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THE EFFECT OF ANTI ROLL BAR ON THE HEAVY VEHICLE HANDLING

MOHD KAMIL AZRAN BIN MD NADZAR

MECHANICAL ENGINEERING
UNIVERSITI TEKNOLOGI PETRONAS
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TABLE OF CONTENT

LIST OF FIGURES

LIST OF TABLES

CERTIFICATION OF APPROVAL

The Effect Of Anti Roll Bar On The Heavy Vehicle Handling

By

Mohd Kamil Azran bin Md Nadzar

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Approved by,

____

(Name of Main Supervisor)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
December 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.

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MOHD KAMIL AZRAN BIN MD NADZAR

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Lastly, I offer my regards and blessings to all of those who supported me in any respect during the completion of the project.

ABSTRACT

Vehicle rollover contributes to huge number of tragic accident in recent years. Heavier vehicle have greater tendency to rollover due to their lower roll stiffness and smaller rollover threshold. While common passenger car could overcome the stability issue by means of speed reduction, having higher points of centre of gravity, it is impossible for heavier vehicle to seek balance between stability and speed. Simple balance spring device known as anti roll bar has been introduced to passenger car to enhance the roll stiffness of the suspension system without having to trade off speed and ride comfort. A system with the same goal has been developed for heavier vehicle. However, the system requires active feedback control system which resulted in the creation of a complex safety control system known as active anti roll system. Although the system is proven effective in preventing rollover of heavier vehicle, it is not really feasible due to the high cost of installation and maintenance. This project aim is to introduce the simple anti roll bar into lightweight truck and study the effect of the device in the vehicle. The device is design in such a way that it complies to the stiffness requirement of the lightweight truck studied in this project. The stiffness value is determined by many factors mainly contributed by the truck itself. The first part of this project is to find out the suitable stiffness value for the anti roll bar. Two sets of simulation is performed, one without any anti roll bar installed and the other one is with anti roll bar. The first simulation is to find out the right boundary condition to be set into the second simulation. It is found out that the critical value for vehicle speed is 50kmph and the cornering degree is 70^{o}. The next simulation uses these two values as the boundary condition to find the most suitable value of anti roll bar stiffness. From the simulation, it is found out that stiffness value is most suitable at 13,000 N/mm. The next step is to design the anti roll bar based on the stiffness value obtained. The anti roll bar is analyzed under various test including stress, deformation and force analysis. The anti roll bar design from the condition provided by the lightweight truck works well in preventing vehicle roll over. In order to design an anti roll bar for a vehicle, several criteria must be taken into consideration; especially the geometry and the bar roll stiffness.

CHAPTER 1
INTRODUCTION

1.1 Project background

Private vehicle especially in the class of sedan and wagon already have lower centre of gravity. Though this should have contributed to better vehicle stability, but when entering a corner, the vehicles tend to lean outward from the turning direction. Usually, a driver will be feeling this outward shifting of the body weight and the driver will be responding by reducing the vehicles' speed. This is done in order to prevent the car from tipping over towards the leaning direction. At certain speed, changing the vehicle direction (i.e. cornering) will result in the changing of the height of the vehicle's centre of gravity.

Performance cars which are usually brought to participate in rally or track race would be installed with an anti roll bar each at the front and rear suspension system. These kinds of cars need the ability to enter a corner without having to reduce its speed. However, they also must not roll over due to high speed while cornering.
Anti roll bar is a simple torsion bar that could help increase vehicle stability especially during cornering. The function of the anti-roll bar is implicit in its name - that is to resist the roll of the vehicle when cornering. The front anti-roll bars on our F-bodies are connected to the front lower control arms by end links and the center section of the bar is attached to the frame rails with bushings. The suspension does not know the difference between one wheel hitting a bump or the vehicle leaning. When either case occurs, the anti-roll bar twists and resists this motion. For example, during right cornering, left front wheel moves up into the wheel well and the right wheel moves down, twisting the bar. This twisting motion eliminates some of the body roll by making the entire front end squat. It increases the vehicle suspension's roll stiffness and consequently resists the motion of the chassis. This simple device works well in allowing a vehicle to make a corner in higher speed without being overturned due to the shifted centre of gravity point.

The idea is to redesign this device and introduce it to trucks. The effect of the existence of this device on heavy vehicle will be studied. In this case, the group of heavy vehicles under study covers only the lightweight truck. While higher centre of gravity is a major issue when tackling the stability problem of those trucks, nothing much can be done to lower the point due to the nature and requirement of the truck dimensions itself. Heavy trucks are usually equipped with active anti roll system. Even though the system works well in countering the stability problem, it brings along higher cost for installation and maintenance. Such condition is not suitable for a lightweight truck (i.e. 1 tonne truck). Simpler and much cheaper solution could be achieved with an anti roll bar suitable for this kind of truck.

1.2 Problem statement

Vehicle rollover is one of the main contributors to the fatal road accident. It is more critical when discussing about heavy vehicle. There are approximately 15,000 road tragedies involving rollover of commercial trucks [1]. It is estimated that 4 percent of all truck accident involves rollover, and from this number, 12 percents contribute to fatalities [2].

Although the incident is preventable through the maneuver of a highly skilled driver, majority of the occurrence could only be avoided with the introduction of external safety control system. Moreover, the ability of a driver to recognize the proximity of their vehicle to roll over is highly doubted [3].

Safety control system available for trucks currently is of complex form, involving active feedback control system. Although it is very effective when installed on heavy truck such as the container truck, it is not really viable for lightweight truck due to higher cost of installation and maintenance. Moreover, the complexity of the system could make the maintenance work harder. There must be some other alternatives which are much simpler and cheaper, especially for the small and lightweight truck.

1.3 Project Aim, Objective and Scope of Work

1.3.1 Aim

The aim of this project is to study the effect of a simple anti roll bar when installed at a lightweight truck. The idea is to come up with a design of a simple anti roll bar which is suitable to be installed on a lightweight truck. A model of a lightweight truck will be then simulated to conclude the effectiveness of the device against the rollover problem of such truck. This project is hoped to give some ideas on simple solutions which are available in avoiding vehicle rollover. Besides having a complex set of a safety control system, simpler solution must also be explored to provide more options to prevent such incident, especially for a lightweight truck.

1.3.2 Objective

This project has several objectives to be achieved:

1. To study the effect and determine the effectiveness of the device when installed at a lightweight truck. The device is expected to increase the truck's suspension roll stiffness and reduce the motion of the truck chassis.
2. To design a simple anti roll bar suitable for installation on a lightweight truck from the stiffness value of the anti roll bar obtained from the simulation. The dimensions, the geometry and the material of the device must be able to withstand the force exerted on the truck suspension.

1.3.3 Scope of Work

1. The vehicle under study is from heavy vehicle group. The focus of this project is however the effect experienced by a lightweight truck.
2. The rollover effect investigated in this project is caused by vehicle cornering.
3. The anti roll bar designed must be suitable and able to withstand the greater load exerted by a lightweight truck, although the basic idea of the device came from those installed on common passenger car.
4. The anti roll bar is of solid type.

5. The scenario in the analysis is defined as the worst case scenario as such: the vehicle is turning in the manner of which the inside tires are lifted off from the ground in which all the vehicle weight is shifted onto the outside tires.
6. The load exerted by onto the anti roll bar is defined as such: 1) maximum allowable load from a fully loaded heavy vehicle. 2) the load on the vehicle is static 3) load distribution is uniform onto each of the vehicle's tire
7. The design of the anti roll bar will be done using CATIA. Several type of anti roll bar is designed. The first basic design is taken from the simplest design usually found in automotive engineering textbook. The second design is taken from the previously studied and tested anti roll bar but is modified in term of its dimension to suit the size requirement of the lightweight truck under study. The third design is taken from the real anti roll bar device which is captured into CATIA.
8. Analysis on the anti roll bars will be done using ANSYS. There are several types of analysis which will be conducted. They are load and moment analysis, stress analysis and the bar stiffness analysis.
9. The simulation of the lightweight truck model will be done using ADAMS.

1.4 Project Overview

The project will require a designing stage to take place during the construction of the design of the anti roll bar. The design is then will be subjected to analysis using a computer software. Several tests on the material properties and design feasibility will be carried out until it meets the project requirement and specification. The anti roll bar design will be then used for simulation and analysis using data from a real lightweight truck. This is done in order to determine the workability and effectiveness of the anti roll bar design when installed on a lightweight truck.

1.5 Planning & Research

At this stage, a clear goal and objective are set to drive the project on the right path. In order to ensure the project is achievable and realistic, a Gantt chart is created as a form of project plan. The project plan will help the time and resource management besides meeting the milestone within the given time frame.

There are several approaches used in gathering the information about the project. Interview with UTP lecturers is a need since some of them are specialized in the related cluster, thus having vas knowledge that can help with the project development. Besides that, review on previous works by other researchers also help in giving strong literature review and contribute some ideas and insight on this project.

1.6 Design

After a collection of data is obtained, construction of the design on an anti roll bar took place. The designing process will be done using CATIA. All specification such as the dimensions, geometry and material properties is determined during this stage.

1.7 Analysis

During the analysis stage, the designed anti roll bar will be subjected to testing to determine the suitability and workability of the device under live load. The analysis will be carried out using ANSYS. Several data will be crucial during this stage such as force exerted by truck suspensions on the device.

1.8 Simulation

The designed anti roll bar will be then used in a simulation of the movement of a lightweight truck in ADAMS. This stage is to determine the effectiveness of the device when installed on a truck model.

1.9 Project flow overview

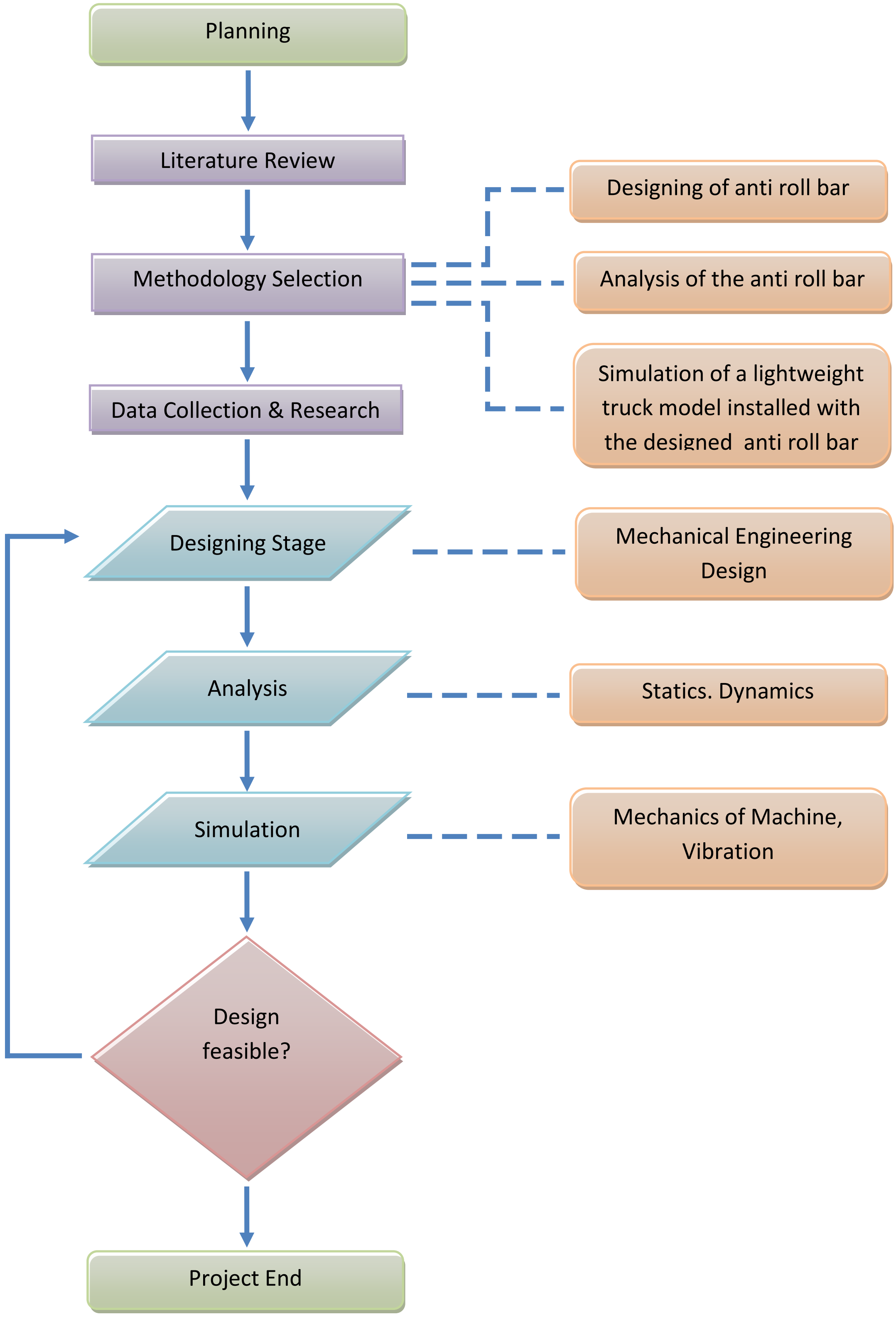


Figure 1.1: Project Flowchart & Overview

CHAPTER 2

LITERATURE REVIEW & THEORIES

Commercial trucks have low level of basic roll stability, which set them apart from lighter vehicle. Roll stability is measured by the static rollover threshold which is expressed as lateral acceleration in gravitational units [1]. Passenger cars have rollover threshold of greater than 1g. However, lightweight trucks, vans and SUVs have rollover threshold which range from 0.8 to 1.2g [4]. Loaded truck usually have rollover threshold of well below 0.5g).

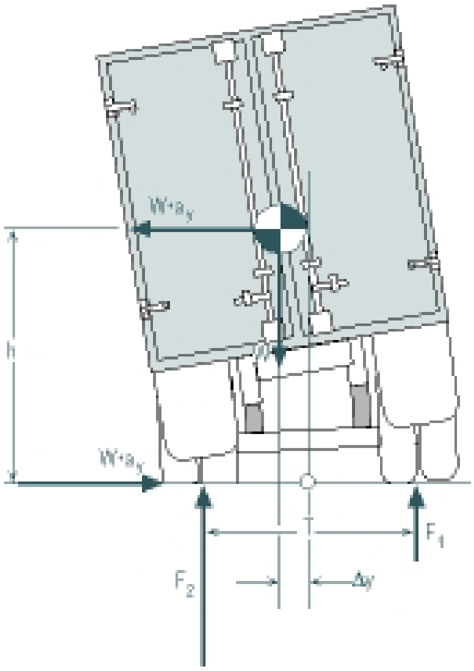


Figure 2.1: A simplified free body diagram of a heavy vehicle in steady turn

There are increasing proposal of using anti roll bar control system to improve vehicle stability and consequently reducing the tendency of vehicle rollover. When entering a corner, vehicle, regardless of the height of its centre of gravity, will tilt out of corners under the influence of lateral acceleration. At this moment, the centre of sprung mass is shifted outboard of the vehicle centerline. This will generate a destabilizing moment that reduces the vehicle roll stability [5]. Figure 2.1 illustrates the forces involved during the cornering of a heavy vehicle. Notice the leaning of the vehicle body which could cause the vehicle overturn.

There are two basic elements that constitute the total roll stiffness of a vehicle suspension system, which are the spring of the suspension system and the auxiliary effect from other components. Anti roll bar is a device which falls under the latter category [6]. Anti-roll bar is a suspension element used at the front, rear, or at both ends of a car that reduces body roll by resisting any unequal vertical motion between the pair of wheels to which it is connected. Installation of a device of this kind could actually greatly help stabilizing the vehicle chassis as it stiffens the suspension springing when the body rolls or one wheel goes over a bump or dip in the road [7].

When determining the kind of anti roll bar which is suitable for a lightweight truck, it is important to consider the difference in the load exerted by the suspension system between those found in the truck and the usual passenger car. The design of the anti roll bar should not be the same as those installed on the passenger car. Bend location on the device should be reduced, taking into account the higher load exerted by the suspension system of a truck [8]. Many cases of fractured anti-roll bars after a 100,000 km of travel are reported. All of the bars are fractured at nearly the same bended location. Even without carrying external forces, an anti roll bar will continuously be subject to lateral displacement because the suspension's transverse links guide its end on circular path which cause considerable bending moments and lateral movement of the bar [9].

