GRADO EN INGENIERÍA EN TECNOLOGÍAS INDUSTRIALES

TRABAJO FIN DE GRADO

Analysis and design of a roll-heave decoupled front suspension for a Formula Student prototype

Autor: José Díaz de Rábago Areilza

Director: Antonio Hernández-Ros Briales

Madrid

Declaro, bajo mi responsabilidad, que el Proyecto presentado con el título

Analysis and design of a roll-heave decoupled front suspension for a Formula Student prototype

en la ETS de Ingeniería - ICAI de la Universidad Pontificia Comillas en el

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Fdo.: José Díaz de Rábago Areilza Fecha: 10/07/ 2024

Autorizada la entrega del proyecto

EL DIRECTOR DEL PROYECTO

Fdo.: Antonio Hernández-Ros Briales Fecha: 10/ 07/ 2024

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Acknowledgements

Having reached the end of my engineering degree, I look back with gratitude to all the people who have been there with unwavering support through difficult times.

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ANALYSIS AND DESIGN OF A ROLL-HEAVE DECOUPLED FRONT SUSPENSION FOR A FORMULA STUDENT PROTOTYPE

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Supervisor: Hernández-Ros Briales, Antonio.
Collaborating Entity: ICAI– Universidad Pontificia Comillas

ABSTRACT

In Formula Student, the design of the suspension system is the most important part for the dynamic control of the vehicle and therefore vehicle performance. The thesis describes the process of studying and designing a new suspension system for a Formula Student prototype. This design will have as a base the current general suspension geometry and therefore will have to completely adapt to it. The main goal of the project is to understand the advantages and design using CAD a system capable of controlling the motions of the vehicle independently from each other.

Keywords: Formula Student, Vehicle dynamics, Suspension systems, Suspension design, Roll-heave decoupling, Dynamic analysis, FEA validation, CAD modelling, Tyre dynamics, SolidWorks.

1. Introduction

Formula Student is an international engineering competition for universities in which teams from all over the world design, manufacture and compete their formula type prototypes. The main focus of the competition is on the engaging of students in real world projects, where they can learn to understand the realities of a big scale project like designing, manufacturing and competing a race car. During the competition, the vehicle is put to the test on track, and the team is assessed on the design decisions behind the project's development. [1]

The importance of a well-designed and calibrated suspension system as well as a good understanding of vehicle dynamics is fundamental in Formula Student, as the on-track performance is mostly gained by the maximization of the vehicles available grip at low speeds and its handling characteristics. The optimization of both of these metrics is done through a correct design of the suspension and a correct understanding of vehicle dynamics.

2. State of the art

The correct design of a suspension system is done through the understanding of vehicle dynamics.

Vehicle dynamics is a field that is centred around the analysis of the forces a vehicle is subjected to, and how it reacts to them. It not only describes the reaction forces generated, but also the relative movement of the vehicle. The field of vehicle dynamics in regard to suspension design can be separated into three sub fields.

- **Tyre mechanics:** Define how the most important part of the vehicle, the tyres, behave under any circumstances depending on all the possible analysable parameters.

This field is fundamental on the understanding of the vehicles behaviour because the tyres are the part responsible for gripping the tarmac, which means they are the way the vehicle moves. The study of the effect of specially tyre position on the grip of the tyres is fundamental. [2],[3]

-Kinematics: Kinematics study the relative movement of the wheels and the vehicle, defining every possible position the tyre can be in. The kinematic behaviour of the vehicle also defines the other relative movements of the vehicle such as the steering and the compression of the spring-damper systems. This means that the kinematic behaviour of the car defines the handling characteristics, as well as the maximum available traction for the tyres at any given moment.

-Load control: The understanding and control of the variation in the loads seen by the tyres at any moment due to the different accelerations the vehicle is subjected to, is fundamental in obtaining not only the desired performance from the tyres at any point, but also, the correct stability of the vehicle when reacting to the changing loads presented by the constantly changing accelerations.

Once these fields are understood and put together, the result is a complete model of the possible dynamic behaviour of the vehicle. In this model, one can observe three main rotational movements across the vertical, lateral and longitudinal axis of the vehicle. These are known as yaw, pitch and yaw, and greatly affect the kinematics of the vehicle when moving.

