Σχολή Μηχανικών
Παραγωγής και Διοίκησης

Diploma Thesis

VEHICLE MECHANICAL DESIGN WITH FOCUS ON THE FRONT AND REAR SUSPENSION THAT MEET THE 2024 FSTUC VEHICLE DYNAMICS ATTRIBUTES

GATOS V. GEORGIOS

Department of Production Engineering and Management

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Ευχαριστίες

Αρχικά θα ήθελα να εκφράσω τις ευχαριστίες μου προς τον καθηγητή και επιβλέπων του μεγαλύτερου μέρους της παρούσας διπλωματικής κύριο Διομήδη Κατζουράκη ο οποίος με καθοδήγησε και με συμβούλευσε καθ’ όλη την διάρκεια της εκπόνησης της διπλωματικής μου εργασίας καθώς επίσης και στον κύριο Παναγιώτη Αλευρά για την σημαντική βοήθεια με σκοπό την ολοκλήρωση της. Θα ήθελα επίσης να ευχαριστήσω όλη την ομάδα της FSTUC για την συνεργασία. Τέλος θα ήθελα να εκφράσω την ευγνωμοσύνη μου στους γονείς μου και στην οικογένεια μου που με στήριξαν και ήταν δίπλα μου καθ’ όλη την διάρκεια των σπουδών μου, καθώς και τους φίλους μου και στην κοπέλα μου για όλη την στήριξη.

Περίληψη

Η παρούσα διπλωματική εργασία συγκεντρώνει την μεθοδολογία για την ανάπτυξη του συστήματος ανάρτησης σε ένα αγωνιστικό όχημα Formula Student. Παρατίθενται γενικές πρακτικές ανάπτυξης ενός προϊόντος που χρησιμοποιούνται στη αυτοκινητοβιομηχανία με έμφαση στην δυναμική οχημάτων και στον σχεδιασμό συστήματος ανάρτησης. Βασικός στόχος της διπλωματικής ήταν ο σχεδιασμός του μπροστά και πίσω συστήματος ανάρτησης βασισμένο σε ορθές μηχανικές πρακτικές. Σε πρώτη φάση σχεδιάστηκε και μελετήθηκε η συμπεριφορά της μπροστά ανάρτησης του προηγούμενου οχήματος της ομάδας και στην συνέχεια μετά από ανάλυση της κινηματικής και δυναμικής συμπεριφοράς προχωρήσαμε στον σχεδιασμό του συστήματος ανάρτησης για το νέο μονοθέσιο. Το πλαίσιο και το σύστημα ανάρτησης μαζί με τα περισσότερα από τα εξαρτήματα που τα αποτελούν κατασκευάστηκαν στα εργαστήρια του Πολυτεχνείου Κρήτης.

Λέξεις κλειδιά: suspension design,CAD, vehicle dynamics

Abstract

This thesis summarizes the methodology for developing a Formula Student Vehicle Suspension System. It presents a streamlined version of the automotive development process, with a focus on Vehicle Dynamics and Suspension Design. The primary task was to design the front and rear suspension for the FSTUC 2024 vehicle based on Vehicle Design Engineering principles. First, a virtual model of the 2023 front suspension geometry was created. The goal was to analyze the behavior of this suspension. With the results of these analyses, the development of the FSTUC 2024 Front and Rear Suspension could begin. The Chassis and the Suspension Systems were developed within the context of the present thesis inhouse at the Technical University of Crete.

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1 Introduction

1.1 Outline

The main task of this thesis was to design the suspension of a Formula Student car by using tools like CAD for analyzing geometries and 3D designs and using MATLAB in order to verify simply suspension calculations. It was quite a challenging problem because we must think about the rules and regulations of the competition, timeline economics as well as manufacturing limitations in order to design a car that meets our goals.

Picture 1: The 2023 Car during the manufacturing Phase

1.2 Formula Student Competition

Formula Student is the largest engineering competition in the world were students design build and race formula type racecars within certain regulations.

The Competition has both Static and Dynamic Events. Static events are Business Plan Presentation, Cost Manufacturing and Design Event. For the dynamic events the teams need to compete on Acceleration, Autocross ,Skidpad,Endurance and Efficiency.(Formula Student Rules 2025 D1, Dynamic Events General)

1.3 Competition Regulations

Ground Clearance

T2.2.1: The minimum static ground clearance of any portion of the vehicle, other than the tires, including a driver, must be 30 mm. If an active suspension system is installed, the static ground clearance is measured in the lowest adjustable position.

Suspension

T2.5.1 The vehicle must be equipped with fully operational front and rear suspension systems including shock absorbers and a usable wheel travel of at least 50 mm and a minimum jounce of 25 mm with driver seated.

T2.5.2 All suspension mounting points must be visible at technical inspection, either by direct view or by removing any covers.

1.4 Problem Statement

The design of the suspension system of a Formula Student vehicle presents a complex engineering challenge. The suspension system should be designed in that way that provides optimal handling, stability and safety under racing conditions. Some of the key requirements are overall vehicle performance, optimal weight distribution, packaging, minimizing weight and compliance with the regulations of the competition and of course keep it as simple as possible.

1.5 State-of-the-art

The development of a Formula Student car is a highly collaborative process involving various aspects of automotive engineering, with the chassis and suspension systems being key components in achieving optimum vehicle performance.

Over the years, advances in materials, simulation tools and design methodologies have significantly improved the development of Formula Student vehicles. Most teams use a chassis spaceframe made from carbon steel while others prefer to build a monocoque as it is lighter. Now regarding the suspension, most teams use a pushrod configuration at the rear and at the front both pushrod and pullrod configurations and direct mount suspension can be seen in various cars.

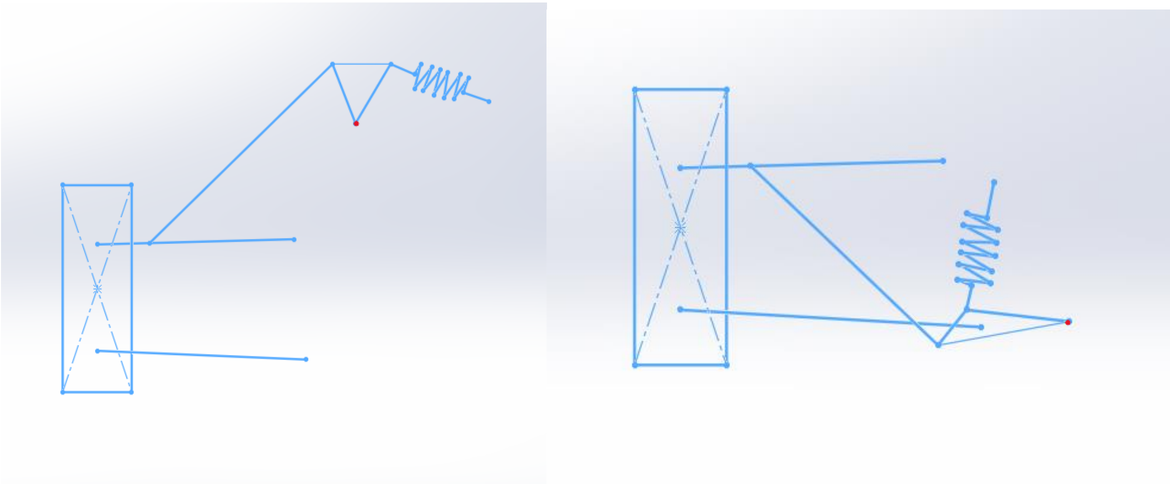


Fig1 : Pushrod and Pullrod Configuration

1.6 Objective

The design of the Car within the Rules and Regulations that meets the target attributes

Fixed:

- Wheelbase/trackwidth: 1.6 m, 1.2 m
- Wheels/tires: 13” constrained by competition
- Front/rear ride rates: 2Hz
- Anti-lift and Anti-squat: 0%
- Front/rear static camber: -1 deg
- Front/rear roll center height: As close as possible to the ground

Adjustable:

- Front/rear static toe: 0.1 deg (adjustable)
- Dynamic toe: none (adjustable via toe link vertical movement)
- Damping: critical on jounce (rebound 3x more damping jounce)
- Understeer targets: 2 deg (road wheel angle)/9.81 m/s^2 at 6 m/s^2 (adjustable via stiffness balancing and dynamic toe adjustment)

1.7 Methodology

The Methodology in the following outlines the approach taken to design the Suspension system for FSTUC’S Vehicle. First a virtual model of the front suspension from the previous year vehicle was created. Gathering data from the previous year’s vehicle was important to design the new suspension. 3D

Models were created for analyzing the behavior of the front Suspension. After that we identify weaknesses and set some design goals for the new suspension system and compare them with other teams to understand the trade-offs they have made in terms of suspension geometry. The Chassis was designed firstly in order to meet regulations of the competition and safety of the driver, and secondly to have optimal interaction with the components such as wishbones, engine, suspension. For the new vehicle various designs had occurred for both front and rear suspension and kinematic analysis were done to understand the geometry of the Suspension. After validating the final suspension design kinematic behavior, we move further into more intrusive design of the component's simulation and parts selection such as bolts, nuts and ball-joints.

2. SUSPENSION GEOMETRY ATTRIBUTES

This chapter explains the parameters and characteristics that are critical to the design of the suspension system and the dynamic behavior of the vehicle. The aim is to provide readers with an understanding of the document's theoretical basis.

2.1 Attributes

An attribute is a high-level aspect of how the user of the vehicle perceives the vehicle.

Attributes and Functions are used to establish processes and structures for requirement setting and verification within a vehicle engineering organization. Such processes and structures are important to enable a good overall design of such a complex product as a vehicle intended for mass production at affordable cost for the customers/users. Fig. 2 tries to give an overview, with reference to the well-known V&V process, how a vehicle program generally is organized.

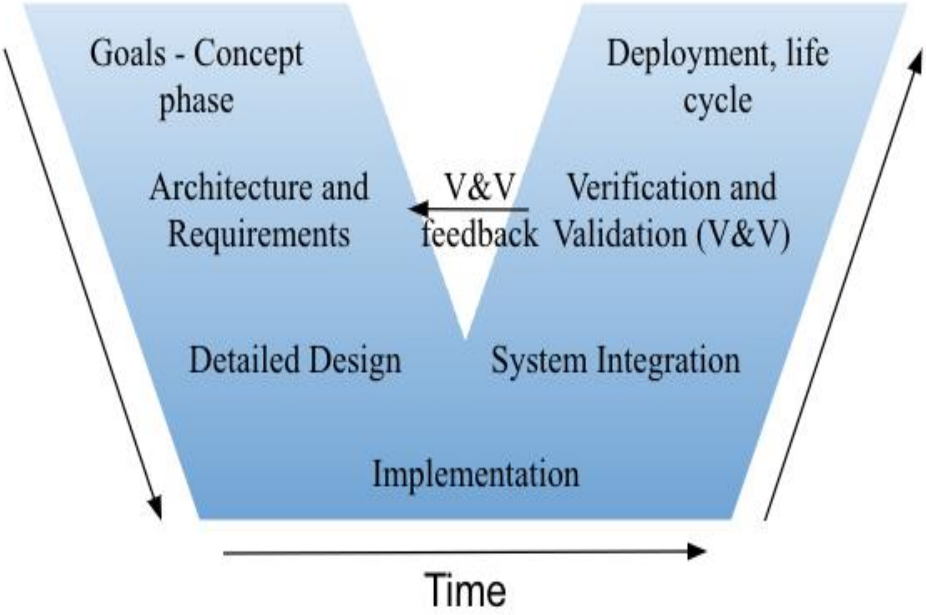


Fig 2: V-process applied to a vehicle program

2.2 Functions

A function is more specific than an attribute. Ideally, it defines a measure which is possible to set (quantitative) requirements on. The (complete vehicle) function does not primarily stipulate any specific subsystem. However, the realization of a function in a particular vehicle program normally only engages a limited subset of all subsystems. So, the function will pose requirements on that subsystem.

2.3 Requirements

A requirement shall be possible to verify as pass or fail. A requirement on the complete vehicle is typically formulated as:

"The vehicle shall … do something or have measure … $<or>or\approx$ … number [unit] … … under certain conditions."

<u>Examples of requirements</u>

• The vehicle shall accelerate from 50 to 100 km/h on level road in <5 s when acceleration pedal is fully applied.

• The vehicle shall decelerate from 100 to 0 km/h on level road in <35 m when brake pedal is fully applied, without locking any rear wheel.

• The vehicle shall turn with outermost wheel edge on a diameter <11m when turning with full-lock steering at low speed.

• The vehicle shall have a characteristic speed of 70 km/h (±10 km/h) on level ground and high-friction road conditions and any recommended tires.

2.4 Hardpoints

A hardpoint refers to a fixed, specific location on a vehicle's structure or chassis where components are mounted. These points are critical to the design and engineering process because they define the geometry and positioning of major parts like the suspension, engine, transmission, or steering system. Hardpoints determine the fundamental structure and performance characteristics of the vehicle.

3. SUSPENSION

3.1 Suspension Configurations

3.1.1 Double wishbone

A double wishbone suspension is an independent suspension design for automobiles using two wishbones. Each wishbone has two mounting points to the chassis and one joint at the knuckle/upright. The damper /spring component could mount directly to the wishbone to control vertical movement or could be mounted to a suspension rod with a rocker arm in order to control the movement and pack better the suspension system when it comes to aerodynamics.