During their research on anti roll bar fracture, Bayrakcen, Tasgetiren & Aslantas modeled the device via ANSYS 7.1/Mechanical module. After the definition of the geometry, the material properties developed by the mentioned experimental studies are entered into the program and a static stress analysis is carried out. Figure 2.2 shows the meshing of the anti roll bar used in their research.

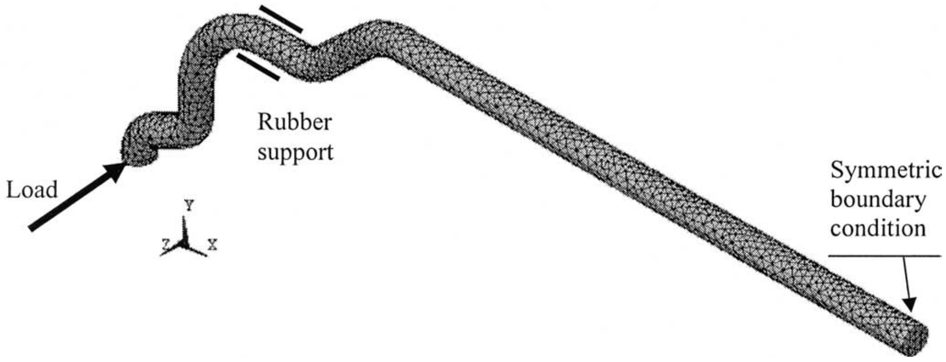


Figure 2.2: Finite element analysis of an anti roll bar

In order to achieve higher spring rate in roll at the front axle of any vehicle, despite the softer bounce rate, anti roll bars are adopted. It reacts only to antimetric travel of the two wheels and has no effect on symmetrical travel. Nowadays, anti roll bar is preferably of kind of torsion bar, especially on front wheels. Wheel load differences and tire slip angles are influenced mainly by several factors. The distributions of roll spring rates between front and rear axles play the major factor. However, the more prominent effect comes from incorporation of anti roll bars or compound springs. An anti roll bar on the front axle increases the total roll rate of the vehicle and reduces the roll angle, hence the wheel load difference at the rear axle. Anti roll bars increases the spring rate of the vehicle without changing the bump rate. Increasing the device rate leads to reduction in the lifting tendency. Since, the bar also reduce the reduce angle, the change of inclination of the trajectories of the tire contact points which lead to jacking up will also reduced greatly [9].

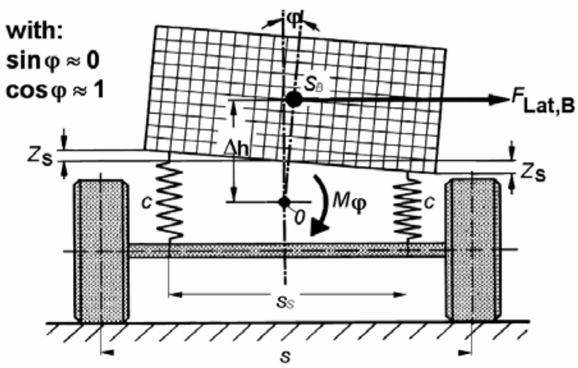


Figure 2.3: Equivalent model for a vehicle suspension system

The centrifugal force $F_{Lat,B}$, acting at the centre of gravity of the body will contribute to a torque M_ϕ around the roll axis as illustrates in the Figure 2.3:

$$M_\varphi = F_{LatB} \cdot \Delta h \cdot \cos\varphi + m_B \cdot g \cdot \sin\varphi \qquad (2.1)$$

with Δh: vertical distance between the centre of gravity of the body SB and the roll axis.

The inclination of the roll axis in the vehicle longitudinal plane is thus neglected. The torque created by the body weight is also usually neglected (sin ϕ << cos ϕ).

With these simplifications, the aligning torques applied by the body springs around the roll axis is given by:

$$F_{LatB} \cdot \Delta H = 2 \cdot \frac{S_{Sf}}{2} \cdot C_{Bf} \cdot f_{Sf} + 2 \cdot \frac{S_{Sr}}{2} \cdot C_{Br} \cdot f_{Sr} \qquad (2.2)$$

with: $S_{Sf,r}$: Spring track width in front, rear

$f_{Sf,r}$: Spring compression in front, rear

$C_{Bf,r}$: Body spring rigidity in front, rear

If $f_F \approx \varphi . \frac{S_s}{2}$, one will finally find that for the roll angle, φ

$$\varphi = \frac{2 \cdot \Delta H}{C_{Bf} \cdot S_{Sf}{}^2 + C_{Br} \cdot S_{Sr}{}^2} F_{LatB} \qquad (2.3)$$

The roll angle 'ϕ' is thus inversely proportional to the square of the spring track width. For a small body inclination, the spring track width should therefore be as high as possible while driving along curves.

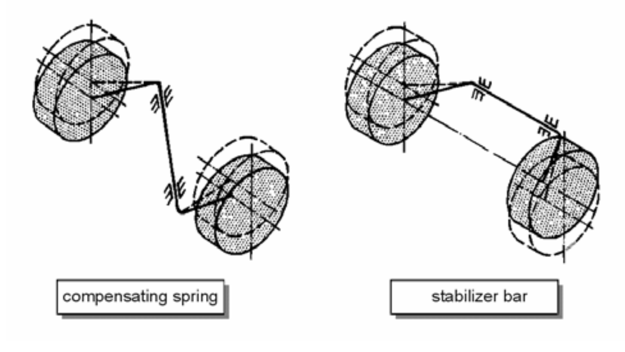


Figure 2.4: Functional spring of anti roll (stabilizer) spring and compensating spring

During the rolling motion of the body, the anti roll bar (stabilizer) such in Figure 2.4 is exerted with torque and thus provides a self-aligning torque around the roll axis, which reduces body inclination. In case of a pure lifting motion of the body suspension on the axle considered, the stabilizer will have no effect. A stabilizer track with a width S_{Stab}, to which the stabilizer stiffness C_{Stab} relates, is defined analogous to the spring track width S_F. Stabilizer stiffness C_{Stab} then corresponds to the stabilizer force at the ends of the stabilizer bar referred to half the differential compression of these ends.

Thus, the roll angle applies is:

$$\varphi = \frac{2 \cdot \Delta H \cdot F_{LatB}}{C_{Bf} \cdot {S_{Ff}}^2 + C_{Stabf} \cdot {S_{Stabf}}^2 + C_{Br} \cdot {S_{Fr}}^2 + C_{Stabr} \cdot {S_{Stabr}}^2} \tag{2.4}$$

While static level modifications could be eliminated by load-sensitive regulating springs, curve inclination can only be reduced in small limits by means of stabilizers. The relationship between body inclination, body natural frequency and body acceleration is therefore considered in the following:

$$\varphi = \frac{2 \cdot \Delta H \cdot F_{LatB}}{(C_B + C_{Stab}) \cdot {S_S}^2} \tag{2.5}$$

By using integral force:

$$F_{latB} = \frac{m_B \cdot V^2}{r} \tag{2.6}$$

We will get:

$$\varphi = \frac{2 \cdot \Delta H \cdot m_B \cdot V^2}{(C_B + C_{Stab}) \cdot {S_S}^2 \cdot r} \tag{2.7}$$

With the equation:

$$f_{eB} = \frac{1}{2\pi} \cdot \sqrt{\frac{2 \cdot C_B}{m_B}} \tag{2.8}$$

For the body eigenfrequency of the stroke oscillation, resolved to m_B

$$m_B = \frac{2 \cdot C_B}{4\pi^2 \cdot {f_{eB}}^2} \tag{2.9}$$

The roll angle is then given by:

$$\varphi = \frac{2 \cdot \Delta H \cdot V^2 \cdot 2 \cdot C_B}{(C_B + C_{Stab}) \cdot {S_S}^2 \cdot r \cdot 4 \cdot \pi^2 \cdot f_{eB}} \tag{2.10}$$

Should the ratio of the shares of the rolling moment M_ϕ supported by the stabilizers on front and rear axles differ from the ratio of the shares supported by the body suspension, or if only one axle has a stabilizer spring installed, then not only the roll angle will be reduced, but the distribution of the wheel load differentials, which occur during cornering between the wheels of the right and left vehicle sides, among front and rear axle will be influenced [10].

In any stress analysis, there must be some consideration on the yield criterion as a tool to measure the yielding of a material. Von Mises is a plasticity theory that works well on ductile material such as metals. Prior to yield, material response is assumed to be elastic. Von Mises yield criterion can be formulated in terms of the von Mises stress, σ_V. The stress is used to predict yielding of a material. Yielding begins when the elastic energy of distortion reaches a critical value. For this, the von Mises criterion is also known as the maximum distortion strain energy criterion. This comes from the relation between and the elastic strain energy of distortion with the elastic shear modulus [11].

Figure 2.5 shows the comparison between von Mises stress and Tresca shear. Observe that Tresca's yield surface is circumscribed by von Mises'. Therefore, it predicts plastic yielding already for stress states that are still elastic according to the von Mises criterion. As a model for plastic material behavior, Tresca's criterion is therefore more 'conservative', which basically means 'on the safer side'.

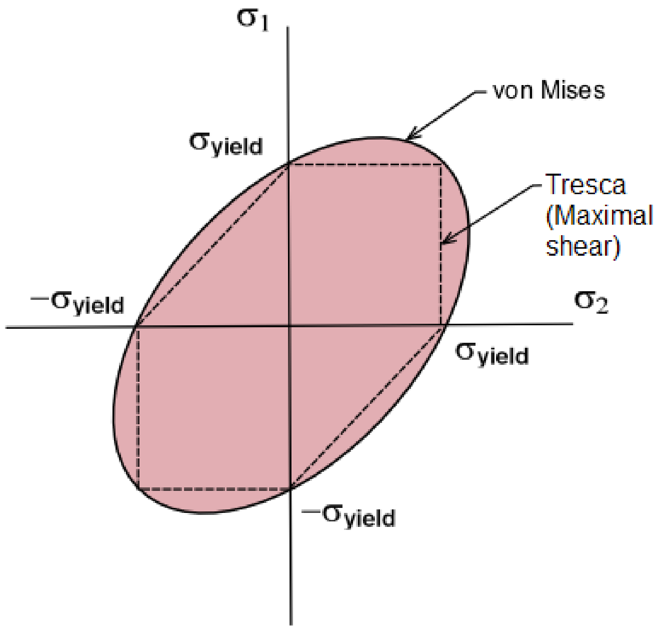


Figure 2.5: Projection of the von Mises yield criterion into the σ1, σ2 plane

CHAPTER 3
METHODOLOGY

3.1 PART 1: Vehicle Modeling & Simulation

The first part of this project is to model and simulate the vehicle studied in ADAMS Car software. The purposes of doing so are described in the following points:

- To obtain the center of gravity and mass moment of inertia of the vehicle
- To determine the boundary conditions (vehicle speed and cornering degree) for vehicle step steer simulation
- To determine the suitable value of the anti roll bar stiffness in step steer simulation
- To study the vehicle behavior in a cornering event when equipped with an anti roll bar

Table 3.1 indicates all technical specifications for a Class 3 Heavy Vehicle which is under Light Duty Category. The division of heavy vehicle category is according to the gross vehicle weight rating (GVWR). GVWR is the maximum allowable total mass of a road vehicle when loaded including the weight of the vehicle itself plus fuel, passengers, cargo, and trailer tongue weight. Class 3 Category vehicles have GVWR between 10,000 lbs – 14,000 lbs.

Table 3.1: Vehicle technical data sheet

NPR/NPR HD Diesel Specifications		
Base Model Description	**NPR DIESEL**	**NPR HD DIESEL**
Dimensions		
-Wheelbase (in.)	109, 132.5, 150.0, 176.0	
-Cab to Axle (in.)	86.5, 110.0, 127.5, 153.5	
-Cab to End of Frame (in.)	129.6, 153.1, 170.6, 196.6	
-Overall Length (in.)	200.5, 224.0, 241.5, 267.5	
-Body Length (ft.)	10-12, 14, 16-18, 20	
-Overall width (in.)	73	
GVWR/GCWR	12,000/18,000 lbs.	14,500/20,500 lbs.
Body/Payload Allowance [1]	6,140-6,328 lbs.	8,557-8,792 lbs.
GAWR		
-Front	5,360 lbs.	
-Rear	8,840 lbs.	9,880 lbs.
Front Axle Capacity	6,830 lbs.	
Rear Axle		
-Capacity	14,550 lbs.	

-Ratio (AT)	4.555	4.777 (AT) 4.300 (MT)
Suspension, Front & Rear[2]		
-Type	Tapered/Multi-Leaf	
-Front Suspension Capacity	8,440 lbs.	
-Rear Suspension Capacity	9,880 lbs.	
Frame		
-Section Modulus	7.20 in.^{3}	
-Resistance Bending Moment	316,800 lbs.-ft./in.	
Service Brakes		
-System	Vacuum/Hydraulic w/4-Channel ABS	
-Front	Disc	
-Rear	Drum	
Exhaust Brake	Vacuum Operated	
Engine		
-Type	Turbo/Intercooled Diesel 4HK1-TC	
-Displacement	5.2 L (317 CID)	
-Oil Level Indicator	Dash-mounted oil level check switch and light	

There are two weights rating numbers in the table above which is GVWR and GCWR. However, we are more interested in the GVWR because GCWR indicates the maximum total mass of the vehicle inclusive of towed vehicle linked to the vehicle. In the case defined in this project, GVWR is more relevant since it represent the maximum loads on the vehicle only. As indicated in the table, this vehicle GVWR is about 12,000 lbs which is ideal for our definition of heavy vehicle in this project.

3.1.1 Location of vehicle's Center of Gravity (CG)

In order to perform cornering test on modeled vehicle, it is important to firstly determine the location of its center of gravity (CG). Passengers' cars, having lower center of gravity are far more stable than heavier vehicle such as truck. This fact becomes more apparent when we consider a cornering event. Apart from the weight difference of the two vehicles, the location of CG plays vital part in influencing the roll effect onto the chassis of the vehicles during the course. Using the truck dimensions obtained from Daihatsu technical data sheet, the truck is re-modeled in CATIA for further analysis and measurement. The following data in Table 3.2 of dimensions are used.

Table 3.2: Vehicle dimensions

Body Parts	Dimensions
Head length	1.4 m
Head height	2.1 m
Head/ body width	1.8m
Body length	4.3 m
Body height	2.5 m

Figure 3.1 shows the model of truck studied in this project. The shape and its solid characteristic play important part in determining the center of gravity. The value of inertia and the center of gravity is then measured using CATIA.

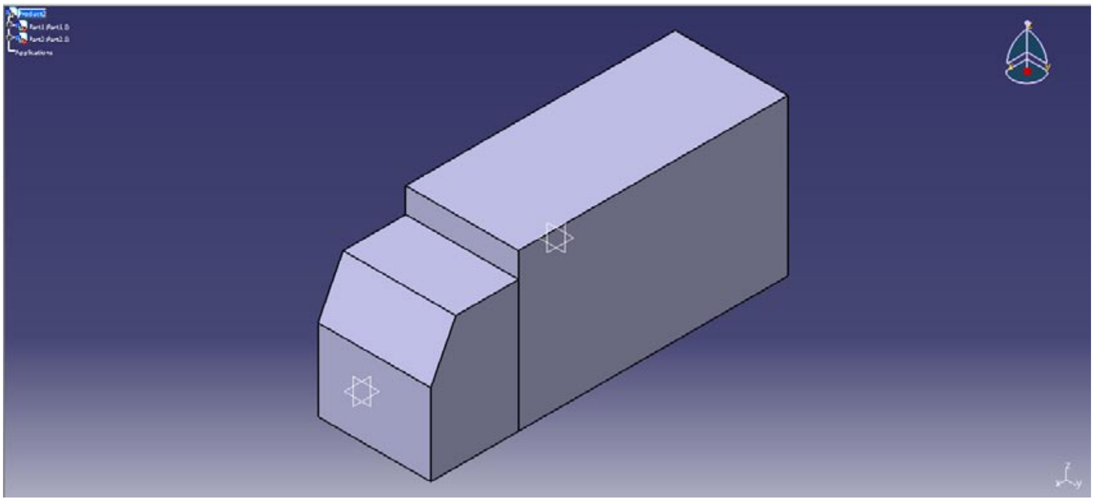


Figure 3.1 Truck model in CATIA

Figure 3.2 shows the point at which the center of gravity of this truck coincides. The blue line represents y-axis, the red line represents x-axis and the green line represents z-axis. The point at which these three lines meet is where the center of gravity is located. The location is given by CATIA as such: Gx = 2322 mm, Gy = 900 mm, Gz = 1225 mm. This coordinate will then integrated in ADAMS Car for the next simulation.

