University of Southern Queensland
Faculty of Engineering and surveying

Redesign of an FSAE Race Car's Steering and Suspension System

A dissertation submitted by

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In fulfilment of the requirements
of

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Towards the degree of

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Abstract

The chosen project is based on the redesign of the steering and suspension system for the University of Southern Queensland's 2008 Formula SAE (Society of Automotive Engineers) or FSAE vehicle.

In 2008 USQ's FSAE team was forced to abandon the competition due to a crash into a barrier which was thought to have been caused by the car's suspension and steering system. If USQ plans to enter a car into future FSAE competition it seemed appropriate that the current suspension and steering system be revised.

The project aimed to uncover any problems with the 2008 vehicle and then use these findings, coupled with appropriate research, to create a new steering and suspension system that possesses improved performance. Although there is no team for the competition this year it is intended that all work completed will be able to be utilised by future groups in years to come.

Completion of the project has seen the design of geometry for the suspension arms, suspension actuation mechanisms, uprights as well as the steering rack and arms. Additionally, concepts in the way of 3d models have been established for the suspension and steering systems.

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ENG4111 Research Project Part 1 & ENG4112 Research Project Part 2

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Signature

Date

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Chapter 1

Introduction

1.1 Problem Definition

The steering and suspension systems are crucial to successful operation of any variety of car. Due to the large responsibility that the these two major components share coupled with the fact that race cars are capable of reaching very high speeds and accelerations, it is obvious that consequences of failure or improper setup of the suspension and/or steering could be quite catastrophic. In 2008 the University of Southern Queensland (USQ) Faculty of Engineering and Surveying entered a car into the Formula SAE Australasia (FSAE-A) competition only to be forced to abandon the event due to a crash into a barrier following a wide exit on a corner. Although there is some uncertainty into the cause of the crash, there was mention that it appeared that one of the front wheels was jacked up off the ground which in effect hindered the car's ability steer. Supporting the argument that the steering and suspension was to blame for the incident, students also noted that the 2008 car had some odd handling characteristics and particularly recognised that steering was rather heavy. It is believed that one of the main causes behind the inferior suspension and steering setups was due to design of these parts being compromised by an early chassis construction.

If USQ plans to enter a car into the 2012 FSAE-A competition it seemed appropriate that the current suspension be revised such that the car could be driven in a fast, manageable and most importantly, safe manner.

1.2 Project Objectives

The project aims to uncover any problems with the 2008 vehicle and then use these findings, coupled with appropriate research, to create a new steering and suspension system that possesses improved performance.

While striving to achieve these major aims, project work will focus on a number of basic objectives. Firstly and most importantly, flaws present in the 2008 vehicle must be removed or improved upon; clearly it is important that the project improves on the past car in some way otherwise all work would be a waste of time. Additionally, it is intended that the final design will be easily adaptable and adjustable such that future USQ teams can incorporate or modify the suspension and steering systems without too much hassle. Lastly, it is hoped that the work documented in the dissertation will be able to serve as a significant aid to these future teams whether they nominate to use the final design produced from this project or even if they choose to start from scratch.

An outline of specific project tasks is as follows:-

1. Research information on currently used automotive steering and suspension systems.
2. Research the existing rules and restrictions for Formula SAE-A race car steering and suspension design.
3. Critically evaluate existing alternatives for steering and suspension designs.
4. Critically evaluate researched methods of testing and adjusting the steering and suspension.
5. Repair University of Southern Queensland's 2008 Formula SAE-A race car before testing and analysing its design.
6. Develop preliminary design of the chosen steering and suspension systems.

As time and resources permit:

7. Manufacture and install prototype into Formula SAE-A racer and evaluate.
8. Test and obtain feedback from drivers and modify designs as needed.

1.3 Overview of the Formula SAE competition

> *The Formula SAE ® Series competitions challenge teams of university undergraduate and graduate students to conceive, design, fabricate and compete with small, formula style, vehicles.*
>
> *SAE International, 2010, p6*

Expanding on this, the competition occurs annually on both a regional (i.e. Australasia) and international level; if successful at the regional round teams are offered to represent their country in the international competitions against universities from all around the world which have all followed the same rules in creating their own formula SAE race cars.

Due to the limited number of restrictions on the overall vehicle design, teams have a large degree of design flexibility and the opportunity to express their creativity and imaginations. However, all design will typically be centred around a number of common goals. As the competition tracks are normally very tight with few opportunities to achieve top speed, vehicles must have exceptional accelerating, braking and handling performance.

Additionally, teams are expected to complete the design task from the perspective of a design firm that is producing 1000 examples of the car for a non-professional, weekend, competition market. Production costs per vehicle created must stay below AU$50000, demanding the car be economic to manufacture as well as assemble and consist of materials and parts that are readily available and perform cost efficiently. Other factors that teams will potentially consider are also the aesthetics, ergonomics and manufacturability.

1.3.1 Judging

Each team is scored under two categories; static events and dynamic events which are broken into the following sub categories seen over the page.

Static Events and Maximum Scores

Technical Inspection	No points

Where the car is inspected to check compliance with the FSAE rules. If any part of the car does not comply or is deemed to be a concern, the team must correct the issue and have the car re-inspected before any other events can proceed

Cost and Manufacturing 100 points
This event consists of three requirements; a cost report detailing the cost of every part on the race car, a discussion session with the competition cost judges where the cost of the car as well as the team's ability to prepare accurate engineering and manufacturing cost estimate, and lastly, a real case scenario in which students are asked to respond to a challenge related to cost of manufacturing of their vehicle

Presentation 75 points
Teams are asked to complete a presentation aimed at convincing the executives of a corporation that their car is best suited to the application of the amateur, weekend competition market and that it can be profitably manufactured and marketed

Design 150 points
Engineering effort and how the engineering meets the intent of the market is judged through medium of a design report and design spec sheet that must be submitted prior to the actual competition date

Total 325 points

Dynamic Events and Maximum Scores

Acceleration 75 points
Where the car's acceleration is tested with a 75m timed sprint where there is to be no 'burnouts' at any time

Skid Pad 50 points
The skid pad test aims to gauge the car's cornering ability on a flat surface while making a constant radius turn

Autocross 150 points
Tests the car's manoeuvrability and handling on a tight course without the hindrance of competing cars

Fuel Economy 100 points
Endurance 300 points
The endurance event requires the car be driven for approximately 22km with no modification or repair and aims to test the durability and reliability. The event operates in conjunction with the fuel economy testing

Total 675 points

Grand total 1000 points

1.3.2 Vehicle Requirements

General Requirements

The race car must be open-wheeled and open cockpit with four wheels that are not in a straight line. Additionally, there are to be no openings through the bodywork into the driver compartment (other than the cockpit opening, the car must have a minimum wheel base of 1525 mm, a difference in tracks in either the front or back of no less that 75% of the larger track, and lastly, all items to be inspected by the technical inspectors must be clearly visible without the use of instruments.

Engine and Drivetrain Requirements

A piston engine using a four stroke primary heat cycle with displacement not exceeding 610cc is required. The engines are able to be modified within the restrictions of the rules with turbochargers and supercharges approved for use. A major consideration though, is the 20mm (for petrol fuelled cars) or 19mm (E-85 fuelled cars) restrictor that must be placed in the intake system between the throttle and the engine which all air has to flow through. Teams are allowed to couple their engine setup to any transmission and drivetrain.

1.3.3 Suspension Requirements

As quoted from the 2011 FSAE rule book:

B6.1.1 The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit.

B5.8.1 To keep the driver's legs away from moving or sharp components, all moving suspension and steering components, and other sharp edges inside the cockpit between the front roll hoop and a vertical plane 100 mm (4 inches) rearward of the pedals, must be shielded with a shield made of a solid material. Moving components include, but are not limited to springs, shock absorbers, rocker arms, antiroll/sway bars, steering racks and steering column CV joints.

B5.8.2 Covers over suspension and steering components must be removable to allow inspection of the mounting points.

B6.1.1 The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit.

B6.1.2 All suspension mounting points must be visible at Technical Inspection, either by direct view or by removing any covers.

B6.2 Ground Clearance

There is no minimum ground clearance requirement. However, teams are reminded that under Rule D1.1.2 any vehicle condition which could, among other things, "… compromise the track surface" is a valid reason for exclusion from an event. Any vehicle contact that creates a hazardous condition or which could damage either the track surface or the timing system is cause for declaring a vehicle DQ.

SAE International, 2010, p43

1.3.4 Steering Requirements

As quoted from the 2011 FSAE rule book:

B6.5.1 The steering wheel must be mechanically connected to the wheels, i.e. "steer-by-wire" is prohibited.

B6.5.2 The steering system must have positive steering stops that prevent the steering linkages from locking up (the inversion of a four-bar linkage at one of the pivots). The stops may be placed on the uprights or on the rack and must prevent the tires from contacting suspension, body, or frame members during the track events.

B6.5.3 Allowable steering system free play is limited to seven degrees (7°) total measured at the steering wheel.

B6.5.4 The steering wheel must be attached to the column with a quick disconnect. The driver must be able to operate the quick disconnect while in the normal driving position with gloves on.

B6.5.5 The steering wheel must have a continuous perimeter that is near circular or near oval, i.e. the outer perimeter profile can have some straight sections, but no concave sections. “H”, “Figure 8”, or cutout wheels are not allowed.

B6.5.6 In any angular position, the top of the steering wheel must be no higher than the top-most surface of the Front Hoop.

SAE International, 2010, p44

1.4 Suspension and Steering System Definition

The following figure provides assemblies of the front and rear suspension and steering systems for an FSAE race car. The key components of these systems are numbered and listed below. Throughout the dissertation these components will be referred to and thus an early introduction into their appearances and applications will allow the reader to gain a much better understanding of the author’s work. With reference to the figure over the page:

1. **Coil over shock absorber**
2. **Tyre**
3. **Wheel**
4. **Steering arm**
5. **Tie rod**
6. **Rack and pinion**
7. **Rocker (or bellcrank)**
8. **Push rod**
9. **Suspension arm (or suspension linkage/ wishbone)**
10. **Upright**
11. **Toe link**

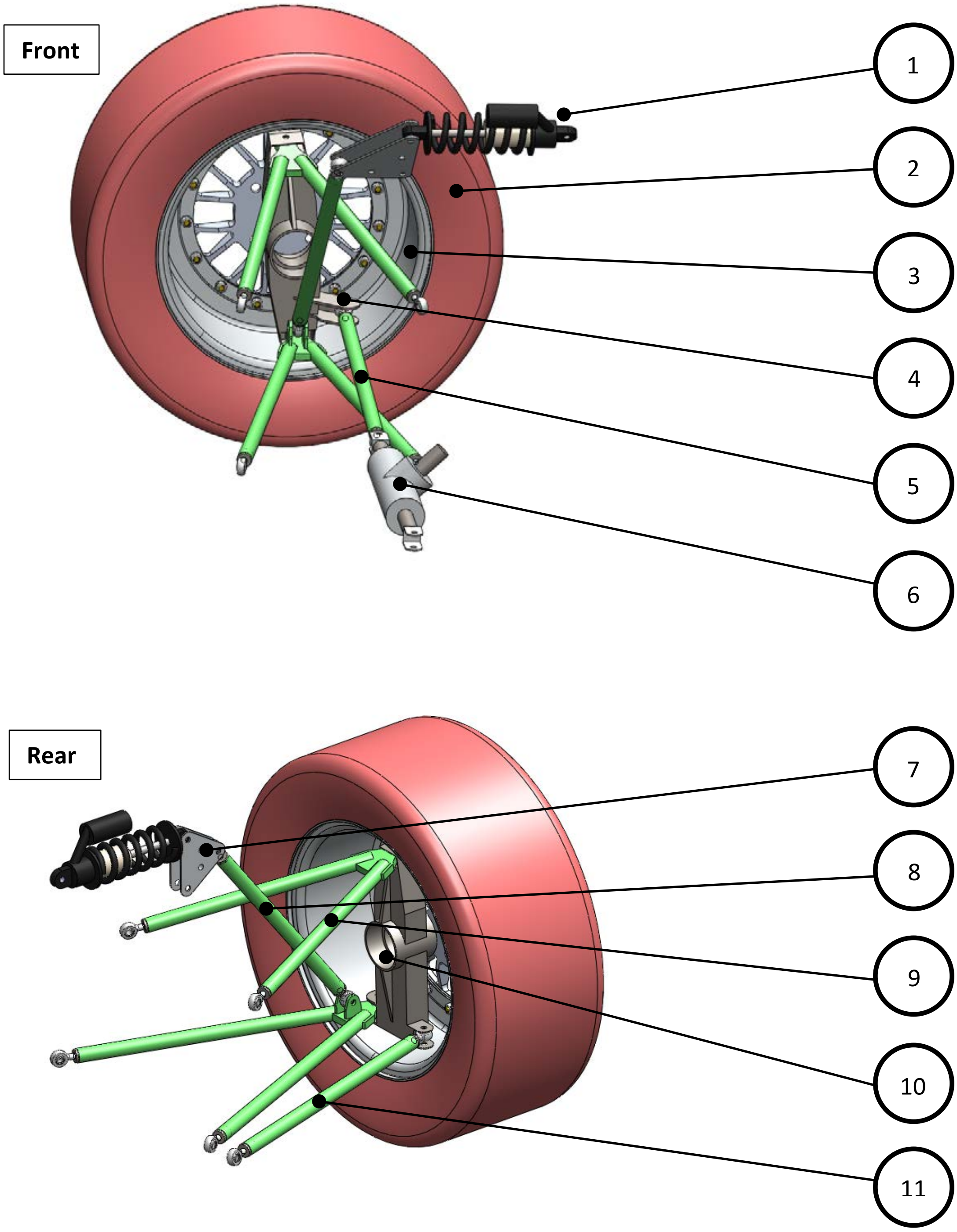


Figure 1.1: *Part list for an FSAE suspension and steering system*

1.5 Dissertation Overview

Chapter 1 has provided a brief introduction to the project and provided some insight into the motivation behind the topic selection. It has also described the project aims and objectives, defined the key elements of an FSAE suspension and steering system and has provided a summary of the FSAE competition.

Chapter 2 presents the project literature review in which fundamental concepts relevant to the design and analysis of suspension and steering systems, commonly used racing suspension and steering mechanisms, and lastly, design methods and recommendations regarding the suspension and steering systems, are all discussed.

Chapter 3 documents the findings of the 2008 vehicle analysis. This analysis aimed to uncover any issues that may have led to poor handling performance of the past vehicle in hope to find causes of the vehicle's crash at the 2008 FSAE-A competition.

Chapter 4 outlays the founding steps that establish a basis for design to begin. This involves specification of the project's design plan or methodology, allocation of performance targets for the final suspension and steering system and the documentation of the design decisions made regarding the type of suspension used, selection of tyres and wheels, and finally, the nomination of the track and wheel base dimensions.

Chapter 5 details the design of the suspension arm and upright geometry that was refined through iteration in *Wingeo3*, a suspension geometry program. For this iteration process, preliminary decisions related to the vehicles handling, methodology, result evaluation criteria and results are all discussed.

Chapter 6 looks at the design of the suspension actuation mechanisms along with the selection of shock absorbers and spring stiffness's. The work completed in this chapter carries on from chapter 5 where the *Wingeo3* suspension geometry model is furthered to include these suspension actuation devices.

Chapter 7 considers the steering geometry by which the location, general size and required ratio of the steering rack and pinion is defined. The chapter also draws on design documented in chapter 5 where the orientation of the uprights defined from the *Wingeo3* analysis is used to position the steering arms.

Chapter 8 summarises the physical component design for the suspension and steering system where the intended design concepts, materials and manufacturing processes for each part are all listed.

Chapter 9 discusses the findings of an evaluation of the designed suspension and steering system. The evaluation criteria applied is derived from the performance targets listed in chapter 4.

Chapter 10 summarises the analysis of project work and the conclusions drawn from such analysis. The chapter also provides insight into further work including completion of the full suspension and steering system design process documented in the design plan (chapter 4), more thorough design evaluation, and lastly, potential integration, manufacture and assembly of the system.

Chapter 2

Literature Review

2.1 Chapter Overview

As the author possesses very little knowledge in the way of vehicle design or motorsport engineering, the literature review process was crucial to the project's completion and success. The review analysed three areas related to the suspension and steering system and was aimed at forming a solid knowledge and skill basis for design to found off. These three areas concerning the suspension and steering system consist of the fundamental concepts regarding the analysis and understanding of these mechanisms, commonly used designs, and lastly, the processes employed to design these systems.

2.2 Objective of the Suspension System

The function of a suspension system for a road vehicle is quite simple. That is to reduce the shock and vibration experienced by occupants or cargo due to irregularities on the driving surface and to ensure all wheels maintain contact with the driving surface to promote stability and control of the vehicle (Bastow Et al, 2004, p3). From a more racing sort of view, Puhn (1976, p27) states that the suspension links the wheels of car to the chassis and aims to give the car the best possible handling qualities. Further explaining this phenomenon, Crahan (2004, p169) mentions that the tyres of a car that is being driven will experience a large degree of deformation by external and internal loads, and that the suspensions system is responsible for compensating for these deformations and loads in order to maximize tyre adhesion which is expected to provide improved handling performance.

2.3 Fundamental Concepts

2.3.1 Load Transfer

Unsprung Weight

The unsprung weight of a vehicle is the fraction of the total weight that is not supported by the suspension springs and will usually consist of the wheels, tires, hubs, hub carriers, brakes (if mounted outside the car's chassis), and lastly, roughly 50% of the weight due to drive shafts, springs and shocks as well as the suspension links. (Smith, 1978, p29)

Sprung Weight

This is basically the opposite of the aforementioned definition above. Again taking information from Carroll Smith's book entitled 'Tune to Win' (1979, p29) it is stated that the sprung weight is the portion of total car weight which is supported by the suspension springs. This weight is much larger than the unsprung weight as it consists of weight from the majority of car components which would include the chassis, engine, driver, fuel, gearbox and other components housed in the chassis.

Centre of Gravity (CG)

The definition of centre of gravity for a car is no different than that of a simple object such as a cube. Essentially, it is a 3 dimensional balance point where if the car was suspended by, it would be able to balance with no rotational movement. Recognising this concept, it is clear that the centre of gravity of the car will be located at where mass is most highly concentrated which for a race car is typically around the engine and associated drive components. It is also expected that all accelerative forces experienced by a vehicle will act through its centre of gravity. It is recommended that the centre of gravity for a vehicle be kept as low as possible to reduce the moment generated as the vehicle experiences lateral acceleration. (Smith, 1978, p29)

Polar Moment of Inertia

The polar moment of inertia is based from Newton's own laws of inertia and refers to the ease with which an object can be rotated about an axis. High concentrations of mass far from this axis will inhibit the rotation about the given axis where as if most mass is located at the axis location rotation will be easier (Crummey, 2011). Applying this concept to a car, the rotation axis is through the vehicle's centre of gravity, acting perpendicular to the ground plane and any mass concentrations distant from this axis in the plan view will affect the car's steering and cornering response. (Smith, 1978, p31)

Mass Centroid Axis

The mass centroid axis is found by dividing the car into a number of segments along its length and then calculating the centre of gravity for each of these segments before finally linking all these centre of gravity points with a line. This is obviously very hard to calculate and so generally a straight line approximation that gives an appropriate distribution of the car's mass in the vertical plane is applied. (Smith, 1978, p29)

Roll Centre

When a car experiences centrifugal cornering forces the sprung mass between both the front and rear axles will tend to rotate around a centre which is also located in a transverse plane to the axles. These points are called the roll centres and are the locations at which lateral forces generated by the tyres on the road will act upon the chassis. It should also be noted that the roll centre of the front and rear of the car are usually at different locations on the transverse planes defined by the car's axles. Figure 2.1 over the page details the process of finding the roll centre for the widely used four bar independent suspension system.

First, lines corresponding with the angle of the upper and lower linkages are extended until they meet at a point which is called the instantaneous centre. From this instantaneous centre a straight line is then drawn back to a point defined by the middle of the tyre's contact patch. Where this line meets the centreline of the vehicle is the roll centre. This is a simplified case though, with the roll centre will only moving up and down as the wheels move up and down where in reality it is found that the roll centre actually moves quite a lot and not just in the vertical axis. (Smith, 1978, p29)

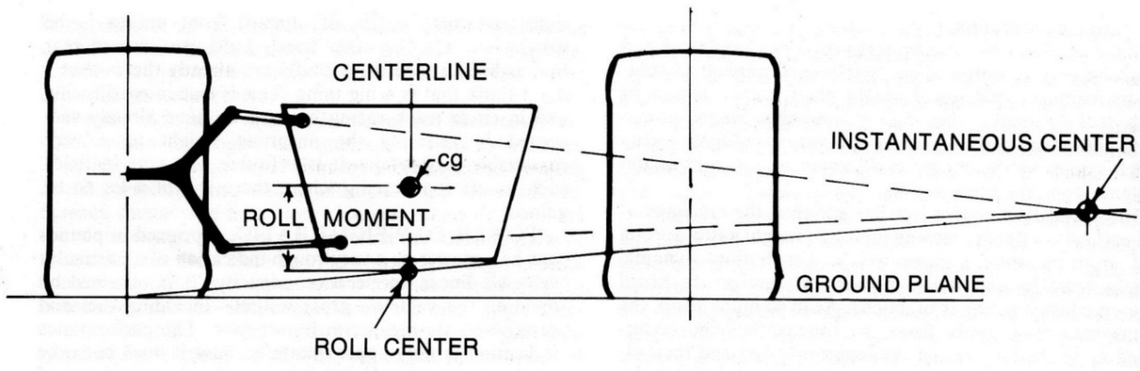


Figure 2.1: Determination of roll centre and moment arm. (Smith, 1978, p30)

Roll Axis

The roll axis is the line that would connect the roll centre at the front axle to roll centre at the rear axle. Building on the fact that front and rear roll centres will not always be at the same point at the front or rear of the vehicle, the roll axis will usually not be parallel to the ground plane. (Smith, 1978, p29)

Roll Moment

Also visualised on figure 2.1, the roll moment is the distance between the centre of gravity at the transverse plane defined by the axle, and the roll centre. In order to calculate the roll moment for the vehicle as a whole and not just either axle location, it is required to find the transverse plane that the overall centre of gravity of the car is located in and then at this cross section, determine the distance between the mass centroid axis and the roll axis. The relation of all these parameters can be observed over the page on figure 2.2 (Smith, 1978, p30)

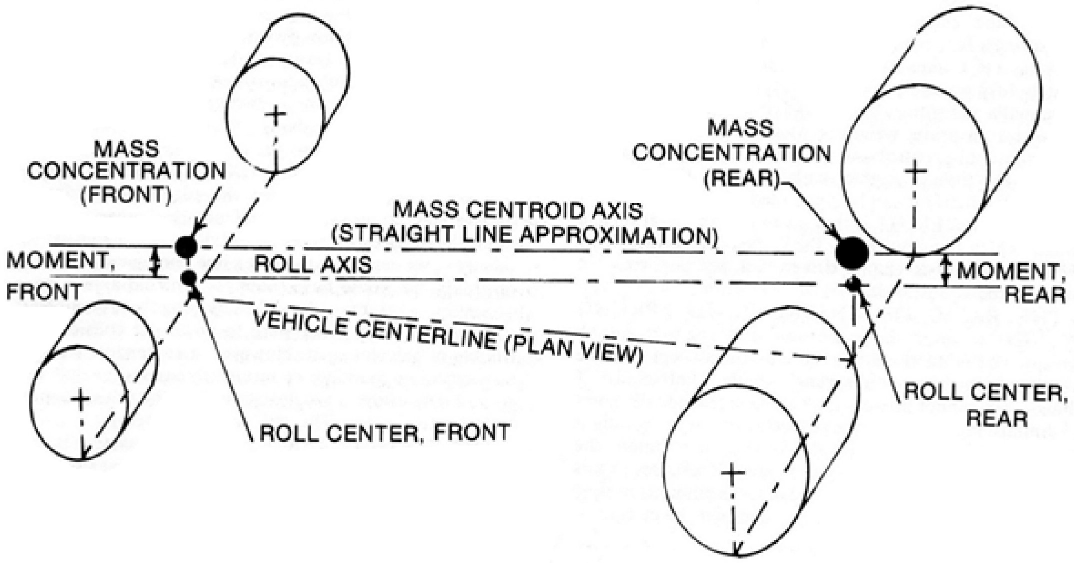


Figure 2.2: Relationship between roll axis, mass centroid axis and roll moments.
(Smith, 1978, p30)

Dynamic Load Transfer

According the Carroll Smith (1978, p31), dynamic load transfer "is the load transferred from one wheel to another due to the moments about the vehicle's center of gravity or its roll centers as the vehicle is accelerated in one sense or another."

Longitudinal Load Transfer

Longitudinal load transfer is the result of the cars mass accelerating from the front of the vehicle to the back or the back to the front under accelerating or braking respectively. It is important to mention that "The total weight of the vehicle does not change; load is merely transferred from the wheels at one end of the car to the wheels at the other end" (Smith, 1978, p29). The amount of load transfer that occurs is governed by the following formula which is also detailed by Carroll Smith:

$$Longitudinal\ load\ transfer(N) = Acceleration\ (g) \times \frac{Weight(N) \times cg\ height(m)}{Wheelbase(m)}$$

Note: Weight is defined as the weight that rests on the wheel set that is being analysed i.e. front or back and wheelbase is the distance between the centre contact patch of the front tyres to the centre of contact patch of the rear tyres.

Dive and Squat:

Dive and squat are fundamentally the same concept except reversed. Dive is where the front end of the dips down under braking due to the longitudinal weight transfer from the back of the car to the front acting on the front springs. Squat is where the back springs are compressed due to longitudinal weight transfer from the front of the car to the back which in effect causes the end of the vehicle to depress towards the ground plane.

Lateral Load Transfer

In essence the lateral load transfer experienced by a vehicle is the same principle as the longitudinal transfer only just rotated 90 degrees such that load is either transferred from the right to the left under a left hand corner and from the left to the right in a right hand corner. Similarly this load transfer can be calculated using the following formula defined by Carroll Smith (1978, p36)

$$Lateral\ load\ transfer(N) = \frac{Lateral\ acceleration\ (g) \times Weight(N) \times cg\ height(m)}{Track\ width(m)}$$

Note: Weight is defined as the weight that rests on the wheel set that is being analysed i.e. front or back and track width is the distance between the centre of the contact patch of the right and left tyres.

Bump and Droop:

Bump and droop are positions of independent suspension under certain scenarios. Bump occurs when the wheels hit a bump on the track surface whereas droop occurs when the wheels drop into a depression in the track surface. Bump and droop movements also associate with the suspension travel terms, rebound and jounce where jounce describes the upwards movement of the wheel or movement in bump while rebound describes the downwards travel of the wheel or droop movement. These principles are best seen on the figure below with the bump condition on the left and the droop on the right.

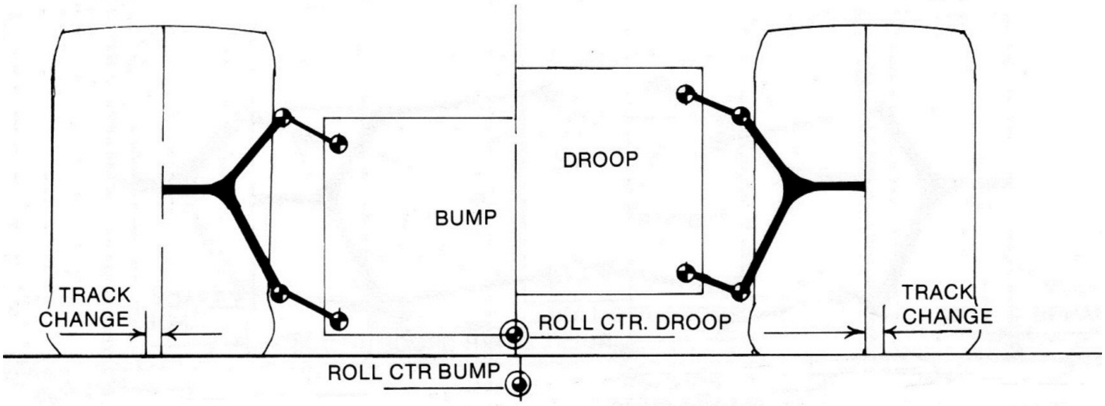


Figure 2.3: Bump and Droop behaviour of double wishbone set up. (Smith, 1978, p51)

Jacking:

Any vehicle possessing independent suspension with its roll centre above the ground plane will exhibit some extent of jacking and is where the car will appear to lift itself up while cornering. This effect may be visualised on the following figure and occurs when the reaction force acting on the tyre acts through the roll centre to balance the centrifugal force generated as the car is turning. This effect is highly undesired as it raises the centre of gravity and places the suspension linkage in the droop position which results in poor tyre camber, in effect, hindering the tyre's adhesion to the track surface. This phenomenon is experienced a lot more significantly in vehicles possessing a high roll centre and narrow track width. (Smith, 1978, p38)

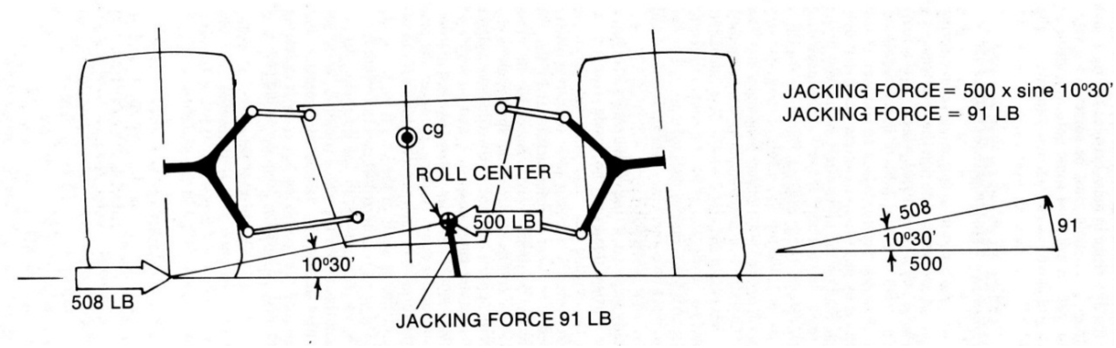


Figure 2.4: Calculating jacking force for a double wishbone set up. (Smith, 1978, p39)

2.4 Modern Day Suspension Configurations

Springs and Dampers

Quoted directly from (Smith, 1978, p64), "In order to make the contact between the tires' contact patches and the track surface as continuous as possible and to avoid shaking the car and/or driver apart, racing cars must have some sort or other of springs." Expanding on this, (Gran Turismo, 2010, p101) states that the springs are responsible for keeping the car body at a constant height and are an important factor in the handling and stability of a vehicle. Currently there are four types of spring commonly utilised in cars which are the coil, leaf, torsion bar and air springs. However, the most popular variation applied in race cars is the coil spring. (Longhurst, 2011)

The stiffness of a spring coupled with the geometry of the suspension will define the wheel rate of the vehicle. The wheel rate of the vehicle is the rate at the wheel moves up and down vertically and is essentially the spring rate measured at the wheel. (Smith, 1978, p64)

Dampers and springs go hand in hand; the springs absorb shocks whereas the dampers dampen the energy stored in the springs as they absorb these shocks. Without dampers the vehicle body would continue to oscillate up and down at its natural frequency after travelling over a disturbance in the road, as when compressed the springs store large amounts of kinetic energy which when released, forces the springs to extend back to their full length. This force is sometimes strong enough to put the vehicle's wheels in full droop. Where dampers come in is then to stop this post bump extension of the springs such that the car's body stays at a roughly constant height. (Smith, 1978, p74)

The damper achieves its function through the use of oil or gas which is forced (as the spring compresses or extends) through a small valve which is often adjustable to alter how stiff the suspension performs. (Longhurst, 2011)

Anti-Roll Bar

The anti-roll bar or anti-sway bar is a type of spring which is often incorporated into a suspension design where higher roll stiffness is required than is able to be supplied by the existing springs that act on each individual wheel. It can only be applied to independent suspension systems and mounts to both ends of the lower suspension arms. The bar is also constrained by mounts featured on the chassis which allow rotation of the bar as the car wheels oscillate up and down. As a vehicle navigates a corner the car will tilt toward the outside of the turn as the suspension on that side of the car experiences the largest forces. What the anti-roll bar aims to do is equalise the amount of force shared by the suspension systems on each side of the car so the car body doesn't roll as much. With appropriate adjustment, the anti-roll bar can be adapted to counter understeer or oversteer. For a simple representation of the anti-roll bar in action, please view figure 2.5 below. (Gran Turismo, 2010)

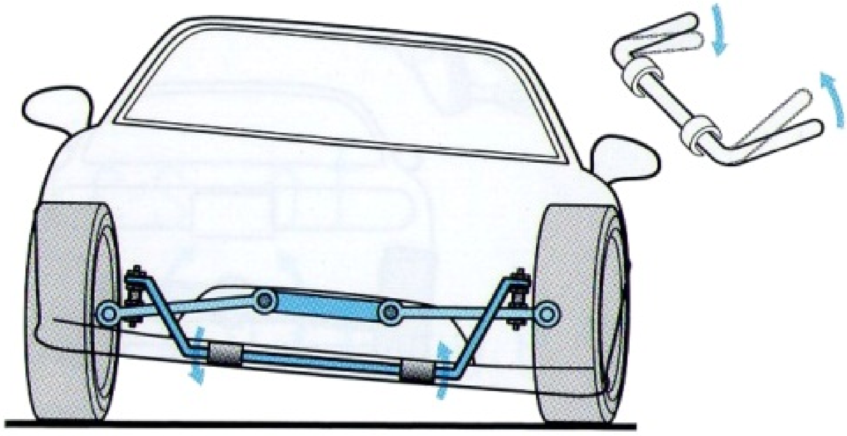


Figure 2.5: Typical anti-roll bar in action on a cornering vehicle. (Gran Turismo, 2010)

Front Suspension Mechanisms

All cars used in track racing and a large majority that are used on public roads all employ independent front suspension. The two most commonly used of these independent suspension types are detailed in the proceeding sections.

Double Wishbones

The double wishbone or four bar linkage suspension configuration is probably the most widely used racing suspension design and also makes up a significant proportion in the domestic market. Its operation is quite straight forward and can be shown on figure 2.6 below. The ends of the two wishbone arms and top end of the shock absorber will mount to the chassis. Here it is seen that as the wheel moves up, the shock absorber is compressed thus reducing the effect of forces induced by the ground surface that are felt by the chassis. To allow the wheels to be steered, the wishbone arms feature ball joints on the top and bottom so that the upright can pivot and rotate as needed.

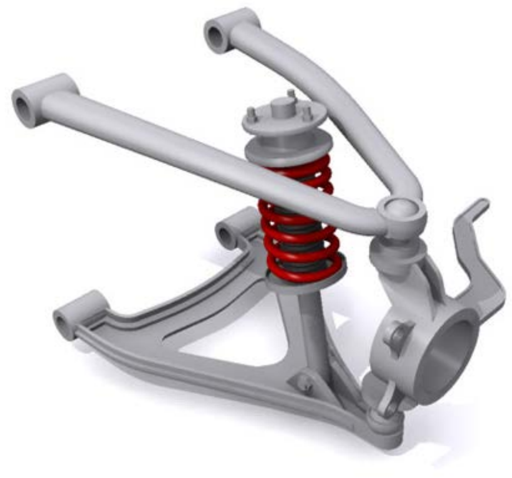


Figure 2.6: Typical double wishbone suspension layout. (Longhurst, 2011)

This design has a number of benefits including the fact that it provides a large amount of room for adjustment, allows decent tyre camber control resulting in enhanced handling characteristics, has high strength and rigidity, if an impact occurs and the suspension suffers damage it is unlikely that all the components will need replacing, and finally, it permits a low unsprung weight for the vehicle as only a small portion of the linkage weights are unsprung.

On the other hand, the double wishbone also holds a number of disadvantages which comprise of relatively higher build and installation costs, large lateral space requirements and the fact that they can sometimes be quite heavy which adds to the sprung weight of the car. (Severson, 2009)

Under the double wishbone configuration there are also a number of geometry variations that can be used to alter the vehicles handling properties. These variations include:

Equal Length and Parallel Arms:

This geometry is created when the upper and lower wishbone linkages are made the same length and thus form a parallelogram. As the wheels move up and down there is no wheel camber change but there is notable track width change. Further still, when the vehicle's sprung mass rolls a certain amount, the camber will change by the exact same amount with the outside wheel cambering in the positive direction. This condition is not to be desired as the contact patch of the tire becomes reduced, diminishing the amount of grip available to the vehicle. For this arrangment the roll centre is taken to remain at ground level and to stay there under suspension actuation. (Smith, 1978, p47)

Unequal Length and Parallel Arms:

As the name denotes, this design is where the arms are of unequal length but still remain parallel. The upper link is typically the shorter one in order to induce a negative camber angle when the car hits a bump and either a negative or positive camber when the linkages go into droop. The amount of camber change will be governed by the relative lengths of the upper and lower linkages. Like before, the wheels are forced into camber angles defined by the roll direction of the car however this time the positive camber of the outside wheel is reduced and the negative camber of the inside wheel increased. Roll centre movement under these conditions will remain fairly small and consistent, thus roll moment will remain fairly constant as well. Additionally, the location of this roll centre will generally be very low. (Smith, 1978, p47-54)

Unequal Length and Non-parallel Arms:

The third and most commonly used set up in racing is the unequal and non-parallel arm design. It goes a step on from the unequal and parallel configuration where, in the static position of a vehicles suspension, linkages are different lengths and non-parallel to each other. In doing this, the design allows even better camber control of the wheels and allows the designer to locate the roll centre wherever deemed appropriate. (Smith, 1978, p54)

For graphical representation of these three variations please consult figure 2.7 below.