3.1.2 Push Rod configuration

This configuration has one upper wishbone, one lower wishbone and the suspension rod that it is connected to the rocker arm and then the spring. The main idea is that when the tire bumps the wheel moves upwards an the suspension is compressed (Pushed).

3.1.3 Pull rod configuration

Again, there is an upper wishbone ,a lower wishbone and the suspension rod that is connected to the rocker arm and then the spring. The difference is that the suspension rod is mounted higher at the wheel assembly and at a low point on the chassis.

The rocker arms have a pivoted point that they can turn around it and it attached to the chassis. Usually, they have one mount for the Suspension rod and one for the spring/damper element but usually have another mount used for anti-roll bars attachment. The rocker arms can be used in order to convert the motion ratio of the suspension system and give the engineers more freedom to design the suspension systems.

3.2 Geometry Definitions

3.2.1 Wheelbase and Trackwidth

Wheelbase is the distance of the center the front and rear tire of the vehicle in side view (Fig 2).In general long Wheelbase decreases pitch while short ensures better maneuverability.

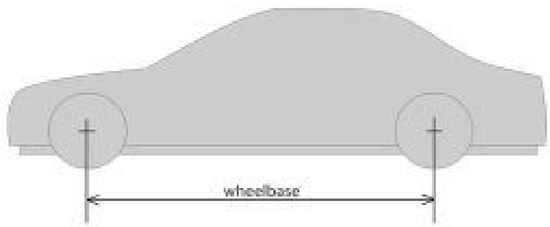


Fig 3 :Wheelbase

Trackwidth is the distance between centers of left and right tire on the same axis when in front view. Wider track width reduces body roll and it can be different for front and rear axis.

3.2.2 Camber Angle

Camber angle is the angle between the wheel plane and the vertical, positive when the top of the wheel leans outward.

(Chalmers Vehicle Dynamics Compendium for course MMF062 2.2.1.2 Sector)

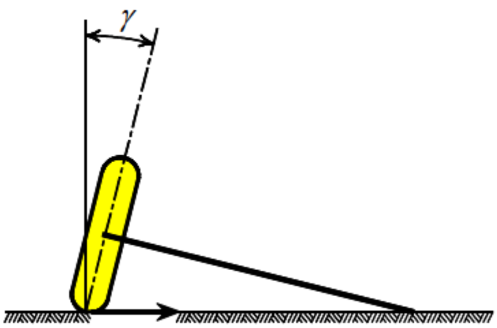


Fig 4:Camber Angle ¨γ¨

3.2.3 Positive Camber Angle

Positive camber ε_0 is usually defined if the wheel leans outward from the car body(Fig.4).The camber angles with respect to the car will be denoted ε and with respect to the ground γ.

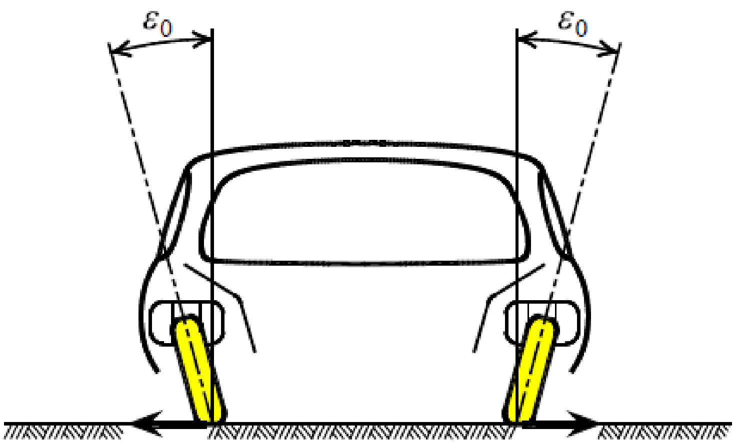


Fig. 5: Positive Camber Angle

3.2.4 Negative Camber Angle

Negative camber ε_0 is usually defined if the wheel leans inward from the car body(Fig 5).

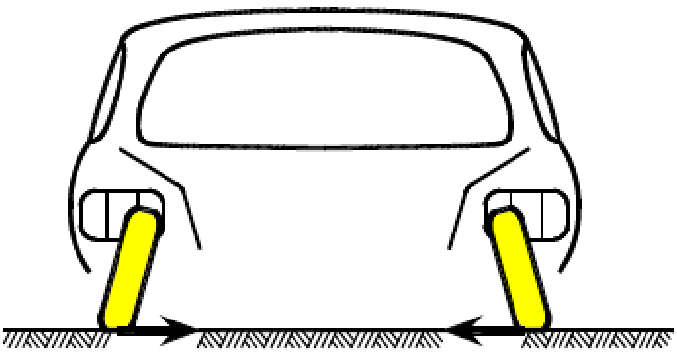


Fig 6:Negative Camber Angle

Negative Camber Setup in general improves handling and road grip, providing an equal load throughout the contact patch. It also minimizes tire vibration and allows for better cornering at high speeds. Although it can lead to oversteering and make the vehicle move toward any road irregularities or bumps.

3.2.5 Toe angle

Toe in/Toe out

Toe angle (or toe-in) is defined as an axle with two wheels (not for a single wheel), as the difference between right and left wheel's steering angles. Toe angle is positive if front ends of the wheels are pointing inward. Hence, toe can be called toe-in. Negative toe can be called toe-out. Toe angle generates opposing lateral forces on each side. The toe angles vary with the tire forces, due to suspension linkage geometry and elasticity in suspension bushings. (Chalmers Vehicle Dynamics Compendium for course MMF062 2.2.1.4 Sector)

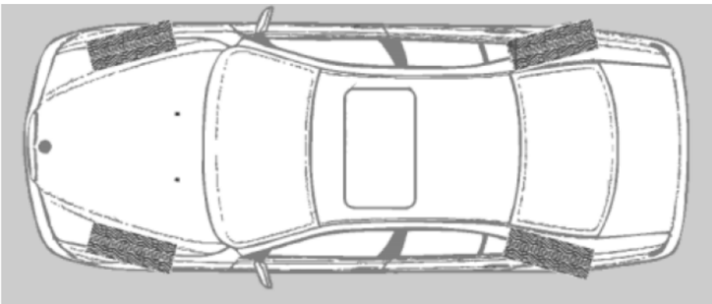


Fig 7:Toe in

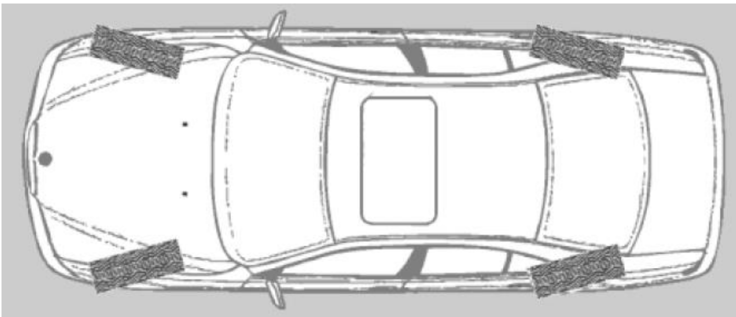


Fig 8:Toe out

3.2.6 Caster Angle

The angle in a side-view between the steering axis and the vertical axis is called Caster angle. It is positive if the top of the steering axis is inclined backwards. Caster angle provides an additional aligning torque, see 2.5.6.2 and Section 4.2.3.2, and changes the camber angle when the wheel is steered (Chalmers Vehicle Dynamics Compendium for course MMF062 2.2.1.3 Sector)

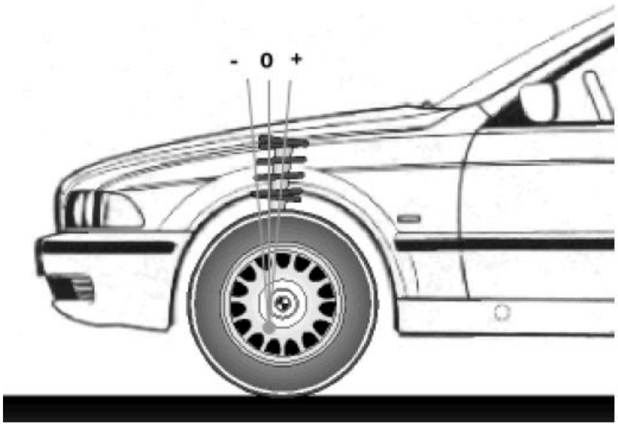


Fig 9:Caster angle

3.2.7 Sprung and Unsprung mass

Sprung mass is the mass of anything that is carried by the car's Suspension. This includes the chassis, engine , transmission, electronics ,interior components and driver. Sprung mass moves up and down as the vehicle travels over surfaces.

Unsprung mass is the mass that is not supported by the vehicle's suspension. This includes the wheels, tires, brakes ,wheel hubs and bearings, uprights ,axle shafts and differential.

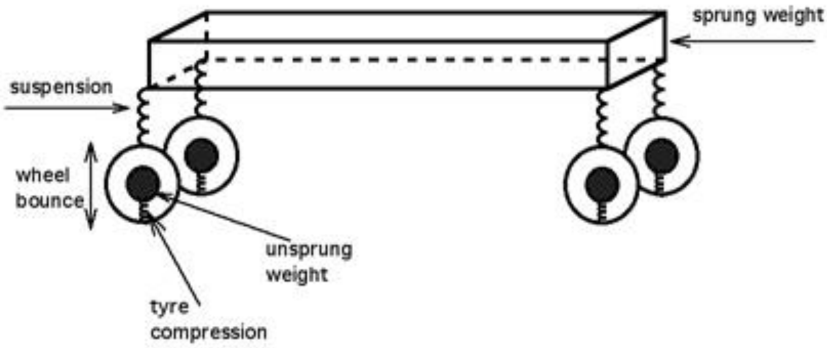


Fig 10: Sprung and Unsprung weight(https://en.wikipedia.org/wiki/Unsprung_mass)

3.2.8 Torsional Stiffness

Torsional Stiffness is determined by the amount of Torque required to deflect the chassis by unit degree. The differences in forces from the suspension links that leads to a Twisting Moment. The moment will deflect the chassis about its Roll Axis. In a perfectly rigid chassis, the front and rear roll angles remain identical, as assumed in suspension design calculations. When the chassis allows for twist, weight transfer deviates from intended values, affecting handling predictability. To minimize these effects, race cars typically aim for a chassis stiffness 3-5 times greater than suspension roll stiffness, while passenger cars range from 7-40 times. Despite these guidelines, there is no standardized value for optimal torsional stiffness in vehicle design.

3.2.9 Roll Centers and roll axis

Roll center is a theoretical point in the front view of a vehicle which the body rolls around it and it is determined by the design of the Suspension. The line connecting front and rear axis roll center is the roll axis. The roll center is the notational point which the cornering forces in the suspension react to the vehicles body.

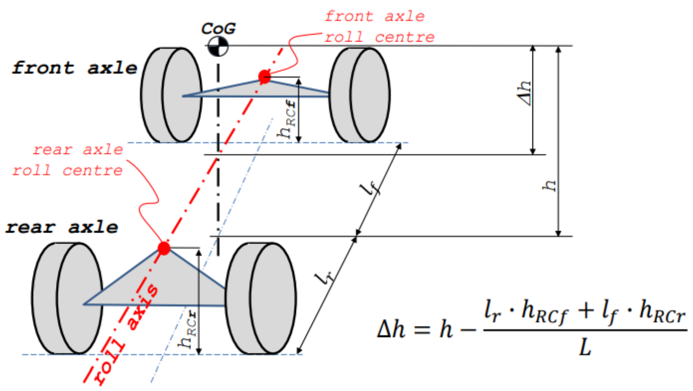


Fig 11: Roll axis for a two axle vehicle(Vehicle Dynamics Compendium for course MMF062 Chalmers Edition)

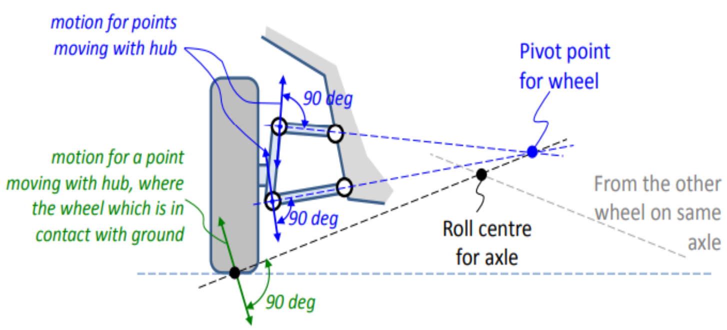


Fig 12: Example of how to appoint the pivot point for one wheel and roll centre for axle with double wishbone suspension.(Vehicle Dynamics Compendium for course MMF062 Chalmers Edition)

3.2.10 Ride Rate or Ride Frequency

Suspensions frequencies are the rate that the spring oscillates after applying a load or hitting a bump. A suspension with stiff springs and a high frequency will oscillate more quickly than the same car with soft springs. This does not include the effects of dampers. Lower frequencies produce a softer suspension with more mechanical grip, however the response will be slower in transient while higher frequencies create less suspension travel for a given track, allowing lower ride heights, and in turn, lowering the center of gravity.

0.5 - 1.5 Hz for passenger cars

1.5 - 2.0 Hz for sedan racecars and moderate downforce formula cars

3.0 - 5.0+ Hz for high downforce racecars

4. FSTUC 2023 VIRTUAL MODEL

A virtual model of the front suspension of the 2023 car was needed, so we tried to copy the geometry of the real car and design each component in SolidWorks. This can be done by taking physical measurements of the car and 3D scanning. We decided through testing that the rear suspension system was not working properly, so we redesigned it for the FSTUC 2024 car.