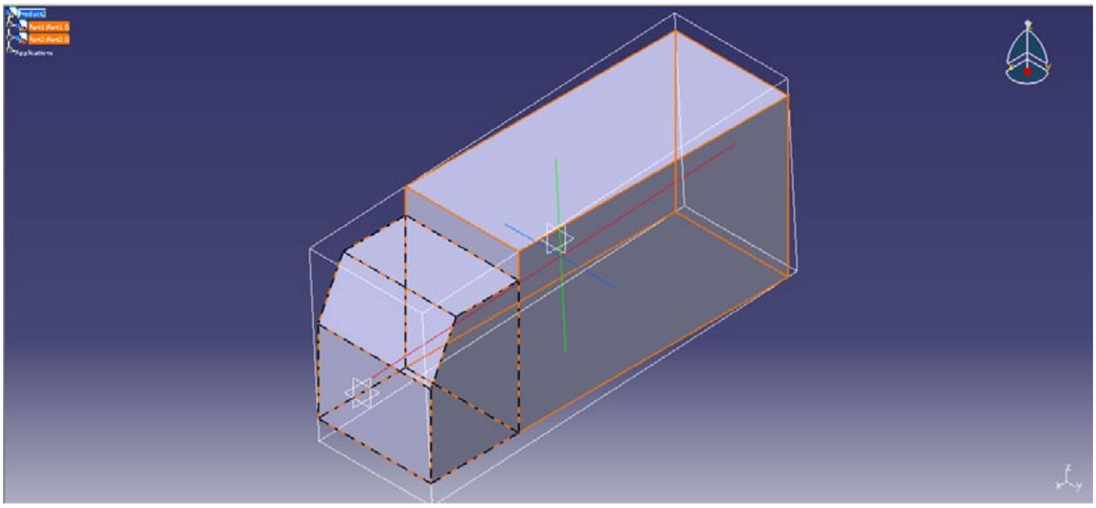


Figure 3.2 Model used to determine the truck center of gravity

3.1.2 Vehicle Rendering in ADAMS Car

Figure 3.3 shows the modeled vehicle in ADAMS Car. ADAMS Car 2007 does not has any template for truck. However, closest or near accurate can be obtained with some alteration made especially to the model hard point/ dimensions to replicate the shape and the size of a truck. Apart from that, values such as the vehicle weight and center of gravity also can be modified to suit the need of this project. Most accurate result can be obtained using ADAMS Car 2010; however, the unavailability of the software must be compensated using the readily available software.

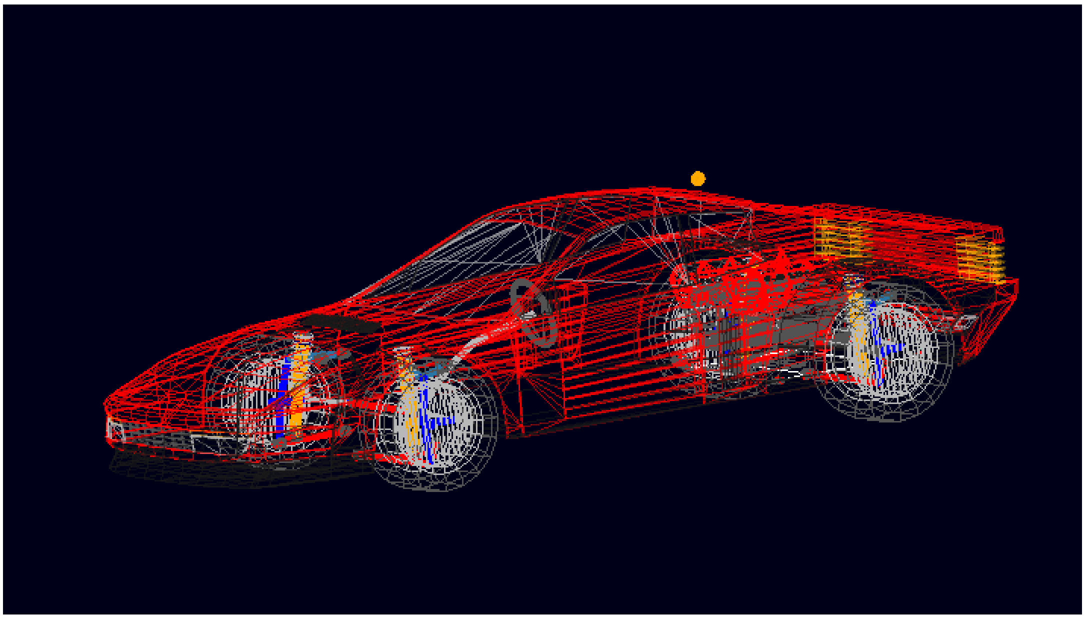


Figure 3.3 Vehicle model in ADAMS Car

Notice in the figure, there is a yellow point depicted. This point represents the corrected center of gravity using the values previously obtained from CATIA. ADAMS Car integrates the use of reference points (0.0, 0.0, 0.0). Hence, it is important to locate this reference point first before any modification can be made. This is a must to prevent any deviation from the correct dimensions assume throughout this project. The location of center of gravity as depicted in the figure is at outside above the model. This implies the use of the corrected value in our simulation. The height of a truck is definitely much higher than that of the model of the available template in ADAMS Car, i.e. a red Ferrari. Hence, it is only make sense if the location of the center of gravity is much higher than the Ferrari, thus, the location is logically must be outside above the car. This further verifies the reliability of this simulation. The value of the truck weight and its mass moment of inertia also are modified into the template to allow the replication of true situation to be made possible.

Although the template still remains the shape of the Ferrari, the engineering and technical data are already representing that of a truck. The Ferrari shape depicted throughout the simulation is merely aesthetic in order to show the movement of any vital part of the vehicle.

3.1.3 Step Steer Simulation (without anti roll bar)

Steep steer simulation is an event where a vehicle is driven in a straight line for a few seconds before it is turn at a certain amount of degree to make a cornering. The important variables in this simulation which will determine the end result is the vehicle speed and cornering degree. Hence, in line with our intention to study the behavior of the vehicle throughout a cornering event and after it exit the corner, this simulation is ideal to be utilized for that purpose.

ADAMS Car is used to run a simulation to determine the critical speed and road corner degree at which the vehicle will roll over. The cornering event considered in this simulation is based on the boundary conditions determine earlier of which the situation attended is when the truck enters a corner. The vehicle should enter and leave the corner in a straight line, unless it fails o make a perfect turn such as during vehicle roll over.

There are two important variables that must be considered when studying a roll over event besides the center of gravity of the vehicle itself. They are the road cornering degree and the velocity of the vehicle when it drives through the corner. The most common road cornering degrees are chosen from the General Estimate System [2]. The velocities of the vehicle are selected from the most possible speed a truck could made when entering a corner.

The simulation is performed at duration of 15 seconds. At 5 seconds lapse, the truck begins to turn i.e. entering the corner. The truck will take 3 seconds to finish the corner before beginning to move at a straight line again. The following table 3.3 summarizes the values of the road cornering degree and the truck velocity used in the simulation.

Table 3.3: Sets of variables for simulation

Road Cornering Degree (^{o})	Truck Velocity (kmph)
30	30
	40
	50
	60
40	30
	40
	50
	60
50	30
	40
	50
	60
60	30
	40
	50
	60
70	30
	40
	50
	60
80	30
	40
	50
	60
90	30

	40
	50
	60

The observation on the end result for each simulation will be added later during result recording. A graph of damper displacement against simulation time is plotted for every simulation performed. The graph will give us the insight on the vehicle tire location. There are two types of graphs that can be produced of which one will show a smooth cornering and the other will show abrupt change in the tire location which indicates that vehicle roll over event is happening.

3.1.4 Steep Steer Simulation (with anti roll bar) & selection of ARB stiffness

The value of spring stiffness of the anti roll bar installed to our vehicle is varied to determine which value is suitable to resist the rolling motion of the vehicle when it enters a corner. The following table summarizes the values of stiffness chosen for this test. In the previous simulation, we have selected the most suitable value of the boundary conditions for the next simulation.

The step steer simulation is again used in this part, where the boundary conditions are now already fixed. The changing variable in this simulation is the anti roll bar stiffness. In theory, the higher the stiffness value, the greater the ability of the bar to help resist the rolling motion of a vehicle. However, having a bar with too high stiffness value could affect the vehicle ride comfort. Hence, given the highest risk scenario define in this simulation (the highest possible vehicle speed and cornering degree), we are going to select the lowest stiffness value that could help resist the rolling motion. The stiffness value as in the Table 3.4 follow are chosen based on the weight of the vehicle which influences the most on the vehicle body roll.

Table 3.4: Anti roll bar stiffness value

Anti Roll Bar Stiffness (N/mm)
10,000
11,000
12,000
13,000
14,000
15,000
16,000

17,000
18,000
19,000
20,000

3.1.5 Anti roll bar design and geometry determination

The value of ARB spring stiffness is obtained from the following equation. Notice that the geometry factors are involved in calculating this value. Hence, by having a fixed length due to the width of the vehicle, and fixed diameter for example, one can vary any other geometry of the ARB design.

$$R = \frac{\pi G d^4}{16 L b^2} \qquad (3.1.5.1)$$

In which R is the ARB spring stiffness, G is the modulus of the material, d is the bar outside diameter, L is the length of the bar and b is the length of the arms of the bar. This equation can also be expressed as:

$$R = \frac{\pi G d^4}{(0.4244 \times A^2) \times B \times (0.2264 \times C^2)} \qquad (3.1.5.2)$$

In this case, because there are many values of bar stiffness that we can select, we should choose the lowest stiffness possible. This is to achieve the balance between ride comforts and roll stiffness. High stiffness value will cause the vehicle to lose it ride comforts, in which even smallest bump on the ride will have vibrating effect transferred to the components of the car. If this situation is prolonged, the vehicle will face the risk of failure due to fatigue and vibrations.

3.2 PART 2: Analysis on the Anti Roll Bar

3.2.1 Anti roll bar material specification

The material selected for the anti roll bar is AISI/ SAE 4130. The material is selected based on the study by Bayrakceken, Tasgetiren & Aslantas [8] and the suggestion in an SAE website.

3.2.2 Anti roll bar geometrical parameters

The overall width of the vehicle is 73 inch which is about 1860 mm. This is the crucial number that we will use while modifying the anti roll bars to fit our heavy vehicle. Usually, an anti roll bar is distanced at about 3 inches from the truck tire on both ends. Considering the width of common medium duty truck tire which is about 9 inches, we will get the overall length of an anti roll bar:

$$ARB\ Length\ (in.) = Overal\ \ Width - 2(Tire\ Width) - 2(ARB\ distances)$$
$$ARB\ Length = 73 - (2 \times 9) - (2 \times 3)$$
$$ARB\ Length = 49\ in.$$

The overall length of the anti roll bar will be the basis of the modification on available anti roll bar taken from the previously tested design and the actual real anti roll bar design.

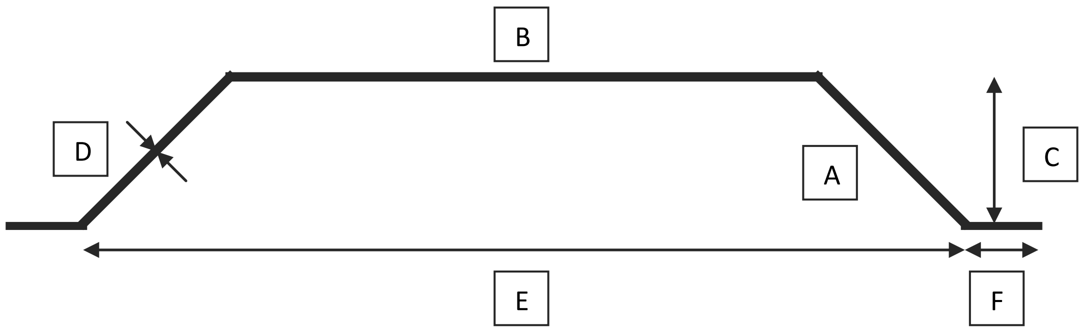


Figure 3.4: Anti roll bar representation

Consider the above linkage in Figure 3.4 representing a simple anti roll bar. In determining the most suitable design especially with regards to the bar stiffness, there are four most important variables which will determine the size of the bar. They are the bar length, the arm length, the vertical length and the bar outside diameter. The bar length (B) is the distance from one curve to the other curve. The arm length (A) is the whole distance of the bended arm. The vertical length (C) is the distance measured vertically from the bar length and the tip of the bended arm. The outside diameter (D) is used if the design use solid bar. All these variables (A, B, C, D) may be varied according to the stiffness required by the designer. However, the design is confined within our overall length (E). In our case, the overall length is found to be about 49 inches. Since the goal is not to come up with new anti roll bar, hence, the available anti roll bars will be modified in term of its dimension to suit or size requirement.

$$R = \frac{\pi G\, d^4}{(0.4244 \times A^2) \times B \times (0.2264 \times C^2)} \qquad (3.2.2.1)$$

Using the value into the above equation, we obtain the other parameters for the dimensions as follow

Table 3.5: Anti roll bar geometry parameters & dimensions

Anti Roll Bar Geometry Parameters & Dimensions	
ARB Overall Length	49 in. (1244.6 mm)
Bar Length (B)	36.5 in (927.1 mm)
Arm Length (A)	9.6 in (243.84 mm)
Connector Arm Length (F)	3 in (76.2 mm)
Diameter (D)	2 in (50.8 mm)
Bend Angle	30^{o}

Figure 3.5a until Figure 3.5d show the base design of an anti roll bar design. This design appears in a lot of automotive engineering textbook. This design works well in explaining the basic principle of an anti roll bar. The function of the bended arms is to give the opposing torque towards the rollng effect on the vehicle body. The arms are shown in their most simplest form in order to simplify the understanding of this bar functions. Eventhough this kind of bar design will not appear int the real life application, but it provides good basis for understanding the effect of loads during the analysis later.

