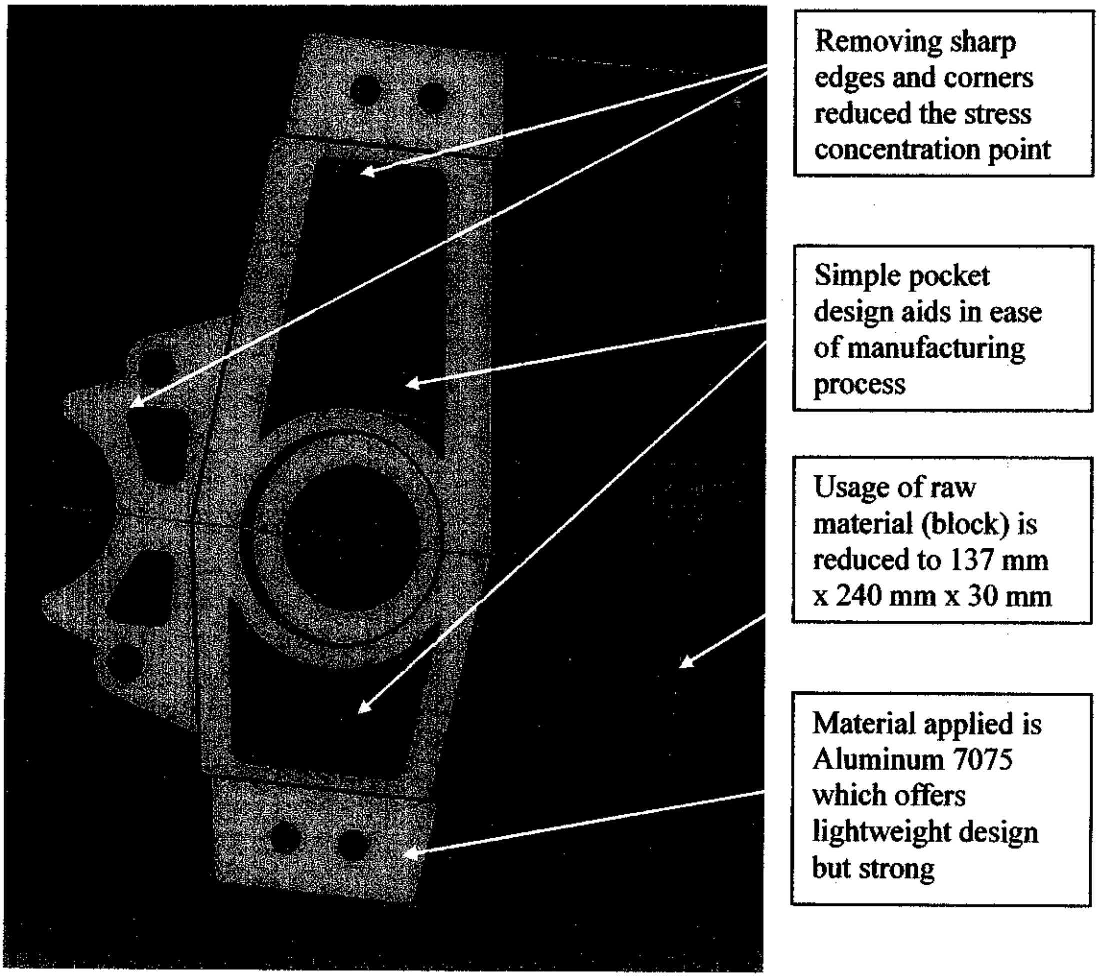


Figure 4.3: New upright design

Criteria	2007 Design	Points	**2008 Design**	**Points**
Material	Aluminum 7075 (Billet)		Aluminum 7075 (Billets)	
Weight	0.864 kg (per piece of the front upright)	4	0.838 kg (per piece of the front upright)	3
Manufacturing process	Outsource CNC: 100%	3	In-house CNC: 100%	5
Material wastage	40% to 50%	2	25% to 35%	4
Stress concentration points	High	3	Low to medium	4
(1 to 5)	TOTAL	12	**TOTAL**	**16**

Table 4.1: Comparison between 2007 and 2008 Upright Design

4.1.3 Clevis New Design

First Design

- Bulky design
- Higher stress concentration on sharp edges and corners
- Assembly: bolts and nuts
- Material: Aluminum 7075

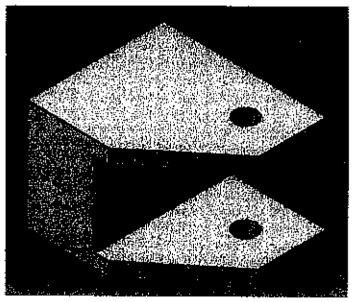


Figure 4.4: Initial clevice design

Second Design

- Fillet sharp edges with 5 mm radius
- Lower stress concentration on edges and corners
- Low rigidity connection between chassis frame and clevice.

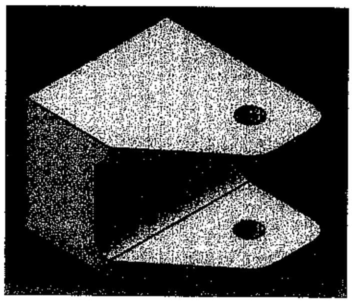


Figure 4.5 : Second clevice design

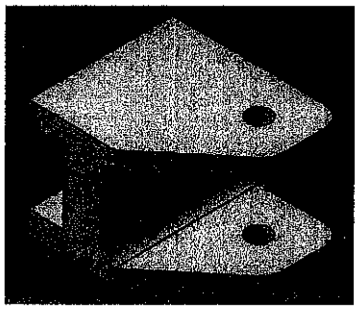
Finalized Design - Improving the contact patch with chassis profile - Weight reduction - Assembly : bolts and nuts or TIG welding	 *Figure 4.6 : Finalized clevice design*

Table 4.2: Design Steps of Clevice

4.2 Loading Condition

4.2.1 During Wheel Collision with Pothole

When the vehicle passes through a pothole, the vertical load is a result of the corner mass multiplied by the gravitational acceleration. Assuming there is no absorption from the tires, thus transferring 100% of the vertical load to the suspension arms and linkages.

$$F = mg$$

Assuming the car to be 250 kg and has 40/60 weight distribution (front and rear)
Material Properties of Aluminum 7075 in APPENDIX A-2

$$m = 40/100\ (250\ \text{kg})$$
$$m = 100\ \text{kg}/2$$
$$m = 50\ \text{kg}$$

$$F = mg$$
$$F = (50\ \text{kg})\ (9.81\ \text{m/s}^2)$$
$$F = \mathbf{490.5\ N}$$

