CERTIFICATION OF APPROVAL

Design of a
Steering Linkage Mechanism for a
Simple Three Wheeled Car

by

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Approved by,

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JUNE 2010

CERTIFICATION OF ORIGINALITY

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

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ABDUL RAUF MOHAMAD NAHAR

ABSTRACT

One of the factors in achieving high energy efficiency in terms of fuel consumption of a car is the need to have simple and light steering system, and a three wheeled car is one of the ways to achieve this. This project attempts to design a steering linkage mechanism for a three wheeled vehicle, with two in front and one rear wheel, rear driven. Analytical and critical studies were made prior and parallel with the design to assure stability and achieved required requirements. Steering system must provide the means to steer the vehicle with stability, predictability, and enough cornering radial. Studies include analytical and critical evidence to prove the designs are logical and able to maximize the car's efficiency.

This project uses the convergence design method where several designs were made to achieve a better design solution. The steering requirements were determined based on researches on several literatures prior to the designing stage. The designs were based on the Ackermann conditions, where the turning radius of the car, the steering ratio, and the behavior of the car during turning were studied. The design requirements were obtained using graphical and mathematical approach. The results are the position and dimensions of the linkages.

ADAMS is used to simulate the movements of the linkages during the turn, where the positions and the dimensions of the linkages were used to plot the hard points in ADAMS. The results from the simulation gives a plot of the front tire turning angles against steering wheel movement turning angle. This plot is analyzed to show the steering ratio and the behavior of the turning linkages. This design is then modified to get another design solution to improve the initial design.

Conclusion of the project is the dimensions of the linkages, the steering ratio of the designed linkages, and the behavior of the designed steering linkages during a turn.

ACKNOWLEDGEMENT

In the name of Allah and praise to Him the Almighty that in his will and given strength that this project managed to be completed. The author would like to acknowledge and thank everyone that has involved and give support and guidance in completing this final year project. The author would also like to give appreciation to Final Year Project coordinator that has made necessary arrangement in order for me to complete this project without any difficulties.

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CHAPTER 1
INTRODUCTION

1.1 BACKGROUND

Complex and heavy steering, brake, and suspension system for a normal car are not energy efficient because of energy losses. 80-85% of the portion of the car fuel energy is lost before it gets to the wheel. About 95 percent of the resulting wheelpower hauls the car itself, so that less than one percent of the fuel energy actually ends up hauling the driver [1].

One of the ways to achieve this is to reduce the number of wheels. It is either to have two rear wheels, or two front wheels. There are several advantages and disadvantages for both designs [2].

There are some differences in three wheeled car dynamic characteristic compared to a four wheeler [3][4]. Thus, a different approach is needed in designing a three wheel steering mechanism.

Geometrical analysis is a very important step in designing a steering system. If the linkages of the steering system are properly designed, with the right angles and height, it could decrease the load applied to each components, joints, and links [4]. Thus, it is possible to use much lighter material. It is the first step in designing a steering system.

When discussing a steering geometry, it means a broad subject of how the linkages of the steering are connected to get the car turns in a predictable, stable way. This connection does not only dictate the path of the relative motion, it also controls the forces that transmitted between them. Any geometry design must be designed to meet the needs of the particular needs, or constraints, of the particular vehicle for which it is to be applied. Thus, there is no single best geometry, but rather, a lot of solutions could be found [5]. Of course there is a lot of compromise to be made. It is usual to find several geometrical solutions, depending on the compromised made. Complete understanding of the kinematics and dynamics on the mechanism is required in order to optimize the geometry.

Three wheeled cars has been produced by giants automotive company, thus suggesting that there's a "do and don't" in designing a three wheeler steering mechanism [3]. In general, a car should be able to:

- Resist losing the rear end in turns
- Able to travel at high speed without continual steering corrections to counteract weaving
- Resist from tipping over in turns and encountering changes in road surfaces if sliding
- Resist from swapping ends in hard breaking due to weight transfer

A good design of a steering linkage will be able to improve the car steering performance, assure that minimum turning radius will be met, and reduces load acting on each links [3].

To measure the performances, several parameters are looked upon. The parameters to gauge the performance of the design in this project are:

- Steering movement angle to steer angle ratio
- Car turning radius about its centre of gravity with a minimum 30^o inside wheel turn angle
- The Ackermann geometry

Steering movement angle to steer angle ratio is ratio of the steering wheel movement against the tire angle movement, also known as the steering ratio [6]. It defines the "sensitivity" of the car. In manual steering installation, the geometry is the only way to control the sensitivity. In high speed car applications, such as in racing industries, the desirable sensitivity of the car is very high, thus smaller ratio. In a passenger car, the ratio is from 12:1 to 20:1 [6]. As an example BMW M3 CSL have 14.5:1 to 15.4:1 [7]. Since the nature of the project is to design a simple and basic car, it is assumed that the car is small and the feel should be such as a race car. Thus a smaller ratio, near to 1:1 is preferable [8].

Having a sensitive car that cannot turn around specific radii is useless as the car must be able turn around corners. Thus, the linkage should be design to be able to turn around a specific minimum radius. To achieve this, is a trivial task. However, to achieve this without compromising the Ackermann geometry is different. Thus, it is proper to get the angles by solving the Ackermann geometry.

Ackermann geometry is the base of this project. Rudolf Ackermann discovered and defined this principle in the 19^{th} century. Ackermann steering has a huge impact on many different vehicles. In general, during a turn, one of the front tires will scrub more than the others. This will results in a loss of efficiency since the forces will be transferred to the scrubbing. Ackermann geometry is simply a mathematical approach to describe the geometry, and is only true when it met several conditions, such as low constant speed throughout a turn, static turning angle, and based on the bicycle model. In practical, perfect Ackermann geometry is impossible as the car will turn in speed, the road is not always flat, and the driver always adjusts the angles in a turn. However, the more the car conforms to the Ackermann steering geometry theory, the more efficient the car is. Thus in this project, the Ackermann geometry will be the base of every calculations [9].

Since this project is about designing a three wheeled car steering mechanism, it is proper to choose which of the three wheel car concept first. There are several types of three wheeled car. There is the two front wheels and one rear wheel, two rear wheels and one front wheels, and two front wheels with the back wheel that steer the car. Each has its own advantages and disadvantages.

Correct steering geometry is particularly important for low powered vehicles, because tire scrub as you turn, the energy wasted can significantly reduce the velocity, thus reduces the overall efficiency. The design method often used to minimize this effect is also useful for lightweight electric or solar vehicle [10].

There are many aspects in designing a linkage for the steering mechanism of a three wheeled car, but knowing that a few of these aspects alone is enough to cover a lot of steering dynamics, it is safe to proceed the design with more attention on these few aspects.

1.2 PROBLEM STATEMENT

Three wheeled car is usually designed to support a lighter load or for use in urban area. When designing a three wheeled car, the mechanism on how the car will steer play a major role. It is very important to assure that the linkage actually works. In this report, the mechanism of a type of a three wheel car is designed. Design parameters are the reliability, simplicity versus complexity, and feasibility.

1.3 OBJECTIVES

The objective of the study is to:

- Design an appropriate steering linkages mechanism
- To analyze on the design; specifically:
    - Lengths and angles of each links in the linkage
    - The car steering ratio
    - Minimum turning radius of the car about its centre gravity with a minimum 30^o inside wheel turn angle
    - The Ackermann geometry

This includes:

- Research of appropriates literature materials that involves three wheeled car designs
- Research on the design constraints and design parameters

1.4 SCOPE OF STUDY

In this report, only the mechanism of the steering would be analyzed for the design. Further analysis includes the steering ratio, the Ackermann effect, turning radius, and feasibility.