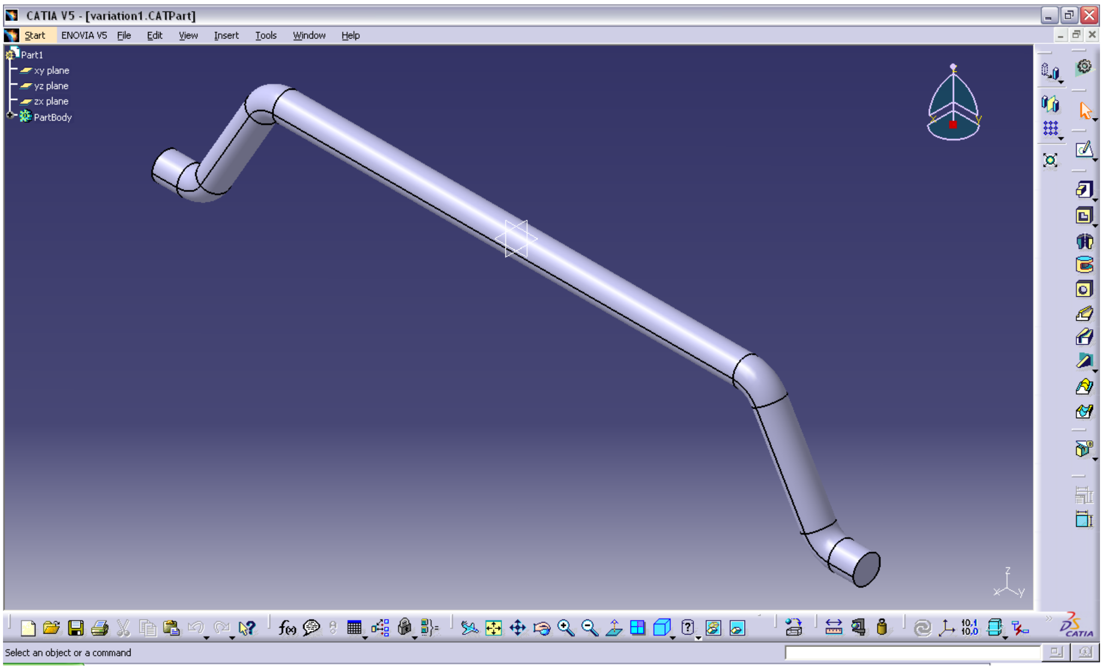


Figure 3.5a: Base design of an anti roll bar

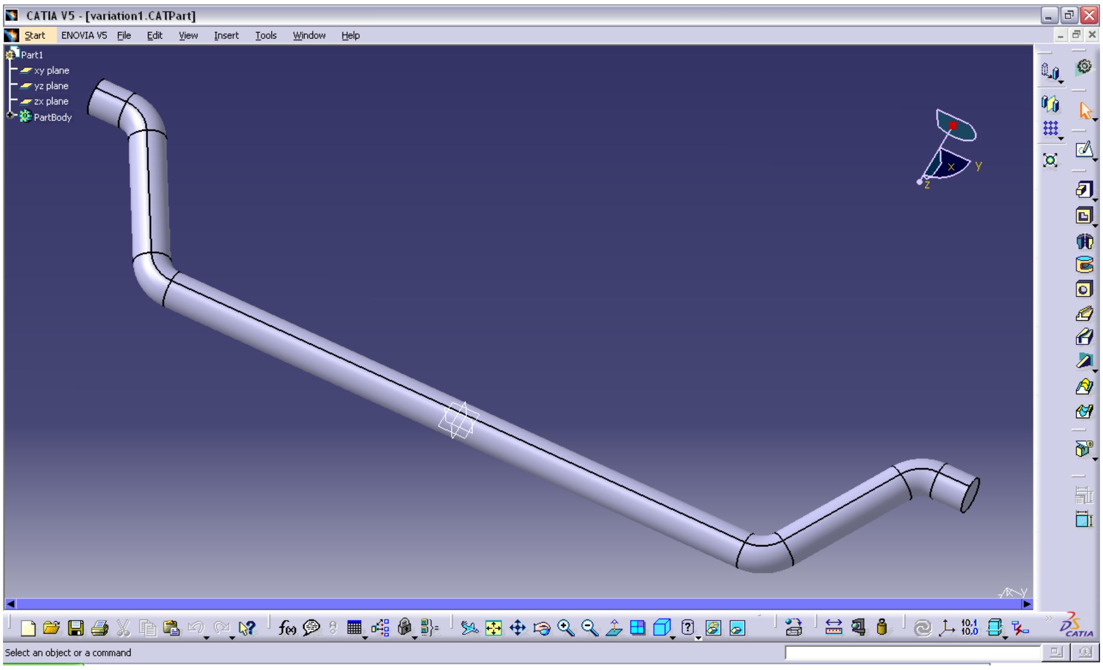


Figure 3.5b: Base design of an anti roll bar

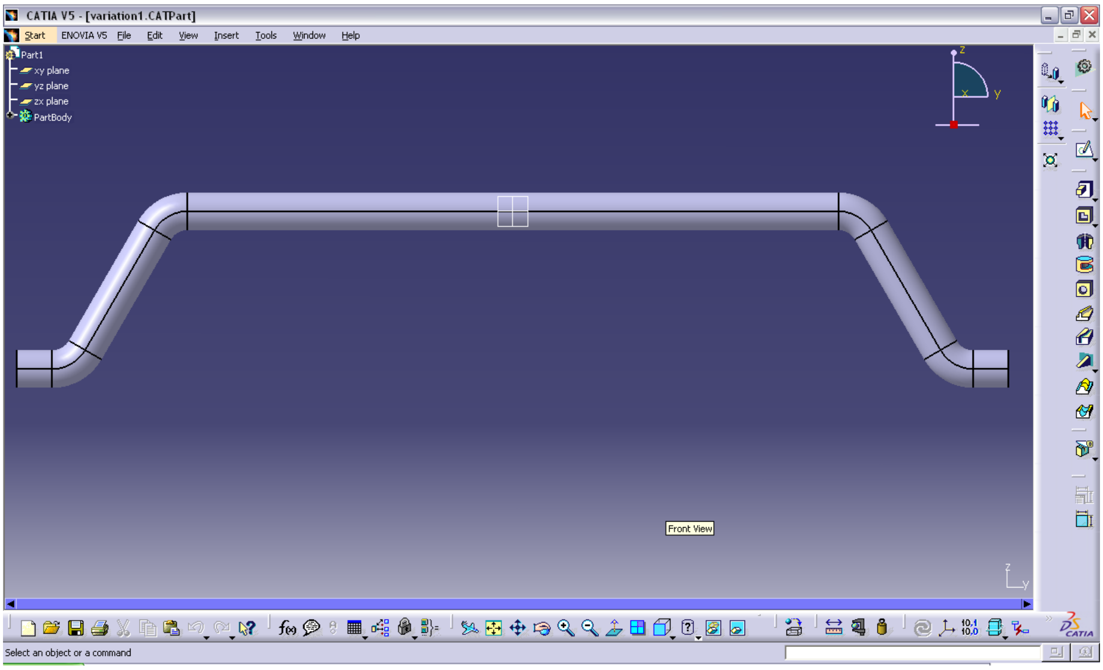


Figure 3.5c: Base design of an anti roll bar

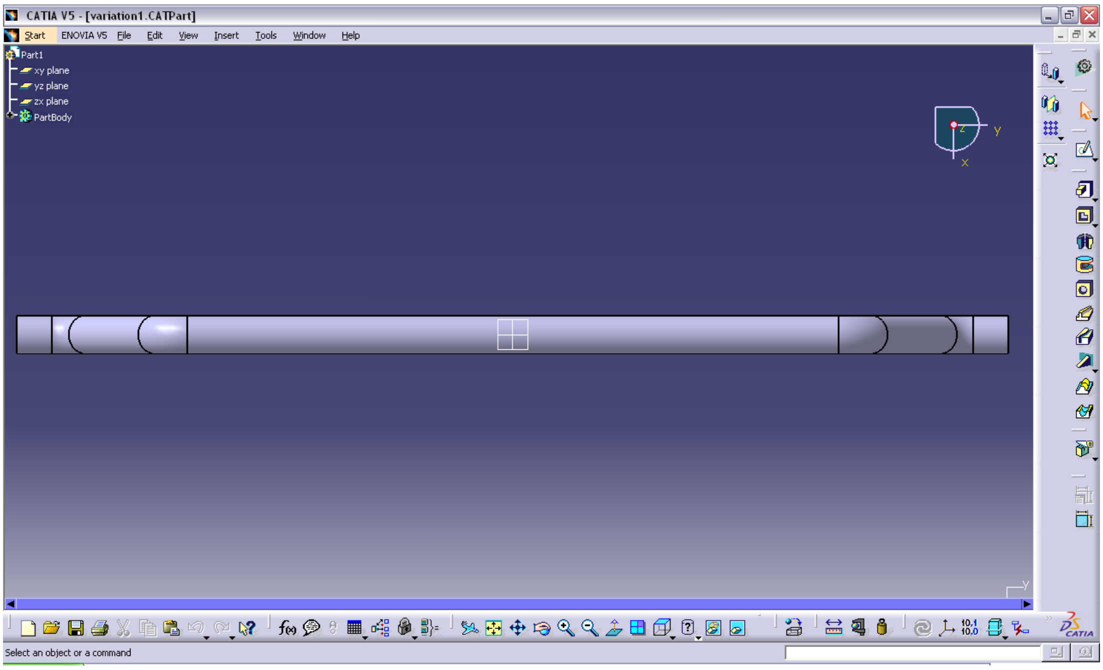


Figure 3.5d: Base design of an anti roll bar

3.2.3 Boundary condition for anti roll bar analysis

In order to perform analysis and run simulation on the anti roll bar, boundary condition of the situation must be firstly determined. The situation replicated in this analysis is that of when the truck entered a corner where the truck is tilted off the ground as a result of rolling. This will be considered as the worst case scenario in the analysis. It is selected as such because the analysis is seeking for studying of what will happen to the anti roll bar when a vehicle is in such condition.

The anti roll bar is installed at the head of the vehicle. The action of cornering is initiated at this part when the vehicle's tires turn. In order to control the rolling effect on the vehicle, anti roll bar must be installed at the place where vehicle tilting will begin, that is at the front tires of a vehicle. Hence, in the analysis performed throughout the project, only the head part of the vehicle would be considered.

Figure 3.6: A free body diagram indicating the boundary condition

Figure 3.6 illustrates the boundary condition of the anti roll bar in the situation defined above. At one end, the anti roll bar is fixed supported to replicate the condition where it is attached to a tire which is in contact with the ground. It is not freely move in vertical or horizontal direction nor is able to slip in any direction. The point, known as Point A, which is fixed supported will be the pivot point when the vehicle rolls over and tilted off the ground at one of its tire. The other end of the anti roll bar, Point B, is where the moment resulted from the rolling action of the vehicle body took place. Point B is freely hanging to replicate the condition where the truck is tilted at one end.

3.2.4 Tilt angle

The tilt angle for the vehicle studied in this project is obtained from the research done by a group of researchers from The Ohio University and The National Highway

Traffic Safety Administration along with other researchers from various institute. [12] The angle is computed through vehicle tilt table test. In this test, a vehicle is placed on a hydraulic platform that will simulate the roll plane behavior of a vehicle in a steady turn. The test vehicle will slowly tilt to an angle on a table inclined in the roll direction.

Figure 3.7 shows the components involve in the tilted condition. In this state, one component gravity gSinΦ acts laterally while the other component gCosΦ acts perpendicular to the simulated road surface (the table surface). Assuming that gCosΦ simulates the gravity, then the simulated lateral acceleration is gTanΦ. Therefore, if the tilt table angle is slowly increased, the tangent of the tilt angle at which the vehicle rolls over can estimate the lateral acceleration at which the static roll stability of the vehicle is reached. The result of all the tests are tabulated in a database span over various type of vehicle, in which one of them is the type of vehicle focused in this study [12].

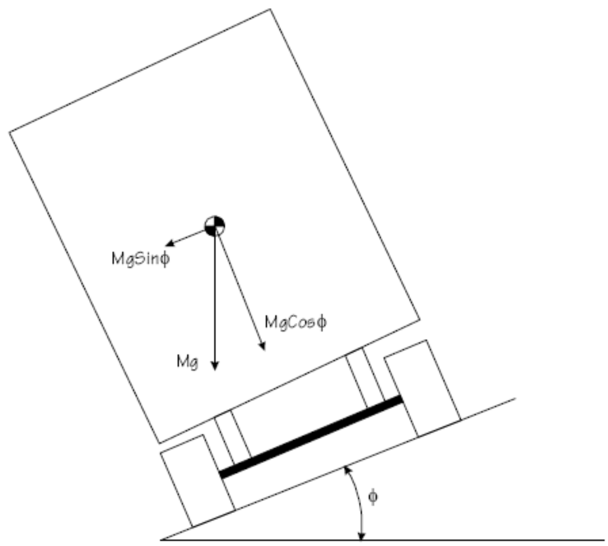


Figure 3.7: Free body diagram of a vehicle during tilt table test

3.2.5 Shifted location of vehicle's centre of gravity

The centre of gravity of the vehicle's head is located as followed. When the vehicle is tilted at 25^o as the result of body rolling, the location of the CG will shifted to a new location as indicated in Figure 4.5. Using sketch in CATIA, the new location of the CG can be determined easily.

Figure 3.8 shows the sketch of vehicle head from the front perspective both during normal condition and tilted condition. The sketch is drawn in CATIA for graphical representation and analysis of the mentioned condition.

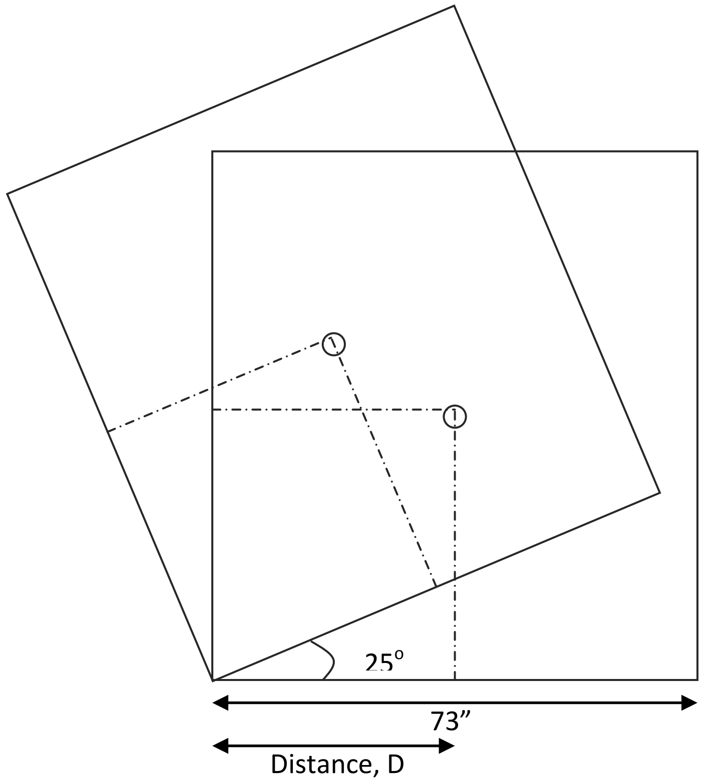


Figure 3.8: Simple diagram showing the tilted condition of the truck

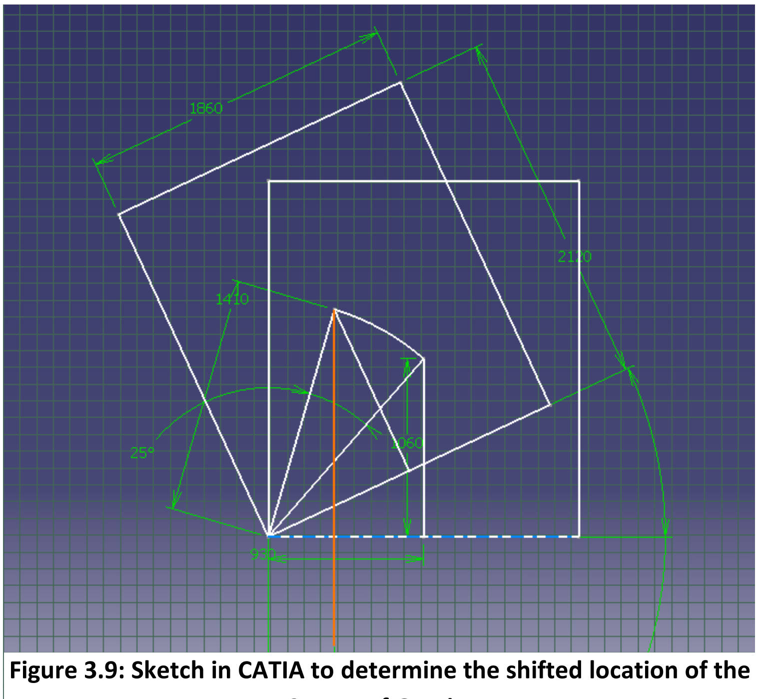


Figure 3.9: Sketch in CATIA to determine the shifted location of the Centre of Gravity

Notice the orange line in Figure 3.9 which is drawn vertically. The line is used to estimate the horizontal distance from the pivot point to the shifted CG as indicated by the orange line. Using constraint features in CATIA, the distance is determined to be **395 mm**. The new CG location and its distance from the pivot point will be used to calculate the force and the moment involved in this situation.