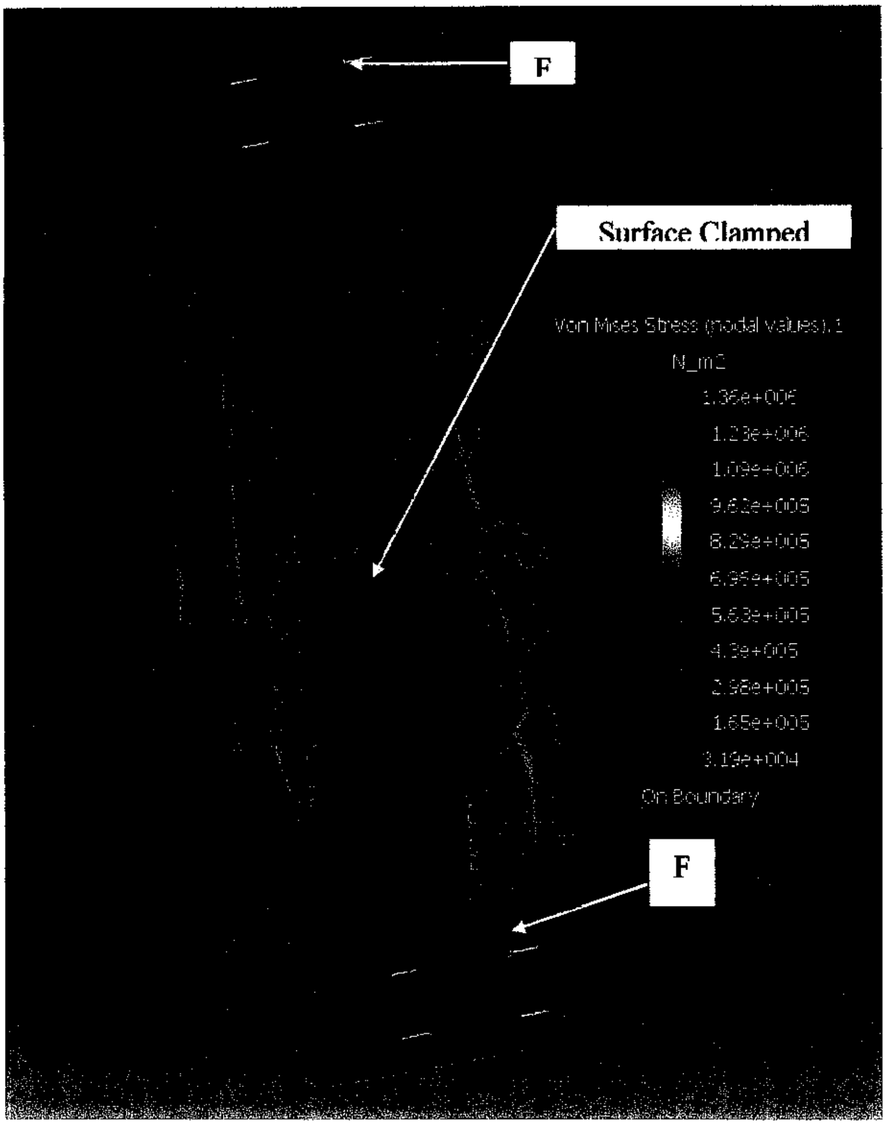


Figure 4.7: Results for collision with pothole

The wheel upright is constructed using Aluminum 7075-T6. Wheel loading is acted on the shaft surface which is connected to the wheel. During pothole collision, the gravitational impact is absorbed by the wheel and transferred to the spindle, fixed to the upright. Stress is mainly concentrated at the top and lower outboard mounts. These mounts are supported through the through holes (mounting point for outboard mounts). Material properties of Aluminium 7075-T6 that is in concern of the analysis is the Yield Strength. The Yield Strength is 95 Mega Pascal. From the result of the analysis, the maximum stress is only 1.36 Megapascal.

4.2.2 During Side Collision at 100 km/h Speed

The impact simulation is based on several assumptions made. These assumptions are:

- Direct impact, thus the force and momentum rate is transferred directly to the suspension components.
- Collision time is set to be 1 second
- Vehicle mass of 250 kg and 40/60 weight distribution (front and rear)

$$\text{Velocity, } V = 100 \text{ km/h}$$
$$= 27.788 \text{ m/s}$$

$$\text{Force (momentum, F)} = m(v - u)/t$$
$$= (250 \text{ kg})(0 - 27.788 \text{ m/s})(1 \text{ s})$$
$$= \mathbf{6947\ N}$$

During side impact collision of speed 100km/h, the exerted force on the bearing perpendicular surface is 6974N. From this standpoint, the loading is more critical and relatively more important. The maximum stress achieved from this analysis is 17.1 Megapascal. However, the factor of safety is 5.5.

