# UNIVERSITY OF NAIROBI

**SCHOOL OF ENGINEERING**

**DEPARTMENT OF MECHANICAL AND MANUFACTURING ENGINEERING**

**FINAL YEAR PROJECT REPORT**

**[FME 561 &562]**

**PROJECT TITLE:**

**DESIGN OF A STEERING SYSTEM FOR A FORMULA ONE STUDENT RACE CAR**

**PROJECT CODE: JMO 03/2016**

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**PROJECT SUBMITTED IN PARTIAL FULFILLMENT OF THE AWARD OF THE DEGREE OF BACHELOR OF SCIENCE IN MECHANICAL AND MANUFACTURING ENGINEERING**

**MAY 2016**

**DECLARATION**
We, the undersigned, declare that this project is of our original work and has not been submitted for a degree award in any other institution of higher learning or published anywhere else.

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This project has been submitted for examination with the approval of our project supervisor.

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**DATE………………………………………………..**

**DEDICATION**

To our families, friends and the Academic Staff at the Department of Mechanical Engineering for their guidance and support from the start to the end of our undergraduate studies at the University of Nairobi.

**ACKNOLEDGEMENTS**
In the build up to making this project, we would like to show much appreciation to our supervisor Prof. Julius M. Ogola for the advice, direction, wisdom and trouble-shooting that he did to aid in the completion of this project. We are very much indebted to you.

We also would like to thank our families and friends who gave us support and encouragement on our studies in the university and thus to the eventful culmination of our undergraduate studies.

We also thank our fellow student Salman Bharadia for the aid he gave us in the making of the CAD drawings.

We would also like to thank God for the much he has given us to be able to complete this project.

LIST OF SYMBOLS USED

| | |
|---|---|
| SAE | Society of Automotive Engineers |
| CAD | Computer Aided Design |
| FEA | Finite Element Analysis |
| $D_p$ | Pinion diameter |
| $F_r$ | Friction force |
| mg | Weight |
| $F_L$ | Lateral force applied from the steering wheel |
| N | Normal reaction |
| μ | Friction coefficient |
| $r_{pinion}$ | Pinion radius |
| $R_{steering}$ | Wheel steering wheel radius |
| $\tau_{max}$ | Shear stresses |
| T | Torque in the steering column |
| J | Inertia for hollow columns |
| $N_{cr}$ | Critical force |
| E | Modulus of elasticity |
| I | Area moment of inertia |
| x | Reduction factor |
| A | Area |
| $f_y$ | Yield strength of the mild steel |
| $\gamma_{M1}$ | Partial factor |

| | |
|---|---|
| $i$ | Radius of gyration |
| $f_y$ | Yield strength of the mild steel |
| $\alpha$ | Imperfection parameter |
| $W_f$ | Load on front axle |
| $W$ | Total weight of car |

ABSTRACT

The aim of the project was to design a steering system for a Formula One Student Race Car to reduce the turning radius. Every year, students from various universities around the world come together to compete in the Formula SAE Race Car competition and face various challenges to emerge as the best. This project which was started last year at The University of Nairobi is also aimed at proving that students at this institution are capable of taking part in competitions at the international arena.

Research was greatly conducted through various Automotive books and journals concerned with the steering systems that guided us through the working principles of a Formula One Car. The steering system is an essential part of the race car since manoeuvrability and control is crucial in fast cars. Thorough consideration was put on what kind of steering system will work best on the student race car and the result was the manual rack and pinion steering system. The rack and pinion steering system is the simplest and most widely used, it is cheap and easy to maintain. The steering system can be found on the Honda civic which is famous in the Student Race cars.

The conceptual design was first put to life using Computer Aided Design (CAD) and visualized for a better understanding of how the system works. The components used in the design were engineered not to fail and under specifications. A computer simulation (ANSYS) was used to make sure the components don't fail under stresses.

Reducing the turning radius of the Race Car is achieved by changing the position of the centre of gravity to the rear and increasing the tire angle. Changing the position of the centre of gravity promotes over-steer which in turn reduces the turning radius. Over-steer can be used as an advantage at high speeds without having to reduce speeds at corners. Over-steer has to be controlled to prevent spinning. Increasing the inner wheel angle was also a vital modification in reducing the turning radius. Despite this, increasing the inner wheel angle caused an increase in the space required for turning.

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# CHAPTER ONE

## INTRODUCTION

## 1.1 Background

Currently, the most advanced cars on the planet are the formula one cars and thus the steering are as much advanced as the car is. Modern steering wheels have a lot of functions with a lot of buttons fitted on them. Over the years there has been a lot of changes. The current steering wheel cost about Ksh.5million and with carbon fibre construction which makes it weigh at about 1.3kgs.

The very first vehicles hardly had any steering wheel and were steered by some long poles called tillers. With advancements in technology, steering wheels were invented. By this time formula one cars were invented, the steering wheel had the following characteristics:

- The steering was from normal road cars and it was made of wood. Due to this, drivers mainly wore gloves during driving to avoid getting wooden splinters in their hands.
- The steering wheels had a large diameter to reduce the effort in turning. Power steering had not been invented yet and thus it was very difficult to make the car turn.

Since then, the vehicles have become so advanced to the point that virtually every part of the car can be monitored on the steering wheel.The incredible design in this form of car has motivated mechanical students all around the world to take part in an international competition organized by SAE. Nowadays, Formula Student is the most prestigious competition of its kind. This competition seeks to bring students from 120 universities round the world to design, manufacture, develop and compete with a car, like a small formula one team. The aim of this challenge is to help young talented engineers to develop in the professional world.

When we observe a car making a turn, in order to do the turn comfortably, the wheel that is in the exterior side of the turn covers a larger distance. On the contrary the rear wheels in any time of the turn are parallels. This causes a problem where the outer wheels cover a larger distance than the inner wheels. This will cause skidding of the car. Skidding is reduced in the rear wheels with the presence of a differential. Although there is a differential, the skidding effect will still be felt if the car made a turn at very high speed.

This effect would have been solved if the formula one car had four wheel drive which would also add to the car grip, but the four wheel drive is discouraged due to the weight it adds up to the car. This weight reduces its chances of winning races.in order to counter-act this problem, we use the Ackerman principle.

This principal involved the geometrical solution to this, for which all wheels were to have their axles arranged as radii of a circle with a common centre point. As the rear wheels are fixed, this centre point must be on a line extended from the rear axle. Intersecting the axes of the front wheels on this line as well requires that the inside front wheel is turned, when steering, through a greater angle than the outside wheel.

## 1.2 Problem Statement

Over the years formula one cars have developed far beyond what was expected initially and cars have been able to reach very top speeds. At these top speeds, the most important component is that the vehicle can be able to handle and manoeuvre appropriately to avoid any accidents. Most accidents occur around a corner whereby the car is most unstable. So the car can easily skid when moving round a corner. Despite this, cars usually have to move slower around corners so as to be able to maintain balance and manoeuvre the corner with ease. For this design of the student race car, we had to find an easy way to manoeuvre round a corner so as to use the least time possible and to make sure that the student race car is safe to drive round the corners at high speeds.

## 1.3 Project Objectives

1. Determine and design a wheel steering system suitable for inclusion in the ongoing development of FORMULA 1 student race car.
2. To determine effects of the modified parameters on the turning radius of the race car.
3. Develop a computer simulation of the steering system.

# CHAPTER TWO

LITERATURE REVIEW

## 2.1 Introduction

The most conventional steering arrangement is to turn the front wheels using a hand-operated steering wheel which is positioned in front of the driver, via the steering column, which may contain universal joints (which may also be part of the collapsible steering column design), to allow it to deviate somewhat from a straight line. [1]

Other arrangements are sometimes found on different types of vehicles, for example, a tiller or rear–wheel steering. Tracked vehicles such as bulldozers and tanks usually employ differential steering, that is, the tracks are made to move at different speeds or even in opposite directions, using clutches and brakes, to bring about a change of course or direction. [1]

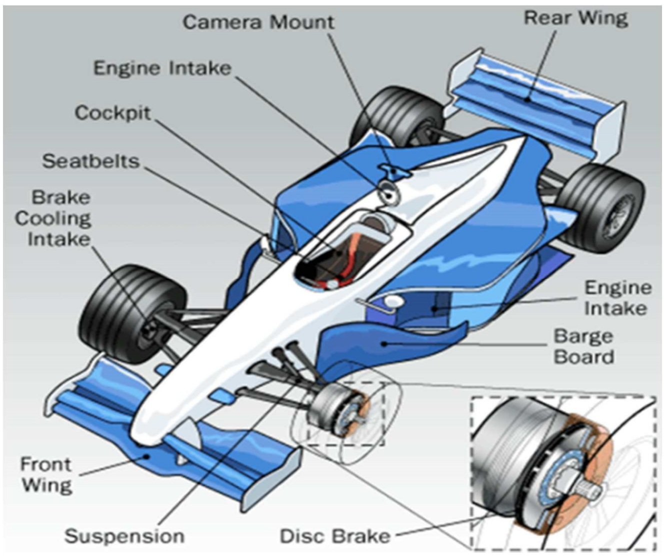


*Figure 1: Parts of an F1 Race Car (http://www.formula1.com)*

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## 2.2 Steering System

The design and type of the steering rack depends on the system chosen. The steering systems used are divided into:

a) Power assisted

b) Manual steering systems

Each is designed to help the driver to turn easily for optimal performance with different configuration of the vehicle. The type of steering system suitable for our light weight and simple design will only include the manual steering system due to cost and complexity factors.

### 2.2.1 Manual Steering Systems

The manual steering systems are used on light weighted vehicles, or vehicles which have the biggest distribution of mass on the rear wheels and can be easily turned with manual steering at low speed. The systems are fast and accurate and it provides a reliable design. [2]

However, it will become more difficult to handle the vehicle at low speed if wider tires are used or more weight is distributed to the front wheels. These concerns play a big role when analysing if manual steering should be used. [2]

The manual steering system incorporates:

- steering wheel and column
- a manual gear box and pitman arm or a rack and pinion assembly
- linkages, steering knuckles and ball joints
- wheel spindle assemblies

There are different types of manual steering gear systems:

a) Worm and roller

b) Worm and sector

c) Worm and nut

d) Cam and lever

e) Rack and pinion

a) Worm and sector

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]

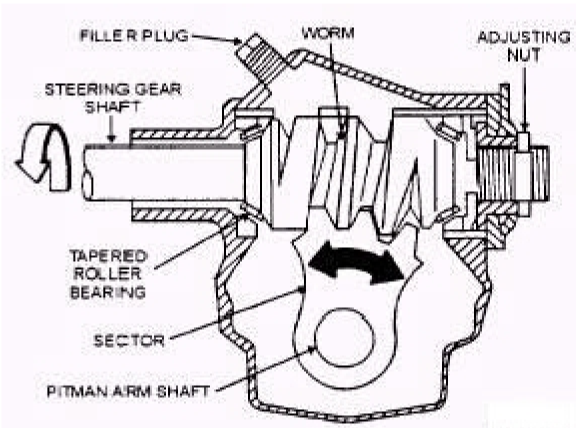


*Figure 2: Worm and sector steering gear system (http://www.tpub.com/basae/213.htm)*

b) Worm and Roller

The manual worm and roller steering gear has a three-turn worm gear at the lower end of the steering shaft. Instead of a sector on the pitman arm shaft, the gearbox has a roller assembly (usually with two roller teeth) that engages the worm gear. The assembly is mounted on anti-frictional bearings. When the roller teeth follow the worm, the rotary motion is transmitted to the pitman arm shaft, pitman arm and into the steering linkage. [3]

The worm has an hourglass shape for variable steering ratio and better contact for the worm and roller. The variable steering ratio will result that the wheels turns faster at some positions than others. This will provide more steering control at the centre of the worm, and more rapid steering as the wheels are turned. [2]

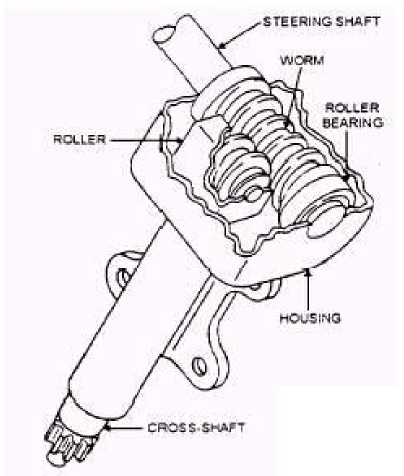


*Figure 3: Worm and roller steering gear system (http://www.tpub.com/basae/213.htm)*

c) <u>Worm and Nut</u>

The worm and nut steering gear comes in different combinations where the recirculating ball is the most common type. The recirculation ball combination offers the connection of the nut on a row of balls on the worm gear to reduce friction. Ball guides returns the balls as the nut moves up and down. The ball nut is shaped to fit the sector gear. [2, p. 16]

When the steering wheel is turned, the steering shaft rotates along with the worm gear fitted at the end of it. The recirculation balls starts to move, and this moves the ball nut up and down along the worm. This turns the pitman arm. [2, p. 16]

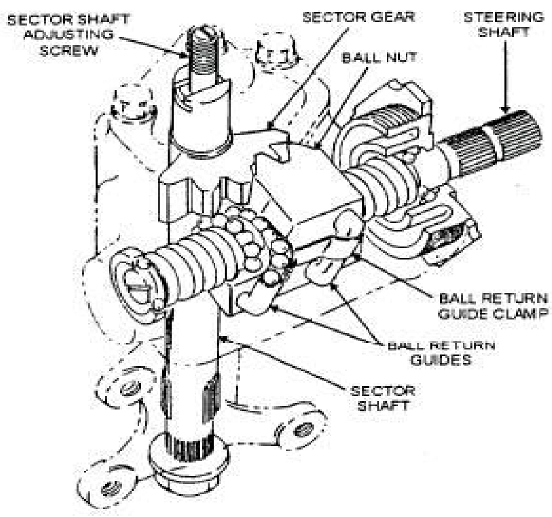


*Figure 4: Worm and nut steering gear system (http://www.tpub.com/basae/213.htm)*

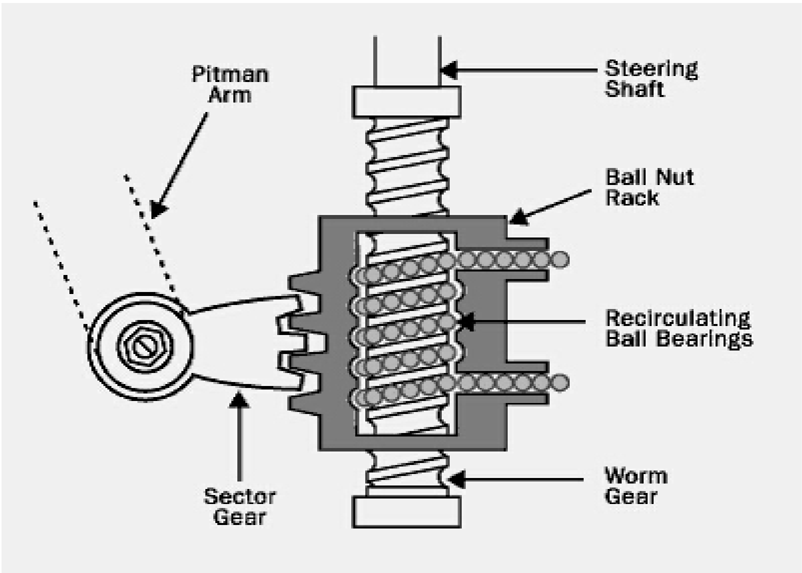


*Figure 5: Dissected worm and nut steering gear system (http://auto.howstuffworks.com/steering3.htm)*

d) Cam and Lever

In the cam and lever gear, two studs are connected on the lever and engage the cam. As the steering wheel is turned, the steering shaft will rotate and move the studs back and forth which move the lever back and forth. This will cause a rotation in the pitman arm. The lever is increased in angle compared to the cam, which will result in a more rapid move of the lever as it nears the ends, as in the worm and nut gear. [2, p. 17]

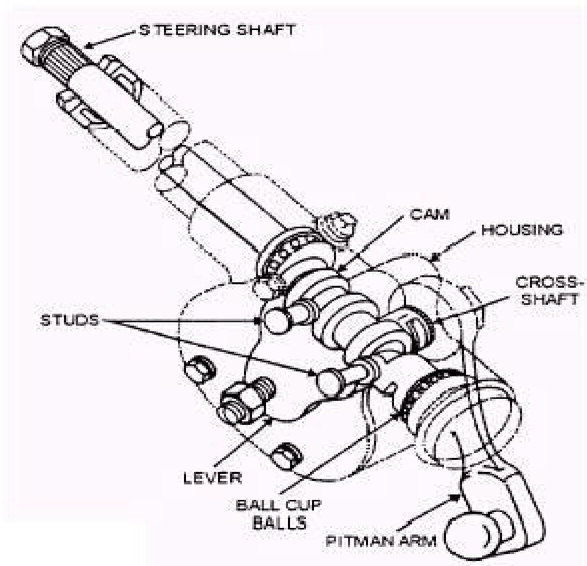


*Figure 6: Cam and lever steering gear system (http://www.tpub.com/basae/214.htm)*

e) Rack and pinion

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. [3] [2, p. 17]

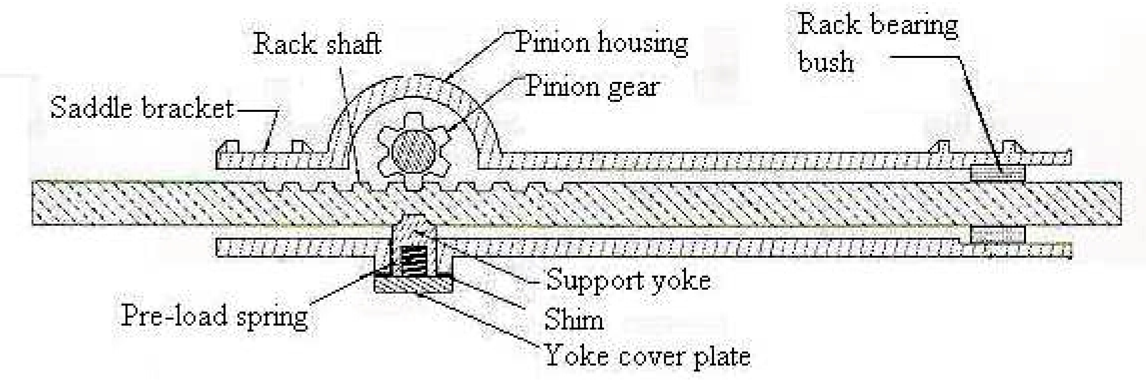


*Figure 7: Rack and pinion steering gear system (http://what-when-how.com/automobile/steering-components-automobile/)*