3.2.6 Force and moment calculation

In order to calculate the moment acting at the end of the anti roll bar, as indicated in the Figure 3.4, we need to define the type and the location of the force/ moment acting on the anti roll bar. Figure 3.8 shows the free body diagram of the anti roll bar, with the length of 49 inch, when tilted at angle 25^o. Point A in the diagram is the only point in contact with the ground. There is a normal force, F_n acting in vertical direction in reaction to the weight of the truck's head. The magnitude of Fn is equal to the weight exerted by the truck mass. Hence,

$$F_n = W_T \tag{3.2.6.1}$$

The shifted CG plays some role in exerting force on the anti roll bar. The location where the force at CG, F_{CG}, must be firstly determined. As depicted in Figure 3.8, the width of the truck's head is 73 inches. The distance from the CG to the pivot point as measured in the CATIA as shown in Figure 3.9 is about 395mm.

Considering the vehicle's parameter as mentioned before; we have to take into account the width of the truck's tires which width is about 9 inches or 228.6 mm. The distance or the tolerance between the tire to the one end of the anti roll bar is assumed to be 3 inches or 76.2 mm based on measurement on the real truck suspension system.

The calculation for the horizontal distance, D between the shifted CG points to the one end of the anti roll bar which is closed to pivot point as shown as follows:

$$Distance, D = Distance\ (CG\ to\ Pivot\ Point\ A) - Tires'\,Width - Tolerance\ (ARB\ to\ tire)$$

$$Distance, D = 395 - 228.6 - 76.2 = 90.2\ mm$$

The force at the shifted CG is acting at the distance, D =90.2 mm. The force, F_{CG}, is equal to $W_T = m_T g$, where W_T is the weight of the truck and m_T is the mass of the truck. These are shown in Figure 3.11.

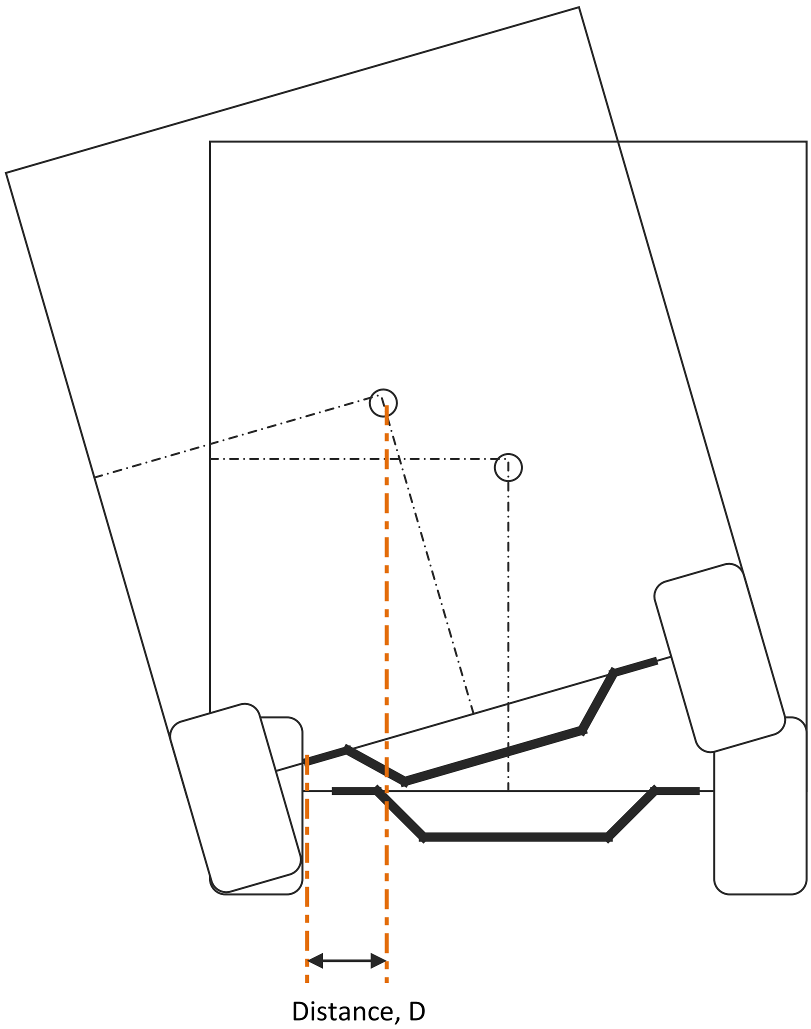


Figure 3.10: The free body diagram indicating the distance, D

Hence,

$$W_T = m_T g$$

$$W_T = (2853\ kg)\left(9.81\frac{m}{s}\right)$$

$$W_T = 27987.93\ N$$

Figure 3.11 shows the location of these forces, its components and the moments associated with each of them.

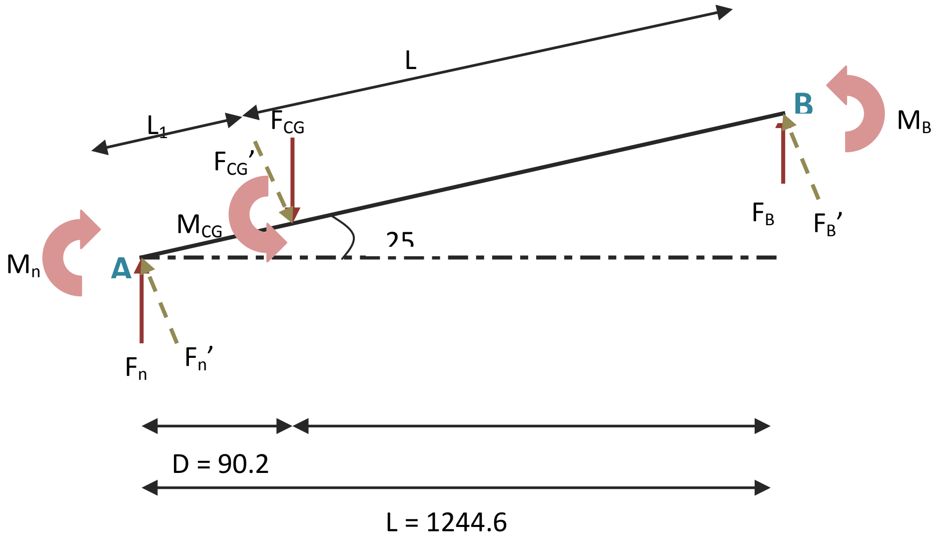


Figure 3.11: Free body diagram indicating all forces, force components and moments involved in the situation

Each point at the end of the anti roll bar is labeled A and B. F_n is the normal force resulted from the weight exerted by the mass of the truck. F_n' is the component of the normal force resulted by the 25^o tilt. M_n is the moment resulted from the F_n. F_{CG} is the force exerted at the centre of gravity of the truck which coincides with the anti roll bar. F_{CG}' is the component of the force resulted by the 25^o tilt. M_{CG} is the moment resulted from F_{CG}. F_B is the force resulted from the upward pulling effect from the truck at point B when tilted at 25^o. M_B is the moment resulted from F_B. L_1 is the length component of the horizontal distance between the centre of gravity and Point A.

First, the length L_1 must be calculated in order to determine the magnitude of the moment MCG.

$$\cos 25° = \frac{D}{L_1}$$

$$L_1 = \frac{D}{\cos 25°}$$

$$L_1 = \frac{90.2\ mm}{\cos 25°}$$

$$L_1 = 99.5\ mm$$

The correspondent L_2 could be determined simply by:

$$L_2 = L - L_1$$

$$L_2 = 1244.6\ mm - 99.5\ mm$$

$$L_2 = 1145.1\ mm = 1.1451\ m$$

Next, each force component must be determined to replicate the tilted position of the anti roll bar:

$$\cos 25° = \frac{F_n{}'}{F_n}$$

$$F_n{}' = F_n \cos 25°$$

$$F_n{}' = 27987.93 \cos 25°$$

$$F_n{}' = 25365.68\ N$$

$$\cos 25° = \frac{F_{CG}{}'}{F_{CG}}$$

$$F_{CG}{}' = F_{CG} \cos 25°$$

$$F_{CG}{}' = 27987.93 \cos 25°$$

$$F_{CG}{}' = 25365.68\ N$$

The calculation to determine the moment M_B is as follow, positive in counterclockwise:

$$\sum M_B = M_B + M_{CG} - M_n = 0$$

$$M_B = M_n - M_{CG}$$

$$M_B = F_n{}'(L) - F_{CG}{}'(L_2)$$

$$M_B = (25365.68)(1.2446) - 25365.68(1.1451)$$

$$M_B = 2523.89\ Nm$$

CHAPTER 4

RESULTS & DISCUSSION

4.1 Summary of methodology, calculation & result

Table 4.1 summarizes all important data obtained throughout this project.

Table 4. 1: Vehicle and material specification & summary of result

Vehicle Specification	
Model	Isuzu NPR HD Diesel
Vehicle type	Class 3 Heavy Vehicle
Category	Light Duty
GVWR*	12,000 lbs (5443 kg)
Payload Allowance**	2870 kg
Normal Occupants Mass	70 x 2 = 140 kg
Head Weight	5443 kg – 2870 kg = 2713 kg
Head Weight + Occupants***	2713 kg + 140 kg = 2853 kg
Overall Length	5.7 m
Body Length	4.3 m
Head Length	1.4 m
Head Height	2.1 m
Head Width	1.8 m
Material Specification	
Code Name	AISI/ SAE 4130
Type	Alloy Steel
Chemical Composition	Nickel, Chromium, Molybdenum, Steel
Density	7700 – 8030 kg/m^3
Poisson's Ratio	0.27 – 8.03
Elastic Modulus, λ	190 – 210 GPa
Tensile Ultimate Strength, σ_{UTS}	744.6 MPa
Tensile Yield Strength, σ_Y	472.3 MPa
Compressive Yield Strength	472.3 MPa
Calculation Result Summary	
ARB Overall Length	49 in. (1244.6 mm)
Bar Length (B)	36.5 in (927.1 mm)
Arm Length (A)	9.6 in (243.84 mm)
Connector Arm Length (F)	3 in (76.2 mm)
Diameter (D)	2 in (50.8 mm)
Bend Angle	30^o
Tilt Angle	25^o
Tilted CG Location	90.2 mm (from fixed pivot point B)
Load From Truck	27987.93 N
Moment B	2523.84 Nm (acting at point A)

** truck body weight + full load + occupants*

*** body weight + full load*

**** full load on the truck head section*

4.2 Result of simulations and the analysis performed

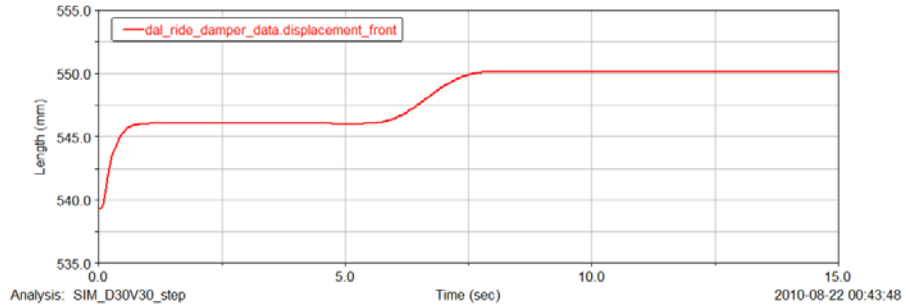
4.2.1 Step Steer Simulation (without anti roll bar)

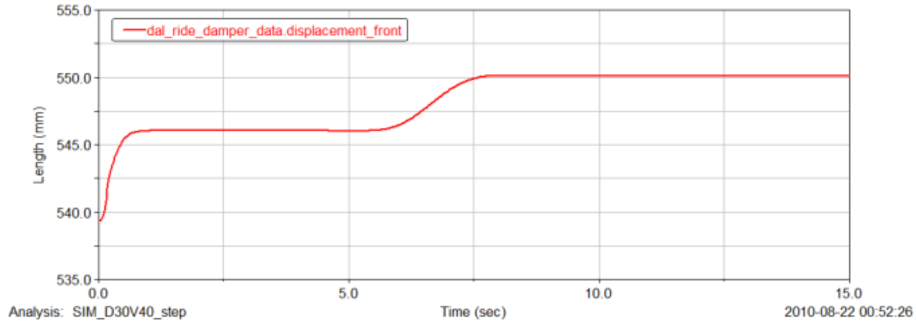
The observation from the step steer simulation is summarized in the following Table 4.2. Two obvious observations are used to indicate the condition of the vehicle during cornering , whether it succeed in making a steady cornering or fail and end up rolled over.

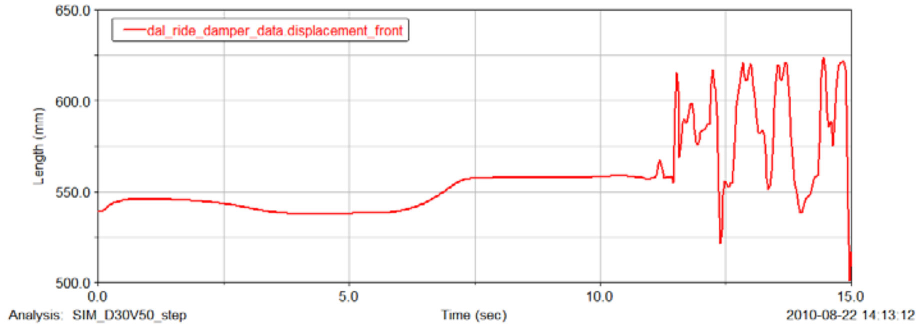
Table 4.2: Simulation observation result

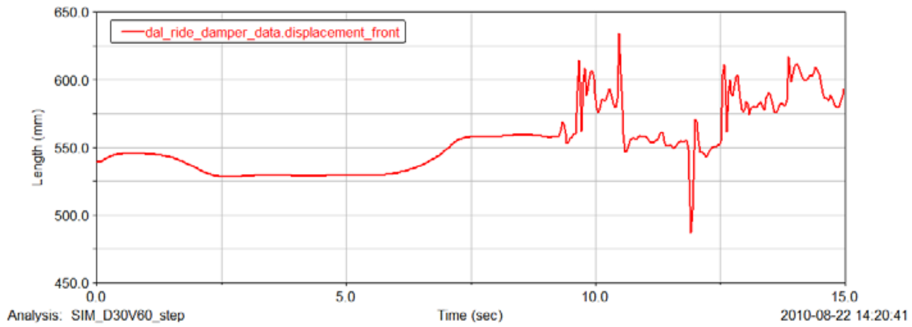
Road Cornering Degree (^{o})	Truck Velocity (kmph)	Observation
30	30	Steady Cornering
	40	Steady Cornering
	50	Roll Over
	60	Roll Over
40	30	Steady Cornering
	40	Steady Cornering
	50	Roll Over
	60	Roll Over
50	30	Steady Cornering
	40	Steady Cornering
	50	Roll Over
	60	Roll Over
60	30	Steady Cornering
	40	Steady Cornering
	50	Roll Over
	60	Roll Over
70	30	Steady Cornering
	40	Steady Cornering
	50	Roll Over
	60	Roll Over
80	30	Steady Cornering
	40	Roll Over
	50	Roll Over
	60	Roll Over
90	30	Steady Cornering
	40	Roll Over
	50	Roll Over
	60	Roll Over

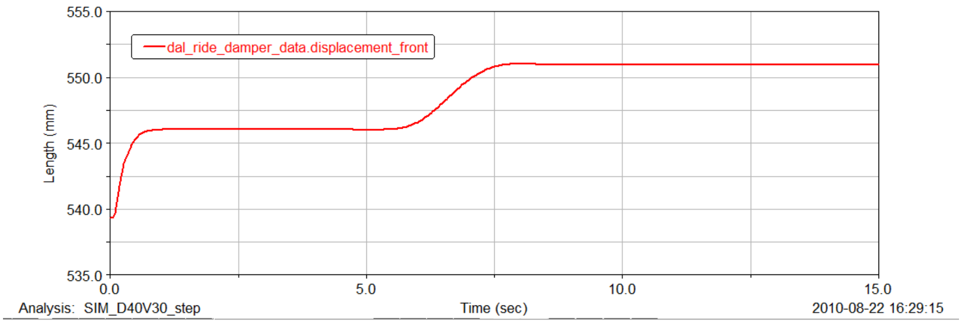
The graph of damper displacement against the simulation time is plotted for every simulation and the results are shown in the Figure 4.1. From Table 4.2, it is understood that the truck is most stable when entering corner of any degree at velocity of 30 kmph. The most critical velocity is 50 kmph at which the truck will start to roll over. At extreme degree of cornering such as at 80^o and 90^o, the truck starts to roll over at even lesser velocity i.e. 40 kmph.

