DESIGN OF SUSPENSION SYSTEM FOR FSAE VEHICLE

Submitted in partial fulfilment of the requirements of the degree of

Bachelor of Engineering

By

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Project Report Approval

This project report entitled **"DESIGN OF SUSPENSION SYSTEM FOR FSAE VEHICLE"**

By

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is approved for the degree of Bachelor of Mechanical Engineering.

Examiners

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Chairman

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Date:

Place:

Declaration

We declare that this written submission represents our ideas in our own words and where others' ideas or words have been included, we have adequately cited and referenced the original sources. We also declare that we have adhered to all principles of academic honesty and integrity and have not misrepresented or fabricated or falsified any idea/data/fact/source in our submission. We understand that any violation of the above will be cause for disciplinary action by the Institute and can also evoke penal action from the sources, which have thus not been properly cited, or from whom proper permission has not been taken when needed.

Anirudh Madhavan (ME801) ____________________

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Date: ______________

Project Report Certificate

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is certified for the submission for the degree of Bachelor of Mechanical Engineering.

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ABSTRACT

The chosen project aims to design a suspension system for HYPERION RACING’S 2014 formula student vehicle or FSAE vehicle representing Pillai Institute of Information Technology, New Panvel at Formula Student Germany to be held at Hockenheim in July 2014

The objective of the team is to provide most optimum and highly responsive suspension system in order to enable quality handling of the vehicle. The team studied and tuned vehicle behaviour at various track conditions taking an iterative approach towards designing the system considering driver ergonomics, kinematic parameters such as **centre of gravity, vehicle roll axis** etc. Suspension is the sole link of the vehicle to the road hence it is of high importance to design an optimum suspension system. We strive to achieve our objective keeping in mind future planning for our next vehicles.

Completion of the project has seen the design of geometry for the **suspension arms, suspension actuation mechanisms, uprights**. Additionally, concepts in the way of 3d models drafted in genuine SOLIDWORKS STUDENT EDITION 2014 software have been established for the suspension system also Optimum Kinematics was used to analyse the design in real time

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Abbreviation and Nomenclature

δ - Sag at static deflection

W - Weight of car

K - Spring stiffness

C - Dynamic capacity, Kgf

P - Equivalent load, Kgf

mr - million revolution

L10 - Life of bearing at 90% survival rate

Fr- Radial load, Kgf

Fa- Axial load, Kgf

X - Radial factor

Y- Thrust factor

Chapter 1

Introduction

1.1 Problem definition

- The steering and suspension systems are crucial to successful operation of any variety of cars. 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.

1.2 Objective

- The objective is to provide high performance suspension keeping in mind adjustability and adaptability such that future teams incorporate or modify the design without any hassle.

1.3 An outline of specific project tasks is as follows

- Research information on currently used automotive suspension systems.
- Research the existing rules and restrictions for Formula SAE race car steering and suspension design.
- Critically evaluate existing alternatives for suspension designs.
- Critically evaluate researched methods of testing and adjusting the suspension.
- Develop preliminary design of the chosen suspension systems.

As time and resources permit:

- Manufacture and install prototype into FSG car and evaluate.
- Test and obtain feedback from drivers and modify designs as needed.

1.4 Overview of 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.
- Expanding on this, the competition occurs annually on 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.

- 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, and competition market. Other factors that teams will potentially consider are also the aesthetics, ergonomics and manufacturability.

1.5 Vehicle requirements:

1) 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.

2) Suspension requirements

As quoted from the 2014 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 notlimited 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.
- 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.

1.6 Suspension System Definition

- The following figure provides assemblies of the front and rear suspension systems for an FSAE race car. The key components of these systems are numbered and listed below. Throughout the synopsis 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 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 bell crank)
8. Push rod
9. Suspension arm (or suspension linkage/ wishbone)
10. Upright
11. Toe link

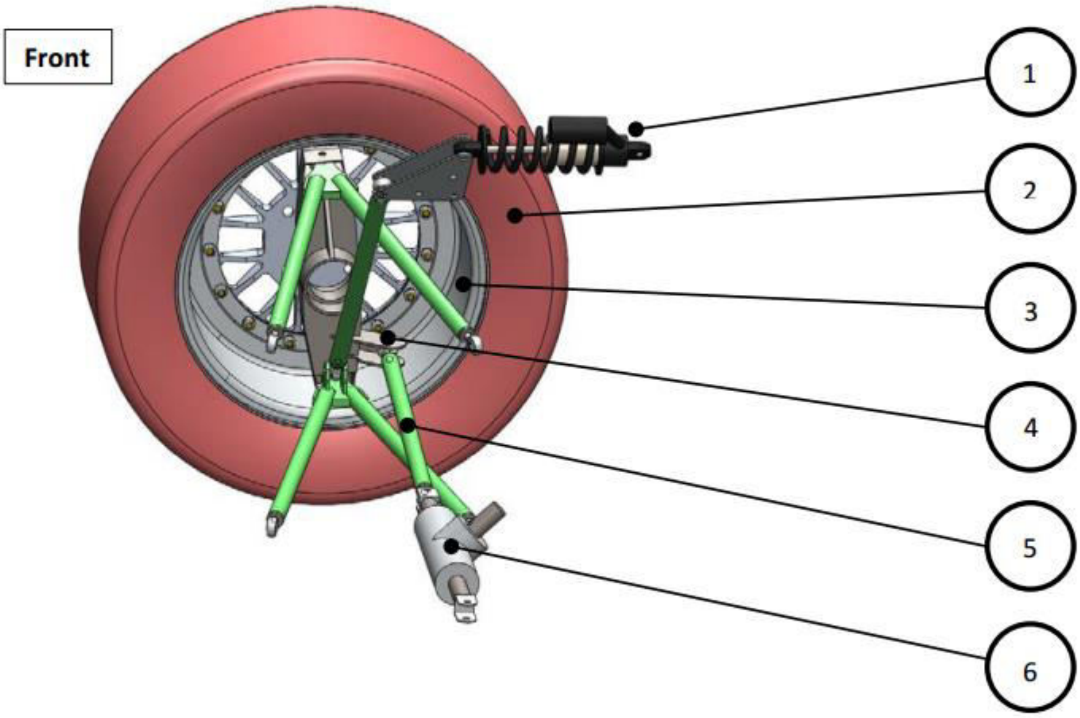


Fig1: Front Wheel Assembly

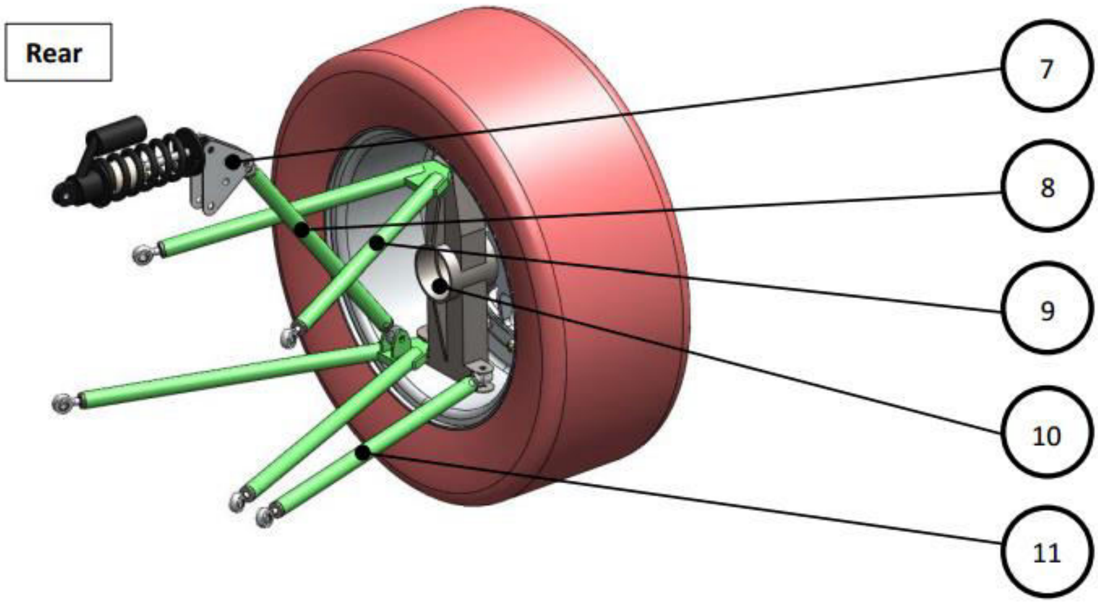


Fig 2: Rear wheel assembly

Chapter 2

Literature review

2.1 Chapter overview

- This chapter intends to educate any layman about suspension design of FSAE vehicle.The review analysed three areas related to the suspension system and was aimed at forming a solid knowledge and skill basis for design. It consists of 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

1) **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)

2) **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.

3) 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)

4) Polar Moment of Inertia

- The polar moment of inertia is based from Newton's 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, p3)

5) 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)

6) 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** 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)

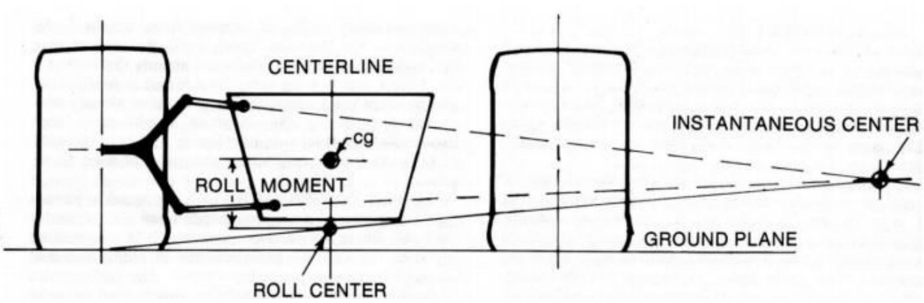


Fig 3: Determination of roll centre and moment arm (Caroll Smith, pg30)

7) 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)

8) Roll Moment

- Also visualised on figure, 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 (Smith, 1978, p30)

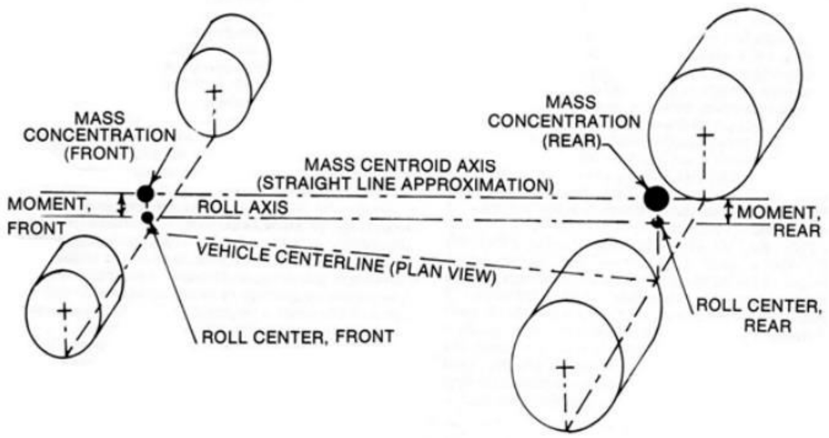


Fig 4: Relationship between roll axis, mass centroid axis and roll moments.