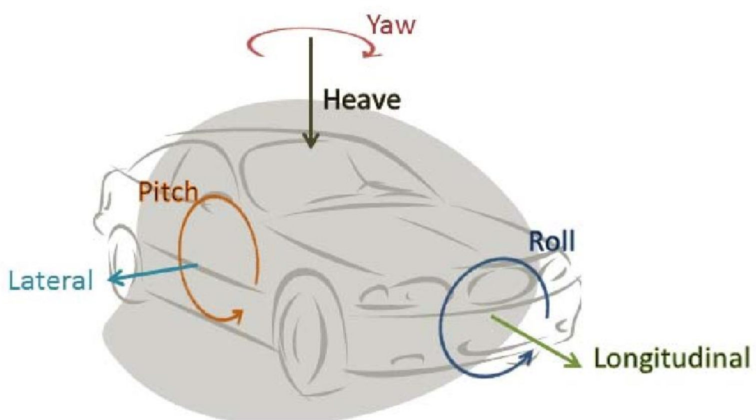

Image 1. Rotational movements of the vehicle. [4]

The other part which is needed to understand how a suspension is designed, is how a suspension system works. The general overview of the mechanisms of a suspension system is this. A suspension system is comprised of rigid structures which define the kinematics of the system and elastic and dampening elements which define the dynamics of the system. The elastic systems can be interconnected between the wheels to better control the relative motions of the connected wheels.

3. Objectives

The main objective for the project is to analyse, design and integrate a new front suspension system which will be able to control independently the vertical (longitudinal) dynamics of the front axle from the lateral dynamics, with the aim being the improvement of the cornering dynamics of the vehicle. Ease of manufacturing, a parametric design to be able to adapt to the system to possible future prototypes and the reduction of the prototypes centre of mass are also objectives of the design.

4. Design and verification

The design process for the project was divided into two: the dynamic response design and the physical design.

The dynamic response design consisted in the formulation of mathematical equivalent systems to the vertical and lateral behaviours of the car and the selection of the design parameters to obtain the desired response. The formulations of these systems consisted in a one degree of freedom model for the roll movement and a two degree of freedom system for the vertical (Heave) movement.

For the roll movement a one degree of freedom model was chosen because the objective for the roll control is mainly to achieve the desired roll angle for a given constant lateral acceleration, in order to maintain the wheels at the desired position relative to the road during high lateral acceleration manoeuvres.

For the heave movement a two degree of freedom model was chosen to mainly take into account other possible factors such as wheel hop and how they affect the system, as well as a way to better understand the dynamic behaviour of the vehicle under longitudinal acceleration.

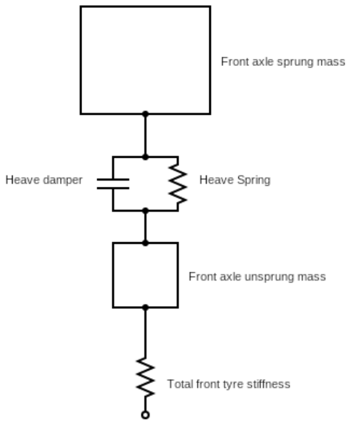

Image 2. Two degree of freedom model designed for the heave movement

The final design parameters of the dynamic response design were:

System freq. (Hz)	3.91
Wheel hop freq. (Hz)	24.36
Roll gradient (°/g)	1
Heave spring stiffness (N/m)	43780
Roll spring stiffness (N/m)	10000

Table 1. Dynamic response design parameters

The physical design consisted in in the modelling of a system that could decouple the roll and heave movements on the dampers and springs so when the axle is rolling only the roll elements are actuated, and when the axle is moving vertically only the heave elements are being actuated.

In order to achieve this, two asymmetric rockers were designed which accommodate between them a spring-damper element parallel to the ground that controls heave, and one which crosses diagonally that controls roll.

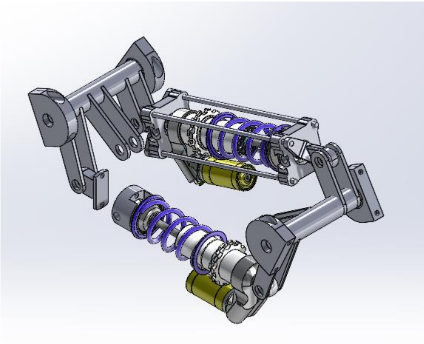

Image 3. Roll-heave decoupled system

The motion ratio of the rockers was one of the most important design factors as, together with the preexisting kinematics of the vehicle define the compression rate for the spring-damper elements. The final design having a 1.2 motion ratio for heave and a 0.8 motion ratio for roll.

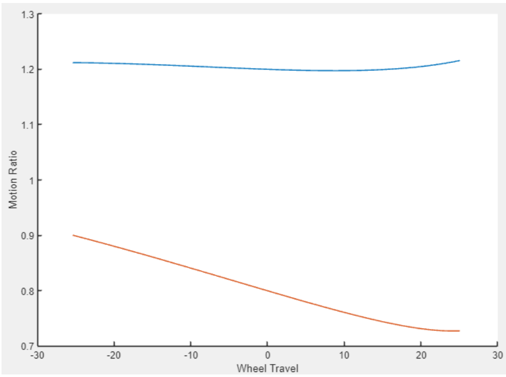

Image 4. Graph showing the effective motion ratios for heave (blue) and roll(red)

To be able to incorporate the preexisting spring-damper elements into the design an extension had to be designed to fit the heave element and a “double spring” device had to also be designed in order for the roll spring to be able to be compressed by for compression and traction strokes.