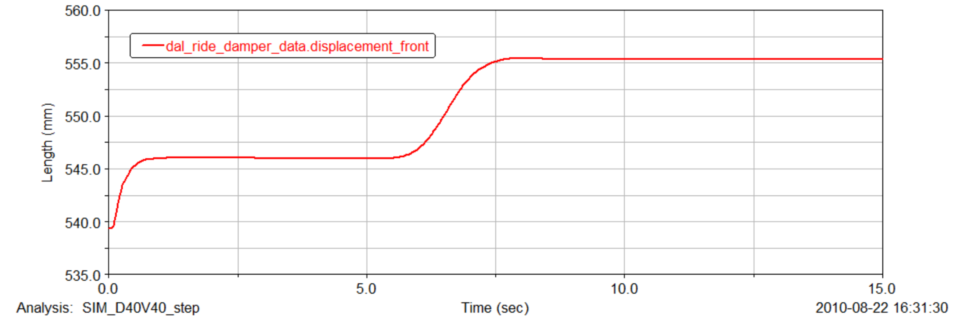


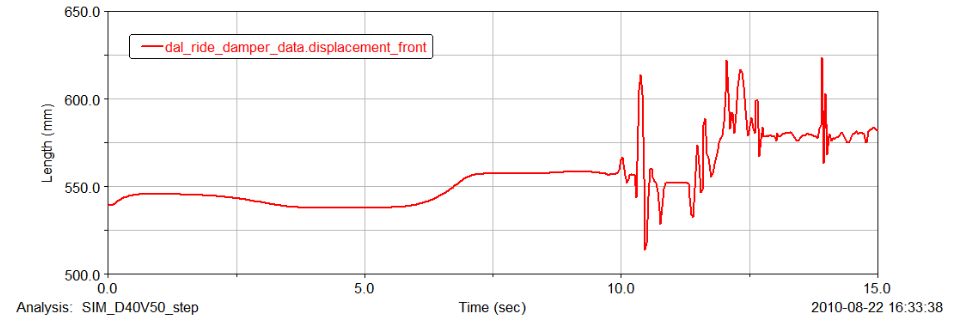


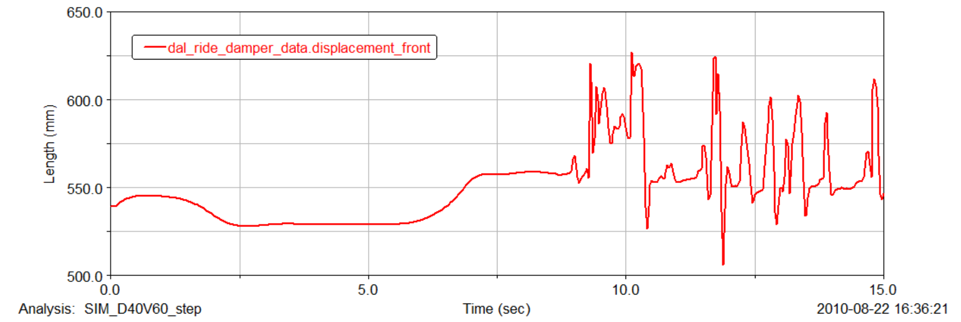





dal_ride_damper_data.displacement_front
Length (mm)
555.0
550.0
545.0
540.0
535.0
0.0
5.0
10.0
15.0
Time (sec)
Analysis: SIM_D40V30_step
2010-08-22 16:29:15


dal_ride_damper_data.displacement_front
Length (mm)
560.0
555.0
550.0
545.0
540.0
535.0
0.0
5.0
10.0
15.0
Time (sec)
Analysis: SIM_D40V40_step
2010-08-22 16:31:30


dal_ride_damper_data.displacement_front
Length (mm)
650.0
600.0
550.0
500.0
0.0
5.0
10.0
15.0
Time (sec)
Analysis: SIM_D40V50_step
2010-08-22 16:33:38


dal_ride_damper_data.displacement_front
Length (mm)
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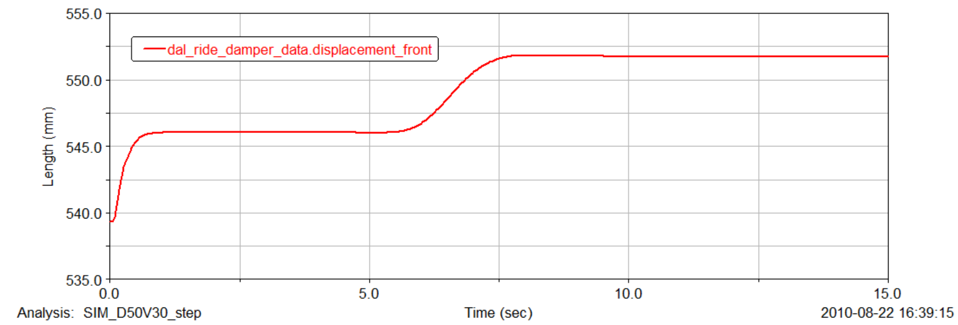
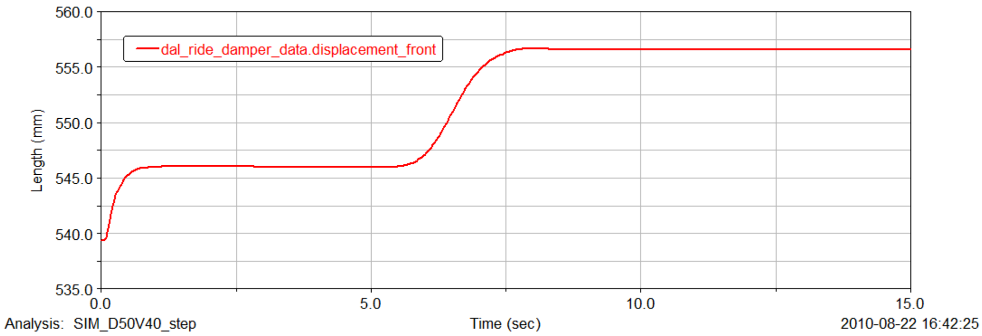
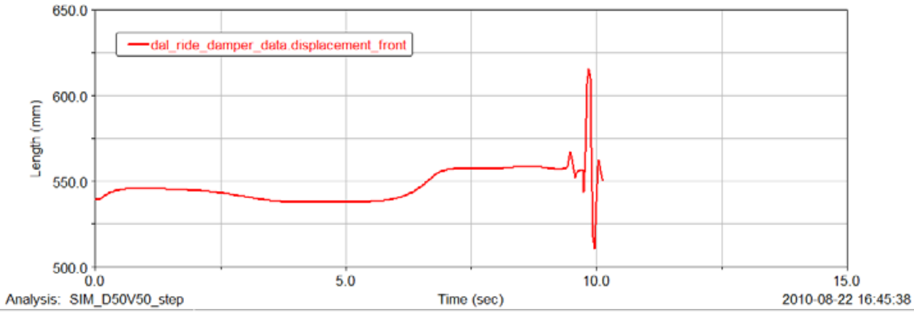
600.0
550.0
500.0
0.0
5.0
10.0
15.0
Time (sec)
Analysis: SIM_D40V60_step
2010-08-22 16:36:21

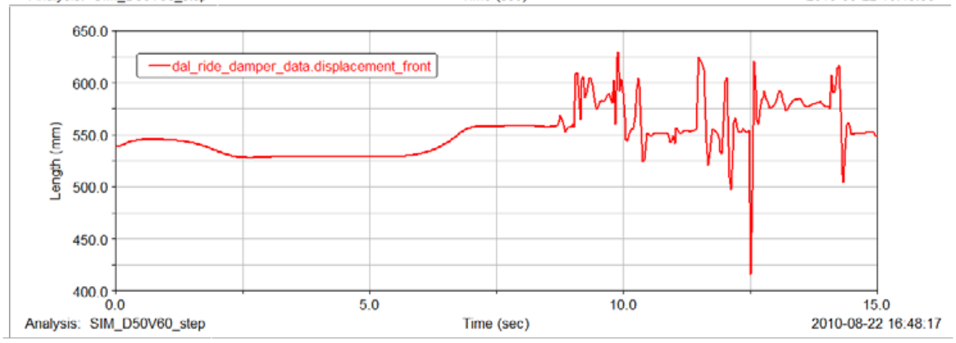

dal_ride_damper_data.displacement_front
Length (mm)
555.0
550.0
545.0
540.0
535.0
0.0
5.0
10.0
15.0
Time (sec)
Analysis: SIM_D50V30_step
2010-08-22 16:39:15



560.0
555.0
550.0
545.0
540.0
535.0
Length (mm)
dal_ride_damper_data.displacement_front
0.0
5.0
10.0
15.0
Analysis: SIM_D50V40_step
Time (sec)
2010-08-22 16:42:25



650.0
600.0
550.0
500.0
Length (mm)
dal_ride_damper_data.displacement_front
0.0
5.0
10.0
15.0
Analysis: SIM_D50V50_step
Time (sec)
2010-08-22 16:45:38



650.0
600.0
550.0
500.0
450.0
400.0
Length (mm)
dal_ride_damper_data.displacement_front
0.0
5.0
10.0
15.0
Analysis: SIM_D50V60_step
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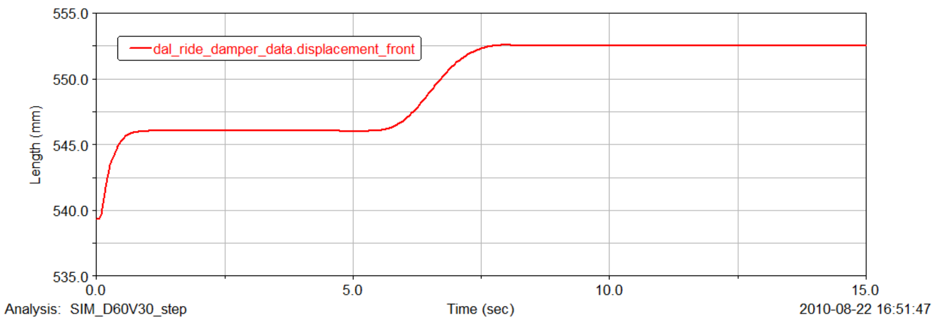

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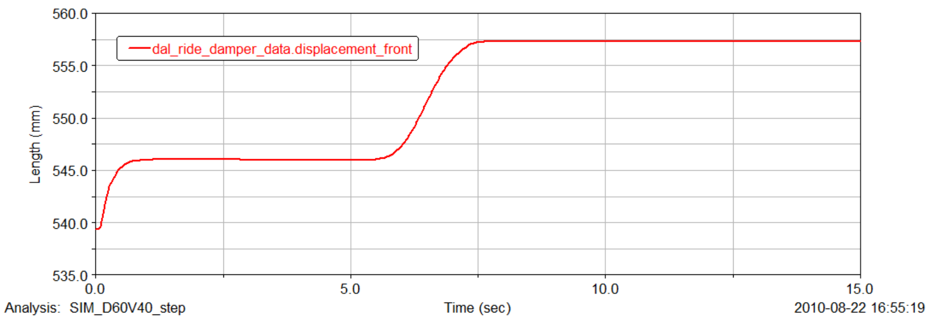