9) 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."

10) 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:

$$longitudinalloadtransfer(N) = \frac{acceleration(g) \times weight(N) \times CGheight(mm)}{wheelbase(mm)}$$

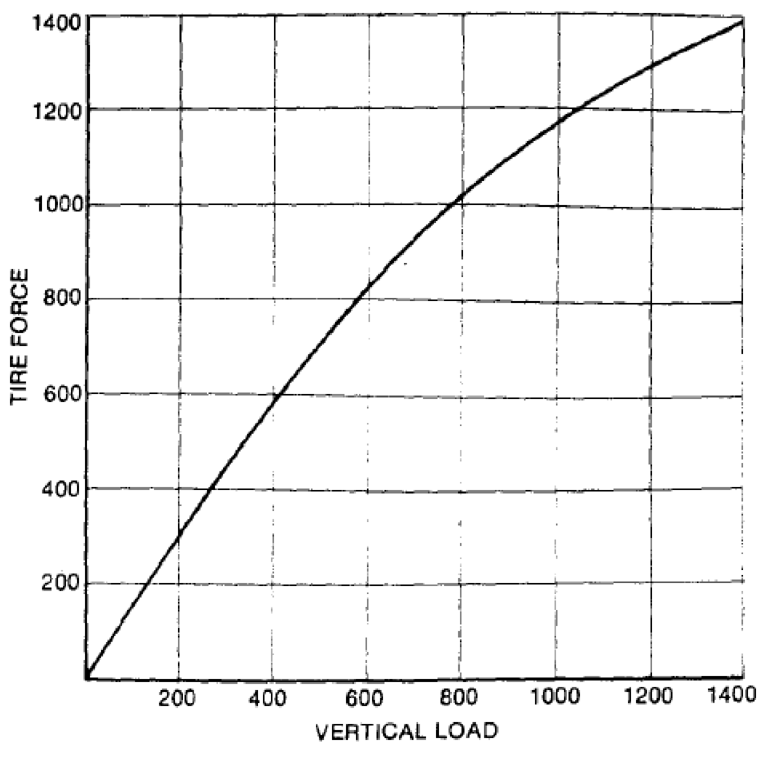


Fig 5

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.

11) 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.

12) 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)

$$lateral\ load\ transfer(N) = \frac{lateralacceleration(g) \times weight(N) \times CGheight(mm)}{trackwidth(mm)}$$

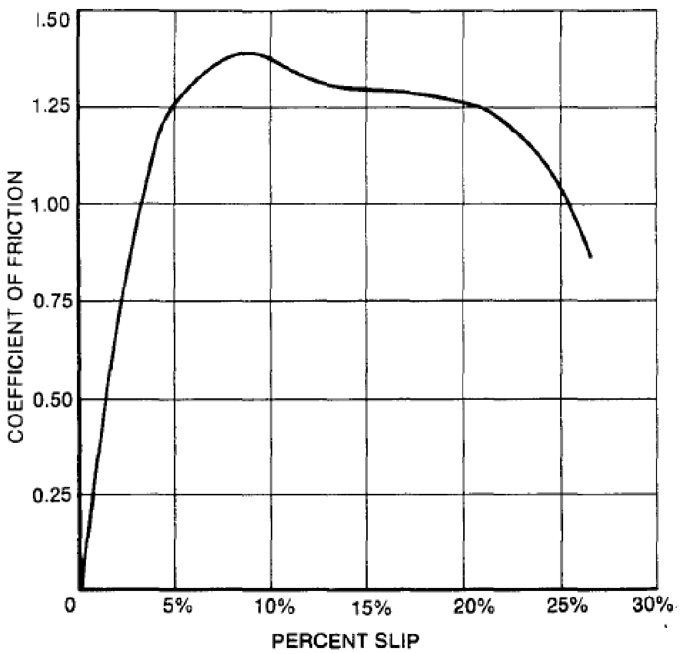


Fig 6: Tyre behaviour

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

13) 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 he 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.

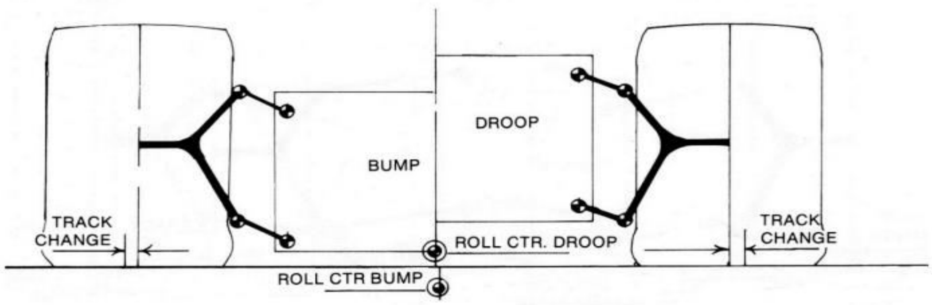


Fig 7: Bump and Droop behaviour of double wishbone set up. (Smith, 1978, p51)

14) 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)

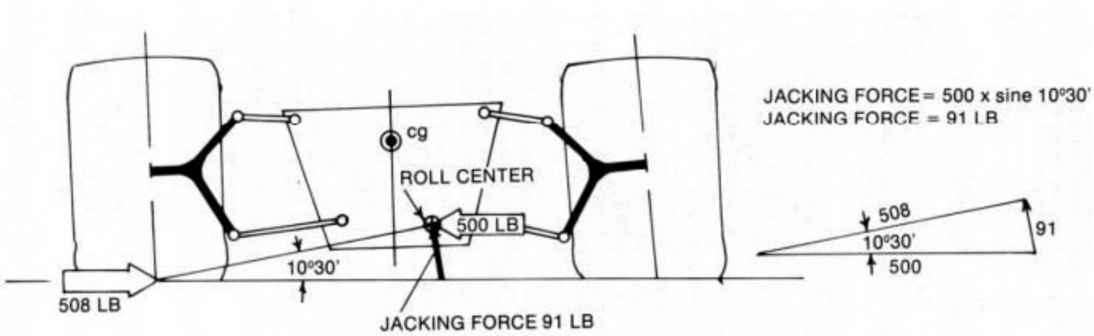


Fig 8:Calculating jacking force for a double wishbone set up. (Smith, 1978, p39)

2.4 Modern Day Suspension Configurations

1) 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)

2) 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 under steer or over steer. For a simple representation of the anti-roll bar in action, see the figure below

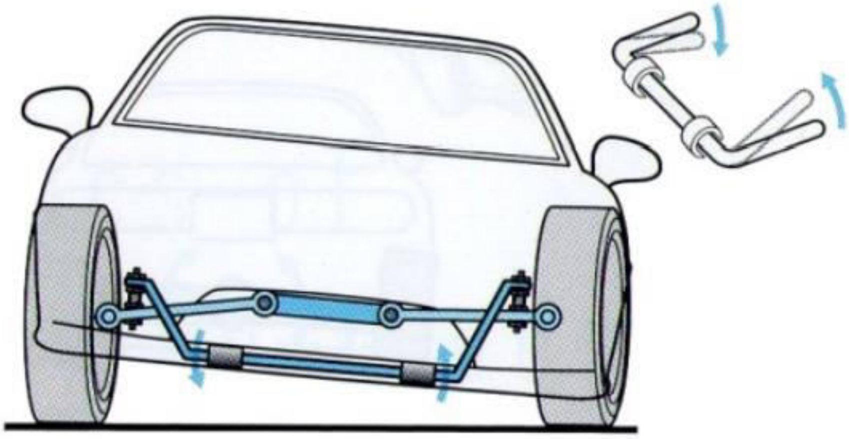


Fig 9: Typical anti-roll bar in action on a cornering vehicle

3) 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.

4) 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 below. The ends of the two wishbone arms and top end of the shock absorber will mount to the chassis.

Fig 10:Typical double wishbone suspension layout

- 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.

- 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. 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:

5) 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 arrangement the roll centre is taken to remain at ground level and to stay there under suspension actuation. (Smith, 1978, p47)

6) 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)

7) 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)

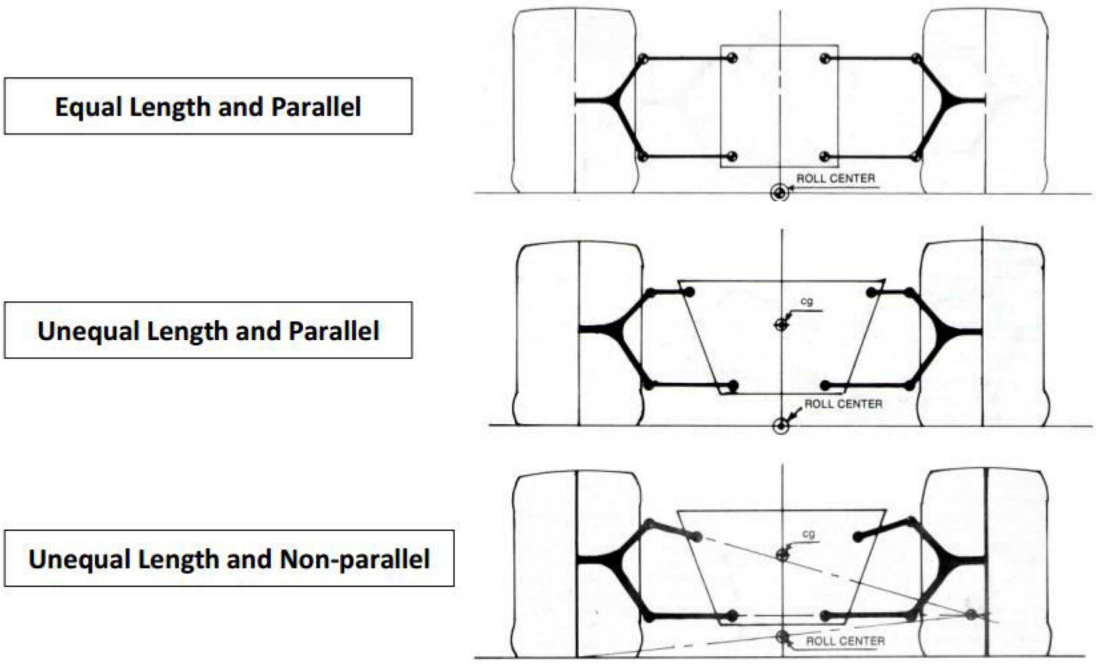


Fig 11: Different variations of the double wishbone suspension arrangement.