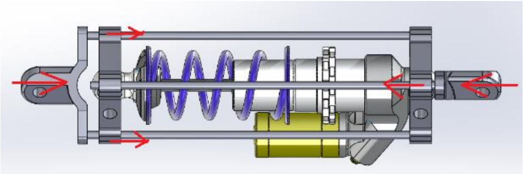

Image 5. Double spring mechanism in compression

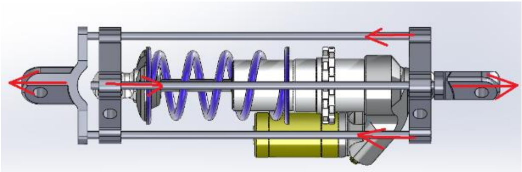

Image 6. Double spring mechanism in traction

After the main design phase was finished, the validation of the designs begun. By analysing the physical design in a Finite Element Analysis software, the validation of the design was finished obtaining a Factor of safety of 1.3 at the lowest value.

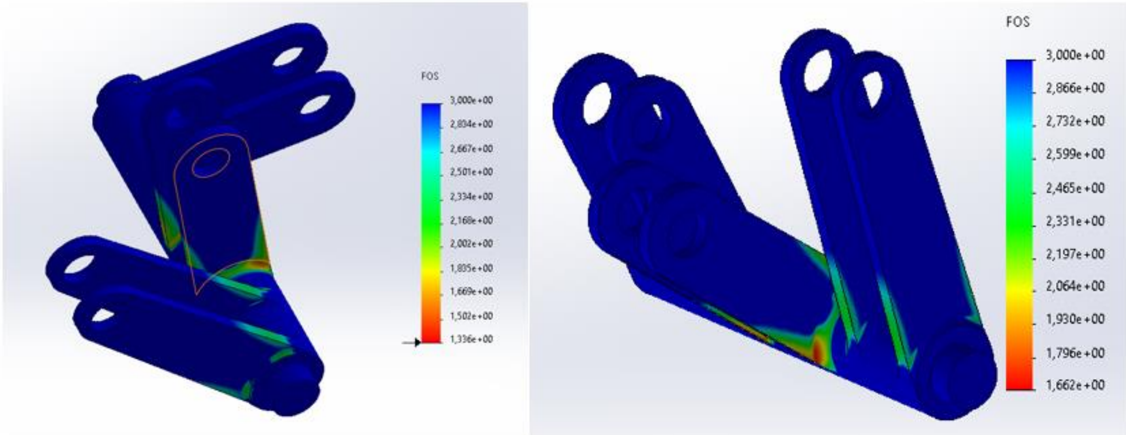

Image 7. Simulation in SolidWorks of the designed pieces and their Factor of safety

5. Conclusions

The design and understanding of the roll-heave suspension system has been completed and verified, and the design can be taken as successful. The next step to take is to bring the design to the real world and validate it in real life. The main objectives achieved were:

- The design of the front suspension to incorporate a roll-heave decoupled system.

- A better understanding and mathematical modelling of the dynamic behaviour of the vehicle.

- An improved dynamic behaviour of the vehicle by the correct selection of spring stiffnesses for both heave and roll.

- Better understanding of the kinematic behaviour of the vehicle's suspension.

6. References

[1] FSG, Formula Student Rules 2024. Version 1.1, 2024.

[2] W. F. Milliken and D. L. Milliken, Race Car Vehicle Dynamics, 1995.

[3] C. Smith, Tune to Win, 1978.

[4] Hasan-Farag, Rana Raouf. "Active neuro-fuzzy integrated vehicle dynamics controller to improve the vehicle handling and stability at complicated manoeuvres.", 2016.

ANÁLISIS Y DISEÑO DE UNA SUSPENSIÓN DELANTERA DESACOPLADA EN BALANCEO-CABECEO PARA UN PROTOTIPO DE FÓRMULA STUDENT

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Director: Hernández-Ros Briales, Antonio.
Entidad Colaboradora: ICAI– Universidad Pontificia Comillas

RESUMEN DEL PROYECTO

En la Fórmula Student, el diseño del sistema de suspensión la parte más importante para el control dinámico y por lo tanto para el rendimiento del vehículo. Este trabajo describe el proceso detrás del estudio y el diseño de un nuevo sistema de suspensión para un prototipo de fórmula Student. El diseño tendrá de base la actual geometría de suspensión y por lo tanto tendrá que adaptarse a esta. El objetivo principal del proyecto es entender las ventajas de un sistema capaz de controlar de manera independiente todos los movimientos del vehículo y diseñar este sistema en CAD.