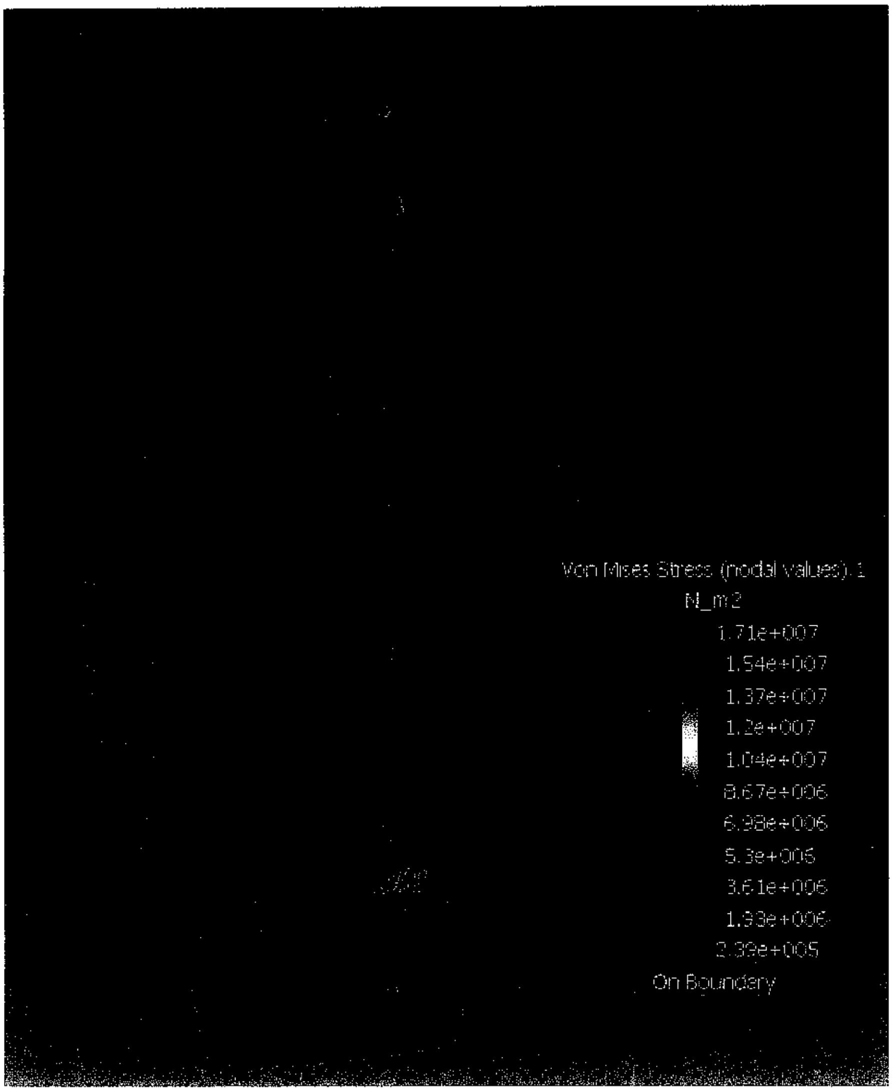


Figure 4.8: Results from side collision at 100 km/h speed

4.2.3 Steering Movement Ratio

The rack and pinion mechanism is designed to transfer the circular input motion of the pinion into linear output movement of the rack. It was measured that for a full travel of the rack of 120 mm the pinion has to be rotated 2.25 turns. Therefore for one turn, the rack travel will be:

$$X_0 = 120 \text{ mm} / 2.25 \text{ turns}$$
$$= 53.33 \text{ mm}$$

Considering the pinion to make one revolution then the input steering movement is:

$$x_i = 2 \text{ x phi x R}$$

Where, R = 155 mm is the radius of the steering wheel.

And the output rack movement is:

$$X_0 = 2 \text{ x phi x r} = 53.33 \text{ mm}$$

$$r = 53.33 \text{ mm} / 2 \text{ x phi}$$

$$= 8.48 \approx 8.5$$

Then, the movement ratio can be calculated as input movement over output:

$$MR = x_i / x_0$$

$$= 155 / 8.5$$

$$= 18.23$$

Therefore the movement ratio is 18.23:1

We needed to know the movement ratio in order to determine the output load transmitted to the tie rods for a given input load. For an effort of 20 N applied by each hand on the steering wheel and considering no friction, the output load will be:

$$F_0 = F_i \text{ x MR}$$

$$= 2 \text{ x } 20 \text{ x } 18.23$$

$$= 729.2 \text{ N}$$

Therefore the load transmitted to the tie rods is 729.2 N.

4.2.4 Pinion Analysis

Based on the load transfer from steering wheel, the moments generated at the pinion are,

$$\text{Torque} = F_0 \times R$$

Where, R = 155 mm is the radius of the steering wheel.

Thus, the torque generated,

$$\begin{aligned} T &= F_0 \times R \\ &= 729.2 \text{ N} \times 0.155 \text{ m} \\ &= 113.026 \text{ Nm} \end{aligned}$$

Figure 4.9: Von Mises Stress of the pinion

4.3 Static Analysis

4.3.1 Kinematics Analysis Result on Equation Derivation

Iterations have been made by using Microsoft Excel from Equations 11. The iterations resulted on finding the camber angle and roll center from inputs given.
The inputs are:

Inputs							
[redacted]							
54.32	370.00	57.00	398.18	209.36	0.00	394.00	57.00

Table 4.3: Front Arm Configuration.

Another input is the track width, t which was set to be 1276 mm. All inputs above have been set to match the packaging with the chassis section. All inputs are based on the equations.

Olley Constants							
[redacted]		P2	Q2	U1	V1	U2	V2
0.000165	0.146811	0.500285	-27.7208	0.009384	8.368216	28.51623	-1580.09

Table 4.4: Maurice Olley's Constant.

The constants above are based on the inputs given to the equations.