On Fig 12 we can see the CAD model of the front suspension that was used on the previous car. It is a Push rod double wishbone suspension that is used commonly on formula student cars. The Suspension rod and the wishbones were created through physical measurements. Also, the wheel hub model designed using hand calculations. Tire model was a simple model with the correct dimensions, and the upright is the real model that was manufactured.

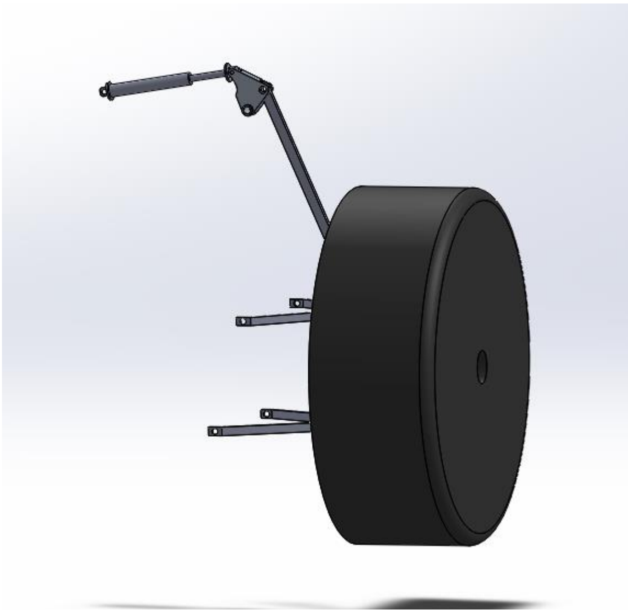


Fig 13: CAD model of Front Suspension

Fig 14 : Detailed view of the Rocker Arm

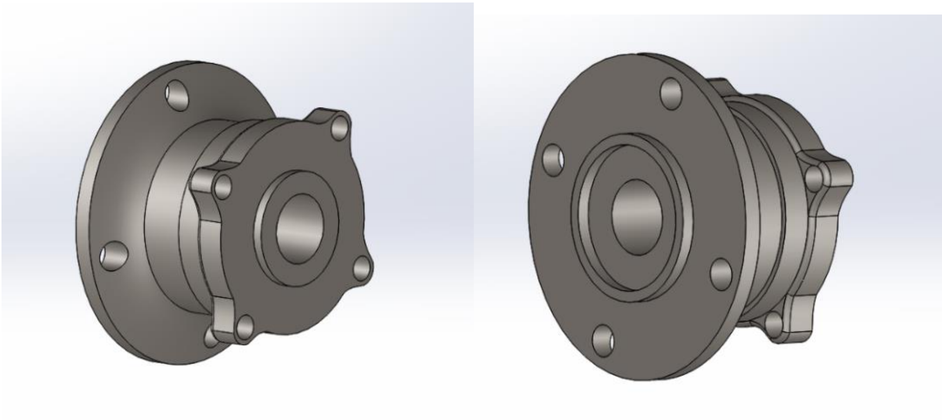


Fig 15: Wheel hub from smart 541

After the creation of the virtual model, we measure the displacement on the wheel and on the damper at the vehicle, then we measure the displacements on the Cad design. The main idea is to compare the two different Motion Ratios and compare them with the hand calculations.

4.1 Measurements on the real vehicle

In order to measure the vertical displacement of the wheel we attached a laser measurement, while lifting the wheel with a small crane. In every measurement of the wheel, we also measure the displacement of the damper with a steel vernier caliper.

Table 1:Measurements on the Real Vehicle

Spring(mm)	Wheel(mm)
0	0
2	5
6	11
10	16
14.6	21
21	27
24.6	33
29.8	38
34.5	43
37	47

After the measurements the Motion Ratio (Table 2) is Calculated for each measurement and then an average motion ratio

$$Motion\ Ratio = \frac{Spring\ Displacement}{Wheel\ Displacement}$$

Equation 1:Motion Ratio

0.4
0.545
0.625
0.695
0.777
0.745
0.784
0.802
0.787

Table 2:Motion Ratio

The average Motion Ratio is 0.67.That means that for 100 mm the wheel moves the spring/damper-moves 67 mm.

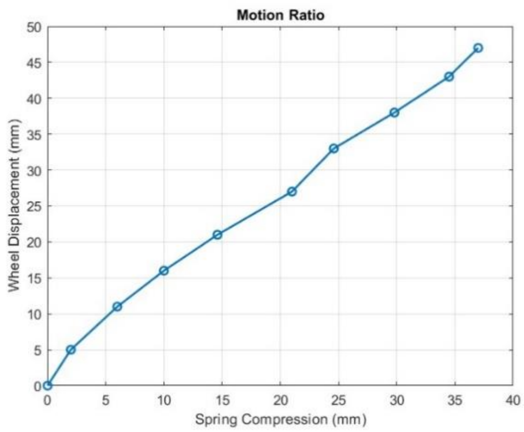


Fig 16 :Motion Ratio

Motion Ratio can also calculated by the Equation 2 below after we measure the following dimensions,L1,L2,L3,L4.

$$Motion\ Ratio = \frac{L2}{L1} * \frac{L3}{L4} = \frac{167}{355} * \frac{65}{43} = 0.71$$

Equation 2:Motion Ratio Calculation

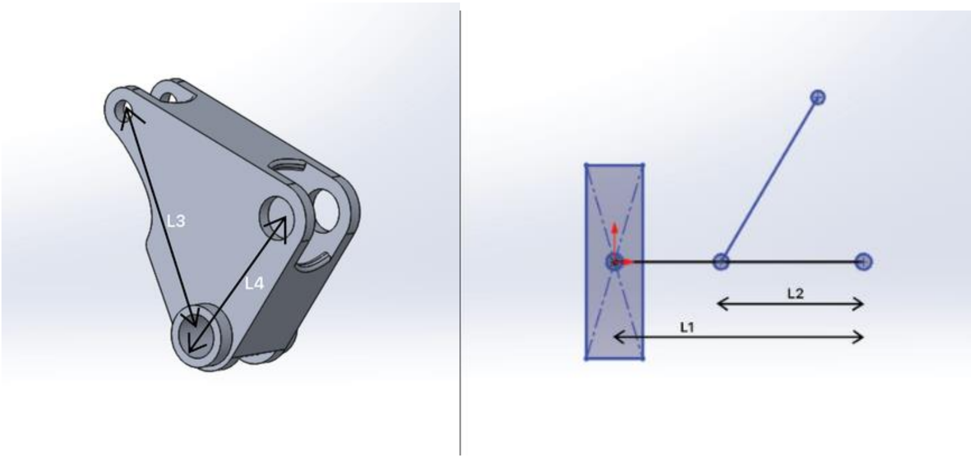


Fig 17 :Dimensions needed for the calculation

4.2 CAD Design and Calculations

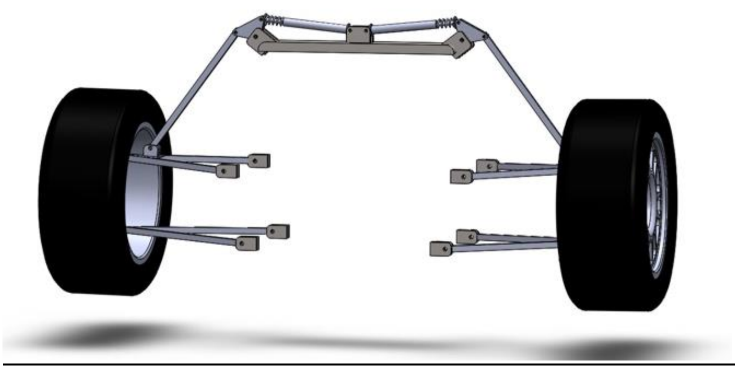


Fig 18: Cad Model of the Front Suspension

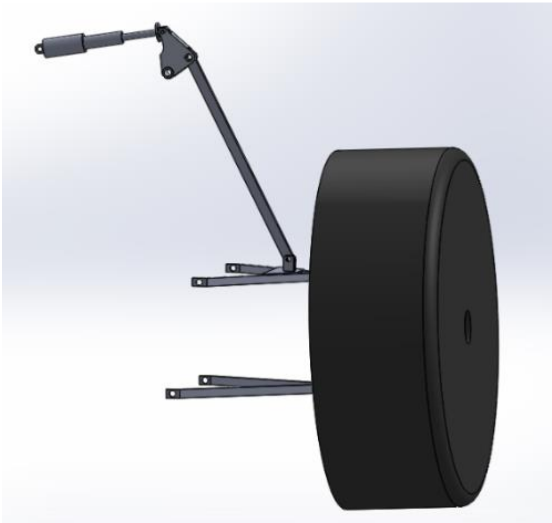


Fig 19 : Cad Model of the Front Suspension

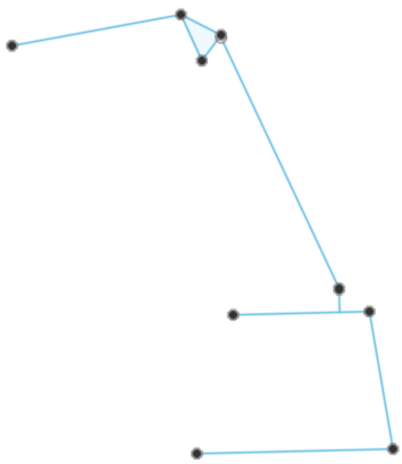


Figure 20: Suspension Geometry of Front Suspension

Using SolidWorks motion study, we created a motion simulation of the tire on the vertical axis. We need to ensure correct mates to have motion study without errors. After that two sensors were added on SolidWorks to measure the spring displacement and the wheel displacement. The data was extracted from SolidWorks for post process analysis.

Table 3:Measurements on the Cad Model

Spring mm	Wheel mm
0	0
3.52	3.98
7.28	8.18
11.19	12.48
14.4	15.97
18.19	20.05
23.96	26.2
30.51	33.12

After the measurements the Motion Ratio for each pair of values was calculated

0,884422
0,889976
0,896635
0,901691
0,907232
0,914504
0,921196

Table 4:Motion Ratio

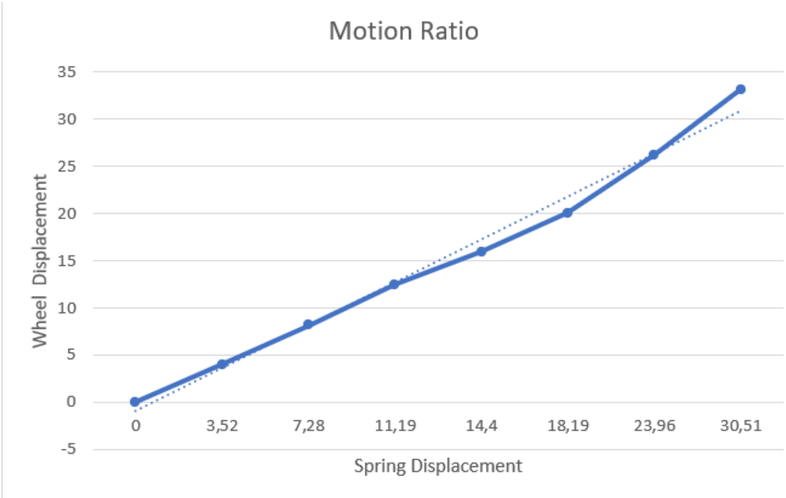


Fig 21 : Motion Ratio

The average Motion Ratio is 0.89.

After the calculations of both real vehicle's Motion Ratio and Cad model's Motion Ratio we could see that there is a difference between the 2 values. We can perform more intensive modifications to the virtual model but we decided that we have enough data from the performance of the suspension. A matlab script was used in order to plot these values on a same graph(Fig 5).

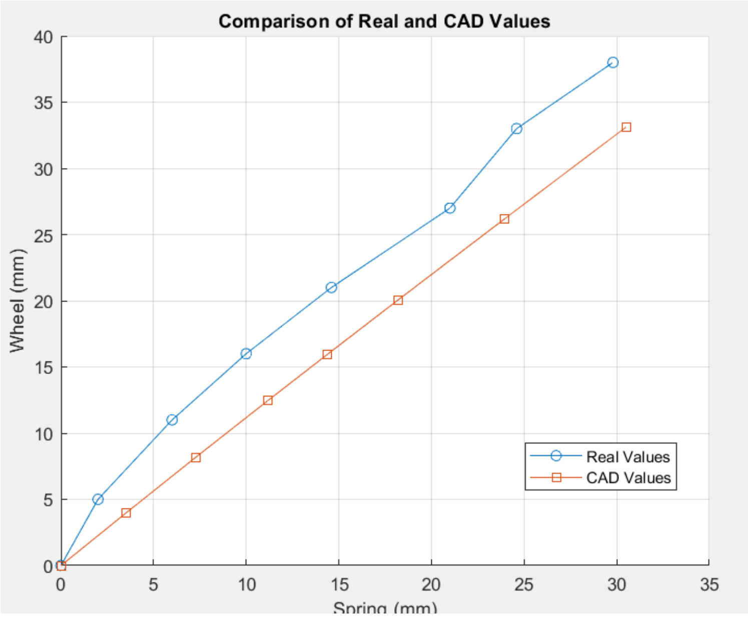


Fig 22: Comparison between the Real values and Cad values

4.3 Load Distribution

Vehicle Weight (Without driver):300 kg

Driver’s Weight:70 kg

Vehicle weight with Driver :370 kg

Assuming a 45:55 weight distribution ,45% on the front and 55% on the rear of the vehicle.