Activities that have been accomplished are research for related literature review, planning for the car design, searching for the design parameters, drawing of a conceptual drawing, mathematical design of the car and simulation of the steering linkages mechanism using ADAMS software.

CHAPTER 2
LITERATURE REVIEW

2.1 TYPES OF THREE WHEELED CAR

These types of car, by design, are basically a triangle shape. The car either has two wheels up front and one in the rear or two wheels in the rear and one up front depending on where the passenger sits the location of the engine and the placement of other critical mechanical components. Having one wheel up front and two in the back is known as delta configuration. The benefit of this delta configuration is its low cost. Most cars applied this concept, which have the engine driving the rear wheels and leave steering to the front one [11].

The second type is called the tadpole or reverse strike. It is the opposite of the delta, which is the car, has two wheels up front and one in the back. It is either steered by front wheels, or rear wheel. This design is much more stable than the delta setup because the back wheel drives the vehicle while the two wheels up front are controlling the steering. Besides that, it also contributes to aerodynamic benefit since the car is shaped almost like a teardrop which is wide and round up front and tapering off in the rear. This will allows air to flow easily over the vehicle's body. This tadpole design has becoming more and more favorable to designers due to its stability, aerodynamics, and ability to house a fuel-efficient engine [11].

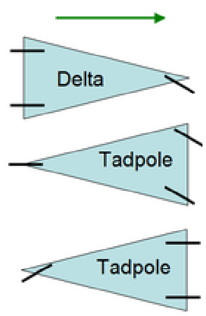


Figure 1: Types of Three Wheelers [11]

2.2 STEERING GEOMETRY ON A FOUR WHEEL CAR

2.2.1 Ackermann Angles

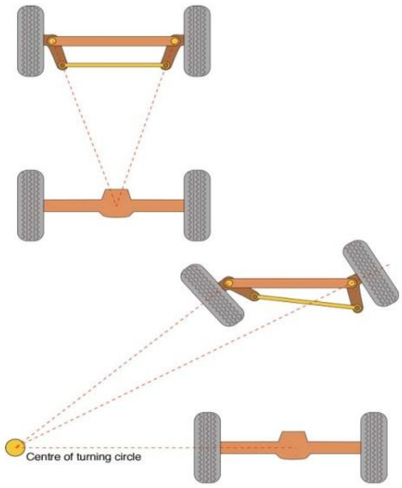


Figure 2: Ackermann Steering Geometry [12]

For Ackermann Steering:

$$\delta_i = \tan^{-1}\left(\frac{L}{R}\right) \qquad \delta_o = \tan^{-1}\left(\frac{L}{B+R}\right)$$

Where δ_i and δ_o are the angle of turn required on the inside and outside wheels respectively from the straight ahead position in order to achieve perfect Ackermann. L is the wheelbase; R is the turning radius from centre of rotation to inside wheel; and B is the track width of the car [12].

The Ackermann design was invented to solve the problem of the inside and outside wheels having to turn at different radii as a car turned a corner [12]. This effect caused the inside wheel to scrub during the maneuver therefore by incorporating a design which utilized Ackermann theory the inside wheel could be made to turn more than the outside wheel. 'Perfect Ackermann' occurs when lines are drawn normal from each wheel and they both converge at a distance equal to the radius of the corner.

Ackermann general condition equation [13]:

$1/\tan \theta_o - 1/\tan \theta_i = B/L$

Where, referring to ***Figure 3***:

θ_o = turn angle of the wheel on the outside of the turn

θ_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

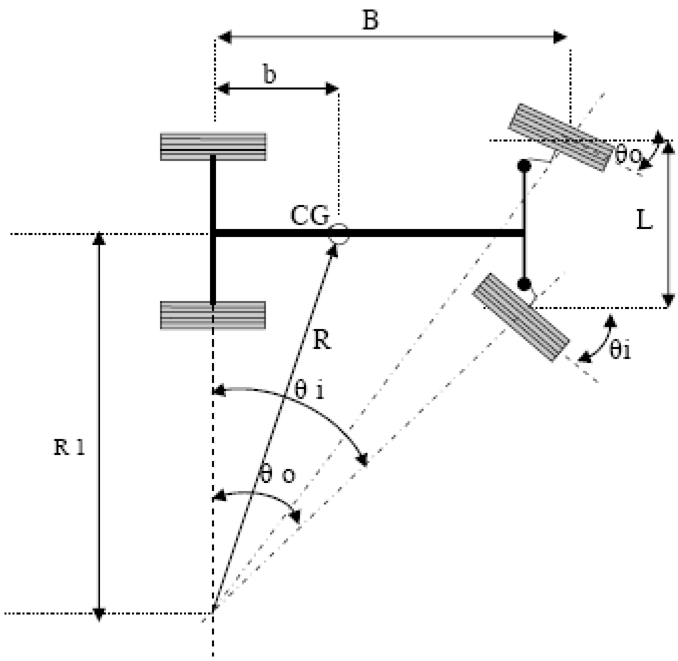


Figure 3: Ackermann Conditions [13]

From this equations, the outer wheel turning angle at the outside turn for a given inside wheel angle can be calculated. The resulting angle could be further calculated to find the minimum turning radius at it centre of gravity using the geometry [13].

One of the team from a Shell Eco Marathon challenge did a case study on their car [14]. This particular team analysis showed that due to the extremely large radius of the corners on the Rockingham oval track (188m), the angle of turn required was 0.463 degrees for the inside wheel and 0.461 degrees for the outside wheel. Whilst the Ackermann angles would be essential for good performance at a tighter track, the analysis showed that due to large radii corners of Rockingham track, any performance advantage through using Ackermann angles would be negligible.

2.3 BASIC CONCEPTS ON THREE WHEELED CAR

Correct steering geometry is particularly important for three wheels vehicles, especially when the power plant produce low power, like the solar racer car, a tricycle, or a small single cylinder engine, because if tires scrub as you turn, the energy wasted can significantly slow you down [10]. It can also end up being costly due to the need to replace tires often. This design method is really popular in many mulitrack vehicles.

There are several aspects to steering design [3]:

- The Ackermann steering geometry which dictates the steering linkage to turn the wheels at the correct angle when taking a corner.
- Minimizing bump and brake steer
- Stability and self-centering effect

2.3.1 Ackermann steering geometry

When a three wheeler takes on a corner, it turns around a point along the line of its rear axle. As the diagram shows, the two front wheels will have to turn through slightly different angles so that they are also guiding the vehicle round this point, but at the same time not over constraint or 'fighting' the turn by scrubbing.

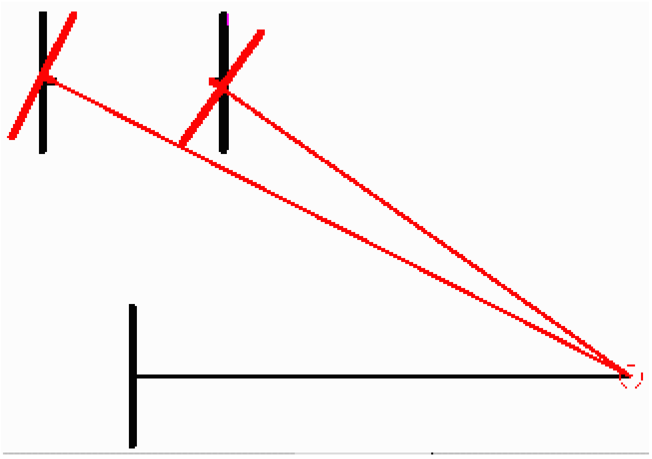


Figure 4: Ackermann Geometry for a three wheeled car [10]

Figure 4 shows the inside wheel turns through a greater angle than the outer.