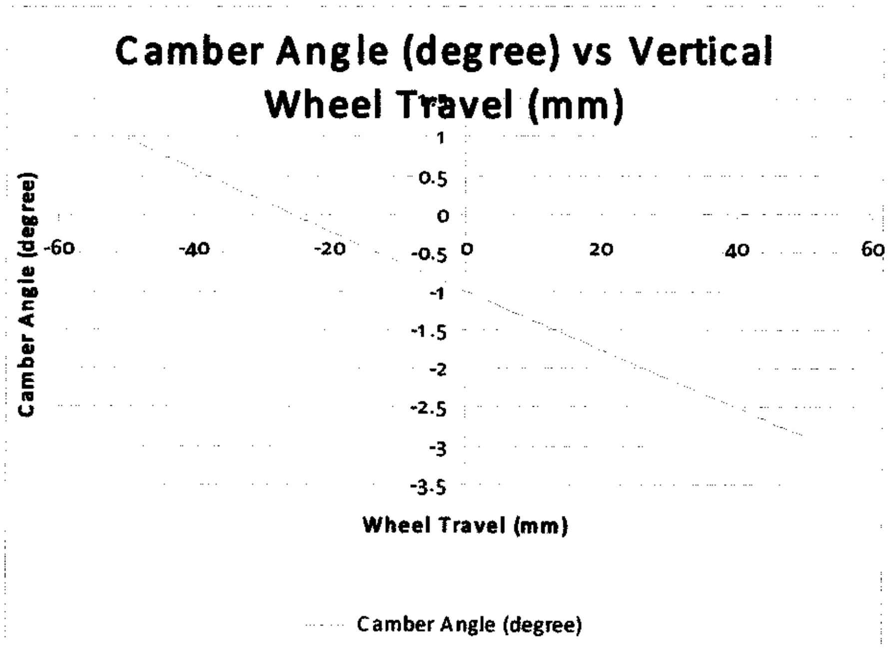


Figure 4.10: Camber Angle versus Vertical Wheel Travel

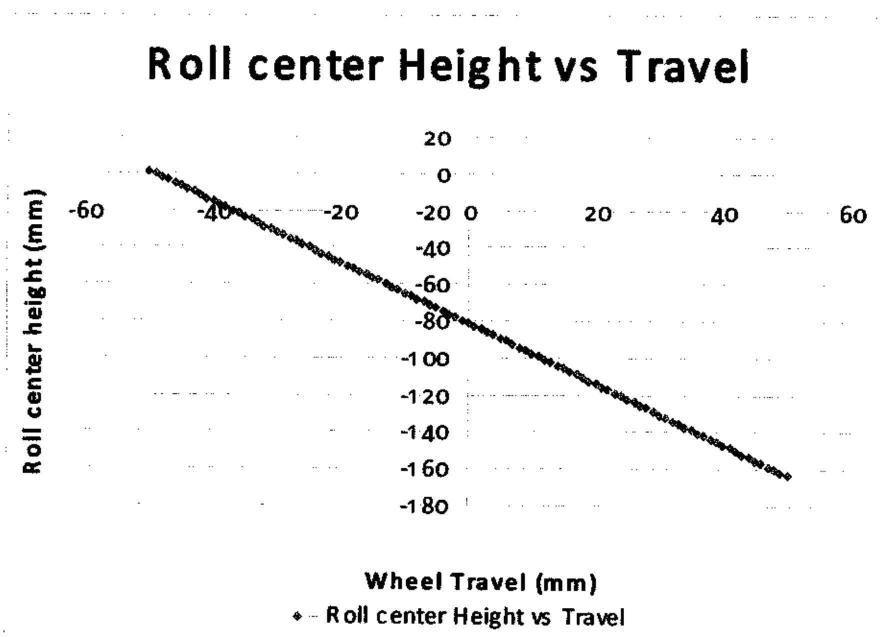


Figure 4.11: Roll Center Height versus Vertical Wheel Travel

4.3.2 Bound and Rebound Kinematics paths for suspension performance using Suspension Analyzer V2.0

The suspension geometry design is evaluated using the Suspension Analyzer V2.0. This software is programmed to analyze static and kinematics geometry behavior according to dive, roll and steer rates. The following is the suspension links pick-up points. The only limitation is this software cannot simulate the kinematics behavior according to the autocross track layout.

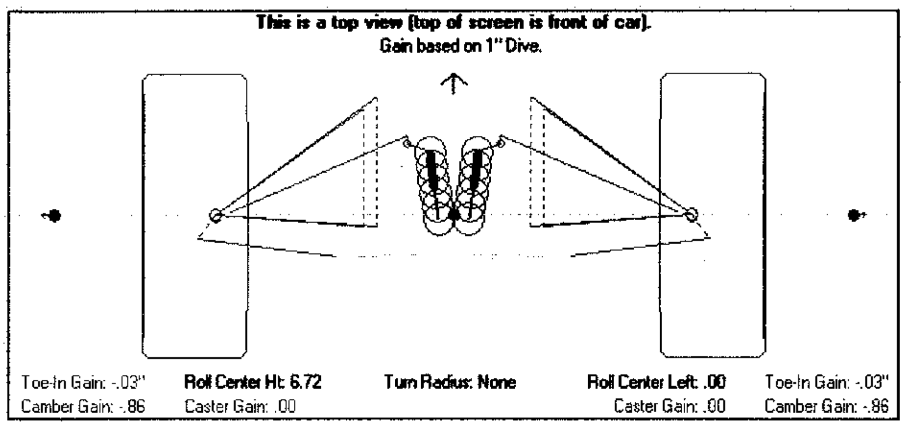


Figure 4.12: Top View of Suspension Layout

From the Figure 4.13 and 4.14 below, the results from Suspension Analyzer illustrate there were such difference in the camber angle and roll center height determined based on derivation method. The roll center height calculated was -80mm while the Suspension Analyzer recorded that the roll center height was -68mm. These slight changes occur due to simplification made to the equation used.