Weight at front: $0.45 * 370 = 166.5\ kg$

Weight on each tire:$\frac{166.5}{2} = 83.25\ kg$

Weight at the rear:$0.55 * 370 = 203.5\ kg$

Weight on each tire=$\frac{203.5}{2} = 101.75\ kg$

Wheelbase:1600 mm

Track Width [Front]=1250mm

Track Width[Rear]=1280 mm

4.4 Load Calculations

Longitudinal Forces during braking

Considering Max. Acceleration of 1g: $9.81\ m/s^2$

Force at the Front side=mass at the rear side*acceleration

Mass at the rear=203.5 kg

Force= $203.5 * 9.81 = 1996.33\ N$

Force on 1 Wheel= $1996.2 = 998.1\ N$

Longitudinal Force = 998.1 N

Lateral Forces

Lateral Forces are because of 2 reasons. Centrifugal Force and lateral load transfer from outside to inside while turning.

Centrifugal Force

Assuming the vehicle takes a turn of 7m radius with a speed of 35 kph.

R=7m

V=35kph=9.72 m/s

$$Centrifugal\ Force = \frac{M * v^2}{R} = 4993.85\ N$$

Centrifugal acceleration=$13.49\ \mathrm{m/s^2}$

Now consider if all the weight at the front side comes on the wheel assembly. The force will be Force due to lateral load transfer $= 0.45 * 370 * 9.81 = 1633.36\ N$

4.5 Suspension Characteristics

Motion Ratio: 0.70

Kspring: 50 N/mm

$$K = (MR^2) * Kspring = 22528\ N/m$$

$$\omega = \sqrt{\left(\frac{K}{M}\right)} = 22.33\ rad/sec$$

$$Ride\ \ Frequency = \frac{\omega}{2 * \pi} = 3.54\ Hz$$

After examining these results, we concluded that the front suspension system had a motion ratio close to 1, with a value of 0.70 (average motion ratio based on CAD Design and actual measurements). This means that for every 100 mm of vertical movement, the spring compresses by 70 mm. This means that half of the spring's stiffness is observed in the wheel. We also had a higher frequency, which is not desirable according to the literature. We decided to try a direct mount suspension for the new front suspension this year. Another significant problem in the suspension assembly was the alignment of the spring/damper, rocker arm and the suspension rod, with the result that not all the force was applied on the rod to compress the spring and, therefore, differences between the actual measurements and the CAD values.

5. FSTUC'S 2024 CAR DEVELOPMENT

The first thing was to define basic attributes such us the Wheelbase (greater or equal 1555 mm from the competition regulations) and front and rear wheel track. Thus, the wheelbase of the Car is 1600 mm and front wheel track is 1210 and rear wheel track is 1250 mm. The size of the rims we decided to use was 13 inch Aluminum OZ racing.

Firstly, the Chassis designed on SolidWorks in order to follow the regulations of the competition. The chassis was designed to maintain a certain minimum distance from the road (Min 30 mm) from the regulations of the competition. After that an assembly was created with many components to design the whole virtual model of the car.

The rear trackwidth was determined by using the same axle shafts as last year. After that, the design of many components started in order to manufacture them like the Uprights and wishbones which are main part of the suspension system.

5.1 Targets

Weight:~270 Kg

Center of gravity is as close as possible to the ground.

5.2 Weight distribution

Assuming a weight distribution of 55%-45% where 55% on the rear and total weight of the vehicle with the driver is 300 kg(Net weight:230 kg ,driver's weight :70 kg)

Mass at front:$0.45 * 300 = 135\ kg$

Mass on each front tire:$\frac{135}{2} = 67.5\ kg$

Mass at the rear:$0.55 * 300 = 165\ kg$

Mass on each rear tire=$\frac{165}{2} = 82.5\ kg$

Wheelbase:1600 mm

Trackwidth Rear:1250 mm

Trackwidth Front:~1210mm

	Weight		
no	Item	Mass(kg)	Weight(N)
1	Driver	70	700
2	Engine	48	480
3	Chassis	50	500
4	Front Suspension	2	20
5	Rear Suspension	4	40
6	Fuel Tank	10	100
7	Differential Assembly	8	80
8	Driveshafts	7	70
9	Bodywork	1	10
10	Steering Column-rack	15	150
11	Cockpit(Seat-dashboard-shifter)	5	50
12	Exhaust	5	50
13	Front Wheel assembly(Wishbones Hub-Uprights-Wheels)	25	250
14	Rear Wheel assembly(Wishbones Hub-Uprights-Wheels)	27	270
15	Intake Components	2	20
	Total	279	2790

Fig 23 : Vehicle and components weight

5.2.1 Load calculations

Longitudinal Forces during braking

Considering Max. Acceleration of 1g: $9.81\ m/s^2$

Force at the Front side=mass at the rear side*acceleration

Mass at the rear=165 kg

Force= $165 * 9.81 = 1618.65\ N$

Force on 1 Wheel= $\frac{1618.65}{2} = 809.325\ N$

Longitudinal Force = 556.525N

Lateral Forces

Lateral Forces are because of 2 reasons. Centrifugal Force and lateral load transfer from outside to inside while turning.

Centrifugal Force

Assuming the vehicle takes a turn of 7m radius with a speed of 35 kph.

R=7m

V=35kph=9.72 m/s

$$Centrifugal\ Force = \frac{M * v^2}{R} = 4049.07\ N$$

Centrifugal acceleration=$13.5\ \mathrm{m/s^2}$ or 1.38 g.

Now consider if all the weight at the front side comes on the wheel assembly. The force will be Force due to lateral load transfer $= 0.55 * 300 * 9.81 = 1618.65\ N$

5.3 CHASSIS DESIGN

The chassis was designed within the competition regulations. Front Hoop and Roll Hoop tubes have an outer diameter of 25 mm and wall thickness of 2.5 mm. All the other tubes have an outer diameter of 25 mm and a wall thickness of 2 mm. It was made of AISI4130 also known as chromyl steel. The mechanical properties of the AISI4130 are listed below

Table 5:AISI Steel Properties

Properties	Metric
Tensile Strenght, Ultimate	760 MPA
Tensile Strength, Yield	435 MPA
Modulus of Elasticity	205 GPA
Bulk Modulus	140 GPA
Poisson's Ratio	0.29

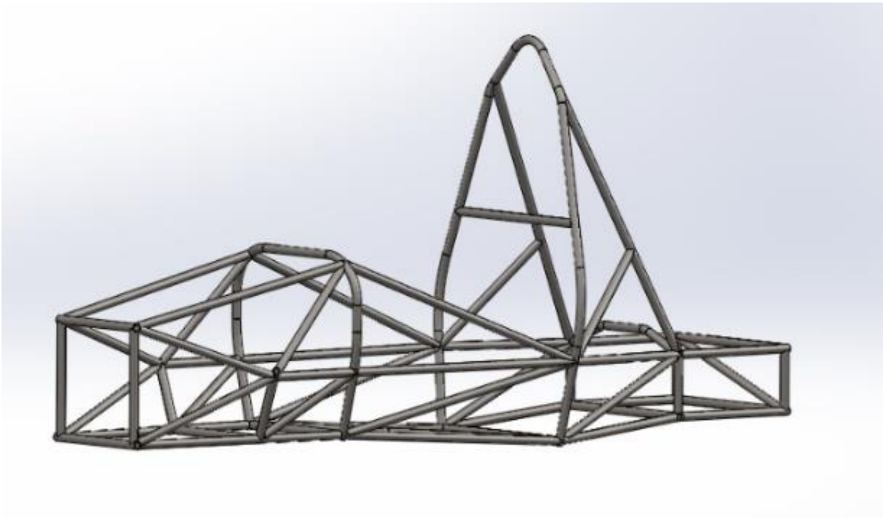
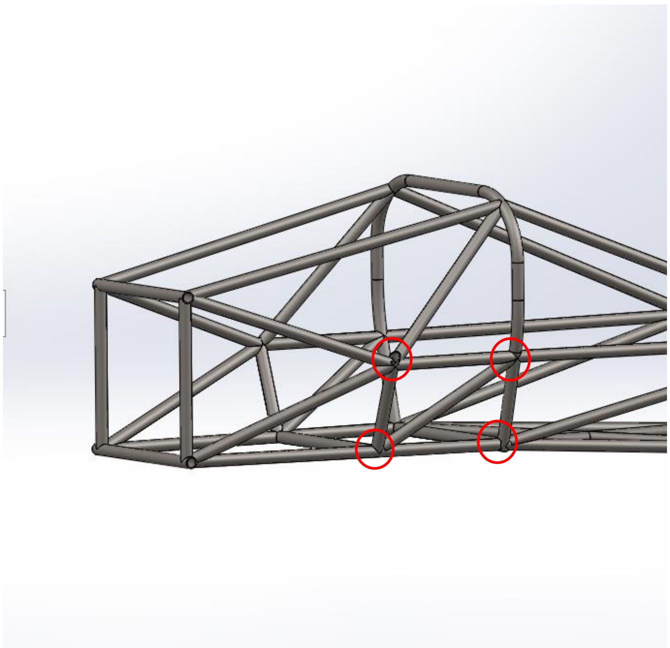


Fig 23 :Chassis

The total mass of the chassis from SolidWorks was 37 Kg.

Below we can see the hardpoints where the wishbones will be attached to the chassis both on front and rear.



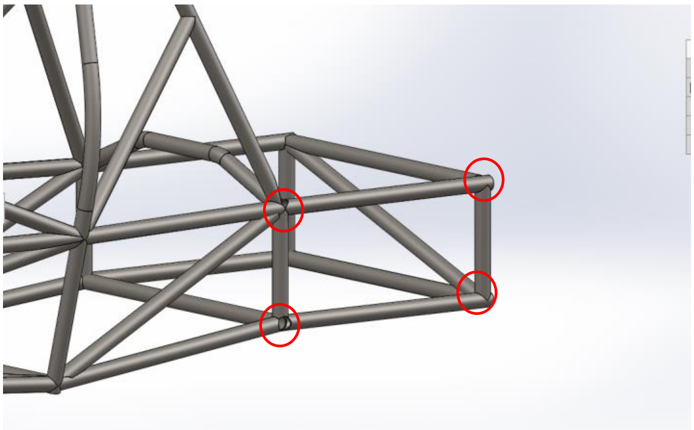


Fig 24:Front and rear hardpoints(Wishbone mounts)

The chassis was tested at three simulations through SolidWorks. A front impact, a side impact and a torsional analysis.

5.3.1 Front Impact

The front impact is done to find out the amount of bending stress and the deformation in the chassis if the car hits a solid body from the front. The rear was set fixed and the force were applied at the front bulkhead nodes. The Force was calculated on a crash of 5G.