8) 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) 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)

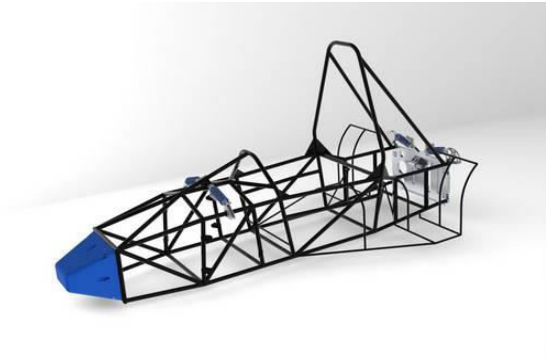


Fig 12: Inboard shock absorber

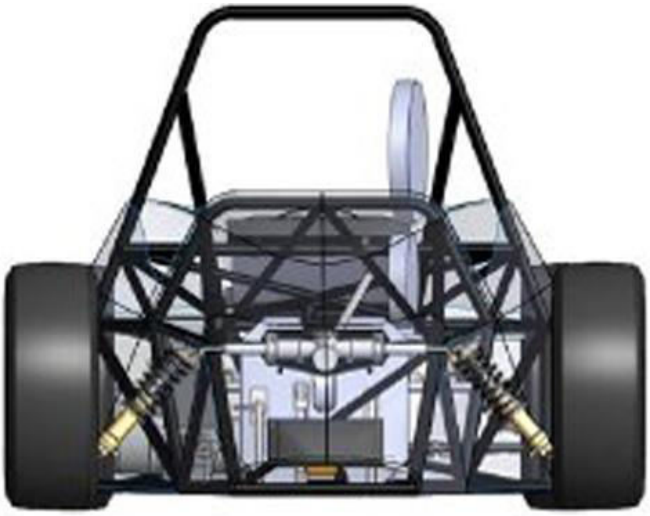


Fig 13: Outboard shock absorber

9) 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)

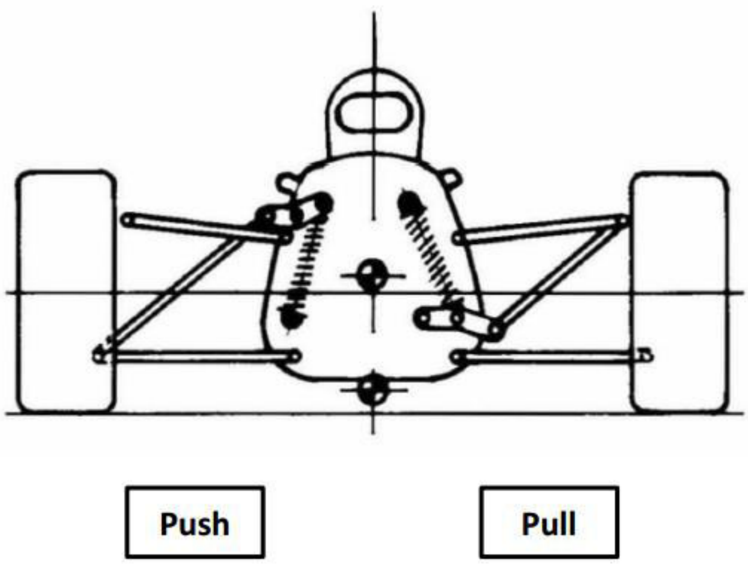


Fig 14:Push and pull inboard suspension configurations

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.

10) 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. 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 in the figure.

Fig 15:Typical MacPherson strut suspension layout.

- 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.

11) Rear Suspension Mechanisms

- Although a large array of commercial and domestic vehicles still uses 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.

12) 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.

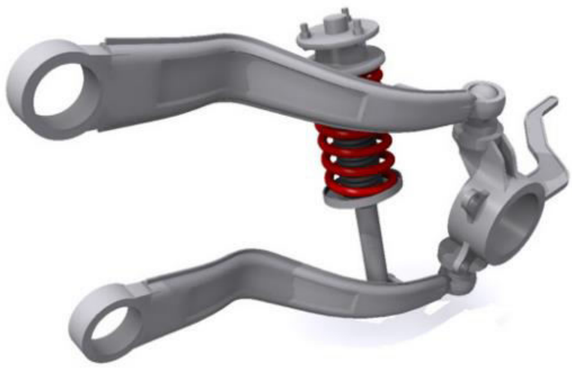


Fig 16: Typical trailing arm suspension layout

13) 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 above figure 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.

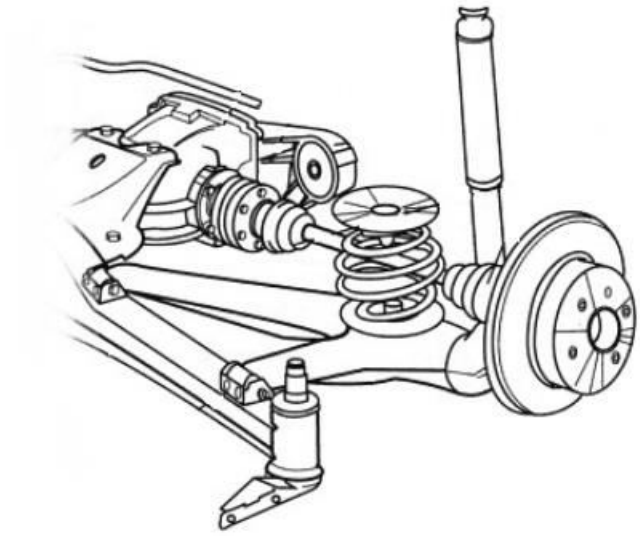


Fig 17: Typical semi trailing axis suspension layout.

14) 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. This design aims to provide a large degree of adjustability without compromising certain suspension characteristics.

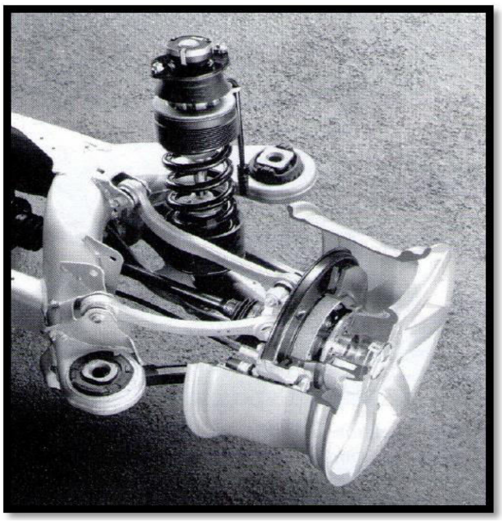


Fig 18:Single rear-wheel suspension of the Mercedes-Benz SL500.

15) **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 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.

2.6 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:
- 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.
- The next option basically uses a computer program to carry out the above mentioned process, saving significant amounts of time and resources.
- 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.

2.7 Chapter Summary

The literature review completed for the project is very extensive. The information uncovered has provided a decent knowledge basis and starting point for design to commence.

Chapter 3

Design Plan and Founding Decisions

3.1 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 FSG competition neither did every author's work agree with the design beliefs held by the project researcher.
- 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.

3.2 Project Design Plan

1) **Analyse the Formula SAE 2014 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 Formula Student Germany (FSG).
- Consider but don't design around any 'possible future rules changes' noted in the FSG rulebook

2) **Establish realistic performance targets for the FSG vehicle's suspension systems**

- 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.

3) Selection 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 and what has been proven by other competing teams at the FSG.

4) Make founding design decisions

- Type of suspension nominated (SLA Pushrod actuated).
- Wheel base and track proposed, again based on 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 to be designed.
- Values nominated may not be a representation of the final design as iteration will later be used to optimise the suspension geometry using Optimum K.

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 Optimum K by Optimum G.
- 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 FSG car and compare each set of results using MICROSOFT EXCEL 2014.

7) Choose dampers

- Selection of dampers based on required stroke, length, weight, degree of adjustability, quality and cost.
- What has been successful on past FSG 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 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 to 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 designed.
- 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.
- Integration with suspension model followed by testing in Optimum K to assess performance
- More iteration used in order to discover an arrangement that will satisfy the existing suspension design and offer performance characteristics (Self steer, steer angle etc) within reasonable range of what is desired.
- Front upright design fully completed with accommodation of the steering arms Physical design of components.

11) Physical design of components

- All components modelled in SolidWorks followed by finite element analysis (FEA) in Solidworks itself 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.

3.3 Performance Targets

The targets for the new suspension and steering systems needs to be realistic but also has to subsidise a final design that should allow the vehicle to compete reasonably well in terms of handling. The following list outlines the initial performance targets for the project's design. It should be recognised that all targets are listed in order of importance.

1) 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, our 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.

2) 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 that the driver needs during the race.

3) Economical

- Referring back to the motivations behind the FSG competition, providing an economical design is well rewarded as the competition is centred around mass producing the 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.

4) Maximised grip

5) Quick response handling

- Targets 4 and 5 are further goals related to the actual operation of the vehicle. 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 FSG circuits.

6) 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.

7) 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 occurs 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.

8) Simple

- Taking on board the guidance provided earlier by Prof. DurgaRao, it is also believed that design should be kept as simple as possible. On top of it simplicity is a target to be achieved 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.