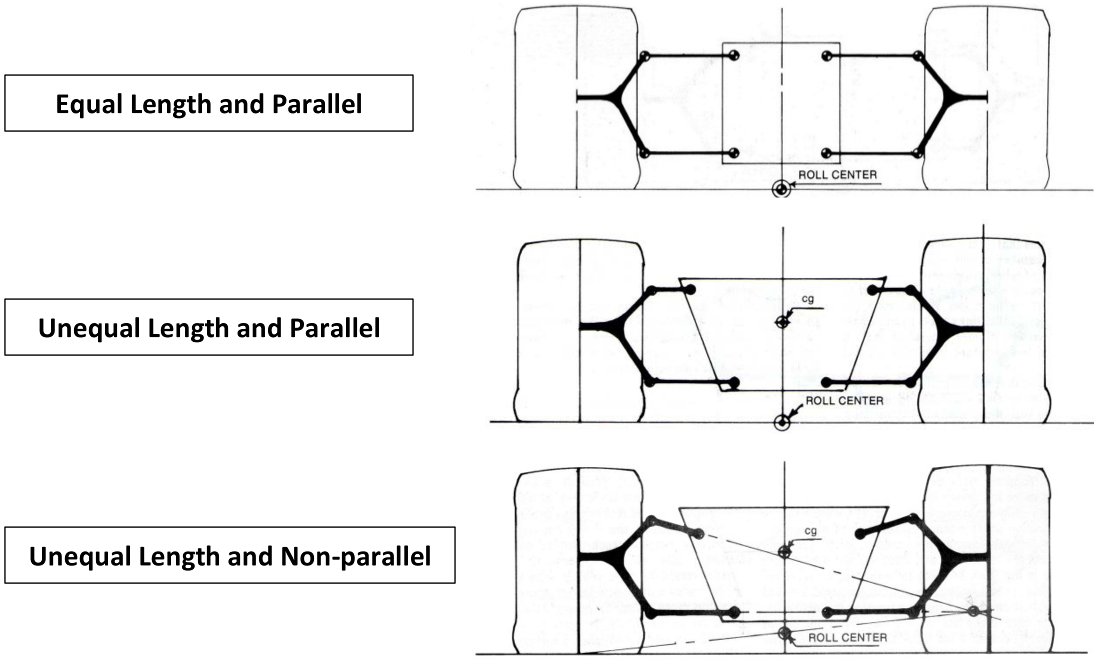


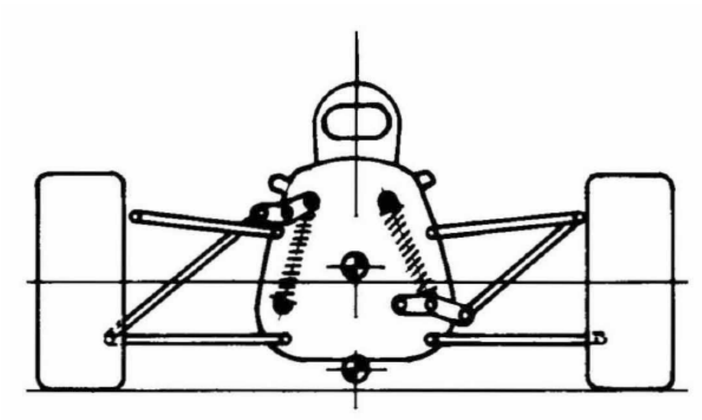
Figure 2.7: Different variations of the double wishbone suspension arrangement. (Smith, 1978, p47-54)

Outboard and Inboard Shock Absorber Positioning:

Outboard and inboard position of the shock is as simple as it sounds; an outboard design places the coil over outside the body of the vehicle (as shown in figure 2.6) whereas an inboard configuration allows the shock absorber to be place inside the car body or chassis by using appropriate actuating rods and rocker arms. The latter method presents a number of benefits to the original inboard mounting techniques for reasons including a reduction of vehicle coefficient of drag by taking the coil overs out of the air stream around the car, improved wheel rate control along with ride height adjustment by employment of suitable rocker arms and rods, and lastly, greater flexibility in where the shocks are positioned. (Staniforth, 1991, p79-80)

Push and Pull:

Currently there are two main approaches to designing the inboard suspension system which are the push and pull variations. These may be viewed on the following figure and as seen, will operate using the same fundamental principles whereby up and down wheel movement is transmitted to the shock absorber by means of a rocker arm. What type of mechanism is used will depend on the layout of the vehicle and the desired loading paths for the suspension design; one method may integrate better much better than the other. It is therefore not uncommon to see a vehicle utilising a push system at one end of the car and a pull at the other. (Staniforth, 1991, p80)



Push **Pull**

Figure 2.8: Push and pull inboard suspension configurations. (Staniforth, 1991, p62)

As can be seen above, the push set up pushes on the rocker to actuate the shock absorber whereas the pull method pulls on the rocker.

MacPherson Strut

Not quite as widely used as the double wish bone setup in racing, the MacPherson strut configuration is the most commonly employed design in the domestic market (Bastow Et al, 2004, p197). It gets its name from a Ford suspension engineer who patented the design during the 1950's. According to Longhurst (2011), author of 'The Suspension Bible', "The system basically comprises of a strut-type spring and shock absorber combo, which pivots on a ball joint on the single, lower arm… The strut itself is the load-bearing member in this assembly, with the spring and shock absorber merely performing their duty as oppose to actually holding the car up." This ball joint permits steering which is also accommodated by a needle bearing above the shock absorber assembly. The MacPherson configuration is observed below on figure 2.9.

Figure 2.9: Typical MacPherson strut suspension layout. (Longhurst, 2011)

The MacPherson strut doesn't feature as many relative advantages as the double wishbone arrangement but still has a number of notable qualities such as being comparably low cost, requiring less space in plan view, providing high strength and rigidity, and lastly, promoting a lower unsprung weight. The down sides to this design include that the mechanism cannot be used on cars where vertical room is deprived due to significant height of the strut, it generally cannot be applied to body on frame type vehicles as the strut requires a strong mounting point, and finally, the fact that the wheels do not gain camber as the suspension actuates, reduces the handling capability of the vehicle. (Severson, 2009)

Rear Suspension Mechanisms

Although a large array of commercial and domestic vehicles still use solid rear axles, the move in recent years has been to utilise independent suspension on all four wheels. The most widely used of these systems that are specifically aimed at a rear wheel driven car will be detailed below.

Trailing Arm

The trailing arm suspension design uses the same fundamental concepts as the double wishbone setup although rotated 90 degrees so that the axle position is behind the holding points for the suspension linkages. One benefit the system has when in use on the rear of a car is the fact that it does not affect the path of the tyre in the lateral direction as the suspension linkages are parallel to the length of the vehicle and thus front the front or rear of the car the rear wheels will only appear to move up and down and have no apparent rotation. However, it is important to realise that from a side view it is apparent that the wheelbase will alter as the suspension moves up and down. This setup may be seen below in figure 2.10.

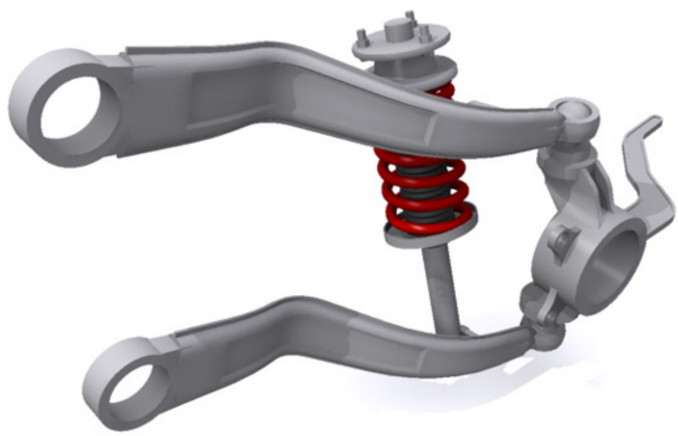


Figure 2.10: Typical trailing arm suspension layout. (Longhurst, 2011)

Semi Trailing Arm

The semi trailing arm suspension mechanism pictured in the following figure is a transformation of the trailing arm design. This change from the normal trailing arm has been to improve the adjustability of the kinematic characteristics particularly in rear wheel applications. Observing figure 2.11 it is noted that the axis which the arm pivots around is angled back towards the centre of the car rather than perpendicular to the length of the car as used in the trailing arm configuration. This results in the change of steer, camber, track, and wheelbase as the rear wheels move up and down. (Bastow Et al, 2004, p217-218)

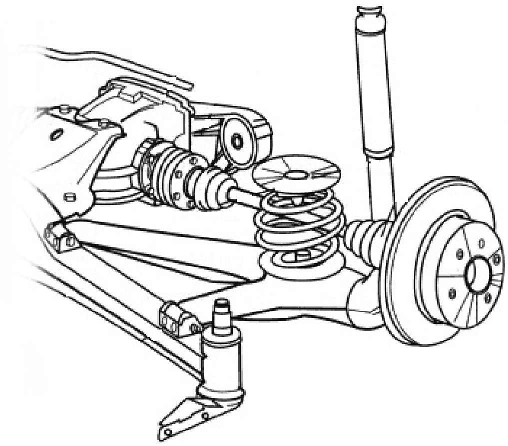


Figure 2.11: Typical semi trailing axis suspension layout. (CadillacOwners.com, 2009)

Multi-Link

Multi-link suspension configurations are in no way as developed as the double wishbone or trailing/semi trailing arm configurations that are used today however their implementation in modern vehicles has seen some very good performance. These systems take the basic double wishbone set up and add arms, modify mounting locations and in some cases, add extra pivot points in the linkage system itself. As there is no set standard for these multi-link mechanisms it is quite hard to explain how they function. One of these systems which is produced by Mercedes Benz on the SL500 model is shown over the page on figure 2.12. This design aims to provide a large degree of adjustability without compromising certain suspension characteristics. (Bastow Et al, 2004, p224-225)

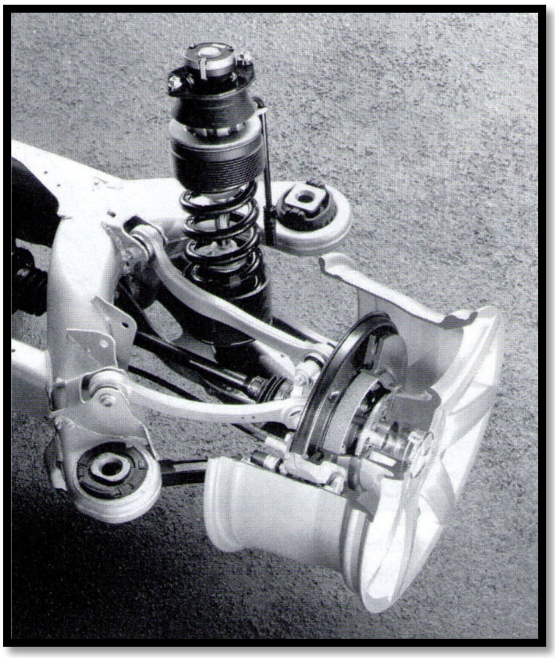


Figure 2.12: Single rear-wheel suspension of the Mercedes-Benz SL500. (Bastow Et al, 2004, p225)

Double Wishbone

As explained earlier, the double wishbone mechanism is a very commonly used design. The principle is exactly the same except at the rear of the vehicle, although steering does not need to be considered. However, if the rear axle is to be driven suitable consideration will need to be made to ensure this driving axle is flexible so to allow for the suspension movement. This is generally achieved using CV or constant velocity joints.

2.5 Objective of the Steering System

According to Bastow Et al (2004, p83), "The function of the steering system is clearly to afford the driver directional control of the vehicle, and to provide this control with sufficient accuracy to choose the best course around corners, to avoid other vehicles and stationary obstructions, and to manoeuvre the car efficiently at low speed…" This view is also supported by the author and is believed applicable to domestic and commercial vehicles as well as racing cars.

2.6 Fundamental Concepts

2.6.1 Geometry

Camber Angle

As shown on the following figure, the camber angle is the angle that the inclination of the vehicle's tyres makes with the vertical axis. In this case the camber is negative as the top of the tyre leans in towards the centre of the car. A positive camber is the opposite of this. According to Bastow Et al (2004, p10), increasing positive camber will enlarge the slip angle for a specific cornering force, decrease the largest possible cornering force possible by the vehicle but will also slow down the onset of 'breakaway' which is assumed to mean the car starting to slide. On the other hand, by increasing negative camber the opposite will occur with a higher cornering force and less time for the car to break away. It should also be mentioned though that generally vehicles will be designed with a relatively small camber angle statically applied.

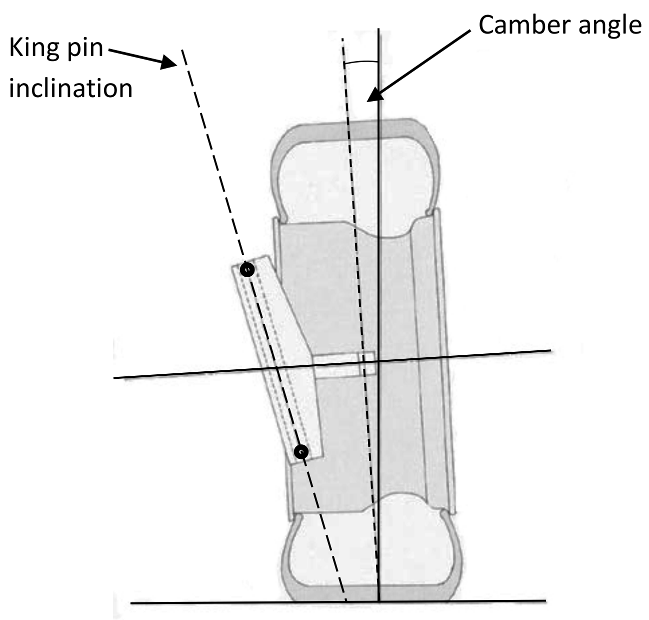


Figure 2.13: Camber angle and kingpin inclination. (Kalver, 2001)

Kingpin Inclination

The kingpin inclination of a vehicle may also be seen on figure 2.13. Taking guidance from Bastow Et al (2004, p10) again, it is stated that the "kingpin inclination (often shortened to KPI) is the angle, viewed in end elevation, between the vertical and the steering (kingpin) axis." In figure 2.13 a positive kingpin inclination is shown because convention is opposite to the camber angle; positive kingpin is when the kingpin axis angles in towards the centre of the vehicle whereas negative inclination is the opposite. Another key parameter that is linked to kingpin inclination is kingpin offset or scrub radius. It is the lateral measurement between the meeting point of the centre of the tyre's contact patch and the kingpin axis, with the ground plane. Convention is that positive offset will be when the kingpin offset is outboard of the kingpin axis.

Caster Angle

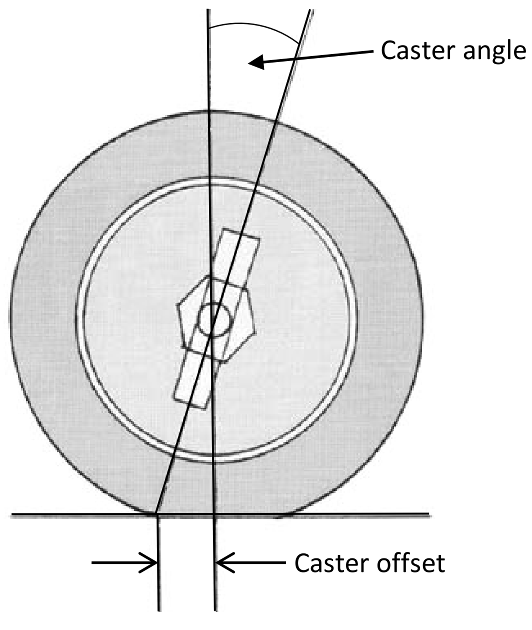


Figure 2.14: Caster angle and offset. *(Kalver, 2001)*

Caster angle also relates to the kingpin or steering axis although describes the angle of it when viewing the vehicle from side on. As seen above, it is the angle that the kingpin axis makes with the vertical. It is positive when the kingpin axis meets the ground ahead of the vertical axis drawn through the wheel centre so in figure 2.14 back over the page, if the car is assumed to be facing right, the caster angle is therefore negative. In a vehicle with negative caster angle, the steering wheels will tend to self-align as the vehicle moves forward. Another term that is often associated with the caster angle is the caster offset or mechanical trail which is also seen on figure 2.14. (Bastow Et al, 2004, p11)

Toe Angle

Toe angle is the angle that a wheel makes with a line drawn parallel to the length of the car, when viewed from above. This concept is seen below on figure 2.15. When the front wheels point away from each other, the condition is called toe out whereas when the front wheels point inwards, the vehicle is said to have toe in. Generally designers will opt for toe in for the reason being that when the vehicle experiences an upsetting force such as a bump or a wind gust, the toe in will promote stability as the front wheels naturally want to steer into a location central to the car's body. Toe out on the other hand will produce some very unstable behaviour under these conditions when the slip angle of the more heavily laden wheel increases. In general, toe in will provide greater straight line stability whereas a controlled amount of toe out can improve the car's turn in ability to a corner. *(Smith, 1978, p38)*

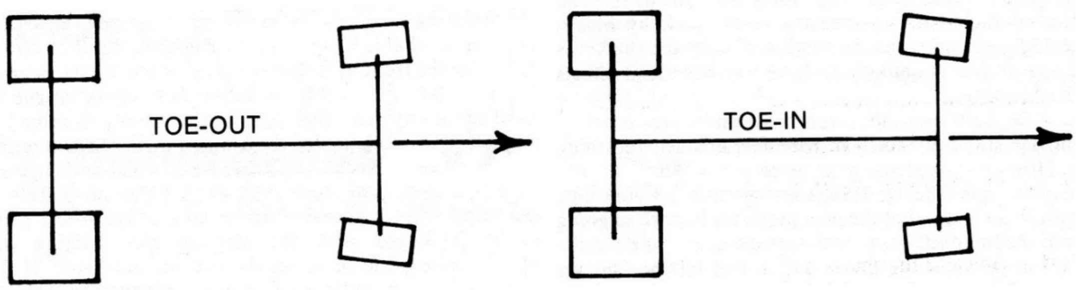
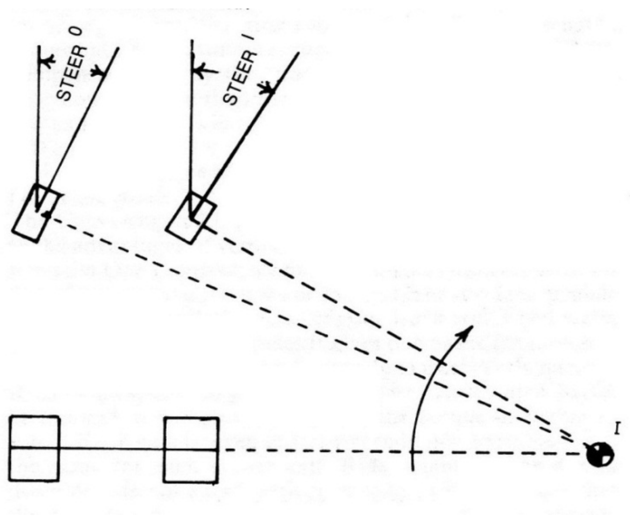


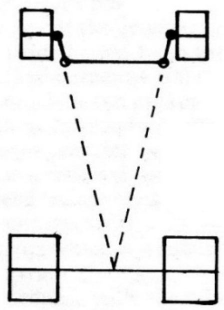
Figure 2.15: Toe angle settings. *(Smith, 1978, p38)*

Ackermann Geometry

The Ackermann steering geometry takes its name from a London agent that patented the design in 1816. The geometry allows the outer front wheel to cover a larger radius than the inside wheel. As a result both wheels will follow individual radii without skidding or scrubbing as the vehicle corners. This effect may be over the page on figure 2.16 (a). In order to achieve this geometry, the steering arms on the front wheels must angle inward and meet at the centre of the rear axle (figure 2.16 (b)). These days only a select number of vehicles will employ full Ackermann geometry. This is because as cornering speed increases, the wheels of a car will adopt slip angles and effects of the Ackermann geometry will become obsolete. For these reasons only slower vehicles that require restricted turning circles will be the ones to use this full geometry whereas the majority of all modern vehicles will only utilise a small amount of Ackermann compensation.



(a)



(b)

Figure 2.16: Ackermann geometry. *(Smith, 1978, p60)*

2.6.2 Steering Behaviours

Bump Steer

Bump steer is the phenomenon that occurs when the front wheels of a vehicle vary their toe angle's as the suspension moves up and down, potentially causing the car to dart around even under no steering input resulting in highly unwanted driver uncertainty and poor handling feel (Staniforth, 1999, p190). According to Smith (1978, p62), when designing around bump steer, large amounts of toe out occurring due to the suspension movement should be avoided at all costs. He also mentions though, that bump steer can be used to the designers benefit by altering the response of the vehicle while cornering.

Roll Steer

Roll steer is also another steering effect that is generated through movement of the suspension system. It is believed that this concept is best clarified by Smith (1978, p63) where he mentions roll steer is "…the self steering action of any automobile in response to lateral acceleration and consists of slip angle changes due to camber change, toe change and the inertias of the sprung mass." This effect will be present in all double wishbone set ups although can be limited by reducing the gross weight of the car, centre of gravity height, eliminating deflection in the suspension and associated chassis mounting components, and lastly, by adjusting bump steer.

Slip Angle

Slip angle is the angle made by the direction of the tyre contact patch with the direction of overall velocity of vehicle. This principle is best demonstrated by observing figure 2.17 below which also highlights the lateral forces imposed on the wheel as it corners.

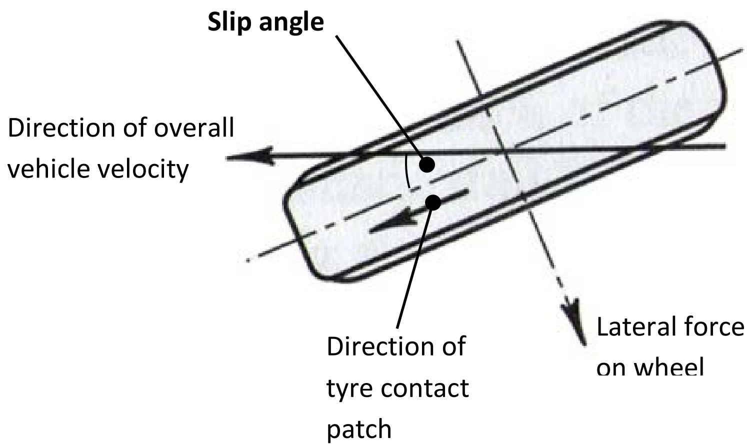


Figure 2.17: Slip angle. (Milliken and Milliken, 2002, p54)

Oversteer, Understeer and Neutral Steer

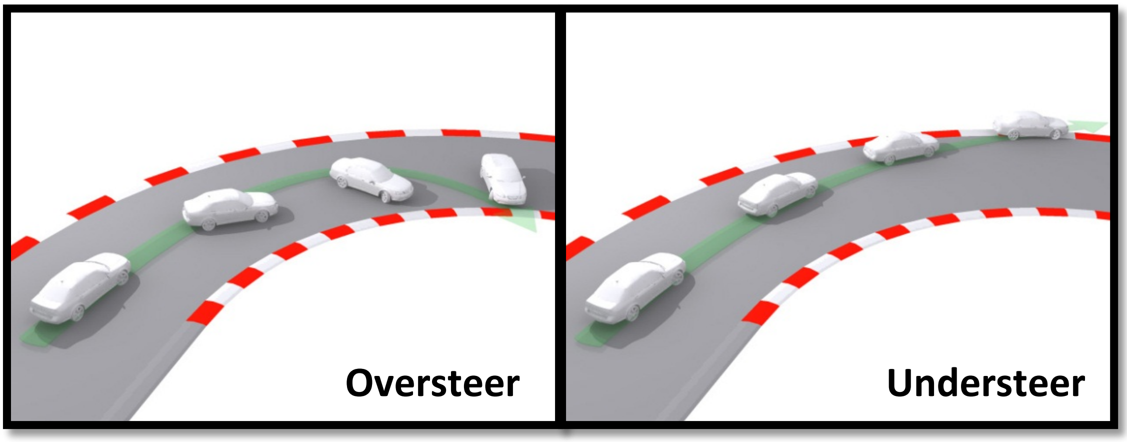


Figure 2.18: Oversteer and understeer effects on a vehicle. (Longhurst, 2011)

Olley (1947) describes oversteer as the occurrence when the front wheel slip angles are smaller than the rear ones and for understeer, when the front wheel slip angles were larger than the rear. Building on this, Longhurst (2011) mentions that oversteer is where the car loses grip at the rear wheels causing the car to turn more than expected by the driver whereas understeer induces the opposite of this behaviour. These concepts can be pictured above on figure 2.18. It should also be mentioned that when the slip angles for the front and rear wheels are equal, then the vehicle is said to be neutral steering.

2.7 Modern Day Steering Configurations

Currently there are 2 main designs used for steering mechanisms in racing vehicles, these are the Pitman arm and the rack and pinion. Each of these designs has a number of slight variations in order to tailor it to the various applications in which they are used.

Pitman Arm

Out of the two above mentioned configurations, the Pitman arm is the lesser used for a number of reasons. These include poor steering feel, a large degree of play or 'slop' in the mechanism's actuation and the fact that it is relatively complex. As seen on the following figure, the mechanism inputs rotational movement from the steering wheel and passes it through a box which houses suitable components in order to convert this rotation around the steering shaft, to rotation around a horizontal axis perpendicular to the box. This converted rotation swings the Pitman arm side to side depending on the given steering input. The end of the Pitman arm connects to a link that then converts the Pitman arm swing into linear movement of the track and tie rods which effect, steers the wheels of the car. Shown below on the second figure (2.20) is the most commonly used mechanism in the Pitman arm 'box' which converts the rotation of the steering wheel to rotation of the Pitman arm. This is the worm and recirculating ball design. (Longhurst, 2011)

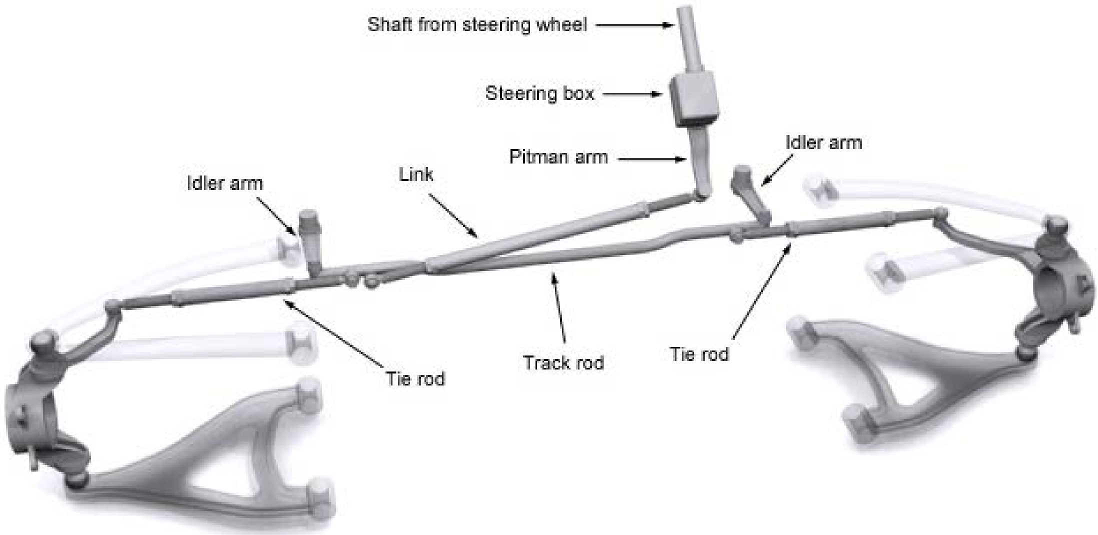


Figure 2.19: Typical Pitman arm steering arrangement and associated components. (Longhurst, 2011)

Figure 2.20: Worm and recirculating ball steering box. (Longhurst, 2011)

Rack and Pinion

The rack and pinion mechanism is bar far the most commonly used design today. This is due "to it's relative simplicity and low cost." and the fact that they "give a much better feel for the driver, and there isn't the slop or slack associated with steering box pitman arm type systems." (Longhurst, 2011). Where the Pitman arm design featured a track rod in between the tie rods, the rack and pinion set up consists of a toothed bar. Meshed to this bar and contained within the steering rack housing is the pinion gear which is then attached to the steering shaft that links the whole steering mechanism to the steering wheel. This arrangement may be observed below with the top figure demonstrating the rack incorporated in a full steering assembly and the bottom defining the actual mechanism.

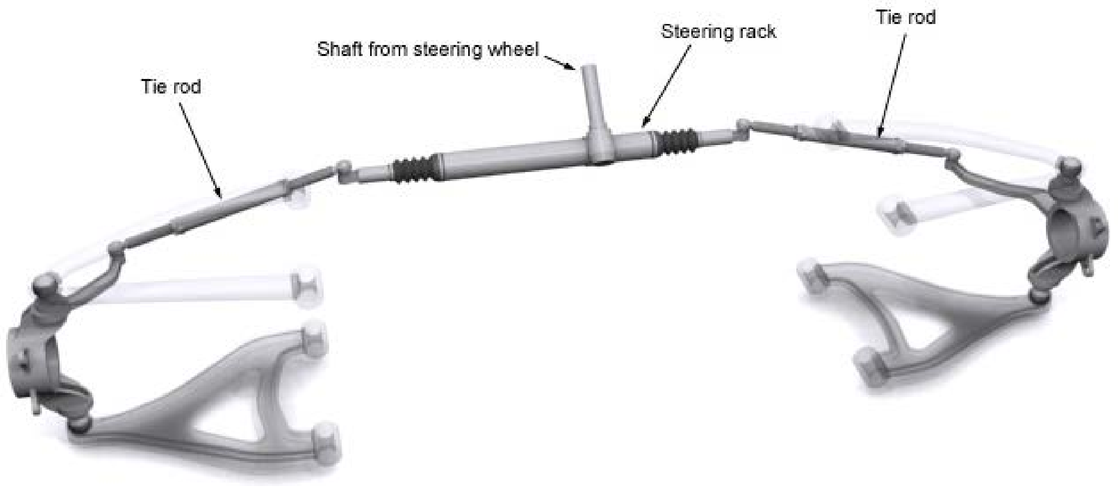


Figure 2.21: Typical rack and pinion arrangement and associated components. (Longhurst, 2011)

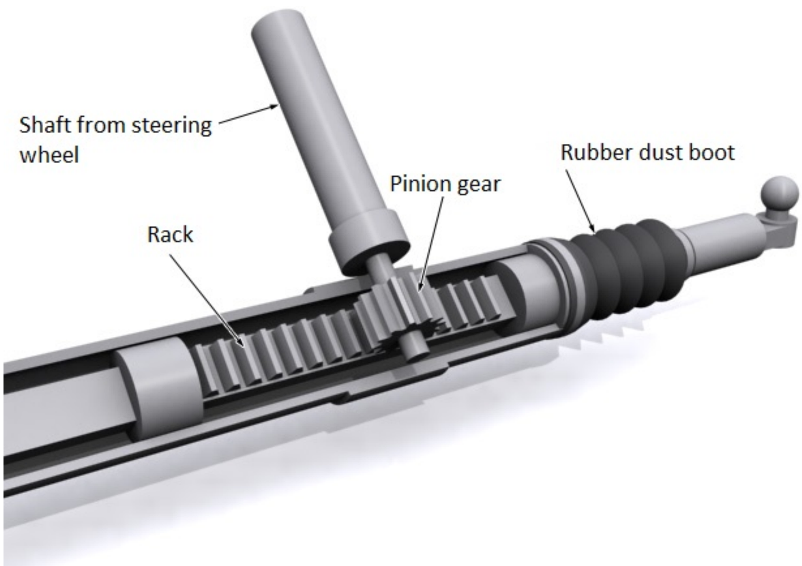


Figure 2.22: Rack and pinion mechanism. (Longhurst, 2011)

By changing the pinion and rack gear ratio, the steering ratio is also able to be adjusted and this is the only method of adjusting the mechanism. It is clear that this system is a lot simpler than the previously mentioned Pitman arm configuration and is another reason why rack and pinion is more commonly used for all applications. (Longhurst, 2011)

2.8 Existing Design Procedures

As the author's knowledge on motorsport engineering and vehicle mechanics is quite limited, a large part of the project research was aimed at uncovering information regarding the structure and process of designing a suspension and steering system from the beginning to end. Upon concluding the research it was observed that there were a large number of documents detailing the design of the suspension and steering systems however out of these only a few provided thorough guidance on this process.

Allan Staniforth's Complete Design Guide

In a book entitled 'Competition Car Suspension', Allan Staniforth delivers a very detailed, step by step guide for the design of a vehicle's steering and suspension systems. This guide is very suitable for the project as it is written for the use of an amateur suspension designer. The following list provides a summary of Allan's recommendations in order they are to be completed.

1. **Regulations**

   Here it is simply mentioned to make sure that the designer has a thorough understanding of the rules and regulations that will relate to the application of which he/she is designing for.

2. **Tyres**

   Next step is to select the tyres to be used based off desired handling characteristics, weight considerations and regulations defined by the given competition rules.

3. Wheels

Obviously the choice of wheel will be limited by the selection of a tyre and thus will also depend on the design considerations mentioned in the previous step.

4. Hubs and Uprights

Following selection of the wheels it is then recommended to design the hubs and uprights. Essentially these components are separate parts but Staniforth reinforces that they are very closely inter-related and thus can be designed in one step. Key features that the designer will have to select include the wheel offset or inset, method of attachment for the wheels as well as the location of the upper and lower suspension pick-up points. An experienced designer will commonly design these components with the geometry desired and then simply manufacture the parts before carrying on with the rest of design whereas an amateur is more likely to create parts with flexibility down the track in mind by making components easily adjustable and simple to physically modify.

5. Geometry

Further design of the suspension system is unable to proceed without the identification of the suspension linkages' pickup points, lengths and angles. Here Allan notes that keeping the centre of gravity and roll centre low is a major goal and specifically mentions that the roll centre should form the basis to start the geometry design from. To aid in this analysis Staniforth also suggests a number of alternative methodologies available to design the geometry which are as follows:

i) Copy an existing design by means of either receiving design information from applicable friendships or contacts or alternatively purchasing the designs. This method requires that the majority of the design will mirror the copied designs otherwise problems will be incurred later down the track when adaptations have to be made in order to finish the overall design for the chosen application.

ii) Draw the initial proposed layout before redrawing the same configuration many times under different suspension movements. Allan makes particular note that this process is highly in-efficient and is rarely used in practice.

iii) The next option basically uses a computer program to carry out the above mentioned process, saving significant amounts of time and resources.

iv) Use a string computer. This was one of Allan's creations and essentially consisted of a full scale model of an unequal, non-parallel double wishbone configuration which fully simulated the suspension movement of the car. A string length was used to derive the behaviour of the roll centre.

6. Springs

Allan suggests the springs or coils for the car should be selected by analysing the required coil rate, the leverage on it, and the sprung weight of the car that will rest on them. He also places a recognisable amount of emphasis on the frequency of the suspension system and the role this plays on selecting the right spring stiffness.

7. Dampers

According to Staniforth, "The precise relationship between a damper, the coil surrounding it and the rest of the car is an extremely subtle and sensitive one, even in this day and age often being fine tuned by testing and "seat of the pants" feel once the car is running." He also dictates that all the damper sees and has to deal with is the coil and thus is unaffected by inclinations, wheel rate, etc. Knowledge of this relationship between damper and coil will obviously vary between different designers and it is expected that a damper will be selected accordingly based off these understandings and beliefs.

8. Anti-Roll Bars

Allan notes that there is only a certain amount that the springs of a vehicle can resist roll as the coils can only be made strong to a specific limit where manufacture or design integration becomes impossible. Following this he suggests the use of a bar to improve the cars stiffness without altering the springs excessively. The first step in the bar selection and design process is to decide where the bars will be placed in relation to the overall car layout. Considerations when positioning the bars will include clearing the driver's legs, gearbox and other key components in the vicinity, the ability to mount on rigid, low friction locations, accessibility for adjustment, and lastly, to avoid fouling on any other parts of the vehicle.

The next step recommended is to calculate the car's roll moment before deriving the stiffness provided by the springs. Once these values are known, Staniforth insists that the required amount of roll should be chosen and then the designer should work back in order to generate the required stiffness of the bars. He states that under cornering force of 1g, the maximum permissible roll angle should be 2.5° for a saloon car and 1.5° for a single seater race car under the assumption that weights and leverage ratios are within check of those used in a typical saloon or single seater.

Once the required stiffness is known the designer can move on to select appropriate bar geometry to achieve this required stiffness. Key dimensions for the bar will include the diameter and wall thickness.

9. **Steering**

The major decision to be made by the designer regarding the design of the steering mechanism is to select the location of the rack and pinion. This location will be influenced by the need to protect and avoid the driver's legs and to avoid significant amounts of bump steer. Three methods to achieve this goal are then presented in graphical form. For full view of these methods please consult appendix B.

Drexel University 2003 Formula SAE Team's Design Approach

Although brief, the method outlined in a sample proposal document for the 2003 Drexel University Formula SAE team (2003) demonstrates a structured plan towards the design of the suspension system for their car. The process reads as follows:

1. **Determine the general type of suspension**
2. **Determine the vehicle track width**
3. **Determine the tire size and rim diameter**
4. **Determine the suspension geometry**

- Mounting location of the suspension to the chassis
- Length and inclination of the a-arms
- Roll centre

- Pitch centre
- Determined using William Mitchell's software: *Racing by the Numbers*
- Base geometry selected from previous year's cars and then iteration used to optimise final dimensions

5. **Design Uprights**

   - Finite Element Analysis utilised to test design concepts
   - Kingpin and caster angles defined

6. **Perform Simulations**

   - *ADAMS/Car simulation* software utilised to again test and compare different geometries and adjustments

7. **Make Iterations**

   - Different geometry/adjustment combinations are tested to again iterate to find an optimal arrangement

8. **Manufacture Parts**

   - Suspension system assembled in Pro/Engineer solid modelling software to check for compatibility and clearance
   - *MasterCAM* software used to assist machining operations

9. **Assemble Vehicle**
10. **Test/tune/optimize the vehicle with the driver**

Again these findings are very useful as they are written for an FSAE vehicle which is the exact nature of the project. It should also be mentioned that the methods employed by the Drexel FSAE team strongly agree with the steps recommended by Allan Staniforth although noticeably exclude any consideration for springs and dampers along with steering system design and integration. Additionally, as this paper is purely a sample document the findings and information expressed in it may not be valid and so care should be taken if this research is to be used.

Gabriel de Paula Eduardo's Design Journal for a Formula SAE Car

Gabriel de Paula Eduardo's design journal (2005) for an FSAE car's suspension system does not directly provide a step by step tutorial on the matter but rather documents the critical aspects of the Universidade de São Paulo (A Brazilian University) FSAE team's design process in a more broad sense. Although this is the case, the journal presents some very useful guidance and once again the information is quite relevant as design is orientated around the FSAE competition. The numbered list that follows is derived from the journal and translated into individual steps to conform to the other design procedures that are documented in this dissertation section.

1. **Analyse the rules**

    - Key constraints regarding the suspension system noted

2. **Define design targets**

    - These targets included compliance with the rules, low system weight, maximised mechanical grip, quick response, accurate transmission of feedback to the driver, and finally, easy adjustment
    - Following establishment of these targets, the top ten teams in the competition were referenced in a benchmarking process to obtain a basis for dimensions needed later on in the design process

3. **Select tyres**

    - "Short race durations, low vehicle mass, and low-speed courses all indicate a need for a tire that reaches its operating temperature quickly." (2005, p1)
    - The type of tyre will define how close the lower ball joint can be to the ground
    - Designers should be aware of the effects on the entire package due to the type of tyre nominated

- Hoosier 20” x 6” R25A tyres selected

4. **Select wheels**

- Strongly influenced by the tyre selection
- Other considerations: clearance for upright, brake calliper and rotor as well as the upright, cost, availability, bolt circle, and weight
- 3 piece, 6” by 13” wheels nominated

5. **Elect Fundamental dimensions and performance parameters**

- Completed with reference to results from the earlier benchmarking process
- Track width and wheelbase the starting point: dynamic load transfer as well packaging are crucial considerations
- Next, parameters including camber gain, roll centre placement, and scrub radius are chosen based on how the designer intends the car to perform

6. **Setup suspension model incorporating desired performance parameters**

- Ball joint locations, inner control arm pivot points, and track width must be known
- The easiest way to model the suspension geometry is in a kinematics software program although if unavailable, a suitable Computer Aided Design (CAD) program can be applied
- To achieve desired parameters iteration is necessary and it needs to be acknowledged that compromises are inevitable
- Consequences of an alteration to the model should be considered prior to carrying out the change
- Amount of scrub radius should be balanced such that steering forces are not excessive and there is sufficient feel delivered to the driver
- Final parameters used by the team incorporated a scrub radius of 9.5mm, 7 degrees of kingpin inclination, and 4 degrees of caster
- The final geometry created presented a roll centre height at the front of 35.6mm below the ground and 35.6mm above the ground at the rear

7. **Calculate required spring stiffness, finalise suspension actuation method and select shock absorbers**

    - Location of shock absorbers nominated (inboard or outboard, push or pull)
    - ‘Quarter-car’ model and ‘vehicle frequency analysis’ used to determine required appropriate shock absorbers
    - Team selected modified bicycle dampers

8. **Analyse of the chosen suspension system in roll**
9. **Simulate the final design**

    - Final design simulated in a suitable dynamics program to gauge the performance of the system

Even though this article provides a great deal of information on the design of a suspension system for an FSAE car, it is clear that a number of important steps are left out that were incorporated between the methods suggested by Staniforth and the Drexel University FSAE team. These missing stages include the consideration of an anti-roll bar, steering design integration, and lastly, the manufacture of the design followed by physical testing.