Parameter	Value
Car weight with driver	310 Kg
Velocity	70 Km/h
Force	14000 N

Table 6: Parameters Front Impact

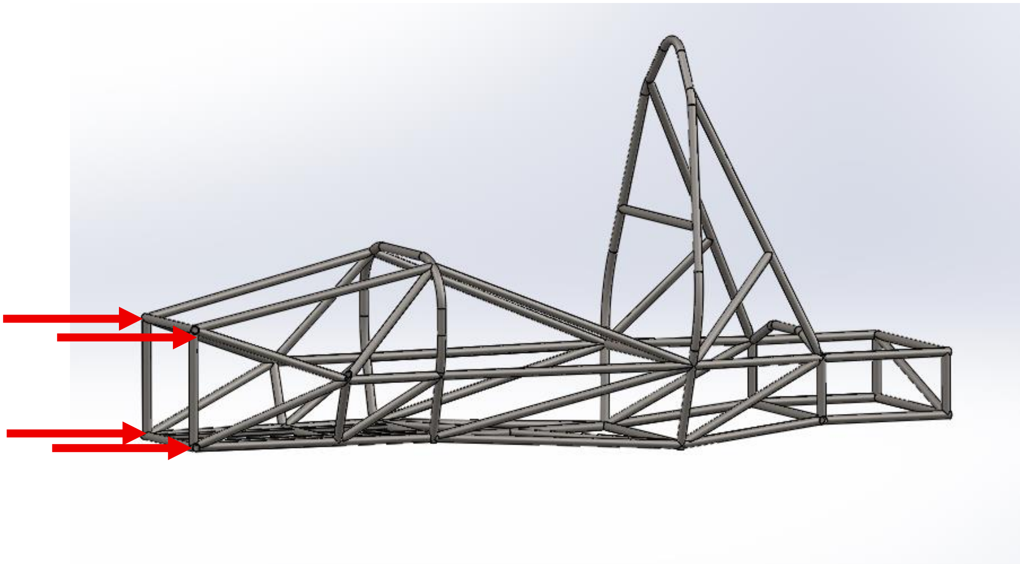


Fig 25: Points of application of force

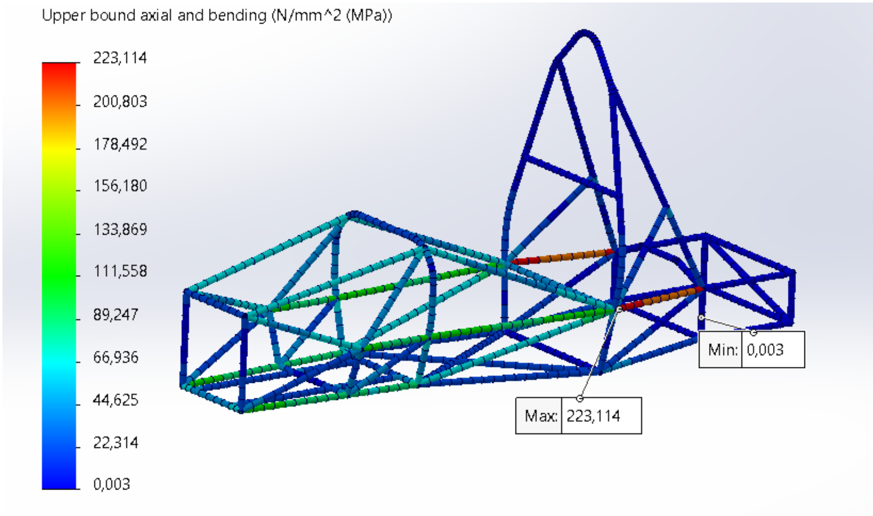


Fig.26 :Von Misses Stress Analysis, Front Impact

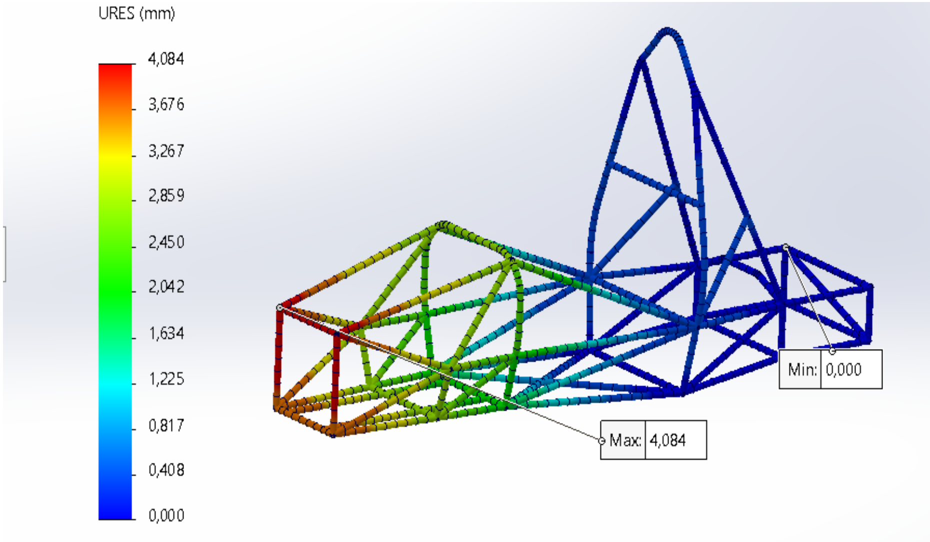


Fig 27 :Deformation ,Front Impact

Von Misses	223.11 MPA
Defor-mation	4,084 mm

Table 7:Results Front Impact

5.3.2 Side Impact

Assuming a velocity of 50 Km/h. One side of the side impact structure is fixed and the on the other side load was applied. The force of 10000 N was applied at the side impact structure

Parameter	Value
Car weight with driver	310 Kg
Velocity	50 Km/h

Table 8:Parameters Side Impact

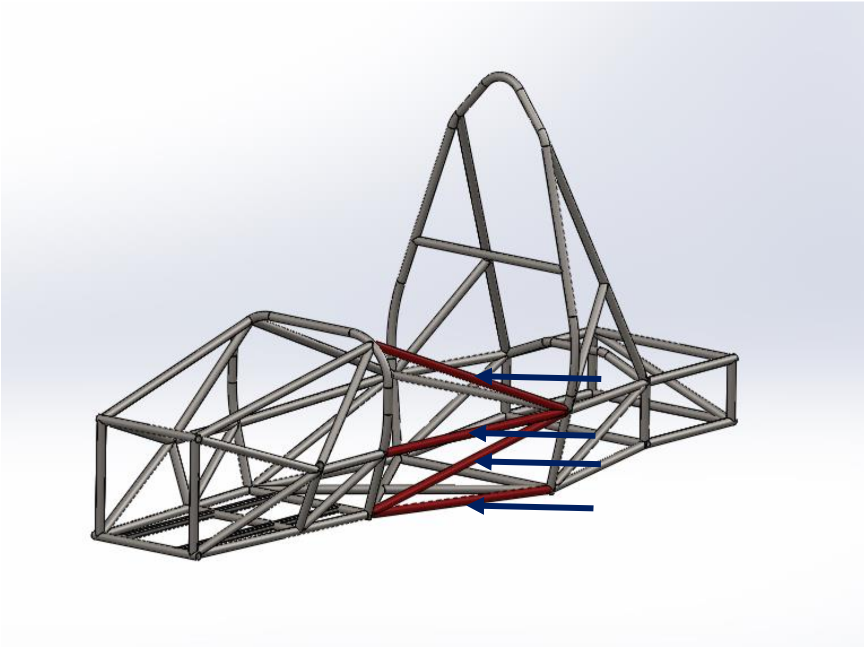


Fig 28 :Points of application of force

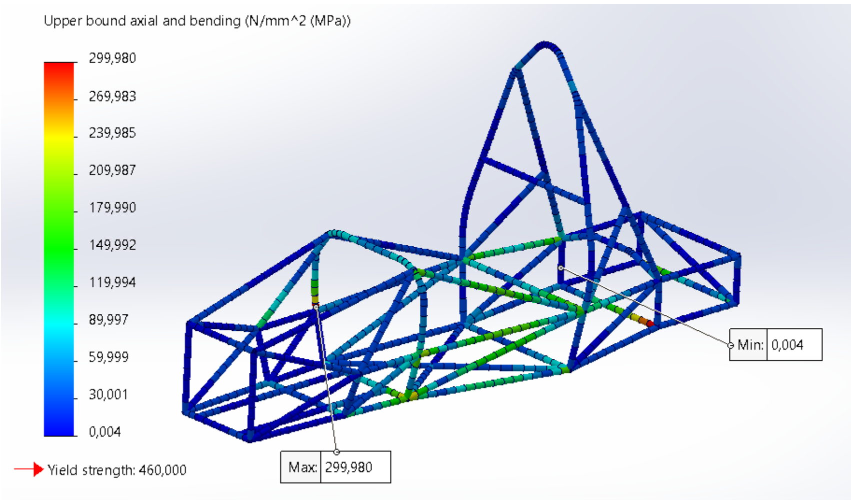


Fig 29: Von misses Stress Side impact

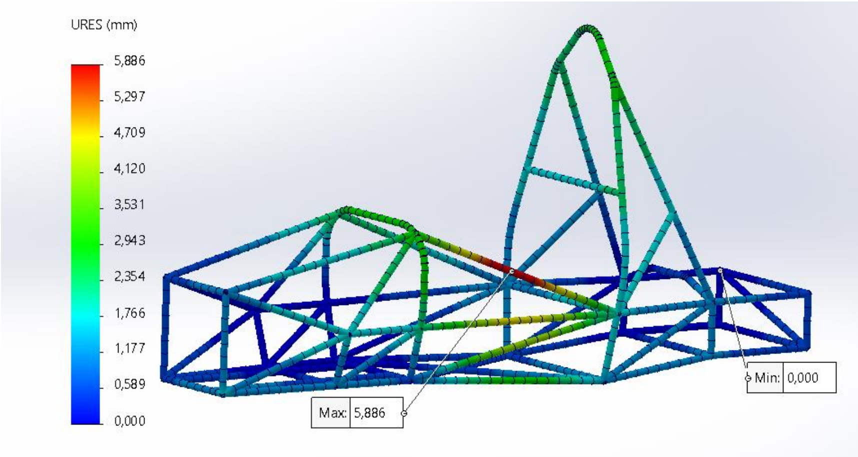


Fig 30: Deformation Side Impact

Von Misses	299.9 MPA
Defor-mation	5.886 mm

Table 9: Results Side Impact

5.3.3 Torsional Stiffness

In order to check the torsional stiffness of the chassis a simulation model was made. The rear suspension nodes of the chassis were set as fixed well 2 forces were added at the wishbone-upright simplified assembly demonstrating the appliable force on the real car and the torque produced based on the distance from the point where the force occurs to the Roll axis of the car. The Force used was 2000 N.

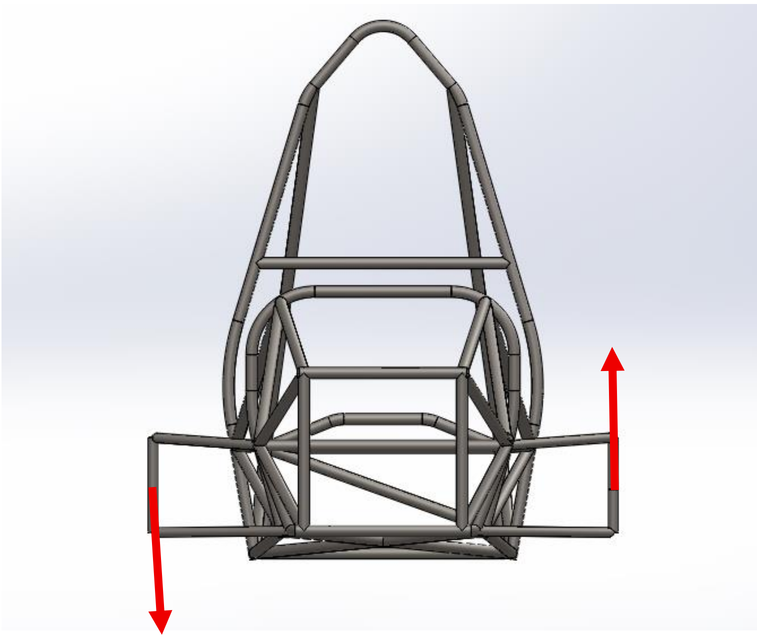


Fig 31 : Points of application of force

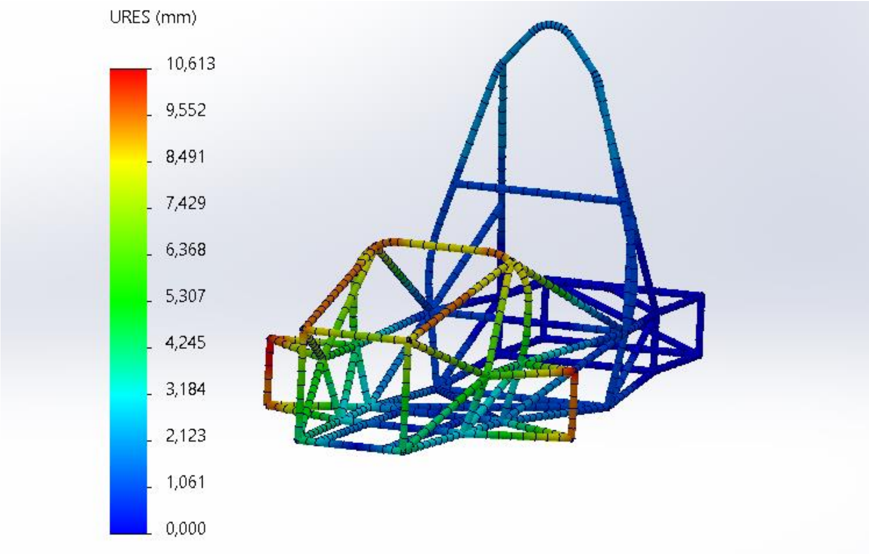


Fig 32 :Deformation Torsional Analysis

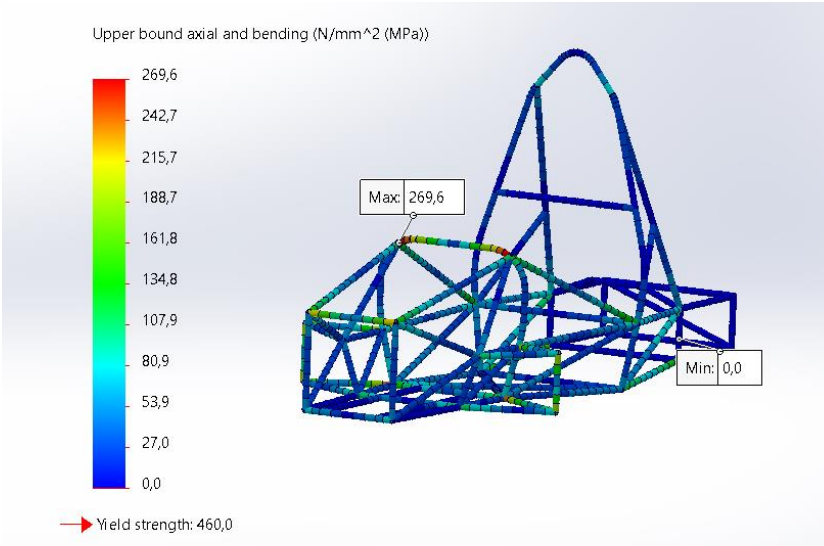


Fig 33:Von Misses Stress Torsional Analysis

From the simulation we can get the linear deflection, and we calculate the angle of twist.