9) Light weight

- 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 find a design that will produce improved performance and also yield a design that does not stretch a typical first year team to great lengths in order to create and build it.

3.4 Founding Design Decisions

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 our FSG 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. 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

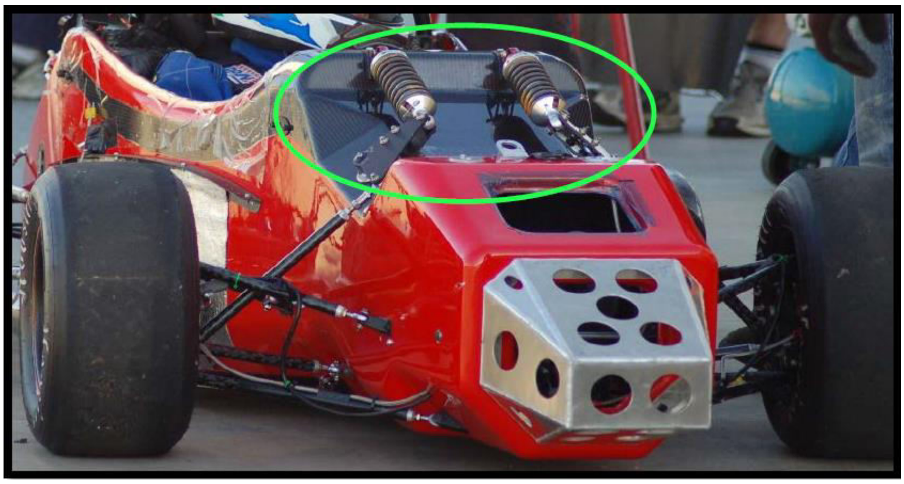


Fig 19: Front layout

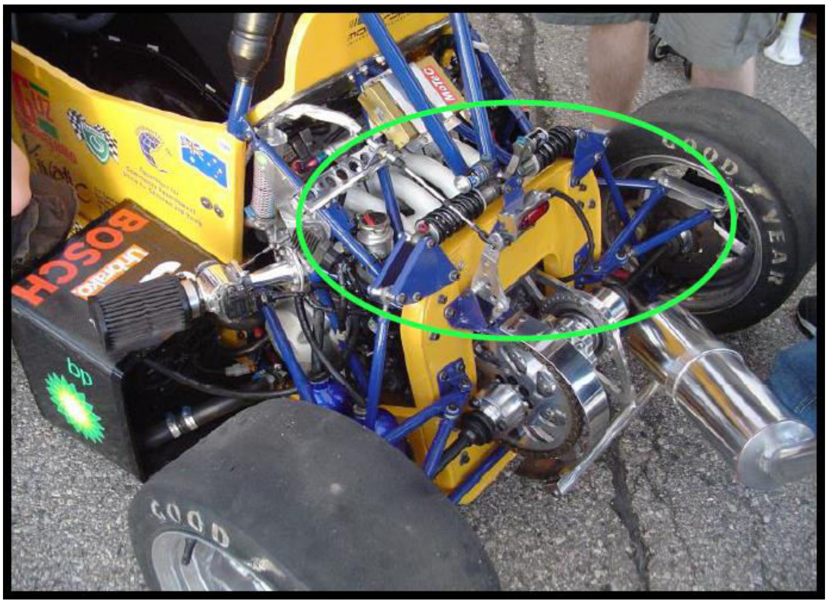


Fig 20: Rear layout

2) Tyres and Wheels

- In the FSG 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.

	10 INCH WHEEL	13 INCH WHEEL
ADVANTAGES	A 10 inch wheel can provide lighter weight, less rotational inertia meaning faster acceleration and a more compact design	13" Wheels on the other hand provide the opposite characteristics due to their larger size.
DISADVANTAGES	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.	

Table 1: Wheel Comparison

- Our 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 FSG vehicle and so using this size would increase the flexibility and adaptability of the design for future incorporation.

3) Wheels

- As mentioned earlier however, all wheels do not provide very good offset and so granted the vehicle with large scrub radiuses.
- A more promising option was found to be wheels obtained from OZ Racing who actually produce wheels specifically designed for the FSG competition. Unfortunately these wheels are made in the Germany and so getting them to India could prove to be quite expensive although these wheels were around half the price (US$245 not including postage) OZ Racing 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 OZ Racing, it was decided to use the model wheel in a 13” x 7” size with22” offset.

The selected wheel model can be seen below.

Fig 21: Wheel

4) Tyres

- With a new wheel nominated, a new tyre also had to be specified to accommodate the rim width. It was chosen that the design would use tyres by Hoosier, with the 20.5x7.0-13 model best suited to the selected wheels. Hoosier tyres provide 20% discount for all teams participating in FSG competition.

3.5 Geometrical allocations

The final wheelbase and track dimensions are based off dimensions of prior FSG 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 and 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 manoeuvrability." 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 FSG 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. 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. Using this value it is believed that handling will be manoeuvrable enough to navigate the tight and technical FSG competition tracks while still providing enough packaging room for components.

Wheelbase	1600(mm)
Front track width	1300(mm)
Rear track width	1200(mm)

Table 2: vehicle dimensions

- The track dimensions nominated are (front – 1300mm, rear – 1200mm) . Employing these dimensions have a number of advantage like the tracks are wide enough to provide sufficient manoeuvrability and lateral weight transfer characteristics while still allowing decent clearance for obstacles on the FSG 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 2014 FSG 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.

3.6 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. Based off opinions in the formulastudent forum and with recommendations from Pat Clarke in mind, the design will attempt to achieve a kingpin inclination of 0°-5.5° and a scrub radius of 0-50mm. These values are expected to provide ease of steering while still providing enough feel for the driver.

3.7 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 FSG 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 various teams 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 4

Suspension geometry

4.1 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.

4.2 Dimensioning the upright

A basic sketch of the upright and wheel assemble was drawn with selected dimensions of tyres and wheel using solidworks