Ackermann geometry is simply steering which achieves this, keeping each front wheel at the correct angle, through the whole range of the steering motion.

Ackermann steering theory is only true during a steady state cornering where all the tires did not change position during the cornering and the car moves at constant velocity. Thus, it is very unlikely that the Ackermann theory will ever be achieved during a turn. Additionally, even with a perfect Ackermann steering, scrubbing is still unavoidable due to dynamic effect where the rear tire tries to push the car forward while the front tires pushing the car to turn, also known as tendency to understeer [3].

2.4 YAW RESPONSE AND CENTRE OF GRAVITY

Stability of a car can be designed by proper choice of the centre of gravity location, longitudinally and in height, the front track and the wheelbase [3].

The yaw response of the vehicle refers to its tendency to rotate about a vertical axis through the centre of gravity. A stable vehicle can undergo side loads as in and not suddenly yaw in such a way as to amplify the tendency to spin. It is possible to yaw slightly in a self-corrective manner. The type of response depends largely upon the location of the centre of gravity. [3]

2.4.1 Slip Angle

When a loaded rolling tire is subjected to a side load, its path is deflected from the direction in which the tire is headed. The angle of deflection is called the slip angle. ***Figure 5*** shows a top view of a tire with side force $\underline{F}$ applied at the axle line, with velocity V along the direction of travel, and slip angle α between the actual path and the tire heading. The axle rotates is assumed always in bearings which keep the tire vertical to the road [3].

Figure 6 shows the front view, with lateral force F acting upon the tire from the ground, and a vertical load N, along the centerline. Lateral force F is equal in magnitude to side force $\underline{F}$ [3].

There is a relationship between the lateral force F and the slip angle, which is found experimentally and reported in the technical literature as plots of lateral force F vs. slip angle for various vertical loads [3].

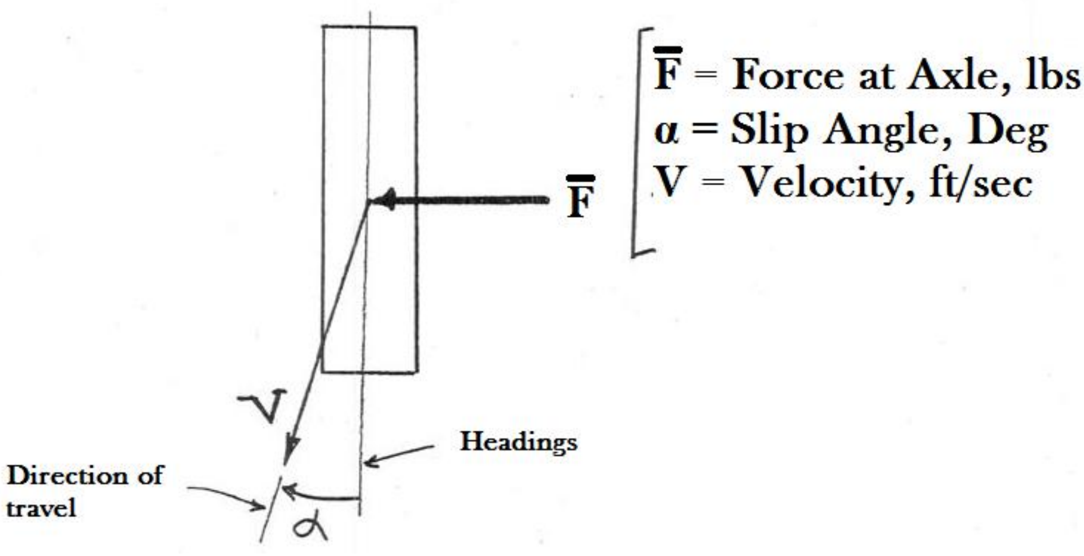


Figure 5: Top view of the tire [3]

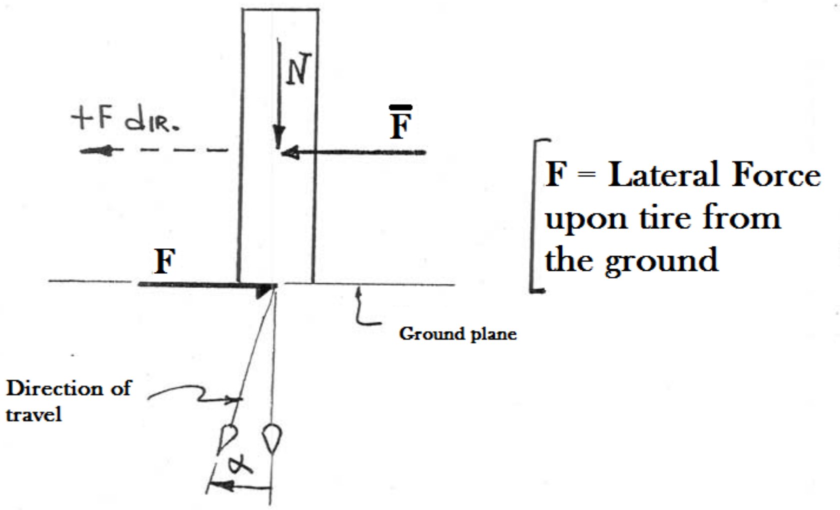


Figure 6: Front view of the tire [3]

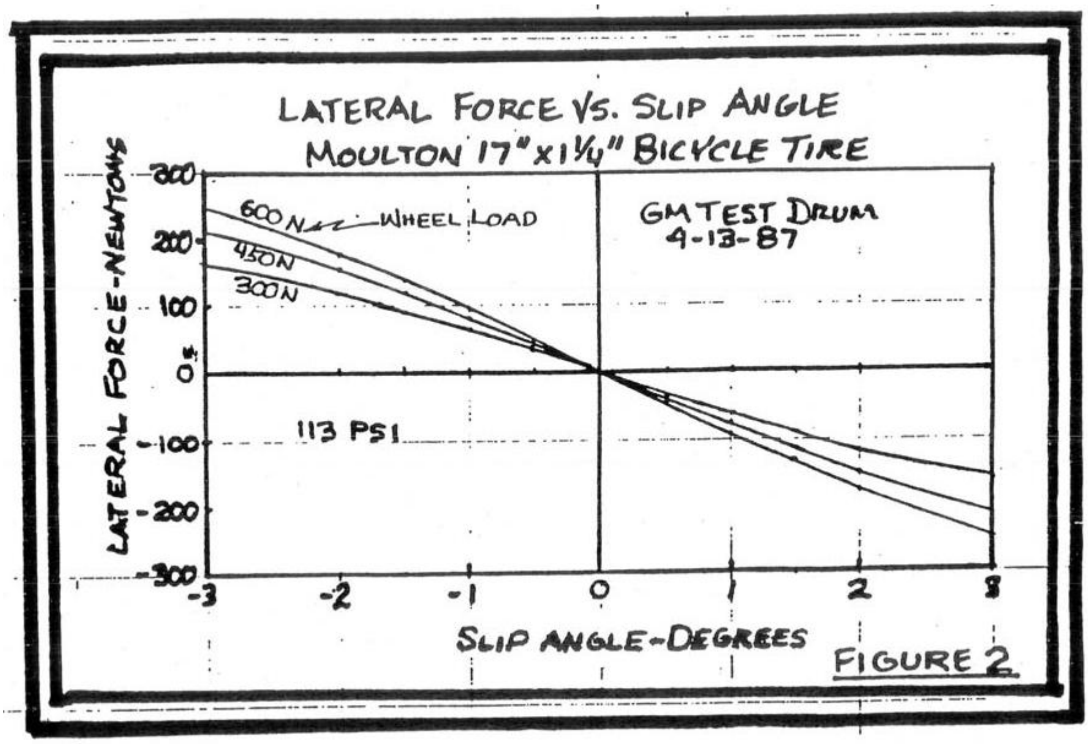


Figure 7: Lateral force vs Slip Angle[3]