Palabras clave: Formula Student, Dinámica vehicular, Sistemas de suspensión, diseño de suspensiones, Desacoplamiento de Cabeceo-Balanceo, Análisis dinámico, Validación en FEA, Modelado en CAD, Dinámica de los neumáticos, SolidWorks.

1. Introducción

Fórmula Student es una competición internacional de ingeniería de universidades en la que equipos de todo el mundo diseñan, fabrican y compiten prototipos estilo fórmula. El principal foco de la competición es el interesar a los estudiantes en proyectos reales, en donde pueden entender las realidades detrás de proyectos grandes como es el diseño, fabricación y competición de un coche de carreras. En la competición el vehículo es probado en la pista y los equipos son evaluados en las decisiones de diseño y liderazgo tomadas durante el desarrollo del proyecto. [1]

La importancia de un sistema de suspensión bien diseñado y con los correctos reglajes, junto con un buen conocimiento de la dinámica vehicular es fundamental en Fórmula Student, ya que el rendimiento en pista es obtenido principalmente por la maximización del agarre a bajas velocidades y la maniobrabilidad del prototipo. La correcta optimización de estos parámetros debe llevarse a cabo a través del diseño de la suspensión y del conocimiento de la dinámica vehicular.

2. Estado del arte

El correcto diseño de un sistema de suspensión debe ser realizado desde un profundo conocimiento de la dinámica vehicular.

La dinámica vehicular es el campo de la ciencia centrado en el análisis de las fuerzas a las que está sometido un vehículo y como el vehículo reacciona a las fuerzas. La dinámica vehicular no solo describe las fuerzas de reacción del vehículo, sino que también define los movimientos relativos de este. La dinámica vehicular, referida al diseño de suspensiones puede ser dividida en tres subcampos.

- Mecánica de los neumáticos: Define cómo la parte más importante del vehículo, los neumáticos, se comportan bajo cualquier circunstancia dependiendo de todos los parámetros analizables. Este campo es fundamental para el entendimiento del comportamiento del vehículo, ya que las ruedas son la parte responsable de agarrarse al asfalto, lo que significa que son la razón por la que el vehículo puede moverse. El estudio del efecto especialmente de la posición relativa de los neumáticos con el suelo en el agarre de estos es fundamental. [2],[3]

-Cinemática: La cinemática estudia el movimiento relativo de las ruedas con el resto del vehículo, definiendo todas las posibles posiciones en las que la rueda puede estar. El comportamiento cinético del vehículo define también el resto de los movimientos relativos como son los del sistema de dirección o los de la compresión de los muelles-amortiguadores. Esto significa que el comportamiento cinético define también la maniobrabilidad del coche y la máxima tracción posible de las ruedas debido a su posición.

-Control de carga: El entendimiento del control y la variación de las cargas vistas por las ruedas en todo momento (debidas a las aceleraciones a las que está sometido el vehículo) es fundamental para obtener, no solo el rendimiento requerido de las ruedas, sino que también es necesario para obtener la correcta estabilidad dinámica del vehículo.

Una vez entendidos estos campos se entiende como al unirlos se puede obtener un modelo dinámico del vehículo. En este modelo se pueden observar tres movimientos rotacionales en los ejes vertical, longitudinal y lateral. Estos movimientos se llaman guiñada, cabeceo y balanceo y afectan en gran nivel a la cinemática del vehículo en movimiento.

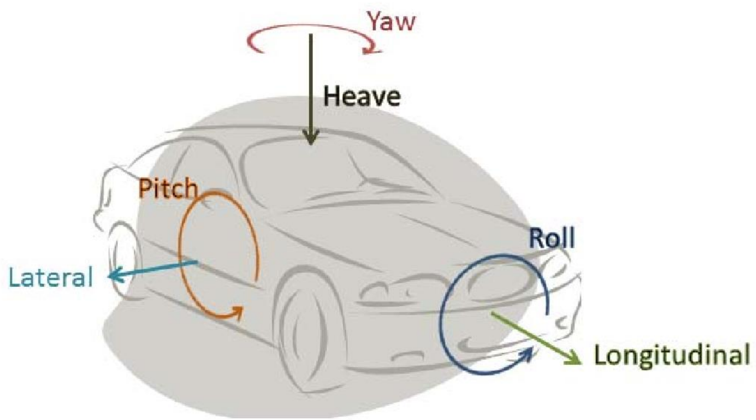

Imagen 1. Movimientos rotacionales del vehículo. [4]

La otra parte fundamental para entender el diseño de suspensiones es el funcionamiento de un sistema de suspensión. En general, el sistema de suspensiones se puede definir así. Un sistema de suspensión está formado por estructuras rígidas que definen la cinemática del sistema y por elementos elásticos y amortiguadores que controlan la dinámica del sistema. Estos elementos elásticos y amortiguadores pueden estar conectados entre sí para poder controlar mejor los movimientos relativos de las ruedas conectadas.