2.9 Design Recommendations

While researching for specific and technical guidance, a significant amount of more general and less design process orientated advice was also discovered. Again, as little expertise is held in regards to the suspension and steering system design processes, a lot of this information was deemed relevant to the project.

In an article presented primarily for teams entering the FSAE competition for the first time, Alan Gruner (2011) makes some very useful recommendations with respect to keeping the design as simple as possible. Firstly, Alan makes it known that a common misconception surrounding the competition is that

"FSAE is a good place to develop new technology. No FSAE team will have the knowledge and resources to develop brand new technology and still meet eligibility requirements for members. You can be innovative. But the place to do it is packaging existing parts and technology."

Building on this, he then goes on to mention that,

"If you have a system from last year's car that works and works well, use it again. Make minor improvements if appropriate but keep major redesign work at a minimum."

The time and energy saved on these design areas can then be applied to other components of the car that were not optimal in previous years. Lastly, Gruner also suggests that for any part or system of a previous year's car that does need significant reconsideration, should be redesigned as simple as possible. However, he also states that it is hardest of all to build something simple and lightweight and that is how to be competitive.

Keeping the design simple comes with a substantial amount of benefits which Alan documents as being reduced cost, reduced weight, less inventory to keep track of, reduced assembly time, and lastly, simpler maintenance procedures as well as easier access to components.

Additionally, Gruner also discusses an issue that the 2008 USQ FSAE team encountered where the suspension and steering systems were compromised for an early chassis creation. Here he establishes that "You must plan ahead where to package all the minor as well as major components." so as to avoid assembling the car and finding that different systems will not integrate and then compromising one or all of offending components.

Theming with this more general advice regarding the competition as a whole, Pat Clarke, an FSAE technical advisor, also provides a large number of helpful recommendations in a technical introduction written for new FSAE teams (2005). In this document Pat also supports the need for simplicity, issuing a number of basic ideas surrounding the FSAE competition which include not trying to build a complex or formula one level vehicle but rather to get a car built that will allow the team to complete the event and gain the satisfaction of competing in the FSAE competition, "…to realise that it is almost impossible to build a winning car at the first attempt.", and "…to realise that it does not matter whether the Judges agree with your design decisions or not, as long as you can cogently argue in favour of your solution" as "There is always more than one correct answer".

Along with these more broad statements, Clarke also presents some great guidance on the design of some of the fundamental components of the FSAE vehicle. These suggestions that affect the design of the steering and suspension systems are as follows:

Planning

- Planning and project management is the most important logistical component
- A detailed plan must be developed before anything is bought, sketches are made or material is utilised
- A successful FSAE car will need to have exemplary handling characteristics, be predictable to drive and easy on its tyres

Tyres and Wheels

- Starting with a 13" wheel is recommended as it allows more flexibility in the suspension design
- Tyres are the most significant selection in the design as the only point the vehicle contacts the road surface is through them

Suspension and Steering

- Camber control as the car rolls is essential
- Good FSAE design has a significant amount of caster with only a small amount of kingpin inclination
- 'Bump steer' should be avoided and any decision affecting the steering geometry should be followed by suitable testing for bump steer
- Roll steer is undesirable
- A large amount of positive Ackermann geometry is recommended for an FSAE car; increase the degree of Ackermann steering by at least 50% over what is thought to be appropriate
- Camber on an FSAE car should never go positive on a loaded wheel while turning
- Stiff suspension rates are not paramount to good performance in the competition and more compliant rates will suite these vehicles better providing wheel angles are controlled
- Design should ensure good control of the roll centre

- Care should be taken to properly react to the force placed on the chassis by front wheel brake torque thus avoiding steering changes or ‘tramp’
- Toe control is important so aim to have a wide toe control base
- When the car is loaded, drive shafts and all constant velocity joints should be aligned in all planes
- Suspension should account for the movement of the drive shafts; tripod joints are probably the best choice

Basic Design Principles

Pat also lists a number of other less specific considerations in a basic design checklist. For all of these suggestions believed relevant to the project, please consult appendix B.

Another key source of information uncovered in the project’s literature review was the FSAE.com forums. Although the information supplied on these forums was not quite as official as the sources listed earlier because it was predominately provided by students and staff associated with universities participating in the FSAE competition, the ideas and opinions expressed were extremely useful and applicable to the project design. This was due to the fact that the people conversing on the various forum topics were basically in the same shoes as the author where students competing in the competition spoke on behalf of experience and true FSAE design background while staff and FSAE technical advisors provided thorough information due to their close association with these student team members and monitoring of design. Pat Clarke was also found to be a regular contributor on these forums, presenting guidance on a number of elements associated with the suspension and steering system design.

2.10 Chapter Summary

The literature review completed for the project is very extensive. This may seem detrimental to the dissertation but it was desired to create a document that future students from USQ could utilise to aid in the design of the suspension and steering systems for the university's FSAE car. After reading a number of past student's dissertations it became apparent that this sort of information was unavailable.

The information uncovered has provided a decent knowledge basis and starting point for design to commence. In particular, the reviewed design processes and recommendations will help form the project's own design plan and methodology while the information regarding the fundamental aspects of the steering and suspension system will aid in the design of new components as well as the analysis of the 2008 vehicle to uncover the causes of its crash which is to be discussed in the following chapter.

Chapter 3

Analysis of the 2008 Vehicle

3.1 Chapter Overview

Based on witness accounts along with feedback from the USQ FSAE team, the suspension and steering system of the 2008 vehicle seemed to be a significant contributor to the car's crash at the 2008 FSAE-A competition. For this reason it was therefore important to analyse the car in hope to define the causes of the crash and the vehicle's poor handling qualities so that future vehicles did not incorporate the same design errors. This chapter details this analysis process and the findings that resulted from it.

3.2 Analysis Process

As briefly mentioned earlier, the USQ's 2008 FSAE vehicle possessed some strange handling characteristics which were believed to be a cause of the 2008 team's crash at the FSAE-A competition. This vehicle may be visualised over the page on figures 3.1 and 3.2. The first photo represents the car competing in the 2008 competition prior to the crash while the second details the current state (22/10/2011) of the vehicle. Unfortunately photos of the 2008 vehicle shortly after the crash were unable to be found.

Initially it was planned to get the 2008 vehicle to a drivable state with components repaired to original dimensions so that physical testing could occur in order to try correlate the feel and handling behaviours of the vehicle with the chosen design parameters and geometry. Unfortunately this was not achieved as although the car was repaired and able to rest on all four wheels, the repaired components were not completed to a standard where physical testing on the vehicle could occur. For this reason the analysis only involved inspecting the car to uncover any physically visible issues before investigating the team's design documentation and resources for that year along with measuring up the car's geometry to uncover any underlying issues in the 2008 design. For a full specification of the 2008 vehicle please consult appendix C.

Figure 3.1: University of Southern Queensland's 2008 FSAE-A car prior to its crash. (Puredrift.com)

Figure 3.2: USQ's 2008 FSAE-A car in its current state (22/10/2011).

3.3 Physically Visible Issues

Inspecting the car to uncover any physically visible issues involved going over the car in the workshop and noting as well as photographing any problems with the car that were physically evident. On completing this task a significant number of problems were discovered, all of which are documented below with corresponding photos of each problem featured in appendix C.

Poor Suspension Actuation

One major problem caused by the layout of the 2008 car is the poor actuation of the shock absorbers at both the front and rear of the vehicle. The consequence of this design is that operation of the suspension is not smooth because the dampers do not see the full force exerted by the movement of the wheel due to friction in the bell cranks or rockers and the fact that pivots in the mechanism experience forces that do not act normal to their axis of rotation. It should be mentioned though, that the root cause of the problem as a whole, is the location of the shock absorbers; if they were placed in a plane common with the rockers, bell cranks and push rods, all pivot points would experience forces only normal to their rotation axis, reducing friction in these joints and promoting smoother suspension travel actuation. Although the problems associated with using such an arrangement can be muted to a degree with the use of spherical bearings where components join and pivot, in the 2008 vehicle the stickiness or un-smoothness of the suspension movement is still felt.

This fault is better explained with reference to figures C.1 and C.2 in appendix C. In these photos the yellow line represents the actuating or push rod of the rear suspension system along with the rocker linkage that allows the up and down movement of the wheel to be transferred to the shock absorbers, the dark blue line signifies force generated on the rocker linkage from the actuating rod, and finally, the light green line dictates the force exerted on the shock absorber imposed by the rocker linkage.

As can be seen the actuating rod will only transfer a limited amount of the wheel's up and down motion to the rocker due to the angle it is placed in relation to the pivot axis of the rocker. Ideally the angle that the push rod makes with the rocker rotation axis should be perpendicular whereas in figure C.2 it can be seen that this angle is significantly less than 90 degrees and thus all forces do not act normal to the rocker's axis of rotation.

Average Quality Shock Absorbers

Generally the USQ teams will utilise a mountain bike shock absorber in the design due to the fact that they are readily available, relatively cheap and feature enough adjustment for the FSAE competition. Although the current shock absorbers may be cheap, it is believed that their quality and specifications are insufficient for the use on an FSAE vehicle. Being an active mountain biker, this view is based on individual experiences, peer feedback, as well as reviews and opinions featured on the internet, television and in magazines, etcetera. The 2008 car's front and rear shock absorbers can be visualised in figures C.3 and C.4 located in appendix C. For reference, the model of the front shock is a DNM ST8-RC and the rear is a DNM MM-20AR.

Inappropriate Manufacturing Procedures

Observing the various suspension and steering components in the lab, it became obvious that there was not a lot of care taken to keep welded components within suitable tolerances of the designed geometry. Clearly this is undesired as it complicates installing and adjusting the suspension and steering systems but could also potentially mean that the final vehicle featured geometry and consequently, performance characteristics significantly varying from those aimed for in the initial design. To provide an example of this component quality, a photo (figure C.5) of the front upper wishbone arm has been featured in appendix C.

Limited Room in Driver Cockpit

Figure C.6 in appendix C represents a view down the corridor in which the driver's legs are placed while operating the vehicle. The red ovals represent the driver's feet in their approximate position on the accelerator pedal (silver colour) and the brake pedal (black).

In reference to this photo it is therefore obvious that space is not in abundance with the drivers legs having to compete for room against the steering rack and steering shaft, as well as the shock absorbers and associated covers (black sections on the floor – the one closest to the camera houses the shocks whereas the one in between the steering shaft protects the drivers legs from the steering rack). One of the checks in the technical inspection which is part of the static events that were mentioned earlier in the introduction chapter is the test to see if the driver can evacuate the car in 5 seconds or less. With a confined cockpit this may not be possible, and on top of this, if there was a real incident requiring the driver to get out as quickly as possible, safety would be compromised.

Suboptimal Steering System Design

The steering system is another area of the vehicle where compromises have been made. This time the team has sacrificed a performance optimised design for cost where the 2008 vehicle features a modified steering rack from a small car meaning the steering ratio was locked at whatever the rack and pinion was manufactured with. On top of this, the steering arms on the uprights are very large meaning that more rack travel is required to achieve full steering range at the wheels. Consequently the steering wheel requires a large degree of rotation to travel from lock to lock and this is not a good thing considering the FSAE tracks feature alternating corner directions with these corners of a very tight nature. Given that the space in the cockpit on the vehicle is not extensive, it is estimated that the drivers of the car would have had a lot of trouble in trying to manoeuvre the steering wheel to make it around the tight track. Ideally it is believed the rack and pinion size and ratio should be selected in order to suit the required dynamic performance characteristics.

3.4 Underlying Issues

These issues are those which are not obvious when just viewing the vehicle and would have to be uncovered by measuring components and geometry along with checking any documentation and computer models created by the 2008 team.

Large Scrub Radius

Upon defining the location of the new design's upright pickup points for the suspension wishbones by electing the desired scrub radius and kingpin inclination, it was realised that the old car, due to the wheels it featured, would possess a very large scrub radius and would thus require an also very large kingpin inclination to minimise this. This finding can be confirmed by observing the following figure which compares the wheel space available in the old wheels (*Superlite* 13" x 5.5") to the new wheels selected for the design (*Keizer* 4L 13" x 7"). Please note that the selection of these wheels is documented later in the dissertation.

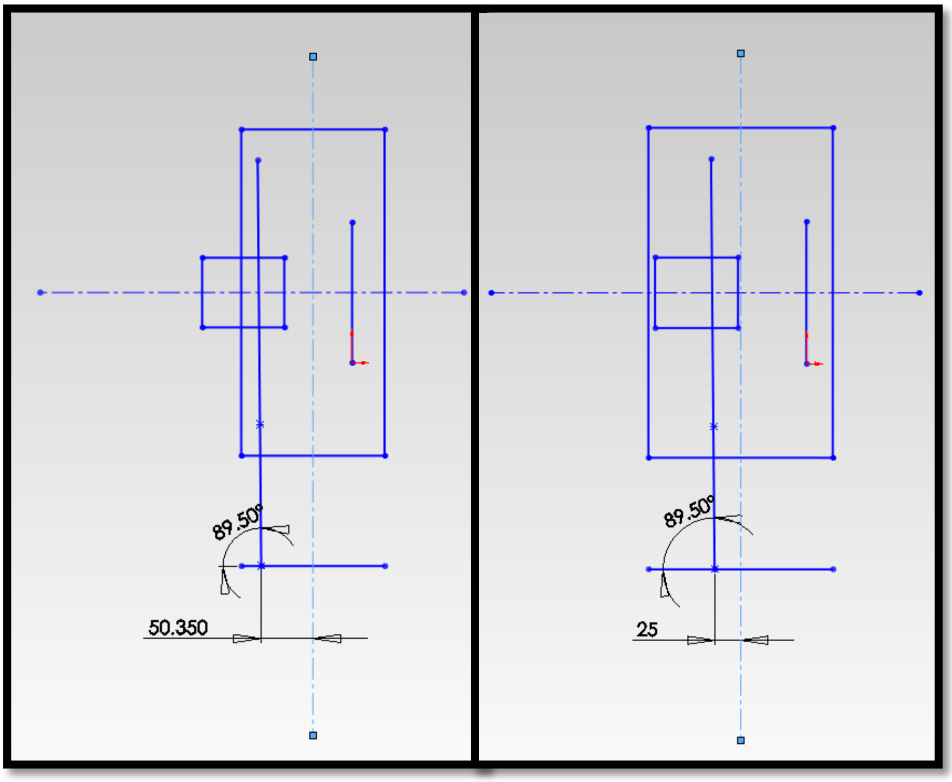


Figure 3.3: Scrub radius of 2008 wheel and 2012 wheel using the same upright design for both

In this figure the large rectangular boxes represent the amount of space available in the wheel and the single horizontal lines below these boxes are an indication of the ground surface. Also, the light blue dotted lines highlight the centre of each wheel. In both these drawings the upright, tyre diameter and hub/axle remain the same such that a viable comparison can be achieved. As can be seen, the 2012 wheel produces the least scrub radius and is approximately half of the value presented by the 2008 wheel. Findings uncovered in the literature review and opinions expressed in the FSAE.com forums suggest the smaller the scrub radius, the better, justifying the move to a wheel featuring larger offset. The scrub radius relates to the steering feel to a large degree with a smaller radius promoting easier steering movement as the friction created by the tyre scrubbing across the road surface as the wheels turn, is reduced. A larger scrub radius means a greater distance from the point where the weight of the car concentrates on the tyre's contact patch and the location where the steering or kingpin axis meets the ground plane which provides a larger moment arm for the frictional forces to act on making it harder for the driver to turn the wheels. It is therefore believed that this flaw may have been a major contributor to the 2008 vehicles poor handling ability by making the car harder to steer and thus manoeuvre around an FSAE track.

Pickup Points for Suspension Arms

Another issue uncovered following the design of respective aspects of the future vehicle was the poor location of the pickup points for the suspension arms. From what is understood, the 2008 vehicle's chassis mount locations were not extremely flexible as the chassis was built before significant suspension design commenced. After completing the designation of these pickup points for the new design it has been realised that only having a narrow range of locations for these suspension arms, puts a significant restriction on the amount of performance that is able to be achieved from the suspension geometry. With the upright specified, movement of these pickup points is the only way to define the static location of the roll centre height and therefore if these points cannot be moved around much than neither can the roll centre meaning that the geometry obtained will not be optimal. On top of this, movement of the roll centre, camber gain of the wheels and many other performance parameters may not be able to be controlled or act in a desired manner under various dynamic vehicle actions.

3.5 Chapter Summary

This chapter has defined the reasons believed to be the cause of the 2008 vehicle's crash as result of analysis concerning physically visible issues obvious by sight along with underlying issues uncovered by measurement of the past vehicle coupled with review of its design and competition documentation. Out of these potential causes it is believed the biggest contributors to the 2008 vehicles crash and poor handling capabilities were the suboptimal steering design, large scrub radius and restricted pickup point locations.

Findings revealed from this analysis have allowed certain decisions concerning design of the new suspension and steering systems to be easily made. These decisions included the nomination of a new wheel to reduce the scrub radius, more flexible design of the suspension arm and upright geometry, specification of new, better quality shock absorbers along with the design of simplified actuation mechanisms that operate in a single plane, and lastly, design of the steering system so that performance was not compromised by an early rack and pinion selection.

Chapter 4

Design Plan and Founding Decisions

4.1 Chapter Overview

One of the major indications to come from the literature was the need to formulate a solid plan and design basis for the project's work. Doing so would ensure the project always had direction and tasks associated with the design were completed with high efficiency. The following chapter documents this plan and founding design decisions along with the processes and justification applied to arrive at these.

The project design plan draws heavily on work and recommendations suggested by a number of author's in the literature review with the final plan made up of steps and ideas deemed most appropriate to the design of an FSAE vehicle. This is expected to provide an optimal pathway for design as not all literature reviewed was specifically aimed at application to the FSAE competition neither did every author's work agree with the design beliefs held by the project researcher.

Founding design decisions have also been strongly influenced by information derived from the literature review but have also been shaped by the problems discovered with the 2008 vehicle. The decisions considered include the type of suspension mechanism used, tyres and wheels, and lastly, early geometrical decisions involving nomination of the wheelbase and track widths as well as the kingpin inclination and scrub radius.

4.2 Project Design Plan

1. **Analyse the Formula SAE 2011 rules**

   - Read through the rules at least twice making note of all regulations affecting the suspension and steering systems
   - Seek documentation on additional rules concerning the Australasian FSAE competition (FSAE-A)
   - Consider but don't design around any 'possible future rules changes' noted in the FSAE rule book

2. **Establish realistic performance targets for the FSAE vehicle's steering and suspension systems**

   - Take into account previous team's vehicles as it is most likely that budgets, part choices and thus, design values will be similar
   - Also consider the vehicles of competing teams for realistic performance goals, particularly teams that have been successful in the past and teams from a similar background to USQ

3. **Select of tyres and wheels**

   - Select an appropriate wheel size based on packaging versus performance
   - Consider the wheel offset to achieve the desired scrub radius
   - Tyres elected based on wheel chosen, what has been successful in the past for USQ and what has been proven by other competing teams at the FSAE competition, tread width and cost

4. **Make founding design decisions**

   - Type of suspension nominated

- Wheel base and track proposed, again based off previous USQ team's vehicles as well as former competitors along with information provided by suspension design experts
- Initial roll centre placement elected
- Selection of camber and caster angles as well as trail
- First designation of ball joint locations and suspension arm connection points on the chassis made
- Values nominated may not be a representation of the final design as iteration will later be used to optimise the suspension geometry

5. Design upright geometry

- Conformance with chosen wheels is paramount
- Initial values for kingpin inclination and scrub radius for the front upright elected
- Pickup points for the upper and lower suspension arms defined based on the chosen kingpin inclination and scrub radius at the front whereas at the rear, points are chosen based on the desired location for the toe link attach point and packaging
- Steering arms on the front upright are neglected at this stage of the design

6. Enter Chosen founding parameters into suspension geometry model

- Nominated modelling software is *Wingeo3* by William Mitchell
- Iterations completed by slightly altering the suspension models in order arrive at a set of geometry that within reason, conforms to performance targets set earlier in the design process and also is able to be integrated with the layout of a typical USQ FSAE car

7. Choose dampers

- Selection of dampers based on required stroke, length, weight, degree of adjustability, quality and cost

- What has been successful on past USQ vehicles along with competing cars is also a consideration

8. Design suspension actuation mechanisms

- This will include choosing the actuation method (inboard or outboard, push or pull) as well as designing any joints, mounting devices and suspension linkages
- First attach points of the push or pull rods on the suspension arms are determined before the location and orientation of the rockers is specified
- Lastly, the a relationship regarding the ratio of the rockers and the geometry of the suspension and orientation of the push or pulls rods is established so that when the springs are nominated, this ratio can be adjusted suitably to provide the desired suspension jounce and rebound

9. Calculate spring stiffness's

- Spring stiffness's and final rocker ratios calculated to conform with design procedures outlined by suspension design experts
- Chosen spring stiffness's analysed for their resistance to chassis roll
- If chassis roll is excessive with the chosen springs, an anti-roll bar is required to be design
- The anti-roll bar stiffness will be defined by the extra stiffness required to keep chassis roll within the desired limits that the normal damper springs are unable to provide

10. Design steering system

- Type of steering nominated
- General positioning of the steering mechanism finalised
- Initial steering geometry established based off previous team design and external sources

- Integration with suspension model followed by testing in *wingeo3* to assess performance
- More iteration used in order to discover an arrangement that will satisfy the existing suspension design and offer performance characteristics within reasonable range of what is desired
- Front upright design fully completed with accommodation of the steering arms

11. Physical design of components

- All components modelled in *SolidWorks* followed by finite element analysis (FEA) in *ANSYS* to verify the design is safe and can withstand all loads imposed by typical vehicle use. A suitable factor of safety (FOS) will need to be chosen in order for this to happen
- If components fail to meet required factor of safety, appropriate revisions will be made to the design before being tested again, thus ensuring the final design is adequate
- Once components are finalised the suspension and steering systems are to be assembled in 3d space to aid in the packaging of other vehicle systems and to verify that there is no fouling of components within the suspension and steering mechanisms as well as other vehicle parts
- If fouling is present suitable redesign will follow before the reconfigured parts are again tested in an FEA

4.3 Performance Targets

The targets for the new suspension and steering systems needed to be realistic but also had to subsidise a final design that allowed the vehicle to compete reasonably well in terms of handling. The following list outlines the initial performance targets for the project's design with the targets in bold regarded as the most crucial aims. It should be recognised that all targets are listed in order of importance.

1. Improvement on 2008 car

Most importantly it is desired to make an improvement on the 2008 car's design and performance as if the project didn't achieve this than it would be a complete waste of time.

2. Easy to drive and inspires confidence

Obviously a car that is easy to drive will inspire confidence which in turn will hopefully allow the car to be driven harder and faster; Chris Snook, the project supervisor made it clear that it is all well and good to reduce weight and try and bleed every bit of performance out of the car but at the end of the day that costs money and takes time. An alternative and cheaper way to achieve faster lap times could be to ensure the car is easy to drive and that the drivers are sufficiently trained.

3. Highly adaptable as well as adjustable and easy to do so

As the car that these components are being designed for is undefined, the design must incorporate a large degree of adjustability to cope with any changes arising from integration with a future USQ FSAE team's vehicle.

4. Economical

Referring back to the motivations behind the FSAE competition, providing an economical design is well rewarded as the competition is centred around mass producing the vehicle under a set budget. The more value for money the design possesses, the larger the tally of points awarded to the team will be. USQ's FSAE teams typically operate on lower funding and this is also another reason the system needs to be kept cheap.

5. Maximised grip

6. Quick response handling

Targets 5 and 6 are further goals related to the actual operation of the vehicle and are pretty self-explanatory. Maximised grip is desired as it allows for increased cornering force while quick response handling is sort after in order to successfully navigate the tight FSAE circuits.

7. Reliable

Although the FSAE competition dynamic events do not demand a lot of reliability out of the vehicle, the design reports and inspections from the judges presented in the static events will bring any potential reliability issues to the surface and will in turn result in a deduction of points from the team's total score.

8. Easy to repair

Extending from above, in the unlikely event that the car does show some poor reliability and loses full functionality, repair must be easy to carry out. If an incident such as this occurred at the competition it would be desirable, for obvious reasons, to have the car up and running as soon as possible and not face complete inability to finish the events.

9. Simple

Taking on board the guidance provided earlier by both Pat Clarke and Alan Gruner, it is also believed that design should be kept as simple as possible. On top of the benefits Clarke and Gruner list, simplicity is a target because as mentioned previously, the author is very new to suspension and steering design and therefore it is believed things should be kept as straightforward as possible to minimise design problems and the delays that come with overcoming these. Keeping things simple could potentially mean that the design is less cluttered as well which is another advantage for a number of reasons including more efficient packaging of components, easier maintenance and repair procedures, and possibly, easier compliance with competition rules regarding the assessment of certain car components by the judges in the technical inspection.

10. Light weight

Stemming off target number 2, it is maintained that the weight of the steering and suspension systems is not terribly important, as long as the design incorporates suitable geometry and is easy to drive to the limits then the design can be considered a success. That's not to say that the weight of each system will not be considered at all. It will be, but only after all preceding targets have been addressed.

These targets aim to found a design that will produce improved performance on the 2008 vehicle but also to yield a design that does not stretch a typical USQ team to great lengths in order to create and build it.

4.4 Founding Design Decisions

4.4.1 Type of Suspension

The chosen suspension configuration used for both the front and rear of the vehicle will be a double wishbone setup using unequal and non-parallel arms with inboard shock absorber placement which uses push rods. The merits and background of such an a arrangement have been discussed in detail earlier although reiterating, this setup was chosen as the double wishbone setup provides a large amount of room for adjustment, allows decent tyre camber control, has high strength and rigidity, good damage protection, and finally, it permits a low unsprung weight for the vehicle. Placing the shock absorbers inboard reduces the vehicle's coefficient of drag, improves wheel rate control along with ride height adjustment, and lastly, allows more flexibility when positioning the shock absorbers. Using a push rod system front and rear is believed to produce the best packaging in a future USQ FSAE vehicle; this arrangement follows that the shock absorbers be mounted up higher in the car and although this raises the overall centre of gravity, for the front it also provides more space in the driver cockpit which was a major downfall of the 2008 vehicle. In the rear, a push rod configuration also suits quite well as with the shock absorbers up higher, associated linkages and components have clearance from the drive train and engine.

An example of these layouts for the front and rear systems can be viewed over the page on figures 4.1 and 4.2. The green highlighted ovals indicate the areas of interest. Obviously the design will not attempt to mimic these setups as this would potentially not be optimal for a USQ vehicle. Instead the layouts will be used as a guideline and then the configuration of past USQ vehicles will be observed to determine a system that will integrate the best with a typical USQ design.

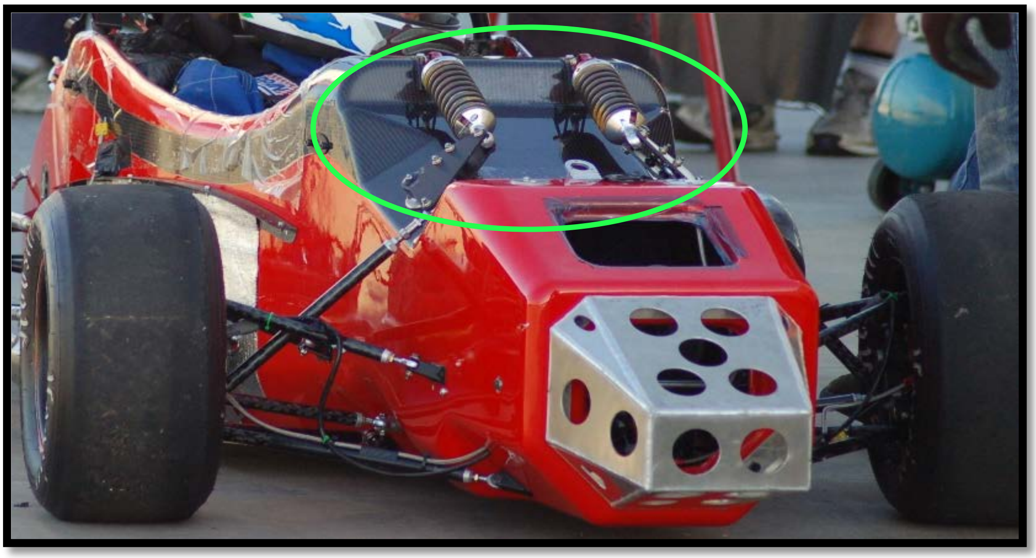


Figure 4.1: Example of the intended layout for the front suspension system – RMIT's 2009 FSAE vehicle.   (RMIT Racing, 2010)

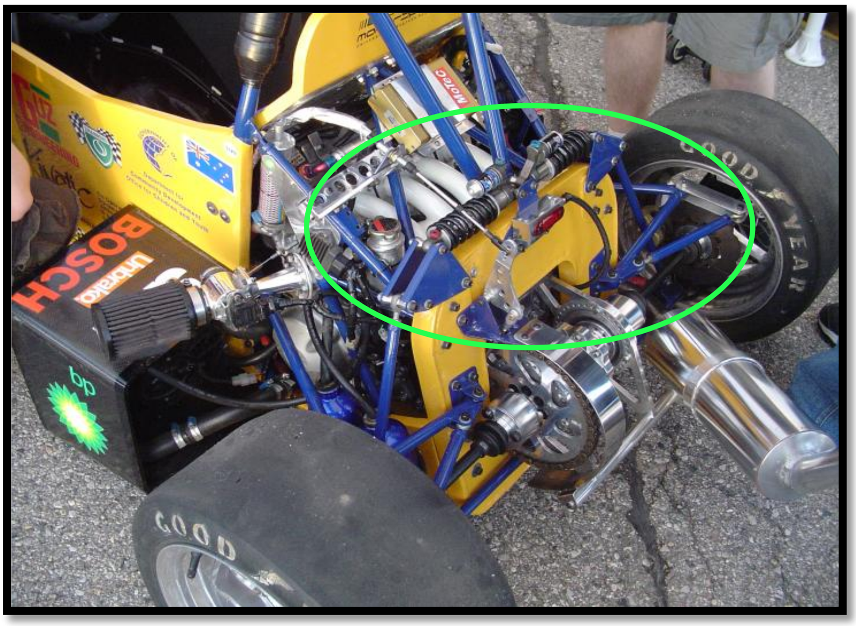


Figure 4.2: Example of the intended layout for the rear suspension system – UWA's 2004 FSAE vehicle.   (FSAE.com, 2004)

4.4.2 Tyres and Wheels

10inch vs 13inch Diameter

In the FSAE competition there are two types of wheel size typically used, 10inch or 13inch diameter. Each size offers a number of pros and cons and may suit one team better to another. A 10 inch wheel can provide lighter weight, less rotational inertia meaning faster acceleration and a more compact design. However, being smaller means that there is less packaging room for the components associated with the suspension, braking and steering systems, making design of these mechanisms more complicated. 13inch wheels on the other hand provide the opposite characteristics due to their larger size. The new design intends to use a 13inch wheel mainly because this size would present the good packaging which in turn would simplify the suspension and steering design processes and for a beginning designer this is desired. A 13inch wheel is also believed to suit the typical packaging and layout of a USQ FSAE vehicle and so using this size would increase the flexibility and adaptability of the design for future incorporation.

Wheels

The initial plan for selecting the wheels and tyres was to use components from the 2008 vehicle. As mentioned earlier however, the old wheels did not provide very good offset and so granted the vehicle with large scrub radiuses. To improve the scrub radius it is possible to increase the kingpin inclination although this should be avoided as it goes against the advice of expert Pat Clarke and the general consensus of the FSAE.com forums because it will increase the amount that camber will change positively when the front wheels are steered. Positively cambered tyres do not grip well and continuation to expose the tyres to this treatment will damage them much more rapidly than if they were to never go into positive camber. (Clarke, 2005)

As the project is not considering any hub or axle redesign and the current calliper placement provides minimal clearances as it is, the only way to rectify the large scrub radius was to go with a larger offset rim. An investigation into the various options available for a potential new wheel uncovered that there was not a significant range of options to choose from in Australia; there was always the option of finding something second hand that would fit and be relatively economical although experience from past USQ teams would suggest this may not be the optimal solution. There was also a wheel provided by *Watanabe* which is an Australian producer and would allow for desired offset, material and weight to be elected. However, these wheels were quite expensive with the cheapest wheel matched to the FSAE car priced at AU$460 per wheel.

A more promising option was found to be wheels obtained from *Keizer* who actually produce wheels specifically designed for the FSAE competition. Unfortunately these wheels are made in the United States and so getting them to Australia could prove to be quite expensive although these wheels were around half the price (US$245 not including postage) of the cheapest from Watanabe and so raised postage costs would be offset. Keizer also provides a wide range of wheels within this FSAE model range, allowing the designer to choose from a wide selection of offsets, materials and designs. After analysing the full series of FSAE wheels from Keizer, it was decided to use the 4L model wheel in a 13" x 7" size with 6 inches of backspacing. This was in contrast to the wheels on the 2008 vehicle which were *Superlite* 13" x 5.5" with 4.25 inches of backspacing. The elected wheel model can be pictured below.

Figure 4.3: Chosen wheel for the future design – Keizer 4L series 13" x 7", 6" backspacing

Tyres

With a new wheel nominated, a new tyre also had to be specified to accommodate the increased rim width. It was chosen that the design would use the same tyres by *Hoosier* as the last car, with the 20.5 x 7.0-13 R25B model best suited to the selected wheels. *Hoosier* tyres have been a proven success in the FSAE competition and they also offer a %20 discount to all FSAE teams. On top of this, USQ have never had any issues in the past with these tyres.

4.4.3 Geometrical Allocations

Wheelbase and Track

Vehicle	Wheelbase (mm)	Front Track (mm)	Rear Track (mm)
2006	1600	1237	1155
2007	1640	1271	1175
2008	1670	1271	1175
Future cars	1600	1300	1175

Table 4.1: Wheelbase and track dimensions of the final design along with past USQ cars

The final wheelbase and track dimensions (highlighted in green on the above table) are based off dimensions of prior USQ FSAE vehicles as well as guidance provided by Carroll Smith in his book entitled 'Tune to Win' (1978, p56). It is believed that the wheelbase and track is influenced by a number of factors when designing a race car. These main factors should be the desired performance and handling characteristics of the vehicle, the overall size of the vehicle and thus ability to navigate through tight track sections, and lastly, how everything is to be packaged on the car.

However, it is also anticipated that the selection of these dimensions is also influenced, although note directly, by some other, less obvious factors including the team's available budget and degree of sponsorship, availability of parts as well as engineering faculty and university supervision and guidance. Therefore, as the project work is intended for a future team that hasn't planned any design, it is supported that the proposed design shouldn't stray too far from past trends but should also comply, to some degree, with recommendations listed by a proven expert, Carroll Smith.

In brief, Carroll states that a '…racing car with a long wheelbase and relatively narrow track widths will be very stable in a straight line at the expense of cornering power and maneuverability." while a vehicle possessing a shorter wheelbase coupled with wide tracks will be "…less stable, harder to drive to its limits, more manoeuvrable and will develop more cornering power." Additionally, it is made clear that a longer wheelbase will reduce longitudinal load transfer and pitching moments as well as allowing more room to put things whereas a short wheelbase with wider tracks will reduce lateral load transfer, provide room for longer suspension links, but will also increase frontal area of a the vehicle thus inhibiting its aerodynamic properties. Given the typically low speeds reached in the FSAE competition, this last disadvantage was not a major consideration.

Adding to this, Smith also mentions the importance of using a considerably wider front track than on the rear and that the lower the cornering speed, the greater the importance of this. By doing this, the vehicle will experience increased resistance to diagonal weight transfer while cornering which in effect, reduces the tendency of the car to “trip over itself” and/or to travel wide in the turn.

The final dimensions may be viewed below and as can be seen, values do not stray too far from previous year’s cars. On top of this, the chosen geometry mostly conforms to the endorsements suggested by Smith. The proposed wheelbase of 1600mm is not a massive leap from past USQ design and in fact, is the same measurement used in 2006. Using this value it is believed that handling will be manoeuvrable enough to navigate the tight and technical FSAE competition tracks while still providing enough packaging room for components.

The track dimensions nominated (front – 1300mm, rear – 1175mm) also do not vary too much from past designs. Employing these dimensions has a number of advantages including the ability to utilise the 2008 car’s drive train components as the rear track remains the same, that the tracks are wide enough to provide sufficient manoeuvrability and lateral weight transfer characteristics while still allowing decent clearance for obstacles on the FSAE course, and finally, that the geometry complies with Carroll Smith’s suggestions of a considerably wider front than rear track which in turn provides the benefits listed earlier.

Most importantly, these dimensions are within the rules stated for the 2011 FSAE competition which imply that the minimum wheelbase allowed is 1525mm while the front and rear tracks do not have limits but must have a difference in tracks in either the front or rear of no less that 75% of the larger track.

Kingpin Inclination and Scrub Radius

With the tyres and wheels selected it was now possible to define the vehicle’s scrub radius and kingpin inclination. As mentioned earlier, the goal for specifying kingpin inclination and scrub radius is to keep them both as small as possible and to maintain a fine balance between these two parameters. Based off opinions in the FSAE.com forum and with recommendations from Pat Clarke in mind, the design will attempt to achieve a kingpin inclination of 0°-3° and a scrub radius of 0-30mm. These values are expected to provide ease of steering while still providing enough feel for the driver.

4.5 Chapter Summary

In this chapter the proposed design plan formulated based on information uncovered in the literature review and what was believed most appropriate to a USQ FSAE team, along with the founding design decisions that would shape the final suspension and steering system, have been discussed. In completing these project tasks, the decision to use a double wishbone, push rod activated suspension configuration at the front and rear of the car has been made, selection of a new wheel model offering a larger offset to reduce the design's scrub radius and kingpin inclination has occurred, a tyre matching the chosen wheel has been picked, wheelbase and track width dimensions have been finalised based off past USQ team design and expert recommendation, and lastly, the desired kingpin inclination and scrub radius believed to offer the best steering and suspension performance have been specified.

Chapter 5

Suspension Geometry

5.1 Chapter Overview

With the founding decisions regarding the suspension system made which included the type of configuration used, the wheel and tyre model, and finally, geometrical choices regarding the wheelbase and track widths, scrub radius and kingpin inclination, the geometry that defined the suspension design could now be established. The following chapter details the results of the geometry design and the process used to arrive at the final solution.

5.2 Dimensioning the Uprights

The first component to be sized was the upright and the new wheels selected in the previous chapter defined the limits on its size. To determine these limits along with the scrub radius and kingpin inclination, simple 2d sketches (one for the front and two for the rear) detailing the space available inside the wheel were created in *SolidWorks*. The sketch representing the chosen configuration for the front upright may be seen over the page on figure 5.1 with the sketches for the rear setup featured in appendix D. With reference to these figures, the nominated pickup points are circled in red.

Front Uprights

For the front arrangement, the sizing of the upright was defined by the desired scrub radius and kingpin inclination. Without any axle or hub redesign the final design arrived at a scrub radius of 25mm with a static kingpin inclination (without camber incorporated) of 0.5°. These values are within the limits nominated earlier in the design process and are expected to provide a good balance between steering ease and steering feel. To arrive at these dimensions a number of different options were tested, each one with an altered scrub radius. Each option was then evaluated in regards to the performance it would provide along with how realistic the final upright dimensions were in terms of being able to fabricate the component and how much strength and rigidity would be provided by such an arrangement; as there was no intention to reconfigure the hub, axles and consequently the bearing housing too, there was a limit to where the upright pickup points for the suspension arms could be located. For smaller scrub radius's the pickup points would become very close to the outer edge of the bearing housing section of the upright, rendering these options unviable.