Calculations

- $Torsional\ Stiffness = \frac{Torque\ Load}{Angular\ Deflection}$

  $Torque\ Load = 2000 * \frac{1}{2} * 1.2 = 1200\ Nm$
- $\theta = Angular\ deflection = \tan^{-1}\left(\frac{vertical\ displacement}{\frac{1}{2}trackwidth}\right) = \tan^{-1}(0.017683) = 1.01\ deg$
- $Torsional\ Stiffness = \frac{1200}{1.01} = 1188\ Nm/deg$

Von Misses Stress	269.6 MPA
Deformation	10.61mm
Torsional Stiffness	1188 Nm/deg

Table 10:Results Torsional Stiffness

5.4 Front Suspension Design

The main idea for the Front Suspension was to mount the Suspension directly from the chassis to the lower wishbone, without using extra suspension rod and rocker arm. This was decided due to time restrictions and the need to create a simple system as soon as possible.

5.4.1 Design Targets

Ride Frequency of 2 Hz

Static Camber: Negative 1 degree

Roll Center as close as possible to the ground

5.4.2 Methodology

After we set the first target, that was the ride frequency, we need to calculate the motion Ratio that satisfies the 2 Hz including also the springs. The motion ratio was calculated to be 0.50.

In order to select the Hardpoint on where the suspension should be mounted on the lower wishbone we had to check 3 positions. One at the nearest point, one at the middle of the wishbone, and one as close as possible to the knuckle. After that we calculate the motion ratio for each setup and the camber at full compression. The methodology was simple. First, we created a simple dynamic Cad Model of the front Suspension, then to change the setup we simply change the mate that connects the Suspension to lower wishbone arm. Three sensors were added in order to calculate each measure, one measures the vertical distance of the wheel center from the road, one the damper displacement, and one the camber angle. On the pictures below we can see the 3 setups that we checked. Then sensors data are exported to an excel file for post-process and then on a Matlab script so we can plot them and perform the calculations.

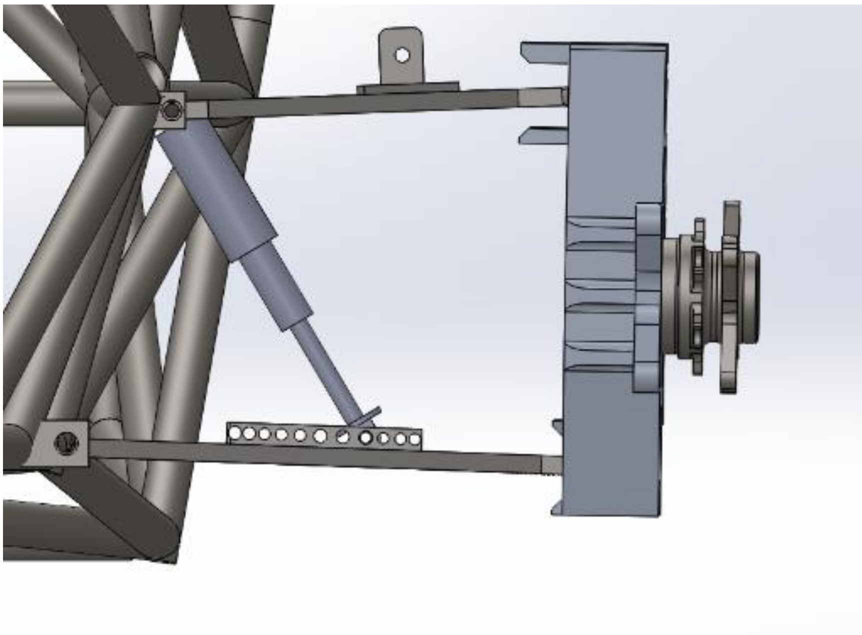


Fig 34 : Setup 1

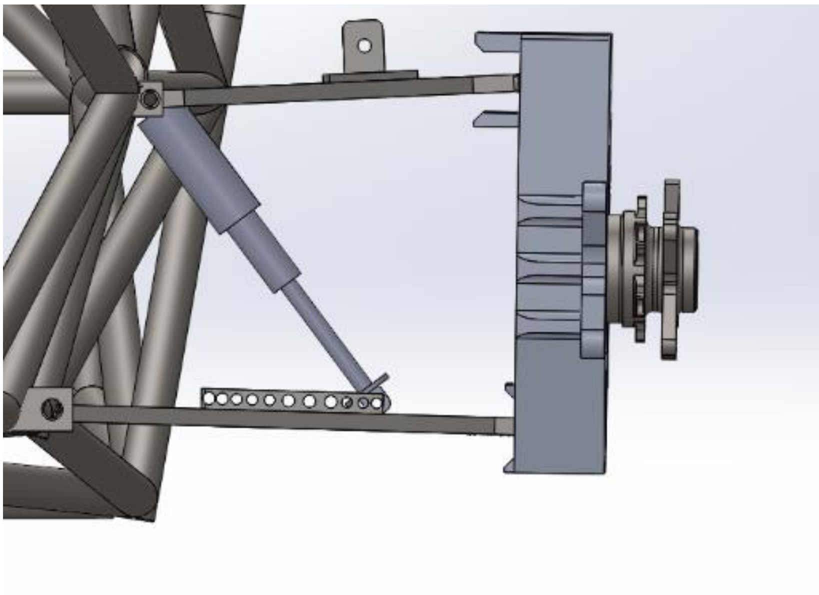


Fig 35: Setup 2

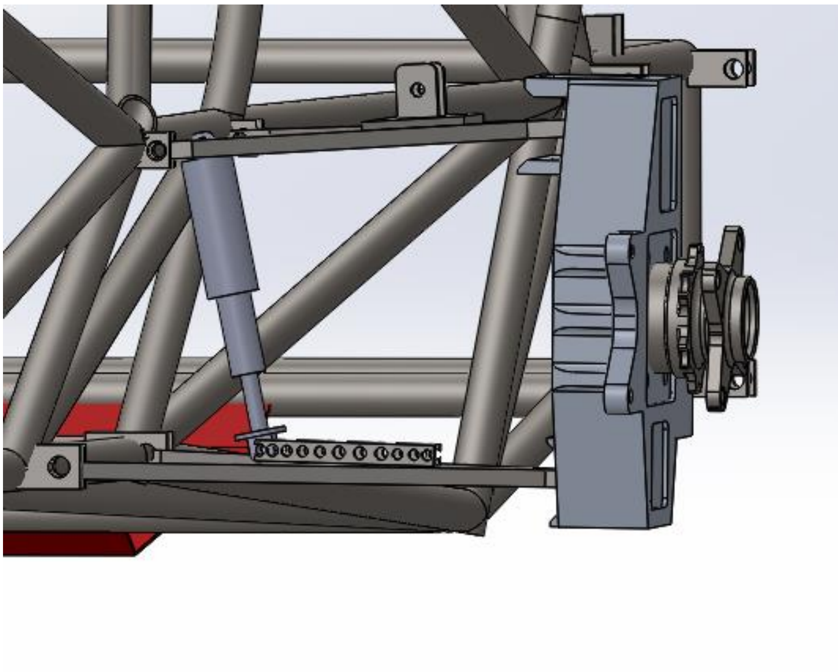


Fig 36 : Setup 3

Below we can see the Motion Ratio for each Setup. As we can see we have an average Motion Ratio of 0.50

```
Motion Ratio for CAD MR 1 Measurements: 0.49005
Motion Ratio for CAD MR 2 Measurements: 0.51783
Motion Ratio for CAD MR 3 Measurements: 0.47675
```

Fig 37 :Motion Ratio for the 3 Setups.

Below we can see the 3 sets of measurements plotted on a same graph. On y axis we have the wheel measurements and on x axis the Spring measurements. We can see that Setup 2(Green) is more linear compared to the other 2 setups.

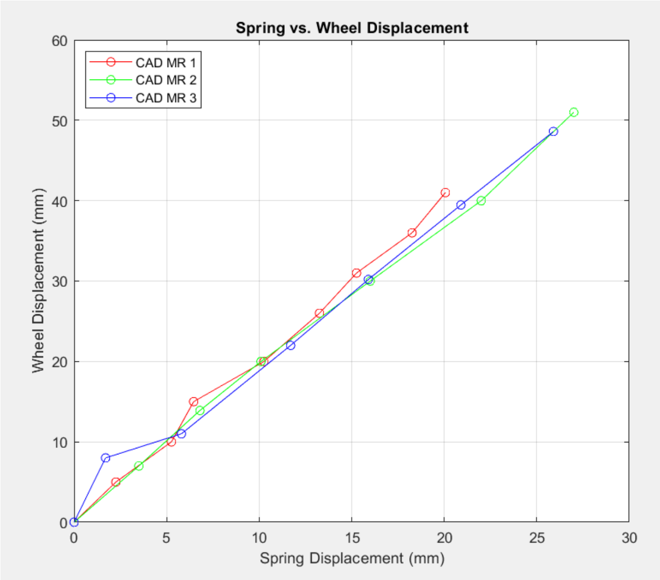


Fig 38: Plot of the 3 Setups Measurements

Also, we compared the camber angle for every setup at full spring compression.

The 3 setups had very similar Motion Ratio's and the 1 and 3 setup required a lot vertical movement of the wheel to achieve full compression. So Setup 2 was selected as it offers a more predictable behavior.

After that a few changes were done at the Cad model in order to satisfy the static camber goal of -1 degree.

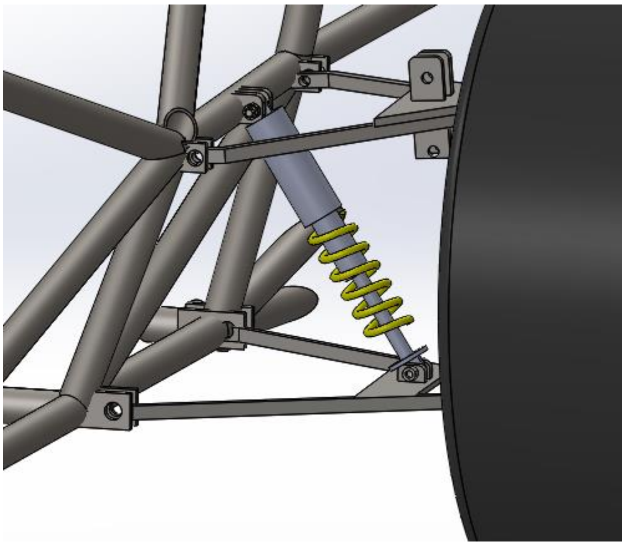


Fig 39: Front Suspension Design through the development phase.

5.4.3 Suspension Characteristics

Motion Ratio :0.48

Static Camber:1.16 degrees negative

Roll Center Height was calculated at 42 mm above the ground

Table 11:Motion Ratio and Camber Angles of the 3 Setups

Setup	Motion Ratio	Camber at full compression	Static Camber
Setup 1	0.49	+1.51	+0.67
Setup 2	0.51	+1.37	+0.6
Setup 3	0.47	+1.18	+0.6

Table 12 :Wheel and Spring Displacement and Camber gain

Spring (mm)	Wheel(mm)	Camber Negative(Degrees)
0	0	1.16
2.37	5.02	1.25
5.87	13.39	1.42
8.79	18.49	1.56
17	3535	2.03
19.54	40.7	2.19
21.35	44.39	2.32

Table 13:Motion Ratio for each pair of values

0.472
0.437
0.475
0.478
0.480
0.480

The average motion ratio is 0.48

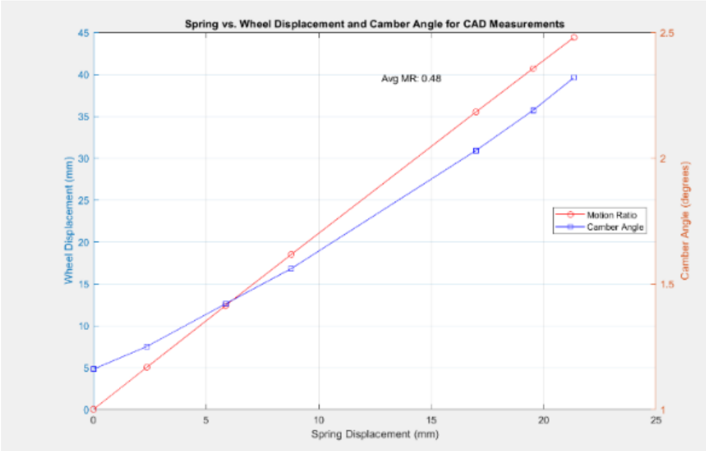


Fig 40 :Motion Ratio and Camber gain

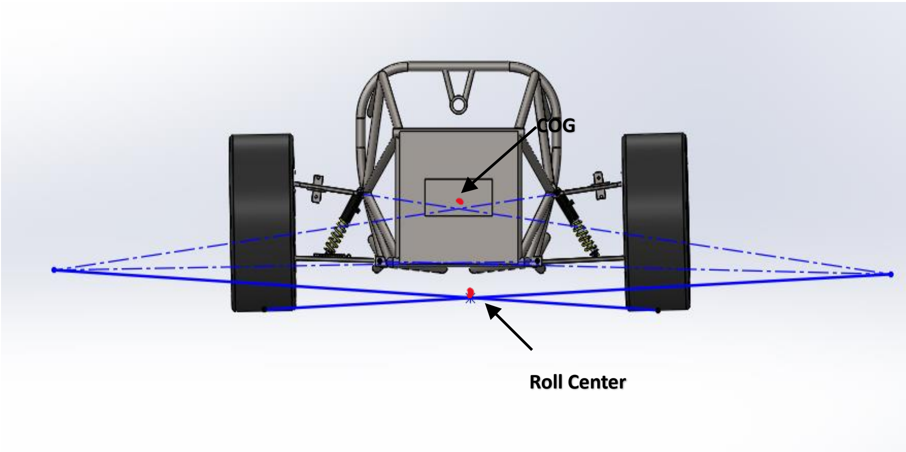
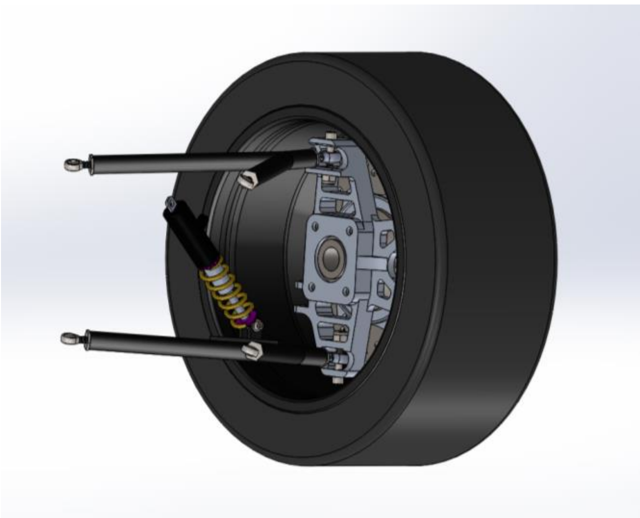


Fig 41 :Vehicle's Front Suspension Roll Center and COG.