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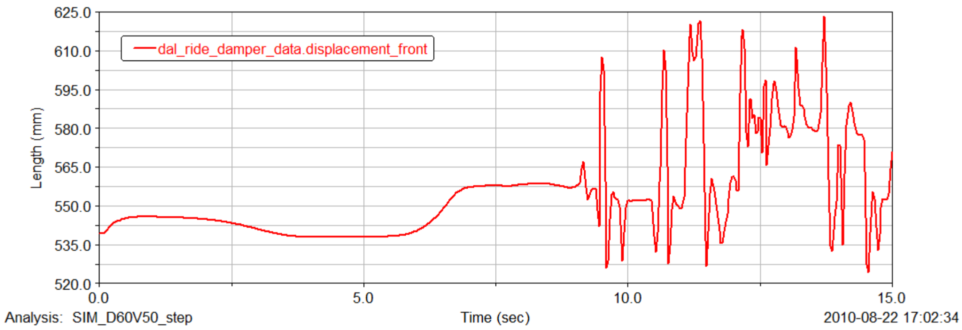
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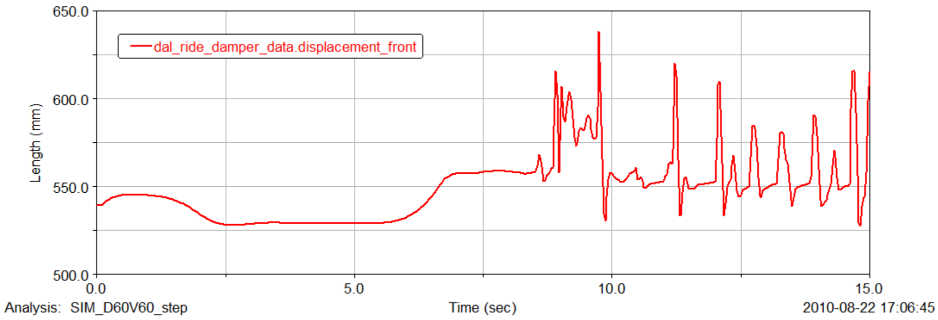
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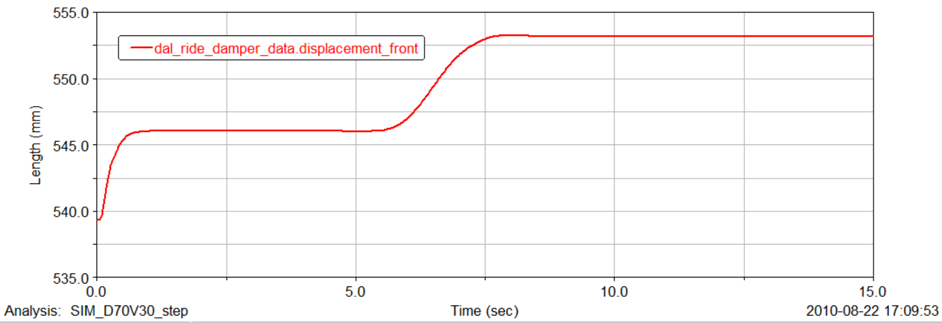
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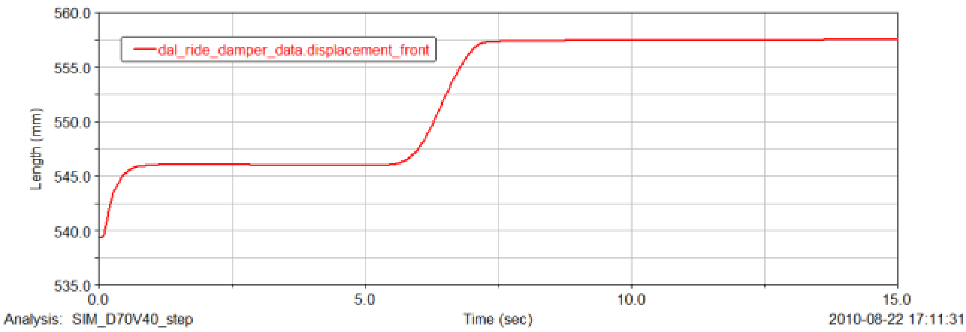
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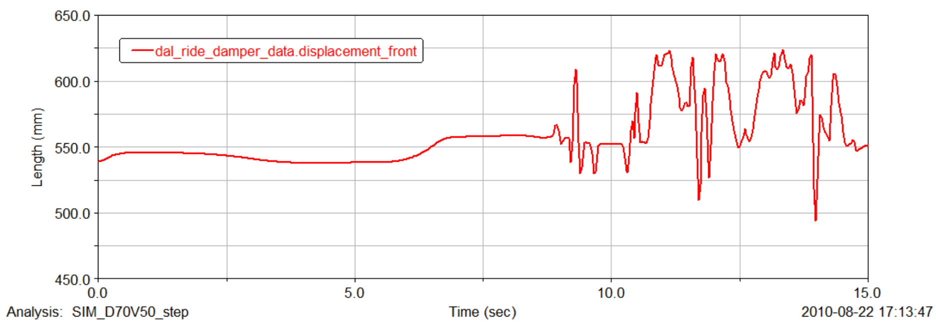
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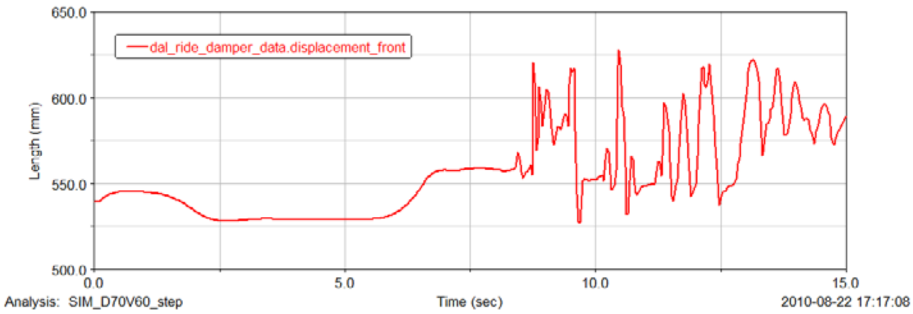
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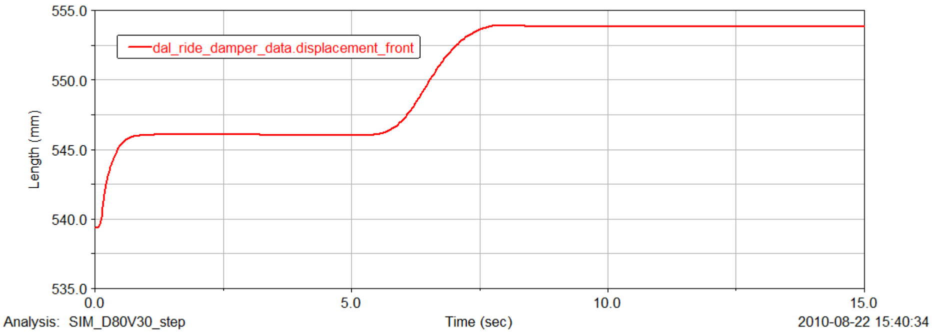
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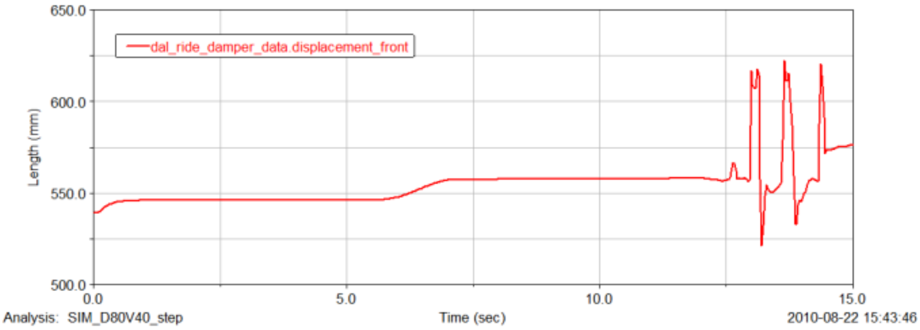
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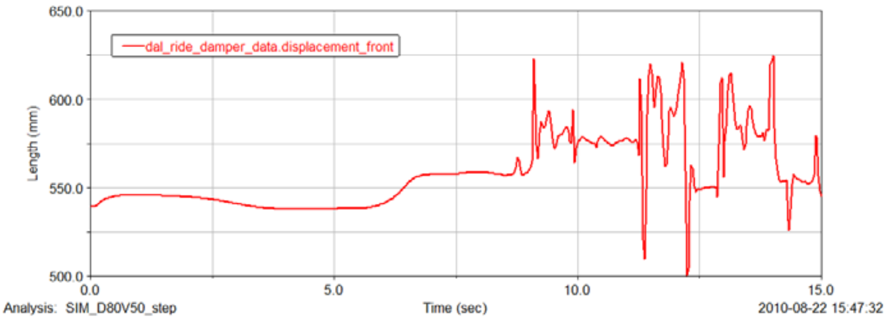
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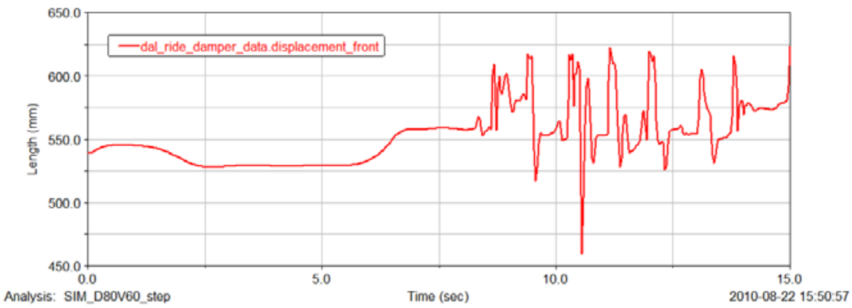
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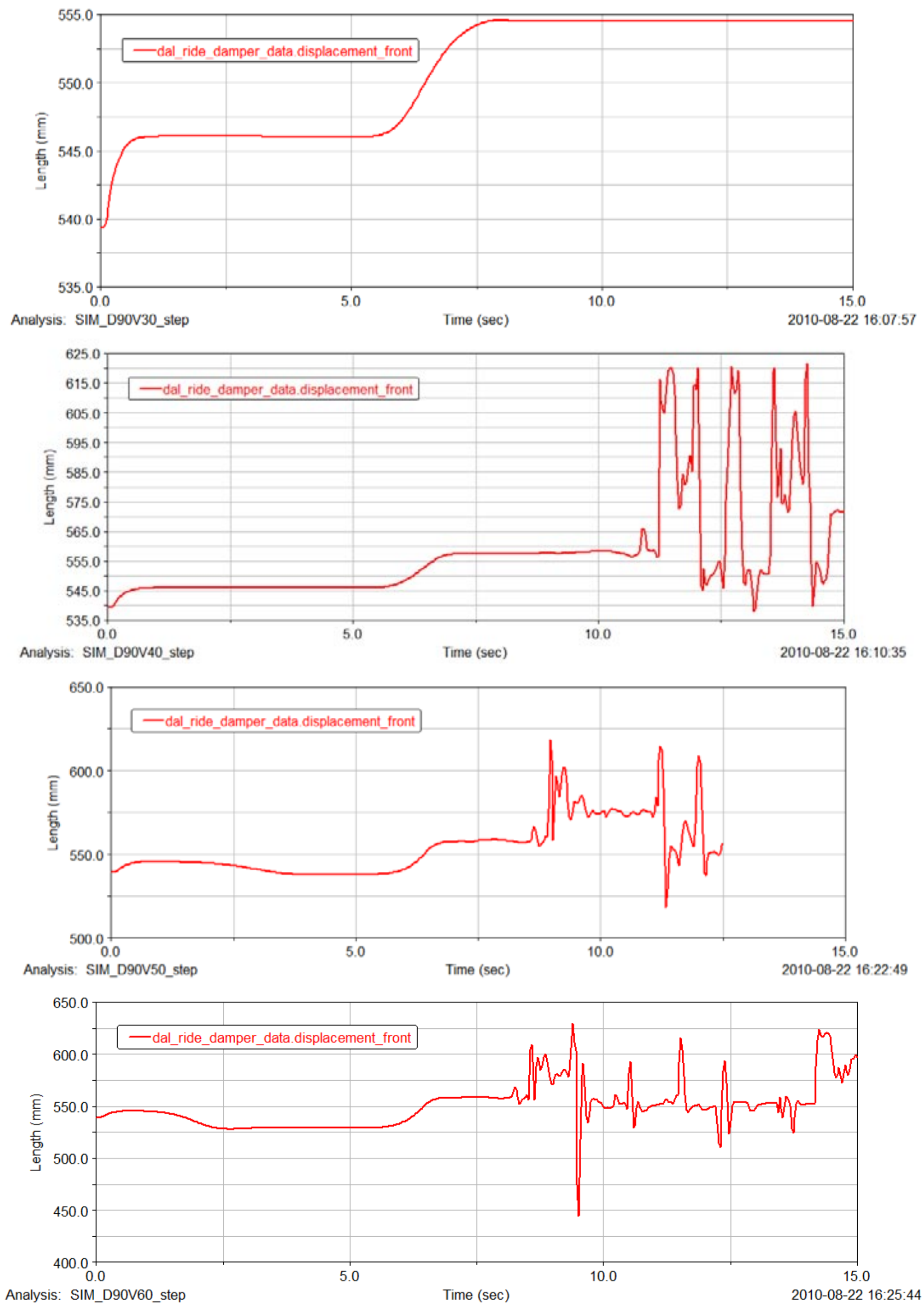

Figure 4. 1 Set of graph from simulations

Figure 4.1 above shows the result of the truck left damper displacement which directly connected to the truck's tires. The displacement shows the vertical movement of the truck's tires during the event of cornering. The left graph shows smooth vertical displacement which implies steady cornering. The right cornering depicts abrupt change and inconsistent change in the damper displacement which implies the failure during the cornering. The imbalance lateral force that act from the left tire to the right tire (from inside of the corner towards the outside) of the truck causes the vehicle to lose its grip on the road and roll over.

Any vehicle entering a corner will experience changes of lateral force either from right to the left or left to the right depending on the cornering direction. However, lower centre of gravity location will reduce the shifting of this point towards the cornering direction. Although the center of gravity of the vehicle is still shifted, but the difference is small and insignificant to cause the inside tire of the vehicle to be lifted off the ground.

Different scenario, however, occurs when studying a heavier vehicle. The location of its center of gravity is higher as explained previously. When entering a corner, the center of gravity of such vehicle shifted more significantly and contributes to loss of the vehicle balance. This situation is magnified when we consider the weight of the vehicle itself. Larger value of weight will exert larger lateral force. The weight value coupled with the high location of center of gravity will result in large pulling force outward of the corner which in turn cause instability and roll over.

The following figure shows the plotted graph of every simulation as summarized in the Table 4.2. D#V# stands for the value of the degree and the velocity used during a particular simulation. For example, D30V30 means 30^o of cornering at 30 kmph velocity.

Figures 4.2 show vehicle condition during the event of roll over. The Ferrari represents what will happen to the truck when roll over happens. Figure 4.12a shows the vehicle during entering the corners, Figure 4.12b shows the vehicle starts to roll over and Figure 4.11c shows the vehicle after rolled over. These figures were taken from the simulation if which the vehicle is travelling at 50 kmph when it enters a corner of 60^o.

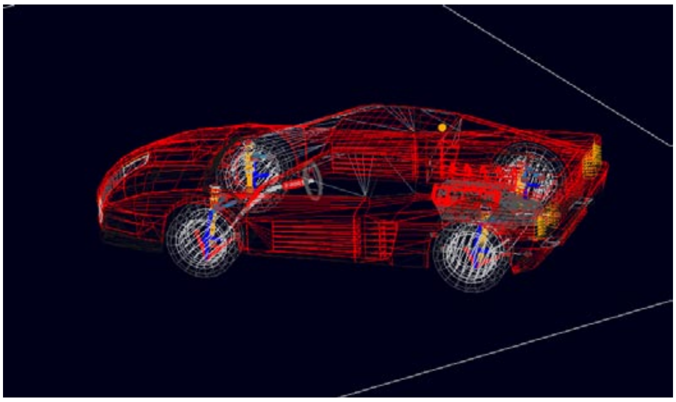

Figure 4.2a: Vehicle entering the corner

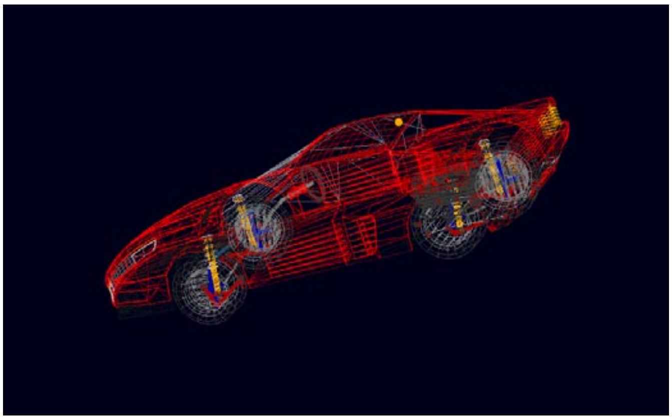

Figure 4.2b: Vehicle starts to roll over

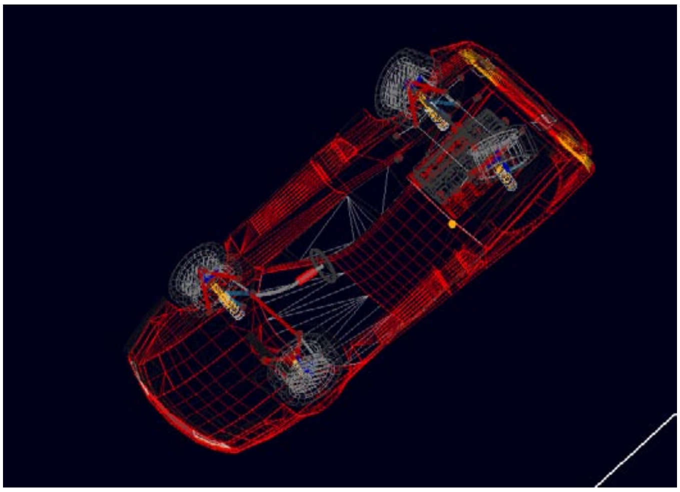

Figure 4.2c: Vehicle after rolled over

It is understood that the most critical velocity at which the truck will start to roll over is at 50 kmph. This value will be used as a boundary condition at which the roll stiffness of the anti roll bar will be tested.

Two categories of cornering are defined as dangerous cornering and extreme cornering condition. Dangerous cornering is when a vehicle tries to make 60^o turn and extreme cornering is when a vehicle makes 90^o turn. Between these two values, a truck can only possible to move at 50 kmph at the most 70^o turn. This has been proven during the simulation previously carried out using ADAMS Car 2007. Hence, the boundary conditions for the coming anti roll bar analysis are velocity at 50 kmph and road cornering degree of 70^o. Several values of roll stiffness will be tested to determine the most suitable stiffness for a truck.

4.2.2 Step Steer Simulation (with anti roll bar)

The value of spring stiffness of the anti roll bar installed to our vehicle is varied to determine which value is suitable to resist the rolling motion of the vehicle when it enters a corner. Table 4.3 summarizes the values of stiffness chosen for this test. Noted that as mentioned earlier, that the boundary condition is for vehicle speed at 50 kmph and cornering degree at 70^o based on the weight of the vehicle itself which give the most influence.

Table 4.3: Simulation result with stiffness varied

ARB Spring Stiffness (N/mm)	Observation
10,000	Roll Over
11,000	Roll Over
12,000	Roll Over
13,000	Steady Cornering
14,000	Steady Cornering
15,000	Steady Cornering
16,000	Steady Cornering
17,000	Steady Cornering
18,000	Steady Cornering
19,000	Steady Cornering
20,000	Steady Cornering

The value of ARB spring stiffness is obtained from the equation (16) or (17). Notice that the geometry factors are involved in calculating this value. Hence, by having a fixed length due to the width of the vehicle, and fixed diameter for example, one can vary any other geometry of the ARB design.

The vehicle started to gain balance and steadily enter the corner when we fixed the value of the bar stiffness at 13,000 N/mm. The next values chosen also caused the car to regain balance and enter the corner steadily.