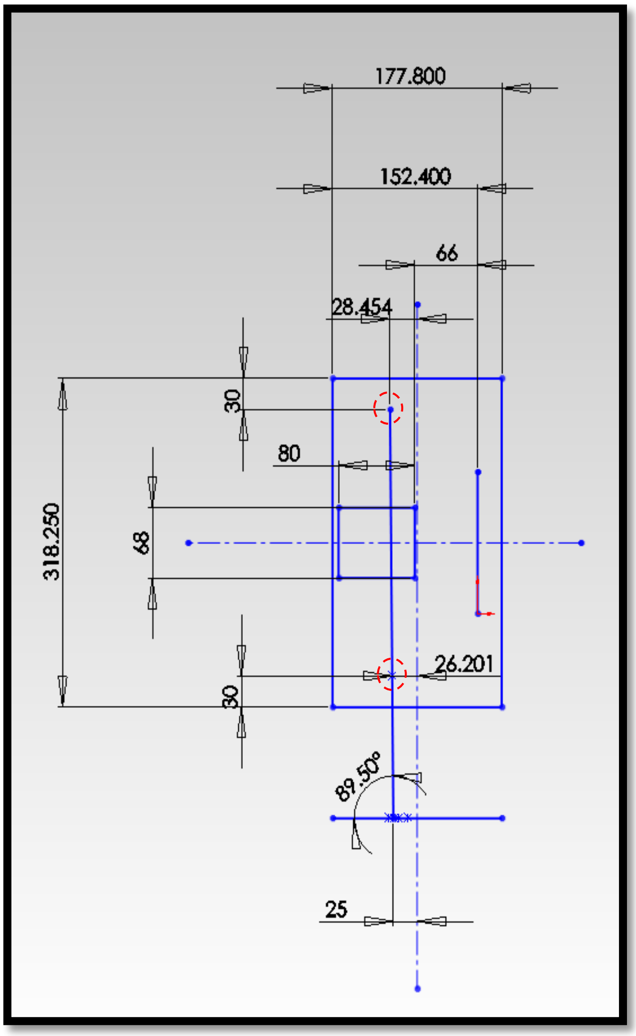


Figure 5.1: SolidWorks sketch detailing the position of the front upright pickup points – as viewed from the rear of the vehicle.

Rear Uprights

The rear upright was a bit simpler to size as the scrub radius and kingpin inclination were not deemed as important in the rear as wheels aren't steered. The main consideration here was the packaging because the upright has to accommodate both the pickup points for the suspension arms but also the pickup point for the toe link, a rod that attaches to the chassis and upright alongside the suspension arm mounts in order to provide toe angle adjustment.

As can be seen from the figure of the final dimensions for the rear upright, the design places the mounting location for this toe link at the base of the upright at the same height as the pickup point for the lower suspension arm. Additionally, the toe link mount is rearward of this suspension arm pickup point and both these lower points extend off the centre of the upright the same distance. As little information was uncovered regarding the location of this toe link in the literature review, the final decision was based off debate on the FSAE.com forums along with basic engineering judgement. Having the toe link placed lower would lower the car's centre of gravity and on top of this there are generally more places to mount this link to the chassis when it is in a lower position because up higher there is the drivetrain and engine. Also, placing the toe link rearward of the lower suspension pickup would induce a positive caster angle and trail which in turn, would mean that as the rear wheels steered due to roll (in a corner), they would not automatically seek to get back into a straight line, putting less strain on the toe link itself along with the upright and all associated mounts and joints. Lastly, by having the toe link pickup and the lower suspension arm pickup evenly spaced apart from the centre of the upright and at the same height, the loads emplaced on the rear suspension geometry would be better shared and so individual parts would be faced with less strain.

The selection of the specific location for the pickup points was also a case of trying to evenly distribute the loads experienced by the geometry along with fitting the upright inside the wheel. The final locations ensure that the distance between the lower suspension pickup point and the toe link pickup point is enough to provide a rigid structure between the toe link, lower suspension arm and the chassis while still allowing a decent amount distance between the base of the upright and the bearing housing such that the rear suspension geometry, as a whole, is structurally sound. As mentioned earlier, the sketches highlighting the final arrangements for the rear uprights can be seen in appendix D with the pickup points circled in red.

5.3 Suspension Geometry Evaluation

In order to arrive at a suspension geometry that provided optimal performance, it was believed that an iteration and evaluation process was required. The chosen method utilised the *Wingeo3* suspension geometry program written by William Mitchell and involved 3 sets of iteration; the first tested a largely varying group of geometries, the second then took the best geometry layout to come from the first iteration and vary certain parameters to determine the effect these changes had on the performance characteristics of the layout before electing the best setup based on these results, and finally, the third iteration then took this refined layout and further tested the geometry under certain varied parameters that involved altering the static camber, caster and trail. Once the effects of these changes were known, the geometry design was completed by choosing the geometry parameters or characteristics that provided the best performance in regard to the chosen evaluation criteria which outlined the areas on which each iteration was assessed. The evaluation procedure, assessment criteria and review of iteration results was elected completed with guidance from information uncovered in the literature review but also took debate and opinions expressed on the FSAE.com forums into consideration.

5.3.1 Initial Decisions

Camber

For each geometry the static camber remained the same such that a viable comparison could be made between each option. The value for this camber was set at -1° for both the front and rear which was based off what USQ and other teams had run in the past. This was not going to be the final value used in the design as this would be decided later after the third set of iteration.

Caster and Trail

Like the camber for each iteration, the caster and trail also remained the same. For the front this was set at 6° caster with a trail of 41.589mm. These values were based off advice from Pat Clarke along with opinions expressed on the FSAE.com forums, although again it was expected that these initial specifications would be refined after the final iteration process. For the rear, caster and trail were defined by the sizing of the upright as these values were not going to change in the final design and did not need to be refined as the rear wheels are not physically steered. Based off the dimensions of the rear upright, the caster was consequently 11.937° with a trail of 83.613mm.

Roll Centre

Although it wasn't possible to specify an exact location of the roll centre and maintain it for each iteration as this would defeat the purpose of trying different geometries, it was possible to reinforce a common trend throughout all iterations regarding the placement of the roll centre. According to Carroll Smith (1978, p54), the front roll centre should always be lower than the rear although not by too much. Therefore each iteration featured this layout with the front roll centre below the rear.

Placement of Pickup Points

While the iteration process needed to test a wide range of configurations, each of these geometries had to be realistic in terms of being able to accommodate a typical USQ FSAE vehicle; it would be simple to create a suspension geometry that delivered very good performance although it may not be able to mount to a typical FSAE chassis and thus would be totally useless to the overall design. Hence the mounting points of the suspension arms on the chassis were located based off dimensions of the 2008 vehicle along with reference to competitor's vehicles and their respective packaging and layouts.

Anti-dive/Anti-squat Geometry

It was decided that each iteration and the final design would not consider any form of anti-dive or anti-squat geometry as it is believed that an FSAE car does not need it. Carroll Smith (1978, p35) also supports this decision where he mentions that sports racing cars don't need it due to their "inherently sensitive natures" and the fact that they can't tolerate the upsets that this geometry can cause. Pat Clarke is another design expert to support the absence this geometry in the USQ design, stating that anti-dive and anti-squat are seen to best advantage on cars where aerodynamic control is important. As this design along with all previous USQ vehicles, have not considered any significant aerodynamic devices, the control provided by anti-dive and anti-squat geometry is not required.

Other Data

All other data required to form the first iteration's geometry came from decisions made earlier on in the design process or from information provided by manufacturers of the components chosen to be incorporated in the design. These dimensions included the wheelbase and tracks as well as the diameter of the tyres which was 21" or 533.4mm (Hoosier, 2011).

With these decisions made, the above dimensions were entered into *Wingeo3* to form the first geometry iteration to analyse. This was achieved by inputting dimensions in the top left of the *Wingeo3* program window. An example of a fully defined geometry is provided below on figure 5.2. As seen on the following page, figure 5.3 details *Wingeo3*'s very rough representation of how the geometry would appear which is typically shown in the lower half of the program window.

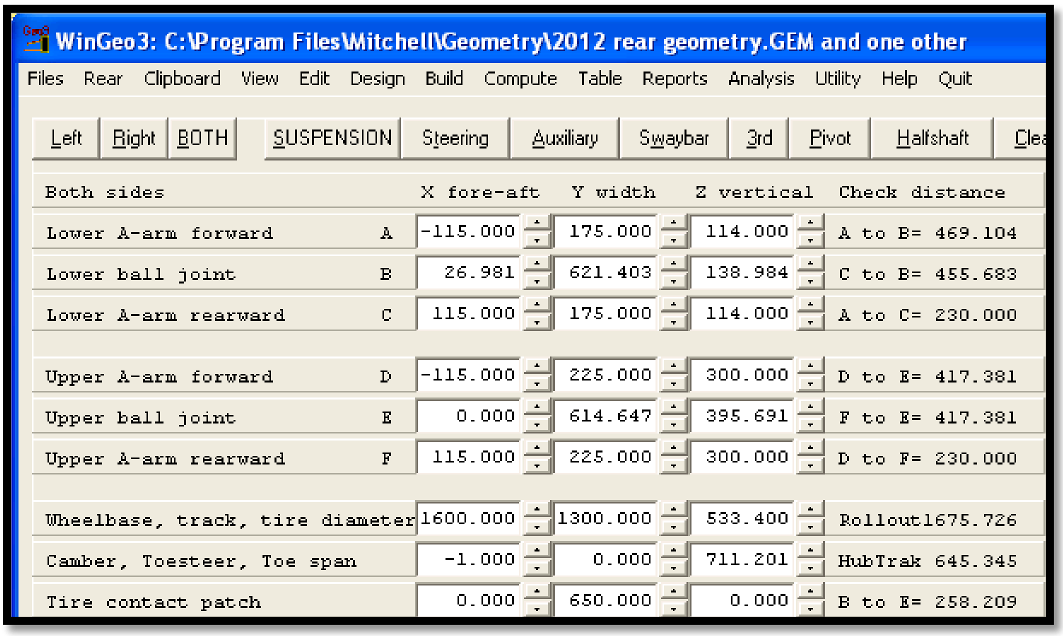


Figure 5.2: Example of suspension geometry data inputted into wingeo3.

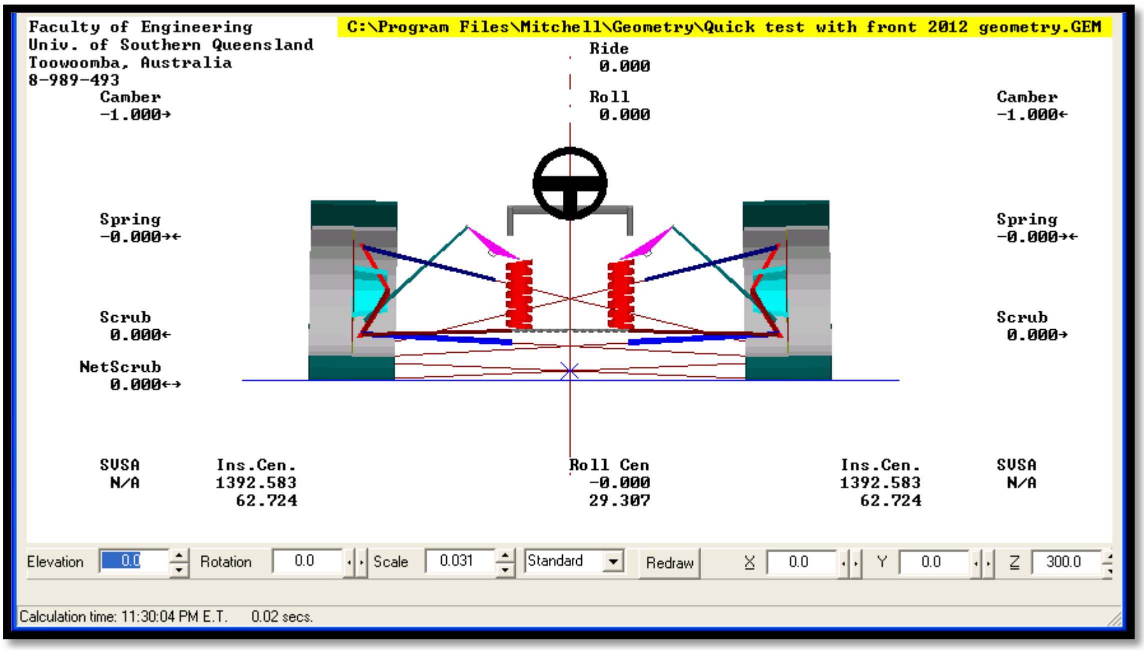


Figure 5.3: Representation of suspension geometry in Wingeo3 (corresponding to geometry data specified in figure 5.2).

5.3.2 Wingeo3 Suspension Geometry Program Analysis

In *Wingeo3* each set of geometry was tested under 3 conditions; in bump, droop and roll with no combinations of these conditions. Although this simplified the model quite a bit, it is believed that these 3 vehicle behaviours were sufficient in order to evaluate the performance of each iteration. More specifically, bump and droop were tested by raising or lowering the chassis height by 50mm, a value thought to cover the typical movement that an FSAE car would experience plus a little extra; the rule book specifies the vehicle must exhibit at least 25mm of bump and droop wheel travel.

The amount of roll tested was nominated as 3° either way and was derived as an appropriate value by considering the maximum roll in degrees, per g of acceleration (°/g) for past USQ vehicles (1.41°/g) and then multiplying this value by a typical max lateral acceleration that an FSAE vehicle would experience in a turn. In the static events of the FSAE competition, competitor's vehicles are tested on a tilt table that is stated to emulate 1.7g's of cornering force to test if the car will keep all wheels on the ground (SAE International, 2010, p45). This value was then assumed as a general maximum lateral force experienced by an FSAE car and so the largest roll for a USQ vehicle could be anticipated as (1.41°/g) x 1.7g = 2.397°. The 3° was then obtained by rounding up to provide a safety barrier and to evaluate each geometry at an absolute worst case scenario.

Summarising, the bump and droop were tested by altering the chassis height from -50mm below its equilibrium (bump) to 50mm above its equilibrium (droop) while the roll was cycled from -3° (right corner) to 3° (left corner).

What *Wingeo3* does when analysing a geometry is test the suspension setup between the maximum limits of vehicle movement (such as the ones listed above) at equally spaced increments of these maximums, specified by the user. At each increment between these limits it then records the position of the geometry allowing the user, once the analysis has been run, to plot various parameters defined by the migration of the geometry points, against the respective movement of the vehicle. For example, figure 5.4 below details a plot of the camber on the right wheel as the chassis rolls from -3° to 3°. However, there are many other parameters that can be plotted along with many other vehicle behaviours other than roll.

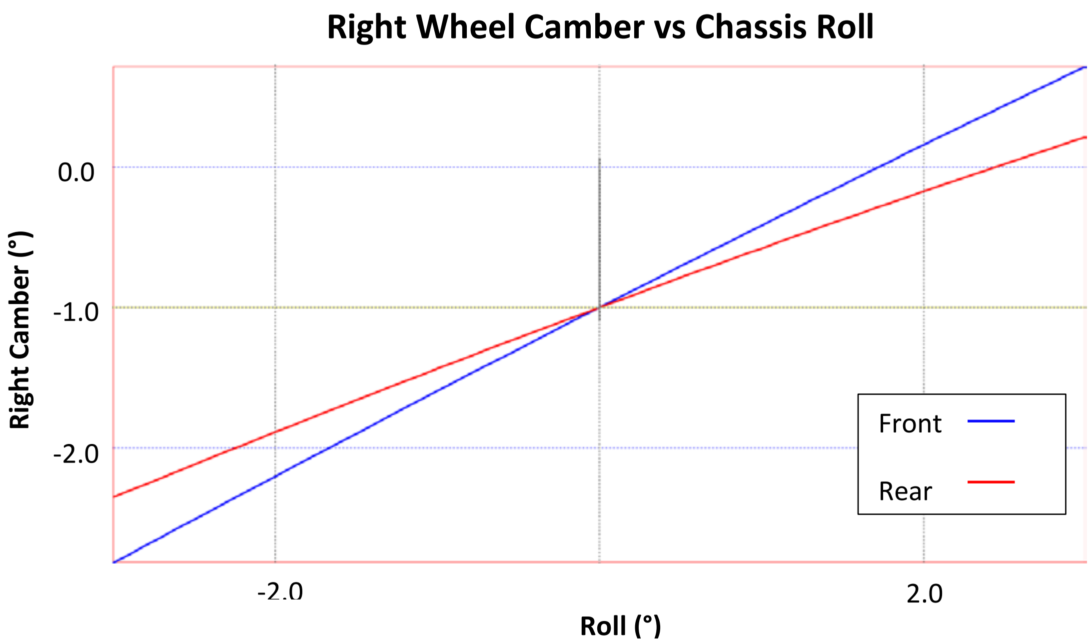


Figure 5.4: Wingeo3 plot of camber - chassis roll from -3° to 3°.

The chosen parameters to observe for each different geometry setup were the camber of the right wheel along with the roll centre location and these would be monitored for both the front and rear of the vehicle. As mentioned earlier there are are many other parameters that can be observed although it is believed that the camber and roll centre movement provided enough information about each geometry for a fair comparision to be made. This is based off material uncovered in the literature review.

Along with the *Wingeo3* plots, the location of the roll centre and camber at the maximum limits of vehicle movement were documented in an excel file (1 spread sheet per set of iterations) such that each different geometry could be compared in a simple manner. The table detailing the results for the final set of iterations is shown below as an example of one of these spread sheets.

Iteration	**RC static location (mm)**	**Camber in max droop (°)**	**Camber in max bump (°)**	**RC location in max droop (mm)**	**RC location in max bump (mm)**	**Camber in 3° roll (°)**	**Camber in -3° roll (°)**	**RC location in 3° roll (mm)**	**RC location in -3° roll (mm)**	**Displacement of RC (mm)**
i: Refined geometry from iteration 2 only requiring caster, trail and camber alterations	F: +28.864 R: +43.927	F: +0.594 R: +1.524	F: -2.860 R: -3.872	F: +90.081 R: +101.322	F: -30.869 R: -11.310	F: +0.766 R: +0.273	F: -2.889 R: -2.421	F:+27.423v, -157.045h R: +43.888v, -49.454h	F:+27.423v, +157.045h R: +43.888v, +49.454h	F: -1.441v, ±157.045h R: -0.039v, ±49.454h
ii: Front trail reduced from -43mm to -15mm	F: +28.864 R: +43.927	F: +0.593 R: +1.524	F: -2.857 R: -3.872	F: +89.988 R: +101.322	F: -30.850 R: -11.310	F: +0.766 R: +0.273	F: -2.891 R: -2.421	F:+27.406v, -156.316h R: +43.888v, -49.454h	F:+27.406v, +156.316h R: +43.888v, +49.454h	F: -1.458v, ±156.316h R: -0.039v, ±49.454h
iii: Front caster increased from -6° to -10°, trail left the same as in iteration i	F: +26.013 R: +43.927	F: +0.434 R: +1.524	F: -2.709 R: -3.872	F: +87.164 R: +101.322	F: -33.655 R: -11.310	F: +0.870 R: +0.273	F: -2.997 R: -2.421	F:+24.411v, -173.276h R: +43.888v, -49.454h	F:+24.411v, +173.276h R: +43.888v, +49.454h	F: -1.602v, ±173.276h R: -0.039v, ±49.454h
iv: Cambers of front and rear wheels adjusted to try and meet evaluation criteria (rear at -0.5° and the front at -1.5°). Everything else as in i	F: +29.443 R: +43.272	F: +0.104 R: +2.011	F: -3.374 R: -3.352	F: +90.609 R: +100.779	F: -30.208 R: -12.116	F: +0.256 R: +0.784	F: -3.382 R: -1.929	F:+28.063v, -152.918h R: +43.206v, -51.359h	F:+28.063v, +152.918h R: +43.206v, +51.359h	F: -1.38v, ±152.918h R: -0.066v, ±51.359h

Table 5.1: Results of iteration set 3 demonstrating the display of results in a spread sheet.

5.3.2.1 Testing Procedure

In summary, the following method describes the process used to test each separate geometry:

1. Geometry entered into Wingeo3
2. Geometry tested in bump and droop
3. Camber and roll centre location at max bump and droop noted in excel spread sheet
4. Geometry tested in roll
5. Camber and roll centre location at roll extremities noted in excel spread sheet

6. Plots created in Wingeo3 detailing the change of camber and roll centre location with chassis roll
7. Characteristics of plots noted in excel spread sheet

This procedure was used for every iteration group. After testing each geometry setup using this process, the results obtained needed to be assessed. The following chapter section details the criteria applied to make these assessments.

5.3.3 Evaluation Criteria

Relative camber of the front and rear:

According to Carroll Smith (1978, p54), the "…front camber curve should keep the laden wheel more upright in roll than the rear" as the front of the vehicle needs as much camber compensation as possible to stop the front end from washing out on the entrance to a corner; as a vehicle enters a corner lateral load transfer will compress the outboard front spring inducing a lot of roll at the front end of the vehicle.

Camber gain:

Does the camber go positive? This is particularly important in roll as the camber of the tyre will relate to the grip available from it. As most radial tyres will cope with a significant amount of negative camber but not a lot of positive (Clarke, 2004), this is quite important. Clarke also makes it clear that "Camber on a FSAE car should never go positive on a loaded wheel in cornering".

Location of roll centre:

The static height of the roll centre will dictate a number of characteristics and these need to be considered when comparing each geometry. Lower roll centres will minimise jacking forces, increase lateral movement of the roll centre (bad) and increase the roll moment (bad) while higher static roll centres will do the opposite. (Smith, 1978)

Movement of the roll centre:

Smith (1978, p54) also recommends that the front and rear roll centre movements should be approximately equal and in the same direction. On top of this, Staniforth (1999, p179) suggests that the movement of the roll centres should be restricted so that handling feel does not dramatically change as the vehicle goes through its various movements. Pat Clarke (2004) is another expert supporting the need to keep the roll centres from moving around excessively where he mentions that “A mobile roll axis will send confusing feedback to the driver, making accurate control difficult.” and makes note to “Pay attention to the migration of the roll centres under all conditions.”

Achievability of geometry:

This simply involves analysing each geometry for its ability to be integrated into a typical USQ FSAE vehicle package.

5.3.4 Results

Results of 1st Set of Iteration

The full set of results for iteration set 1 may be viewed in appendix D (table D.1). The green highlighted row in this table represents the best geometry layout based off the proposed evaluation criteria and was therefore the layout to be refined in the second round of iterations. As seen, the layout of this suspension is an unequal and unparallel setup featuring roll centres above ground and fairly low.

At this stage it was also appropriate that a few more decisions were made about the desired geometry otherwise further iteration would become unnecessarily excessive. Firstly, the roll centre locations were nominated to stay between the ground plane and 60mm above it as this placement proved successful in the first set of iteration and would result in shorter roll moments meaning less time to reach an equilibrium state when entering a corner as well as less lateral load transfer, while still allowing manageable jacking forces. Additionally, it was decided that the mounting points for the lower suspension arms were not to go below 100mm above the ground as placing them below this level would potentially cause integration problems with the chassis and may cause lower components on the vehicle to contact the track surface in bump.

Results of 2nd Set of Iteration

The aim of the second set of iterations was to identify some patterns in the geometry to find out what geometry characteristics contributed positively to the evaluation criteria listed above. These modifications all centred around the unequal and unparallel layout and tested the effect of varying the relative lengths of the upper and lower suspension arms, the height that the pickup points were mounted on the chassis, and lastly, the static roll centre heights. Based off these tests the following conclusions were drawn:

- With a larger difference in relative size of the upper and lower suspension arms (with constant roll centre heights) camber gain is slightly reduced in bump and droop as well as in roll, there is less roll centre movement in bump and droop, and finally, likeness of the front and rear camber curves decreases marginally. No significant patterns are recognised with the behaviour of the roll centre in in roll. To clarify, this test was carried out by maintaining the lower suspension arm geometry while shortening the top arm by moving the chassis mounting points outwards.

- Raising the chassis pickup points while maintaining constant roll centre heights reduces the amount of camber gain in bump and droop but increases it in roll, does not significantly affect the roll centre movement in bump and droop but increases it in roll, increases the likeness of the front and rear camber curves, and finally, increases the consistency of the front and rear roll centre migration. The process applied to test this behaviour involved lowering the height of the mounts for the suspension arms on the chassis while preserving the same roll centre location in each geometry configuration.

- Raising the roll centres by altering the height of the upper chassis pickup points while maintaining the ratio of roll centre heights between front and rear increases the amount of camber gain in bump and droop while decreasing it in roll, does not significantly change roll centre movement in bump and droop but causes its migration to decrease in roll, does not change the shape of the camber curves for the front and rear significantly, and lastly, improves the consistency between the movement of the front and rear roll centres.

- Combining raised roll centres with lower chassis pickup points produces the same results as each geometry alteration on its own although the front and rear camber curves are not as similar in this case.

Note: The table detailing these results in full is contained in appendix D (table D.2).

Using these findings a refined geometry that would provide optimised dimensions both in terms of performance as well as applicability to a USQ vehicle was defined. This geometry was then applied as the first setup to test in the third and final set of iterations where it was developed to further satisfy the evaluation criteria.

5.3.5 Final Geometry Configuration

Like the second set of iteration, the third also sought to find patterns in the behaviour produced when modifying and controlling elements of the geometry. In this instance though, the parameters modified were the caster, trail and camber. Based off the results of this iteration set, the following conclusions were drawn:

- Modifying trail does not change roll centre movement or camber behaviour much in bump, droop or roll.

- Altering caster produces changes more significant than when modifying the trail although these differences in the geometry behaviour are still not great. In particular it reduces the amount of camber gain in bump and droop while increasing it in roll, does not significantly affect the roll centre movement in bump and droop but increases it in roll, decreases the likeness of the front and rear camber curves in roll, and finally, does not detract much from the consistency that the front and rear centres move around with.

- Altering static camber of the wheels does not notably change anything apart from the extremities that camber reach in bump, droop and roll.

Interpreting these findings along with those to come from all previous iteration sets, the final geometry was established. The following table summarises this geometry.

Parameters	Front	Rear
Wheelbase	1600mm	
Track	1300mm	1175mm
Camber	-1.5°	-1°
Kingpin inclination	2°	1°
Scrub radius	25.0mm	15.5mm
Caster angle	-6°	11.936°
Trail	-20mm	83.642mm
Roll centre position (vertical)	28.979mm above ground	46.013mm above ground

Table 5.2: Summary of chosen suspension geometry.

With reference back to the elected evaluation criteria, the final geometry can be justified as the optimal solution for a USQ FSAE vehicle. Firstly, by nominating to have static cambers of -1.5° and -0.5° for the front and rear respectively along with raising the pickup points for the rear on the chassis, the front wheels always possess less camber (positive or negative) than the rear in roll, thus conforming to advice from Carroll Smith suggesting that the front wheel should be kept more upright than the rear in roll. The degree of camber gain for this setup prevents camber from going positive at the front of the vehicle but not at the rear where at the very worst case scenario of 3° chassis roll with the front right wheel is cambered at -0.105° while the rear angles at 0.698°. Although the rear camber goes positive, it is believed that this is sufficient for the design because it does not go positive by a substantial amount and additionally, better camber gain properties would not be able to be achieved from this geometry without other compromises being made and without going outside the packaging allowed by a typical USQ FSAE vehicle. These camber related behaviours may be viewed in the following figure highlighting the camber change against chassis roll for the final geometry.

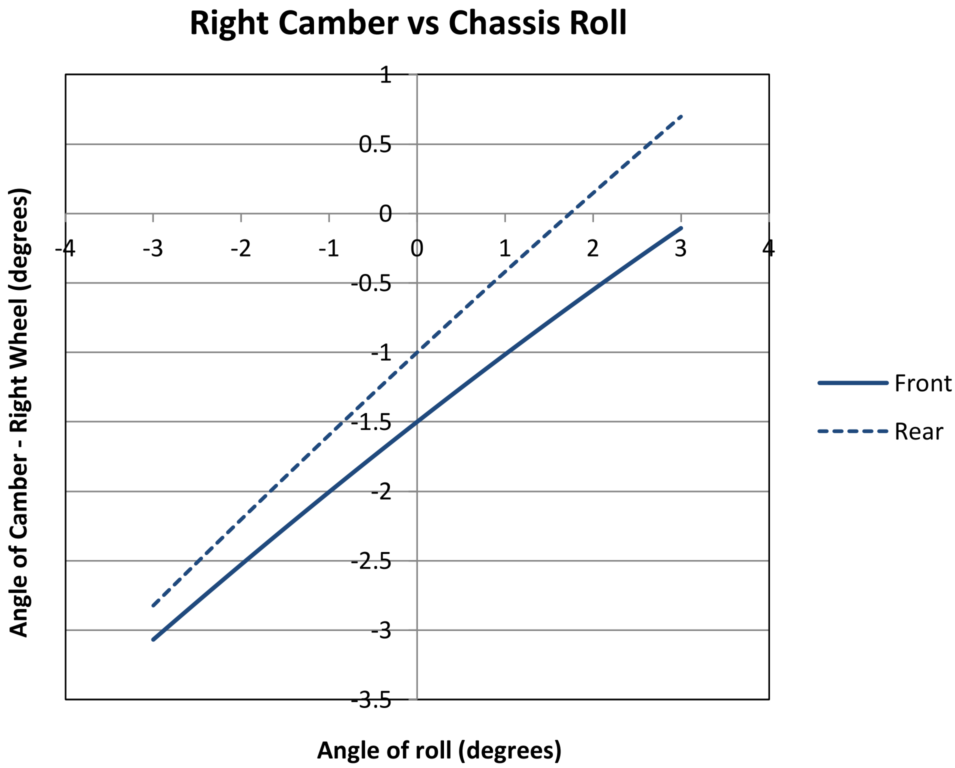


Figure 5.5: Right wheel camber plotted against chassis roll for the final geometry design.

The placement of the roll centre in this geometry was chosen for a number of important reasons. First of all, the rear roll centre height is slightly higher than the front to again comply with instruction from Smith. On top of this, the low to medium height static roll centre locations are hoped to produce manageable jacking forces, suitable control over the movement of the roll centres, and lastly, good turn in response for the vehicle due to the relatively small roll moment resulting in less rotational inertia. The actual movement of the roll centres is also expected to produce good handling results where again the design complies with the ideas of the suspension design experts; movement of the roll centres is fairly consistent between the front and the rear and does not reach large magnitudes in either the vertical or horizontal directions. These performance characteristics regarding the roll centre can be observed in the following figure which demonstrates the path the roll centre travels as the chassis is rolled from 3° to -3°. This figure has been plotted in *Matlab* using tabulated data exported from *Wingeo3.*

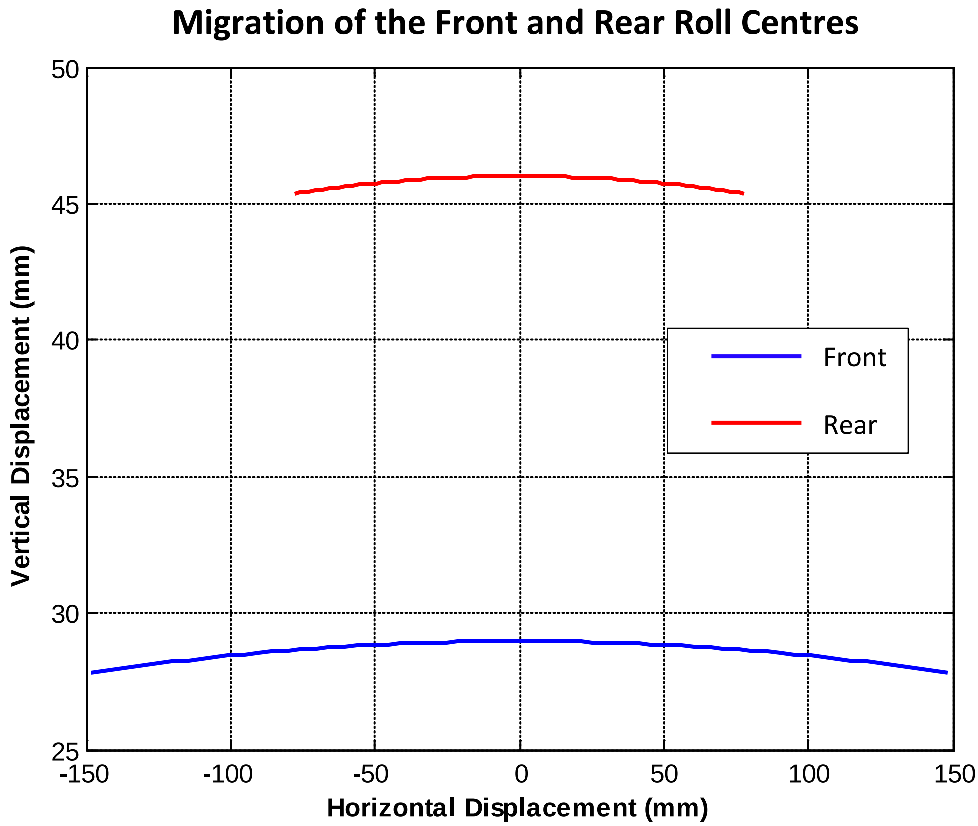


Figure 5.6: Migration of the front and rear roll centres under 3° to -3° chassis roll.

Last of all and most importantly, it is strongly believed that the elected suspension geometry will be adaptable to a future USQ FSAE vehicle. For a representation of the final suspension geometry arrived at through use of the chosen evaluation please view the following figure which presents the design in 3d space using a *SolidWorks* sketch. All geometry points listed are referenced from the centre of the vehicle at the front between the centre of the tyre contact patches. Also, the black arrows indicate the positive directions of the x, y and z axes and the coordinate points listed over the page are in the form (x, y, z) with dimensions shown in millimetres.

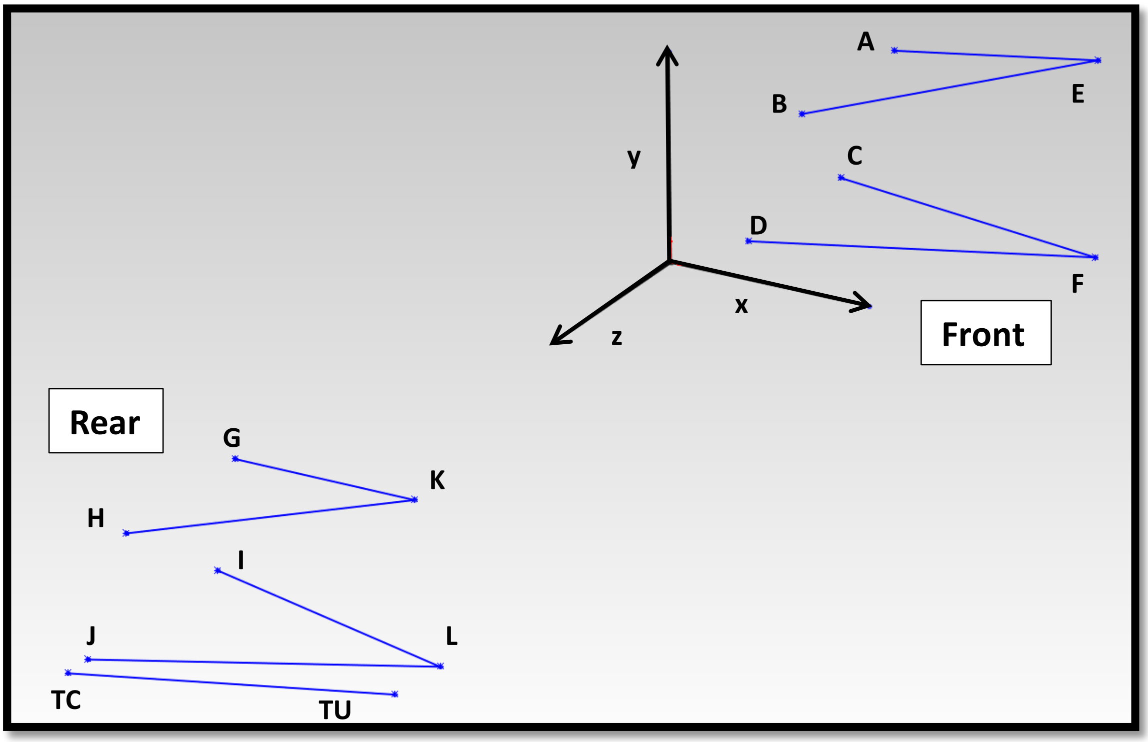


Figure 5.7: Geometry of final suspension design shown in SolidWorks.

A = (260, 300, -115)

B = (260, 300, 115)

C = (180, 110, -115)

D = (180, 110, 115)

E = (611.195, 394.991, -21.482)

F = (620.191, 138.352, 5.492)

G = (225, 330, 1465)

H = (225, 330, 1735)

I = (160, 140, 1400)

J = (160, 140, 1723)

K = (565.094, 395.494, 1600)

L = (569.388, 149.532, 1548)

TC = Toe link attach point on the chassis = (160, 140, 1773)

TU = Toe link attach point on the upright = (564, 149.532, 1652)

5.4 Chapter Summary

In this chapter the derivation process and final selection of the design's suspension geometry has been detailed. This included establishing the upright geometry based on kingpin inclination, scrub radius and packaging, defining the iteration evaluation process and associated design decisions and evaluation criteria, and lastly, discussion of the results obtained from the iteration evaluation process. The final geometry arrived at is expected to provide decent performance characteristics while also being able to be easily integrated with a typical USQ FSAE vehicle.

Chapter 6

Suspension Actuation

6.1 Chapter Overview

Once the suspension arm geometry had been defined using the *Wingeo3* iteration process discussed in the previous chapter, the selection and placement of the shock absorbers as well as design of their associated actuation mechanisms began. Chapter 6 documents the process used to design these elements of the suspension system along with the final solution believed to be optimal for application to a typical USQ FSAE vehicle.

6.2 Selection of Shock Absorber Model

The first step in this segment of the design process was to specify a shock absorber for the vehicle. To do this, a number of preliminary decisions were made which included selecting the amount of wheel travel desired, nominating an approximate range for the motion ratio of the wheel travel to shock absorber movement, and finally, whether or not the front and rear systems would utilise the same shock absorber model.

The minimum wheel travel permitted in the competition is specified as 50.8mm; 25.4mm jounce travel and 25.4mm rebound (2010, p43). For an FSAE vehicle it is believed that less wheel travel is better as this permits a lower ride height and thus centre of gravity height also. Additionally, because the tracks aren't typically rough it is supported that high amounts of travel are unessential. Therefore the design aimed for the minimum wheel travel of 50.8mm. In selecting a shock absorber it was thus important to find a model that provided enough stroke travel in both directions after sag from the static weight of the vehicle and motion ratio (wheel travel: spring travel) were taken into consideration.

Although the motion ratio was likely to be changed later on in the design process when the rockers and push rods were designed, it was believed important to make an early decision on an appropriate range for this design parameter in order to define what size of shock absorber was suitable. Having a motion ratio of 1:1 or larger meant that the shock absorber required would be smaller in size therefore improving the suspension system's packaging and weight. As the wheel travel was desired to be kept at a minimum and the size of available shock absorbers had a limit on how small they could be, it therefore seemed logical that the motion ratio should not stray too far from 1:1 as this allowed a relatively small shock absorber but also meant that the minimum wheel travel was able to be achieved. Pat Clarke (2004) also supports the decision to keep the motion ratio fairly low although this is for different reasons aside from the shock absorber size required. In particular he mentions to "Beware of bellcranks with aggressive multiplication ratios as these make the car very sensitive to minor adjustments."

While not as influential on the vehicle's performance as the two previous preliminary decisions, the choice of whether to use the same shock absorbers on the front and rear would also affect the way the car behaved. It was expected that using the same model front and rear would improve the predictability of the vehicle under various handling manoeuvres thus enhancing the feel experienced by the driver. Therefore it was decided to use a shock absorber model common at the front and rear of the car.