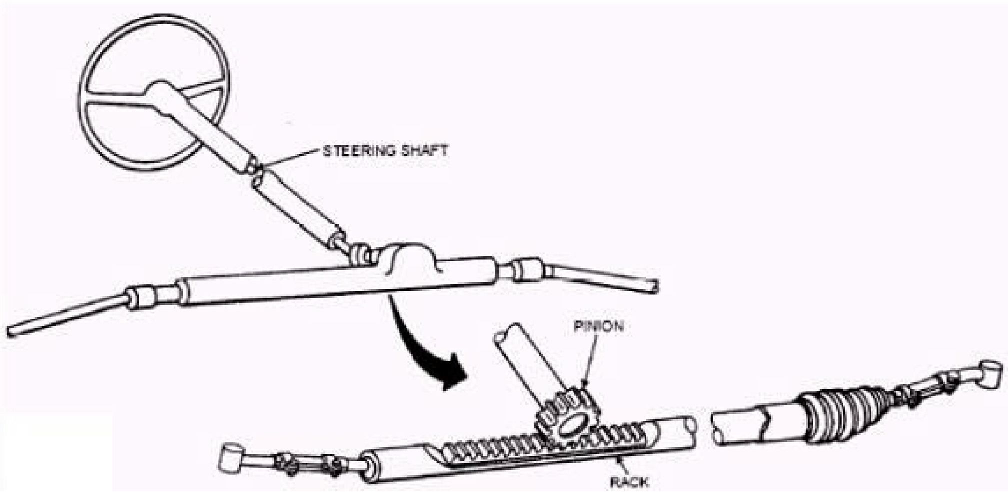


*Figure 8: Illustrative positioning of the rack and pinion gear system (http://www.tpub.com/basae/214.htm)*

### 2.2.2 Power assisted

Power assisted steering systems are used to amplify the turning moment applied to the steering wheels for heavier vehicles which might be hard to turn with a manually steering system at low speeds. This is practical when the car is at a standstill and the wheels have to be turned, i.e. when parking. [2, p. 14]

A power assisted steering system is supported by a hydraulic pump driven by the motor which directs pressurized oil, a boost, to the steering gear and helps to push or pull the rack in either of the directions. The boost is applied to the steering linkage or the steering gear. A flow control valve limits the fluid flow to the cylinder, and a pressure relief valve controls the pressure. [2, p. 14]

The system can also be electrical driven. This is more efficient as the electric power steering only needs to assist when wheels are turned and are not run constantly with the engine as the hydraulically driven system. It also works even if the motor is not running and by the elimination of the pump, hoses and fluids the weight is reduced. There is no leakage of fluids and it runs quieter as there is no pump. [2, p. 14]

Common types of power assisted steering systems are:

a) Power rack and pinion

b) Power re-circulating ball

a) Power rack and pinion

Power rack and pinion steering assemblies are hydraulic/ mechanical unit with an integral piston and rack assembly. An internal rotary valve directs power steering fluid flow and controls pressure to reduce steering effort. The rack and pinion is used to steer the car in the event of power steering failure, or if the engine (which drives the pump) stalls. [3]

When the steering wheel is turned, resistance is created by the weight of the car and tire-to-road friction, causing a torsion bar in the rotary valve to deflect. This changes the position of the valve spool and sleeve, thereby directing fluid under pressure to the proper end of the power cylinder. [3]

The difference in pressure on either side of the piston (which is attached to the rack) helps move the rack to reduce turning effort. The fluid in the other end of the power cylinder is forced to the control valve and back to the pump reservoir. When the steering effort stops, the control valve is centred by the twisting force of the torsion bar, pressure is equalized on both sides of the piston, and the front wheels return to a straight ahead position. [3]

As the pump is connected to the engine it only works when the engine is running. This is the reason why it is hard to turn the steering wheel when the car is turned off.

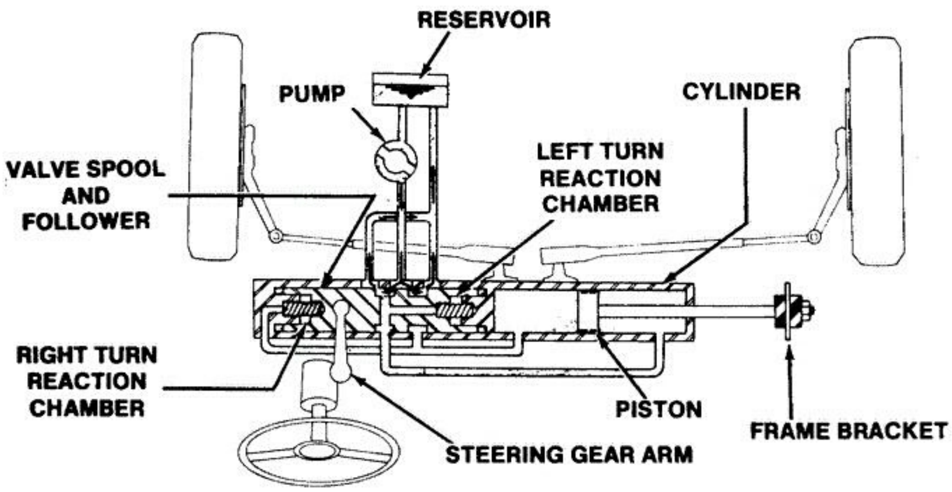


*Figure 9: Power assisted rack and pinion [2, p. 14]*

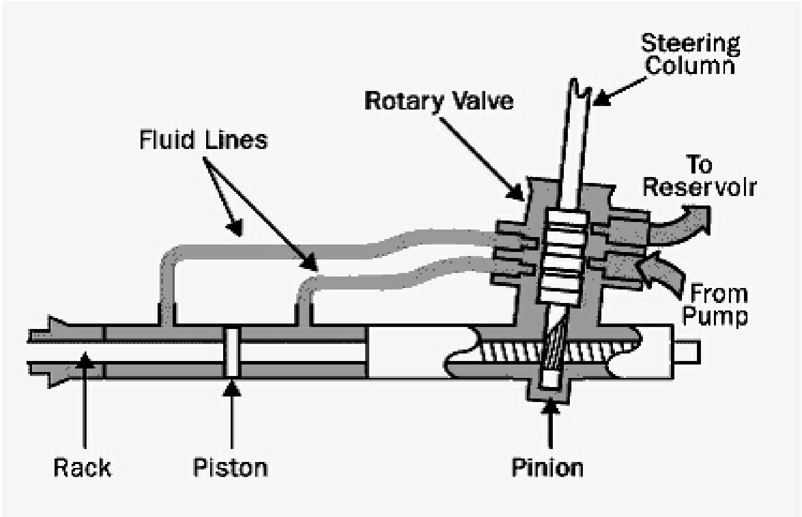


*Figure 10: Dissected power assisted rack and pinion (http://auto.howstuffworks.com/steering2.htm)*

b) Power recirculating ball

This power steering gear uses a recirculating ball system in which steel balls act as rolling threads between the steering worm shaft and the rack piston. The key to its operation is a rotary valve that directs power steering fluid under pressure to either side of the rack piston. The rack piston converts hydraulic power to mechanical force. The rack piston moves up inside the gear when the worm shaft turns right. It moves down when the worm shaft turns left. During these actions, the steel balls recirculate within the rack piston, which is power assisted in movement by hydraulic pressure. [3]

## 2.3 Geometric Parameters

### 2.3.1 Ackermann Condition

Ackerman steering geometry is used to change the dynamic toe setting, by increasing front wheel toe out as the car is turned into the corner.  Racers are interested because of the potential to influence the handling of the car on corner entry and mid corner. [4]

The typical steering system, in a road or race car, has tie-rod linkages and steering arms that form an approximate parallelogram, which skews to one side as the wheels turn.  If the steering arms are parallel, then both wheels are steered to the same angle.  If the steering arms are angled, as shown in the figure below, this is known as Ackerman geometry.  The inside wheel is steered to a greater angle then the outside wheel, allowing the inside wheel to steer a tighter radius.  The steering arm angles as drawn show 100% Ackerman. [4]

When a car goes round a corner, it turns around a point along the line of its rear axle, which means the two front wheels will have to turn through slightly different angles so that they are also guiding the vehicle round this point, and not fighting the turn by scrubbing. [5]

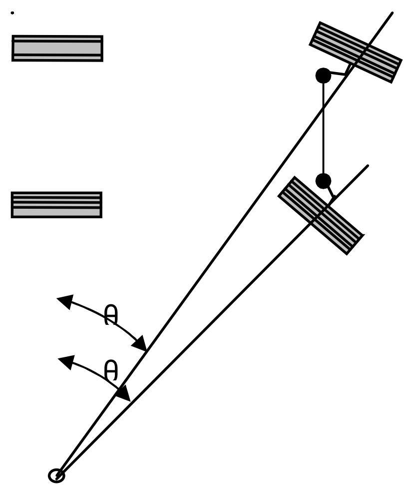


*Figure 11: Ackermann Principle [6]*

Ackerman geometry results when the steering is done behind the front axle and the steering arms point toward the centre of the rear axle as seen on Figure below

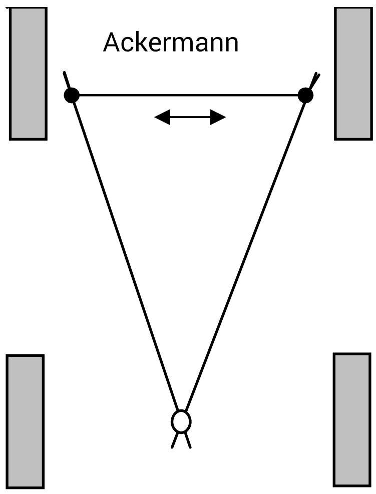


*Figure 12: Ackermann Construction [6]*

For the Ackermann analysis the Ackermann condition is used to determine the relationship between inner and outer wheel in a turn and the radius of turn.

General equation:

$$\frac{1}{\tan\theta_o} - \frac{1}{\tan\theta_i} = \frac{B}{L}$$

Where:

$\theta_o$= turn angle of the wheel on the outside of the turn

$\theta_i$= turn angle of the wheel on the inside of the turn

B= track width

L= wheel base

b= distance from rear axle to centre of mass

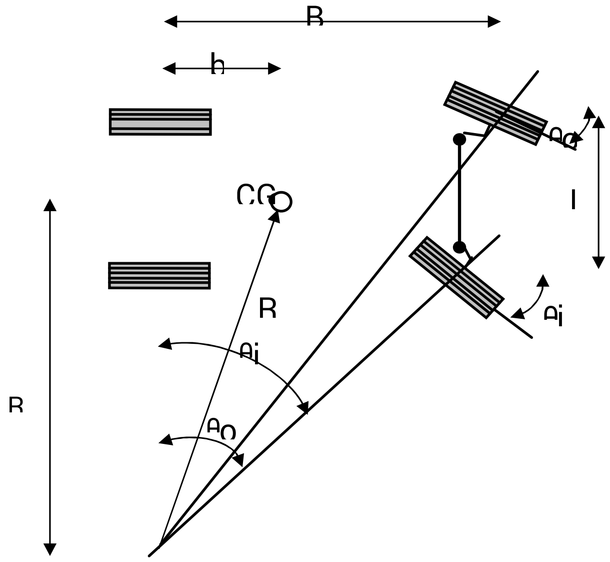


*Figure 13: Ackermann Condition [6, p. 40]*

#### 2.3.2 Wheel toe

The toe in/out configuration explains the position of the wheels relative to the neutral toe position where the distance between the front parts of the wheels equals the distance between the rear parts. The tie rods are adjusted to give the desired toe.

The toe configuration affects the handling of the vehicle and its performance on the straights and corners. Tire wearing will also be affected by the configuration chosen. By analysing the race track, the best decision of the toe configuration can be made. [2, p. 20]

- Toe in

In a “toe in” configuration, or positive toe, the front wheels are turned inwards giving a shorter distance between the front parts of the wheels and bigger distance at the rear parts. By increasing the “toe in” configuration, better stability can be achieved on the straights, but it will give less turning response in the curves. [2, p. 21]

- Toe out

The distance at the front parts of the wheels are bigger compared to the rear parts.

This is also called negative toe. The “toe out” position is more common in racing as the wheels are aligned in a position that encourages the initiation of a turn. The “toe out” may appear in five forms at the vehicle:

a) *Static “toe out”* - This is the “toe out” as a result of the adjustments of the tie rods. The tie rods are adjusted in a way that the wheels are “toed out”.

b) *“Toe out” due to the tie rods configuration*- By using a shorter tie rod on the front left side compared to the front right side, the left wheel is steered at a larger angle than the right wheel when turning to the left. However, when turning to the right, this configuration will give a “toe in” position.

c) *Toe out on Ackerman steering*- When using an Ackerman configuration, the “toe out” will occur when turning the wheels. This means that the toe out due to the Ackerman configuration only occurs in turns.

d) *Toe out due to bump steer*- When riding, the ride motions and the body roll can lead to toe out.

e) *Toe out due to slip angles*- As the vehicle is turning, the outside contact patch between the wheel and the road will experience heavier load than the inside contact patch which result in a larger slip angle for the outside patch than the inside patch. As a result, the contact patches can be toed out.

[2, p. 21]

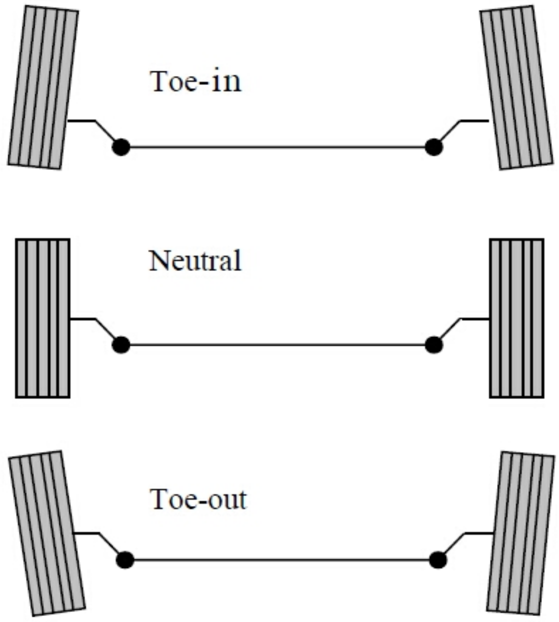


*Figure 14: Diagrammatic representations of toe in, toe out and neutral conditions [6, p. 26]*

#### 2.3.3 Camber Angle

The camber angle is the angle between the vertical axis of the wheel and the vertical axis of the vehicle. The angle is negative if the wheel leans towards and positive if it leans away from the chassis. [2, p. 22]

The cornering forces on a wheel are dependable on the wheels angle on the road surface, and so the camber angle plays a major role on the forces acting on the car. It can also be used to increase the temperature in the wheels to their proper operating temperature by giving more negative camber angle. [2, p. 22]

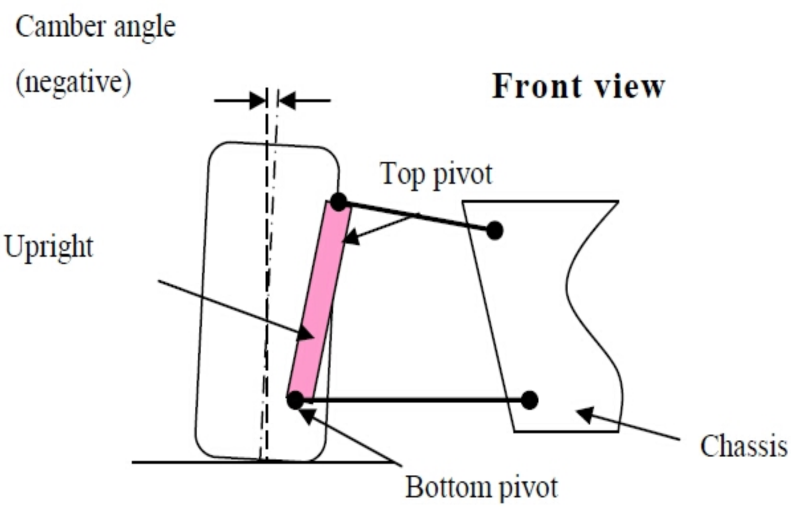


*Figure 15: Camber Angle [6, p. 22]*

#### 2.3.4 Caster Angle

The caster angle is the angle between the pivot line and the vertical line at the centre of the wheel. The angle adds damping to the steering system as it controls the steering; too much caster angle makes the steering heavy. The caster can be positive or negative:

- **Positive**: If the top pivot is placed further to the rear than the bottom pivot – axis tilted forward
- **Negative**: If the top pivot is placed further to the front than the bottom pivot – axis tilted backward

Positive caster angle enhance straight-line stability when driving forward as it straightens the wheels. This happens because the steering axis, which points forward, pulls the wheel along when the car moves. As the caster angle is increased, camber gain can be achieved in corners. [2, p. 22]

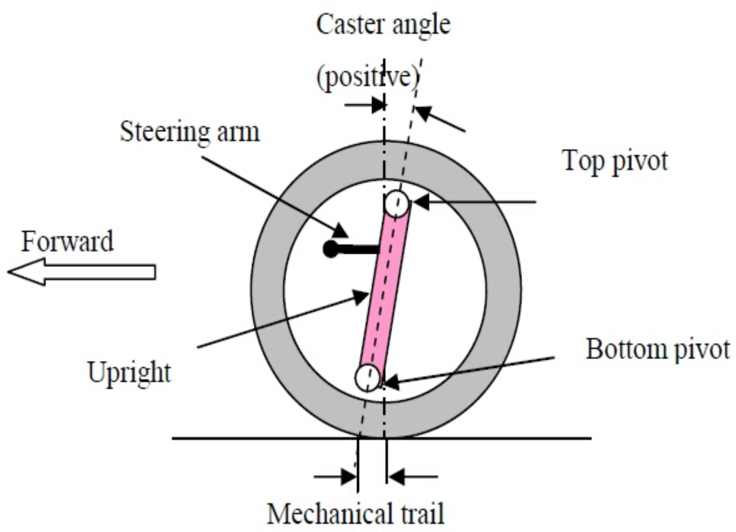


*Figure 16: Caster Angle [6, p. 23]*

### 2.3.5 Bump-steer

If the vehicle experience bumps on the track, the wheels may have the tendency to steer themselves without the driver doing any changes to the steering wheel. This is undesirable and known as bump-steer. The wheels will change between toe out and toe in as the suspension compress and de-compress during the bump. The steering wheel must be moved constantly to keep the vehicle in a constant turn. The wheel will also tend to toe out in a sharp turn as some of the weight is distributed to the outer wheel and hence makes the suspension on the outer wheel to compress. Bump-steer will also cause increase tire wear.

Bump-steer can be avoided by designing the same length and angle on the tie rod and the lover a-arm, and by ensuring that they both travel along the same arc during a bump.

By comparison; if the tie rod got a shorter length, the travelling arc would also be shorter. The shorter arc would pull on the wheel and make it toe in during a bump.

It can also be controlled by introducing shims on the connection point between the tie rod and the bolt, which will increase the elimination of the toe experienced. However, it is important that the size of the shims is within the limit of what the bolt can handle, as the shim introduces a moment to the bolt when the lateral force is acting on the tire. The bigger the lateral force, the smaller the shim should be to avoid big moment on the bolt. [2, p. 22]

# CHAPTER THREE

## RESEARCH METHODOLOGY

The formula 1 race car is a reflection of how modern automotive engineering has progressed over the years. The modern car technology has progressed mostly through the formula 1 industry with dire need to create the fastest and safest car available. Different components and systems have been invented through the need to make work easier and more accurate.