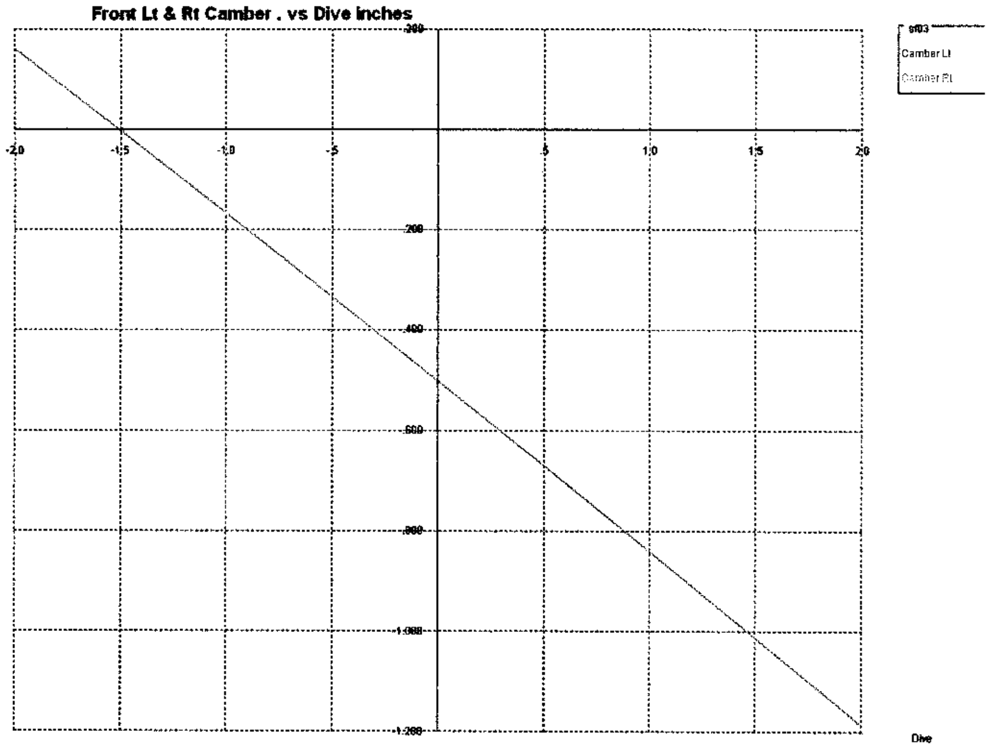


Figure 4.13: Graph Generated of Camber angle

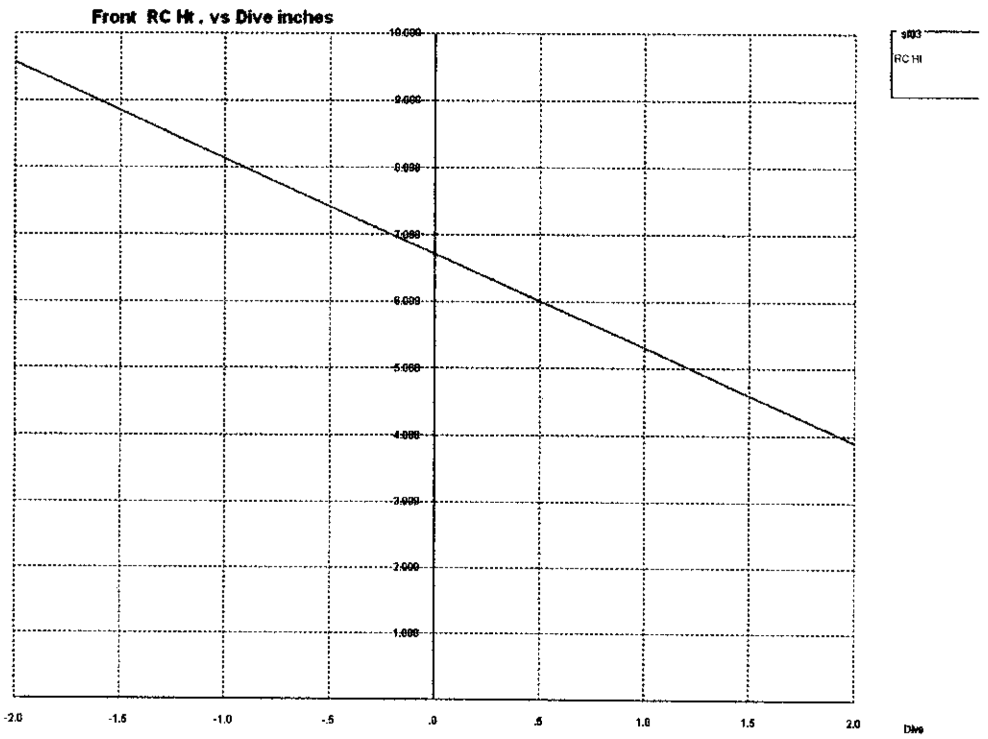


Figure 4.14: Graph Generated on Roll Center Height

4.3.3 Lateral Load Transfer Determination

Iteration has been made on Microsoft Excel based on the Lateral Load Transfer equation. The iteration results on finding Lateral load transfer of the vehicle. From here we can get the force values that acted on outer and inner of wheels. Few assumptions had been made:

- Steady-state cornering.
- Neglect change in body height and angles due to steering and slip angles of the tire.

The inputs required for determining the Lateral Load Transfer,

Center of Gravity (CofG) height	0.4	m
Roll couple (d)	0.1	m
Total Roll Stiffness	1000	N/deg
Roll Stiffness Front (ksf)	500	N/deg
Roll Stiffness Rear (ksr)	500	N/deg
Track Width Front (Tf)	1.28	m
Track Width Rear (Tr)	1.15	m
Wheelbase (WB)	1.65	m
Roll Center Front (hf) height	0.06	m
Roll Center Rear (hr) height	-0.04	m
Total Mass (Mtotal)	280	kg
70% sprung		
Sprung Mass Front (msf)	98	kg
Sprung Mass Rear (msr)	98	kg
30% unsprung		
Unsprung Mass Front [illegible]	42	kg
Unsprung Mass Rear [illegible]	42	kg
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]

Table 4.5: Input Parameters

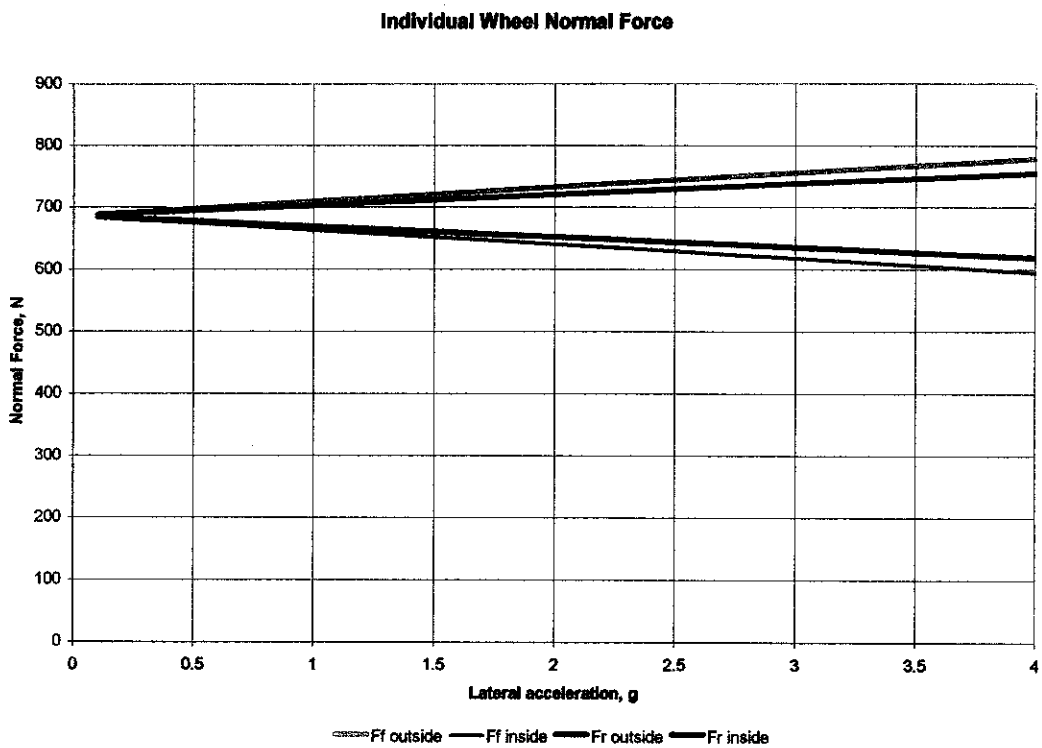


Figure 4.15: Individual Wheel Normal Force versus lateral acceleration.

The lateral acceleration is set from $\alpha = 0$ to 4g (where $g = 9.81\ m/s^2$). The cornering radius is set to be 7.5 meters. Load is transferred from the inside track to the outside track when cornering because of the height of CG. The vehicle loads on the outer wheels increases while on the inner wheel the loads decreases.