With these goals in mind, a shock absorber model that allowed them to be achieved was chosen. In selecting a model a very detailed and extensive analysis could have be carried out as there are so many types, manufacturers and models of shock absorbers that are applicable to an FSAE vehicle. To narrow this range down and to simplify the process, the selection only considered mountain bike shocks. Although this limited the design from a true optimal solution as not all possibilities were considered, it was alleged that this was not a major issue because the shock absorbers do not contribute as heavily to the overall performance of a design as some of the other components in a steering and suspension system.

With that being said, it is not believed that mountain bike shocks aren't a good solution for the shock absorbers in an FSAE vehicle. These types of shock absorbers are cheap, readily available, compact and light weight. However, they do typically come with a couple of disadvantages being that the damping and adjustment is not optimised for a motorised vehicle, let alone an FSAE car.

After checking through the range of shock absorbers from renowned (for range and price) sellers *EBay*, *Jenson USA*, *Chain Reaction Cycles* and *BikeWagon* using size, cost and adjustability as the selection criteria, two shock absorbers models were narrowed down. These were the 2005 *Manitou* Swinger 4-Way Rear Shock 200mm long x 50mm stroke and the 2012 *Fox Shox* Van RC Coil Rear Shock 190mm long x 50mm stroke.

The sizing of these shocks is very similar was chosen based off calculation of an approximate value of the required stroke needed to provide the minimum 25mm rebound travel specified by the 2011 FSAE competition rules using a 1:1 motion ratio after the sag from the weight of an FSAE vehicle was applied. This provided a rough indication of the stroke size needed to achieve a motion ratio in the range of 1:1 and hence, the overall size of the shock absorber that was best suited to the FSAE car's design. To do this, the smallest spring stiffness available for each model was taken (250lb/in for both) and then the heaviest weight of a past USQ vehicle (201.8kg on the rear wheels for the 2008 vehicle) was used to calculate the deflection or sag of the shock absorber:

$$W = ks$$

$$\frac{W}{k} = s$$

$$\frac{\left(\frac{201.8}{2}\right)(kg/shock\ absorber) \times 9.81\ (m/s^2)}{\frac{(250 \times 4.45) \times 1000}{1 \times 25.4}(N/m)} = s$$

$$s \cong 22.6mm$$

The formula applied in this calculation is referenced from R. C. Hibbler's text book entitled 'Engineering Mechanics – Statics' 11th Edition (2007, p84)

This result therefore indicated that the required stroke for the shock absorber needed to be at least approximately 48mm in order to provide the 25.4mm of jounce travel with a motion ratio of 1:1. However, this calculation also uncovered that as the required rebound travel was 25.4mm also, the 1:1 motion ratio could not be applied in the design because even with the softest spring, the largest jounce travel available would only be 22.6mm. This was not an issue though because a larger motion ratio could be defined later in the design process. That aside, the two models selected to compare suited the most as they provided stroke lengths best matched to the approximately required wheel travel. This meant that the design's wheel travel could be the same or very close to, the minimum value specified in the rules, which as mentioned earlier was a desirable outcome. The required length for the shock absorbers did not involve any calculation and was simply chosen by finding a model with minimal overall size that also provided a stroke length suited to the desired wheel travel talked about earlier.

Apart from size, there were also two other factors considered when selecting the final model, cost and adjustability. Out of these two considerations cost brought the biggest influence because as mentioned earlier, USQ teams typically don't operate on a large budget and the adjustability would not contribute as significantly to the performance of the suspension as some other aspects of the overall suspension system. The following table provides a summary of each shock with pictures of them shown in figures 6.1 and 6.2 over the page.

Properties	Manitou Swinger 4-Way	Fox Shox Van RC
Eye to Eye Length	200mm	190mm
Stroke	50mm	50mm
Mass (without spring)	426g	387g
Adjustment	Rebound, SPV platform pressure, SPV Volume, preload	Rebound, air pressure, low speed compression, preload
Price (with spring and bushings)	US$111.09	US$310.00

Table 6.1: Properties of the potential shock absorbers.

As can be seen the *Manitou* shock was quite a lot cheaper than the *Fox Shox* model and adjustments provided by each shock were very similar. The only other things separating the two models was mass with the *Fox Shox* weighing in slightly less than the *Manitou,* as well as the length where the *Fox Shox* was shortest by 10mm. Based on these features it was decided to opt for the *Manitou* Swinger due to its significantly lower price and comparable length, mass and adjustability to the more expensive *Fox Shox* Van RC. Ultimately it was believed the *Manitou* possessed better value for money and would suit a USQ team's design much better because of this.

Figure 6.1: 2005 Manitou Swinger 4-Way Rear Shock

Figure 6.2: 2012 Fox Shox Van RC Coil Rear Shock

6.3 Position of Shock Absorbers

The general position of the shock absorbers was discussed earlier in chapter 4, section 4.3 where it was stated that the shock absorbers would mount above the driver's legs at the front of the vehicle and above the engine and drivetrain at the rear with the shocks being actuated via push rods and rockers. Further refinement of these positions by specifying exact locations of the shock absorbers represented the first step in the technical design of the suspension actuation mechanisms. Without these placements the design of the rockers and push rods was unable to occur.

When selecting shock absorber locations, the design's packaging, simplicity, accessibility, and applicability to a typical USQ vehicle were all considered. However, it is important to mention that these locations were not locked in as later analysis involving the sizing of the rocker's would potentially change the required placement of the shock absorbers. Although these preliminary positions for the shocks aren't listed in the dissertation, values representing the final 3d coordinates of the shock absorber placements can be seen at the end of the chapter in figure 6.5.

6.4 Actuation Mechanisms and Spring Stiffness's

The actuation mechanisms and spring stiffness's were required to be designed in one process as both these decisions had to agree with each other and there was no direct way to calculate them both separately. The method applied to complete the design of these suspension system elements therefore involved iteration in order to find an optimal solution. The exact process employed is listed below.

1. Find the spring rate required for a 1:1 motion ratio
2. Test the sag of the suspension with the chosen spring rate and1:1 motion ratio
3. If the minimum travel (jounce or rebound) allowed by the competition is unable to be achieved than calculate the motion ratio needed for this to occur
4. Test the new motion ratio for required spring rate using natural frequency method
5. If the new spring rate differs from the original spring stiffness calculated for a 1:1 motion ratio than the 1:1 sag calculation must be repeated using the updated spring stiffness in order to calculate a new motion ratio that permits the minimum wheel travel before the whole process (steps 1-5) is repeated so that a motion ratio and spring stiffness that agree with each other can be found
6. Once motion ratio and spring stiffness is defined, design the actuation mechanism by finding the required rocker ratios and push rod lengths

This methodology relies on the requirement that the spring stiffness's are only chosen from the range that Manitou provide for their products. For example if, a required stiffness is found to be 150 lbs/in then the actual spring stiffness used in further calculation would be 250 lbs/in as this is the lowest and thus closest spring stiffness that *Manitou* provides (Chain Reaction Cycles, 2011). If the stiffness required ended up being between two set values from *Manitou* then the spring offering the closest stiffness to that required would be taken. If this wasn't the case then a much more extensive iteration process would be required in order to settle on a spring stiffness. However, this would provide a more optimised solution and so if further work was to be completed on the project than it is believed that extra work on this area of the suspension and steering system design would be well warranted.

There are a number of approaches suggested by suspension design experts used to select the correct spring rate for a racing vehicle. Allan Staniforth (1991, p186) along with William and Douglas Milliken (1995, p601) support that the spring's natural frequency can be used as a goal and starting point in selection of the vehicle's springs. Carroll Smith (1978, p69) on the other hand, holds a conflicting view on the frequencies associated with suspension rates where he states "we don't need to know about them". Based off the fact that little was known about the vehicle package being designed for and discussion on the FSAE.com forums supporting the need to consider suspension frequencies, it was nominated to use the methods documented by Staniforth and W. and D. Milliken.

The required spring stiffness for the vehicle was defined by analysis of the suspension movement in decreasing change of ride height or bump.

Ride Analysis

The frequency method outlined by Staniforth along with W. and D. Milliken involves selecting a desired natural frequency and then working back to find the required spring stiffness required to achieve this natural frequency. This exact relationship is defined by the following formulas which are documented in Allan Staniforth's book entitled 'Competition Car Suspension' (1991, p186)

$$Wheel\ Rate\ (lbs/in) = \left(\frac{Wheel\ Frequency\ (CPM)}{187.8}\right)^2 \times Sprung\ Weight\ (lbs)$$

$$Coil\ Rate\ (lbs/in) = Wheel\ Rate\ (lbs/in) \times Suspension\ Leverage^2$$

Using the heaviest sprung weight from a past USQ vehicle along with a desired wheel frequency of 145 cycles per minute (CPM) for a circuit racing car as recommended by Staniforth (1991, p183), the results of these equations indicated that for a motion ratio of 1:1, a spring weight of 119 lbs/in was needed at the front of the car and a weight of 133 lbs/in at the rear. As the lightest spring available for the chosen *Manitou* Swinger 4-Way shock is 250 lbs/in, it was therefore elected to use a spring weight of 250 lbs/in front and rear for the further calculations.

6.4.1 Motion Ratio Based on Required 1:1 Spring Stiffness's

As noted above, the first spring stiffness to run through the iteration process was 250 lbs/in at both the front and rear of the vehicle. The sag travels produced by these spring rates for a motion ratio of 1:1 at the front and rear were found to be 20.3mm and 22.6mm respectively. The formula used to achieve these values is featured earlier in the chapter under section 6.2 with the full worked calculations shown in appendix E. Therefore the 1:1 motion ratio didn't offer the required 25.4mm rebound specified in the rules and so the motion ratios were adjusted to increase the travel of the wheel versus the movement of the shock absorber. These motion ratios were calculated by dividing the minimum rebound allowed in the competition (25.4mm) by the previously derived sag travel at the front and rear. Completing this analysis found that at the front a ratio of 1.26 was required whereas at the rear 1.13 was appropriate. The working associated with these values is also shown in appendix E.

To verify if the calculated motion ratios agreed with the chosen spring stiffness's, the natural frequency coil rate calculations were repeated although using the updated motion ratio. On completion of these workings it was discovered that the actual required spring weights were 189 lbs/in at the front and 170 lbs/in at the rear once again meaning that a 250 lbs/in spring stiffness would suit the design calculations for the front and rear as this was the softest spring available from *Manitou*. This also meant that the motion ratio established earlier to achieve the minimum rebound travel of 25.4mm was suitable for the 250 lbs/in spring stiffness and that the iteration process did not need to continue.

6.4.2 Actuation Mechanism Design

With the motion ratio set as 1.26:1 at the front and 1.13:1 at the rear and the shock absorbers selected and located in the vehicle, the actuation mechanisms were now able to be designed. This process involved positioning and sizing the suspension rockers that actuated the shock absorbers along with the push rods that actuated the rockers. While doing this, the packaging, simplicity, accessibility and applicability of the design were again taken on board.

The first step in this process was to define the attach points of the push rods on the lower suspension arms. In selecting these locations it was desired to keep the push rods, rockers and shock absorbers in the same plane to promote simplicity and improve the smoothness and quality of the suspension actuation by reducing friction and ensuring all pivots and bearings only experienced forces normal to their rotating axes. This arrangement may be viewed on the following figure which indicates the actuation mechanisms represented in *SolidWorks* assemblies.

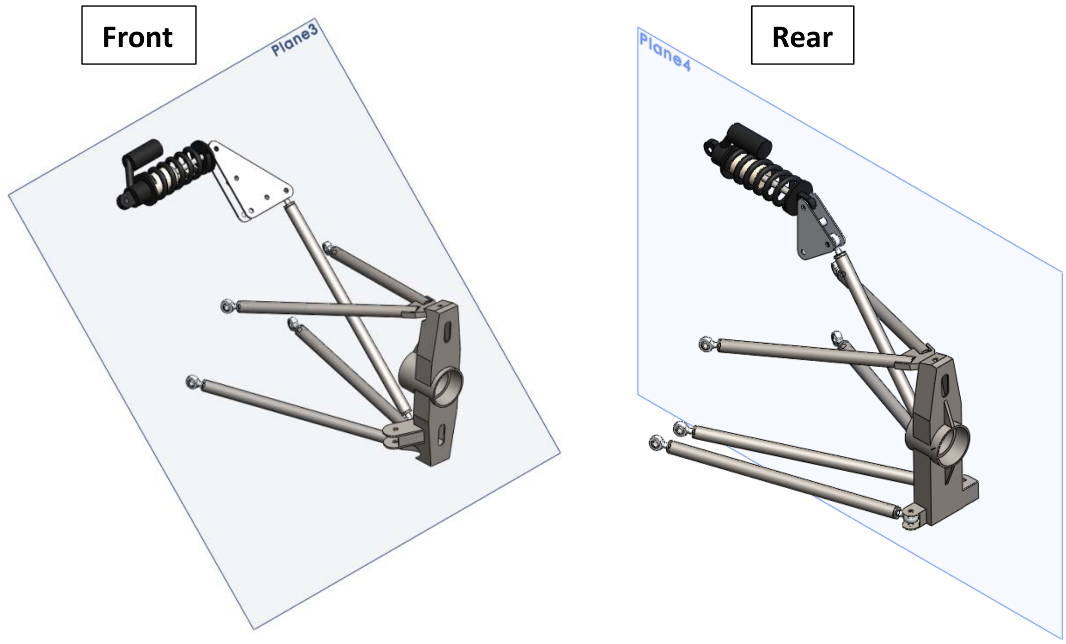


Figure 6.3: Suspension actuation mechanisms shown in their planes.

As noted in chapter 4 where the analysis of the 2008 vehicle was discussed, the suspension actuation on this car was not smooth due to actuation mechanisms that featured components out of plane which therefore put non normal forces on the cars pivot axes and bearings. Thus by nominating to not do this on the future design was expected to most definitely provide an improvement over the 2008 vehicle.

By choosing to have movement all in one plane for the front system, the position of the lower push rod attach point also defined the horizontal length of the rocker. As the rocker attach points on the shock absorbers were 80mm behind the suspension arm push rod attach point, this dimension was therefore 80mm.

For the rear system, defining the base dimension of the rocker was a little different as because the shock absorbers ran perpendicular to the car's length, the rocker base needed to be angled rather than horizontal in order for the push rod to push on it normally (see figure 6.3). Therefore the location of the lower end of the push rod had no influence on this dimension. The chosen length of the base for the rear rocker was nominated as 63mm.

These decisions defined the angle of the push rods to the vertical and from here the second length of the rocker (the one that would actuate the shock absorbers) was calculated in order to achieve the desired motion ratios calculated earlier. The calculations defining these values may be seen in appendix E and deduced that for the front, the second length of the rocker needed to be 91mm and for the rear the second length needed to be 70mm, therefore producing rockers that appear as below.

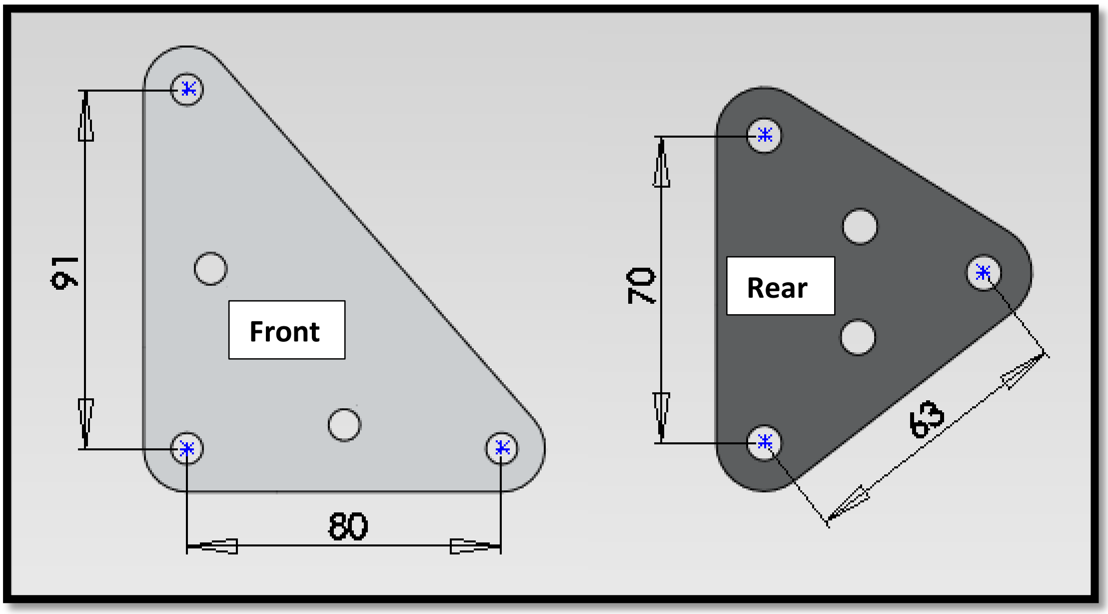


Figure 6.4: Critical dimensions of the front and rear rockers.

To accommodate these lengths a number of considerations were taken into account. For the front, increasing the vertical length of the rocker changed the position of the rocker's pivot location as well as the length of the push rod whereas at the rear of the vehicle, lengthening the vertical side only altered the height that the shock absorbers were placed at.

6.4.3 Frequencies of the System

Working back to find the new wheel frequencies provided by the chosen spring stiffness's, again using relationships discussed by Staniforth (1991, p186), it was found that the revised frequency of the front suspension system was 167 cycles per minute or 2.7833′ Hz and at the rear, 176 cycles per minute or 2.933' Hz. These calculations may be viewed in appendix E.

These frequencies were a little higher than desired 145 CPM frequencies although it was believed that this wasn't too much of a problem. The car may be a little rougher to drive although as the FSAE competition dynamic events do not run for long periods of time this design change is not seen to be an issue. If however, once the car was built, it felt very rough to drive even in short stints, custom made or generic springs could be investigated so that the desired wheel frequencies and handling qualities could be reached.

6.4.4 Design Issues

Although these parts have been located and dimensioned as realistically as possible in terms of application on a USQ vehicle, it is believed that there may need to be a significant amount of modification to the design to accommodate a future team's car. This is because it was hard to locate these components without knowing where potential mounting locations were on the chassis and where other components of the vehicle were packaged. Even though this is the case, it is believed that if a future group were to use this work, the methods and design ideas applied would still provide a significant aid to their design.

Another minor issue of this design exists with the rear actuation system. As can be seen back on figure 6.3, the rear push rod is very close to the upper suspension arm and with flex in the components or adjustment of the geometry, the rod could potentially foul on the arm. Without full analysis of the suspensions actuation and the system's adjustment this is cannot be confirmed though.

It should also be acknowledged that the calculations applied in the actuation mechanism design, as mentioned by W. and D. Milliken (1995, p601), present a simplified way of solving for the spring weight. As observed there is no equations that consider the effects of the vehicle experiencing a bump or rounding a banked turn. If future work was going to be carried out and the design was going to be physically produced, it would be useful to verify the stiffness's derived above with some of these types of calculations in order to achieve an optimal solution.

6.5 Roll Performance

Although the chosen spring stiffness's were suited to the vehicle in order to provide optimal suspension operation in jounce and rebound, the roll stiffness of these springs needed to be evaluated in order to determine whether a third spring or anti-roll bar was required to provide the desired roll performance for the vehicle. Pat Clarke (2004) also supports the need to consider an anti-roll bar for application to an FSAE car where he states anti-roll bars "are a good idea" and that "such devices are invaluable for fine tuning the handling to suit track or weather conditions".

The roll analysis carried out used equations stated by W. and D. Milliken (1995, p601-604) and first calculated the roll stiffness at the front and rear provided by the nominated 250 lbs/in springs along with the moment generated at the front and rear of the car caused by a lateral acceleration of 1g before using these two values to find out the roll rate of the front and rear or how many degrees of roll occurred per 1g of lateral acceleration. The equations used to evaluate these relationships may be viewed below and are a conglomeration of W. and D. Milliken's and the author's work.

$$Spring\ roll\ rate\ (lbs.ft/°) = \frac{Ride\ rate(lbs/in) \times Track^2(in^2)}{1375}$$

W. and D. Milliken, 1995, p601-604

$$\begin{aligned} &Roll\ moment, 1g\ lateral\ acceleration \\ &\qquad = (CG\ height(m) - RC\ height(m)) \times sprung\ weight(kg) \times g(m/s^2) \end{aligned}$$

$$Roll\ rate\ (°/g) = \frac{Roll\ moment, 1g\ lateral\ acceleration\ (N.m)}{Spring\ roll\ rate\ (N.m/°)}$$

Again the sprung weight of the vehicle was assumed to be the worst case or heaviest of past USQ vehicles while the centre of gravity heights were estimated using suitable engineering judgement.

After completing these calculations the roll rate for the front and rear were found to be 0.66 °/g and 1.22 °/g respectively. As a comparison, according to W. and D. Milliken (1995, p605) the typical roll rates for sedans range from 1.0-1.8 °/g while for aero cars the rates are generally within 0.25-0.5 °/g. As there is no magic number for the optimal roll rate of an FSAE car, further work would need to be done in order to determine if an anti-roll bar was needed. It is believed that a viable method to achieve this would be to obtain tyre data so that the maximum allowable camber on the tyres could be determined. From here the stiffness of the anti-roll bar required to stop this maximum allowable camber could be determined.

6.6 Specifications

Once the shock absorbers were chosen and their actuating devices were designed, the geometry points that defined them were entered into the existing *Wingeo3* model where the performance specifications of the system were determined. The table over the page provides a summary of these specifications.

As observed in this table the wheel rate for both the front and the rear increases as the shock absorbers compress. If they decreased, the suspension would not ramp up when the shock absorbers reached their limits of stroke which is very bad as the car would be more prone to bottoming out. In this case, the position and size of the push rods would need to be redesigned.

It can also be seen that these wheel rates do not dramatically increase through the shock absorbers compression and this is also good. Rapid increase of the wheel rate results in poor use of the suspension travel as the when the car encounters a bump the wheel rates stiffen excessively, preventing the shock absorber from reaching its full stroke.

The equations used to derive these wheel rates are provided by Carroll Smith in his book entitled 'Tune to Win' (1978, p65) which use data obtained from testing in Wingeo3. These formulas are featured over the page.

Another finding to come from the brief *Wingeo3* analysis was that the motion ratio for the rear mechanism varied a little from the required value which was 1.13. It was unsure as to why this was and thus potential future work could be to carry out a more extensive investigation into this behavior. The fact that the ratio was a little larger than required was not deemed a major issue though because the required rebound travel was still achievable.

Parameters	Front	Rear
Type of suspension actuation	Inboard, push rod actuated	Inboard, push rod actuated
Shock absorber model	2005 Manitou Swinger 4-Way 200 x 50	2005 Manitou Swinger 4-Way 200 x 50
Spring stiffness	250 lbs/in	250 lbs/in
Static motion ratio	1.26:1	1.13:1
Wheel frequency	3.5 Hz	3.3166' Hz
Wheel travel at 1mm compression	1.249mm	1.227mm
Wheel travel at 25mm compression	30.889mm	30.656mm
Motion ratio at 1mm compression	1.249:1	1.227:1
Motion ratio 25mm compression	1.234:1	1.226:1
Wheel rate at 1mm compression	160.3 lbs/in	166.1 lbs/in
Wheel rate at 25mm compression	164.2 lbs/in	166.3 lbs/in
Ride height adjustment	Push rod length modification via rod ends, pre-load of shock absorber	Push rod length modification via rod ends, pre-load of shock absorber
Roll rate	0.66 °/g	1.22 °/g

Table 6.2: Summary of suspension actuation specifications.

Where,

$$Motion\ ratio = \frac{Wheel\ travel\ (m)}{Spring\ travel\ (m)}$$

$$Wheel\ rate = \frac{Spring\ rate\ (lbs/in)}{Motion\ ratio^2}$$

The geometry coordinates that define the suspension actuation mechanisms are shown over the page. This figure represents the front and rear mechanisms shown in a 3d sketch created with use of *SolidWorks.* All geometry points listed are referenced from the centre of the vehicle at the front between the centre of the tyre contact patches. Also, the black arrows indicate the positive directions of the x, y and z axes and the coordinate points listed over the page are in the form (x, y, z) with dimensions shown in millimetres. The broken lines represent the shock absorbers.

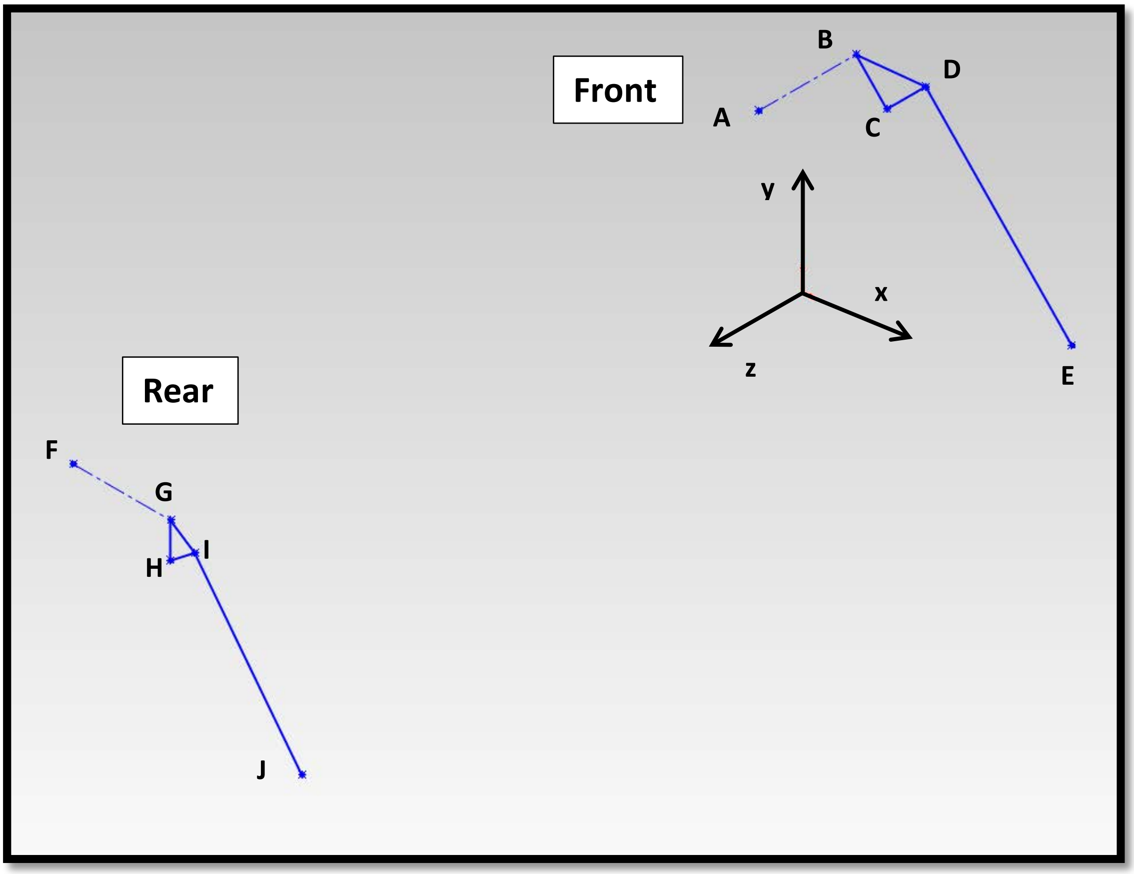


Figure 6.5: Geometry of the suspension actuation mechanisms.

A = Front shock absorber mount = (190, 555.666, 280)

B = Front shock absorber mount = (190, 555.666, 80)

C = Front rocker pivot point = (253.3, 490.3, 80)

D = Front push rod connection to rocker = (253.3, 490.3, 0)

E = Front push rod connection to suspension arm = (570, 163, 0)

F = Rear shock absorber mount = (50, 490, 1548)

G = Rear shock absorber mount = (250, 490, 1548)

H = Rear rocker pivot point = (250, 420, 1548)

I = Rear push rod connection to rocker = (300.1, 458.7, 1548)

J = Rear push rod connection to suspension arm = (520, 174, 1548)

6.7 Chapter Summary

The documentation presented in this chapter has described the design of the suspension actuation mechanisms. This design step has involved selecting the shock absorber models, positioning the shock absorbers, calculating the required spring stiffness's, motion ratios and rocker ratios, before lastly considering the system's roll stiffness and investigating the need for an anti-roll bar.

The chosen shock absorber model is the same at the front and rear of the vehicle and was the 2005 *Manitou* Swinger 4-Way rear shock chosen based on size, wheel travel required, cost and adjustability. The process used to select this model was simplified quite a bit and so a true optimal solution was unable to be achieved. Spring stiffness's derived to suit these shock absorbers were 250 lbs/in at both the front and rear of the vehicle.

The spring stiffness's and system motion ration were calculated in one process as both these design parameters had to agree with each other and there was no direct way to calculate them both separately. The result of this design procedure found the required spring stiffness's to be 250 lbs/in for both the front and rear of the vehicle while the motion ratios were calculated as 1.26:1 and 1.13:1 for the front and rear respectively.

Following the specification of the geometry for the actuation method, further calculation showed that the design featured wheel frequencies of 167 cycles per minute or 2.7833′ Hz at the front while at the rear, 176 cycles per minute or 2.933' Hz. These frequencies were higher than desired although this was not deemed a major problem.

Roll rate analysis carried out uncovered that with only the shock absorber springs installed, roll rates at the front and rear of the vehicle were 0.66 °/g and 1.22 °/g respectively. It then was determined that further analysis was required in order to verify if an anti-roll bar needed to be applied.

Although the design process used to obtain the above listed design parameters was quite extensive it is believed that the final suspension actuation system is not a fully optimal solution. This is due to the fact that the geometry of the system was hard to define without a known chassis configuration and that the method used to select the spring stiffness's was largely simplified.

Chapter 7

Steering Design

7.1 Chapter Overview

The steering geometry and design was the last task completed in defining the full suspension and steering system. In this chapter the placement of the rack and pinion and consideration of Ackermann geometry will be discussed in relation to their influence on the final steering geometry design. Along with this, the component design and manufacture of a steering rack and pinion and consideration of toe angles will also be detailed.

The steering design relies on a number of earlier decisions made concerning the design of the suspension system. These factors include the orientation of the uprights which dictates the required geometry of the steering arm as well as the location of suspension system components which affects the steering system's packaging.

7.2 Placement of Rack

There are four main options when positioning the steering rack. These are to place the rack above the driver's legs at the top of the cockpit or below the driver's legs at the base of the cockpit and also whether to position it forward or rearward of the front axles. Out of these two considerations, the vertical placement of the rack will have the biggest impact on vehicle's design. Each configuration has a number of advantages and disadvantages which are summarised below.

Steering Rack Above Driver's Legs:

Advantages:

- No steering shaft in cockpit
- Steering shaft requires no joints or bends

Disadvantages:

- Raised vehicle centre of gravity
- Steering rack is very close to driver's legs and may cause injury in an accident
- The packaging of the shock absorbers and their actuating mechanisms is compromised as they are also mounted around the top of the vehicle's cockpit. Additionally, on the 2008 vehicle all the electronic devices mount at the top of the cockpit and so these may need to be moved elsewhere for this design to work
- If bump steer is not wanted in the design, mounting the rack up higher will mean that it also needs to be bigger in width to comply with design techniques used to remove bump steer

Steering Rack Below Driver's Legs:

Advantages:

- Lowered vehicle centre of gravity
- No risk of leg injury caused by the rack in an accident
- Rack packages well with the chosen suspension system design
- Smaller rack required in order to reduce bump steer

Disadvantages:

- Steering shaft is in cockpit between drivers legs which potentially creates a hazard when the vehicle needs to be evacuated
- Steering shaft requires a bend and joint to link up with the steering wheel

As observed, positioning the rack low in the cockpit has the least amount of disadvantages by a significant margin and consequently this was the orientation chosen for the design. Additionally, the rack was chosen to be placed rearward of the front axles as this provided the best clearance for the brake rotor on the upright as the steering arm angled back toward the centre of the vehicle to accommodate Ackermann geometry. It is also believed that this will make the foot space in the cockpit less cramped as the rack is positioned under the arch in the driver's legs rather than near their feet where there is less space available.

7.2.1 Height of Rack

The vertical position of the rack has been nominated as 140mm up from the ground as in this location, it is believed that there will be enough room in the cockpit for the driver and the rack should be able to be packaged well with the chassis. This is important as the 2011 FSAE competition rules (2010, p38) state that the vehicle's cockpit must complete a test whereby a template representing the minimum space required in the corridor where the driver's legs are be placed, is passed along the length of the cockpit and if the template is unable to travel this path than the team will not be permitted to compete in any of the dynamic events.

Knowing the height of the rack, the appropriate rack length required could be calculated. Choosing the correct length would eliminate bump steer which as discussed earlier in the literature review, is the undesirable phenomenon whereby the wheels are steered unintentionally as the suspension goes through its travel. To do this, methods suggested by Woodward Steering (2010, p64) were applied.

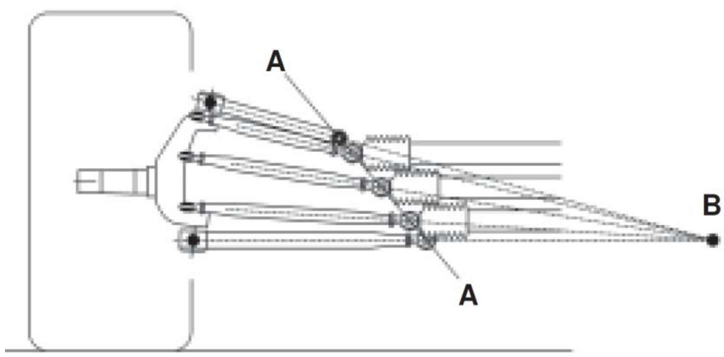


Figure 7.1: Determining length of steering rack and height of steering connection – shown from front or rear of the vehicle. (Woodward Steering, 2010, p64)

With reference to figure 7.1 back over the page, the length of rack is determined by where the steering rack needs to attach to the tie rod which is the link that connects the steering rack to the steering arm on the wheel upright. As highlighted in this figure, in order to eliminate bump steer, this connection point needs to lie on a line drawn from the pickup points of the top suspension arm to the pickup points of the lower suspension arm (line AA). To apply this method to the design's suspension geometry, a simple sketch was established in *SolidWorks* where the critical dimensions were easily measured. This analysis found that the distance from the tie rod connection on the steering rack to the middle of the car needed to be 192.632mm and thus the rack length required was therefore 385.264mm. For the derivation of this dimension see appendix F.

Another measure to eliminate bump steer advocated by Woodward Steering (2010, p64) was to ensure that the steering tie rod aligned with the line drawn from the cars instant centre (point B) and then through the tie rod's connection point on the steering rack. This feature was thus added to the *SolidWorks* sketch created earlier to determine the height of the rack. The placement of the tie rod helped define the geometry of the steering arm needed on the front uprights.

7.3 Ackermann and Steering Range

As stated earlier, Ackermann steering geometry steers the inside wheel further than the outside while cornering in order to reduce scrub of the tyres. Also mentioned was the fact that this type of steering geometry is best suited for low speed vehicles required to make tight corners. For this reason it is was chosen that Ackermann steering should be applied in the cars design. Pat Clarke (2004) also supports the need for Ackermann in an FSAE vehicle where he states in his article aimed at preparing new teams for the FSAE competition, "Does it follow that low speed cars on a tight track need positive Ackerman? You bet it does!"

The amount of Ackermann employed in the steering system is defined by the geometry of the steering arm on the upright. To determine this geometry, *SolidWorks* sketches were again used although to come up with an optimal solution involved a bit more analysis than was required to locate the height of the steering rack. Firstly, it was necessary to find out how far the left and right wheels needed to be steered in order to produce no scrub on the tightest corner on an FSAE competition track.

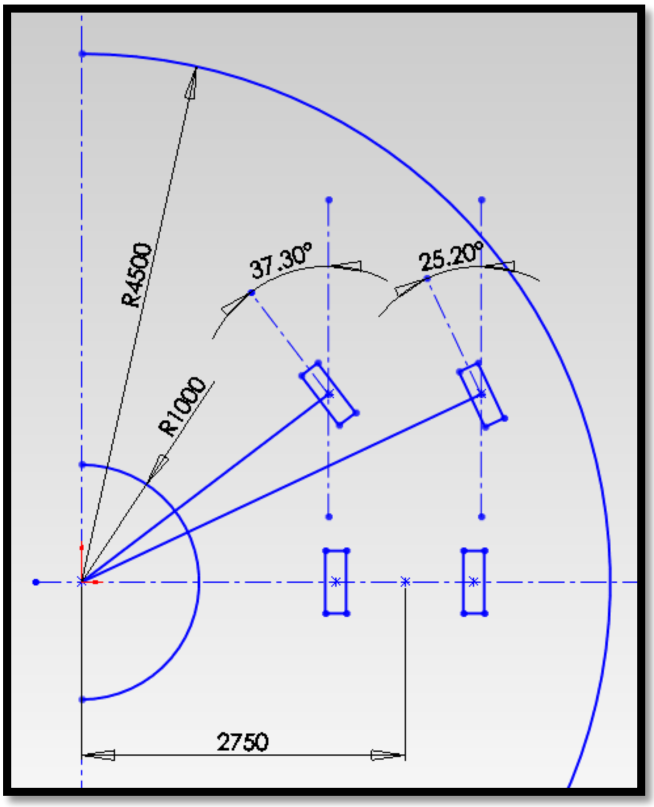


Figure 7.2: Determining front wheel steer angles that produce no scrub on the tightest turn on an FSAE track.

The 2011 FSAE competition rules (2010, p108) state that the minimum outside radius of a turn in an autocross track is 9m. Coupled with the information that the minimum track width is 3.5m which would locate the inner radius of this corner, the smallest radius that the vehicle needed to turn around could therefore be calculated. This was achieved by taking the average of the inner and outer corner radiuses defined by the above limits stated in the rules. The tightest radius required to be turned was thus (1000mm+4500mm)/2 = 2750mm which is shown in the figure above. Also shown on this diagram are the angles of the front wheels required to produce no scrub on this tight corner. These are 37.3° for the unloaded inside wheel and 25.2° for the loaded outside wheel.

These steering ranges were thus made a goal for the steering arm design. To arrive at an optimal solution iteration was used, where the amount of Ackermann and length of the steering arm were modified. Apart from the steering range, the steering ratio (steering wheel angle: wheel angle) was also a consideration for each iteration result. Using a larger ratio would mean that vehicle movement would be less sensitive to driver input and would also mean that the driver would fatigue less over long periods of driving as the force required to turn the wheels is lessened. As a guideline, W. and D. Milliken (1995, p716), state that the steering ratio for race cars typically ranges from 20:1 to 10:1 while a vast majority opinions expressed on the FSAE.com forums indicated that the steering ratios employed in an FSAE vehicle generally ranged from around 4:1 up to around 10:1.

To determine this ratio for each iteration, a desired lock to lock steering wheel range was selected. This needed to be large enough to ensure the steering ratio did not become too low, meaning that the car would be hard to drive over long periods but also small enough so that the driver could make it round the tightest turns on the track without running out of space in the cockpit or getting their arms tangled up trying to get the steering wheel to full lock. Based on these findings along with opinions expressed in the FSAE.com forums, the chosen lock to lock steering range for the new design was 270°.

The iteration process involved altering the Ackermann and steering arm length in *SolidWorks* to find the geometry points of the steering arm. This geometry also complied with the earlier design recommendation by Woodward Steering (2010) regarding the angle of the tie rod. From there the geometry was entered into the *Wingeo3* model where the steering range was tested to its maximum limit. This process was then completed for each iteration and the results tabulated. For the results table and further description of the iteration process, see appendix F.