Final Products and Exploded view



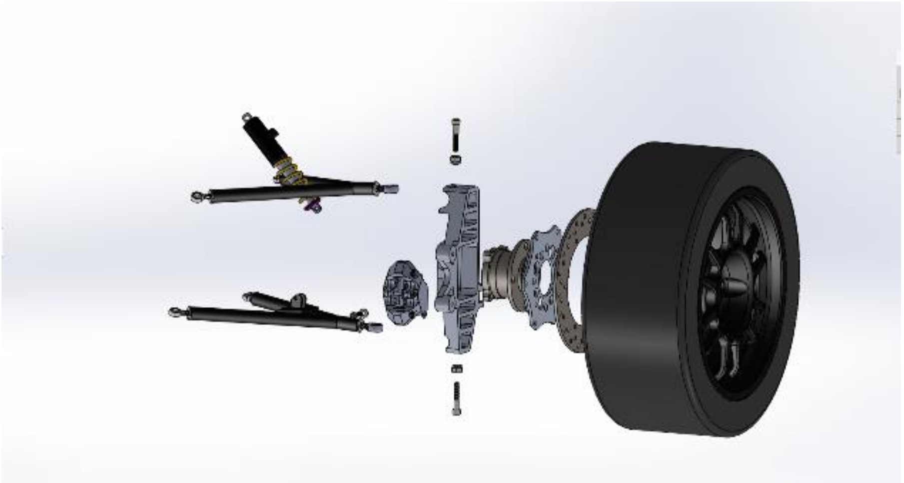


Fig 42 : Final Designs Front Suspension

5.5 Rear Suspension Design

The main idea for the rear suspension was a new design compared to last year's. A push rod double wishbone suspension with rocker arm at the rear of the car and the suspension rod attached at the lower wishbone (Fig 4.6).

The methodology was again the same, setting design goals that satisfy the characteristics of the suspension we want. After that, more interfering modifications to the components would be made and eventually after FEA we would have the whole model.

<u>Design Targets</u>

Ride frequency: 2 Hz

Static Camber: 1degree negative

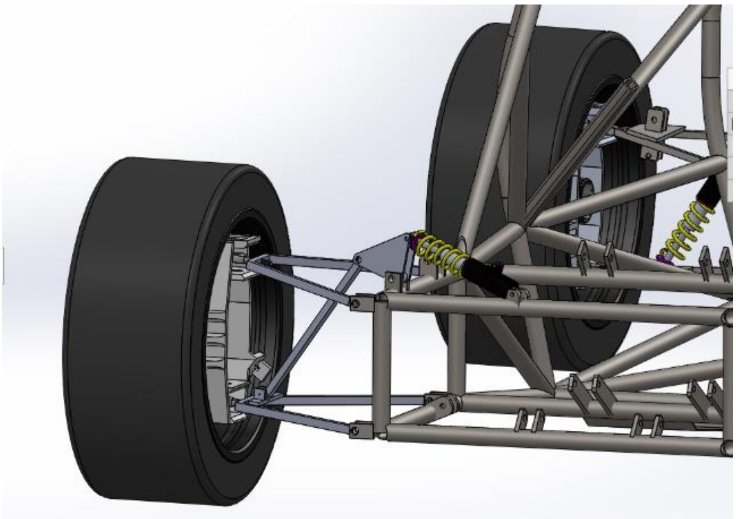


Fig 43:Rear Suspension Design through the development phase.

Table 14: Wheel and Spring Displacement and Camber gain

Spring (mm)	Wheel(mm)	Camber Negative(Degrees)
0	0	1.15
6.17	10.64	1.36
7.74	12.74	1.4
15.33	24.95	1.69
18.61	32.3	1.89
27.19	47.87	2.35

Table 15: Motion Ratio for each pair of values

0.578
0.607
0.614
0.576
0.567

The average motion ratio is 0.59

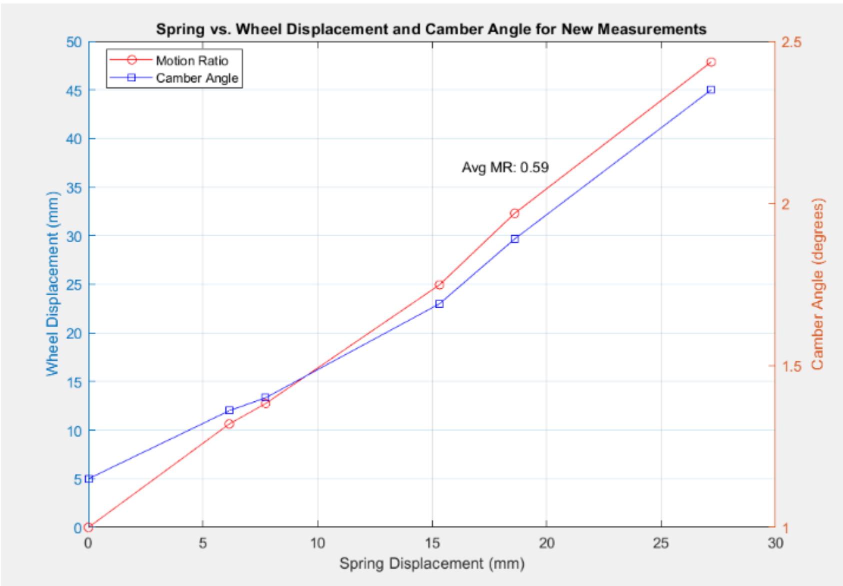


Fig 44:Motion Ratio and Camber gain

Motion Ratio: 0.59

Kspring: 30 N/mm

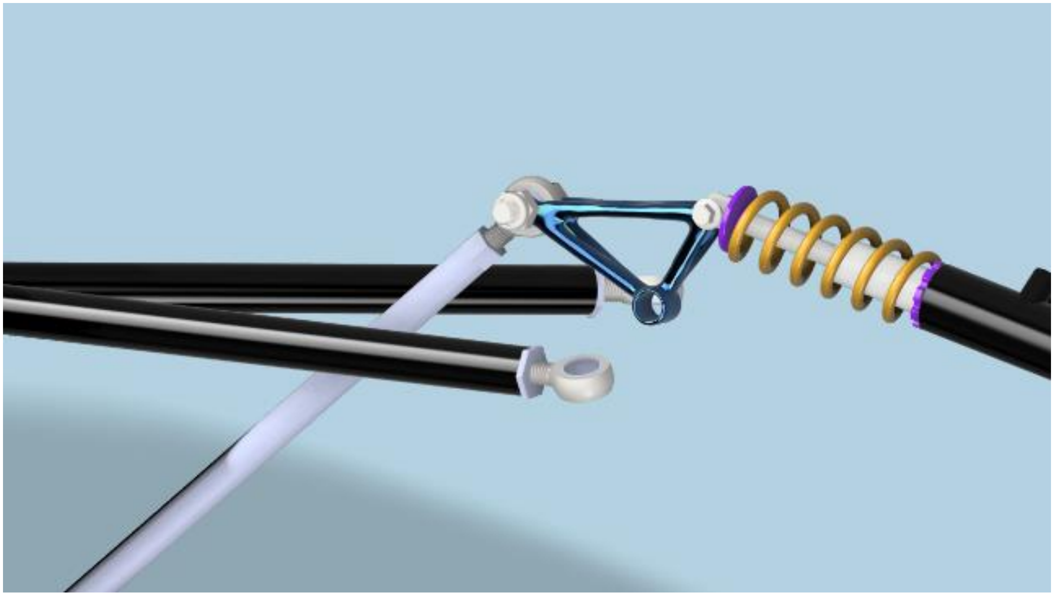
$$K = (MR^2) * Kspring = 10445\ N/m$$

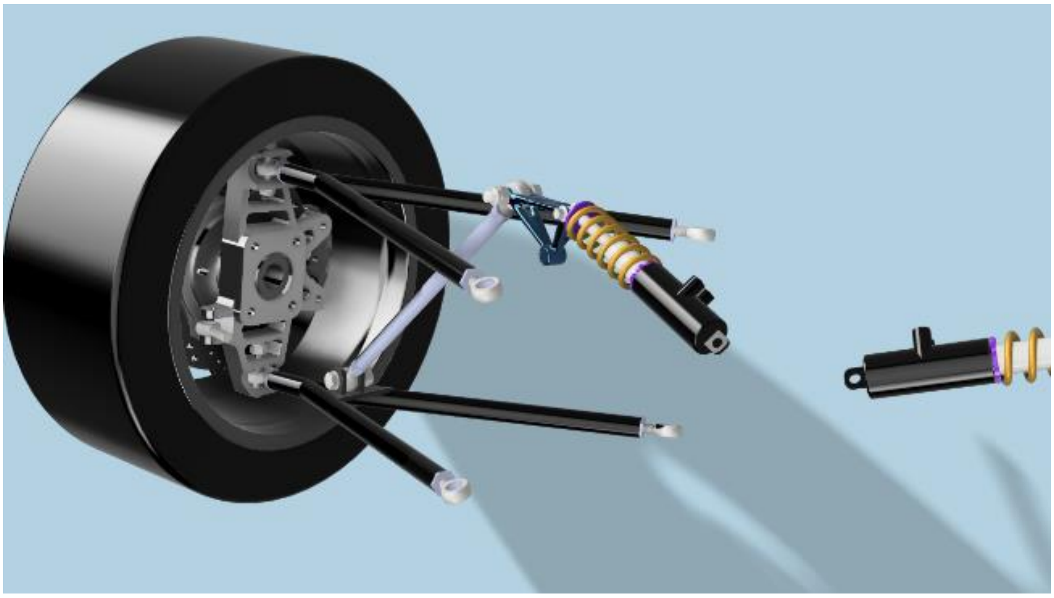
$$\omega = \sqrt{\left(\frac{K}{M}\right)} = 14.45\ rad/sec$$

$$Ride\ \ Frequency = \frac{\omega}{2 * \pi} = 2.29\ Hz$$

The Springs we selected were RF130-36-030 with a total length of 130 mm, inner diameter 36 mm and a Spring constant of 30 N/mm.

The Rear suspension system consists of the upper wishbone, the lower wishbone, the suspension rod, the rocker arm ,the damper/spring element and the Upright-wheel hub assembly.