3. Objetivos

El principal objetivo del proyecto es analizar, diseñar e integrar un nuevo sistema de suspensión delantero que tiene la capacidad de controlar independientemente el desplazamiento vertical del eje (como el que se ve durante el cabeceo), del balanceo del eje, con la meta siendo la mejora del comportamiento del vehículo en curva. La facilidad de fabricación, diseño paramétrico para poder adaptar el sistema a futuros prototipos y la reducción del centro de masa del prototipo son también objetivos del diseño.

4. Diseño y verificación

El proceso de diseño del proyecto se dividió en dos el diseño de la respuesta dinámica, y el diseño físico.

El diseño de la respuesta dinámica consistió en la formulación de modelos matemáticos equivalentes a los sistemas dinámicos de los comportamientos laterales y verticales y la selección de los parámetros de diseño para la obtención de la respuesta dinámica deseada. La formulación de estos sistemas consistió en un sistema de un grado de libertad para el movimiento de cabeceo, y en un sistema de dos grados de libertad para el movimiento vertical.

Para el movimiento de cabeceo un sistema de un grado de libertad fue elegido ya que el objetivo con el control del balanceo es principalmente conseguir cierto ángulo para una aceleración lateral constante, para poder mantener las ruedas en la posición deseada con respecto a la carretera en maniobras de gran aceleración lateral.

Para el movimiento vertical un sistema de dos grados de libertad fue elegido para principalmente tener en cuenta factores de múltiples modos de vibración como es el “salto de rueda” y ver los posible problemas que pueden causar al sistema en general. También para poder entender mejor al comportamiento del vehículo en aceleración longitudinal.

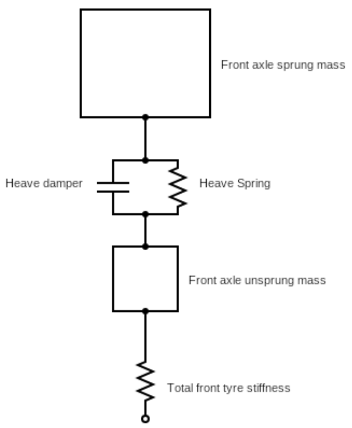

Imagen 2. Modelo de dos grados de Libertad diseñado para el movimiento vertical

Los parámetros finales del diseño de la respuesta dinámica fueron:

Frec. del sistema (Hz)	3.91
Frec. de salto de rueda (Hz)	24.36
Gradiente de balanceo (°/g)	1
Rigidez del muelle v. (N/m)	43780
Rigidez del muelle b. (N/m)	10000

Tabla 1. Parámetros finales del diseño de la respuesta dinámica

El diseño físico consistió en el modelado de un sistema que pudiera desacoplar el balanceo del movimiento vertical en los muelles-amortiguadores, de tal manera que cuando el eje delantero sufre balanceo solo se ven afectados los elementos encargados del balanceo, y cuando el movimiento es vertical, solo se ven afectados los elementos encargados del movimiento vertical.

Para poder obtener un diseño como el descrito, dos balancines asimétricos fueron diseñados para acomodar entre ellos un muelle-amortiguador paralelo con el suelo para encargarse del movimiento vertical y uno en diagonal para encargarse del balanceo.

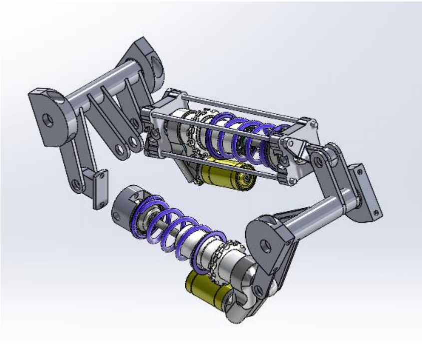

Imagen 3. Sistema desacoplado en balanceo-cabeceo

La relación cinética de los balancines es uno de los factores más importantes en este diseño, ya que junto a la cinemática preexistente definen el ratio de compresión de los muelles-amortiguadores. El diseño final tiene una relación de 1.2 en movimiento vertical y de 0.8 en balanceo.