4.4 Dynamics Analysis

4.4.1 Assembly Setup

In order to run a simulation in ADAMSCAR, a full vehicle assembly must be created. This assembly consists of chassis, front and rear suspensions, steering systems, tires, and anti-roll bars. Also, the hard points of the front and rear suspension had been modified to the specification of the vehicle. Figure 4.16 below indicates the full assembly of the vehicle in ADAMSCAR.

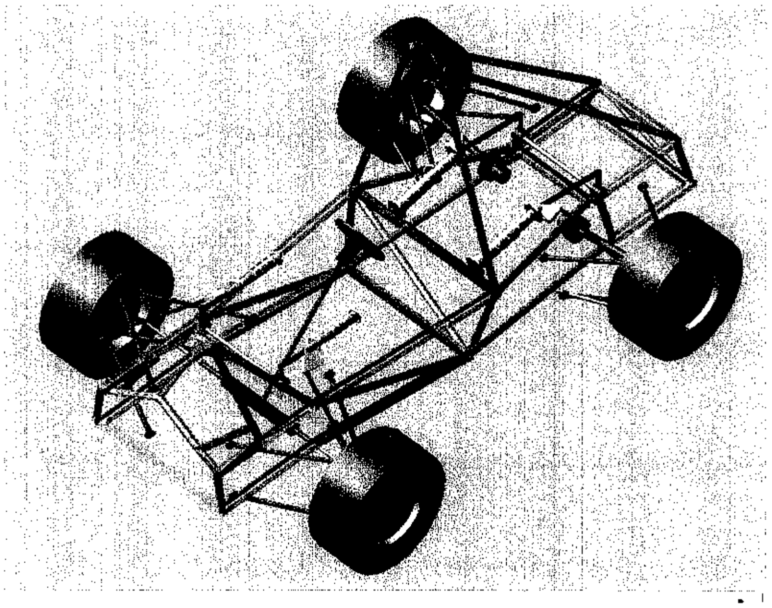


Figure 4.16: Full Vehicle Assembly in ADAMSCAR

4.4.2 Open Loop Step Steer Input Results

Simulations had been made with input of the followings:

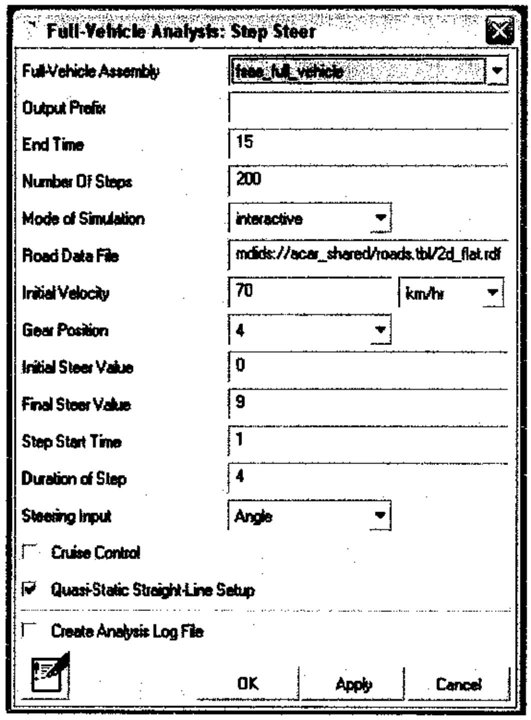


Figure 4.17: Input Parameters

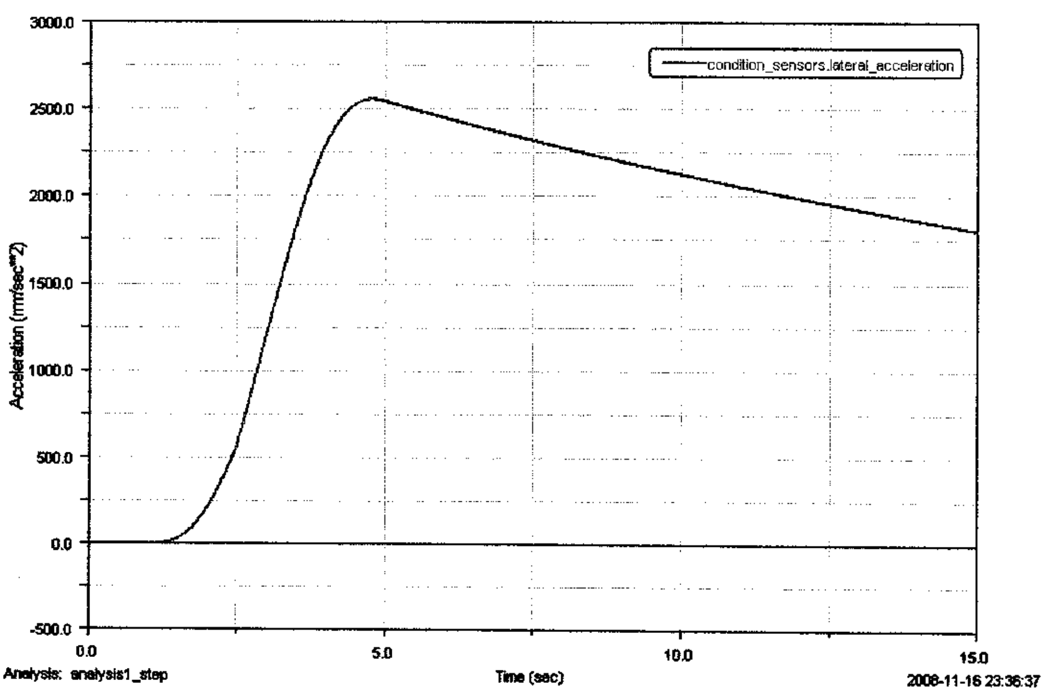


Figure 4.18: Lateral Acceleration versus time

The result shows that the acceleration of the car during cornering is not very smooth. There is some overshoot recorded which still need to be minimized in order to achieved a stable vehicle during cornering. Overall result shows the car behavior is in the desired configuration.