Designing and selection of the most suitable steering system was done through research on the most common ones used in the market with their pros and cons noted. The research was greatly influenced by a set of criterion to lead us to the perfect selection listed below:-

- Cost.
- Availability.
- Complexity.

The main objective is to reduce the turning radius of the vehicle.

The main sources of research was from automotive experts ( Mr. Jack who specialises in car hardware components), automotive books, the internet, reports on the F1 compiled by students that took part in the competitions and articles specialising on the steering system. The various steering systems were carefully considered and the selected system was narrowed down to one that matched our criterion.

The design process was laid out where the initial design was facilitated by determining the dimensions of the race car. The dimensions were carefully chosen by the previous formula 1 student race car project done by the previous year's project. The steering system Ackermann condition which was suggested by Rudolph Ackermann is still in use today for most cars since it is accurate and accounts for difference in tire angles to prevent slip when turning. Various dimensions of the moving components like the connecting rack and tie rod were found from the Ackermann geometry which made it easier to calculate the sizes and movement of the parts with the angles to be made.

Mechanical movements and dimensions were not enough in the design process since mechanical properties of the materials chosen had to be calculate to prevent catastrophic failure of the components. First, the force transmitted through from the tires to the steering arm through the tie rod to the connecting rack then the rack and pinion gear to the steering rod and finally the steering wheel where the driver uses to turn the tire had to be calculated. The components had to be designed to withstand the force without possible failure with a safety factor of 2 or more. The material selection process was carried out according to the ISO standards.

The rack and pinion selection was based on the F1 standards where the ideal ratio of number of turns has to be as low as 1:12 where normal cars have a steering ratio of 1:20 from lock to lock. The idea is based on the crucial factor of time where more turns translates to higher time consumption and lack of proper control of the vehicle on a turn. Hence the pinion gear had to be bigger to reduce the number of turns made by the driver, which consequently increased the effort needed to turn.

Formula 1 race cars are engineered to move at very high speeds up to 360 km/h but unfortunately the race tracks are not straight and hence the car needs to slow down at corners which brings the need to design better cars to move at higher and higher speeds around corners without toppling over or spinning around due to oversteer. Reducing the turning radius enables the car to move at higher speeds around the corner and hence design changes have to be made.

Several design analyses were put to test through varying the turning radius and change in centre of gravity and varying the inner tire angle. The tests and calculations suggested the most suitable weight ratio and tire angles needed to create the best design for maximum effectiveness of turning speeds with the right amount of oversteer without causing the car to spin or topple over. A graphical analysis shows the rate at which each degree of variation affects the movement.

A visualisation of the proposed design was shown clearly on a 3D AUTOCAD drawing where the components, exact sizes and movement of the steering system can be simulated. The components that undergoes stresses and deflections were put to test using a stress and deflection of finite element analysis simulation program called ANSYS. The components were able to withstand the acting forces and therefore passed the test.

# CHAPTER FOUR

# DESIGN OF THE STEERING STYSTEM

## 4.1 Introduction

From the literature review, various kinds of steering systems and designs in existence that have been used have been looked at. Therefore, a system had to be chosen that will be suitable for the student race car and that will be able to achieve the objective.

To achieve all these, guidelines had to be set for the design. These guidelines were set by selecting adequate and essential design parameters. Some other guidelines that were set were how to achieve the desired objective from the problem statement. The turning radius was reduced through:

- Over-steer: the front to back weight distribution was selected as 30:70.
- Selecting a suitable material that is light and strong enough for the design.
- Ensuring that the steering ratio is a value less than 8.

## 4.2 Design Parameters

The design concept is based primarily on the design parameters set up by the previous race car project for the steering wheel. The design parameters are listed below:

| **Parameter** | **Value (mm)** |
|---|---|
| Front axle-to-nose cone | 600 |

| | |
|---|---|
| Rear axle-to-back | 200 |
| Height | 950 |
| Weight | 300 Kilograms |
| Wheelbase | 1600 |
| Length | 2450 |
| Track Width | 1210 |
| Maximum inner wheel turning angle ($\delta_i$) | $40^0$ |

*Table 1: Design Parameters*

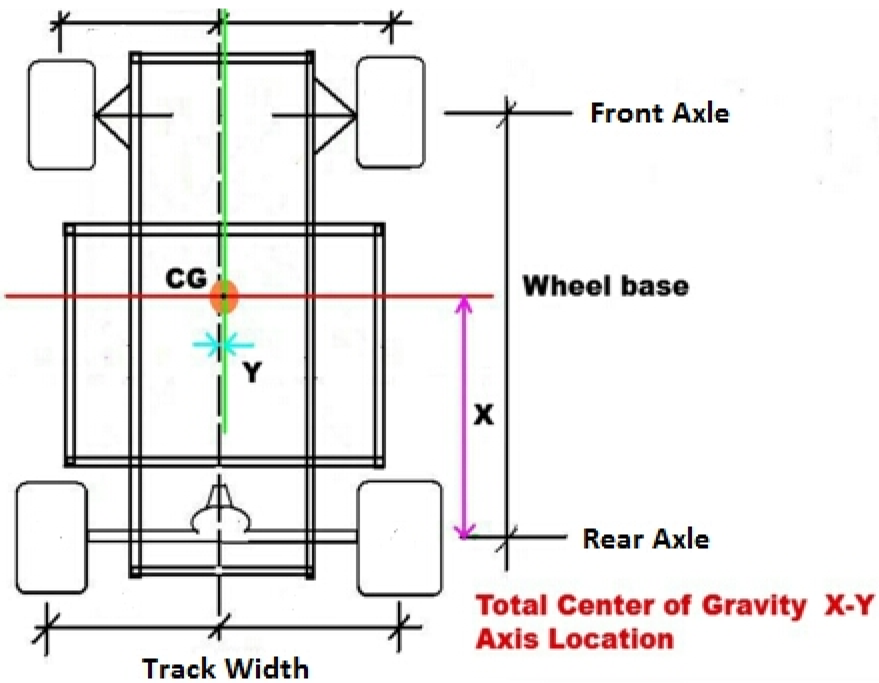


*Figure 17: Positioning of the track width, wheel base and centre of gravity (http://www.racecartuning.com/alignment.html)*

The above parameters are some of the essential vehicle parameters that will be used to find the turning radius of the vehicle. As stated above, the weight distribution of the student race car is 30:70. Thus the weight at the front is 90Kgs and the weight at the back is 210Kgs. This weight distribution is measured from the center of gravity to the absolute ends of the race car. The indicated length is the length from the nose to the rear wing.

## 4.3 Working Principle of a Two Wheel Steering System

This project involved design of a two wheel steering system which will reduce the turning radius. For the Formula 1 student race has its steering system positioned centrally, unlike in normal road cars where the steering system is either to the right or left. This makes the maneuvering ability easier. From the many available forms of the steering system, the rack and pinion steering system was selected due to its ease in usage and maintenance. It has fewer parts, it is lighter, it gives better and quicker feedback and it is easy to repair.

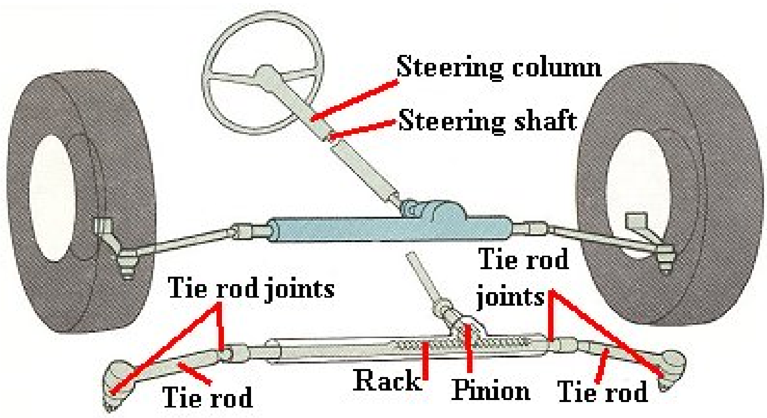


*Figure 18: Parts of a steering system (http://m5carblog.blogspot.co.ke/2013/02/steering.html)*

A rack-and-pinion gear set is enclosed in a metal tube, with each end of the rack protruding from the tube. A rod, called a tie rod, connects to each end of the rack. The pinion gear is attached to the steering shaft. When you turn the steering wheel, the gear spins, moving the rack. The tie rod at each end of the rack connects to the steering arm on the spindle. Both the pinion and the rack teeth are helical gears. Helical gearing gives smoother and quieter operation for the driver.

Mechanical advantage is gained by the reduction ratio. The value of this ratio depends on the size of the pinion. A small pinion gives light steering, but it requires many turns of the steering wheel to travel from lock, to lock. A large pinion means the number of turns of the steering column is reduced, but the steering is heavier to turn.

Once the tyres start turning, the Ackermann effect takes place to avoid slipping of the front tyres. Thus the inner front tyre turns at a larger angle than the outer front tyre so as to avoid slipping. As the race car moves round a corner at high speed, the rear tires tend to slip due to most of the weight of the car being at the back; 70% of the weight is at the back. The slipping of the car in this manner causes over-steer and this phenomena reduces the turning radius of the car. The driver should on the other hand maintain the car in control to avoid spinning out of the car due to too much over-steer. This is why the steering ratio should be low enough to make the steering system sensitive.

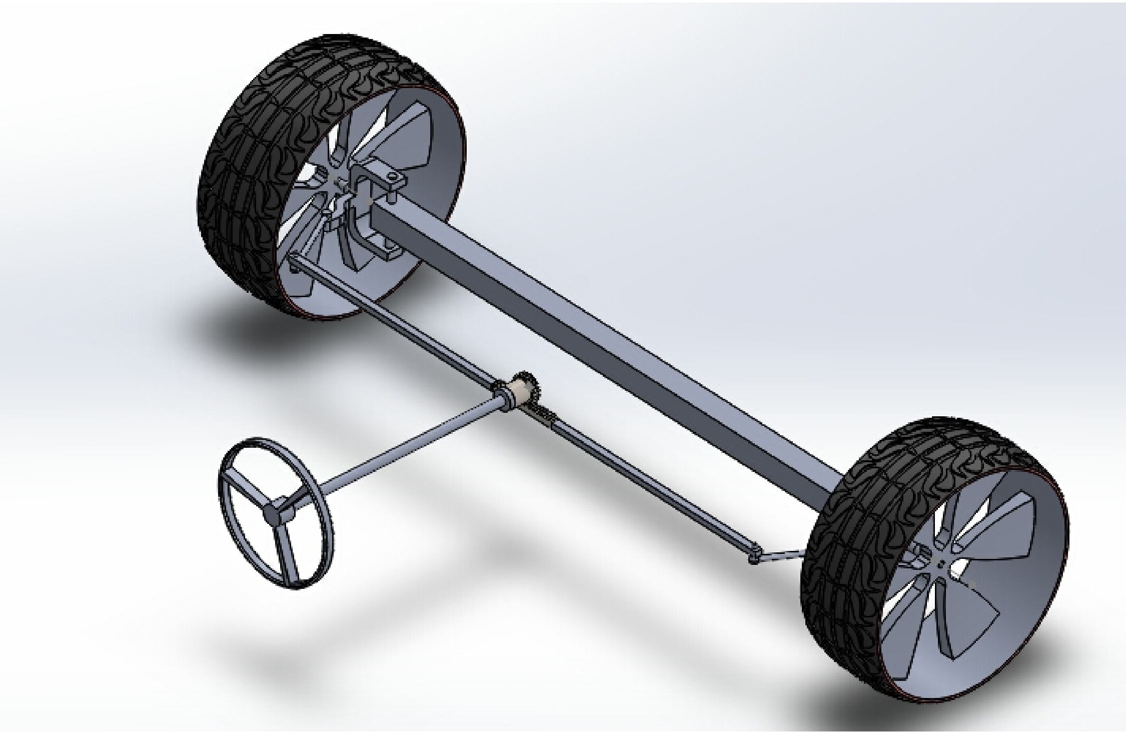

*Figure 19: Simulated design of the steering system*

## 4.4 Fundamental Design Calculations

## 4.4.1 Ackerman angle

The Ackerman steering geometry is a geometric arrangement of linkages in the steering of a car or other vehicle designed to solve the problem of wheels on the inside and outside of a turn needing to trace out circles of different radius.

The Ackerman angle is very important in designing the steering components. We have two components moving together – the left and right steering knuckles, but the relationship between their motions changes as we move them.

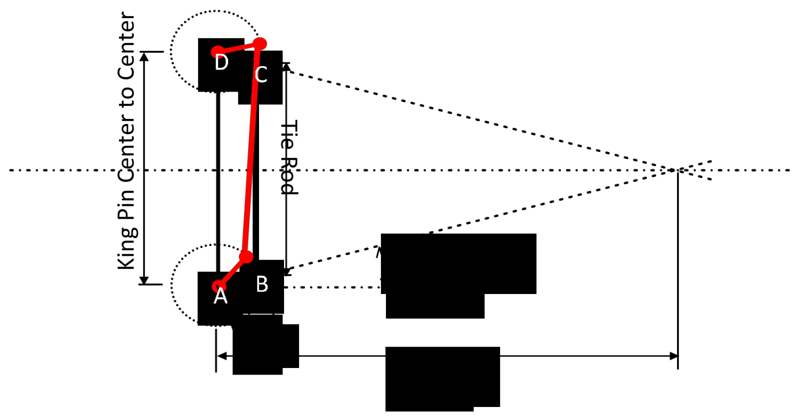


*Figure 20: Ackermann Angle [7]*

$$\frac{\textit{Kingpin centre to centre distance}}{2}$$

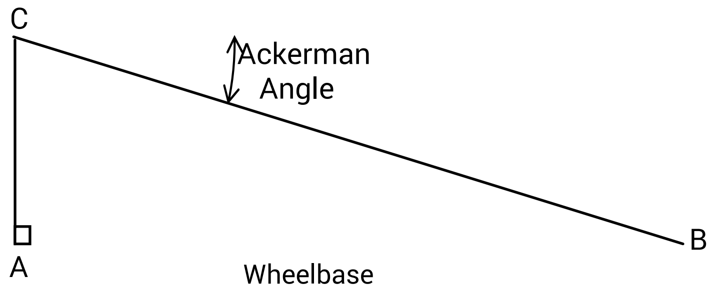


*Figure 21: Relation of ackerman angle, wheel base and kingpin centre to centre distance [7]*

$$\tan\varnothing = \frac{\textit{kingpin centre to centre distance /2}}{\textit{wheelbase}}$$

$$\tan\varnothing = \frac{\frac{1210}{2}}{1600}$$

$\varnothing = 20.712^0$

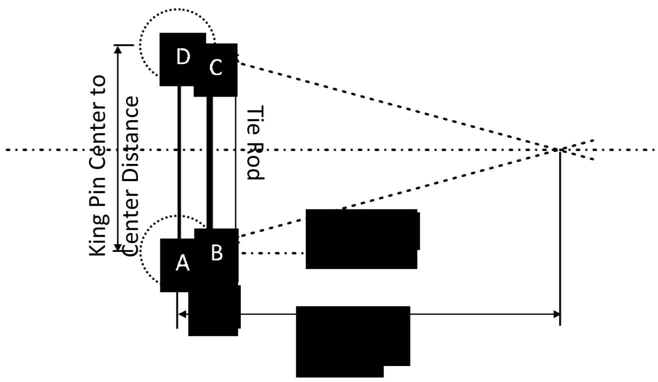


*Figure 22: Ackermann Angle [7]*

The Ackerman angle is calculated as $20.712^0$. This angle is implemented into the design so as to make sure the inner and outer front wheels do not turn at the same angle so as to prevent slip which will wear down the tyres really fast and reduce the speed of the race car during cornering.

### 4.4.2 Tie Rod Length

The tie rods are the connection from your steering system to your wheels. These two systems, however complex, contain a very simple, yet important part, the tie rod. Tie rods play a crucial role in your steering system. Without tie rods, your steering system would fail. Tie rods are the pivot point between your steering system and your steering arm and wheel. The outer tie rod end is adjustable. This means that you can change the length of the tie rod to fix your vehicle's alignment.

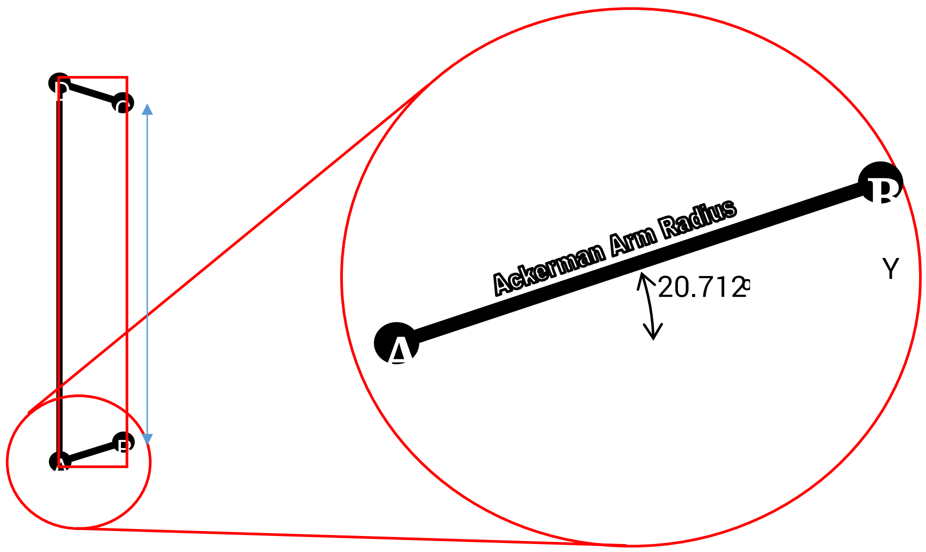


*Figure 23: Ackermann arm radius [7]*

$$\sin\varnothing = \frac{Y}{R}$$

Where: R = Ackerman Arm Radius

Kingpin C-C = kingpin center to center distance

(...Assume R=150 mm) $Y = \sin 20.713 \times 150\ mm$

$Y=53.053\ mm$

Hence: Tie Rod Length = kingpin C-C dist. – 2Y

= 1210 – (2×53.053)

= 1103.9 mm.

-

## 4.5 The Rack and Pinion Design

As stated earlier, the type of gear chosen was the rack and pinion. This steering gear with a shift movement is used not only on small and medium-sized passenger cars, but also on heavier and faster vehicles, such as the Audi A8 and Mercedes E and S Class, plus almost all new light van designs with independent front wheel suspension. The advantages of this system are:

- Simple construction
- Economical and uncomplicated to manufacture
- Easy to operate due to good degree of efficiency;
- Contact between steering rack and pinion is free of play and even internal damping is maintained
- Tie rods can be joined directly to the steering rack
- Minimal steering elasticity compliance
- It is compact (the reason why this type of steering is fitted in all European and

Japanese front-wheel drive vehicles)

- The idler arm (including bearing) and the intermediate rod are no longer needed
- Easy to limit steering rack travel and therefore the steering angle. [8]

### 4.5.1 Determining the displacement of the rack

The lock to lock angle of the tires was agreed to be $80^0$ which meant that when the tires are pointing straight ahead, the maximum deflection of the inner tires is $40^0$.