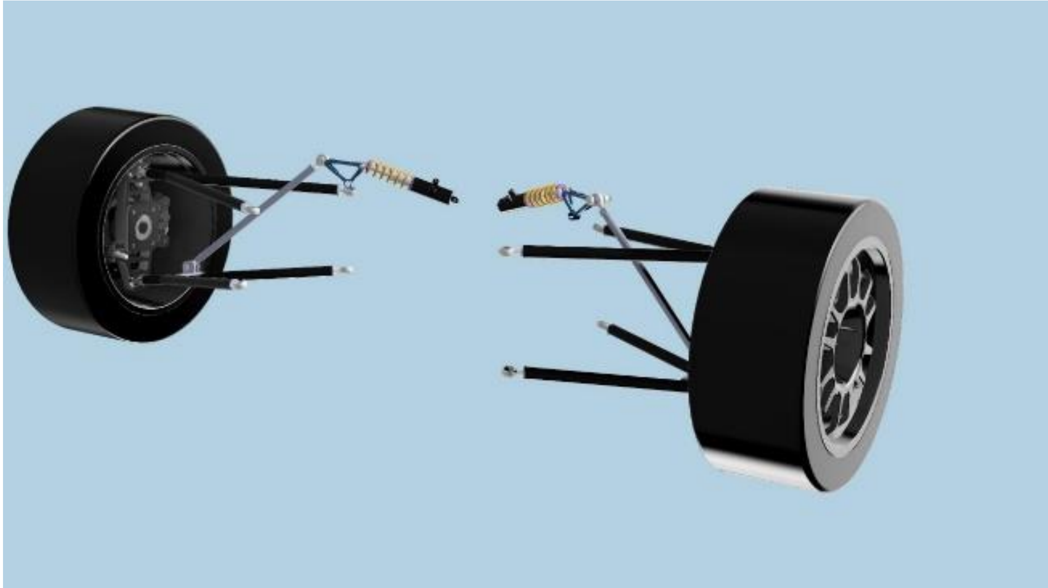


Fig 45:Rear Suspension final design

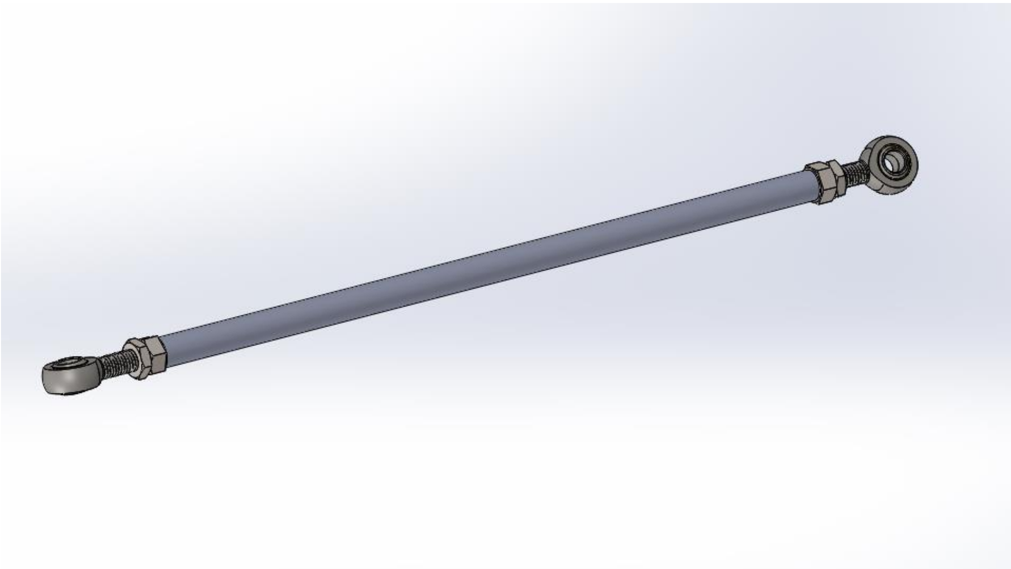


Fig 46: Suspension Rod

Table 16:Rear Suspension Components

Parts	Quantity
10 mm Ball joints	4
Suspension Rod	2
Damper/Spring	2
Rocker Arm	2
Upper Wish-bones	2
Lower Wish-bones	2
Upright	2
Wheelhub	2

10 mm Partially threaded Allen Bolt	2
Hex Nut 10 mm Thin	4
Nut 10 mm	4

5.6 Rocker Arm Design

Main part of the Rear Suspension Assembly is the Rocker Arm. The rocker Arm designed in that way so it can provide the Motion Ratio we need for the Suspension. Taking the weight of the component into consideration, we want to create an optimized part with no excessive material. The part was designed by steel plates 4 mm thick, also support plates were added. The plates will be welded together and weigh 155 gr.(Pic.1).

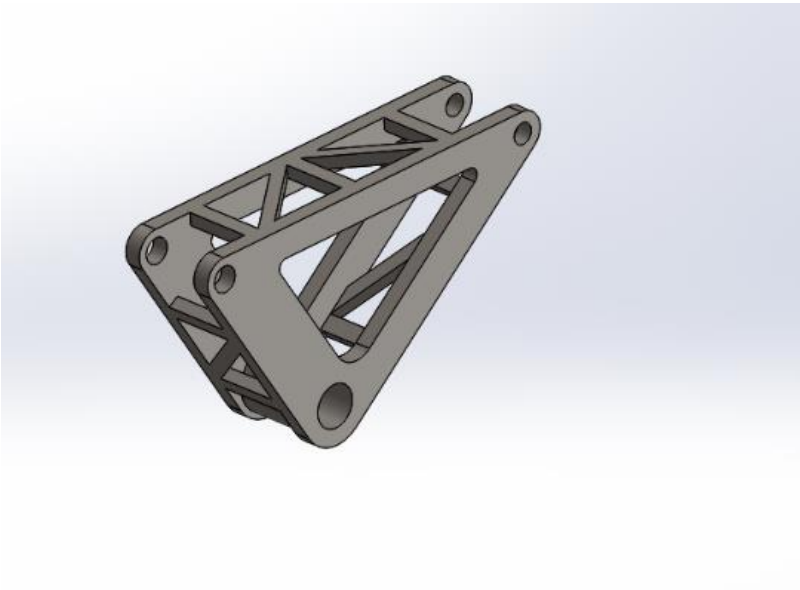


Fig 47 :Rocker Arm

5.6.1 Analysis of the Rocker Arm

Theoretical Calculations were done with a Bump Force of 2400 N that comes from the wheel to the rocker arm through a pushrod.

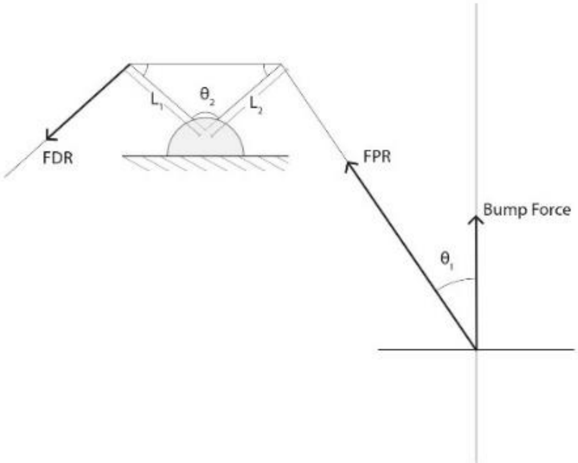


Fig 48:Free Body Diagram of the Suspension

L1=60 mm, L2=70 mm ,Θ1=48°, Θ2=90.5°

Considering a bump condition with a force of 2400 N.

Force at Pushrod end (F_{PR})=$Bump\ Force * cos\Theta 1 = 2400 * \cos(48) = 1605.9\ N$

Rocker Motion Ratio(RMR)=$\left(\frac{L2}{L1}\right) * \sin(\theta 2) = \frac{70}{60} * \sin(90.5) = 0.71$

Force on Rocker Damper end (F_{DR})= $F_{PR} * RMR = 1140.189\ N$

The simulation of the part was done on SolidWorks Software. The values of nodes and elements were found 86068 and 52134 respectively. After performing analysis on the design, it was found that the part has minimum 1.61 value of Factor of Safety and an average factor of safety of 3.

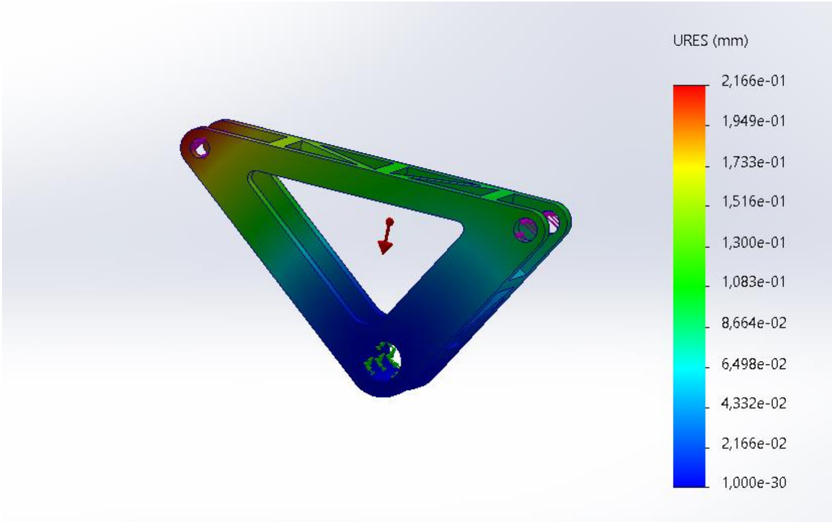


Fig 49: Displacement

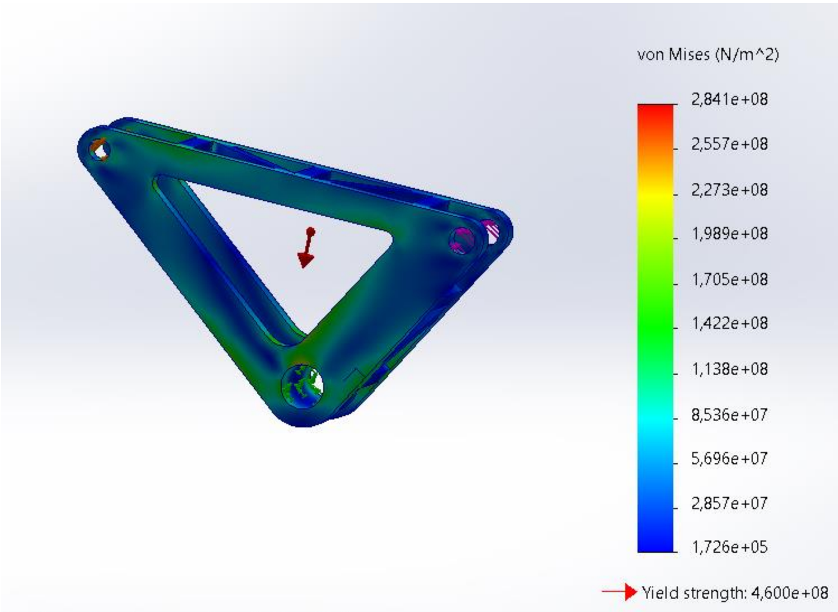


Fig 50: Stress

The second Part (Fig 46) is automated design done on Fusion 360.We designed the three hard-points/constraints, and the software connects them, with various ways. This product will be 3d printed from steel and will weigh about 139 gr and if it is printed by aluminum it will weight 60 gr.

Fig 51:Automated Rocker Arm

5.7 Front and Rear Suspension during Development

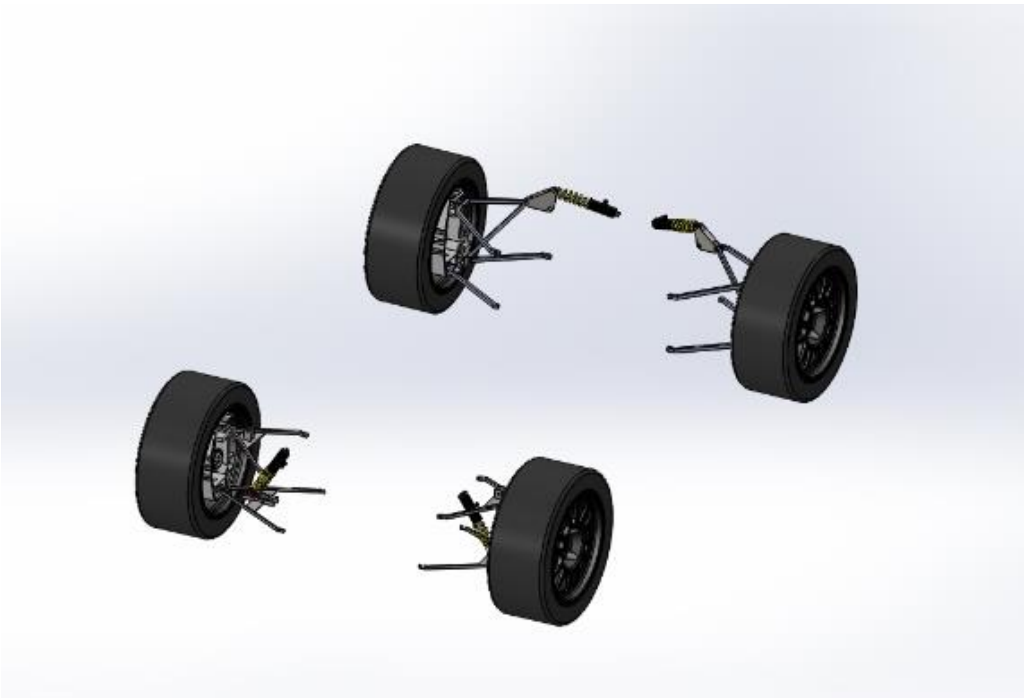


Fig 52:Suspension assembly

6.Conclusion

In conclusion, the team was very satisfied with the job done both on chassis and suspension system of the car. The driver's output was that the car was safe to push, was stable during handling and cornering and had no problems with oversteer/understeer. Although all drivers complained about easy rolling of the car due to low stiffness (1188 Nm/deg). We satisfied most of the goals we have set for the suspension, Ride frequency of 2 Hz, Motion ratio 0.5-0.6 and camber angles. Some things we need to consider for the next car are the integration between the chassis and the suspension components, more accuracy on the inhouse manufacture, considering adding anti roll bars for both front and rear suspension. The design can be improved by adding an antiroll bar, making the suspension a bit stiffer, better packing and removing excess weight from various components. Moreover, this thesis covers the early steps of developing a suspension system using CAD tools, and Matlab. In order to design a better system, the team later use software such as Altair Inspire which can simulate the front and the rear suspension of the car at racing conditions. (Acceleration, skidpad ,etc).

Someone who wants to understand in deep the whole development process should read the thesis of Spyros Karakostas in which Spyros analyzing geometries I designed at IPG and performed dynamic simulations there and then can have a fully view of how we can go from Cad and simple design-calculations to design a whole model of the car on IPG with the correct suspension setup.

7.References

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APPENDIX

Front and rear suspension from various cars