With reference to this table, the highlighted green row represents the chosen design geometry. It was chosen as the superior option because it best suits the steering range required to make the tightest turn on the autocross track and also has an appropriate steering ratio. Although some of the other iterations had a better steering ratio this was only because they could only achieve a small steering range resulting in a higher max ratio of steering angle to angle of the wheels. These other options were not considered as they didn't get close enough to the steering range required to make the tightest turn on the autocross track. As can be seen, the steering system therefore possesses 125% Ackermann, steers the inside and outside wheels at full lock 38.290° and 25.893° respectively, and lastly, will incorporate a steering ratio of 4.21:1.

7.4 Rack and Pinion Design and Manufacture

The selection or design of a steering rack is not considered in the project due to limited time. However, it is important to mention that this should be a significant consideration for future USQ FSAE design. As the 2008 vehicle utilised a steering rack from a small car, performance was compromised and consequently, as mentioned earlier, the steering required extra steering wheel rotation to achieve full lock and did not reach a very large steering range which was a major problem as it made the vehicle harder to manoeuvre around the tight FSAE track. For this reason it is supported that the steering rack and pinion should be the last thing designed in the steering system which would probably mean that the steering rack would need to be physically designed from scratch to achieve the specific ratio required. Although this may prove more costly than sourcing a second hand rack from a domestic car or similar vehicle, it would also provide the optimal steering solution which, as the FSAE competition places so much emphasis on handling ability, is believed to be a justifiable design decision.

If this rack were constructed, based on the steering geometry finalised in the iteration process, the rack and pinion steering c-factor or rack movement per 360° rotation of the steering wheel could be derived. Based on the equation suggested by W. and D. Milliken (1995, p718) the c-factor was calculated to be 77.33'mm/360°. These equations and calculations are featured in appendix F. The c-factor is a value commonly used in the industry to specify the ratio of a rack and pinion. Also, it is believed that if possible, the chassis mounts for the steering rack should have a degree of adjustability. Vertical adjustment would ensure bump steer could be minimised in the system while forward and rearward movement would alter Ackermann and toe angles if required.

7.5 Toe Adjustment

The final consideration for the steering design was the static toe applied to the front and rear wheels. As stated in the literature review, a controlled amount of toe out at the front of the can improve the vehicle's turn in response when entering a corner however, toe out at the rear of the vehicle should be avoided as it produces unpredictable and potentially dangerous handling characteristics. On the other hand, toe in at either the front or rear promotes better straight line stability. Based on this information it was desired to use a manageable amount of static toe out at the front wheels while at the rear either use 0° toe or a slight toe in to promote stability and predictable handling performance. This is the furthest the toe design was taken in the project, once again due to a lack of time available. To arrive at some actual numbers it is believed that the best method would be to do some physical testing on the vehicle to work out what setup was the fastest and what felt best for the driver.

7.6 Specifications

The table shown over the page indicates a summary steering system's specifications. The values shown here have been derived from the design process documented in the preceding sections of this chapter along with testing in the Wingeo3 model. As can be interpreted from this summary, the bump and roll steer is relatively small which based off the information featured in the literature review, is a good thing. Bump and roll steer will cause the vehicle's wheels to steer unintentionally as the vehicle either hits a bump or rolls in a corner (or even both), delivering unpredictable feedback to the driver.

Although these values are small, it is believed that further improvement on these figures could be achieved with adjustments to the steering system once the car had been manufactured and assembled. This would involve using spacers to change the height of the steering arm tie rod connection point and steering rack tie rod connection point but if the bump steer was severe enough, the height of the rack could be adjusted.

The geometry of the steering system is shown in the figure that follows the specification table and lists the locations of the coordinates that define the components of this system in 3d space, once again using *SolidWorks*. Also, like the geometry figures shown in the previous two chapters, the reference point is taken from the middle of the car at a point exactly between the centre of the front tyre contact patches with the black arrows presenting the positive direction for each axis and the coordinate points following the figure in the form of (x,y,z). Also, the broken line represents half of the rack and pinion length,

Parameters	Future Steering System
Steering rack location	140mm vertically from ground plane, 60mm behind front axles (below driver's legs towards the knees)
Steering rack length (left tie rod connection to right tie rod connection)	385.264mm
Steering arm's tie rod connection location	179.447mm vertically from ground plane, 60mm behind front axles, 588.731mm from centre of vehicle
Static Ackermann	125%
Steering arm length	67.044mm
Steering ratio	4.21:1
C-factor	77.33'mm:360°
Steering wheel range	270° lock to lock
Max steer angle of inside wheel	38.290°
Max steer angle of outside wheel	25.893°
Bump Steer at max suspension compression (-25mm ride)	Left wheel : -0.011° toe out Right wheel: -0.011° toe out
Roll steer at max chassis roll (3 degrees)	Inside wheel: -0.025° toe out Outside wheel: -0.019° toe out
Ackermann adjustment method	Adjustable steering rack position
Static toe adjustment method	Adjustable steering rack position, tie rods and toe links

Table 7.1: *Summary of steering specifications.*

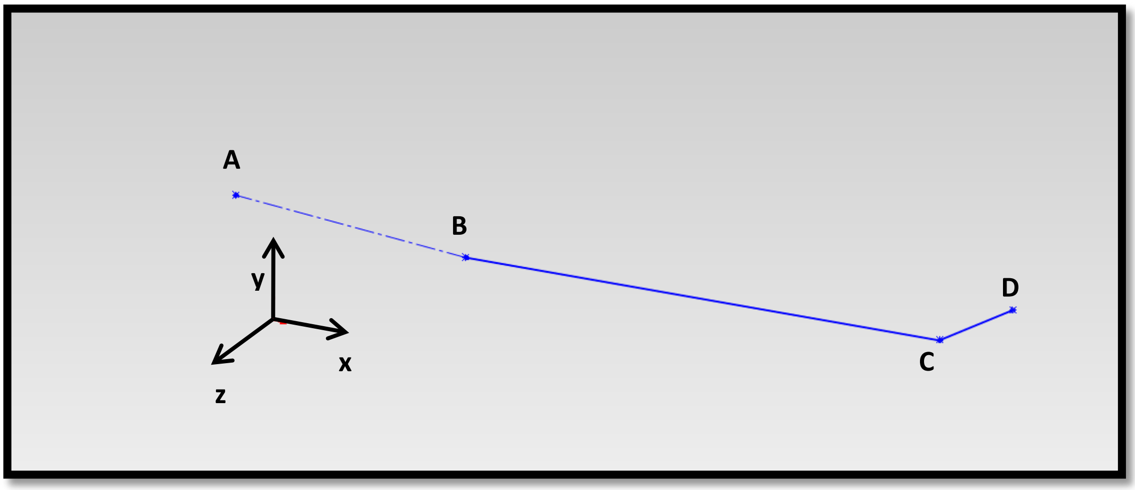


Figure 7.3: *Geometry of the steering system.*

A = Middle of rack and pinion = (0, 140, 60)

B = Rack and pinion tie rod connection point = (192.632, 140, 60)

C = Steering arm tie rod connection point = (588.731, 179.447, 60)

D = Upright steering pivot point = (618.646, 182.426, 0.86)

7.7 Chapter Summary

This chapter has documented the design of the steering system and the processes that have shaped this final design. These processes included positioning the rack and pinion along with choosing the amount of Ackermann employed through use of an iteration process. Additionally, consideration for future manufacture of the rack and pinion as well as toe angle adjustment has also been discussed.

The placement of the steering rack and pinion was chosen to be at the base of the cockpit and rearward of the front axles as this provided the most advantages to the overall design and the optimal amount of Ackermann to be used calculated as 125%. The chosen final design also presented a ratio of 4.21:1 and a c-factor of 77.33'mm/360°.

Toe angle was not specified as it was believed that physical testing would be required to arrive at the appropriate values.

Chapter 8

Component Design

8.1 Chapter Overview

So far the dissertation has only considered the geometry of the whole design. The following chapter defines the next step on from this where parts making up both the suspension and steering geometries are physically modelled. On top of this, each component's material and anticipated manufacturing process is detailed.

8.2 Modelling Process

As the geometry was fully defined for the steering and suspension systems, modelling of the components that made up these systems could ensue. Although, all these parts were modelled, the design of each component is only at a concept stage. This was due to an absence of time towards the conclusion of the project. Therefore the models presented in this chapter are purely an indication of how the system may look like if it were to be designed fully and manufactured. In arriving at these designs inspiration from past USQ vehicles along with designs of past FSAE competitors were used as a guideline. On top of this, general engineering knowledge and judgement were also applied. All models have been created and assembled in *SolidWorks* and there are no detail drawings available due to the limited completeness of the models. Also, for this reason no FEA has been carried out to further optimise the designs.

Assembled, it is also unknown if all the components will interact without fouling as the vehicle goes through the various movements of suspension actuation and steering input. If more time was available this would be a major consideration before final detail drawings could be formed.

8.3 Uprights

The upright concepts for the front and rear are pictured on the following two figures. These components are fairly simple and feature mostly square edges and profiles. If they were to be built it is expected they'd be made from mild steel rectangular and square hollow sections as well as plate, with the circular bearing housing at the centre of the upright, cast and machined or simply machined. The cut outs and holes would be completed with a milling machine and drill. All steel sections would then be welded together to form the upright. Although this bulky steel design would be heavier than some of the low weight setups used that employ lighter materials such as aluminium alloy along with more complex designs, it is also believed the uprights would be more robust, simpler to modify if last minute changes were required, and also easier to repair if an accident in the car damaged them.

Components directly mated to the uprights are only the axle bearings. As no axle design was included in the project, these bearing housings are identical to those featured on the 2008 vehicle. As shown on the concept figures there is also no mounts for the brake callipers. If more time was available then these mounts would be considered however it was felt that they were not imperative to the project's suspension and steering system design outcome.

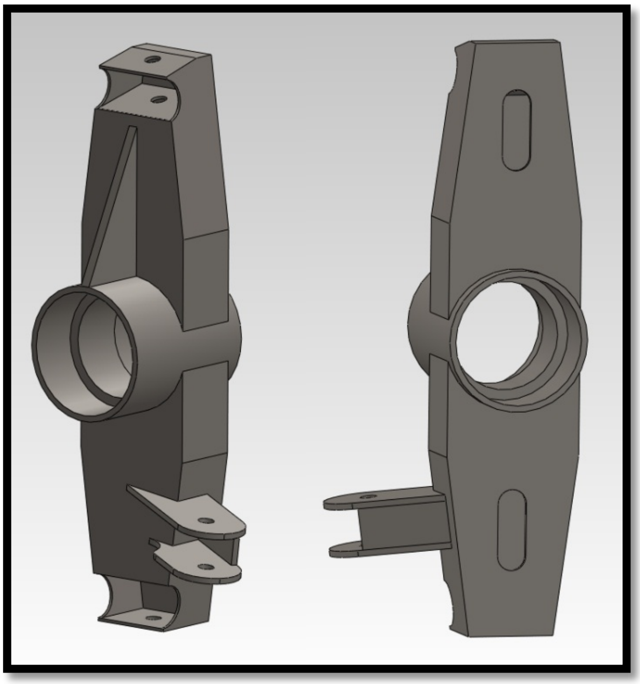


Figure 8.1: Front upright concept.

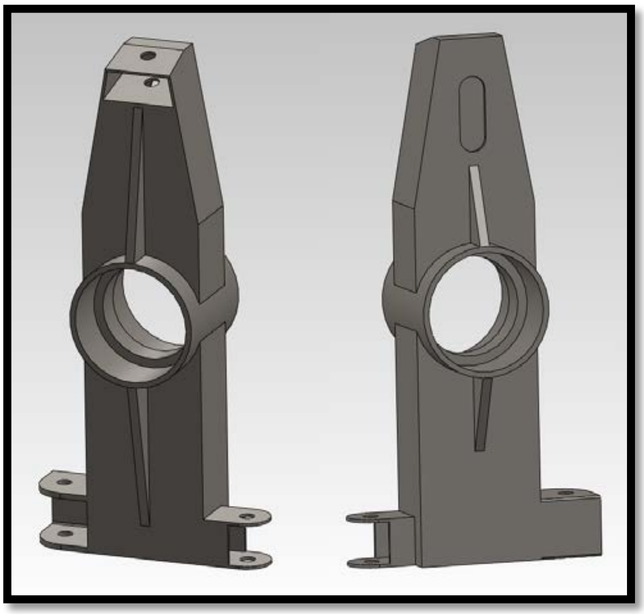


Figure 8.2: Rear upright concept.

8.4 Suspension Arms, Tie rods, Toe Links, Push Rods

Material, construction and design of the suspension arms, tie rods, toe links and push rods is all very similar. As with the uprights, the intended material is mild steel due to its decent strength, rigidity, and design flexibility/repairability. For the tie rods, toe links and push rods this steel would be in the form of circular hollow section (CHS) tubing while the suspension arms would also use this tubing but also incorporate machined steel plate for the push rod mounts along with plasma cut/machined steel profiles to connect each tube and to house the spherical bearing used to accommodate the upright. The size of this CHS tubing has been maintained for all of these components and once again it is intended that each part component will be welded together.

The relative size of this CHS tubing is shown on the following two figures which represent the lower suspension arms for the front and rear. The design of these components takes inspiration from the 2008 vehicle and as seen, is quite simple. Like the 2008 vehicle, one particular issue with this design could be controlling the geometry when the parts are welded. In order to avoid distortion and misalignment of the CHS tube and plasma cut steel sections appropriate welding and jigging processes would be essential.

Directly associated with the suspension arms, tie rods, toe links and push rods is the end plug which is also featured over the page in figure 8.5. This plug is intended to be inserted in the ends of the CHS tube before being welded in place. These plugs will facilitate the rod ends which are required to provide angular rotation of the suspension and steering components and so will feature threads on the inner hole. For components requiring plugs at both ends (tie rods, toe links and push rods) these threads will need to be both right and left handed so that the rod ends do not loosen under operation of the vehicle.

Along with the end plugs, the suspension arms will also mate with the spherical bearings used to support the uprights. These will press into the machined hole in the plasma cut steel section that joins the two CHS tubes of each suspension arm. This bearing will be detailed later in section 8.6.

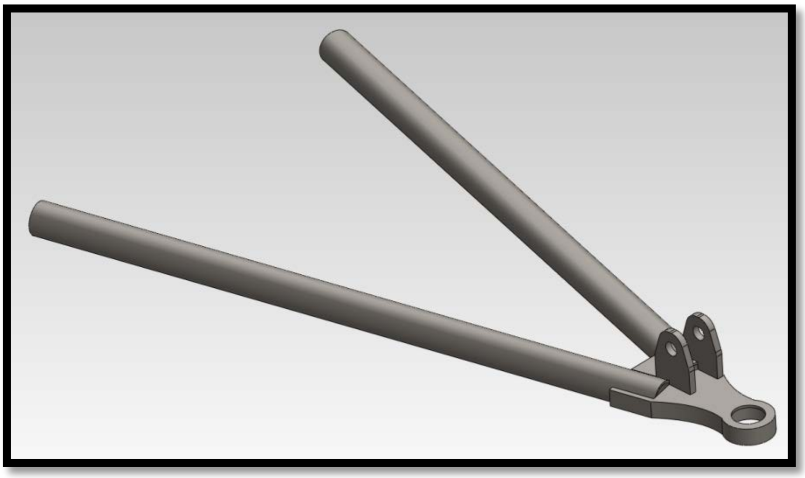


Figure 8.3: Front lower suspension arm concept.

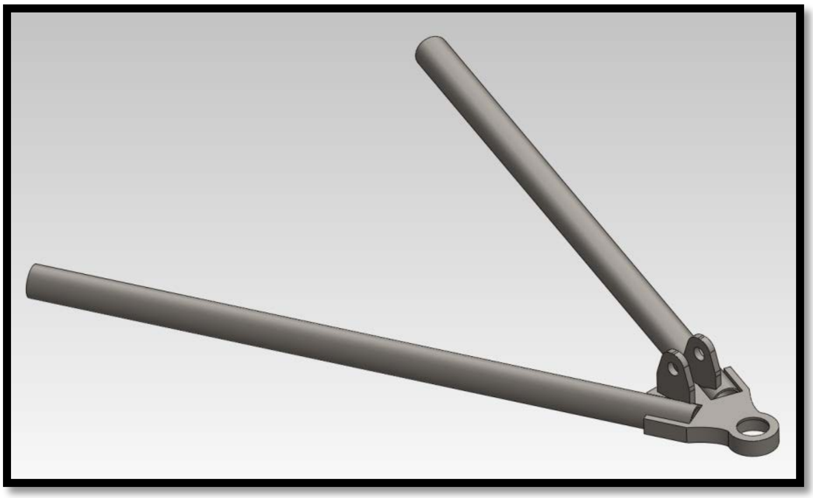


Figure 8.4: Rear lower suspension arm concept.

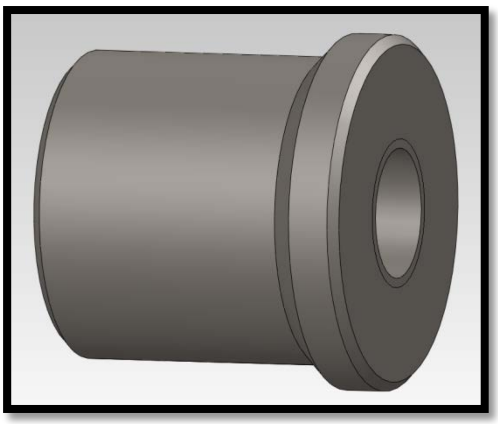


Figure 8.5: End plug for components featuring circular tubing.

8.5 Rockers

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

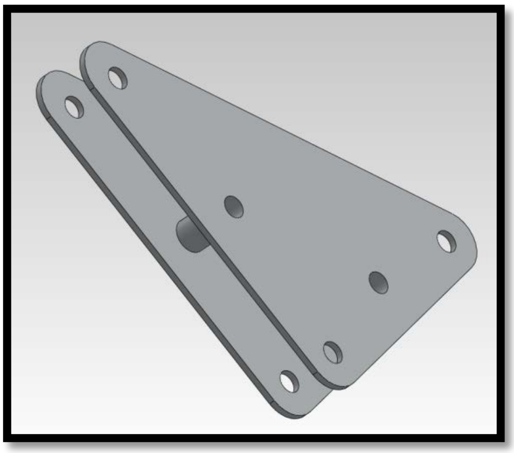


Figure 8.6: Front rocker concept.

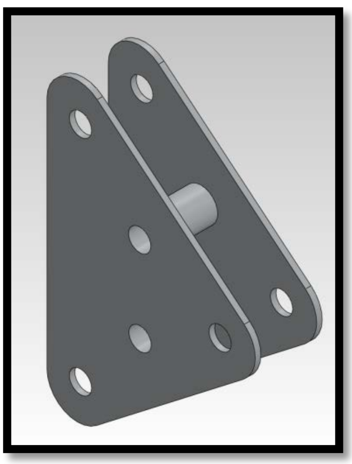


Figure 8.7: Rear rocker concept.

8.6 Steering Rack

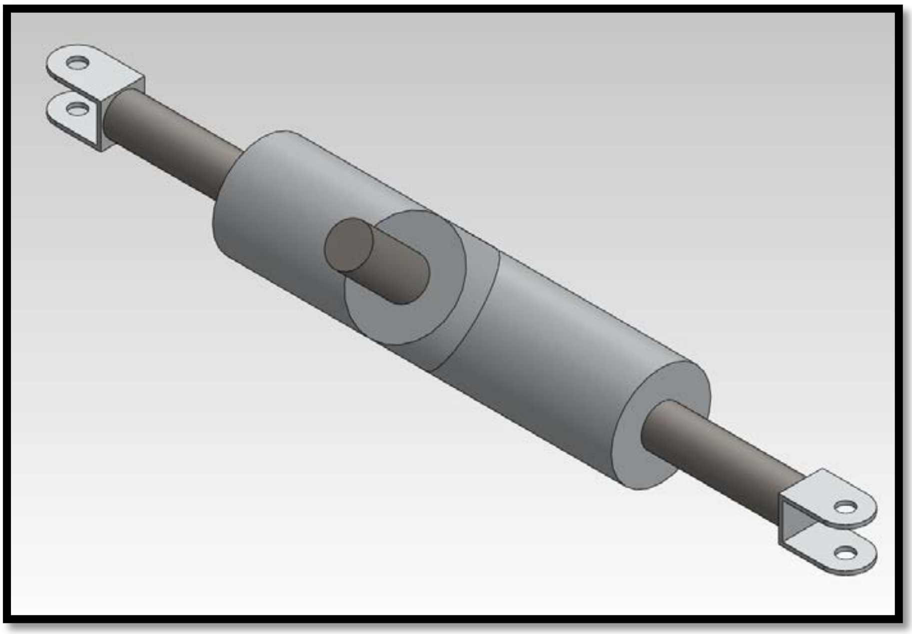


Figure 8.8: Steering rack representation.

Figure 8.8 above represents the relative length of the steering rack required for the design. It should be mentioned though, that all other dimensions of this design are purely a guideline to how the rack may look if it were designed completely.

If this rack and pinion were to be created, it is believed that the housing of the gears would be made from machined aluminium alloy, the rack and pinion gears would be bought complete if possible while the brackets to secure the tie rod ends would be made from either machined or bent aluminium alloy plate.

8.7 Bearings and Fasteners

The only bearings and fasteners considered in the design were the THK SB-12 spherical bearing and the Alinabal AM-5-GP rod end. As previously mentioned, the spherical bearings will be placed in the suspension arms to support the uprights. The rod ends on the other hand are to mate with the end plugs used on the suspension arms, tie rods, toe links and push rods. These parts have been carried over from the 2008 vehicles design. 3d models of these two components may be seen in the following two figures.

Obviously the suspension and steering system also requires a significant number of fasteners although these haven't been featured in the design concept. These would predominantly be nuts, bolts and washers or spacers.

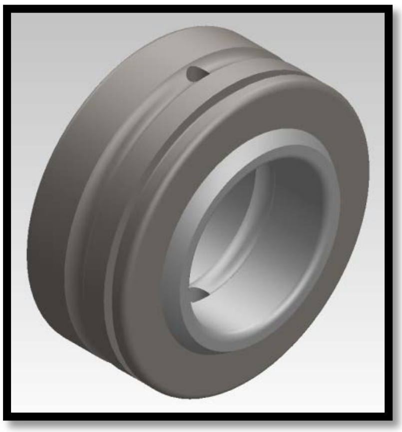


Figure 8.9: THK SB-12 bearing used in the suspension arms.

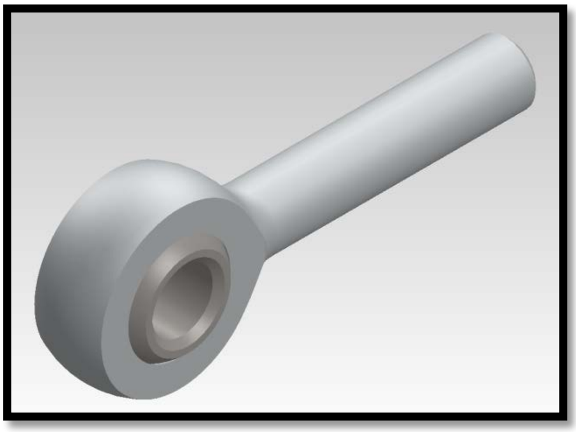


Figure 8.10: Alinabal AM-5-GP rod end.

8.8 Complete Suspension and Steering System

The following two figures define the assembly of the suspension and steering system for the front and rear ends of the vehicle. Along with the components discussed earlier in this chapter, the models also incorporate the chosen *Keizer* 4L 13" x 7" wheel plus a rough representation of the *Manitou* Swinger 4-way shock absorbers. The orientation of these parts has been determined by entering the *Wingeo3* model geometry into a 3d sketch in *SolidWorks* before designing and assembling the components using the reference points created by this sketch.

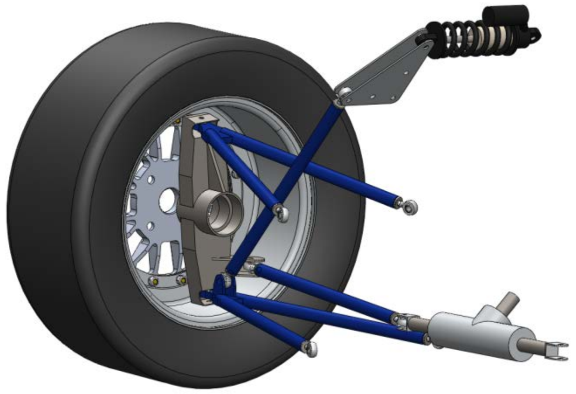


Figure 8.11: Assembly of front suspension and steering system in SolidWorks.

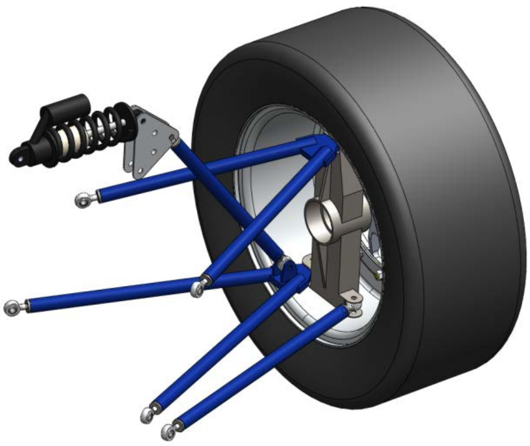


Figure 8.12: Assembly of rear suspension system SolidWorks.

Lastly, the figure shown over the page represents the layout of the whole car rendered in *SolidWorks*. This assembly summarises all work completed in the project and is believed to represent a design that is flexible and adaptable to future USQ FSAE vehicles while also providing optimal performance.

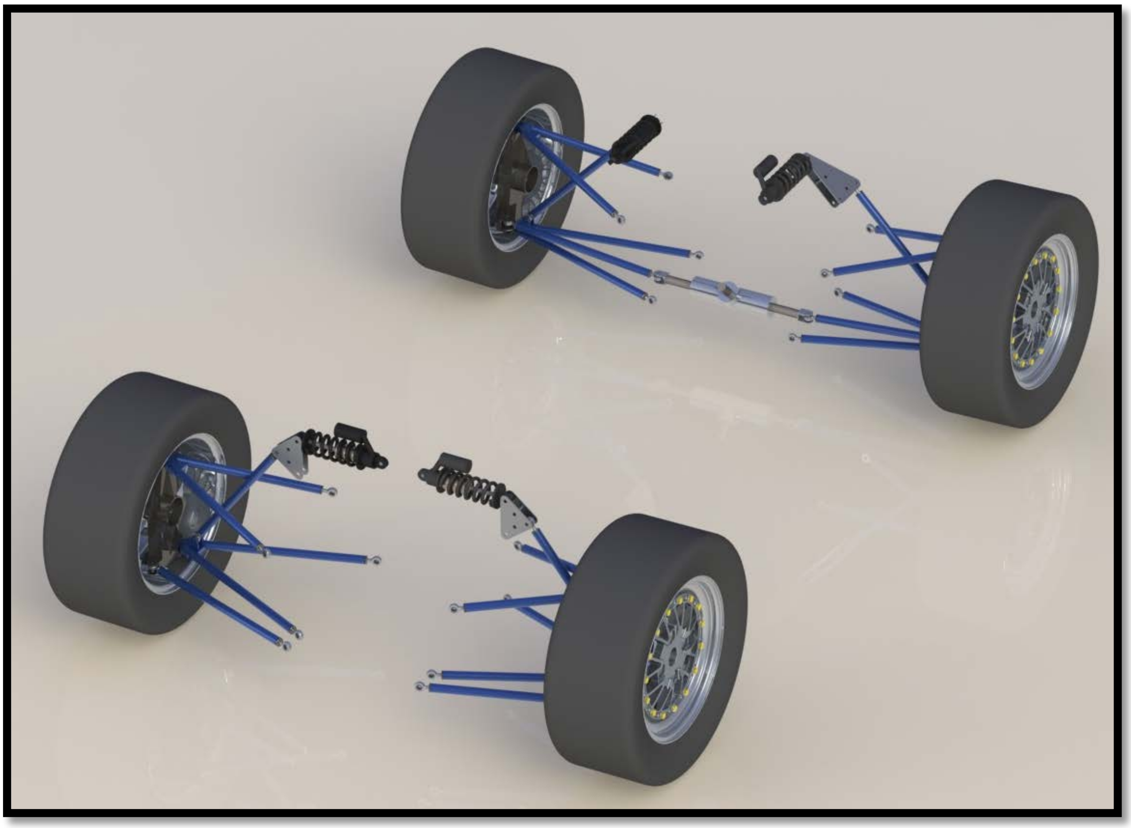


Figure 8.13: Render of full vehicle in SolidWorks.

8.9 Chapter Summary

This chapter has featured components modelled based off the design geometry that was refined in the earlier chapters of the dissertation. It also provides discussion on the system as a whole and provides graphical representation of all components assembled to form this system.

Due to a lack of time the components pictured in the chapter are only at a conceptual stage. The chosen designs are all very simple and are intended to be made from cheaper materials as opposed to more expensive, high performance materials such as carbon fibre, magnesium and titanium. These better performing materials may improve the overall performance of the design but the extra costs associated with their use did not warrant their application to a typical USQ FSAE vehicle.

Chapter 9

Design Evaluation

9.1 Chapter Overview

As with any engineering design work, it is imperative that the suspension and steering system be appropriately tested and evaluated before the design is physically built and put to use in order to verify its integrity and ability to withstand loads associated with its operation.

Because the design has not been produced, testing of the suspension and steering system cannot occur. However, an evaluation is still possible and this process is discussed in the following chapter. The evaluation process utilised involves assessing the design in regards to fulfilment of the design targets listed back in chapter 4.

9.2 Evaluation Criteria

It is believed the best method to evaluate the suspension and steering system would be to refer back to the ranked design targets set before any technical work began; obviously if the final solution fulfilled what was hoped for, than the design would be a success. The following sub sections consider each of these targets and gauge the new design's fulfilment of them. Each section or target, like the list in Chapter 4, is ranked in order of importance with the most valued design goals listed first to the least at the end of the chapter.

Although the steering and suspension systems were not physically produced, evaluation of the projects design was still applicable. However, thorough assessment of the car's dynamic performance was obviously unachievable as the design couldn't be physically tested. Limited

Wingeo3 testing is also another factor that has limited the evaluation process on the design's performance.

9.2.1 Improvement on 2008 car

This was the ultimate goal for the new design as failing to achieve it meant that all project work was in vain. Due to the inability to actually compare a physical car possessing the geometry and components developed in the project to the 2008 USQ vehicle, it was therefore impossible to draw a true conclusion to this assessment. However, based on the design guidance followed and the fact that no serious compromises have been made, it is believed that if the system was to be manufactured and assembled, that the new design would in fact yield better handling than the 2008 vehicle.

9.2.2 Drivability

Drivability of a vehicle incorporating the final suspension and steering designs is also another characteristic of the project's work that was hard to measure as this is a criteria that no computer modelling and testing can assess where the only source of data available is from driver feedback. To warrant that the final design is easy to drive and inspires confidence though, a number of measures have been taken throughout the design process. These include using a smaller scrub radius and kingpin inclination to improve the feel and ease of steering, employing a refined suspension geometry that offers optimal camber and roll centre control to make the car easier to drive harder and making its behaviour on the track more predictable, using better quality shock absorbers and simpler actuation mechanisms that operate in a single plane to provide enhanced suspension feel and operation, and lastly, creation of a steering system that has been designed with limited compromise that offers bettered ergonomics due to less lock to lock steering wheel rotation making it easier for the driver to get round the tightest turns on the track without getting cramped in the cockpit.

9.2.3 Adaptability

One of the major goals throughout the project was to make the design as adaptable and adjustable as possible in order to improve the possibility of integration with a future USQ FSAE vehicle. Measures to facilitate this design target have involved specifying a deep wheel which would allow future teams a large degree of flexibility in the suspension and steering component packaging and to also achieve a decent scrub radius and kingpin inclination, while lastly, allowing for adjustment of the steering rack and pinion even after the car was constructed. Apart from these component-specific allowances there have also been a couple of other procedures incorporated throughout the project to better the design's flexibility. These have included using properties from past USQ vehicles where needed in design calculations and most importantly, documenting the design process, decisions and solutions thoroughly in the dissertation in hope that future students will be able to better understand the suspension and steering design procedure but to also improve the ease with which they are able to interpret the findings of the project in order to apply them to a future vehicle.

Although these steps to promote adaptability have been made it is also believed that there is still a bit more room for improvement. The best way to achieve this would be to add adjustability to more of the components such as the suspension rockers to change their ratio as well as the steering arms so that the steering ratio and steering effort could be modified. Doing so would ensure that once the vehicle was physically assembled, further optimisation could occur based on driver feedback, enhancing the drivability of the car which as discussed in the preceding sub section is a very important attribute to a successful FSAE racer.

9.2.4 Cost

Not only will an economical design ensure that the system is more applicable to a future USQ team, it will also result in a better score at the FSAE competition's cost event. The following table summarises the costs associated with the parts designed for the suspension and steering system. It should be noted that where components are sourced rather than manufactured such as the wheels and shock absorbers, prices indicated do not include postage. On top of this, the costs listed for the components that are manufactured is only a representation of material price which does not include manufacturing costs of these components. Consequently, the total price of $2639.80 is an underestimate of the true costs involved in producing and assembling the design. This becomes obvious when comparing the cost of the project's design to the cost of the 2008 vehicle's suspension and steering systems. For the past vehicle the total coast of the suspension system was $2862.98, while the steering system and tyres and wheels were $696.67 and $2498.09 respectively, providing a total cost for the three of $6057.74.

The main reason that the cost of the new design is expected to supersede that of the 2008 vehicle's is due to the fewer compromises made on the off the shelf items such as the *Keizer* wheels and the custom made steering rack and pinion, where on the 2008 vehicle cheaper second hand components were utilised. Although this is not desirable, it is strongly supported that the cost sacrifices made are well worth the performance gained from this more optimal design.

Component	**Number Required**	**Source**	**Cost ($)**	**Total Cost ($)**
Front upright	2	-	4.40	8.80
Rear upright	2	-	3.85	7.70
Keizer 4L - 13” x 7” wheel	4	Keizer Wheels	240.30	961.20
Hoosier 20.5 x 7.0-13 R25B tyre	4	Hoosier	153.00	612.00
Front lower wishbone	2	-	1.90	3.80
Front upper wishbone	2	-	1.50	3.00
Rear lower wishbone	2	-	1.85	3.70
Rear upper wishbone	2	-	1.50	3.00
Front push rod	2	-	0.65	1.30
Rear push rod	2	-	0.50	1.00
Front rocker	2	-	1.75	3.00
Rear rocker	2	-	0.80	1.60
Manitou Swinger 4-Way 200 x 50 rear shock	4	Bikewagon	56.40	225.60
Manitou shock spring 200	4	Chain Reaction Cycles	36.10	144.40
Manitou shock bushes 8 x 20	4	Chain Reaction cycles	18.75	75.00
Front tie rod	2	-	0.60	1.20
Rear toe link	2	-	0.65	1.30
Steering rack and pinion	1	Estimation	250.00	250.00
Circular tubing end plug	16	-	0.10	1.60
Alinabal AM-5-GP rod end	16	SES Linear Bearings	7.50	120.00
THK SB-12 spherical bearing	8	SES Linear Bearings	16.95	135.60
Other hardware	1	Estimation	75.00	75.00
			Total:	**$2639.80**

Table 9.1: Cost summary for the suspension and steering systems.

With reference to this table, costs listed without a source have been calculated using a number of assumptions. Firstly, the price of the material was obtained by observing the price of commercially available structural members made from the specific material and then calculating the cost of this material per kg based on data provided by the suppliers. Secondly, to work out the cost of each suspension and steering component, the mass of each part was determined in *SolidWorks* by allocating materials inside the program and then multiplying this mass by the earlier determined cost per kg for that material.

Also, the cost listed for the 'other hardware' which includes fasteners and bearings not discussed in the previous component design chapter, is an educated guess at what these parts would cost. This was based off prices for this type of hardware in the 2008 vehicle cost report and typical prices found on the internet.

9.2.5 Grip

Once again this is another criterion that could not be fully evaluated without physical testing of the design. It was also very hard to draw a comparison with the 2008 car as the behaviour of this past vehicle is impossible to predict without a full *Wingeo3* model based off its geometry or being able to physically drive the car. However, it was hypothesized that the grip performance delivered by the new design would be decent as a result of the extensive iteration used to arrive at the final suspension geometry configuration. This is because the camber control provided by the final geometry design is quite decent and an optimal solution and as mentioned earlier in chapter 5, grip available from the tyres is strongly related to the camber placed on the wheels, so maintaining control of this camber is expected to enhance the grip available from the tyres.

9.2.6 Quick response handling

As has been reiterated so many times in the dissertation, the FSAE autocross tracks are usually very tight and technical and is why quick response handling is so important. Design elements that will affect the car's ability to manoeuvre in a quick manner include the roll centre height, sprung and unsprung weight and also the steering ratio. Although the design's quick response handling characteristics couldn't be truly tested as the design wasn't produced, it is supported that its handling response will be desirable as the suspension and steering systems feature above ground roll centre heights to reduce the roll moments which in effect reduces the time required for the car to stabilise as it enters a turn, as well as a low steering ratio meaning that the steering is 'quicker' or more responsive. However, because many of the components in the design are intended to be made from steel, the unsprung and potentially, sprung weights will be relatively high resulting in larger inertial forces that need to be overcome meaning that the time required for the springs and dampers to react to a change in the ground surface or chassis orientation will be increased.

9.2.7 Reliability

The reliability of the design is not assessable at all without a physical example although it is expected that the intended use of mild steel for the majority of components should lend itself well to reliability as steel possesses great strength, rigidity and fatigue resistance. It should also be realised that reliability of the vehicle relies on lot more aspects than just the consistency of the component materials and this is where a completed assembly of the vehicle would be required to do a full assessment.

9.2.8 Ease of Repair

Because the parts used in construction are made from mild steel or aluminium alloy (both weldable) and are relatively simple, the ease of repair for the new design is expected to be quite good. That way, if a component is damaged significantly at the competition it is able to be fixed on site with use of suitable welding equipment. On the other hand, if components were constructed from fibre composites or other non-weldable material, when a component broke the only way to get the car functioning again would be to have a ready supply of spares which for these types of materials is not cheap. Obviously, even though the suspension and steering components are easily repairable it would be wise to carry spares of each part although as steel and aluminium are relatively cheaper than some of the commonly used high performance materials in the FSAE competition, this would be significantly cheaper.

9.2.9 Simplicity

Design simplicity lends itself to many benefits in an FSAE vehicle. In summary, these benefits include simplified design procedures and calculations, better design packaging qualities and potentially easier maintenance and repair procedures as not only would the systems be easier to access, it is likely that as the parts are simple, the repair and maintenance procedures required to fix them would also be simple in nature. Overall it is believed that the project's design work encapsulates a significant amount of simplicity. This has been achieved by using square profiles in the uprights, a majority of circular profiles in the suspension arms, tie rods, toe links and push rods, and lastly, ensuring that the suspension actuation mechanisms only operate in one plane.

9.2.10 Weight

The final design target set in chapter 4 and thus final evaluation criteria is the weight of the design. Table 9.2 below lists the components considered in the project's design along with their masses. As can be seen the total weight of all considered components for the design sums to 49.8092kg. Although no data regarding the weight of the 2008 vehicle was able to be found it is estimated that the new design would be at a very similar mass as the design concepts and nominated and materials of these two designs are quite similar. It is also estimated that the mass of the new design is relatively heavy in comparison to some of the more competitive team's vehicles. While this is the case, the fact that the design's weight may be a little on the heavy side is not a major concern. Obviously the weight of the system is considered the least important design target and so if the final design does not offer light weight characteristics it is believed that the overall performance of the vehicle will not be affected dramatically.