Since the tie rod pushes the steering arm on one side and the other side is fixed, we can determine the displacement of the rack for a $40^0$ deflection.

The length from the fixed point to the tie rod connection is assumed to be 100 mm.

r = 100 mm

Therefore,

$$\text{Displacement} = \frac{40}{360} \times \pi \times 200 \text{ mm}$$

$$= 69.8 \text{ mm}$$

Therefore, to make an angle of $80^0$ the rack has to be twice that length since the pinion will be at the middle of the rack.

Hence:

Rack length = 69.8 × 2 = 139.6 mm ≈ 140 mm

### 4.5.2 Designing the pinion required to displace the rack

For the pinion geometry, it was agreed that we make one revolution of the pinion to cause a lock to lock displacement of the rack. Considering that the steering wheel is required to turn an angle of $180^0$ to the right to make the tires turn an angle of $40^0$ and vice versa for the left turn, the pinion's circumference is equal to the length of the rack.

Therefore, $D_p = 140/\pi$

$$= 44.5 \text{ mm} \approx 45 \text{ mm}$$

### 4.5.3 Number of teeth on pinion

Since for one rotation of the pinion, the rack makes a lock to lock movement, then the length of the rack is pitch circle circumference of the pinion. Taking the module of the pinion as 2, we calculate the teeth of the pinion as:

Pinion circumference = π × module × number of teeth

Where:

Pinion circumference= 140mm

Module=2

$$\text{Number of teeth} = \frac{140}{2\pi} = 22.28$$

This value is rounded up to 23 teeth. This is a more accurate approximation of the number of teeth but we will use 24 teeth in our design to make the pinion stay at the middle of the rack without pre-load.

### 4.5.4 Number of teeth on rack

Since the pinion makes one revolution to displace the rack from lock to lock, the gear ratio is 1. Thus the rack has the same number of teeth as the pinion, which is 24 teeth.

## 4.6 Material Selection

A material has attributes: its density, strength, cost, resistance to corrosion, and so forth. A design demands a certain profile of these: a low density, a high strength, a modest cost and resistance to sea water, perhaps. It is important to start with the full menu of materials in mind; failure to do so may mean a missed opportunity. If an innovative choice is to be made, it must be identified early in the design process.

There are four main steps, which are translation, screening, ranking and supporting information. The first step in tackling it is that of translation, examining the design requirements to identify the constraints that they impose on material choice. The immensely wide choice is narrowed, first, by screening-out the materials that cannot meet the constraints. Further narrowing is achieved by ranking the candidates by their ability to maximize performance. Criteria for screening and ranking are derived from the design requirements for a component by an analysis of function, constraints, objectives, and free variables.

The material selection process begins with the family of materials. From this family, materials are eliminated slowly to remain with the final best material to be used for the design process.

The family of materials includes:

- Ceramics
- Glasses
- Metals
- Polymers
- Elastomers
- Hybrids
- Wood

1. <u>Translation</u>

Function and constraints, objective and free variables define the boundary conditions for selecting a material. Translation involves setting these boundaries that the design is subject to. For our steering system design, these boundaries include:

- To be able to support load
- To contain pressure exerted to the wheels
- To be as light and strong as possible
- To avoid any buckling in the design

- Cheap enough
- To survive under all conditions

2. Screening

Unbiased selection requires that all materials are considered to be candidates until shown to be otherwise. The first of these eliminates candidates that cannot do the job at all because one or more of their attributes lies outside the limits set by the constraints. The materials eliminated were:

- Ceramics- this family was eliminated since its dimensional tolerance is difficult to control during processing, it is weak in tension, it has poor shock resistance, and it can crack when hit with heavy items.
- Glasses- it was eliminated because it is an expensive material, it breaks easily and when broken, the pieces may cut you.
- Elastomer-these were eliminated since they are relatively expensive

3. Ranking

This is the process of arranging materials form the best to the worst based on a certain criteria. In this case it was done according to the stiffness of the remaining families (from the graphs).

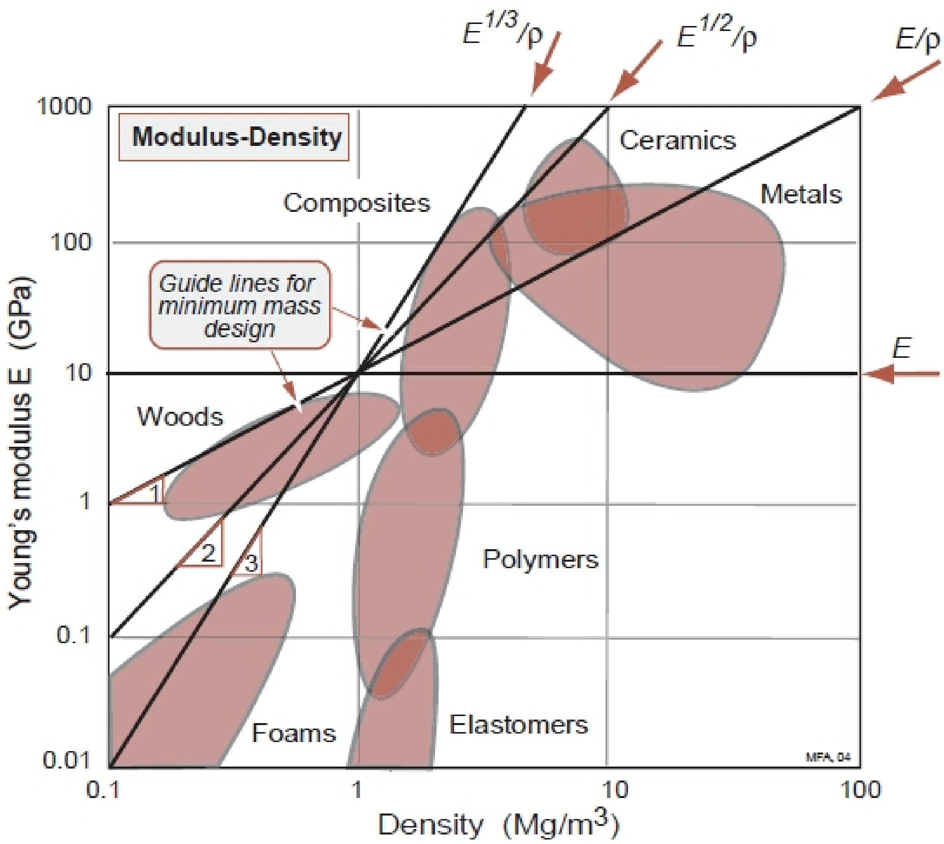


*Figure 24: Young's Modulus for various materials [9]*

The ranking from highest to lowest was:

- Metals
- Hybrids
- Wood
- Polymers

4. <u>Support information</u>

For making the steering system we required a material that is malleable and has high strength properties. From research it was found that metal was the most appropriate material. Most metals are strong enough to withstand an appropriate amount of load and it is relatively easy to cut through.

5. Local conditions

Metal is readily available in the Kenyan market and is fairly cheap. Hence metal is the material best suited for making the drawers.

### 4.6.1 Metal Selection

There are various kinds of metals and we have to select one. So we re-do the material selection steps.

a) Translation

We had to choose one form of metal that is suitable for our design. For our design, we had to have a metal that is:

a. Ductile
b. Malleable
c. Corrosion resistant
d. Tough
e. Cheap

| TOUGHNESS | BRITTLENESS | DUCTILITY | MALLEABILITY | CORROSION RESISTANCE |
|---|---|---|---|---|
| Copper | White Cast Iron | Gold | Gold | Gold |
| Nickel | Gray Cast Iron | Silver | Silver | Platinum |
| Iron | Hardened Steel | Platinum | Aluminum | Silver |
| Magnesium | Bismuth | Iron | Copper | Mercury |
| Zinc | Manganese | Nickel | Tin | Copper |
| Aluminum | Bronzes | Copper | Lead | Lead |
| Lead | Aluminum | Aluminum | Zinc | Tin |
| Tin | Brass | Tungsten | Iron | Nickel |
| Cobalt | Structural Steels | Zinc | | Iron |
| Bismuth | Zinc | Tin | | Zinc |
| | Monel | Lead | | Magnesium |
| | Tin | | | Aluminum |
| | Copper | | | |
| | Iron | | | |

* Metals/alloys are ranked in descending order of having the property named in the column heading

*Table 2: Properties of different Metals [10]*

b) Screening

Using this table, we were able to narrow down to a few metals. These included gold, silver, aluminum, copper, tin, lead, zinc and iron. From these, the following were eliminated using our objectives:

i. Gold and silver were immediately disqualified because from our objectives, we needed a metal that is in-expensive.

ii. Lead was also eliminated since it is quite a heavy metal. It weighs about 320Kgs per cubic foot.

iii. Tin was eliminated since at very low temperatures, it has a tendency to decompose.

c) Ranking

We thus remained with aluminum, copper, zinc and iron. From these we ranked the materials from the best to the worst. We did this using the table above where the materials were arranged in descending order. Thus we arranged them in their various categories as follows in descending order:

| Toughness | Ductility | Malleability | Corrosion Resistance | Cheapness |
|---|---|---|---|---|
| 1. Copper<br>2. Iron<br>3. Zinc<br>4. Aluminum | 1. Iron<br>2. Copper<br>3. Aluminum<br>4. zinc | 1. Aluminum<br>2. Copper<br>3. Zinc<br>4. Iron | 1. Copper<br>2. Iron<br>3. Zinc<br>4. Aluminum | 1. Iron<br>2. Aluminum<br>3. Zinc<br>4. Copper |

*Table 3: Ranking of Metals*

From the ranking, we are able to see that iron is the best suited material for our design.

d) <u>Support Information</u>

Iron in itself is a very brittle material under tension but very strong in compression. Thus modifications have to be done to it so as to strengthen it. To achieve this, we have to increase the carbon content in the iron. Increasing the carbon content results in an increase in yield stress and ultimate tensile stress, while the elongation remains essentially constant. This thus turns the iron into steel.

e) <u>Local Conditions</u>

There are various forms of steel in the market, but we have to pick one that is readily available and cheap enough to build the student race car steering system. We found out that mild steel was easily available to us.

Mild steel has the following mechanical properties:

| **Material Property** | **Magnitude** |
|---|---|
| Modulus of Elasticity | 200GPa |
| Tensile strength | 455MPa |
| Yield strength(tension) | 250MPa |
| Ductility, percent elongation in 50mm | 23 |
| Poisson's ratio | 0.29 |

*Table 4: Properties of Mild Steel*

Thus mild steel will the material used to design most of the steering system components.

-

## 4.7 Calculation of Forces and Torques

### 4.7.1 Force Required to Turn the Tyres

The frictional force caused by the contact between the ground and tyre is transmitted from the tyre, through the steering arm, to the tie rod, all the way to the rack and pinion and finally to the steering wheel where the driver has to overcome this frictional force so as to make the wheels turn. The force that is transmitted is destructive to the mechanical components and can cause failure.

To prevent failure, the force is calculated and the components are designed to withstand such forces.

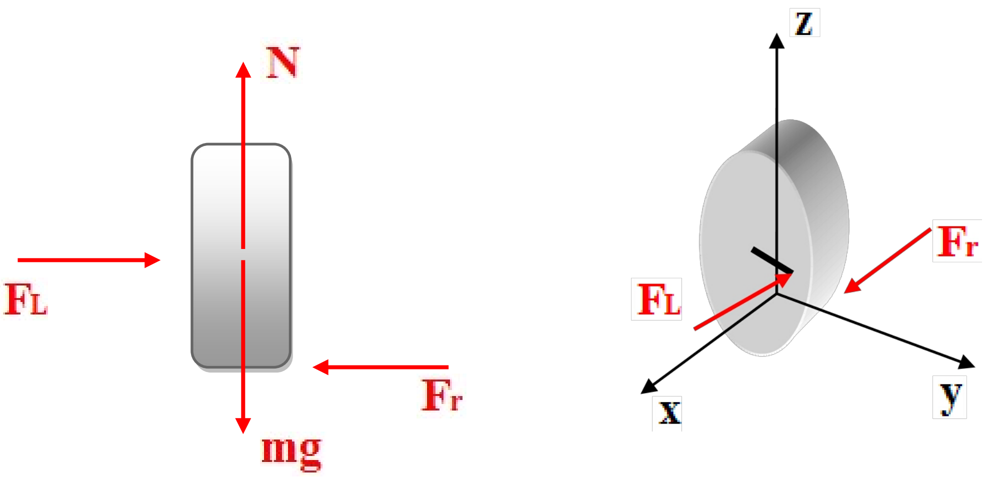


*Figure 25: Forces on the wheel [11]*

**$F_r$ =** friction force

**mg =** weight

**$F_L$ =** lateral force applied from the steering wheel

**N =** normal reaction

The wheel rests on the ground on a surface and not in a point so it appears with two friction forces as we can show in the figure below.

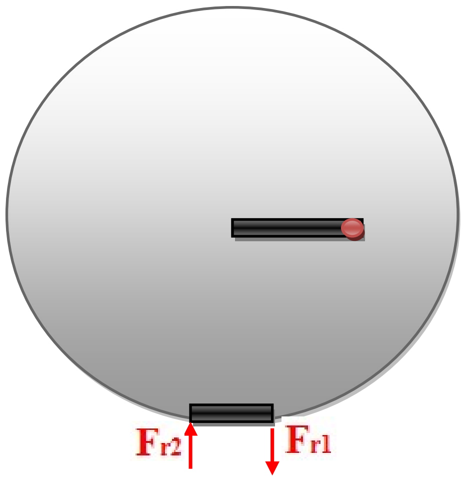


*Figure 26: Application of frictional forces on the wheel [11]*

In the diagram above shows a section of wheel. The black rectangular section at the centre of the wheel is the position of the steering arm, with the red dot being the application point of the lateral force from the steering wheel. The distance between the red dot and the middle of the wheel is the steering arm length (Rs). The small black rectangular section at the bottom represents the contact surface between the wheel and the ground, with $F_{r1}$ and $F_{r2}$ being the friction forces that occur on the contact surface of the wheel. These forces act on a radius r from the middle position of the wheel.

$F_{r1}$ and $F_{r2}$ act in different directions but they are of the same magnitude and under the same moment. Thus we can say:

$$F_{r1} = F_{r2} = F_r$$

Adding up the horizontal forces:

$\Sigma F_x = 0 \qquad F_L - F_r = 0$

$F_L = F_r \qquad (1)$

Adding up the vertical forces:

$\Sigma F_y = 0 \qquad N - mg = 0$

$N = mg \qquad (2)$

Summing up the moments about the centre of the wheel:

$\Sigma M_y = 0$ $\qquad$ $(F_L \times R_s) - (2 \times F_r \times r) = 0$ $\qquad$ (3)

Since this project is limited to the steering system and the other parts of the car are not designed, the weight of the car is assumed.

A typical Formula 1 student race car usually weighs 300 Kgs and the driver can weighs about 80 Kgs.

Therefore, the total mass is (300 + 80) Kgs = 380 Kgs.

To calculate the weight distribution on each tyre, the weight ratio for front to back was taken to be 30:70. That means that the front tyres takes only 30% of the total weight of the car. Therefore the mass on the front tyres is:

$380 \times \frac{30}{100} = 114$ Kgs.

The mass exerted on one tyres will be half of the 114 Kgs which is 57 Kgs.

Hence the weight will be: $57 \times 9.81 = 559.17$ N

$F_r$ may be found using the following formula:

$F_r = \mu \times N$ $\qquad$ Where: $\qquad$ $\mu$ = friction coefficient

$N = mg$

The friction coefficient will be of a higher value in order to establish a safety coefficient. So we take $\mu = 1$.

Now we calculate the friction force:

$$F_r = \mu \times N$$

$$= 1 \times 559.17\text{N}$$

$$= 559.17\text{N}$$

And from equation (1) $Fr = F_L$

Thus:

$F_L = 559.17N$

This is the force that the rack has to transmit to the tie rods and these to the steering arms to move the wheel. According to the conditions given, this will be the minimum force required to cause a turn of the wheels. But since the friction coefficient was rather large, we can assume that this is the force that will be applied by the driver during racing.

### **<u>4.7.2 Torque on Pinion</u>**

Now we can calculate the torque on the pinion. To calculate the torque we use the following equation:

$T = F \times r_{pinion}$

In our case we have a pinion with a diameter of 45 mm so:

$$T = 559.17 \times 45$$

$$= 25162.65 \text{ Nmm}$$

$$= 25.16 \text{ Nm}$$

This is the amount of torque required on the steering wheel to turn the pinion.

Finally the tangential force needed on the steering wheel by the driver to turn the wheels is calculated as below:

$$T = F \times R_{steeringwheel}$$

Where: $R_{steering\ wheel} = 200$ mm

Therefore: $F = \dfrac{T}{R_{\text{steering wheel}}}$

$F = 25.16 \div 0.2m$

$F = 125.8\ N$

This is the highest possible value that can be used to turn the wheels. This is because we took a rather high value of the friction coefficient as a safety factor. The most probable case is that the force to turn the steering wheel will be of a much lower value.

### 4.7.3 Steering Column Stresses

The steering column is located just after the steering wheel and is used to transmit the force from the steering wheel to the pinion. It undergoes a torsional force and thus a shear stress due to the torsion. Thus the column had to be designed with adequate material to design against the shear stress.

We can calculate if the material for the steering column is adequate for the torque that is transmitted by the force on the steering wheel. The steering column support torsion efforts so:

$$\tau_{\max} = \frac{T * r}{J}$$

Where:

$\tau_{max}$ =Shear stresses

T = torque in the steering column

R = radius of the column

J = inertia for hollow columns= $\frac{\pi}{32} \times$ Where: $d_o$ = 18mm; $d_i$ = 16 mm

Therefore;

$$J=\frac{\pi}{32}\times$$

$$=3.872\times10^{-9}\ m^4$$

$$\tau_{max}=\frac{25.16\times 0.018}{3.872\times10^{-9}} = 116.96\ \text{MPa}$$

-

## 4.8 Tie Rod Design

The tie rod connects the steering arm to the rack. The rack is under constant compression and tension. The most probable mode of failure is by buckling, therefore, the design against buckling is essential.

$N_{cr} = \frac{\pi^2 EI}{L^2}$ Where: $N_{cr}$ = Critical force

E = modulus of elasticity

I = area moment of inertia

L = unsupported length of the column

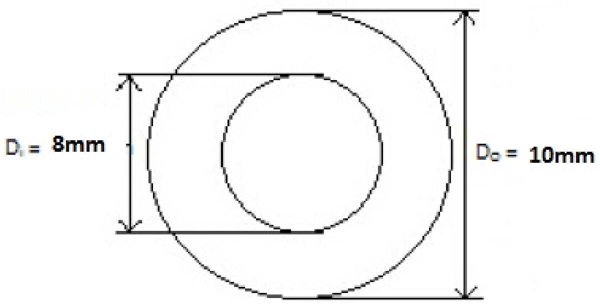


*Figure 27: Tie rod cross-sectional area [2, p. 46]*

The criteria to resist buckling is:

$$\frac{N_{Ed}}{N_{b,Rd}} \leq 1{,}0$$

$N_{Ed}$ is 559.17 N which is the force acting on the tie rod with a safety factor of 1.5.