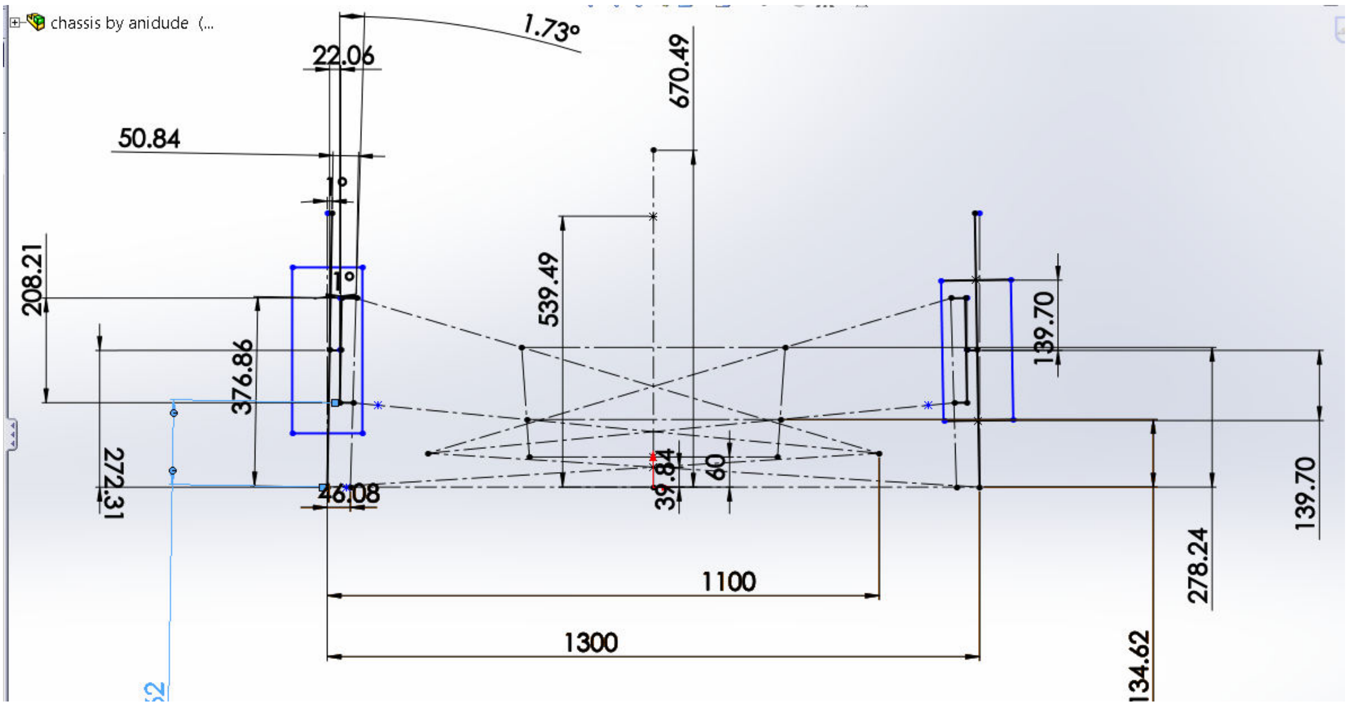


Fig 22: Dimensioning of front wheel assembly on Solidworks

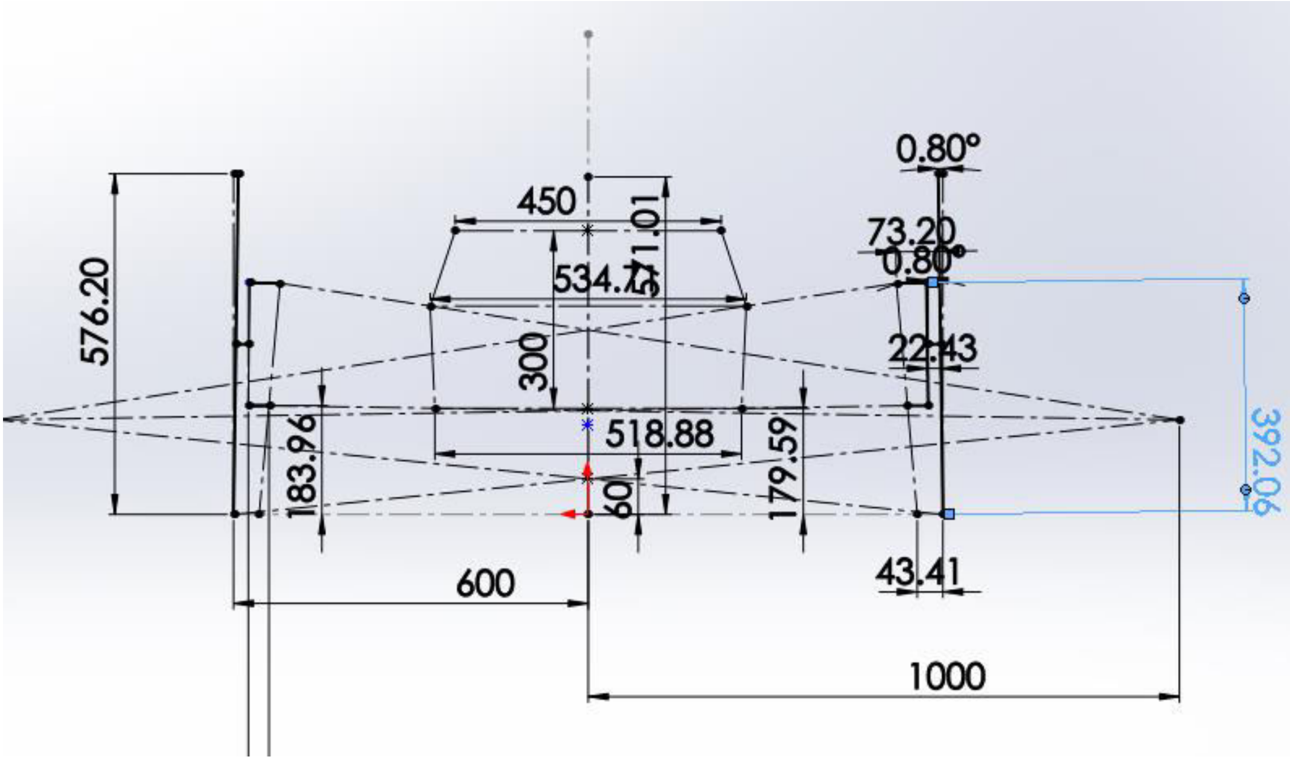


Fig 23: Dimensioning of rear wheel assembly on Solidworks

4.3 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 **optimum kinematics**and involved sets of iteration it first tested a largely varying group of geometries, 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, we 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.

4.4 Initial Decisions

1) **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 front and -0.8˚ for rear. This was not going to be the final value used in the design as this would be decided later after the final set of iteration.

2) **Caster and Trail**

- Like the camber for each iteration, the caster and trail also remained the same. For the front this was set at 5° caster with a trail of 42mm. Based off the dimensions of the rear upright, the caster was consequently 0˚.

3) **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**. An initial value of **1"** was selected for the front roll centre for reference.

4.5 Optimum Kinematics- About the software

- It isasimulation software which offers real time analysis of required suspension geometry allowing the user to analyse the geometry at various motions and track conditions.
- The input being 3-D coordinates of the solidworks sketches, the software creates the 3-D model accordingly.
- The results hence obtained are tabulated on excel spread sheets and compared to give the most optimum geometry.

For example:

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

Fig 24: Iteration spread sheet

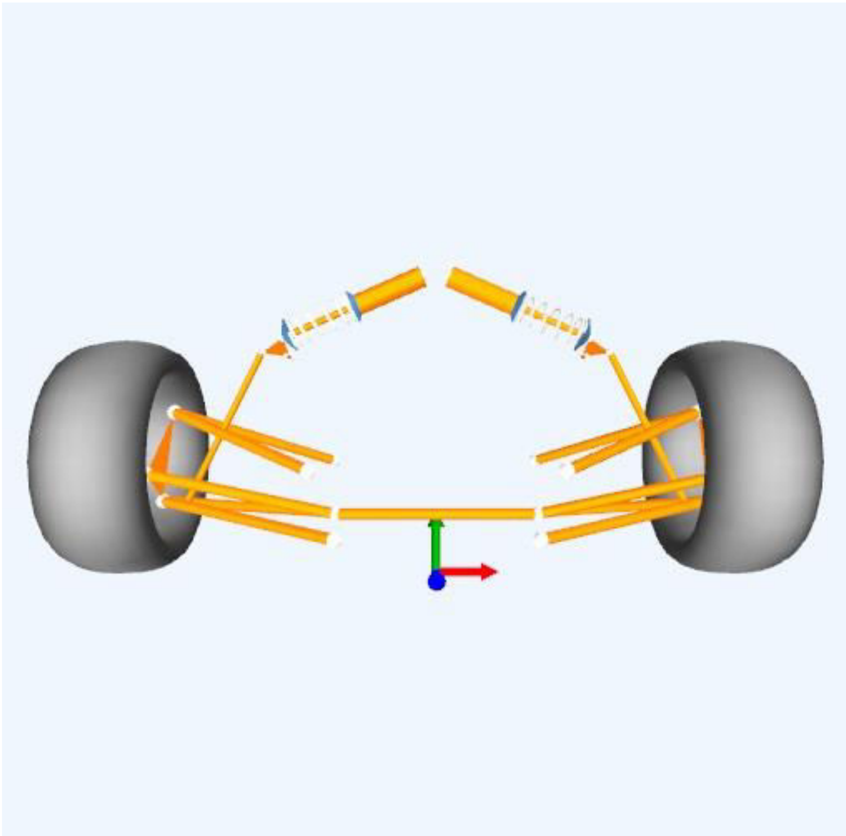


Fig 25: Representation of suspension geometry in optimum k

4.6 Testing Procedure

The following method describes the process used to test each separate geometry:

- Geometry entered into **optimum kinematics.**
- Geometry tested in bump and droop.
- Camber and roll centre location at max bump and droop noted in excel spread sheet
- Geometry tested in roll.
- Camber and roll centre location at roll extremities noted in excel spread sheet.
- Geometrical changes made in the system and changes were noted in the optimum kinematics software hence the best geometry was chosen.

4.7 Evaluation Criteria

1) **Relative camber of the front and rear**

- 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.

2) **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".**

3) **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

4) **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.**”

4.8 Results of iterations

As can be seen in the following final set of spread sheet the evaluation of each condition adheres to above mentioned concepts

H	I	J	K
camber in -3deg roll	**RC location in 3deg roll**	**RC location in -3deg roll**	**comments**
fr=-0.096 rr=0.915	front=-4.785 rear=72.598	front=-1.293 rear=73.377	camber movement is in same direction for both front and rear and roll center movement is equal and same direction
fr=-0.035 rr=-0.915	front=-4.785 rear=72.598	front=-1.293 rear=73.377	camber is in positive direction for heave undesirable rc movement similar to earlier data camber never goes in positive
fr=-0.031 rr=-1.672	front=-8.470 rear=72.598	front=-8.343 rear=73.377	
fr=-0.575 rr=-1.614	front=28.242 rear=74.77	front=28.48 rear=77.57	during roll RC movement is equal for both front and rear camber movement reduced during bump and droop
fr=-2.199 rr=-1.715	front=27.330 rear=78.54	front=27.023 rear=79.226	camber reduced for front while rc movement increased and camber change increased in rear
fr=-2.303 rr=-2.593	front=27.348 rear=77.5	front=27.023 rear=67.90	camber movement reduced rc movement in same direction
fr=2.204 rr=-2.593	front=28.489 rear=66.452	front=28.105 rear=67.939	no significant changes camber movement reduced further
fr=-2.172 rr=-2.63	front=27.586 rear=49.085	front=24.77 rear=51.07	roll center variation is less
fr=-2.0 rr=-2.62	front=28.556 rear=49.32	front=26.77 rear=49.98	**CAMBER NEVER GOES POSITIVE ROLL CENTER MOVEMENT IS IN SAME DIRECTION AND EQUAL DURING ALL MOTIONS**
fr=-1.51 rr=-2.01	front=28.71 rear=49.51	front=30.51 rear=51.246	**SIMILAR VALUES OF CAMBER AND RC OBTAINED**
fr=-1.696 rr=-1.902	front=25.882 rear=49.93	front=29.18 rear=51.046	CAMBER NEVER GOES POSITIVE
fr=-1.881 rr=-1.695	front=25.882 rear=49.93	front=29.18 rear=51.046	camber never goes positive !!
fr=-1.106 rr=-1.656	front=39.299 rear=51.344	front=40.267 rear=52.613	camber goes positive in one marginally other wise desirable results as roll center movement is in same direction
fr=-1.102 rr=-1.652	front=39.299 rear=51.344	front=40.267 rear=52.613	**camber goes positive in one marginally other wise desirable results as roll center movement is in same direction**
fr=-1.12 rr=-1.656	front=39.299 rear=51.344	front=33.089 rear=52,574	desired results

Fig 26: Excel spread sheet representing the iterations performed

4.9 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.

Chapter 5

Suspension actuation

5.1 Chapter Overview

Once the suspension arm geometry had been defined using the Optimum kinematics 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. This chapter documents the process used to design these elements of the suspension system along with the final solution believed to be optimal for application

5.2 Selection of Shock Absorber Model

The first step in this segment of the design process was to specify a shock absorber for the vehicle. 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.

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.

These were the 2013 Manitou Metel 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 2014 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 expected weight of **Hyperion Racing** vehicle was used to calculate the deflection or sag of the shock absorber:

$$w = ks$$

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

$$\frac{\frac{200\times 9.81}{2}}{\frac{250\times 4.45\times 1000}{25.4}} = s$$

$$s = 22mm$$

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. This meant that the design's wheel travel could be the same or very close to, the minimum value specified in the rules, which asmentioned 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.

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 3: comparison of dampers available

Fig 27: Manitou

Fig 28: Fox VAN-RC

5.3 Final decision

Considering lower cost of the Manitou it was chosen as the other parameters are near about the same.