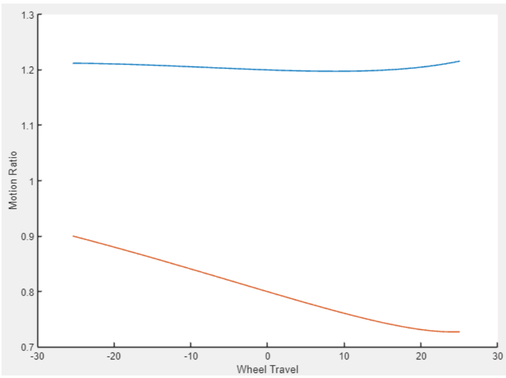

Imagen 4. Gráfico con las relaciones cinéticas para movimiento vertical (azul) y balanceo (rojo)

Para poder incorporar los muelles-amortiguadores previos en el diseño se tuvo que fabricar una extensión para el elemento de movimiento vertical y un sistema de "muelle doble" se tuvo que diseñar para que el elemento de balanceo pudiera ser comprimido para tanto movimientos de tracción como de compresión.

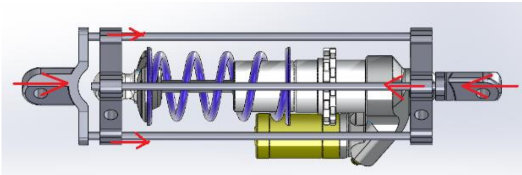

Imagen 5. Mecanismo de doble muelle en compresión

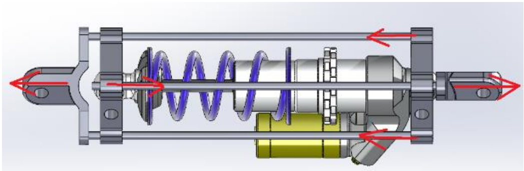

Imagen 6. Mecanismo de doble muelle en tracción

Tras la fase de diseño principal, la fase de validación comenzó. La validación se llevó a cabo en un software de simulación por elementos finitos obteniendo un factor de seguridad mínimo de 1.3.

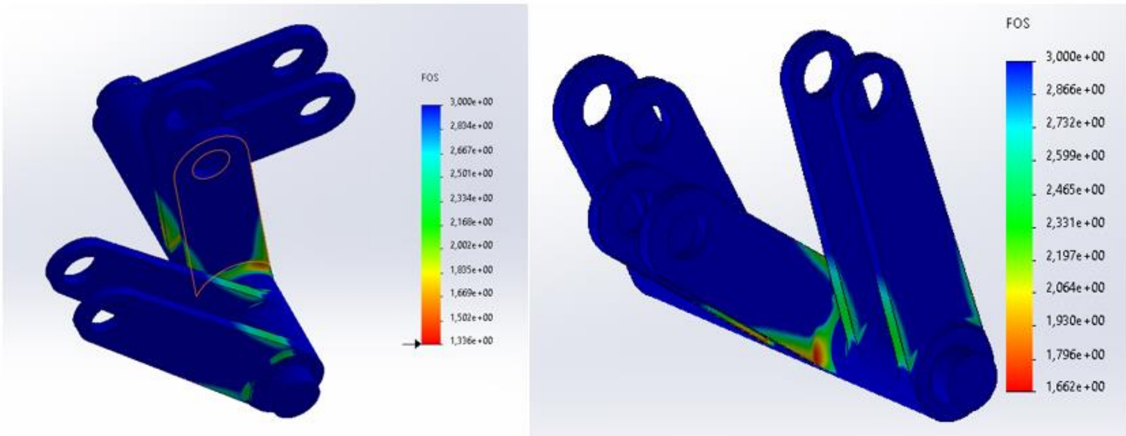

Imagen 7. Simulación en SolidWorks de las piezas diseñadas con su factor de seguridad

5. Conclusiones

El diseño y entendimiento del sistema desacoplado en cabeceo-balanceo ha sido completo y verificado, con el diseño pudiendo ser dado por válido. El siguiente paso será traer el diseño a la realidad y verificarlo correctamente fuera de la simulación. Los principales objetivos conseguidos fueron:

- El diseño de la suspensión delantera para incorporar un sistema desacoplado.

- Un mejor entendimiento y modelado del comportamiento dinámico del vehículo.

- Un mejorado comportamiento dinámico del vehículo gracias a la correcta selección de las rigideces verticales y de balanceo.

- Un mejor entendimiento del comportamiento cinético de la suspensión del vehículo.

6. Referencias

[1] FSG, Formula Student Rules 2024. Version 1.1, 2024.

[2] W. F. Milliken and D. L. Milliken, Race Car Vehicle Dynamics, 1995.

[3] C. Smith, Tune to Win, 1978.

[4] Hasan-Farag, Rana Raouf. “Active neuro-fuzzy integrated vehicle dynamics controller to improve the vehicle handling and stability at complicated manoeuvres.”, 2016.

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Chapter 1. INTRODUCTION

1.1 FORMULA STUDENT

Formula Student is an international student engineering competition, which originated in the United States under the name Formula SAE in 1981. Through the years, it has expanded globally and risen to be the largest and most prestigious student engineering competition in the world, attracting thousands of students and presenting them with the challenge to fully design, manufacture, assembly and compete their prototype vehicles, according to a ruleset defined by the competition.