Therefore with safety factor: 559.17 × 1.5 = 838.755 N

Thus, $N_{Ed}$= 838.755N

$N_{b,Rd}$ is the buckling resistance and is calculated through the formula:

$N_{b,Rd} = \frac{x*A*f_y}{\gamma_{M1}}$ Where: x = reduction factor

A = tie rod area

$f_y$ = yield strength of the mild steel

$\gamma_{M1}$ = partial factor = 1 (for mild steel)

To calculate the reduction factor, we first need to know the non-dimensional slenderness. Non-dimensional slenderness ($\lambda$) is calculated through the formula:

$\overline{\lambda} = \frac{L_{cr}}{i} * \frac{1}{93.9*\varepsilon}$ Where: $L_{cr}$= L = 150mm (since the tie rod as pinned ends)

$\varepsilon = \sqrt{\frac{235}{f_y}}$

i = radius of gyration

$\varepsilon = \sqrt{\frac{235}{f_y}}$ Where: $f_y$ = yield strength of the mild steel = 250MPa

$= \frac{235}{250} = 0.9695$

Radii of gyration:

$i = \sqrt{\frac{I}{A}}$ Where: I = area moment of inertia

A = tie rod area

Moment of Inertia:

$I = \frac{\pi}{64}$ $= 289.81\ mm^4$

Cross-section area:

$A = \frac{\pi}{4}$ □ $= 28.27\ mm^2$

From equation above:

Radii of gyration:

$$i = \frac{289.81\ mm^4}{28.27\ mm^2}$$

= 3.2 mm

Thus:

$\lambda = \frac{150}{3.2} \times \frac{1}{93.9 \times 0.9695} = 0.5149$

Since $\lambda \geq 0.5$, the formula for the reduction factor is given by:

$$x = \frac{1}{\varnothing + \varnothing^2 - \lambda^2}$$

Where: $\varnothing = 0.5 \times$

Thus:

$\varnothing = 0.5 \times$

Where: α = imperfection parameter α= 0.21 for rolled steel section

α= 0.49 for welded steel section

The tie rods will be welded, thus α= 0.49

$\varnothing = 0.5 \times$

= 0.6362

Hence:

$$x = \frac{1}{\varnothing + \ \varnothing^{2} - \lambda^{2}}$$

$$x = \frac{1}{0.6362 + \ 0.6362^{2} - 0.5149^{2}}$$

=0.9902

Thus the buckling resistance is given by:

$$N_{b,Rd} = \frac{x \times A \times f_{y}}{\gamma_{M1}}$$

$$N_{b,Rd} = \frac{0.9902 \times 28.27\,mm^{2} \times 250\ N/\,mm^{2}}{1.0}$$

= 6998.3N

Checking the criteria:

$$\frac{N_{Ed}}{N_{b,Rd}} = \frac{838.755N}{6998.3N} = 0.1199$$

The tie rod will resist buckling with the given dimensions since this value is less than 1.

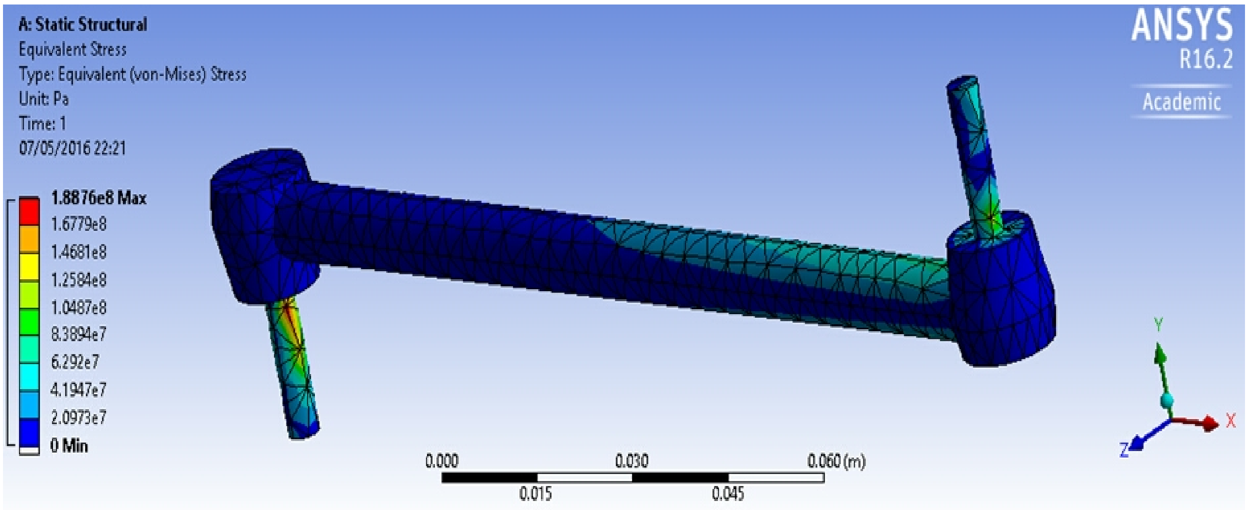


*Figure 28: Stresses on the tie rod*

## 4.9 Turning Radius

As stated in the problem statement, the purpose of the project was to reduce the turning radius, at the minimum cost possible. In this design, it was done through shifting of the centre of gravity of the vehicle just slightly behind the usual student race cars so that the vehicle can oversteer. This will enable the race car to turn at a lesser angle than usual.

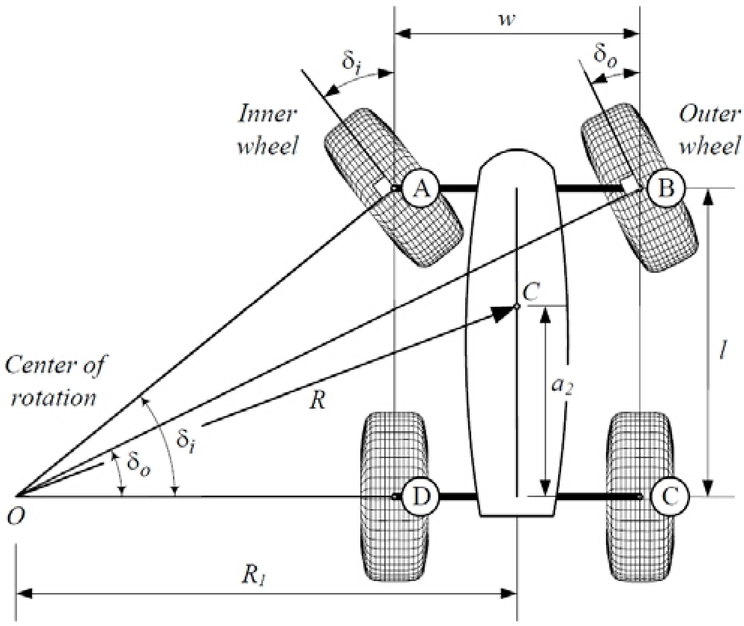

*Figure 29: Turning measurements of a vehicle [12]*

To be able to calculate the turning radius, we will need to know the distance of the front and rear axles from the centre of gravity. This distance will depend on the weight ratio which we gave as 30:70 for front axle weight to back axle weight. It is calculated using the formula:

$$W_f = \frac{W \times a2}{L}$$

Where:

$W_f$ = Load on front axle

W = Total weight of car = 300Kgs

L = Wheelbase = 1600mm

$$W_f = \frac{30}{100} \times 300 = 90\text{Kgs}$$

This is the weight handled by the front axle, which is found front the weight ration and the total weight of the vehicle.

Thus we now can find the distance of the centre of gravity from the rear wheel axle as:

$$a_2 = \frac{Wf \times L}{W}$$

$$= \frac{90 \times 1600}{300}$$

$$= 480\text{mm}$$

As stated in the parameters, the desired maximum inner wheel turning angle was chosen as $40^0$. To calculate the turning radius and the steering ratio, we will need to know the value of the outer wheel angle. This is calculated using the formula:

$$\cot\delta_o - \cot\delta_i = \frac{w}{L}$$

Where:

$\cot\delta = 1.569875$ therefore $\delta = 32.5^0$

$R=$ 0.5

R = 2.557m

## 4.10 Space Required for Turning

The space required for turning is the space between the two circles in which the whole vehicle fits without going out of the circle.

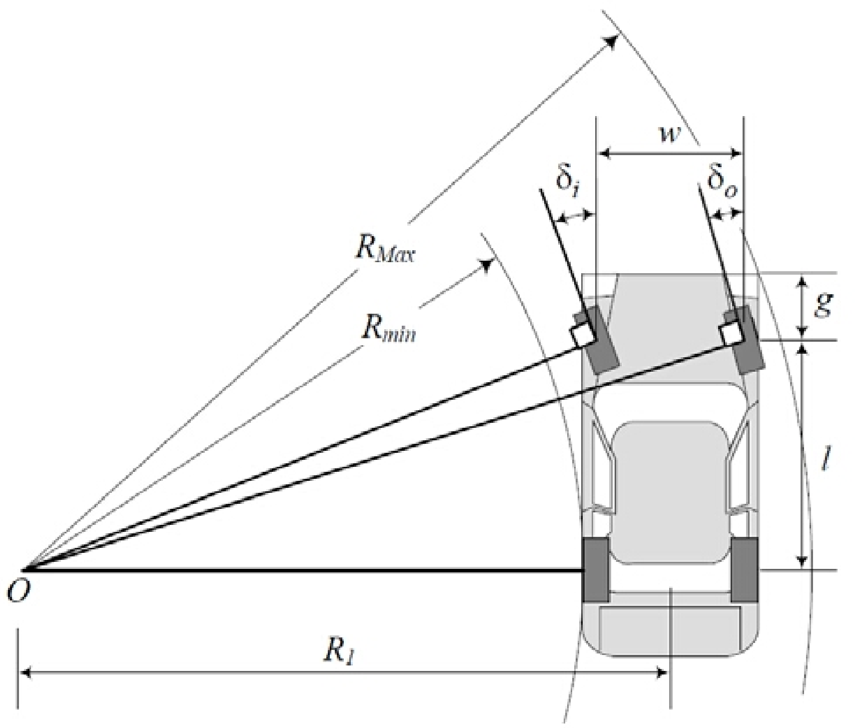


*Figure 30: Space required for turning diagram [12]*

Space required for turning is given by the formula:

$\Delta R = R_{Max} - R_{min}$

$R_{Max}$ and $R_{min}$ are obtained from the formulas below:

$R_{Max}= [(R_{min}+ w)^2+ (L + g)^2]^{0.5}$ where g = distance from front axle to nose

= 600mm

$$R_{min}= R_1 - (\frac{w}{2}) = (\frac{L}{tan\delta i}) = (\frac{L}{tan\delta o}) - w$$

Therefore:

$$R_{min}= (\frac{L}{tan\delta i}) = (\frac{1.6}{\tan 40}) = 1.9068m$$

$R_{Max}= [(R_{min}+ w)^2+ (L + g)^2]^{0.5}$

$= [(1.9068 + 1.210)^2 + (1.6 + 0.6)^2]^{0.5}$

$= 3.815m$

$\Delta R = R_{Max} - R_{min}$

$= 3.815 - 1.9068$

$= 1.9082m$

This is thus the width of the space that will be covered during turning.

## 4.11 Steering Ratio

Steering ratio is the ratio of the number of degrees turned at the steering wheel vs. the number of degrees the front wheels are deflected. Steering ratio gives mechanical advantage to the driver, allowing them to turn the tyres with the weight of the whole car sitting on them, but more importantly, it means the steering wheels doesn't need to be turned a ridiculous number of times to get the wheels to move.

For this design, the pinion is set at the middle and the pinion turns once ($360^0$) for lock to lock movement of the rack. Thus, if the inner wheel turns to its maximum ($40^0$), the pinion would have moved from the middle of the rack, to one far end of the pinion. The pinion will have moved through $180^0$ degrees. The steering ratio is thus given by:

$$\text{Steering Ratio} = \frac{180}{\frac{\delta_o + \delta_i}{2}}$$

$$= \frac{180}{\left(\frac{27.1735 + 40}{2}\right)}$$

$$= 5.35$$

# CHAPTER FIVE

# EFFECTS OF MODIFICATIONS ON THE TURNING RADIUS

## 5.1 Introduction

From the design made. We need to find out whether the modifications made were effective in reducing the turning radius of the formula one student race car. This was accomplished graphically through varying the various properties that influence the turning radius that we modified and finding out its influence on the turning radius.

### 5.1.1 Turning radius vs. Weight on front wheel axle

In our design, we decided to shift the weight of the car slightly to the rear wheel axle. This led to the shifting of the centre of gravity slightly to the back. To find out whether this helped reduce the turning radius, we represented it as below.

| FRONT TO REAR RATIO | FRACTION OF WEIGHT ON FRONT WHEELS | DISTANCE OF C.G FROM REAR AXLE ($a_2$) in mm | TURNING RADIUS (mm) | TURNING RADIUS (m) |
|---|---|---|---|---|
| 30:70 | 0.3 | 480 | 2557.252 | 2.557252 |
| 35:65 | 0.35 | 560 | 2573.468 | 2.573468 |
| 40:60 | 0.4 | 640 | 2592.053 | 2.592053 |
| 45:55 | 0.45 | 720 | 2612.956 | 2.612956 |

| | | | | |
|---|---|---|---|---|
| 50:50 | 0.5 | 800 | 2636.122 | 2.636122 |
| 55:45 | 0.55 | 880 | 2661.492 | 2.661492 |
| 60:40 | 0.6 | 960 | 2689.003 | 2.689003 |
| 65:35 | 0.65 | 1040 | 2718.591 | 2.718591 |
| 70:30 | 0.7 | 1120 | 2750.189 | 2.750189 |

Where C.G is the centre of gravity

*Table 5: Front to rear ratio vs Turning radius*

The values of the turning radius were plotted against the values of the fraction of weight on the front wheels as follows.

From the graph, it can be seen that as the weight of the car is shifted to the front wheels, the turning radius of the car increases. Thus it would be more desirable for the weight to be a little further back so as to reduce the turning radius. This proves that shifting the weight slightly to the back does reduce the turning radius of the formula one student race car.

#### 5.1.2 Turning radius vs. Inner wheel angle

Another modification that was made was to slightly increase the turning angle so as to reduce the turning radius. This was analysed as follows:

| INNER WHEEL ANGLE | OUTER WHEEL ANGLE | cot $\delta$ | TURNING RADIUS |
|---|---|---|---|

| | | | |
|---|---|---|---|
| (degrees) | (degrees) | | (mm) |
| 40 | 27.17352 | 1.569879 | 2557.258 |
| 38 | 26.15625 | 1.658067 | 2695.981 |
| 36 | 25.12208 | 1.754507 | 2847.953 |
| 34 | 24.06868 | 1.860686 | 3015.545 |
| 32 | 22.9936 | 1.97846 | 3201.72 |
| 30 | 21.89424 | 2.110176 | 3410.231 |
| 28 | 20.76784 | 2.258851 | 3645.898 |
| 26 | 19.61143 | 2.428429 | 3915.023 |
| 24 | 18.42186 | 2.624162 | 4226.007 |

| | | | |
|---|---|---|---|
| 22 | 17.1957 | 2.853212 | 4590.304 |
| 20 | 15.92929 | 3.125602 | 5023.947 |

*Table 6: Inner wheel angle vs Turning radius*

This was represented graphically as:

From the graph, it can be noted that the turning radius reduces with increase in the inner wheel angle. This modification did actually help reduce the turning radius of the student race car. From most of the road cars, the maximum inner wheel angle is usually from around 28 degrees to 35 degrees. For our design, we used an inner wheel angle of 40 degrees which ultimately reduced the turning radius by about 1metre.

Through these modification, some other aspects of the car will be affected which will ultimately cause an effect on the turning radius. These aspects were analysed and presented graphically as follows:

### 5.1.3 Space required for turning vs. Inner wheel angle

The amount of space required for turning is usually influenced by the inner wheel angle. This was analysed as below:

| INNER WHEEL ANGLE (degrees) | $R_{min}$ (mm) | $R_{max}$ (mm) | SPACE FOR TURNING (mm) | SPACE FOR TURNING (m) |
|---|---|---|---|---|
| 40 | 1906.806 | 3815.033 | 1908.227 | 1.908227 |
| 38 | 2047.907 | 3931.152 | 1883.245 | 1.883245 |
| 36 | 2202.211 | 4059.949 | 1857.738 | 1.857738 |
| 34 | 2372.098 | 4203.739 | 1831.642 | 1.831642 |
| 32 | 2560.535 | 4365.425 | 1804.89 | 1.80489 |
| 30 | 2771.281 | 4548.692 | 1777.411 | 1.777411 |
| 28 | 3009.162 | 4758.291 | 1749.128 | 1.749128 |
| 26 | 3280.486 | 5000.447 | 1719.96 | 1.71996 |
| 24 | 3593.659 | 5283.478 | 1689.819 | 1.689819 |
| 22 | 3960.139 | 5618.749 | 1658.61 | 1.65861 |
| 20 | 4395.964 | 6022.195 | 1626.231 | 1.626231 |

*Table 7: Inner wheel angle vs Space required for turning*

This was presented graphically as:

It can be seen that an increase in the inner wheel angle will also increase the space required for turning. This is attributed to the slightly more horizontal position that the race car will take to incorporate for the increased inner wheel angle. This will further increase the oversteer effect on the race car when making a turn.

#### <u>5.1.4 Steering Ratio vs. Inner wheel angle</u>

The inner wheel angle will also influence the steering ratio. Thus an analysis had to be done to find out in what manner the steering ratio will be affected.

| INNER WHEEL ANGLE (degrees) | OUTER WHEEL ANGLE (degrees) | S<br>T<br>E<br>E |
|---|---|---|

| | | |
|---|---|---|
| | | RINGRATIO |
| 40 | 27.17352 | 5.3592555 |
| 38 | 26.15625 | 5.61113 |
| 36 | 25.12208 | 5.8898852 |

| | | |
|---|---|---|
| 34 | 24.06868 | 6.1995556 |
| 32 | 22.9936 | 6.546217 |
| 30 | 21.89424 | 6.937186 |
| 28 | 20.76784 | 7.38191 |

| | | |
|---|---|---|
| | | 4 |
| 26 | 19.61143 | 7.8927558 |
| 24 | 18.42186 | 8.4861192 |
| 22 | 17.1957 | 9.1846881 |
| 20 | 15.92929 | 10.019 |

| | | |
|---|---|---|
| | | 6<br>8 |

*Table 8: Inner wheel angle vs Steering ratio*

This is represented graphically as follows:

From the graph, it is noted that an increase in wheel angle causes a decrease in steering ratio. This is a desirable effect since the sensitivity of the steering system is increased. The vehicle handling will be much better with a decreased steering ratio.

# CHAPTER SIX

# COMPUTER AIDED DESIGN OF STEERING SYSTEM

## 6.1Introduction

Various software were used in the design of our steering system. These software were primarily used to visualize, manage and test the design without having to make the actual component or system.