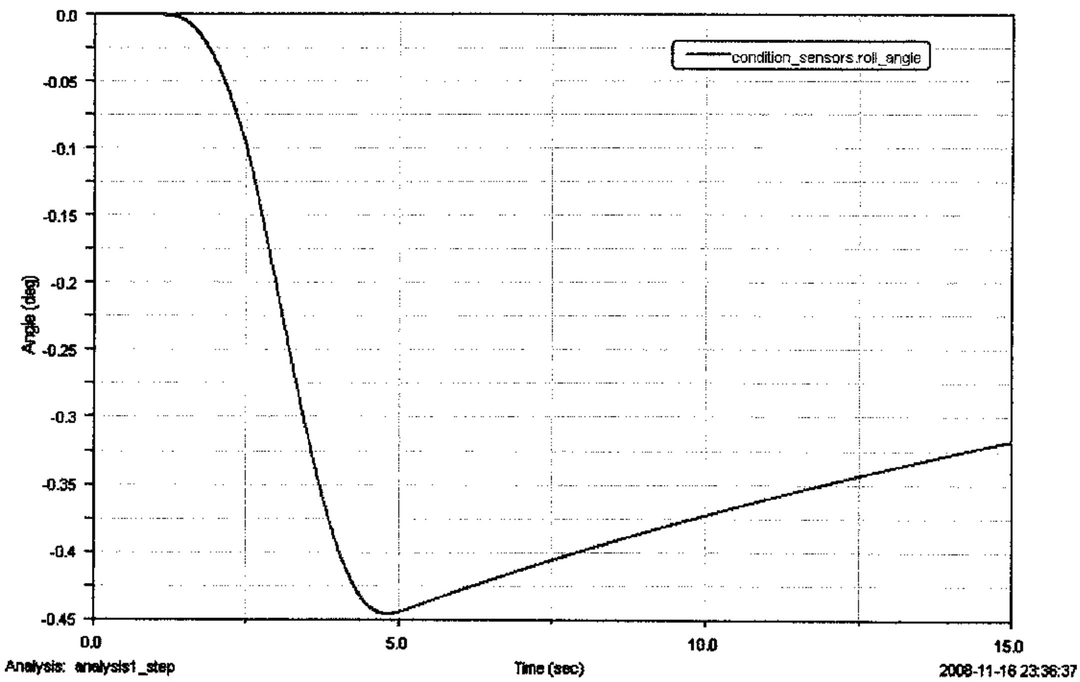


Figure 4.19: Roll Angle versus time

The maximum roll angle plotted is -0.44 degree when the vehicle is entering the corner. The vehicle tends to roll to outer radius of corner and this roll can be reduced by introducing the anti-roll bar in suspension system. There are still an overshoot recorded in the graph in which it should be minimize.

4.5 Fabrication Method

4.5.1 Wishbone Construction

Wishbones were constructed of two sections of circular hollow tubing, two rod end inserts, two rod ends and a spherical bearing housing with spherical bearing. The wishbone that has the pushrod/pullrod connected to it also has a connecting plate assembly. The usage of jig in construction of wishbone, aids in maintaining all the hard points and angles as per drawing. The suspension geometry is critical and by utilizing this jig, it allowed accurate location of the rocker mounts and the suspension pick-up points and also ensured that the rocker mounts were at the correct angle. The failure to do so will leave wishbone members in bending and consequently lead to excessive loading and failure.

Figure 4.20: Wishbone constructed according to Jig [10].

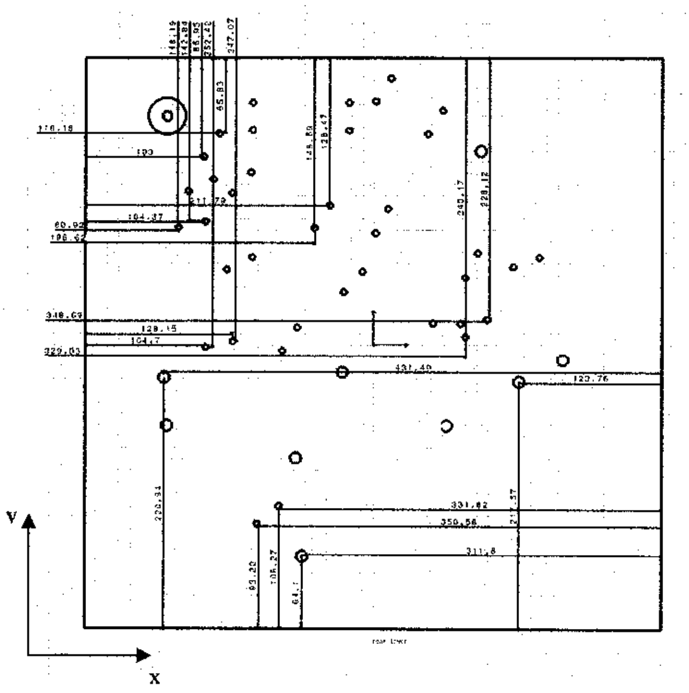


Figure 4.21: Example of x, y coordinates for jig holes [10].

4.5.2 Suspension Mounts (Clevis)

The fabrication process of Suspension Mounts requires either by using conventional milling machine or CNC milling machine. Since the accuracy and tolerance in developing a race car are vital, the usage of CNC milling machine ensures that all the dimensions are followed. These mounts are then being either welded or screwed into the square nodes on the chassis. The TIG welding machine is used if the mounts were needed to be weld on the chassis.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

As a conclusion, the main objective of this project which is to design and perform analysis on the suspension system of small race car has been successfully accomplished. For the modeling, CATIA V5R14 had been used and it had shown the ease of modeling for any such model. Packaging analysis performed via CATIA also shows the suspension design can adopted the chassis configurations without interfering with other components.

Finite Element Analysis (FEA) of the suspension components had been attained, and the results had shown optimal weight to strength ratio. With this in mind, the design of the vehicle dynamics is safe and also expected to be as agile as the design considerations are as of a concern.

Dynamics analysis performed via ADAMSCAR indicates that the vehicle is running smoothly on steady state and transient maneuvers during cornering. Further analysis on ADAMSCAR could aids in better understanding of vehicle behaviors during acceleration and braking.

The design has been improved to attain a better manufacturing process and reduction in material wastage. Attaining a simple yet lightweight and functional design is the key criteria that promise high points for the FSAE team.

5.2 Recommendations

As a recommendation to this project, fabricating the suspension system would be a good start to give realistic impression on the whole system and also gives better understanding on the processes involved. In addition, with the system are being fabricated, physical testing could be done in order to verify that the design and analysis which had been done earlier were correct.

The usage of ADAMSCAR software as a platform of simulation for the whole vehicle assembly should be deeply performed with various simulation conditions. The results obtained will be a good reference on the vehicle behavior before the vehicle could be fabricated.

Continuation on the optimization of the design aids in providing a good and reliable vehicle with accurate manufacturing processes. Improvements in design lead to much reliable vehicle in the future.