In this case, because there are many values of bar stiffness that we can select, we should choose the lowest stiffness possible. This is to achieve the balance between ride comforts and roll stiffness. High stiffness value will cause the vehicle to lose it ride comforts, in which even smallest bump on the ride will have vibrating effect transferred to the components of the car. If this situation is prolonged, the vehicle will face the risk of failure due to fatigue and vibrations.

Hence, we choose to use the bar stiffness of 13,000 N/mm as the reference stiffness value in determining the dimension of our base design for the ARB. In the next step, we are going to get several options for the dimensions based on the previous equations using the stiffness chosen. After that, we are going to design the ARB.

4.2.3 Anti roll bar geometry

The anti roll bar spring stiffness can be calculated using the following equation:

$$R = \frac{\pi G d^4}{16 L b^2} \tag{16}$$

In which R is the ARB spring stiffness, G is the modulus of the material, d is the bar outside diameter, L is the length of the bar and b is the length of the arms of the bar. This equation can also be expressed as:

$$R = \frac{\pi G d^4}{(0.4244 \times A^2) \times B \times (0.2264 \times C^2)} \tag{17}$$

From either of the two equation above, we can obtain the geometry parameters for the newly design anti roll bar.

4.2.4 Force, moment and stress analysis on the anti roll bar

All parameters specified in the previous calculations are used in the simulation for stress analysis and deformation analysis. The simulation is performed using ANSYS Workbench Version 11.0. Parameters used are inclusive of the boundary condition which specifies the fixed support at the pivot point A and the moment M_B which acting positively clockwise at Point B. Point B is not supported and hanging freely to replicate the condition where one of the truck tires is lifted off the ground. There are also types of analysis that would be performed, which are the equivalent stress analysis (Von Misses) and the total deformation analysis.

Figure 4.3 shows the anti roll bar device after being uploaded into ANSYS for simulation preparation. Figure 4.4 shows the meshing performed on the anti roll bar device. Figure 4.5 shows the boundary condition being defined in the software in which the fixed support point and the moment acting at point B are located.

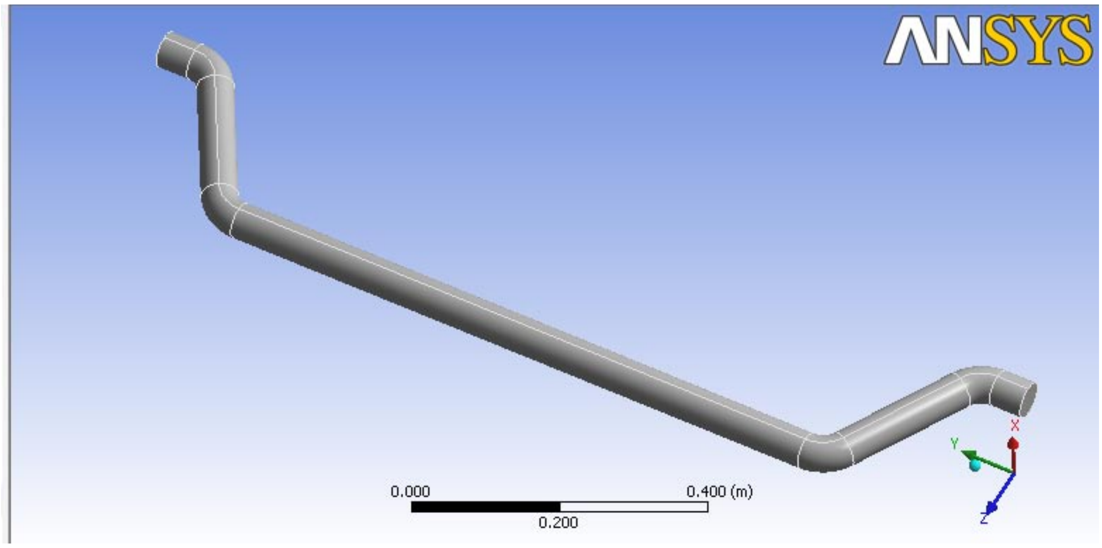

Figure 4.3: The anti roll bar model being uploaded into ANSYS Workbench V11

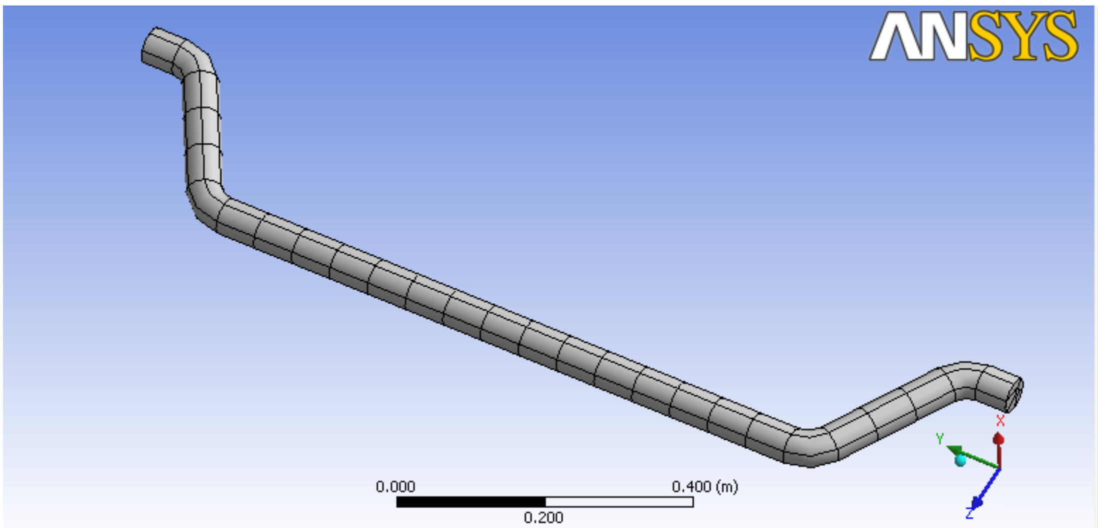

Figure 4.4: The anti roll bar model being meshed in the ANSYS Workbench V11

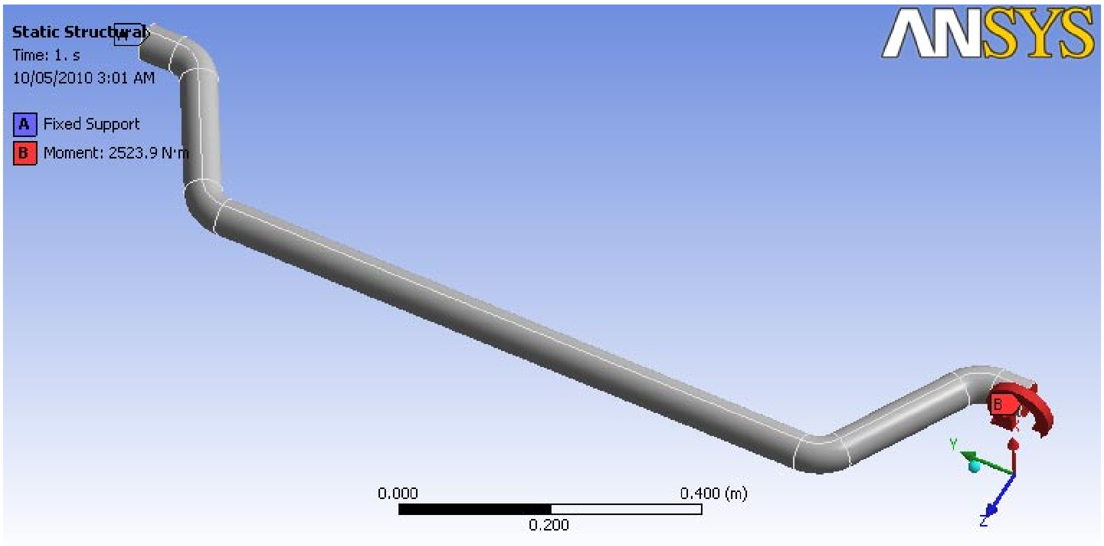

Figure 4.5: The boundary condition and moment involved are being defined

Figure 4.6 shows the stress analysis result on the anti roll bar device. Notice that the highest stress is concentrated at the bending area as indicated by the red marker. Figure 4.7 shows the anti roll bar before deformation analysis is performed. Figure 4.8 shows the condition of the anti roll bar after the analysis is carried out. The highest deformation occurs at Point A in which the anti roll bar is deformed about 68 cm from the initial condition. The red marker indicates the highest deformation occurrence and the blue marker indicates the lowest deformation.

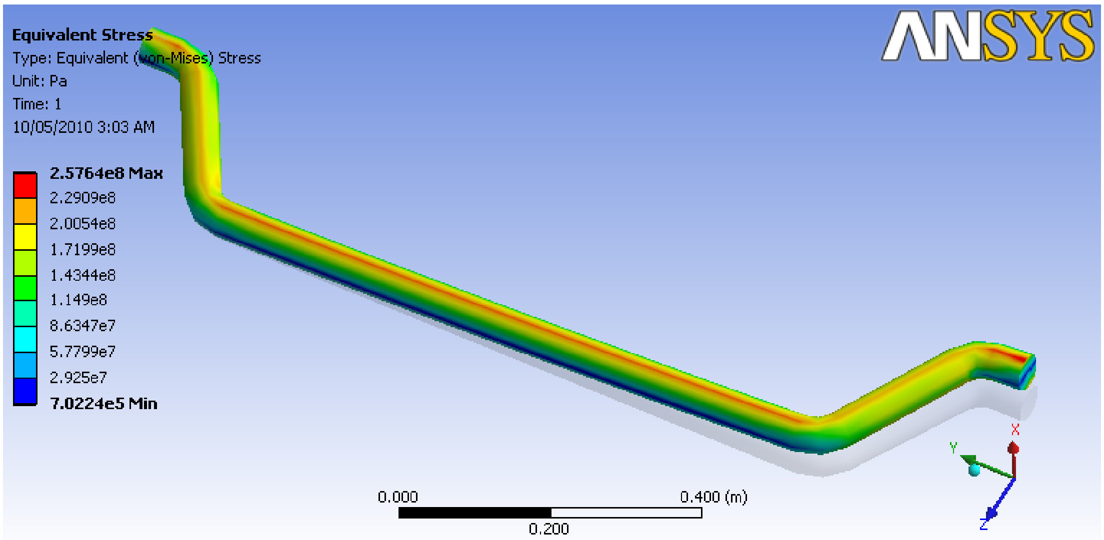

Figure 4.6: The result of the stress analysis on the anti roll bar model

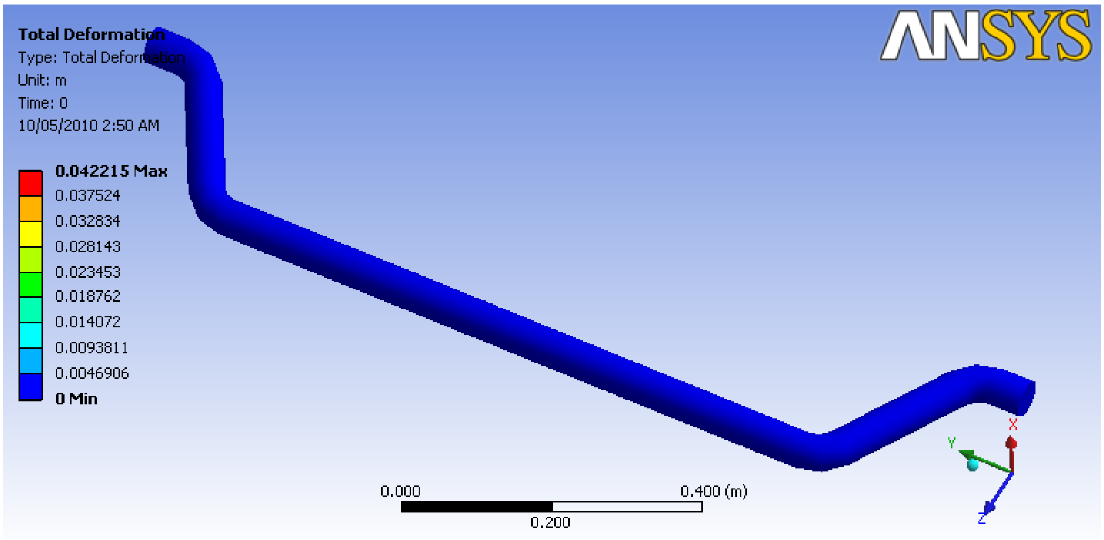

Figure 4.7: The anti roll bar model before the deformation analysis

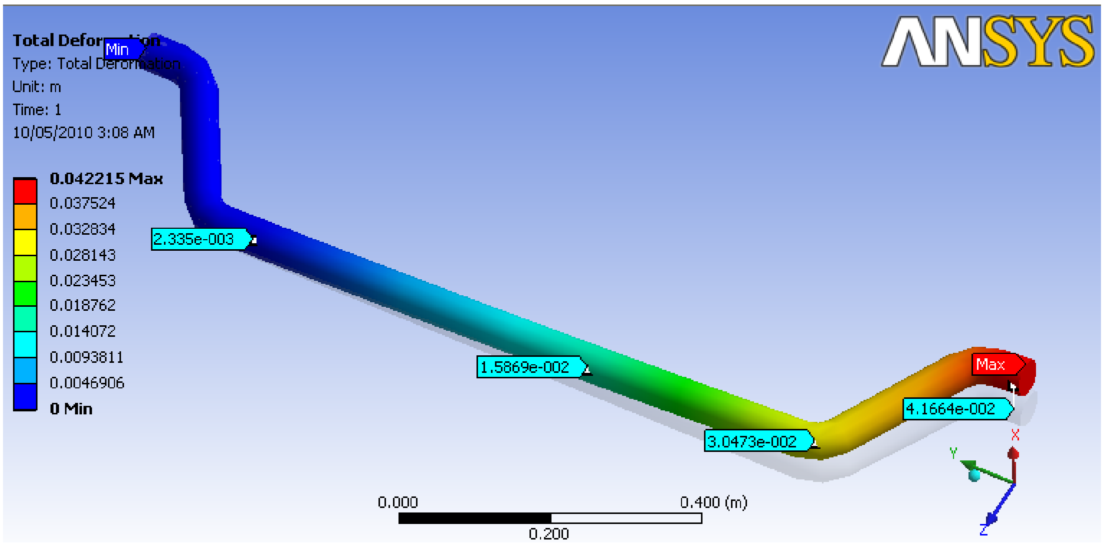

Figure 4.8: The result of the deformation analysis on the anti roll bar model

CHAPTER 5
CONCLUSION & RECOMMENDATIONS

5.1 CONCLUSION

It has been proven that anti roll bar device is capable to prevent vehicle roll over and has been used widely especially in performance car. The simple concept of having a curved rod installed makes it ideal solution and cheaper alternative to fight against vehicle roll over.

Using the same concept taken from that of a passenger car, we managed to prove that this device is also suitable to be used on heavier vehicle. Depending on the type of vehicle, different vehicle might require different design of an anti roll bar. However, all those design is govern by a stiffness value that must be first determined from simulation and test using the vehicle. A suitable stiffness value is a value that is lowest enough to be able to prevent vehicle roll over given a worst case scenario.

Through simulation performed, we have proved that the anti roll bar can prevent roll over among heavier vehicle and influence the handling of these vehicles. The stress, force and moment analysis carried out also prove that the design and the material can sustain the force possibly exerted by the vehicle onto the device.

5.2 RECOMMENDATIONS

The situation studied in this project is during a cornering event. In future, it is recommended to explore this project in situation where a vehicle is driven throughout a slope. It is also recommended to explore different design possible using the same stiffness value and observe how different designs of anti roll can affect heavy vehicle handling. Also, different material could also influent the design since the material plays a vital part in determining the anti roll bar stiffness. One important aspect in this project is the simulation software used in this project. Since we are using ADAMS Car 2007, there is no available template for a truck and we have to work using the only available template – a passenger car, which had been modified to replicate our vehicle. It is recommended to use latest software such as ADAMS

Car 2010 because it already has a template for a truck, and we can use the template to get more accurate result.

CHAPTER 6
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