It can be shown from ***Figure 7*** that the lateral force developed by a tire at a certain slip angle will increase as the vertical load increases, but in a diminishing manner.

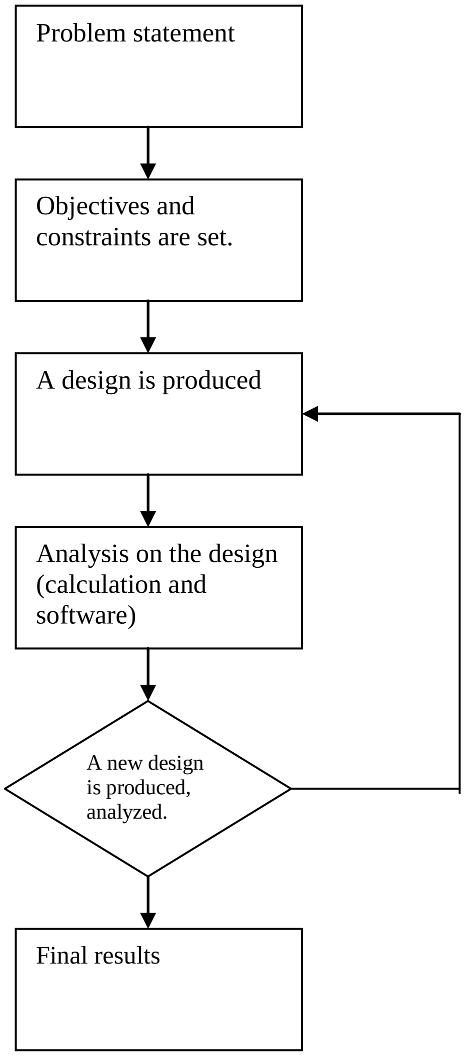
Cornering stiffness, C is the slope at the linear states of the lines. As load increases, C increases in diminishing manners.

A “coefficient of lateral friction”, μ_c can be interpreted as the ratio of the lateral force divided by the vertical load.

CHAPTER 3

METHODOLOGY

3.1 FLOW CHART



- The project starts with attending to the problem
- Objectives are decided and constraints are set
- A first design is produced
- Designs analyzed using software and calculation
- Another design is produced and analyzed. The results is compared with the first design
- This step is repeated until the most preferable design is chosen.

Figure 8: Flow Chart

3.2 PROCEDURE

It is significant to know how the design of the steering mechanism is acquired. The approach used to get to the final design is by trial and error. An initial design is prepared with several assumptions, and it is improved in several areas to reach a better design. The design process involved as the following:

1. Conceptual of the design is prepared
2. Preliminary design is prepared
3. Selection and evaluation criteria is conducted
4. Detail design of the car is prepared

Details of the procedures discussed below:

1. Conceptual of the design is prepared.
    a. Design constraint
        i. The design constraint for the car is decided; however the figure chosen for these are common enough to be used on normal road.
        ii. Maximum wheelbase, β is 250 cm.
        iii. Maximum width of the car should not be half the length of the car, thus must be less than 125 cm.
    b. Design parameter is defined
        i. Steering ratio
        ii. Minimum turning radius about the centre of gravity
        iii. Ackermann geometry
    c. Conceptual drawing
        i. A conceptual drawing of how the steering linkages looks like is drawn
2. Preliminary design is prepared
    a. Using the constraint given, a preliminary design is assumed and drawn.
    b. Using mathematical and graphical approach, outer wheel turning angle can be determined

c. Minimum turning radius at the centre of gravity of the car is determined from the angle
d. The design is drawn in ADAMS to simulate the motion. Analysis on ADAMS shows the exact position of every point during the motion, allowing a plot to be made to show the position of the steering wheel and the position of the tires during a turn, in terms of angles. The simulation also able to show if the design makes a sudden "jerk" on the steer angle during the steering wheel motion.
e. Steering ratio of the design is obtained from the plot.

3. Selection and evaluation criteria
   a. Using the parameters value from the preliminary drawings, another design is made by modifying the preliminary design values
   b. The design is analyzed again using mathematical and graphical approach, and simulated into ADAMS to get the design parameters' values.
   c. Data of the specified parameters is recorded into the results
   d. Results, which is the design parameters, of the two analysis is compared
   e. The better of the two is chosen, and modified again to get a better results
   f. This step is repeated until acceptable parameters' values achieved
4. Detailed design
   a. The final chosen design linkages dimension and angles is acquired
   b. The design is drawn

3.3 DESIGN SPECIFICATION

Maximum track width, β = 250 cm

Maximum wheelbase, L = 125 cm

Minimum turn radius around the centre of gravity at minimum inner wheel turning angle, θ_i, = 500 cm

Front wheel steering mechanism, with two driven front wheels and one rear driver wheel.

CHAPTER 4
RESULTS

4.1 CONCEPTUAL DESIGN

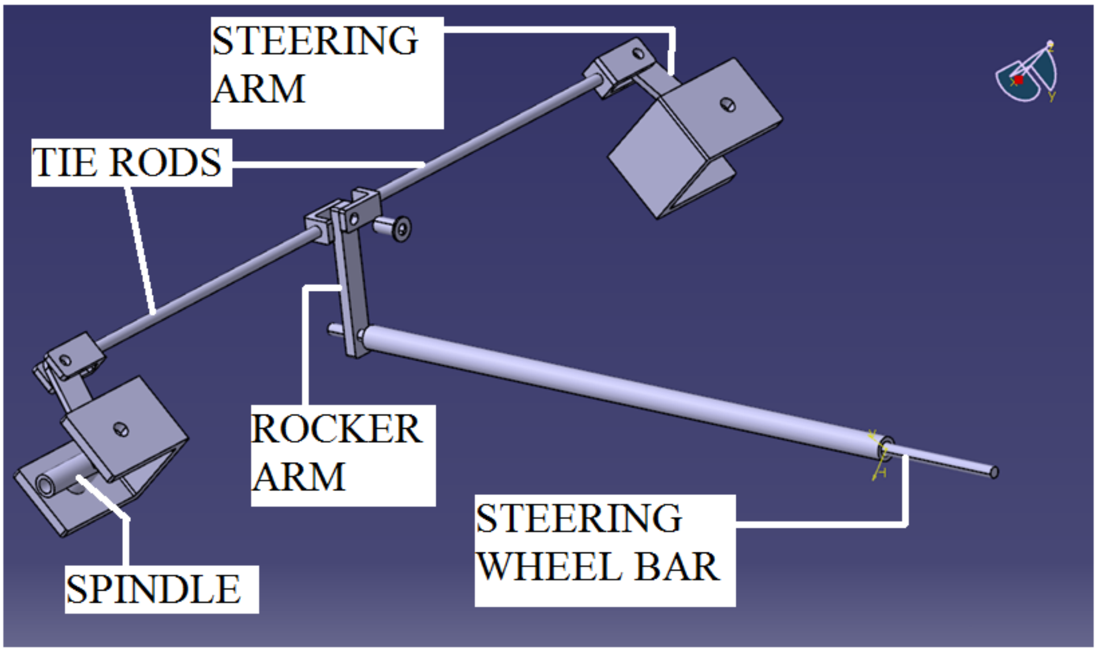


Figure 9: Steering mechanism linkage with labels

Since different author use different names for the linkage, ***Figure 9*** was labeled the before proceeding into further discussions.