The main goal of the competition is the development, construction, and competition of small, formula style, race cars designed in accordance with the rules by university students. The competition not only consists of the development of the car, but also everything related to the project, which includes the team managing, the business planning, the marketing, the logistics, the data analysis, and much more.

This competition challenges students to go through the development of a real engineering project gaining hands on experience, presenting the students with a place to employ their engineering knowledge and put to practice what they learn in a competitive environment. All this work is then tested during the competitions through the static and dynamic events in which not only the car and its performance on track is put to the test, but also the team and their knowledge and ability to carry out the project.

Not only does the competition present the participants with a challenge, but it also presents them with a great community and an environment which share common values and interests. It is also an optimal environment to develop new designs and solutions for problems from the automotive industry, achieving innovation by focusing on performance gain, budgetary restrictions or even the restrictions presented by the rules. These factors also attract many big automotive related businesses to Formula Student, either by directly sponsoring different teams and competitions or indirectly by rewarding applicants with experience in the competition.

Figure 1. The team at Formula Student Spain 2023

The different events on every competition are divided into two different types: the dynamic events, in which, once the prototype has been deemed compliant with the rules, its performance is evaluated in relation to the rest of the field; and the static events, in which the team is evaluated on the design process, the business model of the team and the cost reporting.

The dynamic events are:

Acceleration: It consists of a 75-meter-long track in which the car accelerates to complete the straight in the shortest time, testing the vehicles capacity to launch and accelerate, making teams focus on the power to weight ratio, the ability for the powertrain to respond to torque peaks and the overall longitudinal grip of the vehicle.

Skidpad: It consists of a figure eight track consisting of two circles 15.25 meters in diameter and of a track width of 3 meters, the vehicle must perform two full consecutive laps of each circle and the time is averaged the sum of the best of each circle. This test challenges specially the cornering dynamics of the vehicle, with the suspension both in design and setup being the most important part of the prototype for this event.

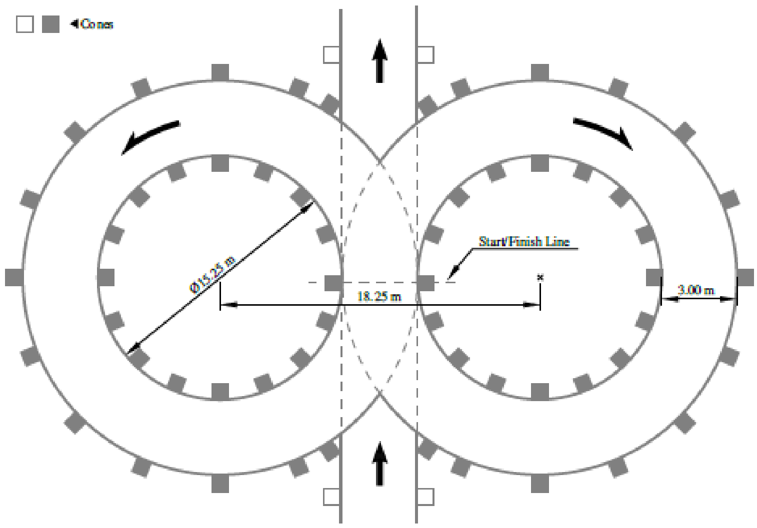

Figure 2. Layout for the Skidpad event. [1]

Autocross: It consists of a timed run of a coned marked track designed to test all the vehicles handling characteristics, as well as its ability to maximize acceleration in short straights. This event essentially tests the performance of every system on the prototype and vehicles performance as a complete package.

Endurance: It is the last event of the competition and consist of a 22-kilometre race around the track at which the autocross is run, where the durability of every component is put to the test. This test challenges, not only the vehicles performance, but also its reliability and ability to maintain a high-performance level during the whole event.

The static events are:

Engineering design: It consists of a presentation of the different departments of the vehicle explaining the design process followed and the logic behind every decision taken. It aims to assess the team's ability to follow coherent design decisions and explain them in an articulate manner.

Cost and Manufacturing: It consists of a detailed report of the costs of the manufacturing of the prototype. It aims to assess the team's ability to estimate the costs, as well as their knowledge of the different manufacturing techniques.

Business plan: It consists of a simulated pitch to potential investors in which the judges focus on the team's ability to communicate their vision, as well as their potential and viability as a project. It aims to assess the financial knowledge as well as the team's entrepreneurial mindset.