A Computer Aided Design (CAD) and Finite Element Analysis (FEA) software were used in the design and testing of the steering system. The Computer Aided Design conceptualized the overall design of the steering system where the components and the working principle and movements of the steering system were projected. When the CAD design was complete, the finite element analysis tests followed. The finite element stress analysis, strain analysis and elastic deformation were then applied on the components. These tests were applied to the components that have an applied force which can cause elastic deformation or buckling.

## 6.2 Computer Aided Design

Computer aided design helped to aid in the creation, modification, analysis and optimization of the design. CAD software is used to increase productivity of the design, improve quality of design and communications through documentation and to create a database for manufacturing. The design process is an iterative procedure as shown in the figure below.

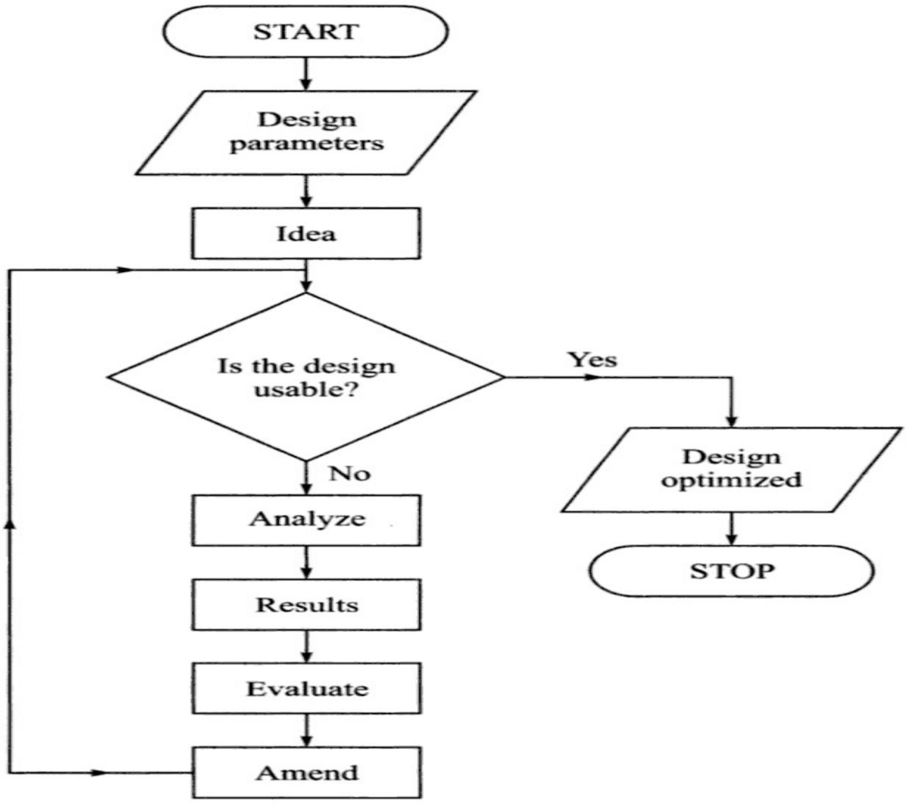


*Figure 31: Computer Aided Design process*

The mechanical engineering CAD software used is the SOLIDWORKS software. The figure below shows the conceptual design:-

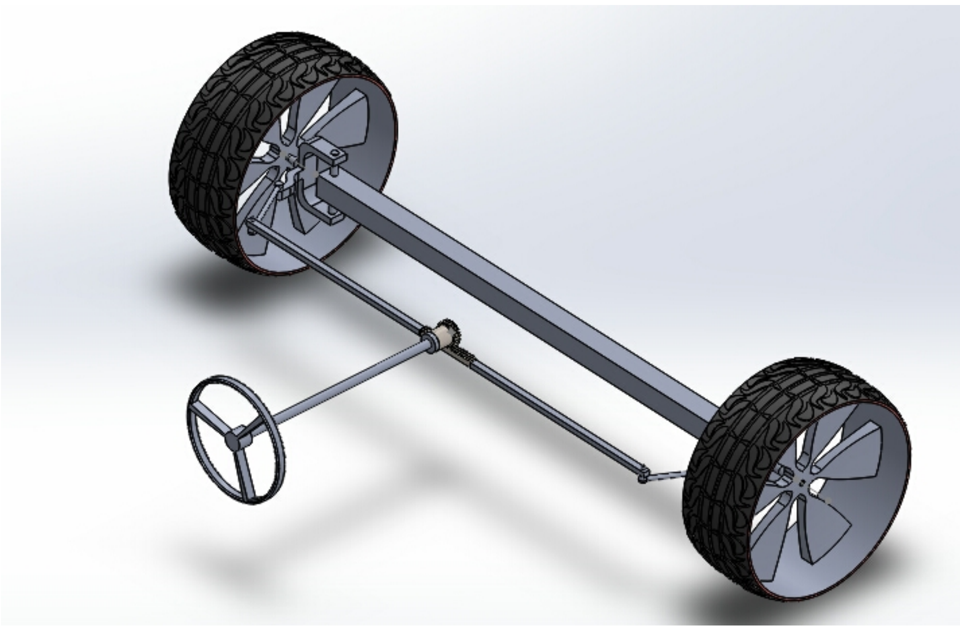

*Figure 32: Isometric view of the simulated steering system*

The CAD design shown above consists of the original design of a rack and pinion steering that can be used in a Formula One student race car. The system uses the Ackermann principle which accounts for the turning radius and wheel angle. The dimensions used to make the computer model was obtained from the calculations in chapter Four.

## 6.3 Finite Element Analysis (FEA)

Finite element analysis (FEA) is a computerized method for predicting how a product reacts to real world situation like applying forces, vibration, heat, fluid flow, and other physical effects.

FEA subdivides a large problem into smaller, simpler, parts, called finite elements. The simple equations that model these finite elements are then assembled into a larger system of equations that models the entire problem.

In the process of analysis, the parts are broken down to three main parts i.e

- Elements
- Nodes
- Meshes

In stress analysis the equations are equilibrium equations of the nodes. There may be several hundred or several thousands of these equations hence computer implementation is mandatory.

The finite element analysis was conducted using the ANSYS Mechanical software program. The software is a finite element analysis tool used for structural analysis, including linear, nonlinear and dynamic studies. The computer simulation provides finite elements to model behaviour, and supports material models and equation solvers for a wide range of mechanical design problems.

The software was used to test for any structural deformations like buckling, directional deformation, equivalent stresses and equivalent elastic strain.

### 6.3.1 Test Procedure

The following are the necessary steps used to test using ANSYS software:

- Meshing the components
- Applying forces and pressures.
- Fixing non-moving faces.
- Applying magnitude of forces.

The components to be tested had already been designed and were easily imported from SOLIDWORKS CAD. The components put under test are as follows:-

- Steering arm
- Tie rod

#### 6.3.1.1 Test and analysis of the steering arm

This component was being tested for equivalent stress, equivalent elastic strain and directional deformation. A force of 559.1N is applied on the face where the steering arm is connected to the caster. The result is shown below:-

| | |
|---|---|
| Magnitude | 559.17N |
| Nodes | 11541 |
| Elements | 6138 |

*Table 9: Steering arm statistics*

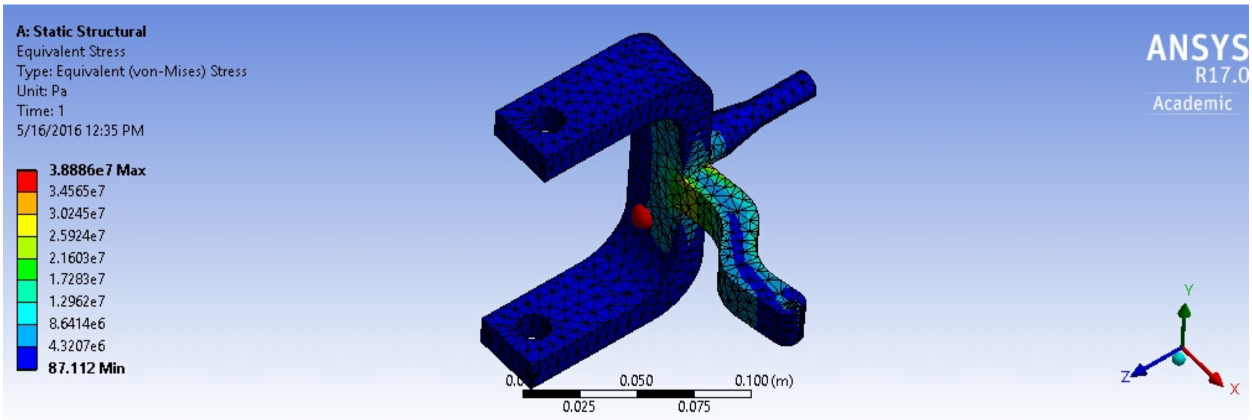

*Figure 33: Equivalent (von-mises) stress*

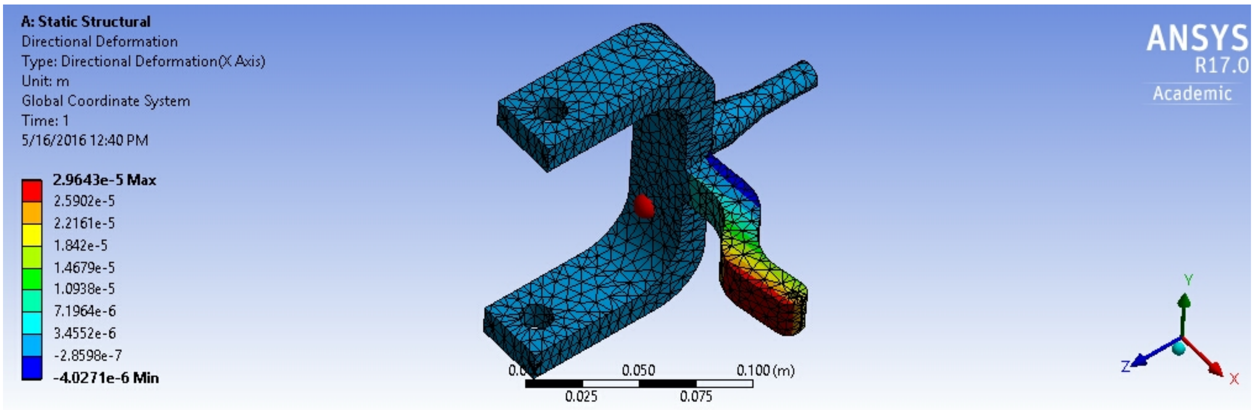


*Figure 34: Directional Deformation*

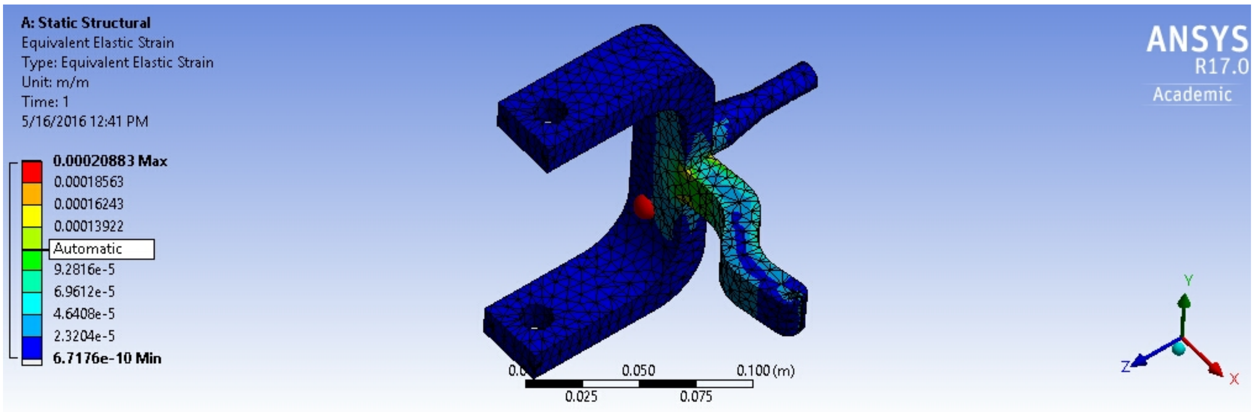


*Figure 35: Equivalent elastic strain across the steering arm*

#### 6.3.1.2 Test Results of the steering arm

| | Equivalent (von-Mises) Stress | Equivalent Elastic Strain | Directional Deformation |
|---|---|---|---|
| Minimum | 87.112 Pa | 6.7176e-010 m/m | -4.0271e-006 m |
| Maximum | 3.8886e+007 Pa | 2.0883e-004 m/m | 2.9643e-005 m |

*Table 10: Steering arm results*

The above table shows the results of the component after an applied force is subjected to a point on the steering arm. The maximum stress is observed to be at the yellow and orange sections on figure 33. The maximum stress was found to be 38.887 Mpa which according to our calculations and material used is allowable.

From the directional deformation test in figure 34, maximum elastic deformation was at the tip of the contact point coloured red. The displacement was by 0.029643mm which is a very small deflection which is allowable.

### 6.3.1.3 Test and analysis of the tie rod

The tie rod was tested for buckling since the forces are acting axially by compression. The component was designed to withstand a force of 559.1N. Theoretical calculations show that the tie rod can withstand the force with a safety factor of 2. The figures below show the component after the load is applied:-

| | |
|---|---|
| Magnitude | 559.17N |
| Nodes | 4473 |
| Elements | 2116 |

*Table 11: Tie rod test statistics*

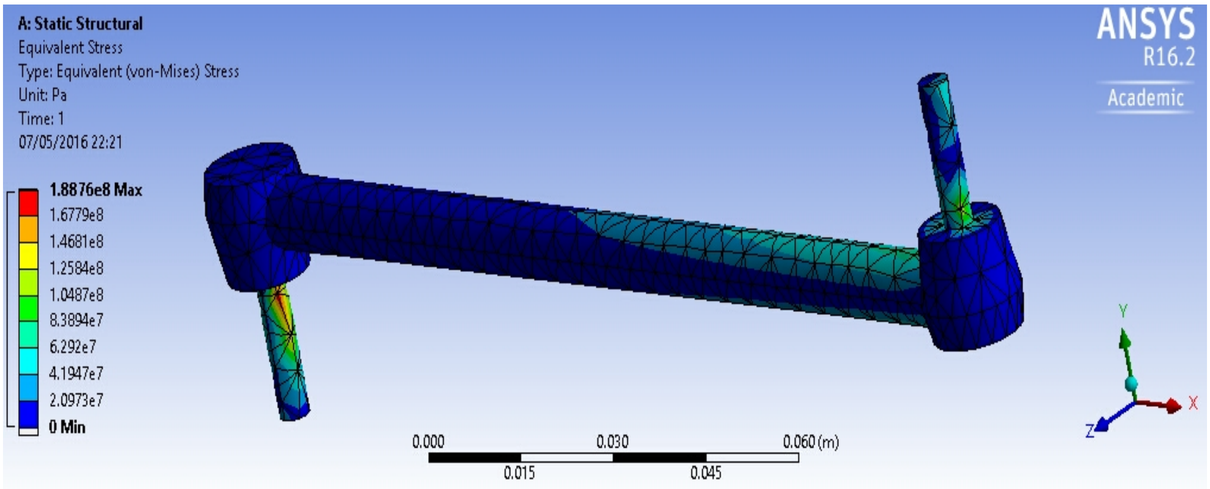


*Figure 36: Equivalent (von-mises) Stress across the tie rod*

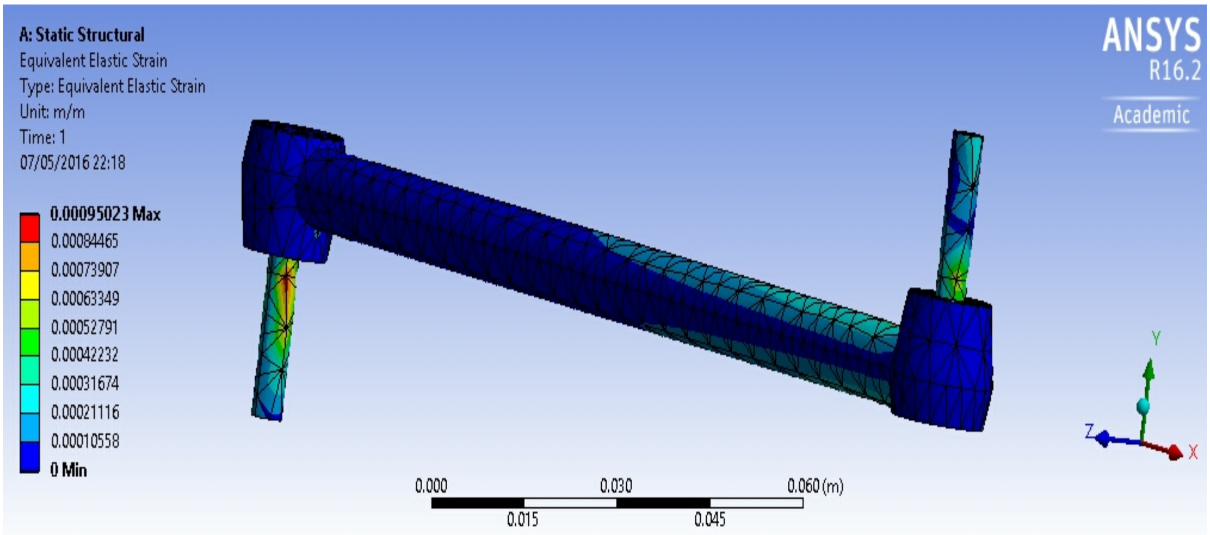


*Figure 37: Equivalent Elastic Strain across the tie rod*

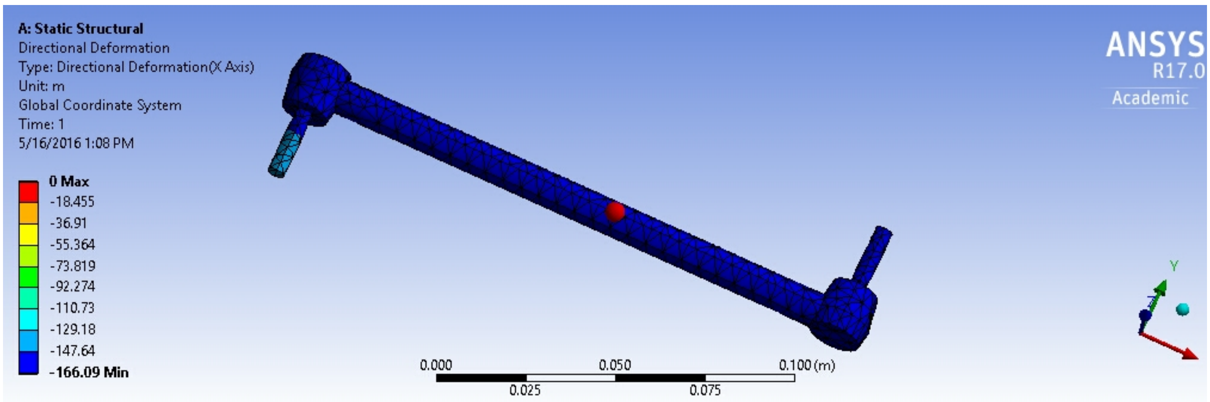

*Figure 38: Directional deformation of the tie rod*

**6.3.1.4 Test results of the tie rod**

| | Equivalent (von-Mises) Stress | Equivalent Elastic Strain | Directional Deformation |
|---|---|---|---|
| Minimum | 2.0973+007 Pa | 0.00010558 | -18.455 |
| Maximum | 1.8876+008 Pa | 0.00095023 | -166.09 |

*Table 12: Tie rod test results*

The above table, results are shown from the simulation of the tie rod that is most likely to fail by buckling. Maximum Stresses on the tie rod is found to be at the pin contact to the tie rod which was 188.78 Mpa, half of the allowable maximum stress of 250 Mpa. The Maximum Elastic was 0.00095023 which is also safe.