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APPENDIX A-1

Material Properties of Aluminium 7075-T6

Material	Aluminium 7075
Young Modulus	7e+010N_m2
Poisson Ratio	0.33
Density	2810kg_m3
Thermal Expansion	2.36e-005_Kdeg
Yield Strength	9.5e+007N_m2

APPENDIX A-2

Input Parameters in Suspension Analyzer V2.0

Suspension Data

Location	Type	Lt Out (X)	Lt Height (Y)	Lt Depth (Z)	Rt Out (X)	Rt Height (Y)	Rt Depth (Z)
Upper Ball Joint, cm	Input	58.321	39	0	58.321	39	0
Upper Frame Pivot, Front,	Input	21.979	34.386	-25.926	21.979	34.386	-25.926
Upper Frame Pivot, Rear,	Input	21.979	34.386	2.715	21.979	34.386	2.715
Lower Ball Joint, cm	Input	58.321	18	-.15	58.321	18	-.15
Lower Frame Pivot, Front,	Input	19	17.998	-28.522	19	17.998	-28.522
Lower Frame Pivot, Rear,	Input	19	17.998	2.715	19	17.998	2.715
Tie Rod on Rack, cm	Input	28.4	18.258	9.9844	28.4	18.258	9.9844
Tie Rod on Spindle, cm	Input	62.521	18.145	5.751	62.521	18.145	5.751
Spring Mount on Frame	Input	3.605	57.39	.866	3.605	57.39	.866
Push Rod Mount on Lower	Input	54.883	20.698	-.811	54.883	20.698	-.811
Bellcrank Axis Front, cm	Input	11.487	50.512	-13.442	11.487	50.512	-13.442
Bellcrank Axis Rear, cm	Input	11.487	50.512	-13.442	11.487	50.512	-13.442
Spring Mount on Bellcrank,	Input	5.744	55.87	-15.154	5.744	55.87	-15.154
Pushrod Mount on	Input	11.352	48.747	-19.161	11.352	48.747	-19.161
Spring Length, cm	Output	16.23	16.23	.00	16.23	16.23	.00
Spring Angle from Front	Output	54.60	54.60	.00	54.60	54.60	.00
Spring Angle from Side	Output	84.58	84.58	.00	84.58	84.58	.00
Spring Rate/Wheel Rate	Input (clc)	61.6	.0		62.49	.0	
Mtn.Ratio	Output	.017			.017		
Track, cm	Input	63.82	63.82	.00	63.82	63.82	.00
Tire Circumference, cm	Input (clc)	214	0	0	214	0	0
Tread Width, cm	Input (clc)	25.4	0	0	25.4	0	0
Camber, deg	Input	-.5	-.50	.00	-.5	-.50	.00
Caster, deg	Output	.41	.41	.00	.41	.41	.00
Caster Trail, cm	Output	.28	.28	.00	.28	.28	.00
Toe In, deg	Input	0	.00	.00	0	.00	.00
Toe In, cm	Output	.00	.00	.00	.00	.00	.00
Ideal Ackrmn Toe In, deg	Output	.00	.00	.00	.00	.00	.00
Ackrmn Error, deg	Output	.00	.00	.00	.00	.00	.00
King Pin Angle, deg	Output	.00	.00	.00	.00	.00	.00
Scrub Radius, cm	Output	5.50	5.50	.00	5.50	5.50	.00
Spindle Angle, deg	Output	-.50			-.50		
Instant Center Height, cm	Output	17.99	17.99	.00	17.99	17.99	.00
Instant Center Left, cm	Output	107.15	107.15	.00	107.15	107.15	.00
Roll Center Height, cm	Output	6.72	6.72	.00	0	0	0
Roll Center Left	Output	.00	.00	.00	0	0	0
Roll Stiffness, kg-m/deg	Output	.0					
Anti Squat, %	Output	.0	.0	.0	.0	.0	.0
Upper Arm Len	Output	36.63	44.88	36.73	36.63	44.88	36.73
Lower Arm Len	Output	39.32	48.49	39.43	39.32	48.49	39.43
Spindle Length, cm	Output	21.00			21.00		
Tie Rod Length , cm	Output	34.38			34.38		
Front View Swing Arm	Output	171.7	171.7	.0	171.7	171.7	.0
Side View Swing Arm	Output	1000.6	1000.6	.0	1000.6	1000.6	.0

APPENDIX A-3

Formula SAE 2008 Rules and Regulations

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3.2.3 Suspension

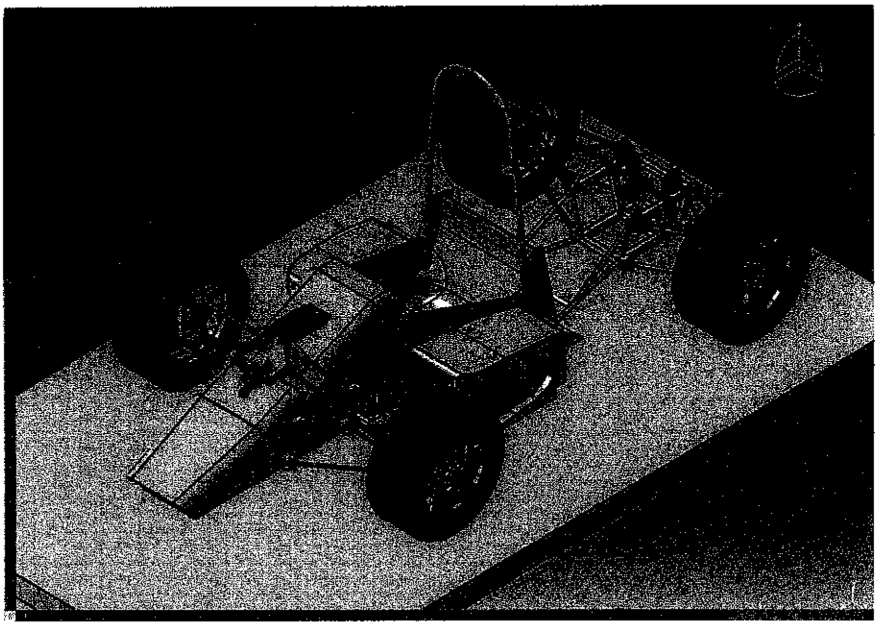
The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit.

All suspension mounting points must be visible at Technical Inspection, either by direct view or by removing any covers.

21 2006 Formula SAE® Rules

APPENDIX A-4

2008 FSAE Car



Isometric View