Fig 29: CAD model of Manitou Metel

5.4 Calculations

1)**Motion ratio, Coil rate, Spring deflection**

Min wheel travel=50.8mm

Jounce=25.4mm

Rebound=25.4mm

Manitou metel- 200mm×50mm(length×stroke)

K=250lb/in=43.78N/m

Selecting motion ratio to be 1,

REAR

W=K*δ

Therefore, δ=W/K

δ-Sag at static condition

Weight=200kg

δ =$\frac{200*9.81}{43.78*2}$ = 22.40mm

To find spring stiffness, from Alan StaniforthPg. 186,

Coil rate (lb. /in)=$(\frac{wheelfrequency}{187.9})^2 * sprungweight$ =$(\frac{145}{187.8})^2$ *374.786

Wheel frequency-145cpm

Coil rate=223.423 lbs. /in

Wheel rate =$\frac{springrate}{1.27^2}$=155lbs/in

FRONT

W=K*δ

Therefore, δ=W/K

δ- Sag at static condition

Weight=200kg

δ=$\frac{200*9.81}{39.12*2}$ = 25.07mm

To find spring stiffness, from Alan StaniforthPg 186,

Coil rate (lb. /in) $=(\frac{wheelfrequency}{187.9})^2 * sprungweight = (\frac{145}{187.8})^2$ *326.284

Wheel frequency-145cpm

Coil rate=195.40 lbs. /in

Wheel rate $=\frac{springrate}{1.12^2}$=199 lbs/in

Motion ratio$=\frac{25.07}{22.40}$=1.12:1

2)Force calculation on wheel (Acceleration, Braking, Cornering)

Weight of car=350kg=772lb

C.G-60:40, C.G height-300mm=12 inches

Wheel base-1600mm=63inches

a) **BRAKING (1.7 g)**

From Caroll smith-Tune To Win pg 33 as shown below

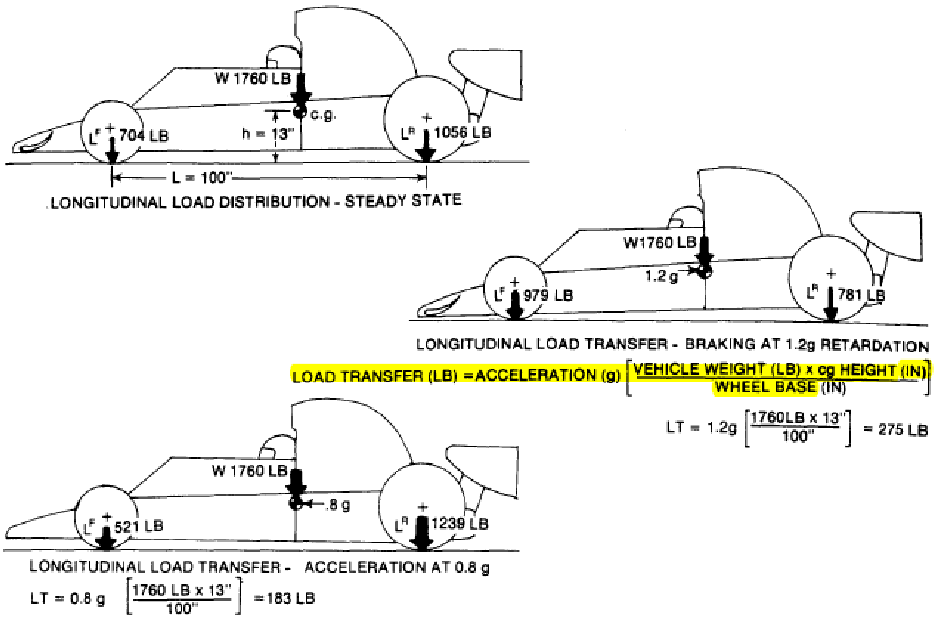


Fig 30: load transfer during vehicle motions

SAMPLE CALCULATION

Assuming (60:40) weight distribution for our car

FRONT REAR

308.8lbs 463.2lbs

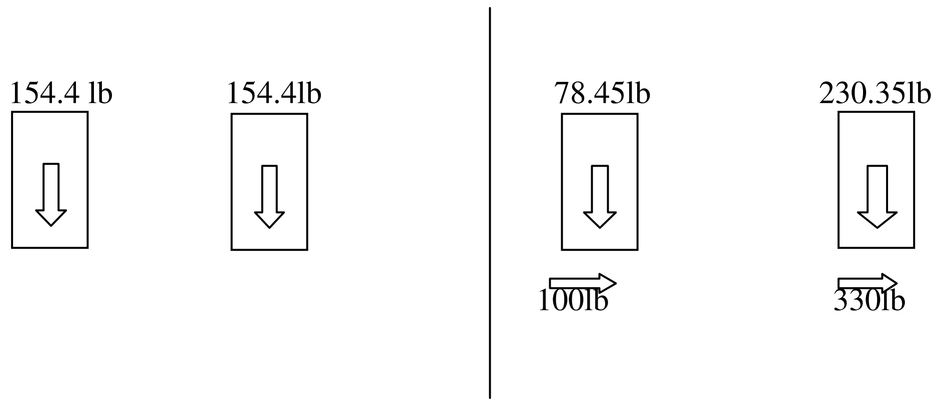
When car is stationery

Load transfer= $1.7 * (\frac{772*12}{63})$=250lb

FRONT REAR

558.2 lb 213.2lb

b) CORNERING (1g)



Load transfer=$\frac{1*772*5.0354}{51.18}$=75.9lb

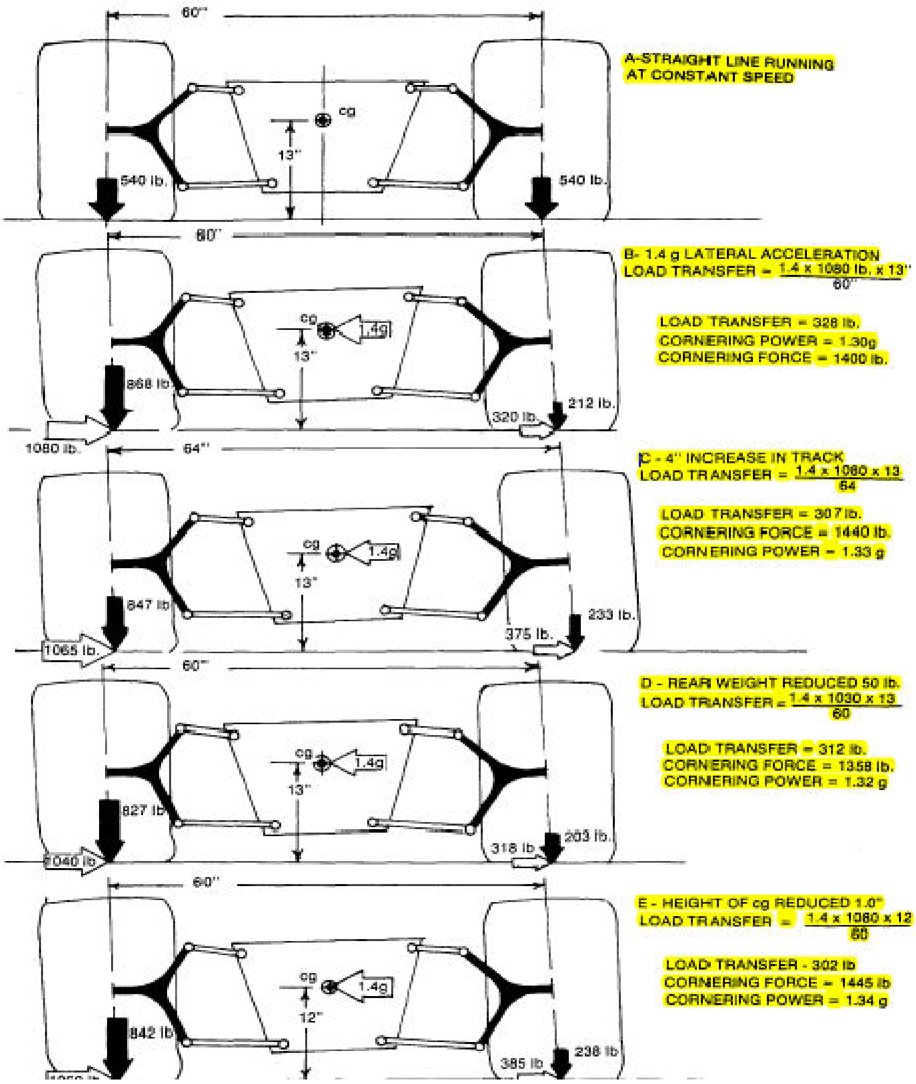


Fig: 28:

Simplified illustration of the relationship between track width gross weight, centre of gravity height and lateral load transfer.

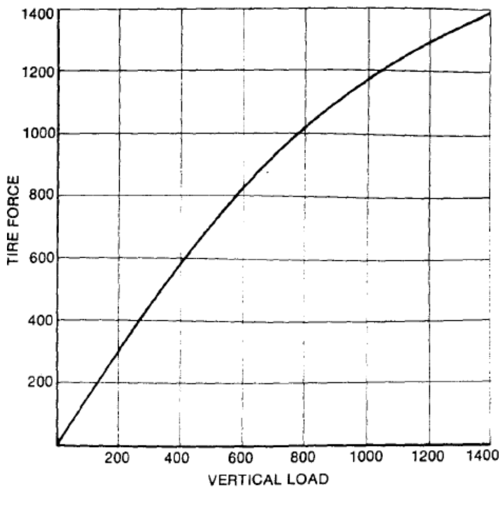
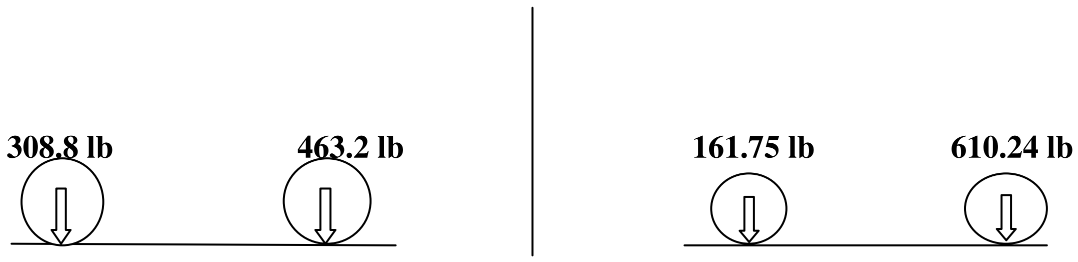


Fig 31: Vertical force vs. Tire force

c) **Acceleration(1g)**



When stationery

During acceleration

Load transfer=$\frac{1*772*12}{63}$

3)Axial force calculation (A-arms, toe link, tie rod)

- The forces transmitted through the various control arms was found out using basic mechanics by transforming the 3-D coordinates of the control arms into 6×6 matrix represented by,
  AX=B
- Where matrix A represents the position vector of each control arms and unit vector in each direction i.e. A is a 6×6 matrix where first 3 rows represents position vector of each control arm and last 3 rows represent unit vector in direction of each control arm.
- While matrix B which is a 6×1 matrix represents the forces on the wheel contact patch at various conditions.
- Matrix X is the amount of forces on each arm

For ex:

Matrix A

(-0.501 -0.795 -0.56 -0.827 0.594 0.93

-0.278 -0.12 -0.008 -0.012 -0.316 0.136

0.819 0.59 0.868 0.56 0.739 0.36

0.806 0.80 0.9645 0.9465 0.737 0.79

0.60 0.60 0.322 0.322 0.34 0.584

0 0 0 0 0.067 0.06)

Matrix B for vehicle under 1g acceleration

(1000

0

0

650

0

0)

- Similarly forces for various condition like cornering, bump was calculated using matrix multiplication.