# CHAPTER SEVEN

## DISCUSSION, CONCLUSION AND RECOMMENDATIONS

### 7.1 Discussion

The aim of the project was to develop the steering system of a student formula one race car with the objective of solving the problem of reducing the turning radius taken round a corner. This was done by the selection of an adequate steering system with the correct principles to aid in reduction of the turning radius at a reduced cost.

The rack and pinion sizes was calculated and was found to be 140mm and 45mm diameter respectively. One revolution of the pinion will make a complete displacement of the rack from lock to lock. This enabled the driver to make sensitive movement of the steering wheel to make a turn, which becomes an advantage on the race track where time is vital.

The material chosen for the steering system had to be tough enough to withstand the forces and stresses that occur in the steering system and at the same time, it had to be light and cheap. Through the material selection process, we were able to eliminate the different kinds of material and we finally remained with mild steel. We thus had to do further calculation to be able to find out whether the mild steel will be able to withstand the forces and stresses in the steering system.

Therefore, we had to calculate the amount of force that has to be used so as to move the tyres. Through calculation, this force was found to be 559.17N, but the driver only had to apply a force of 125.8N to move the tyres. The steering column, which is just right below the steering wheel would undergo some form of torque. This meant there would be shear stress on the column during turning of the steering wheel. The shear stress was calculated as 116.96MPa, which was far below the yield strength and the ultimate tensile strength of mild steel which is 250MPa and 455MPa respectively. This therefore meant that the material selected would be able to hold the forces applied to this system. The tie rods were thus designed using the mild steel and since the most probable mode of failure would be through buckling where a buckling test was carried out. The material was found to be stable under the buckling conditions of the tie rods.

Since all the steering system design parameters had been calculated, we needed an effective way to reduce the turning radius of the student race car. This was done by shifting the centre of gravity a little bit more to the rear axle so that the vehicle can over-steer just a little bit more. Over-steering of the race car leads to reduction of the turning radius, but too much over-steer will lead to spinning out of the car. This is the reason why the rack and pinion system had to be sensitive. This sensitivity of the steering system ensured the driver could keep the race car under control when he/she feels the car will spin out. Another way in which we helped reduce the turning radius was through slightly increasing the inner wheel turning angle of the race car. Thus the turning radius was found to be 2.557m which was rather low as compared to many road cars which have a turning radius of more than 6m. The biggest contributing factor to the reduction of the turning radius was the increase of the inner wheel angle, while the least was the shifting of the centre of gravity slightly to the back. This thus explained the reason why most formula one race cars use a weight distribution ratio of 40:60.

A Finite Element Analysis (ANSYS simulation) of the steering arm and tie rod was carried out to ensure the design will not fail under stresses. The maximum stresses were found to be 38.887 Mpa and 188.76 Mpa for the steering arm and tie rod respectively. These values are way below the yield strength and the ultimate tensile strength of mild steel which is 250MPa and 455MPa respectively.

## 7.2 Conclusion

The aim of the project was to reduce the turning radius of the student race car. This was achieved through shifting of the weight of the race car slightly to the rear axle and also by increasing the turning angle. The modifications made to the steering system were analysed graphically, thus getting the following results:

- The turning radius was found to decrease as you moved the centre of gravity towards the rear axle.

- The turning radius decreased if you increased the inner wheel angle of the tyres.
- As you increase the inner wheel angle, the space that the race car would need to turn would also increase.
- The steering ratio would increase with increase in the inner wheel angle.

This proved that the modifications made did reduce the turning radius of the steering system, but also increased the space required for turning of the student race car. The space required for turning may be further increased by the over-steer effect that the race car will have. Despite all this, this effect can be reduced through the higher steering ratio that the vehicle will attain. This was further aided by the fact that both the space required for turning and the steering ratio increase.

## 7.3 Recommendations

From the data collected in the project, the modifications made reduced the turning radius. Despite this, there are other aspects of the student race that have to be enhanced so that the designed steering system can work effectively and efficiently. These parts include the wheel and tyre system and also the suspension system which are to be designed by other students who will take part in this continuing project.

For the students who will take part in the wheel and tyre system, they would need to design a tyre system that will be able to withstand the impact of the lateral force on the steering system. Due to the oversteering that will take place in the student race car, the rear tyres should be able to handle the forces impacted on them to avoid spinning out of the race car. Thus, the rear wheels should have enough traction to avoid spinning out. This will enable the driver to be able to go around corners with minimal problems.

As for the students who will work on the suspension, they will need to design a system that will be able to handle the lateral force exerted on the wheels and also design the suspension to be able to reduce the bump steer. The lateral forces may cause discomfort to the driver and difficulty in handling. So to enable the cornering to be smooth, the suspension needs to be enhanced. Bump steer will also be a factor during racing since there is usually presence of rubber on the race track due to the wearing if tyres during tyres. This rubber may cause the ride to be bumpy and thus cause deflection of the steering system during driving. Thus the suspension team needs to design against bump steer.

Special emphasis should be put in the physical implementation of the Formula 1 Student Race Car project as it will be able help students to roughly have an introduction of what to expect in the industrial setting. With the continued economic growth of the country, the level of education within The University is also expect to follow suit so as to achieve vision 2030. A small set up of the student race car competition can be set up within the country, with various universities competing among themselves. This can further grow to other African countries with time.

All this can be achieved through the help of the government and various manufacturing and assembly companies, with specific emphasis on the industries that deal with vehicles. With the help of these external sources, The University and engineering students, this vision is very much within our grasp.

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# APPENDICES

## 8.1 APPENDIX 1: Tables of Standards Used in Transmission Calculations

| TOUGHNESS | BRITTLENESS | DUCTILITY | MALLEABILITY | CORROSION RESISTANCE |
|---|---|---|---|---|
| Copper | White Cast Iron | Gold | Gold | Gold |
| Nickel | Gray Cast Iron | Silver | Silver | Platinum |
| Iron | Hardened Steel | Platinum | Aluminum | Silver |
| Magnesium | Bismuth | Iron | Copper | Mercury |
| Zinc | Manganese | Nickel | Tin | Copper |
| Aluminum | Bronzes | Copper | Lead | Lead |
| Lead | Aluminum | Aluminum | Zinc | Tin |
| Tin | Brass | Tungsten | Iron | Nickel |
| Cobalt | Structural Steels | Zinc | | Iron |
| Bismuth | Zinc | Tin | | Zinc |
| | Monel | Lead | | Magnesium |
| | Tin | | | Aluminum |
| | Copper | | | |
| | Iron | | | |
| * Metals/alloys are ranked in descending order of having the property named in the column heading | | | | |

*Table 13: Properties of different metals*

| Diametral Pitch | Circular Pitch (Inches) | Thickness of Tooth on Pitch Line (Inches) | Depth to be Cut in Gear (Inches) (Hobbed Gears) | Addendum (Inches) |
|---|---|---|---|---|
| 3 | 1.0472 | .5236 | .7190 | .3333 |
| 4 | .7854 | .3927 | .5393 | .2500 |
| 5 | .6283 | .3142 | .4314 | .2000 |
| 6 | .5236 | .2618 | .3565 | .1667 |
| 8 | .3927 | .1963 | .2696 | .1250 |
| 10 | .3142 | .1571 | .2157 | .1000 |
| 12 | .2618 | .1309 | .1798 | .0833 |
| 16 | .1963 | .0982 | .1348 | .0625 |
| 20 | .1571 | .0785 | .1120 | .0500 |
| 24 | .1309 | .0654 | .0937 | .0417 |
| 32 | .0982 | .0491 | .0708 | .0312 |
| 48 | .0654 | .0327 | .0478 | .0208 |
| 64 | .0491 | .0245 | .0364 | .0156 |

*Table 14: Gear and teeth dimensions*

*TABLE 4: EFFECTIVE LENGTH OF PRISMATIC COMPRESSION MEMBERS*

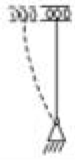
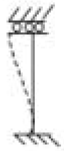
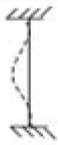
| Boundary Conditions | | | | Schematic representation | Effective Length |
|---|---|---|---|---|---|
| At one end | | At the other end | | | |
| Translation | Rotation | Translation | Rotation | | |
| Restrained | Restrained | Free | Free |  | 2.0L |
| Free | Restrained | Restrained | Free |  | |
| Restrained | Free | Restrained | Free |  | 1.0L |
| Restrained | Restrained | Free | Restrained |  | 1.2L |
| Restrained | Restrained | Restrained | Free |  | 0.8L |
| Restrained | Restrained | Restrained | Restrained |  | 0.65 L |

*Table 15: Effective length of prismatic compression members*

| **Material** | | **(s) Lb. per Sq. In.** |
|---|---|---|
| Plastic | | 5000 |
| Bronze | | 10000 |
| Cast Iron | | 12000 |
| Steel | .20 Carbon (Untreated) | 20000 |
| | .20 Carbon (Case-hardened) | 25000 |
| | .40 Carbon (Untreated) | 25000 |
| | .40 Carbon (Heat-treated) | 30000 |
| | .40 C. Alloy (Heat-treated) | 40000 |

*Table 16: Values of safe static stress*

| Mechanical Properties of Various Materials at Room Temperature | | | | | |
|---|---|---|---|---|---|
| **Metals (wrought)** | ***E*** **(GPa)** | ***Y*** **(MPa)** | **UTS (MPa)** | **Elongation in 50 mm (%)** | **Poisson's ratio ($\nu$)** |
| Aluminum and its alloys | 69–79 | 35–550 | 90–600 | 45–4 | 0.31–0.34 |
| Copper and its alloys | 105–150 | 76–1100 | 140–1310 | 65–3 | 0.33–0.35 |
| Lead and its alloys | 14 | 14 | 20–55 | 50–9 | 0.43 |
| Magnesium and its alloys | 41–45 | 130–305 | 240–380 | 21–5 | 0.29–0.35 |
| Molybdenum and its alloys | 330–360 | 80–2070 | 90–2340 | 40–30 | 0.32 |
| Nickel and its alloys | 180–214 | 105–1200 | 345–1450 | 60–5 | 0.31 |
| Steels | 190–210 | 205–1725 | 415–1750 | 65–2 | 0.28–0.33 |
| Titanium and its alloys | 80–130 | 344–1380 | 415–1450 | 25–7 | 0.31–0.34 |
| Tungsten and its alloys | 350–400 | 550–690 | 620–760 | 0 | 0.27 |
| Zinc and its alloys | 50 | 25–180 | 240–550 | 65–5 | 0.27 |

*Table 17: Mechanical properties of various materials at room temperature*

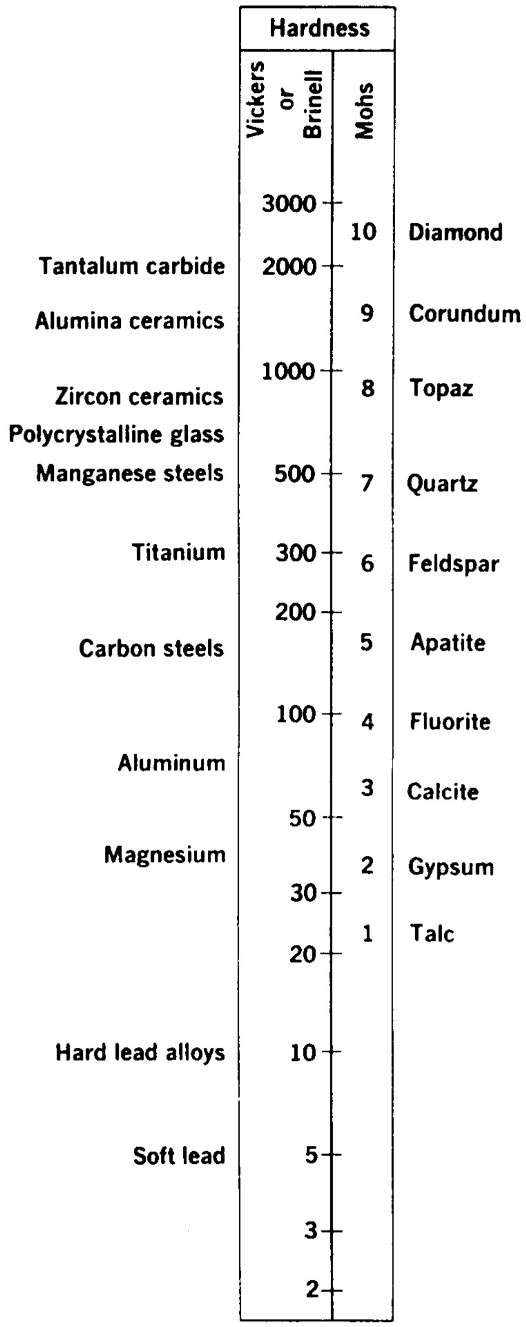



*Figure 39: Approximate relative hardness of metals and ceramics for Mohs scale and indention scale*

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## 8.2 APPENDIX 2: SOLIDWORKS Drawings of Complete Steering System