Steering wheel bar is where the steering wheel will be attached. This steering wheel bar will be rotated when the steering wheel is rotated. This motion will also rotate the rocker arm in appropriate direction. The rocker arm will then move the tie rods almost axially, which in turn will push or pull the steering arm. The steering arm will pivot on the kingpin, thus moving the spindle.

This concept is basically inefficient because of the joints used is not designed properly. A better way is to use ball joints instead of nuts and bolts.

4.2 THE INITIAL DESIGN

4.2.1 Mathematical and graphical approach

Below is the calculation based on a steady-state cornering.

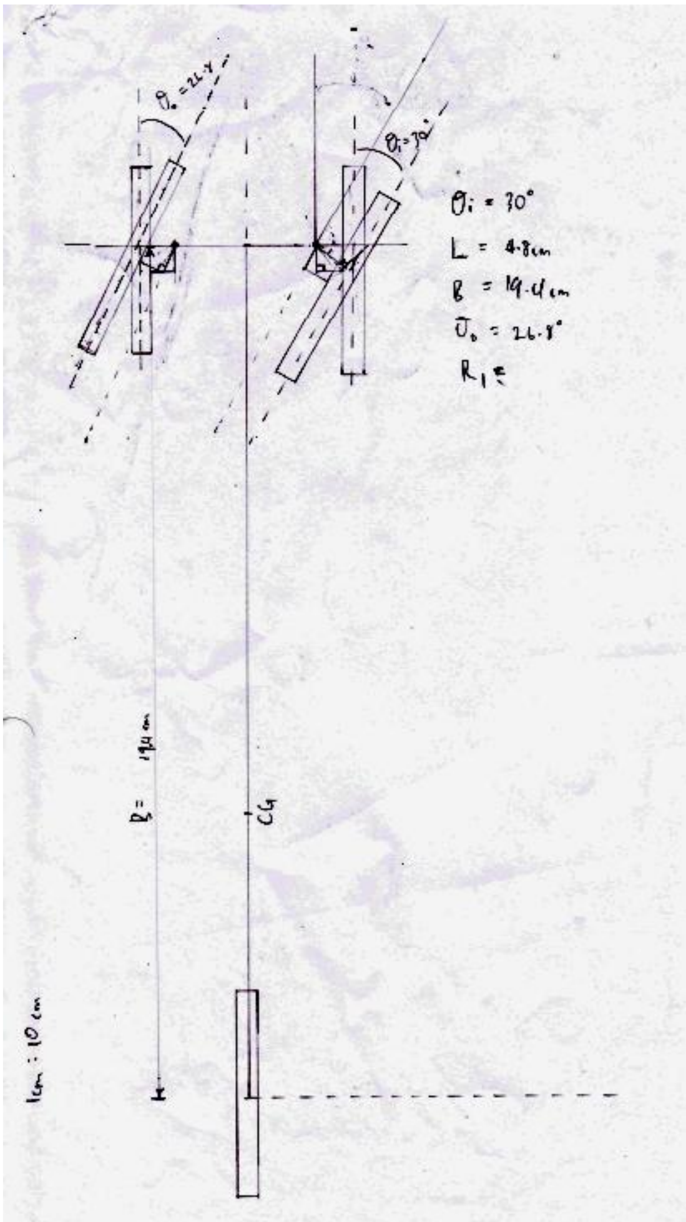


Figure 10: Ackermann Condition for Initial Design

B = 194 cm

L = 48 cm

$\theta_i = 30^o$

Where B is the track width of the car, L is the length of the wheelbase, and θ_i is the desired inner tire turning angle, in degree.

$1/ \tan \theta^o{}_o = 1/\tan \theta^o{}_i + L/B$

$= 1/\tan 30^o + 48 \text{ cm} / 194 \text{ cm}$

$= 1.979^o$

$1 / \tan \theta^o{}_o = 1.979$

$\tan \theta^o{}_o = 0.505^o$

$\theta^o = 26.8^o$

$R_1 = B /\tan \theta_i + L / 2 = 194 / \tan 30^o + 48 / 2 = 360 \text{ cm}$

$$R = \sqrt{R_1^2 + \beta^2}$$

$$= \sqrt{360^2 + 194^2}$$

$= 408.94$ cm ≈ 4 m

Thus, the turning radius of the car is nearly a little bit more than 4m at about 30^o steer angle.

4.2.2 ADAMS analysis

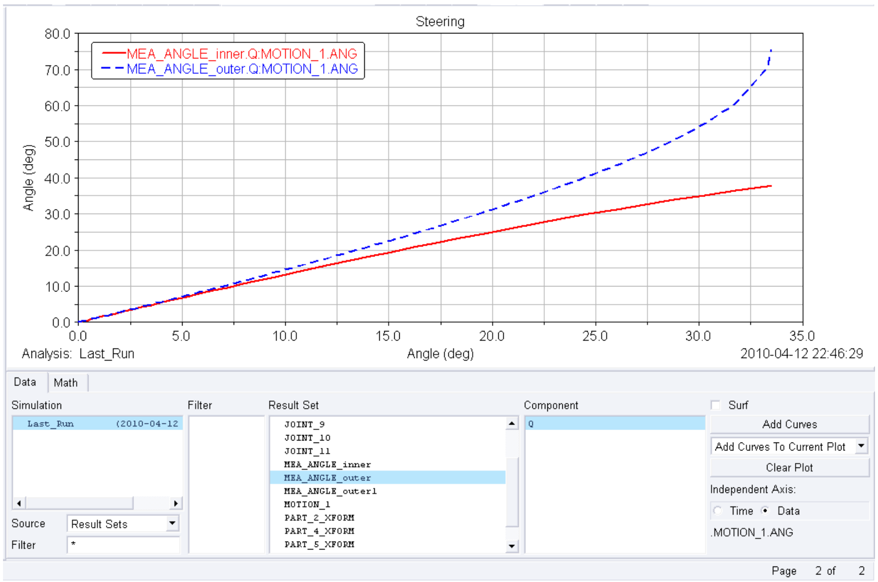


Figure 11: Tire Steer Angle vs. Steering Wheel Turning Angle

The vertical axis shows the steer angle of the car, in this case, the angle of the tire to the front of the car. 0^o turn angle means that the tire does not turn to any directions, and are at its default straight position, while 5^o turn angle means the tire turn 5^o to the right. The horizontal axis shows the steering wheel motion angle, or rotational angle of the steering wheel. 5^o motion angle means that the steering wheel is turned 5^o to the right. The blue line shows the outer tire turning angle while the red line indicates the inner tire turning angle, plotted against steering wheel motion angle. The end of both lines mark the maximum steering wheel movement angle, where the links finally cannot be moved because it is restricted by their degree of freedom.