Component	Number Required	Mass (kg)	Total Mass (kg)
Front upright	2	1.3649	2.7298
Rear upright	2	1.2038	2.4076
Keizer 4L - 13" x 7" wheel	4	3.2	12.8
Hoosier 20.5 x 7.0-13 R25B tyre	4	4.9	19.6
Front lower wishbone	2	0.5962	1.1924
Front upper wishbone	2	0.4645	0.929
Rear lower wishbone	2	0.5785	1.157
Rear upper wishbone	2	0.4656	0.9312
Front push rod	2	0.2009	0.4018
Rear push rod	2	0.1631	0.3262
Front rocker	2	0.1268	0.2536
Rear rocker	2	0.0592	0.1184
Manitou Swinger 4-Way 200 x 50 rear shock	4	0.886	3.544
Front tie rod	2	0.1833	0.3666
Rear toe link	2	0.1958	0.3916
Steering rack and pinion	1	1	1
Circular tubing end plug	16	0.0267	0.4272
Alinabal AM-5-GP rod end	16	0.0363	0.5808
THK SB-12 spherical bearing	8	0.019	0.152
Other hardware	1	0.5	0.5
		Total:	**49.8092 kg**

Table 9.2: Mass summary for the suspension and steering systems.

Once again a number of assumptions were made to fill out the values shown in the table. These included using masses for material defined by *SolidWorks*, taking the mass for the chosen wheel as average mass of all 4L model wheel sizes provided by *Keizer*, making an estimate of the rack and pinion mass, and lastly, using the mass of a used *Hoosier* R25A 20.5" x 7" tyre from a past USQ vehicle as a representation of the mass for a new *Hoosier* R25B 20.5" x 7" tyre. All other listed masses were gained from component specifications listed by suppliers and manufacturers.

9.3 Chapter Summary

The preceding chapter has provided a comprehensive assessment of the final suspension and steering system by evaluating the design using criteria defined by the performance targets established in chapter 4.

As discussed in the chapter, physical assessment was not possible as the design never materialised. However, based on the design decisions and procedure applied that was shaped by information derived from the literature review, it is believed that the design will show decent conformance with these physical performance related evaluation criteria. The final design was also found to conform well to the evaluation criteria that did not require physical testing.

Chapter 10

Conclusions and Future Work

10.1 Chapter Overview

Completing the design of the suspension and steering systems for a future USQ FSAE race car has led the author to a number of findings and conclusions. A summary of these conclusions is provided in the following chapter.

10.2 Conclusions

Firstly, a literature review uncovered information on fundamental concepts relating to the suspension and steering of a car, commonly used racing suspension and steering mechanisms, and lastly, some of the techniques and methods used to design these systems. On completion of the review it was determined that out of all these researched design methods, there would be no one that offered a complete guide applicable to the design of an FSAE vehicle and that a custom design plan containing segments from all reviewed methods would be much more appropriate. Following the literature review, an analysis of the 2011 FSAE competition rules provided a number of limits and further guidelines for the design.

The 2008 vehicle was attempted to be repaired of the damage suffered from its crash at the 2008 FSAE competition although due to a lack of support and time delays on the manufacture of replacement suspension and steering components, the vehicle was not completed on time nor to a safe standard. Therefore the analysis of the car could only be done statically in the lab. This analysis uncovered a large number of issues with the vehicle that all could have potentially contributed to the 2008 vehicle's crash. The biggest of these contributors was postulated to be the large scrub radius associated with the cheap wheels fitted on the vehicle as well as the suboptimal steering design.

Before the geometry design work commenced, founding decisions were made to provide a starting point for the future work. Based on information uncovered in the literature review and applicability to a typical USQ FSAE vehicle, it was nominated to use a double wishbone suspension configuration which actuated shock absorbers that were mounted inside the chassis by use of push rods, for both the front and rear suspension systems. It was also decided to select a new wheel with larger offset in order to improve the large scrub radius possessed by the 2008 vehicle. A revision of the 2008 vehicle's track widths and wheel base based on information discovered in the literature was also carried out in hope to improve vehicle handling capabilities.

Completion of the geometry design for the suspension and steering system found that iteration is key to arriving at an optimal solution as there are no straightforward equations or processes regarding the formulation of this geometry. As a result, the suspension and steering geometries arrived at provided a compromise between a number of performance characteristics. Additionally, it was also discovered that the design process required a large amount of assumption both in the type of analysis used to test geometries in the iteration procedure but also in the selection of the preliminary geometry parameters that would influence how the car handled. The geometry design process could potentially be quite complex and extensive although due to the obvious time constraints associated with designing a whole suspension and steering system in one year, the method used in the project was simplified. It is believed the largest influence on the geometry design was the need to accommodate a typical USQ FSAE vehicle.

Designing the components that made up the geometry to a manufacturable standard was not able to be completed in the project although this was not deemed as an issue because even if detail drawings were produced, it is highly likely that they'd need to be altered in order for the parts to integrate with a future USQ vehicle. The full compatibility of the geometry and component design was also not completed as an assembly able to be manipulated in order to simulate typical vehicle operation would be required and this wasn't available due to a lack of time.

Analysis of the design in relation to targets set before work commenced, although not able to be fully evaluated, indicated that the suspension and steering systems produced, would bring an improvement on the 2008 vehicle. To conduct a full evaluation of the design the components in the system would need to be manufactured and assembled on a vehicle so that physical testing could occur.

In summary, although the work completed on the project has not quite met what was initially desired, it is believed that results derived over the year represent a step in the right direction in regards to revising a suspension and steering system for a future USQ FSAE vehicle. The process whereby components are designed *for* rather than designed *around,* employed in the project is believed will eliminate or improve problems uncovered in the 2008 vehicle. It is also expected that future USQ students will be able to make significance of the work completed by either adapting the design to a future vehicle or by otherwise using the work as guide for a totally new design.

10.3 Future work

The potential future work can be split into two groups, work that can be completed in the short term and work requiring a significant amount of extra design work and thus requiring more time.

10.3.1 Short Term

If work was to continue after the submission of the project dissertation it would be desired to further the design, complete a more extensive analysis in Wingeo3 to test the geometry behavior, and lastly, to document the design process more thoroughly so that future students had a better guide to the suspension and steering design. More specifically, this would involve completing the anti-roll bar selection, furthering the front upright design to accommodate the brake calipers, designing and 3d modelling the steering rack and pinion as well as the steering shaft and associated components, selecting a steering wheel, designing and 3d modelling all chassis mounts required to attached the suspension and steering components, modelling the 2008 vehicle's geometry in Wingeo3 before comparing its behavior with the new design, making the Wingeo3 testing more comprehensive to better represent real life behaviour of a vehicle, refining the component 3d models and producing detail drawings for manufacture, and finally, documenting all this extra analysis as well as expanding on the earlier work done in this dissertation.

10.3.2 Long Term

Providing the short term future work was completed and a USQ vehicle compatible with all design was available, the suspension and steering systems could be produced and integrated with this car. This would also open up opportunities to conduct a more thorough design evaluation as physical testing could occur, allowing the design's performance to be gauged along with the true cost, weight and packaging qualities, ultimately providing an answer to whether the new design was an improvement of the 2008 vehicle.

Reference List

2005 Manitou Swinger 4-Way Rear Shock 200 x 50 2011, online product listing, Bikewagon Salt Lake City, UT USA, viewed 1 October 2011, <http://www.bikewagon.com/Forks-Headsets-Suspension/Rear-Shocks/Rear-Shocks/2005-Manitou-Swinger-p7764460-1-2.html >.

Apex 2010, Gran turismo

Bastow, D., Howard, G. & Whitehead, J. P. 2004. *Car suspension and handling,* London, Pentech.

Cadillac Owners 2009, Madison Ross Media Group, viewed 21 May 2011,<http://www.cadillacforums.com/forums/cadillac-catera-cimarron-forum/178885-sway-bar-bushings-upgrade-log.html>.

Chris Longhurst 2011, *The Suspension Bible*, viewed 20 May 2011, <http://www.carbibles.com/suspension_bible.html>.

Collegiate Formula SAE 2011, tyre specifications, Hoosier Racing Tire Corp. Lakeville, IN USA, viewed 22 September 2011, <http://www.hoosiertire.com/Fsaeinfo.htm>.

Engineers, S. O. A. 2004. *Proceedings of the 2004 Motorsports Engineering Conference and Exhibition,* Warrendale, PA, Society of Automotive Engineers.

Formula SAE 2010, *2011 Formula SAE Rules*, SAE International, United States of America

Fox Van RC Coil Rear Shock '12 2011, online product listing, Jenson USA Riverside, CA USA, viewed 1 October 2011, <http://www.jensonusa.com/store/product/RS259B02-Fox+Van+Rc+Coil+Rear+Shock+12.aspx >.

Gabriel de Paula Eduardo 2005, *Formula SAE Suspension Design*, paper series E, SAE International Brasil, viewed 2 July 2011, <http://www.theoryinpracticeengineering.com/resources/fsae/fsae%20suspension%20design%20brazil%20style.pdf>.

Hibbeler, R. C. 2007, *Engineering Mechanics – Statics*, Prentice Hall, Singapore.

Ira Crummey 2011, *Polar Moment of Inertia*, viewed 27 May 2011, <http://ironduke7.tripod.com/polarmoment.htm>.

Keith Calver 2001, *SUSPENSION - Terminology*, Mini Mania, viewed 23 May 2011, <http://www.minimania.com/web/SCatagory/SUSPENSION/DisplayType/Calver%27s%20Corner/DisplayID/1084/ArticleV.cfm>.

Manitou Shock Bushes/Hardware - 8mm - 14mm 2011, online product listing, Chain Reaction Cycles Ltd. Ballyclare UK, viewed 1 October 2011, <http://www.chainreactioncycles.com/Models.aspx?ModelID=57750>.

Manitou Shock Spring 2011, online product listing, Chain Reaction Cycles Ltd. Ballyclare UK, viewed 1 October 2011, <http://www.chainreactioncycles.com/Models.aspx?ModelID=15335>.

Milliken, W. F & Milliken, D. L., Olley, M. & Society of Automotive Engineers 2002. Chassis design: principles and analysis, Warrendale, PA, Society of Automotive Engineers.

Milliken, W.F. & Milliken, D. L. 1995, *Race Car vehicle dynamics*, SAE International, Warrendale, PA USA.

Olley, M., "Road Manners of the Modern Car." Proc. Inst. Auto. Engrs., Vol. XLI, 1946-1947

Pat Clarke 2005, *FormulaSAE-ANewsletter*, Volume 4, Issue 1, University of New South Wales Canberra, viewed 7 July 2011, <http://www.fsae.unsw.adfa.edu.au/newsletter/2009/other/fsae_03_2005.pdf>.

Pre comp testing 2009 2009, Royal Melbourne Institute of Technology, viewed 22 September 2011, < http://www.fsae.rmit.edu.au/rmit_racing_website/pre_comp_testing_2009.html>.

Senior Design Team MEM-03 AY0203 2002, *Drexel University Formula SAE: Design and Optimization of a Racecar Suspension*, sample project proposal document, Drexel University Philadelphia, viewed 2 July 2011, <http://www.mem.drexel.edu/seniordesign/Samples%20of%20Previous%20Project/Samples%20of%20SD%20Written%20Proposal/MMSD-Sample-Proposal-2.pdf>.

Staniforth, A. 1991, *Competition Car Suspension*, Haynes Publishing, Sparkford UK.

UWA rear bulkhead 2004, FSAE.com - James Waltman Bellingham WA, viewed 22 September 2011, < http://fsae.com/eve/forums/a/tpc/f/8356059423/m/52110721411>.

Woodward Steering 2010, *Configuring a rack for your race car*, technical catalogue, Woodward Machine Corp. Casper, WY USA, viewed 10 October 2011, <http://www.woodwardsteering.com/images/cat05%20pdf%2064-79.pdf>.

Appendix A

Project Specification

University of Southern Queensland Faculty of Engineering and Surveying

ENG 4111/2 Research Project

PROJECT SPECIFICATION

FOR:	**Jock Allen Farrington**
TOPIC:	**Redesign of an FSAE race car steering and suspension system**
SUPERVISOR:	Mr Chris Snook
SPONSORSHIP:	Faculty of Engineering and Surveying
PROJECT AIM:	To provide a compliant steering and suspension system for a competing race car in the 2012 Formula SAE-A competition
PROGRAMME:	**Issue B, 4 October 2011**

1. Research information on currently used automotive steering and suspension systems.
2. Research the existing rules and restrictions for Formula SAE-A race car steering and suspension design.
3. Critically evaluate existing alternatives for steering and suspension designs.
4. Critically evaluate researched methods of testing and adjusting the steering and suspension.
5. Repair University of Southern Queensland's 2008 Formula SAE-A race car before testing and analysing its design.
6. Develop preliminary design of the chosen steering and suspension systems.

As time and resources permit:

7. Manufacture and install prototype into Formula SAE-A racer and evaluate.
8. Test and obtain feedback from drivers and modify designs as needed.

AGREED:________________________(student) ________________________(supervisor)

(dated) ___/___/___

Appendix B

Literature Review Material

Allan Staniforth's Steering Rack Positioning Methods

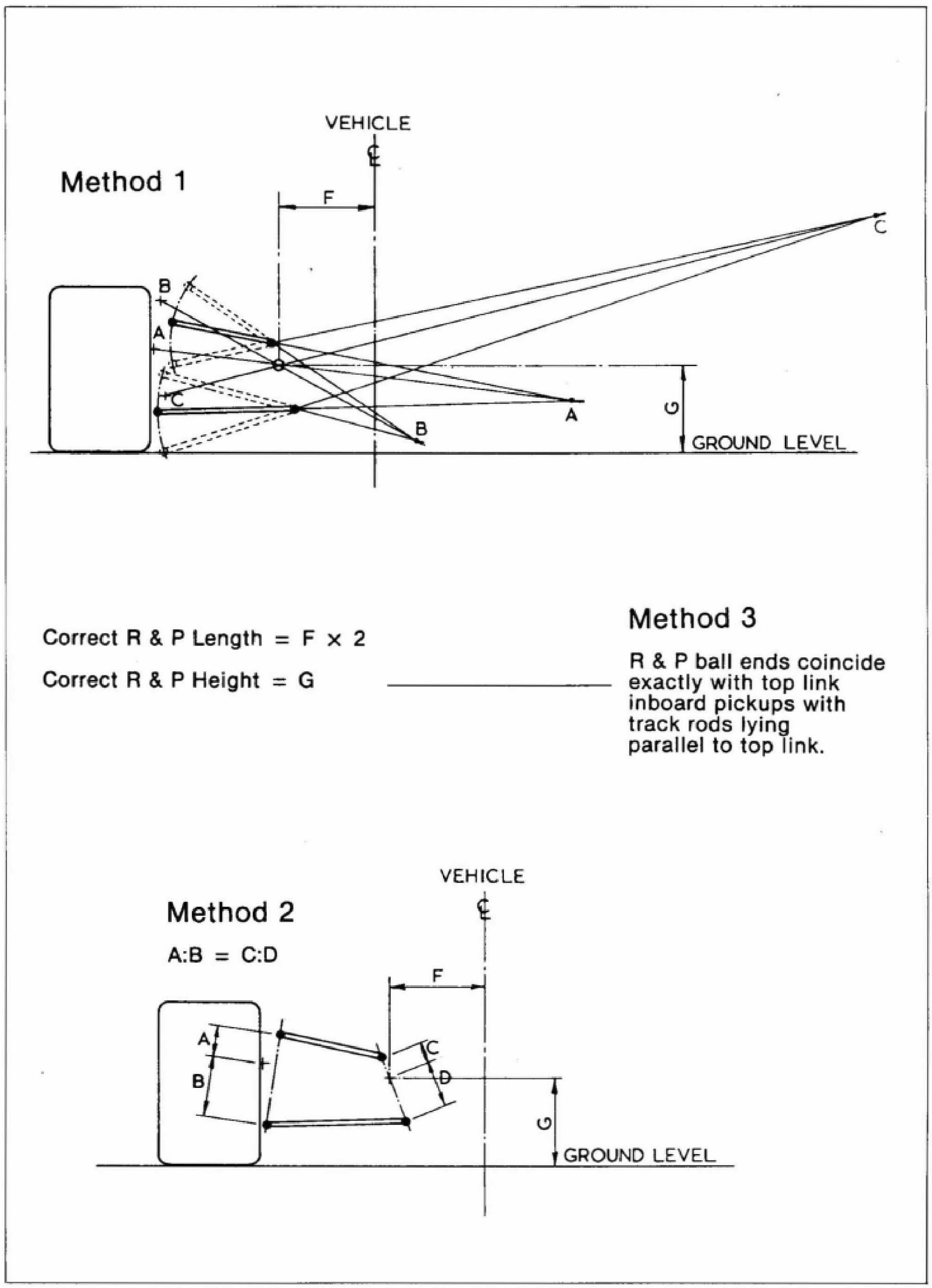


Figure B.1: Allan Staniforth's steering rack positioning methods. (Staniforth, 1991, p191)

Basic Design Checklist – Pat Clarke's Technical Introduction to Formula SAE

1. All load-paths should be direct and obvious to the judges. Judges love isosceles triangles and hate voids and indirect load-paths.

2. Never load a threaded rod end in bending. Apart from it being poor design, Judges hate this and hate seeing it again and again year after year.

3. Chassis stiffness should be such that the suspension can effectively work. If the suspension spring rate is such that the chassis flex becomes the de facto suspension, all your calculations go out the window, rapidly followed by handling and road holding.

4. Weight is bad! Remember the immortal words of the late Colin Chapman, "Add lightness and simplificate". (By the way, Mr Chapman also said "Any suspension will work if you don't let it"…..but Judges watch out for that!)

5. Cars with aggressive caster angles are self-adjusting with regard to corner weights. Therefore it is an absolute waste of time attempting to adjust corner weights unless the chassis is square, in proper alignment, on a flat and level surface and with tyre diameters equal front and rear.

6. Push rod or pull rod suspension is a good idea for the following reasons.

    - It is possible to adjust the ride height or chassis attitude without altering spring preload, and vice versa.
    - By using a rod and bellcrank operation of the suspension components, the motion ratio can be increased to permit more effective damper travel for minor wheel movements.
    - Unsprung weight may be decreased and the mass of the suspension components can be located to lower the CG.

7. Never forget it is 'Wheel rate' that is important, not 'Spring rate'. Work out a simple mathematical equation for the wheel/spring travel ratio to allow easy calculation of the

effects of spring or bellcrank ratio changes. Beware of bellcranks with aggressive multiplication ratios as these make the car very sensitive to minor adjustments.

8. Roll control devices (ARBs) are a good idea. If not needed they can always be disconnected, however, such devices are invaluable for fine-tuning the handling to suit track or weather conditions.

9. Ensure there is an adequate toe control base at the rear of the car, and that the components are stiff enough to prevent unwanted dynamic toe change. The judges will check for this using the old-fashioned 'Manual Labour' method.

10. Things will flex under load, therefore it is a good idea to use spherical bearings at both ends of all suspension units.

Pat Clarke, 2004

Appendix C

Past Vehicles

Figure C.1: Poor actuation of the shock absorbers on the 2008 vehicle as seen from the back of the car.

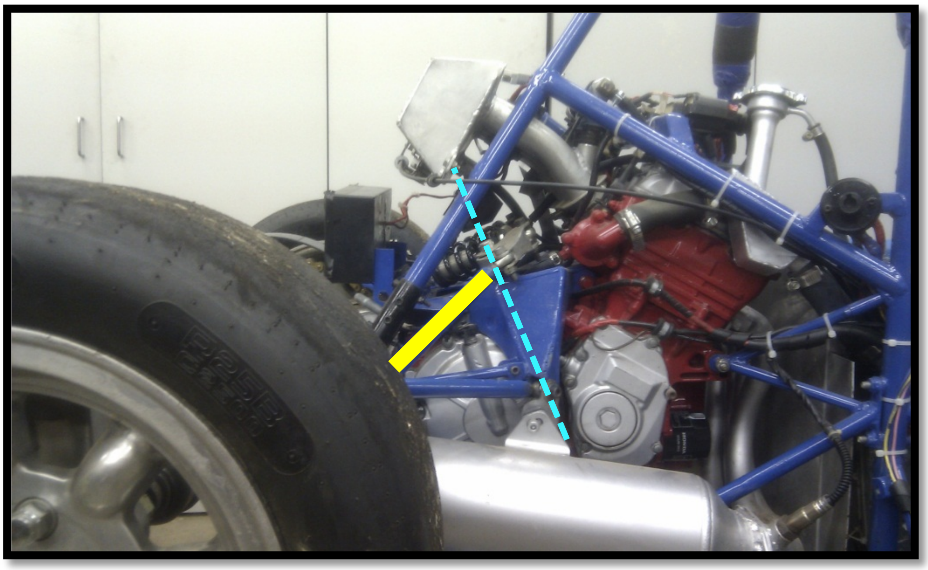


Figure C.2: Poor actuation of the shock absorbers on the 2008 vehicle as seen from the right side of the car.

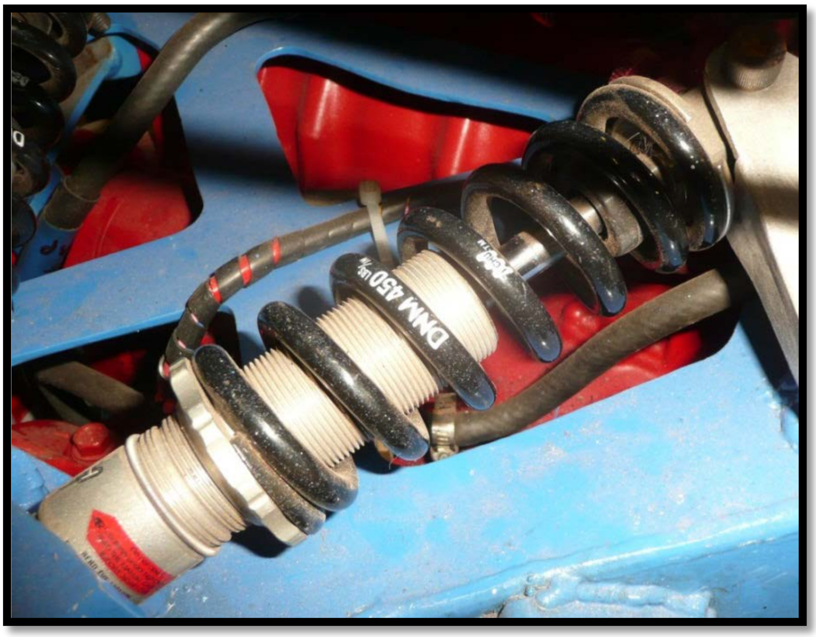


Figure C.3: *Rear shock absorber used in 2008 vehicle.*

Figure C.4: Front shock absorber used in 2008 vehicle.

Figure C.6: Front upper wishbone from 2008 vehicle.

Figure C.5: View down the cockpit on the 2008 vehicle illustrating the limited space.

FSAE-A Design Spec Sheet		**2008**
Competitors: Please replace the sample specification values in the table below with those appropriate for your vehicle and submit this to with your design report. This information will be reviewed by the design judges and may be referred to during the event.		
--Please do not modify format of this sheet. Common formatting will help keep the judges happy!		
--The sample values are fictional and may not represent appropriate design specs.		
Car No	13	
University	University of Southern Queensland	
Dimensions	**Front**	**Rear**
Overall Length, Width, Height	**2903mm, 1424mm, 1220mm**	
Wheelbase	**1670**	
Track	**1271 mm**	**1175mm**
Weight with 68 kg driver	**181.2kg**	**201.8kg**
Suspension Parameters	**Front**	**Rear**
Suspension Type	**Unequal length double wishbone. Pull rod actuated spring/damper unit**	**Unequal length double wishbone. Push rod actuated spring/damper unit.**
Tyre Size and Compound Type	**20x7-13 R25A Hoosier slick**	**20x7-13 R25A Hoosier slick**
Wheels	**Superlight, alloy 13" x 5.5"**	**Superlight, alloy 13" x 5.5"**
Design ride height (chassis to ground)	**55mm**	**55mm**
Center of Gravity Design Height	**307 mm above ground**	
Suspension design travel	**26 mm jounce/ 26 mm rebound**	**26 mm jounce/ 26 mm rebound**
Wheel rate (chassis to wheel center)	**28.7N/mm**	**28.7N/mm**
Roll rate (chassis to wheel center)	**1.41˚/g**	
Sprung mass natural frequency (in vertical direction)	**2.39Hz**	**2.34Hz**
Jounce Damping	**Adjustable**	**Adjustable**
Rebound Damping	**Adjustable**	**Fixed**
Motion ratio	**1.18**	**1.18**
Camber coefficient in bump (deg / in)	**0.219° / 10 mm bump**	**0.329° / 10 mm bump**
Camber coefficient in roll (deg / deg)	**0.773°/°**	**0.686°/°**
Static Toe and adjustment method	**0.5˚ toe out adj. by tie rods**	**0.25˚ mm toe in adj. by toe links**
Static camber and adjustment method	**-1.5° shim adjustable**	**-0.5° shim adjustable**

Front Caster and adjustment method	**2.5° shim adjustable**	
Front Kingpin Axis	**4.2° non-adjustable**	
Kingpin offset and trail	**12.5 mm offset and 1.2 mm trail**	
Static Akerman and adjustment method	**140% ackaman, 3˚ toe out at 3m radius turn, non adjustable**	
Anti dive / Anti Squat	**0%**	**0%**
Roll center position static	**19.27 mm below ground, on CL**	**12.65 mm below ground, on CL**
Roll center position at 1g lateral acc	**17.9 mm below ground, moves 156.6 mm toward inner wheel**	**12.13mm below ground, moves 94 mm toward inner wheel**
Steering System location, Gear ratio, Steer Arm Length	**Rack above legs, running to the front of the upright, 4.86:1, 67.55mm**	
Brake System / Hub & Axle	**Front**	**Rear**
Rotors	**Custom, mild steel**	**Custom, mild steel**
Master Cylinder	**5/8 Alloy**	**3/4 Alloy**
Calipers	**Wilwood Dynalite Single (twin piston, 1.38 dia inch)**	**Wilwood Dynalite Single (twin piston, 1.38 dia inch)**
Hub Bearings	**Tapered roller (Inner & Outer, 32007)**	**Tapered roller (Inner & Outer, 32007)**
Upright Assembly	**Student built fabricated aluminum (5083 & 6061), double sheer ball joints, bottom mount steering arm.**	**Student built fabricated aluminum (5083 & 6061), double sheer ball joints, push rod upright mounted**
Axle type, size, and material		**Spool, 48mm OD 2.2mm thk**
Ergonomics		
Driver Size Adjustments	**Interchangable seat liners. Fixed seat and steering wheel position,** manual pedal box adjustment ±50mm	
Seat (materials, padding)	**Intergrated fibreglass passanger cell shell, interchangeable foam seat liners.**	
Driver Visibility (angle of side view, mirrors?)	**120 degrees. No mirrors.**	
Shift Actuator (type, location)	**L/H manual shifter. Linkage opperated**	
Clutch Actuator (type, location)	**Shift intergrated automatic mechanical clutch actuation.**	
Instrumentation	**Tachometer (3 lights), shift light, water temperature light and neutral light**	
Frame		
Frame Construction	**Steel tube space frame**	
Material	**Cold drawn mild steel tube (350LO)**	
Joining method and material	**MIG and TIG welded**	
Targets (Torsional Stiffness or other)	**1350 Nm/deg torsional stiffness, 30kg with all brackets**	
Torsional stiffness and validation method	**1385 N.m/˚, to be physically validated**	

Bare frame weight with brackets and paint	**41 with intergral brackets**
Crush zone material	**Fibreglass 25mm honeycomb, CSM 450gsm, 2 layers polyester resin**
Crush zone length	**210mm**
Crush zone energy capacity	**7900J**
Powertrain	
Manufacture and Model	**1991 Honda F2, 600cc I-4, DOHC**
Bore / Stroke / Cylinders / Displacement	**67mm/ 45.2mm / 4 / 599cc**
Fuel Type	**98 RON**
Induction	**Atmospheric induction**
Throttle Body / Mechanism	**50mm, butterfly**
Max Power design RPM	**9500rpm**
Max Torque design RPM	**8000 rpm**
Min RPM for 80% max torque	**6000 rpm**
Effective Intake Runner Length	**205mm**
Effective Exhaust runner length	**Primaries 510mm.**
Exhaust header design	**4-1 equal length**
Fuel System (manf'r)	**Fuel injection, sequential (Adaptronic ECU)**
Fuel System Sensors (used in fuel mapping)	**MAP, MAT,TPS, Water Temp**.
Injector location	**Inlet runners, 139mm from back of inlet valve**
Intake Plenum volume	**2000cc**
Compression ratio	**12.0 :1**
Fuel Pressure	**2 Bar, minimum (Variable)**
Ignition System	**4 GM LS1 coil / igniter unit, 1 per cylinder**
Ignition Timing	**Digitally programmable by engine management system (Wasted Spark)**
Oiling System (wet/dry sump, mods)	**Factory wet sump**
Coolant System and Radiator location	**Twin radiators, Honda CBR250, 285x180x25mm. Side-pod mounted, elec (size, spec) fans triggered by ECU**
Fuel Tank Location, Type	**Floor mounted aluminum (1.6mm 5083-H32) tank between seat and firewall**
Muffler	**Modified CBR 1000 muffler**
Other significant engine modifications	
Drivetrain	
Drive Type	**Chain drive, 520 chain**
Differential Type	**Spool (live axle)**
Final Drive Ratio	**4.25:1**
Vehicle Speed @ max power (design) rpm	**(9500RPM)**
1st	**40km/h**
2nd	**56 km/h**
3rd	**75 km/h**
4th	**87 km/h**
5th	**101 km/h**
6th	**111km/h**

Half shaft size and material	**4140 steel 18mm dia, Q&T to 45HRC**
Joint type	**Tripod inner and outer, Custom inboard 4140 housing (Q& T to 53 HRC), standard Suzuki SS80v outboard**
Aerodynamics (if applicable)	
Front Wing (lift/drag coef., material, weight)	N/A
Rear Wing (lift/drag coef., material, weight)	N/A
Undertray (downforce/speed)	N/A
Wing mounting	N/A
Optional Information	
Rear Chassis and suspension	The rear suspenion has a unique layout to allow for fange mounted bearings and easily accesable sprockets and brakes. The frame can be "split" about the main hoop, alowing the motor and rear suspension to be removed as a unit, the rear suspension can then come away from the engine as a single unit.

Table C.1: *2008 USQ FSAE car design specification.*

Appendix D

Suspension Geometry

Upright Geometry Sketches

Front:

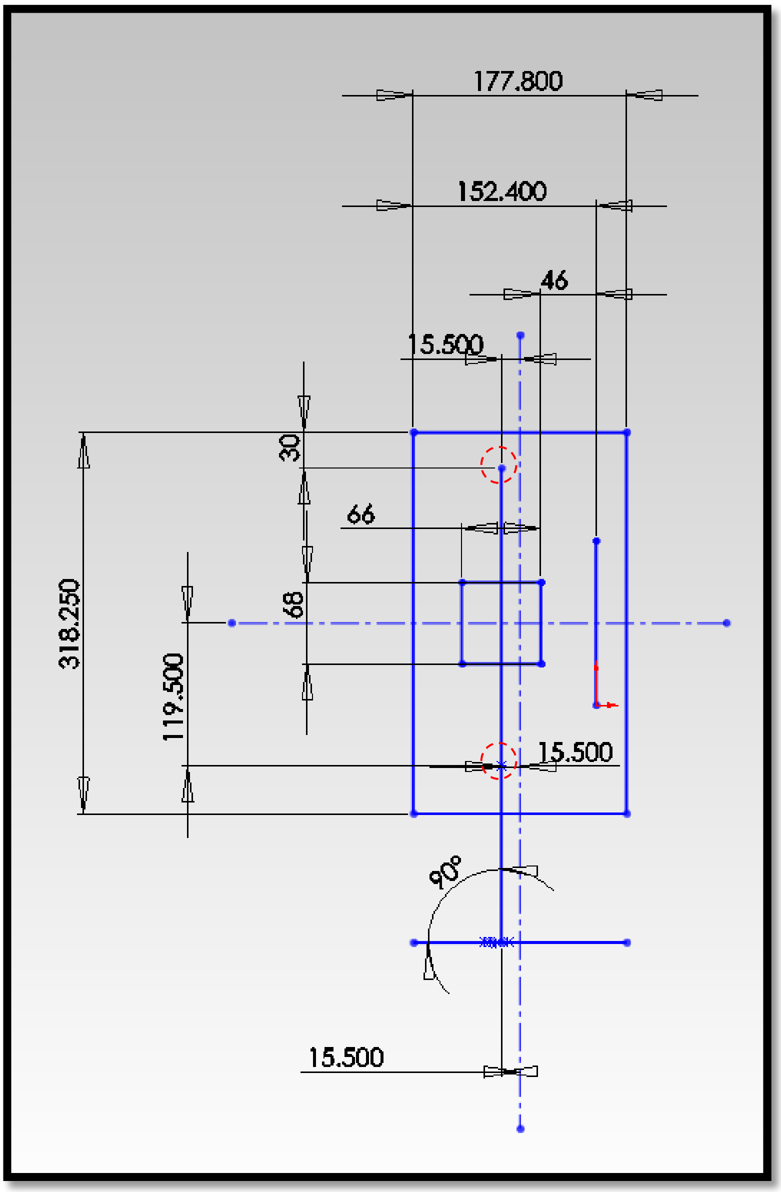


Figure D.1: SolidWorks sketch detailing the position of the rear upright pickup points as viewed from the rear of the vehicle.

Rear:

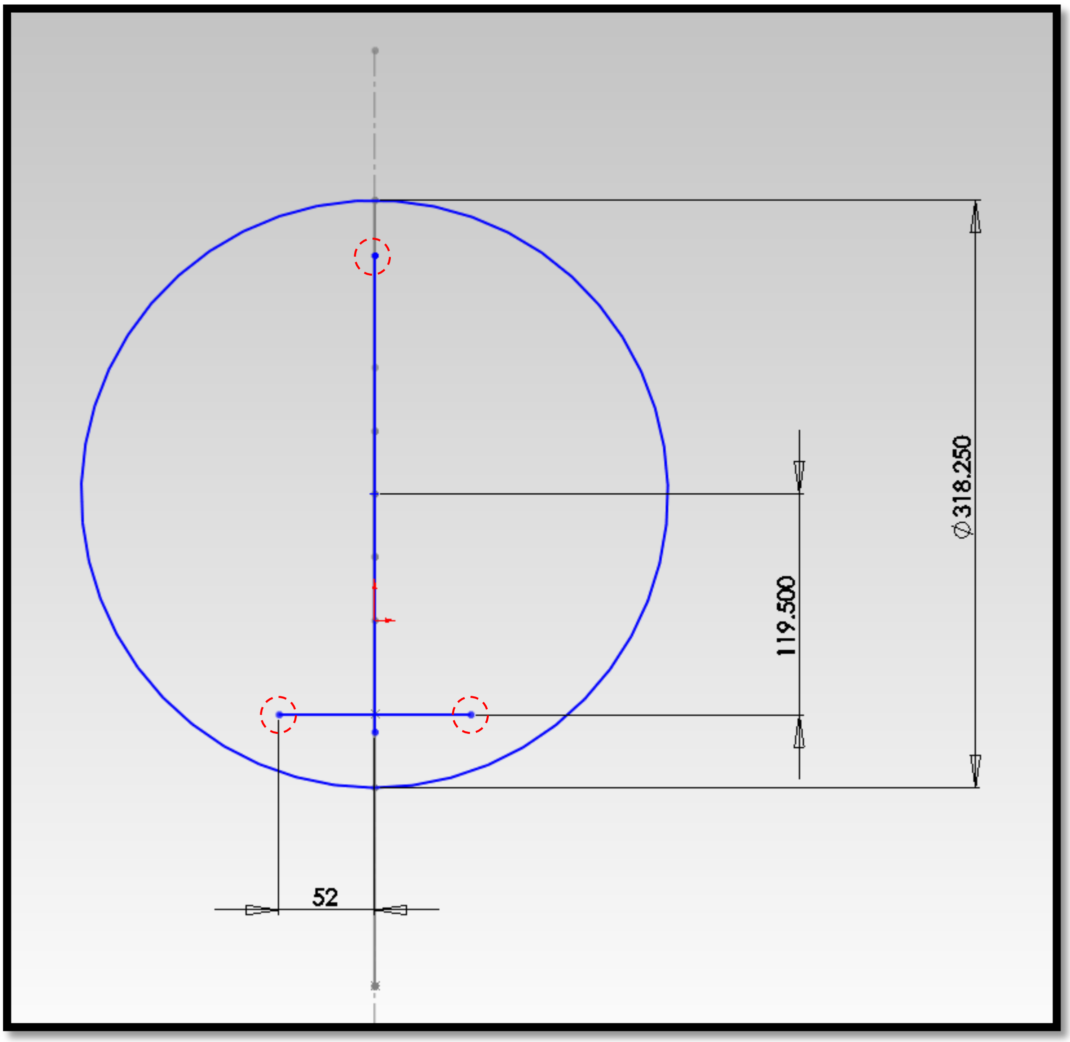


Figure D.2: SolidWorks sketch detailing the position of the rear upright pickup points as viewed from the right of the vehicle.

Wingeo3 Iteration Results (begins over page)

Iteration	RC static location (mm)	Camber in max droop (°)	Camber in max bump (°)	RC location in max droop (mm)	RC location in max bump (mm)	Camber in 3° roll (°)	Camber in -3° roll (°)	Camber curves	RC location in 3° roll (mm)	RC location in -3° roll (mm)	Displacement of RC (mm)	RC movement
1st: Equal and parallel arms	F: +138.984 R: +149.532	F: -1.095 R: -1.052	F: -0.889 R: -1.035	F: +81.648 R: +85.091	F: -81.453 R: -84.999	F: +2.073 R: +1.989	F: -4.066 R: -4.021	Cambers for the front and rear are almost identical throughout the roll of the chassis, extremities of camber are quite large.	F: -10217.9v, -311592h R: -27165.3v, -748316h	F: -10217.9v, +311592h R: -27165.3v, +748316h	F: -10256.884v, ±311592h R: -27314.832v, ±748316h	It is hard to draw any conclusion from the plots as because the suspension arms are parallel, there are a number of discontinuities in the curves. Roll centre moves around quite a lot.
2nd: Unequal and parallel arms	F: +138.984 R: +149.532	F: -1.500 R: -1.564	F: -1.283 R: -1.526	F: +45.584 R: +37.473	F: -44.792 R: -37.185	F: +1.892 R: +1.804	F: -4.252 R: -4.211	Cambers for the front and rear are almost identical throughout the roll of the chassis, extremities of camber are quite large.	F: -51.743v, +10799.6h R: -778.387v, +61552.3h	F: -51.743v, -10799.6h R: -778.387v, -61552.3h	F: -190.727v, ±10799.6h R: -927.919v, ±61451.1h	It is hard to draw any conclusion from the plots as because the suspension arms are parallel, there are a number of discontinuities in the curves. Roll centre moves around quite a lot although not as much as in the first iteration.
3rd:Unequal and non-parallel	F: +65.918 R: +80.852	F: +0.638 R: +1.176	F: -3.391 R: -4.215	F: +110.825 R: +118.462	F: +24.458 R: +48.923	F: +0.452 R: +0.140	F: -2.813 R: -2.556	Front camber curve is a little steeper than the rear. Camber extremities are not as large as in the previous iterations.	F: +66.353v, +48.047h R: +80.919v, +70.332h	F: +66.353v, -48.047h R: +80.919v, -70.332h	F: +0.435v, ±48.047h R: +0.067v, ±70.332h	Roll centres of the front and rear move around fairly consistently and not to a large extent
4th: Unequal and non-parallel - lower roll centres	F: +16.184 R: +26.710	F: -0.079 R: +0.604	F: -2.705 R: -3.685	F: +61.261 R: +64.287	F: -25.202 R: -4.819	F: +0.931 R: +0.483	F: -3.291 R: -2.898	Front camber curve is a little steeper than the rear. Camber extremities are not as large as in iteration 1 and 2 although are larger than the 3rd iteration.	F: +15.817v, +172.696h R: +24.802v, +196.601h	F: +15.817v, -172.696h R: +24.802v, -196.601h	F: -0.367v, ±172.696h R: -1.908v, ±196.601h	Roll centres of the front and rear move around fairly consistently and not to a large extent. The horizontal movement between the front and the rear is very similar (more than iteration 3) although the vertical movement differs a bit more than iteration 3.