	Acceleration(N)		**Cornering(N)**		**Bump(N)**	
	Front	Rear	Front	Rear	Front	Rear
Upper aft	2987	-424	-990	5298	2100	2890
Upper fore	-2700	-1800	760	-3100.9	-1120	2100
Lower aft	2287	-335	3957	-980	2450	890
Lower fore	-3523	3546	-790	1302	-1930	927
Toe link/tie rod	2900	1800	-2869	-1346	-800	2000
Push rod	-1290	2500	2890	-1900	3200	-4790

Table 4: Control Arm Forces

4) Bearing calculation (rocker)

Fr=5.5Kn=550 kgf (Safer limit)

Fa=0

H=2160 hrs

T=70^{o}c

Survival rate =92%

Equivalent load

P=(X*Fr*V+Y*Fa)*S.F

S.F- service factor

From PSG pg 4.2

Therefore, P=(1*500*1.2+0)*2=1100 Kgf

P_{o8}=0.92

$L_{o8}=\frac{HN60}{1*10^6}$

$L_{o8}=\frac{60*2160*500}{1*10^6}$=64.8 mr

From PSG 4.2

$$\frac{L_{o8}}{L_{1o}} = [\frac{\ln(1/P)}{\ln(1/P_{1o})}]^{1/b}$$

b=1.17 for median life

$$\frac{64.8}{L_{1o}} = [\frac{\ln(1/0.92)}{\ln(1/0.9)}]^{1/1.17}$$

L_{1o}=79.144 mr

Dynamic capacity

C= $(\frac{L}{L_{1o}})^{1/K} * P$

K-10/3 from psg

C= $(\frac{79.144}{1})^{3/10} * 1100$

C= 4082.42 Kgf= 48.98 Kn.

Deep groove ball bearing by skf (W629-2Z) was selected for rocker (Bell crank)

5.5 Overall detail

Parameters	Front	Rear
Wheel base	1600mm	
Track width	1300mm	1200mm
Weight distribution	49:51	
Tyre Dimension	20.5×7×13	

Table 5: Vehicle static parameters

Parameters	Front	Rear
Roll centre height	25.4 mm	50.8mm
Static camber	-1˚	-0.8˚
Caster	5˚	0
KPI	5.18	-

Table 6: Vehicle kinematics

5.6 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 FSAE vehicle.

Chapter 6

Component design and cost report

6.1 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 the suspension geometries are physically modelled. On top of this, each component's material and anticipated manufacturing process is detailed.

6.2 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 aluminium (6061-T6) billet, with the circular tapered bearing housing at the centre of the upright. The cut outs and holes would be completed with a milling machine and drill.

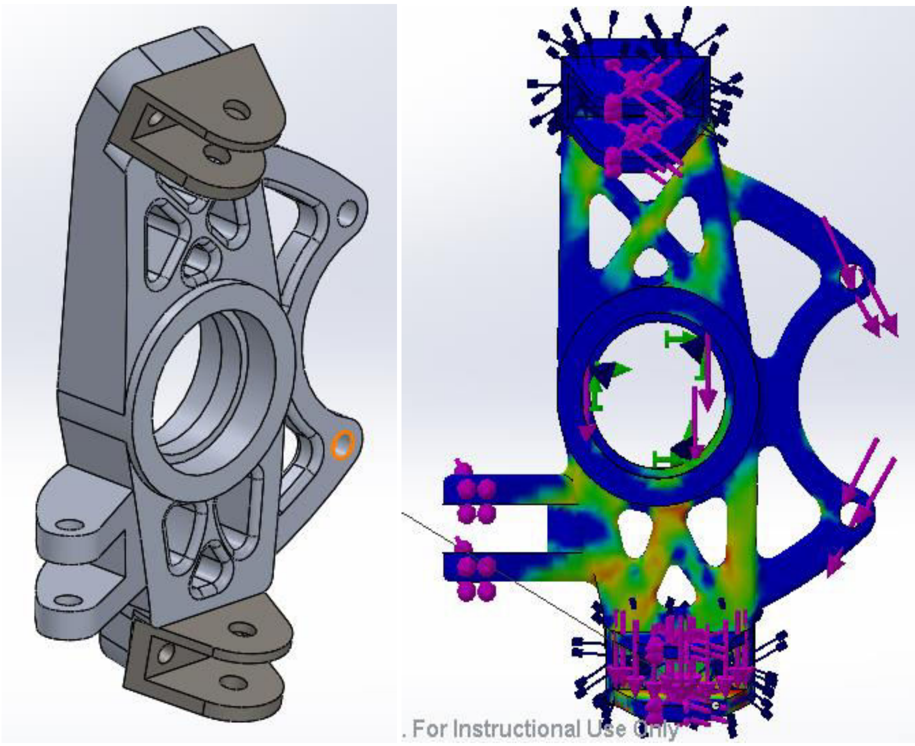


Fig 32: Front upright

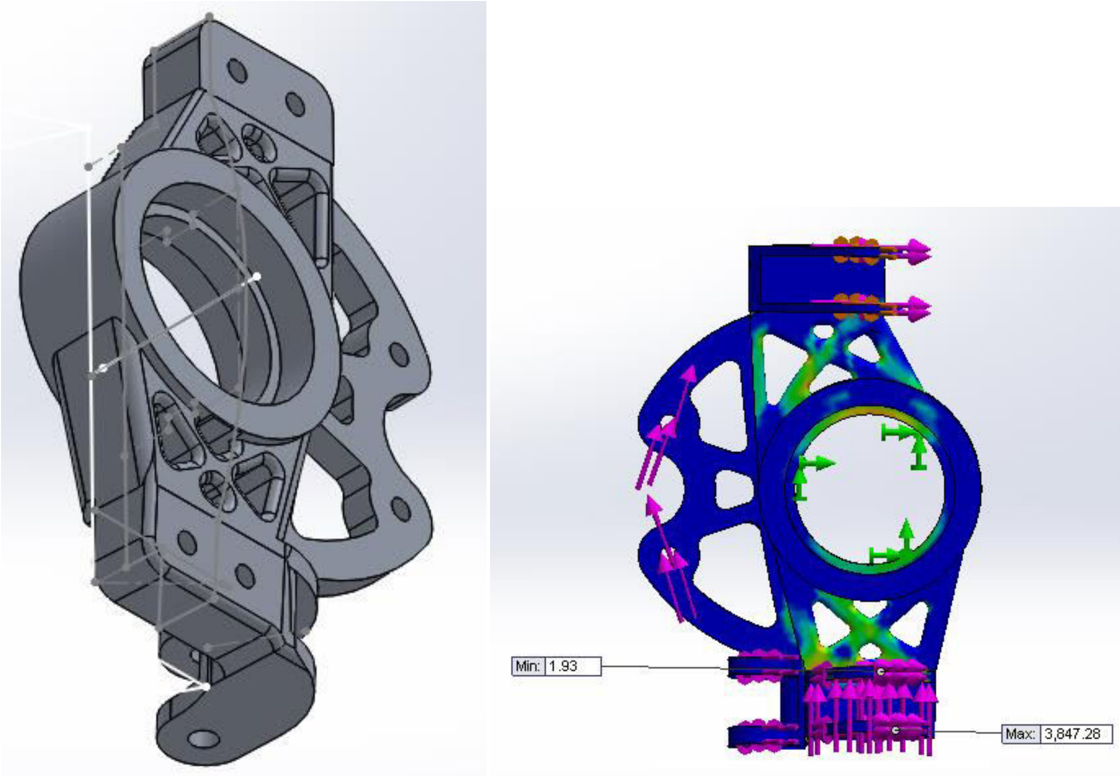


*Fig 33:*Rear upright

6.3 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 verysimilar. The intended material is mild steel due to its decent strength, rigidity, and design flexibility/reparability. 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 laser cut steel tabs to connect each tube to the chassis and to house the spherical rod end bearing. 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 is quite simple and avoiding rod ends in bending (REIB). 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 spherical rod end bearing which are required to provide angular rotation of the suspension components and so will feature threads on the inner side.
- The suspension arms will also mate with the spherical bearings used to support the uprights. These will press into the machined hole in the laser cut steel plate (BSK46)

that joins the two CHS tubes of each suspension arm. This bearing will be detailed later.

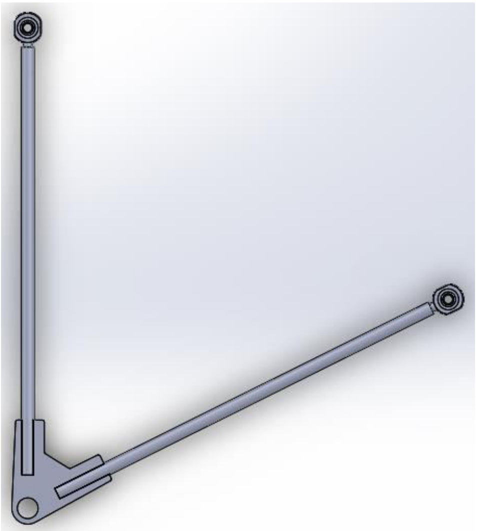


Fig 34: Front upper arm

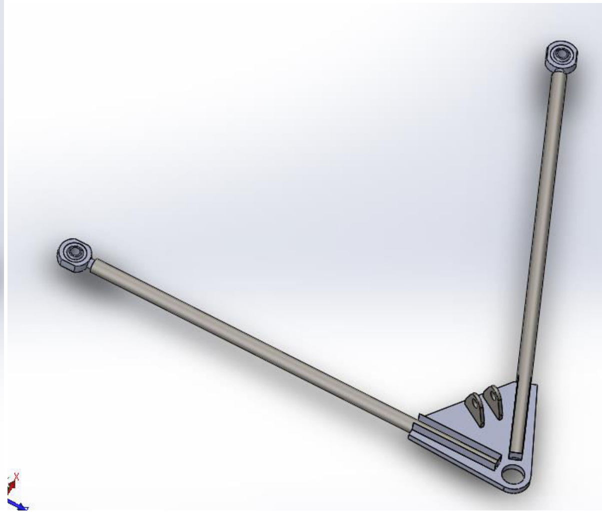


Fig 35: Front lower arm

6.4 Rocker (Bell crank)

- 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(6061-T6) and incorporate two thin plates separated by a hollow circular AL rod 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. 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.