These results are based on models below:

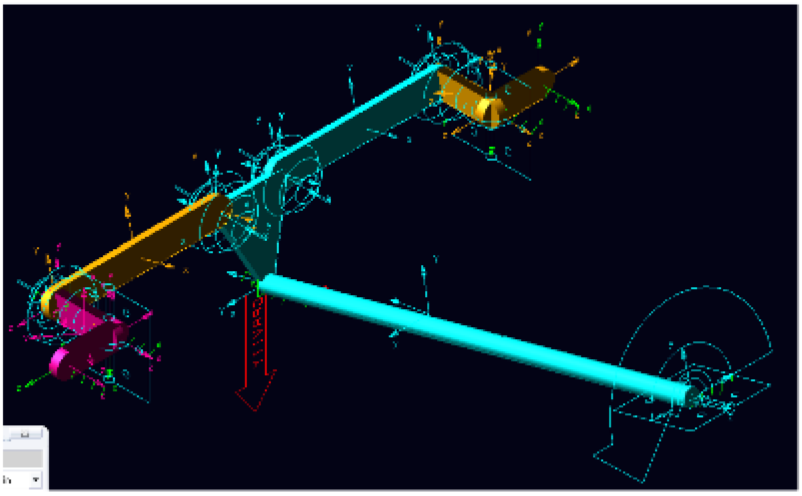


Figure 12: ADAMS (VIEW) steering model (isometric view)

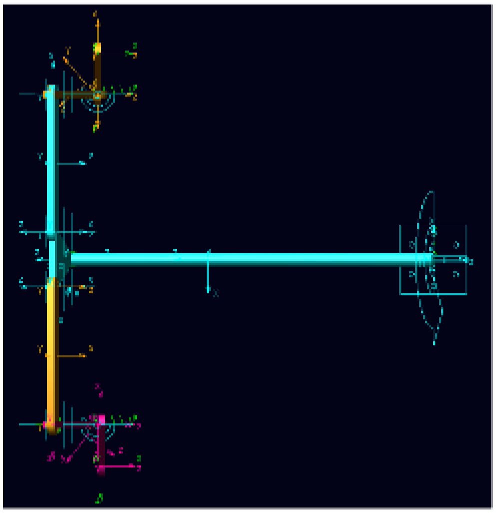


Figure 13: ADAMS (VIEW) steering models (top view)

From this, the resulting length of each links can be obtained.

Length of each tie rods = 20 cm

Length of the spindle = 6.0 cm

Length of the steering arm = 6.5 cm

Length of the rocker arm = 9.0 cm

Length between two tier rods = 8.0 cm

To get the steering ratio,

$$\text{Steering ratio} = \frac{\text{steering wheel motion angle}}{\text{mean turning angle of the tire}}$$

$$= \frac{\text{steering wheel motion angle}}{(\theta°_i + \theta°_o)/2}$$

$$= \frac{30°}{(55° + 35°)/2} = 0.67$$

4.3 FINAL DESIGN

After getting the initial design, a second design is made. The parameters in the second design are modified several times until the best numbers is achieved for the results. At last, this final design is chosen.

4.3.1 Mathematical Analysis and Graphical Approach

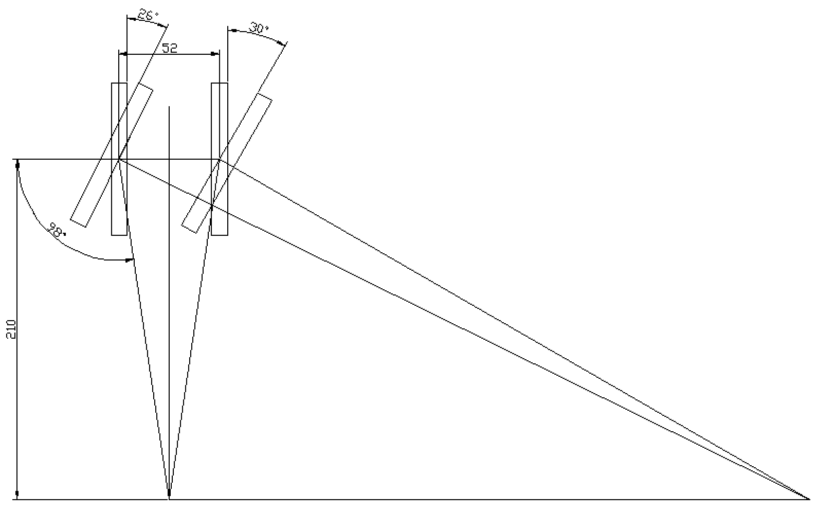


Figure 14: Ackermann Condition

$\beta = 210\ cm$

$L = 52\ cm$

$\theta_i = 30°$

Where β is the track width of the car, L is the length of the wheelbase, and θ_i is the desired inner tire turning degree.

$$\frac{1}{tan\,\theta°_o} = \frac{1}{tan\,\theta°_i} + \frac{L}{\beta} = \frac{1}{tan\,30°} + \frac{52}{210} = 1.979$$

Thus, the turning radius of the car is nearly a little bit more than 4.5 m at about 30^o steer angle. Therefore the minimum radius of turn of the vehicle around its centre of gravity for a maximum inside wheel turn of 30^o is about 4.5 meters.

4.3.2 ADAMS Analysis on the final design

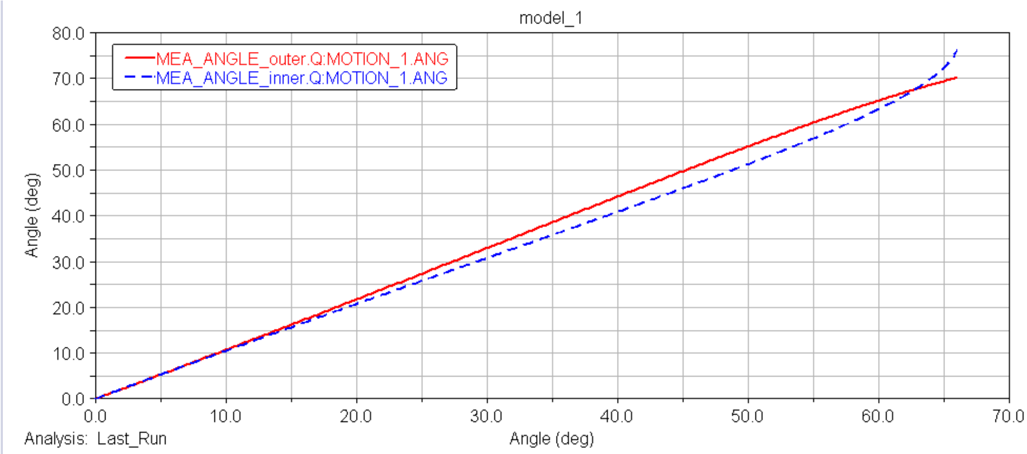


Figure 15: Tire Steer Angle Vs Steering Wheel Turning Angle

The vertical axis in ***Figure 15*** shows the steer angle of the car, in this case, the angle of the tire to the front of the car. 0^o turn angle means that the tire does not turn to any directions, and are at its default straight position, while 5^o turn angle means the tire turn 5^o to the right. The horizontal axis shows the steering wheel motion angle, or rotational angle of the steering wheel. 5^o motion angle means that the steering wheel is turned 5^o to the right. The blue line shows the outer tire turning angle while the red line indicates the inner tire turning angle, plotted against steering wheel motion angle. The end of both lines mark the maximum steering wheel movement angle, where the links finally cannot be moved because it is restricted by their degree of freedom.