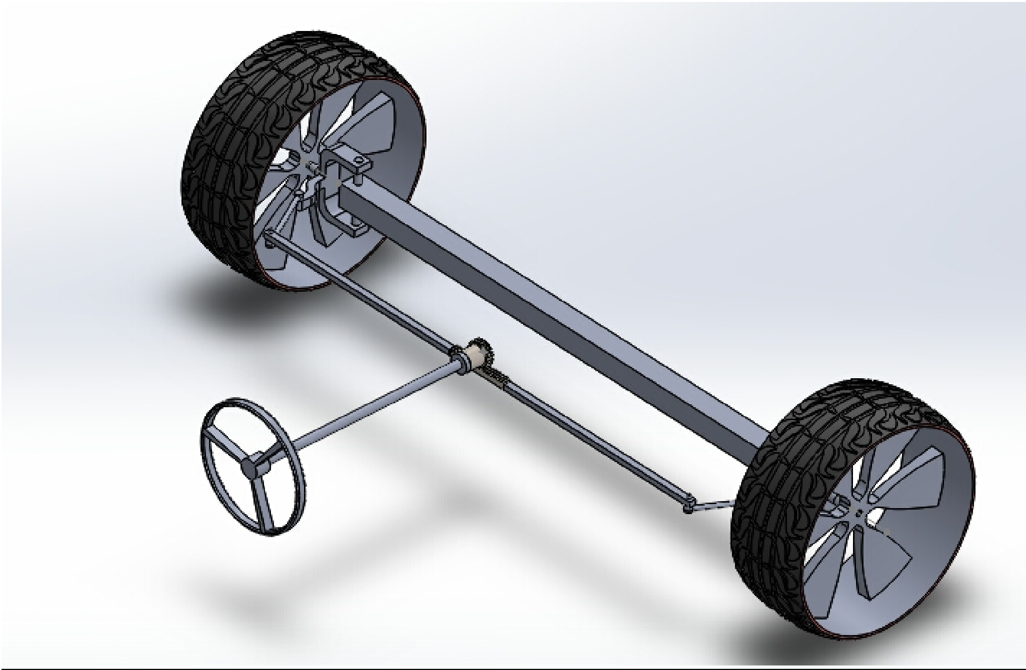

*Figure 40: 3D view of the steering system*

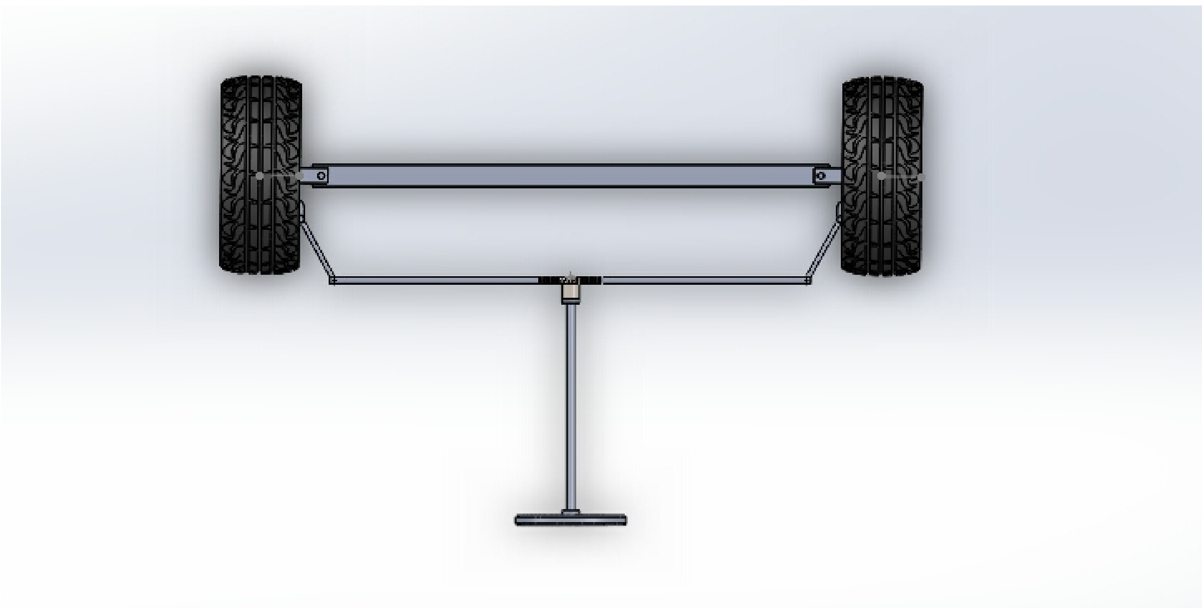

*Figure 41: Plan view of the steering system_*

<u>8.3 APPENDIX 3: Drawings and dimensions of the steering arm and tierods</u>

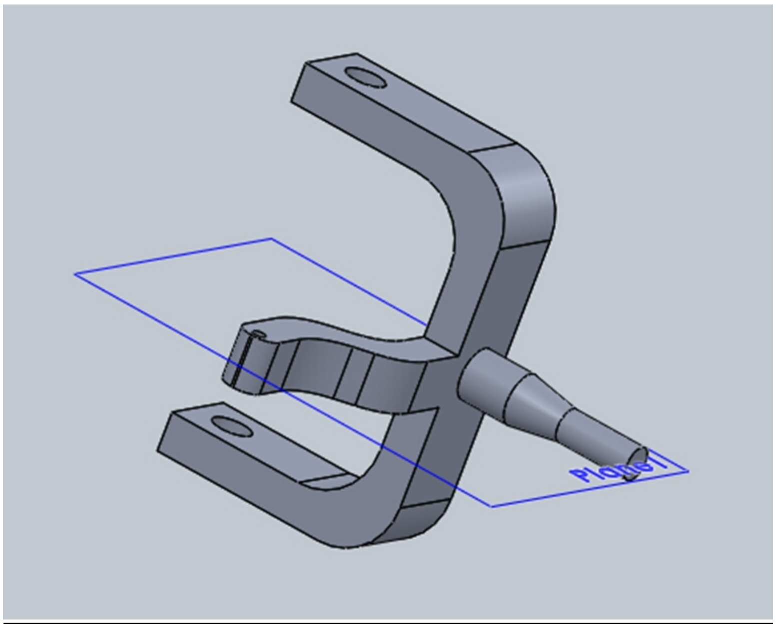


*Figure 42: 3D view of the steering arm*

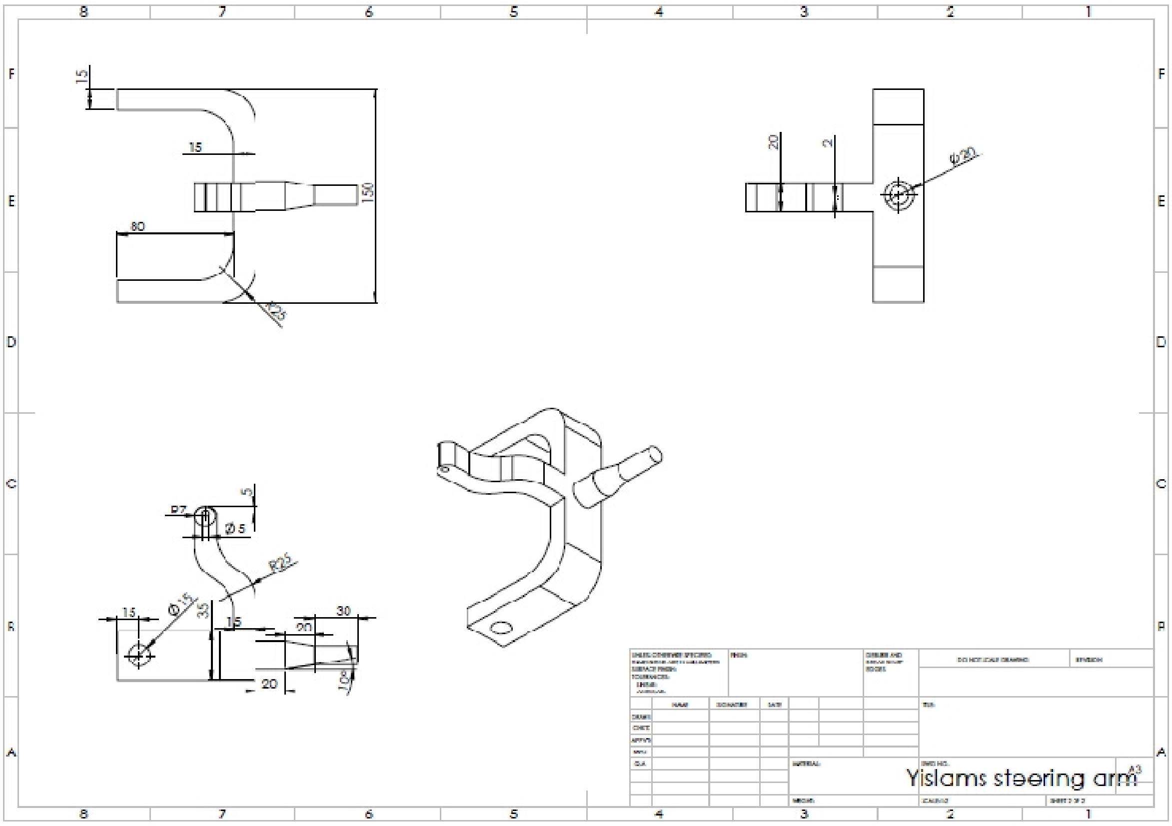


*Figure 43: Drawing of the steering arm*

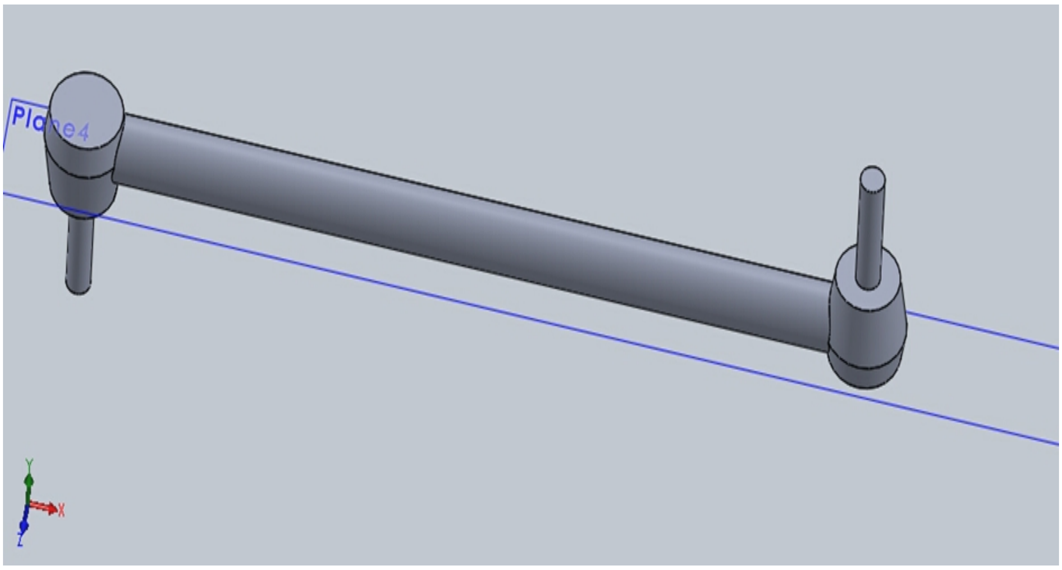

*Figure 44: 3D view of the tie rod*

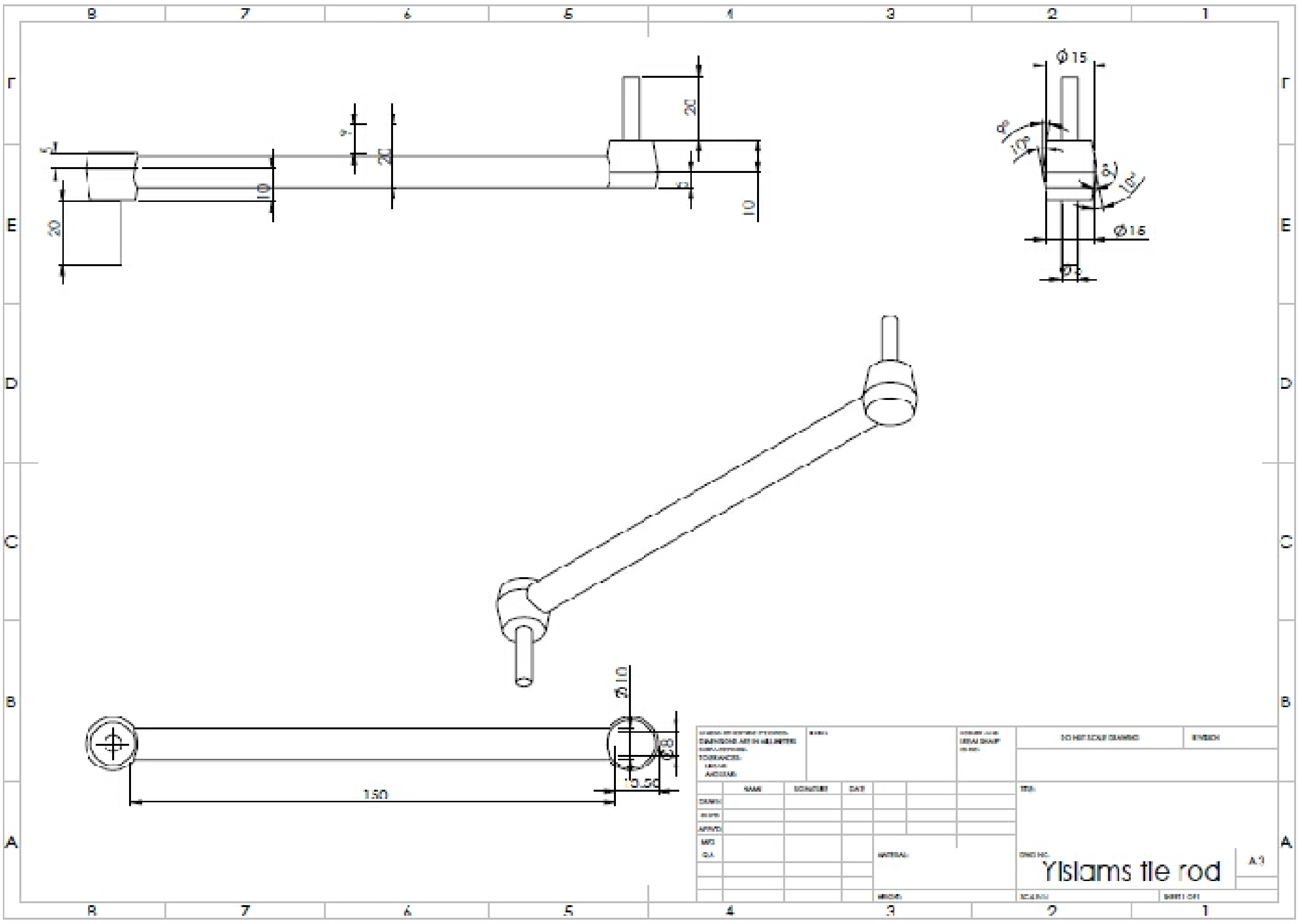


*Figure 45: Drawing of the tie rod_*

# 6.4 APPENDIX 4: Elastic strain and deformation of the steering arm and tie rod

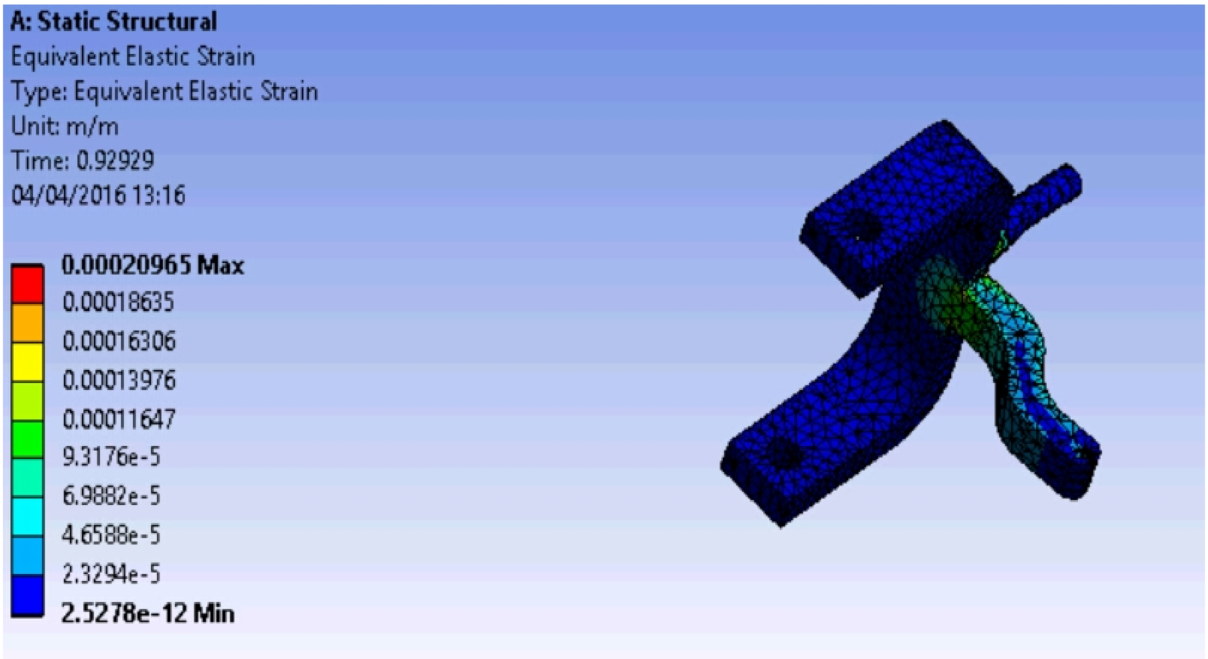


*Figure 46: Equivalent elastic strain across the steering arm*

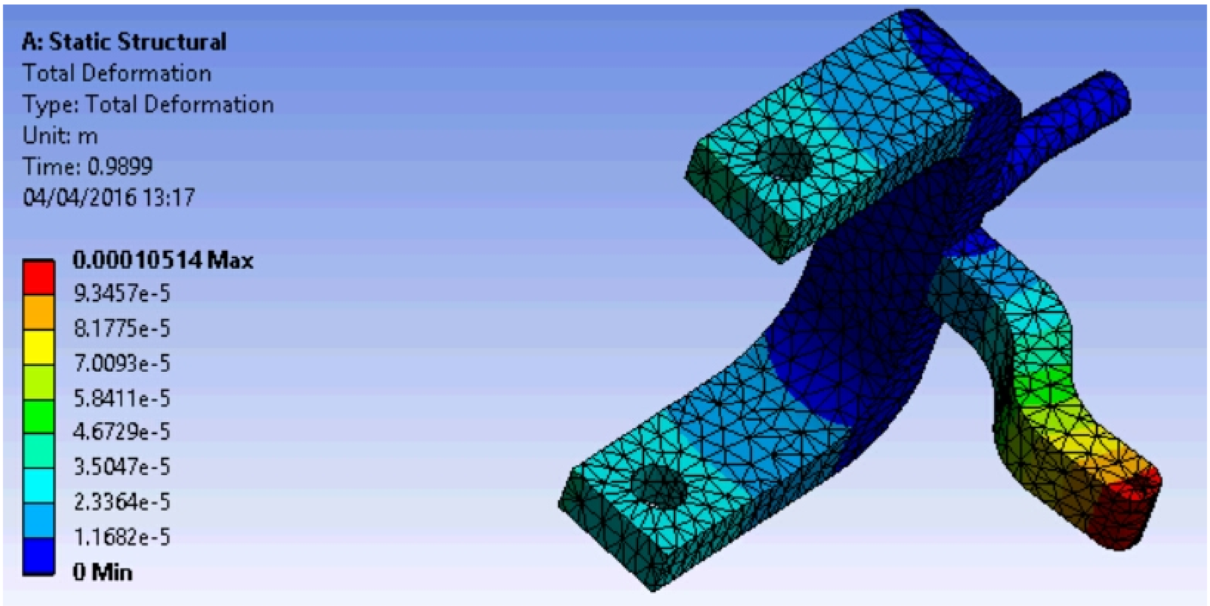


*Figure 47: Total deformation across the steering arm*

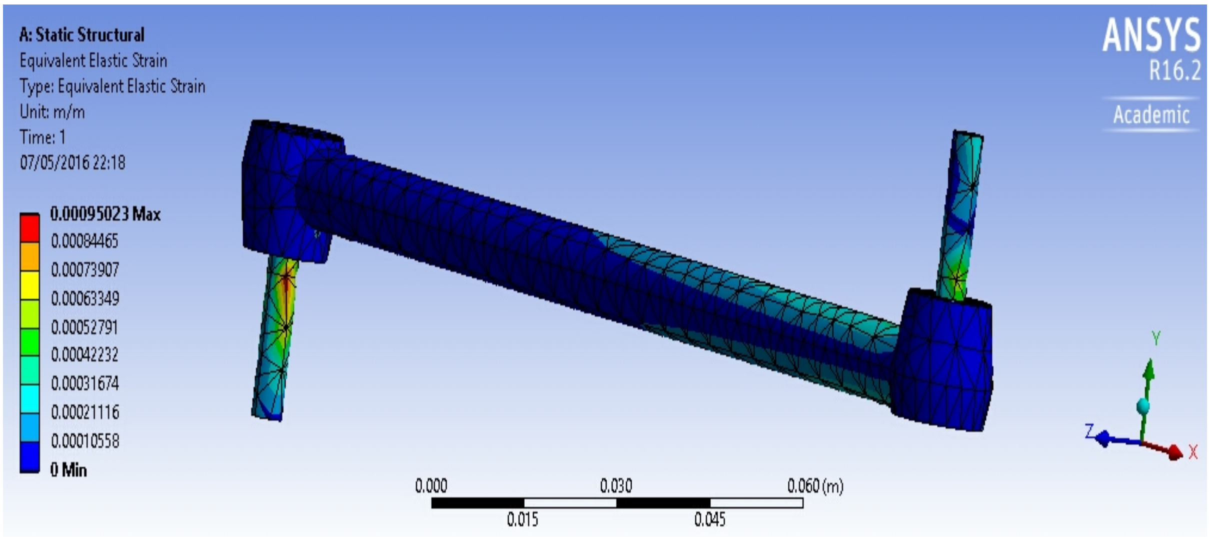


*Figure 48: Equivalent elastic strain across the tie rod*

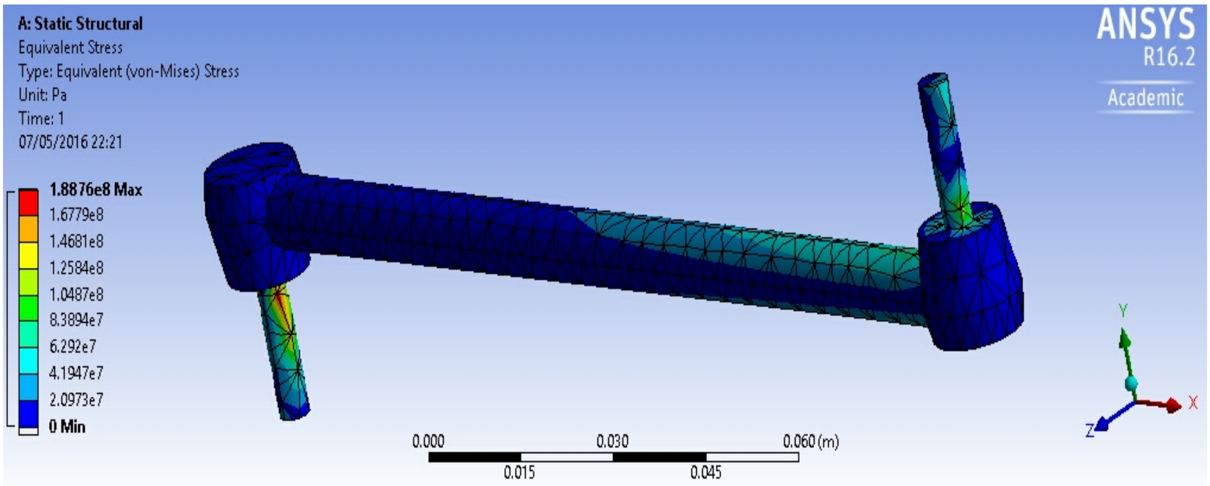


*Figure 49: Equivalent stress across the tie rod*