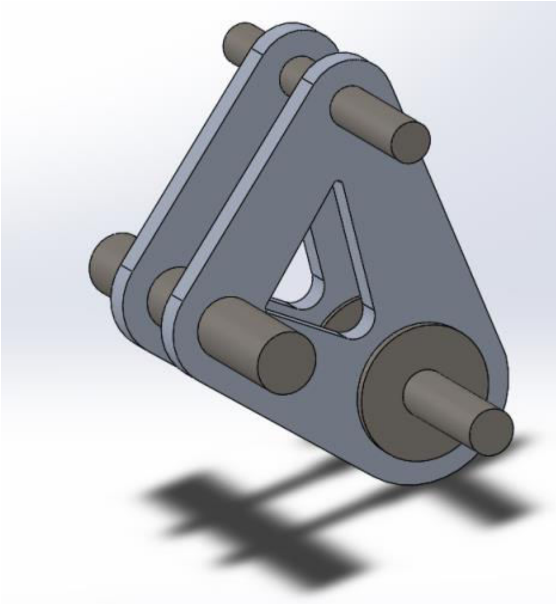


Fig 36: Front rocker

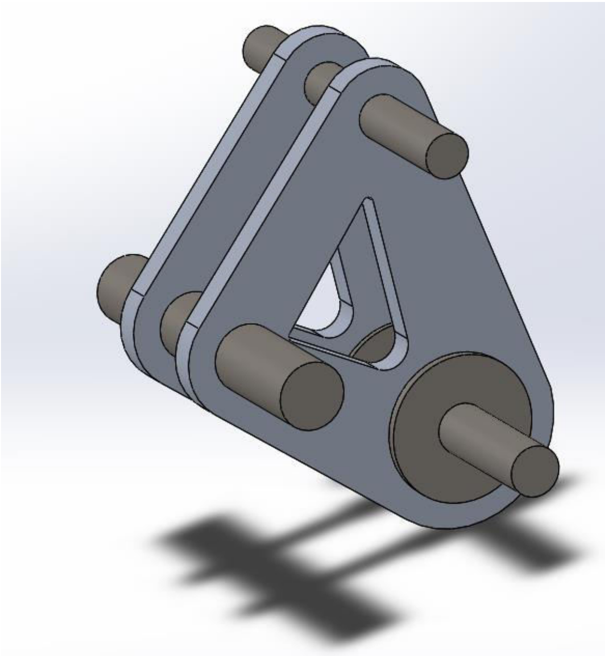


Fig 37: Rear rocker

6.5 Bearings and Fasteners

- The bearings used were SAR1-10 spherical bearing and SAKB8F 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 handused on the suspension arms, tie rods, toe links and push rods. 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.

Fig 38: SAKB8F

Fig 39: SAR1-10

6.6 Complete Suspension System

- The following two figures define the assembly of the suspension 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 *OZ Racing* 13” x 7” wheel plus a rough representation of the *Manitou* Swinger 4-way shock absorbers.

6.7 Cost report

Components	Material	No’s reqd	Source	Total cost (Rs)
Front Upright	Al 6061-T6	2	Local vendor	3500
Rear Upright	Al 6061-T6	2	Local vendor	3500
OZ Racing 13” wheel	-	4	OZ Racing	1,52,000 (upto Dubai)
Hoosier tyre 20.5*6*13	-	8	Hoosier	2,00,000
Material for A-arms , pushrod, toelink etc.	AISI-1018	5 meters	Local vendor	6000
Front Rocker	Al 6061-T6	2	Local vendor	1500
Rear Rocker	Al 6061-T6	2	Local vendor	1500
Manitou metel 4 way adjustable.	-	4	Manitou	26,000
SAKB8F (Rod end bearing)	-	32	Local vendor	2560
SAR1-10(Spherical bearing)	-	8	Local vendor	640
W629-2Z(DGBB)	-	8	Local vendor	1200
Miscellaneous	-	-	-	8000
			Total cost	4,06,400

Table 7: Cost report

6.8 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 all components assembled to form this system.
- 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 FSG vehicle.
- Images of partly assembled view of our car.

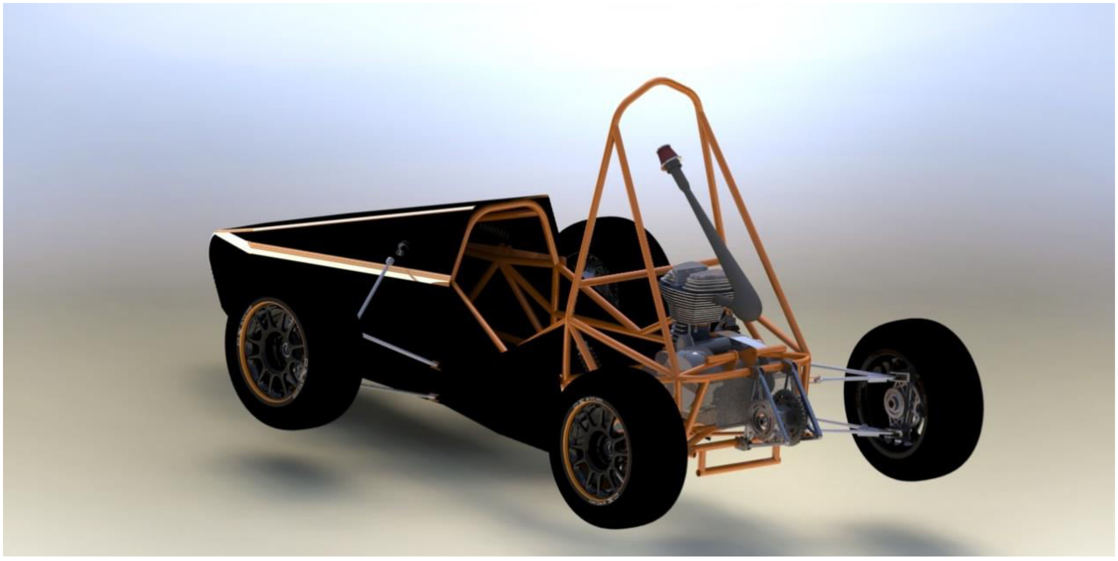


Fig 40: Vehicle assembly Isometric view

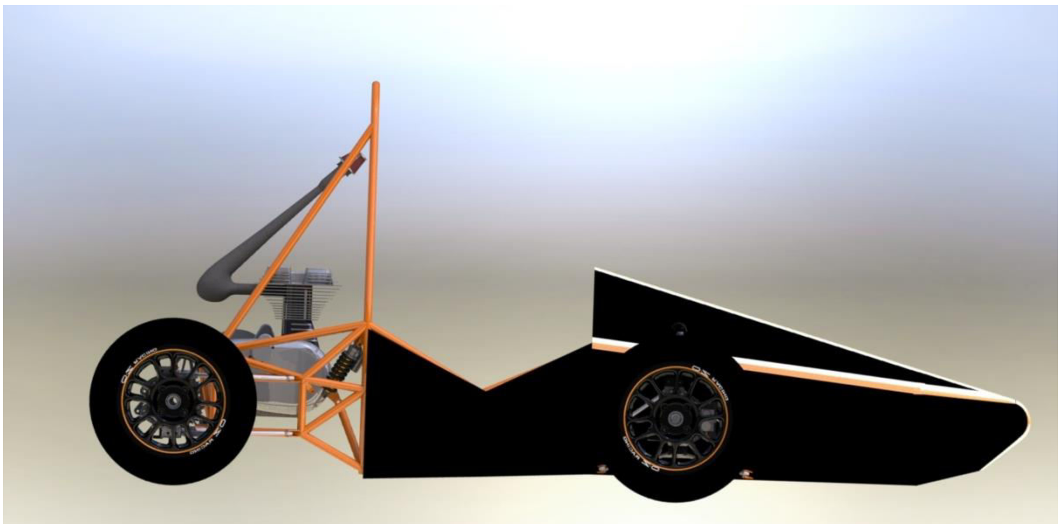


Fig 41: Vehicle assembly S.V

Chapter 7

Conclusions

7.1 Chapter Overview

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

7.2 Conclusions

- A literature review uncovered information on fundamental concepts relating to the suspension and steering of a car, commonly used racing suspension 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 2014 FSAE competition rules provided a number of limits and further guidelines for the design.
- Thereafter the various design procedures used and various design decisions were discussed concluding with the complete assembly of the suspension for the 2014 FSAE vehicle for HYPERION RACING.

Bibliography

- Fox Van RC Coil Rear Shock '12 2011, online product listing, Jenson USA Riverside, http://www.jensonusa.com/store/product/RS259B02 Fox+Van+Rc+Coil+Rear+Shock+12.aspx >.
- 2014 Formula SAE Rules, SAE International.

- Chris Longhurst 2011, The Suspension Bible, http://www.carbibles.com/suspension_bible.html
- Gabriel de Paula Eduardo 2005, Formula SAE Suspension Design, paper series E, SAE International Brasil, http://www.theoryinpracticeengineering.com/resources/fsae/fsae%20suspension%20design%20brazil%20style.pdf.

- Milliken, W.F. & Milliken, D. L. 1995, Race Car vehicle dynamics, SAE International, Warrendale, PA USA
- Staniforth, A. 1991, Competition Car Suspension, Haynes Publishing, Sparkford UK.

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

- Caroll Smith –Tune to Win

- Machine design by V.B. Bhandari
- PSG
- Google images

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