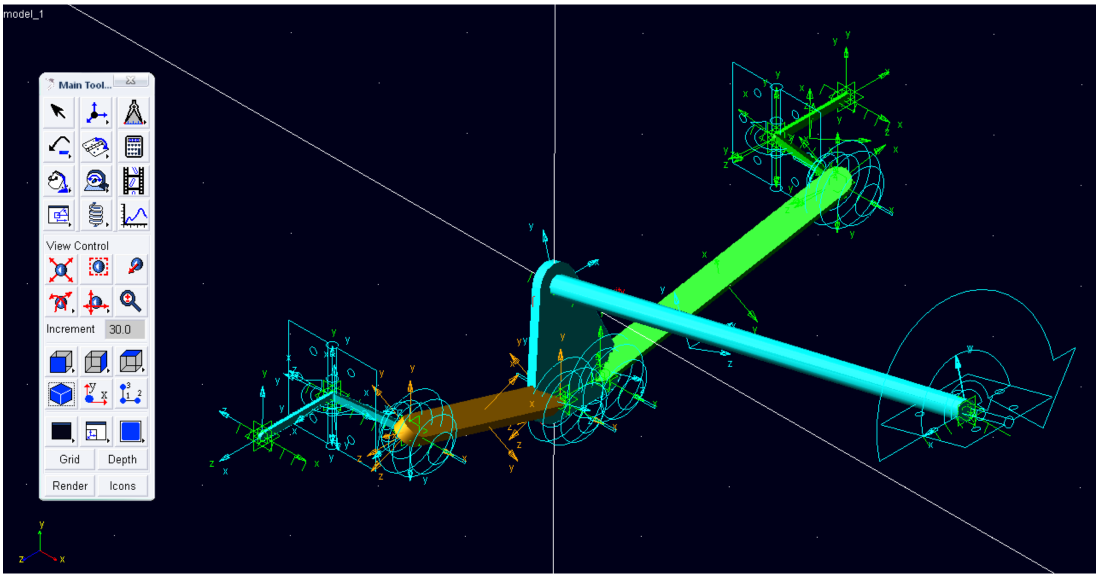


Figure 16: ADAMS Model Final Design Isometric View

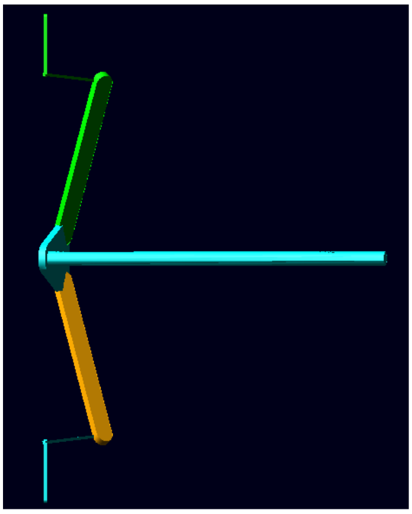


Figure 17: ADAMS Model Final Design Top View

Length of each tie rods = 18.6 cm

Length of the spindle = 6 cm

Length of the steering arm = 6 cm

Length of the rocker arm = 9 cm

Length between two tier rods = 4 cm

Angle of the spindle = 97^{o}

Calculations:

$$\text{Steering ratio} = \frac{\text{steering wheel motion angle}}{\text{mean turning angle of the tire}}$$

CHAPTER 5
DISCUSSION

5.1 ASSUMPTIONS

Since the car will run at low speed (below 35km/h) during the course of the race, it is safe to assume that the car will turn at a constant velocity. For this reason, accelerations and other forces acting on the vehicle are neglected.

To find out the minimum turning radius of the car at a desired angle, the Ackermann angle was used. Although it is highly unlikely for the car to have no slip angle hence scrubbing, the car always runs with three wheels. The Ackermann theory invented to overcome the problem of the outer tire scrubbing during a turn caused by the force from rear tire, which forces the front wheel to stay on its track. Having three tires however significantly reduces the effect because the track of the rear tire is now at the middle of the car. Thus, the error will not change the resulting solution drastically [3].

Tire turning angle of the car is constraint to be at maximum 30˚. This means that the car should be able to take a turn with a minimum radius of 5 m by turning only 30^o of its tire, not the steering wheel. Using the Ackermann formula with mentioned assumptions, the minimum calculated turning radius is about 4.5 m for both cars.

By assuming the circumference of each turn is less than a semicircle, it is safe to calculate the minimum turning radius needed using circumference formula, length of the turn is equal to $\frac{1}{2} * 2\pi R$, where R is the radius of the turn when the length of the turn is given. Although not all turns are a perfectly round shape, it is assumed that the driver of the car will take a turn like a perfectly round shape.

5.2 ANALYSIS ON THE INITIAL DESIGN

5.2.1 Ackermann Geometry Conformance Analysis

From ***Figure 11***, the blue line indicates the outer tire turning angle while the red line indicates the inner tire turning angle.

Both lines are plot against the steering motion movement angle. This is the angle of movement of the steering wheel. The inner tire turning angle increases proportionally with the steering wheel turning angle, almost linearly, while the outer tire turning angle increases exponentially especially when the steering wheel radius turn above 25˚.

The lines also show that at a certain steering wheel movement angle, the tire will turn sharply. This is shown by the outer tire turning angle line which increases like an exponential line. What this means that when the driver turn the steering wheel at around 25^{o}, the car will suddenly turn very sharply, assuming that the slipping occur on the inner tire.

This result shows that the steering mechanism does not follow the Ackermann no-slip angle, as the Ackermann angle should have the inner tire turning radius greater than the outer tire turning radius. This means that one of the tires, which the one with less traction quality, will be “slipping”. Either the inside tire, or the outside tire, will experience the slip. This will cause significance energy losses especially during very low speed turning, as the energy that should be converted by the slipping tire into lateral force of the car moving forward while turning has been converted all into the rotational movement of the slipping tire. In order to know which tire exactly slip, an experiment is usually made, which is the skid pad test, to test the traction of all tires. The test requires a car to take a specific turn around a circle at a constant velocity, and the velocity is increase until the car finally “washed” out from the circle, or roll over. The results will vary greatly depending on the the car, the tire used, the radius of the circle, and the driver ability to keep the car on the circle [15].

5.2.2 Steering Ratio Analysis

The steering ratio of the mechanism calculated shows that the mechanism is very sensitive. The steering ratio is 0.67, which means the tire reacts almost one and a half times the amount of steering motion. Sensitive steering is preferred since the car has limited space for driver movements. Additionally, sensitive steering is also preferred for a long car (relative to its width).

5.2.3 Turning Radius Analysis

Result also shows that it is possible for the mechanism to reach the desired 30˚ turning angle. 30˚ tire turning angle can be achieved at 27˚ steering angle.

Since 30^o turning angle means that the car could turn at a 4 m tight radius, this car is able to take the turn with even smaller radius than 4 m. Note that the design specifications requires a minimum of 5 m. Thus, this design is under specifications.

5.3 ANALYSIS ON THE FINAL DESIGN

5.3.1 Ackermann Geometry Conformance Analysis

From ***Figure 15***, the blue line indicates the outer tire turning angle while the red line indicates the inner tire turning angle.