Table D.1 (part 1): Results of first set of iteration in Wingeo3 program

Iteration	RC static location (mm)	Camber in max droop (°)	Camber in max bump (°)	RC location in max droop (mm)	RC location in max bump (mm)	Camber in 3° roll (°)	Camber in -3° roll (°)	Camber Curves	RC location in 3° roll (mm)	RC location in -3° roll (mm)	Displacement of RC (mm)	RC movement
5th: Unequal and non-parallel - roll centres a little below ground plane	F: -15.202 R: -11.079	F: -0.531 R: +0.209	F: -2.265 R: -2.311	F: +29.986 R: +26.602	F: -56.661 R: -42.687	F: +1.233 R: +0.721	F: -3.595 R: -3.136	Front camber curve is a little steeper than the rear. Camber extremities are not as large as in iteration 1 and 2 although are larger than the 3rd and 4th iteration.	F: -13.003v, -200.826h R: -1.544v, -539.842h	F: -13.003v, +200.826h R: -1.544v, +539.842h	F: +2.199v, ±200.826h R: +9.535v, ±539.842h	Roll centres of the front and rear move around more incosistently than all previous iterations and to a larger extent.
6th: Unequal and non-parallel - roll centres significantly below ground plane	F: -63.981 R: -55.601	F: -2.089 R: -1.260	F: -0.707 R: -1.829	F: -18.551 R: -17.697	F: -108.319 R: -91.329	F: +2.283 R: +1.627	F: -4.655 R: -4.033	Front camber curve is a little steeper than the rear. Camber extremities are very similar to those experienced in iteration 1 (quite significant).	F: -37.183v, -63.520h R:-54.094v, -85.108h	F: -37.183v, +63.520h R:-54.094v, +85.108h	F: +26.798v, ±63.520h R: +1.507v, ±85.108h	Roll centres of the front and rear move around fairly consistently and not to a large extent

Table D.1(part 2): Results of first set of iteration

Iteration	RC static location (mm)	Camber in max droop (°)	Camber in max bump (°)	RC location in max droop (mm)	RC location in max bump (mm)	Camber in 3° roll (°)	Camber in -3° roll (°)	Camber curves	RC location in 3° roll (mm)	RC location in -3° roll (mm)	Displacement of RC (mm)	RC Movement
A: Upper arms mounted as outwards as possible, lower arms placed in most inner possible locations, lower arms at the same height as iteration 6 from previous set of geometries	F: +15.100 R: +26.710	F: -0.079 R: +0.604	F: -2.674 R: -3.685	F: +62.421 R: +64.287	F: -28.543 R: -4.819	F: +0.949 R: +0.483	F: -3.294 R: -2.898	Front camber curve is a little steeper than the rear.	F:+15.366v, +122.034h R: +24.802v, +196.601h	F:+15.366v, -122.034h R: +24.802v, -196.601h	F: +0.266v, ±122.034h R: -1.908v, ±196.601h	Roll centres of the front and rear move around inconsistently and to a large extent
B: Same as iteration A although upper suspension pickup points are moved towards the center of the vehicle and lowered in order to achieve a similar roll centre to iteration A	F: +15.489 R: +26.041	F: +0.071 R: +0.789	F: -2.540 R: -3.438	F: +71.165 R: +75.715	F: -37.860 R: -20.172	F: +1.007 R: +0.580	F: -3.222 R: -2.833	Front camber curve is a little steeper than the rear. Extremities are very close to iteration A.	F:+15.405v, -125.031h R: +26.610v, +27.036h	F:+15.405v, +125.031h R: +26.610v, -27.036h	F: -0.084v, ±125.031h R: +0.569v, ±27.036h	Roll centres of the front and rear move around inconsistently (even more than iteration A). The extremities of movement are not as high as the previous iteration though.
C: Same as iteration B although upper suspension pickup points are moved towards the center of the vehicle even further and lowered in order to achieve a similar roll centre to iteration A	F: +15.778 R: +26.594	F: +0.182 R: +0.954	F: -2.441 R: -3.300	F: +77.643 R: +84.783	F: -44.728 R: -29.722	F: +1.051 R: +0.628	F: -3.169 R: -2.767	Front camber curve is a little steeper than the rear (less difference in the curves than the previous two iterations). Extremities are very close to iteration A and B.	F:+12.487v, -306.463h R: +26.131v, -96.390h	F:+12.487v, +306.463h R: +26.131v, +96.390h	F: -3.291v, ±306.463h R: -0.463v, ±96.390h	Roll centres of the front and rear move around inconsistently and to a large extent (larger than previous two iterations)

Table D.2 (part 1): Results of second set of iteration

Iteration	RC static location (mm)	Camber in max droop (°)	Camber in max bump (°)	RC location in max droop (mm)	RC location in max bump (mm)	Camber in 3° roll (°)	Camber in -3° roll (°)	Camber curves	RC location in 3° roll (mm)	RC location in -3° roll (mm)	Displacement of RC (mm)	RC Movement
D: Same as iteration C although pickup points are lowered while maintaining the same rc location as in previous iterations	F: +15.348 R: +26.396	F: +0.713 R: +2.174	F: -2.998 R: -4.558	F: +76.027 R: +82.422	F: -43.784 R: -27.212	F: +0.675 R: -0.145	F: -2.803 R: -2.018	Front camber curve is steeper than the rear (more difference in the curves than the previous iterations). Extremities are smaller than all other iterations.	F:+12.810v, -278.239h R: +26.454v, -64.114h	F:+12.810v, +278.239h R: +26.454v, +64.114h	F: -2.538v, ±278.239h R: +0.058v, ±64.114h	Roll centres of the front and rear move around inconsistently (not as much as C but more than A and B) and to a large extent (again not as much as C but more than A and B).
E: Same as iteration C although pickup points are raised while maintaining the same rc location as in previous iterations	F: +15.879 R: +26.592	F: -0.712 R: -0.284	F: -1.525 R: -2.043	F: +79.266 R: +86.502	F: -46.662 R: -32.151	F: +1.670 R: +1.398	F: -3.779 R: -3.524	Cambers for the front and rear are almost identical throughout the roll of the chassis, extremities of camber are quite large (larger than all other iterations).	F:+11.225v, -355.329h R: +25.517v, -124.480h	F:+11.225v, +355.329h R: +25.517v, +124.480h	F: -4.654v, ±355.329h R: -1.075v, ±124.480h	Roll centres of the front and rear move around fairly consistently and to a large extent (larger than all previous iterations).
F: Same as iteration C although roll centres are raised a bit while maintaining the same approximate ratio between the front and rear setups	F: +33.196 R: +57.079	F: +0.731 R: +1.957	F: -2.998 R: -4.303	F: +94.200 R: +113.889	F: -26.290 R: +2.560	F: +0.671 R: +0.004	F: -2.797 R: -2.158	Front camber curve is a little steeper than the rear (much like iterations A-C). The extremities of camber are much the same as iteration D.	F:+32.042v, -133.272h R: +57.174v, -32.760h	F:+32.042v, +133.272h R: +57.174v, +32.760h	F: -1.154v, ±133.272h R: +0.095v, ±32.760h	Roll centres of the front and rear move around fairly consistently (similar to iteration E) and to a small extent (second least movement out of previous iterations).
G: Combination of the raised roll centres from F with the lowered mounting points from D	F: +33.444 R: +57.531	F: +1.278 R: +3.196	F: -3.574 R: -5.558	F: +93.239 R: +112.337	F: -24.732 R: +5.118	F: +0.282 R: -0.774	F: -2.419 R: -1.401	Front camber curve is quite a bit steeper than the rear (more difference in the curves than the previous iterations). Extremities are smaller than all other iterations.	F:+32.664v, -115.762h R: +57.785v, -19.532h	F:+32.664v, +115.762h R: +57.785v, +19.532h	F: -0.78v, ±115.762h R: +0.254v, ±19.532h	Roll centres of the front and rear move around fairly consistently (similar to iteration F) and to a small extent (least movement out of previous iterations).

Table D.2 (part 2): Results of second set of iteration.

Iteration	RC static location (mm)	Camber in max droop (°)	Camber in max bump (°)	RC location in max droop (mm)	RC location in max bump (mm)	Camber in 3° roll (°)	Camber in -3° roll (°)	RC location in 3° roll (mm)	RC location in -3° roll (mm)	Displacement of RC (mm)
i: Refined geometry from iteration 2 only requiring caster, trail and camber alterations	F: +28.864 R: +43.927	F: +0.594 R: +1.524	F: -2.860 R: -3.872	F: +90.081 R: +101.322	F: -30.869 R: -11.310	F: +0.766 R: +0.273	F: -2.889 R: -2.421	F:+27.423v, -157.045h R: +43.888v, -49.454h	F:+27.423v, +157.045h R: +43.888v, +49.454h	F: -1.441v, ±157.045h R: -0.039v, ±49.454h
ii: Front trail reduced from -43mm to -15mm	F: +28.864 R: +43.927	F: +0.593 R: +1.524	F: -2.857 R: -3.872	F: +89.988 R: +101.322	F: -30.850 R: -11.310	F: +0.766 R: +0.273	F: -2.891 R: -2.421	F:+27.406v, -156.316h R: +43.888v, -49.454h	F:+27.406v, +156.316h R: +43.888v, +49.454h	F: -1.458v, ±156.316h R: -0.039v, ±49.454h
iii: Front caster increased from -6° to -10°, trail left the same as in iteration i	F: +26.013 R: +43.927	F: +0.434 R: +1.524	F: -2.709 R: -3.872	F: +87.164 R: +101.322	F: -33.655 R: -11.310	F: +0.870 R: +0.273	F: -2.997 R: -2.421	F:+24.411v, -173.276h R: +43.888v, -49.454h	F:+24.411v, +173.276h R: +43.888v, +49.454h	F: -1.602v, ±173.276h R: -0.039v, ±49.454h
iv: Cambers of front and rear wheels adjusted to try and meet evaluation criteria (rear at -0.5° and the front at -1.5°). Everything else as in i	F: +29.443 R: +43.272	F: +0.104 R: +2.011	F: -3.374 R: -3.352	F: +90.609 R: +100.779	F: -30.208 R: -12.116	F: +0.256 R: +0.784	F: -3.382 R: -1.929	F:+28.063v, -152.918h R: +43.206v, -51.359h	F:+28.063v, +152.918h R: +43.206v, +51.359h	F: -1.38v, ±152.918h R: -0.066v, ±51.359h

Table D.3: Results of third set of iteration

Appendix E

Shock Absorber Calculations

Ride Rate Analysis

Known:

$Gravity, g = 9.81\ m/s^2$

$1\ Newton = 0.2248$ Pounds

$Mass\ on\ front\ wheels = 181.2kg$

$Mass\ on\ rear\ wheels = 201.8kg$

$Nominated\ natural\ frequency = 145\ cycles\ per\ minute\ (CPM)$

$$Motion\ ratio = \frac{Wheel\ movement}{Spring\ movement}$$

Assumptions:

- Mass of the vehicle will be approximated as the same as the 2008 vehicle as a worst case measure
- Mass on the front and rear wheel sets will be used as the respective sprung weights for the front and rear of the vehicle
- Natural frequency is elected based off recommendation from Allan Staniforth (1991, p183

Coil rate will be defined by the following formulas taken from Allan Staniforth's book entitled 'Competition Car Suspension' (1991, p186)

$$Wheel\ Rate\ (lbs/in) = \left(\frac{Wheel\ Frequency\ (CPM)}{187.8}\right)^2 \times Sprung\ Weight\ (lbs)$$

$$Coil\ Rate\ (lbs/in) = Wheel\ Rate\ (lbs/in) \times Motion\ ratio^2$$

Front, 1:1 Motion Ratio

$$Wheel\ Rate\ = \left(\frac{145}{187.8}\right)^2 \times \left(\frac{181.2}{2}\right) \times 9.81 \times 0.2248$$

$$Wheel\ Rate = 119.107 \ldots lbs/in$$

$$Wheel\ Rate \cong 119\ lbs/in$$

$$Coil\ Rate = 119\ \times 1^2$$

$$Coil\ Rate\ = 119\ lbs/in$$

Rear, 1:1 Motion Ratio

$$Wheel\ Rate\ = \left(\frac{145}{187.8}\right)^2 \times \left(\frac{201.8}{2}\right) \times 9.81 \times 0.2248$$

$$Wheel\ Rate = 132.648 \ldots lbs/in$$

$$Wheel\ Rate \cong 133\ lbs/in$$

$$Coil\ Rate\ = 133\ \times 1^2$$

$$Coil\ Rate\ = 133\ lbs/in$$

Front, 1st Iteration:

$$Wheel\ Rate = \left(\frac{145}{187.8}\right)^2 \times \left(\frac{181.2}{2}\right) \times 9.81 \times 0.2248$$

$$Wheel\ Rate = 119.107 \ldots lbs/in$$

$$Wheel\ Rate \cong 119\ lbs/in$$

$$Coil\ Rate\ = 119\ \times 1.26^2$$

$$Coil\ Rate\ = 189\ lbs/in$$

Rear, 1st Iteration:

$$Wheel\ Rate = \left(\frac{145}{187.8}\right)^2 \times \left(\frac{201.8}{2}\right) \times 9.81 \times 0.2248$$

$$Wheel\ Rate = 132.648 \ldots lbs/in$$

$$Wheel\ Rate \cong 133\ lbs/in$$

$$Coil\ Rate = 133\ \times 1.13^2$$

$$Coil\ Rate = 170\ lbs/in$$

Sag Calculations

Known:

$Gravity, g = 9.81\ m/s^2$

$1\ Pound = 4.4482$ Newtons

$Mass\ on\ front\ wheels = 181.2kg$

$Mass\ on\ rear\ wheels = 201.8kg$

$Motion\ ratiofront = 1{:}1$

$Motion\ ratio\ rear = 1{:}1$

Assumptions:

- Mass of the vehicle will be approximated as the same as the 2008 vehicle as a worst case measure
- Mass on the front and rear wheel sets will be used as the respective sprung weights for the front and rear of the vehicle

The formula used to calculate the sag for each spring weight has been derived from R. C. Hibbler's text book entitled 'Engineering Mechanics – Statics' 11th Edition (2007, p84)

$$W\ (N) = k\ (N/m) \times s\ (m)$$

$$\frac{W\ (N)}{k\ (N/m)} = s\ (m)$$

Therefore, evaluating with respect to the design process noted in chapter 6:

Front, 1st Iteration:

$$\frac{\left(\frac{181.2}{2}\right)(kg/shock\ absorber) \times 9.81\ (m/s^2)}{\frac{(250 \times 4.4482) \times 1000}{1 \times 25.4}(N/m)} = s$$

$$s \cong 20.3mm$$

Rear, 1st Iteration:

$$\frac{\left(\frac{201.8}{2}\right)(kg/shock\ absorber) \times 9.81\ (m/s^2)}{\frac{(250 \times 4.4482) \times 1000}{1 \times 25.4}(N/m)} = s$$

$$s \cong 22.6mm$$

Required Motion Ratio Calculations

Assumptions:

- Motion ratio will be calculated in order to make the minimum rebound required, not the required jounce

Motion ratio is be calculated by dividing the minimum rebound allowed in the competition (25.4mm) by the previously derived sag travel.

Completing the calculations therefore yields,

Front, 1st iteration:

$$Front\ motion\ ratio = \frac{Minimum\ rebound\ wheel\ travel\ (mm)}{Front\ sag\ wheel\ travel\ (mm)}$$

$$Front\ motion\ ratio = \frac{25.4}{20.3}$$

$$Front\ motion\ ratio \cong 1.26$$

Rear, 1st iteration:

$$Rear\ motion\ ratio = \frac{Minimum\ rebound\ wheel\ travel\ (mm)}{Rear\ sag\ wheel\ travel\ (mm)}$$

$$Rear\ motion\ ratio = \frac{25.4}{22.6}$$

$$Rear\ motion\ ratio \cong 1.13$$

Rocker Calculations

Known:

$Angle\ of\ front\ push\ rod\ to\ the\ vertical, \theta_F = 45.94\ °$

$Angle\ of\ rear\ push\ rod\ to\ the\ vertical, \theta_R = 37.68\ °$

$Base\ length\ of\ front\ rocker = 80mm$

$Base\ length\ of\ rear\ rocker = 63mm$

$Required\ size\ of\ vertical\ rocker\ length\ front = L_F(m)$

$Required\ size\ of\ vertical\ rocker\ length\ rear = L_R(m)$

$Motion\ ratiofront = 1.26{:}\ 1$

$Motion\ ratio\ rear = 1.13{:}\ 1$

To find the required length of the vertical side of the rocker the following equations must be used:

$$Rocker\ multiplication\ factor = \frac{Ratio\ needed\ to\ overcome\ push\ rod\ incline}{Motion\ ratio}$$

$$Rocker\ multiplication\ factor = \frac{1}{\frac{\cos(Angle\ of\ push\ rod\ to\ vertical\ (°))}{1}} \times \frac{1}{Motion\ ratio}$$

$$Required\ size\ of\ vertical\ rocker\ length = Rocker\ multiplication\ factor \times Rocker\ base\ length$$

Applying these relationships to the front and rear of the car therefore yields:

Front:

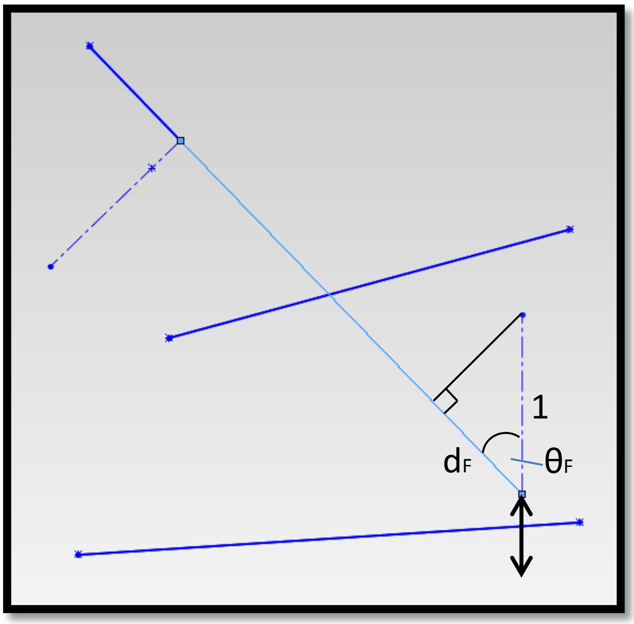


Figure E.1: Inclination of front push rod to the vertical.

$$Rocker\ multiplication\ factor = \frac{1}{\frac{\cos(\theta_F)}{1}} \times \frac{1}{Motion\ Ratio\ Front}$$

$$Rocker\ multiplication\ factor = \frac{1}{\frac{\cos(45.94)}{1}} \times \frac{1}{1.26}$$

$$Rocker\ multiplication\ factor = 1.141\ \ldots$$

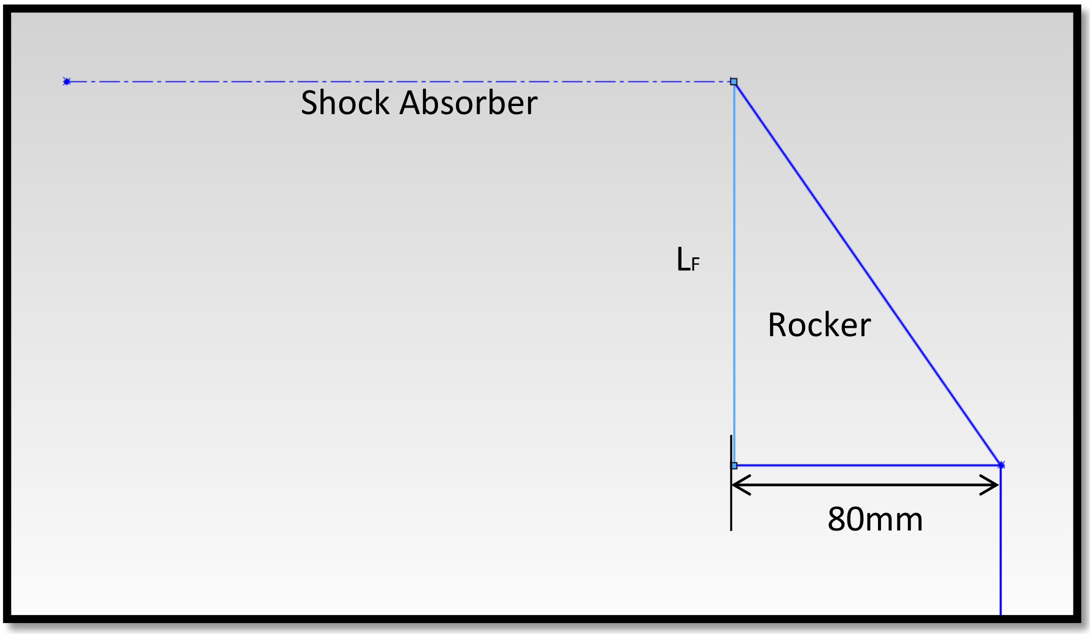


Figure E.2: *Determining the vertical length of the front rocker.*

With reference to the above shown figure,

$$L_F = Rocker\ multiplication\ factor \times Rocker\ base\ length$$

$$L_F = 1.141 \ldots \times 80$$

$$L_F = 91.301 \ldots mm$$

$$L_F \cong 91\ mm$$

Rear:

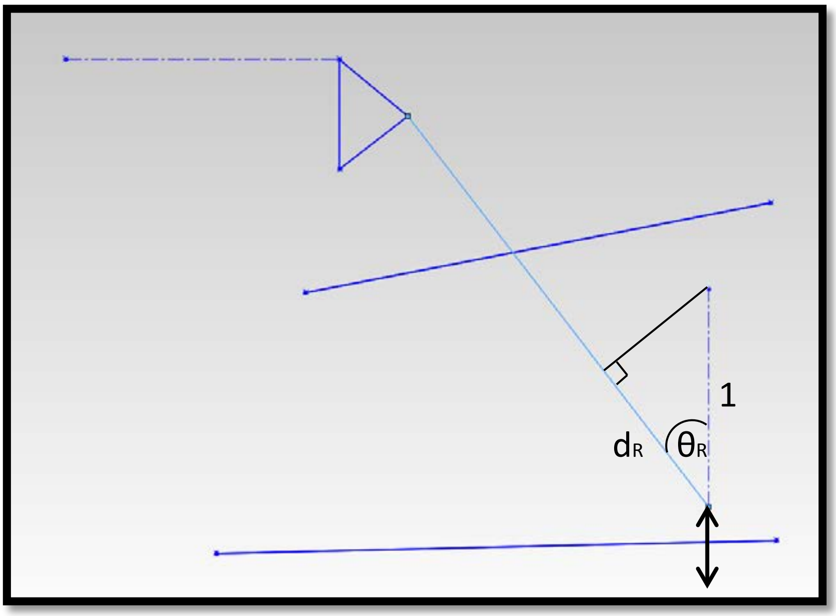


Figure E.3: Inclination of rear push rod to the vertical.

$$Rocker\ multiplication\ factor = \frac{1}{\frac{\cos(\theta_R)}{1}} \times \frac{1}{Motion\ Ratio\ Rear}$$

$$Rocker\ multiplication\ factor = \frac{1}{\frac{\cos(37.68\,)}{1}} \times \frac{1}{1.13}$$

$$Rocker\ multiplication\ factor = 1.118\ ...$$

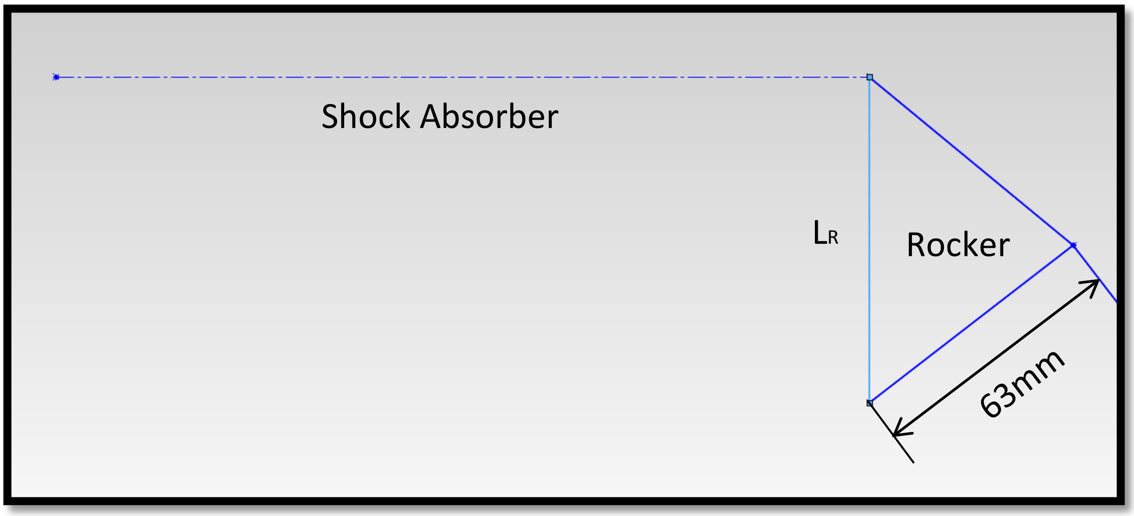


Figure E.4: Determining the vertical length of the rear rocker.

$$L_R = Rocker\ multiplication\ factor \times Rocker\ base\ length$$

$$L_R = 1.118 \ldots \times 63$$

$$L_R = 70.444 \ldots mm$$

$$L_R \cong 70\ mm$$

Frequencies for the Final Design

As the lowest stiffness available for the Manitou Swinger 4-Way shock absorber is 250 lbs/in this is the spring weight selected for the front and the rear. With these stiffness's the natural frequencies of the front and rear suspension set ups will be as follows:

$$Wheel\ Frequency\ (CPM) = 187.8\sqrt{\frac{Wheel\ Rate\ (lbs/in)}{Sprung\ Weight\ (lbs)}}$$

Where,

$$Wheel\ Rate\ (lbs/in) = \frac{Coil\ Rate\ (lbs/in)}{Motion\ ratio^2}$$

Using these equations in the designs achieves the following:

For the front:

$$Wheel\ Rate = \frac{250}{1.26^2}$$

$$Wheel\ Rate = 157.470 \ldots lbs/in$$

$$Wheel\ Frequency = 187.8\sqrt{\frac{157.470 \ldots}{\left(\frac{181.2}{2}\right) \times 9.81 \times 0.2248}}$$

$$Wheel\ Frequency = 166.724 \ldots CPM$$

$$Wheel\ Frequency \cong 167\ CPM$$

$$Wheel\ Frequency = 2.7833'\ Hz$$

For the rear:

$$Wheel\ Rate = \frac{250}{1.13^2}$$

$$Wheel\ Rate = 195.786 \ldots lbs/in$$

$$Wheel\ Frequency = 187.8\sqrt{\frac{195.786\ldots}{\left(\frac{201.8}{2}\right)\times 9.81\times 0.2248}}$$

$$Wheel\ Frequency\ = 176.160\ldots CPM$$

$$Wheel\ Frequency \cong 176\ CPM$$

$$Wheel\ Frequency = 2.933'\ Hz$$

Roll Rate Analysis

Known:

$Gravity, g = 9.81\ m/s^2$

$1\ Pound = 4.4482$ Newtons

$1\ Newton = 0.2248$ Pounds

$Mass\ on\ front\ wheels, m_F = 181.2kg$

$Mass\ on\ rear\ wheels, m_R = 201.8kg$

$Centre\ of\ gravity\ height\ front, h_F = 180mm\ (above\ ground\ plane)$

$Centre\ of\ gravity\ height\ rear, h_R\ = 300mm\ (above\ ground\ plane)$

$Roll\ centre\ height\ front, RC_F\ = 28.979mm\ (above\ ground\ plane)$

$Roll\ centre\ height\ rear, RC_R\ = 46.276mm\ (above\ ground\ plane)$

$Front\ spring\ roll\ rate = K_{\phi SF}\ lbs.ft/°$

$Rear\ spring\ roll\ rate = K_{\phi SR}\ lbs.ft/°$

$Front\ ride\ rate = K_{RF}\ lbs/in$

$Rear\ ride\ rate\ =\ K_{RR}\ lbs/in$

$Front\ ride\ freqency, \omega_F\ =\ K_{RR}\ lbs/in$

$Front\ track, T_F = 1300\ mm$

$Front\ track, T_R =\ 1175mm$

$Roll\ moment\ front = M_{1gF}\ N/m$

$Roll\ moment\ rear = M_{1gR}\ N/m$

Assumptions:

- Mass of the vehicle will be approximated as the same as the 2008 vehicle as a worst case measure
- Mass on the front and rear wheel sets will be used as the respective sprung weights for the front and rear of the vehicle
- Rear suspension is assumed as an independent system, not live axle
- The centre of gravity heights are purely an estimation based on suitable engineering judgement

The following analysis follows the 'simplified calculations approach' recommended by William and Douglas Milliken in their book entitled 'Race Car Vehicle Dynamics' (1995, p601-604)

$$K_{\phi SF} = \frac{K_{RF}{T_F}^2}{1375}$$

$$K_{\phi SR} = \frac{K_{RR}{T_R}^2}{1375}$$

$$M_{1gF} = (h_F - RC_F)m_F g$$

$$M_{1gR} = (h_R - RC_{FR})m_R g$$

$$Roll\ rate\ front = \frac{M_{1gF}}{K_{\phi SF}}$$

$$Roll\ rate\ rear = \frac{M_{1gR}}{K_{\phi SR}}$$

Evaluating these equations:

Front spring roll rate:

$$K_{\phi SF} = \frac{157.470 \ldots \times \left(\frac{1300}{25.4}\right)^2}{1375}$$

$$K_{\phi SF} = 299.996 \ldots lbs.ft/°$$

$$K_{\phi SF} = \frac{299.996 \ldots}{\left(\frac{1000}{12 \times 25.4}\right)} \times 4.4482\ N.m/°$$

$$K_{\phi SF} = 406.737 \ldots\ N.m/°$$

Rear spring roll rate:

$$K_{\phi SR} = \frac{195.786 \ldots \times \left(\frac{1175}{25.4}\right)^2}{1375}$$

$$K_{\phi SR} = 304.711 \ldots lbs.ft/°$$

$$K_{\phi SR} = \frac{304.711 \ldots}{\left(\frac{1000}{12 \times 25.4}\right)} \times 4.4482\ N.m/°$$

$$K_{\phi SR} = 413.131 \ldots N.m/°$$

Roll moment for the front at 1g lateral acceleration:

$$M_{1gF} = (0.18 - 0.028979) \times 181.2 \times 9.81$$

$$M_{1gF} = 268.450 \ldots N.m$$

Roll moment for the rear at 1g lateral acceleration:

$$M_{1gR} = (0.3 - 0.046276) \times 201.8 \times 9.81$$

$$M_{1gR} = 502.286 \ldots N.m$$

And so lastly,

$$Roll\ rate\ front = \frac{268.450\ \ldots}{406.737\ \ldots}$$

$$Roll\ rate\ front = 0.660\ \ldots{}^{\circ}/g$$

$$Roll\ rate\ front \cong 0.66\ {}^{\circ}/g$$

$$Roll\ rate\ rear = \frac{502.286\ \ldots}{413.131\ \ldots}$$

$$Roll\ rate\ rear = 1.215\ \ldots{}^{\circ}/g$$

$$Roll\ rate\ rear \cong 1.22\ {}^{\circ}/g$$

Appendix F

Steering Calculations

Rack Length

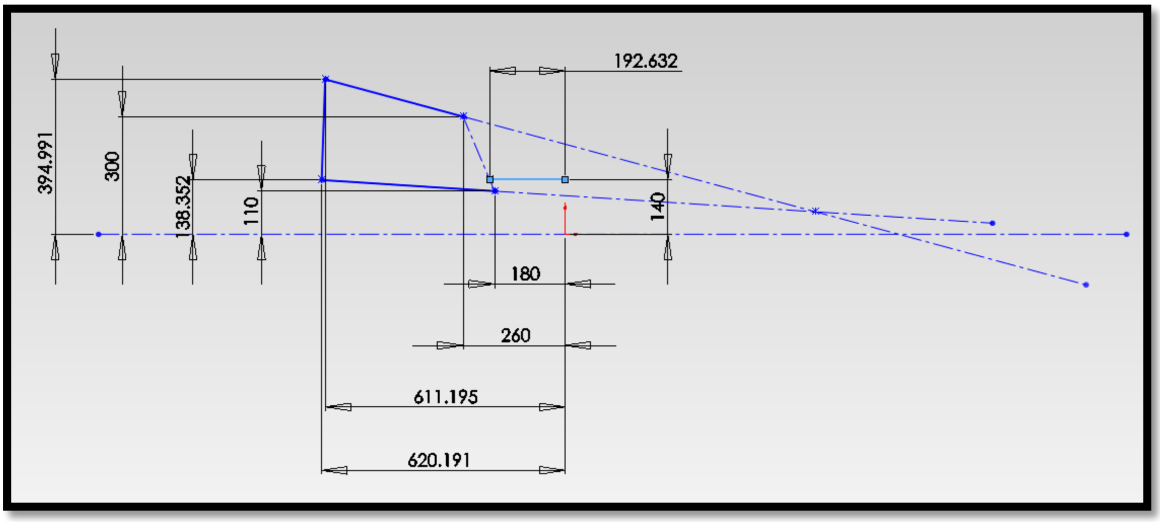


Figure F.1: Determining the rack length in SolidWorks – shown from the rear of the vehicle.

Geometry shown in the above figure was taken from the Wingeo3 suspension geometry designed in chapter 5.

Steering Arm Geometry

With reference to the results table shown over the page, data has been collected from Wingeo3 models testing different steering arm geometry variations. The highlighted green row in this table represents the chosen steering geometry.

Iteration	Max steering angle inside wheel (°)	Max steering angle outside wheel (°)	difference from ideal angle - inside wheel (°)	difference from ideal angle - outside wheel (°)	Wheel steering range (°)	Steering ratio	Rack travel at max steer (mm)
1st: 150% Ackermann, 94.334mm steering arm	31.347	20.985	-5.953	-4.215	26.166	5.16	32
2nd: 150% Ackermann, 70.751mm steering arm	32.972	21.912	-4.328	-3.288	27.442	4.92	25
3rd: 175% Ackermann, 75.128mm steering arm	29.317	19.013	-7.983	-6.187	24.165	5.59	22
4th: 175% Ackermann, 100.171mm steering arm	27.729	18.106	-9.571	-7.094	22.9175	5.89	28
5th: 100% Ackermann, 64.329mm steering arm	36.723	27.457	-0.577	2.257	32.09	4.21	30
6th: 125% Ackermann, 67.044mm steering arm	38.29	25.893	0.99	0.693	32.0915	4.21	29
7th: 140% Ackermann, 69.006mm steering arm	35.975	23.828	-1.325	-1.372	29.9015	4.51	27

Table F.1: Results of steering iteration process to determine steering arm length and Ackermann.

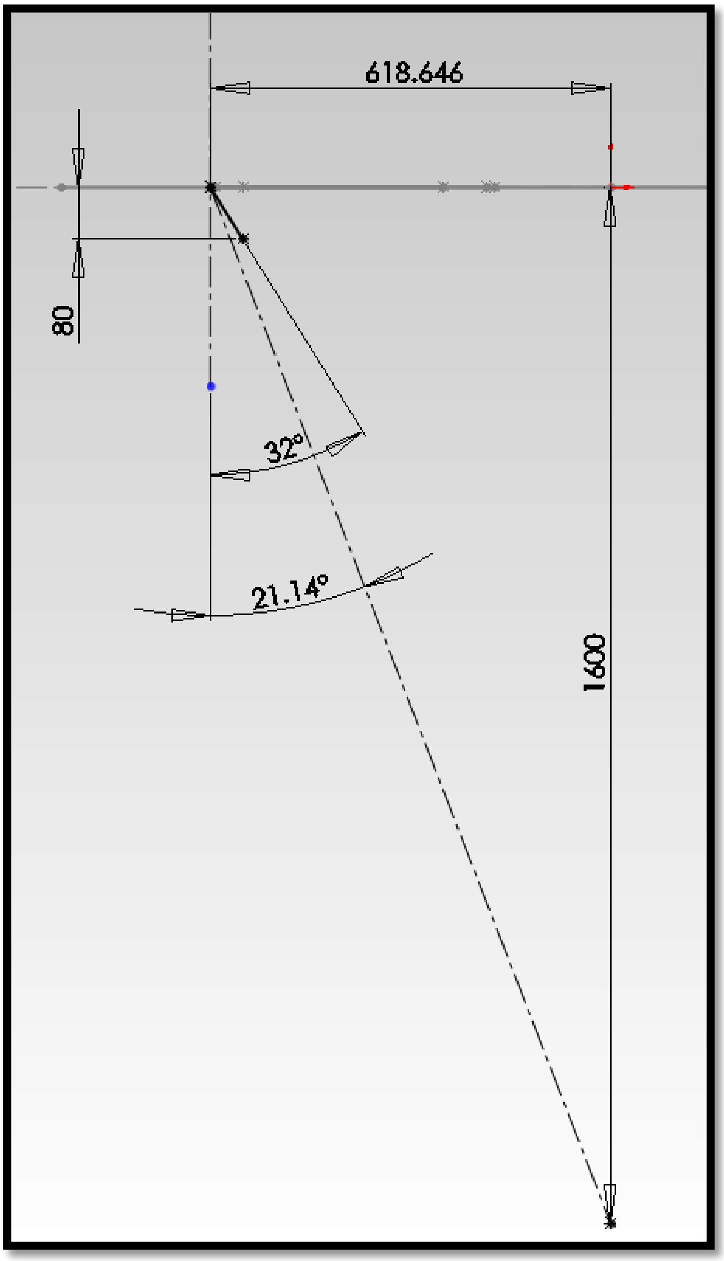


Figure F.2: Determining steering arm geometry in a SolidWorks sketch – viewed from above the vehicle.

The above figure denotes the first iteration's *SolidWorks* sketch showing the steering arm geometry in the horizontal plane. This sketch was the starting point for each iteration and defined the percentage of Ackermann employed along with the steering arm length. For the iteration shown above the Ackermann is 150% as the angle of the steering arm divided by the angle required for 100% Ackermann is equal to approximately 1.5. The steering arm length was determined by how far the steering arm connection point to the tie rod was behind the front axles. Throughout all iteration this distance was only varied between 80mm (as shown on the previous figure) and 60mm as these were believed to be realistic representations of the upper and lower limits that would work on the car.

Rack and Pinion C Factor

Known:

$$Rack\ travel\ at\ 135°\ steering\ wheel\ angle = 29mm$$

$$Rack\ travel\ at\ 1°\ steering\ wheel\ angle = \frac{29mm}{135°}$$

$$Rack\ travel\ at\ 360°\ steering\ wheel\ angle = \frac{29mm}{135°} \times 360 = 77.33'mm$$

As stated by W. and D. Milliken (1995, p718) the formula concerning the rack and pinion C factor are as follows

$$c - factor = \frac{Travel(mm)}{360\ (°\ pinion\ rotation)}$$

Therefore substituting the known values for the steering system:

$$c - factor = \frac{77.33'(mm)}{360(°\ pinion\ rotation)}$$