Both lines are plot against the steering motion movement angle. This is the angle of movement of the steering wheel. Both the inner tire and outer tire turning angles increases proportionally with the steering wheel turning angle, almost linearly. Observe at around 65^o steering wheel turning angle, the steering geometry actually conform to the Ackermann geometry. However, since its turning degree has already goes well beyond 30^o which is the constrained value, the car could never achieve the geometry in practical.

However, the lines increase more linearly. This shows the behavior of the car during turning is much more predictable. The car will not suddenly turn sharply, rather it will turn almost in a constant manner during the driver's course of turning the steering wheel. This is desirable, as opposed to an erratic behavior of turning where the car suddenly turn sharply, or suddenly turn very little, during the course of turning the steering wheel.

This result shows that the steering mechanism does not follow the Ackermann no-slip angle, as the Ackermann angle should have the inner tire turning radius greater than the outer tire turning radius.

5.3.2 Steering Ratio Analysis

The steering ratio of the mechanism calculated shows that the mechanism is very sensitive. The steering ratio is 0.94, which means the tire reacts almost at the same movements as the steering wheels. Sensitive steering is preferred since the car has limited space for driver movements. It is noted that most racing cars has near to 1:1 steering ratio [8].

5.3.3 Turning Radius Analysis

Result also shows that it is possible for the mechanism to reach the desired 30˚ turning angle. 30˚ tire turning angle can be achieved at 25˚ steering angle.

Since 30^{o} turning angle means that the car could turn at a 4.5 m tight radius, this car is able to take the turn with even smaller radius than 4.5 m. Note that the design specifications requires a minimum of 5 m. Thus, this design is under specifications.

5.4 COMPARISON OF THE FINAL DESIGN WITH THE INITIAL DESIGN

Because the method of designing the car is by improving the initial design in steps until a very desirable design parameter's value is achieved, the final design is confirmed to have a better design parameter's value compared to the initial design.

However, the comparison will show how much the design improves from the initial design to the final design.

5.4.1 Comparison on the Ackermann Geometry

Both designs do not conform to the Ackermann geometry. However, from ***Figure 11*** and ***Figure 15***, it is shown that the final design have a better Ackermann conformance. This is shown by the amount of the divergence. Larger divergence means that the outer tire, which is supposed to take a smaller turning angle, is taking a much larger angle during specific point of driver's turning angle of the steering wheel. Smaller divergence shows that although the Ackermann geometry does not achieved, the outer tire that should take a smaller turning angle than the inner tire is actually taking a larger turning angle, but the difference between them is not as large compared to the larger divergence. In simpler words, this means that smaller divergence is more conformed to the Ackermann geometry although it is not yet conforms to the Ackermann geometry.

5.4.2 Comparison between the steering behavior

Also that can be noted from the differences is the steering behavior. ***Figure 11*** and ***Figure 15*** shows that the initial design will have a more erratic turning compared to the final design, which have a more stable turning behavior. This is shown by the curves. A smoother curve in ***Figure 15*** shows a smoother turning behavior, while an exponential curve shows that the car will turn sharply, suddenly, during the course of the driver's turning movement of the steering wheel. Although in some cases, especially for motorsport where the driver has a lot of experience driving a certain car, the driver would want to have some "sudden" turning movements by having a little "erratic" behavior of steering mechanism, it is usually more desirable to have a stable turning behavior, because it is more predictable. In this project, the more stable behavior is more desirable.

5.4.3 Comparison on the steering ratio

Steering ratio is also subjective matter, where a motorsport driver may want a small steering ratio, around 1:1; an ordinary passenger car's driver may want a smoother steering ratio, around 14:1 to 15:1. It is also not unusual to see smaller steering ratio than 1:1 in a race car, such as in the Formula One race car [8].

In this project, the steering ratio of the first design is 0.67, which is near to 1:2, and the final design gives 0.94 which is near to 1:1. Both steering ratio have high sensitivity with steering ratio like a race car. This is desirable, since the nature of the car is small, and the space for the driver is limited, the driver would want a more sensitive car so that they wouldn't need a large space to move the steering wheel. The car shape is also long compared to its width, which is twice its width, means that it is more appropriate to have a smaller steering ratio.

Thus, both designs have acceptable steering ratio.

CHAPTER 6
CONCLUSION AND RECOMMENDATION

6.1 RECOMMENDATION

While castor angle, toe in, and toe out angle is still considered at 0˚ for the time being, camber angle seems to have a greater effect of reducing scrubbing during a cornering. However further study has to be made for its feasibility as camber angle does apply some scrubbing even while not in a turn.

6.2 CONCLUSION

The final design has been achieved with a steering ratio of 0.94. The final design gives more space, better steering angle up to 65^o, with below specifications:

Length of each tie rods = 18.6 cm
Length of the spindle = 6 cm
Length of the steering arm = 6 cm
Length of the rocker arm = 9 cm
Length between two tier rods = 4 cm
Angle of the spindle = 97^o
At 30^o inner wheel turn angle, the car will turn 4.5m about its centre of gravity.

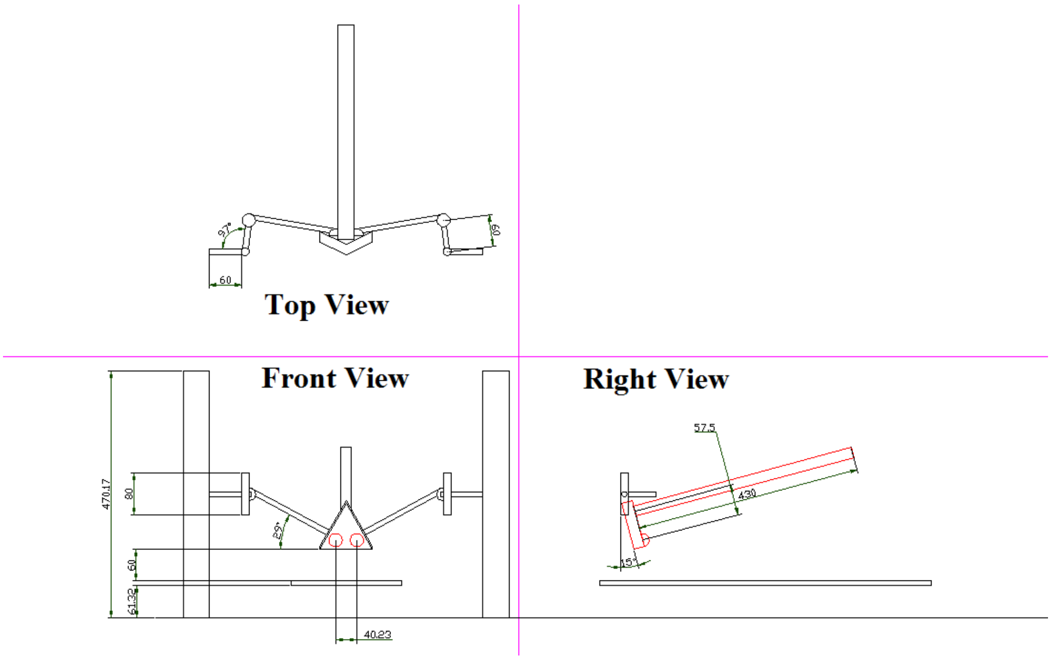


Figure 18: Final Dimensions for the Linkage of Steering Mechanism

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