During a competition the points distribution for the events are:

	CV & EV
Static Events:	
Business Plan Presentation	75 points
Cost and Manufacturing	100 points
Engineering Design	150 points
Dynamic Events:	
Skidpad	50 points
Driverless (DV) Skidpad	75 points
Acceleration	50 points
Driverless (DV) Acceleration	75 points
Autocross	100 points
Driverless (DV) Autocross	-
Endurance	250 points
Efficiency	75 points
Trackdrive	-
Overall	1000 points

Figure 3. Maximum point allocation for the different events in Formula Student [1]

1.2 THE ISC FORMULA STUDENT RACING TEAM

The ISC Formula Student Racing Team is the electric Formula Student Team from ICAI, it was funded in 2017 and the first prototype was produced a year later in 2018. Born as a fully electric team, the first years were a challenge, with the design and integration of an electric power train presenting some great difficulties, but the first running prototype was produced overcoming most issues in the year 2021. This development continued steadily the following years culminating in the IFS-05, the prototype of the season 2022-2023, which was the first prototype to compete in all the dynamic events, and even finishing the endurance, the most gruelling event which most electric teams are not able to finish.

Figure 4. The IFS-05 Competing at the endurance event in Formula Student Spain.

The team is centred in offering a way for the students to be part of a real engineering project and challenging them to be able to solve real problems and push themselves to exceed their own expectations, all of this while pushing to achieve better results every season and passing down the knowledge gained. Currently consisting of 61 members, the team is currently standing 6th from the 22 registered Spanish teams in FS-Electric and 133 in the global FS-Electric ranking.

One of the greatest hurdles that the team needs to overcome is that, because Formula Student in Spain is relatively unknown, the ability to grow from sponsors and partners is greatly diminished in contrast with other countries where these types of projects are more known and supported, this leads to a lower budget as well as a higher threshold to overcome when trying to attract interest in the project, and while Formula Student is an engineering competition, the budget is always a constraint for the development of the project, making “smaller” teams less competitive as a whole.

Taking onto account these limitations, as well as the relative youth of the project, the prospects for the team are mainly, to be able to compete with the top Spanish teams in Formula Student Electric. This means not only with the new electric teams, but also with the teams that are transitioning from combustion backgrounds, which come with a great experience and backing but will be faced with the challenges of developing and integrating an electric powertrain. Based on this, the team should be trying to maximize both the on-track performance of the prototype, as well as the ability to defend the design, business plan and cost and manufacturing.

Through the development of this thesis together with the ISC all these objectives can be approached, as it helps with the development of the vehicle's on track performance, designing a more versatile suspension and documenting it for the design and cost events as well as forming team members by presenting a new design which completely changes the current architecture, making them understand the reasons behind this change, and therefore the reasons behind the design decisions.

1.3 ROLE OF THE SUSPENSION ON A FORMULA STUDENT PROTOTYPE

The suspension system on any road going vehicle ensures the correct contact the vehicle and the road and maximizing the amount of grip available at any point and therefore maximizing the performance the vehicle can offer at any point.

When talking about Formula Student, the importance of the suspension system is greatly increased. This is because at the tracks, and cornering speeds at which the competitions are held, the maximization of the available grip doesn't depend so much on the downforce that can be generated, but more on the optimization of the contact between the tyre and the track, which is defined by the suspension design. All of this means that the design of a Formula Student suspension is the most important design feature for obtaining the desired cornering characteristics.

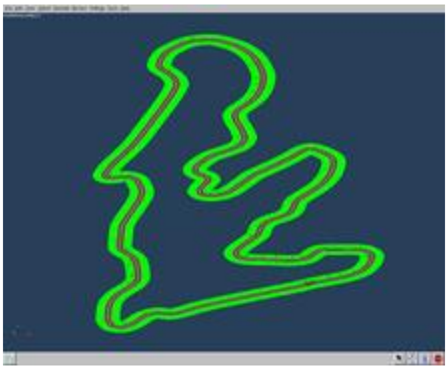

Figure 5. Generic FS track layout. [2]

Other factor in which the suspension design is crucial is the longitudinal dynamics of the vehicle, especially when considering the weight transfer during hard accelerating and decelerating, as when the design is correctly tuned to the vehicle, the traction is optimized and therefore the performance is maximized.

When considering Formula Student as a whole, the correct design of the suspension is also a fundamental piece in the design event, as the suspension interfaces with all the mechanical components of the vehicle. Therefore, a correct design of the suspension consists of not only understanding and applying the vehicle dynamics to the prototype, but also being able to incorporate the design in a coherent manner with the rest of the vehicle.

This integration of the suspension system into a bigger mechanical system should also be applied to the individual components of the suspension to correctly defend the design, so all the interactions between every component of the suspension must be analysed and understood to be able to correctly defend the design as every change to a design can affect the whole system.

Finally, the understanding of the suspension in any type of vehicle, is also fundamental to understand the changes in the set up that need to be made to improve the vehicles performance, which on formula student is fundamental to be able to correctly prepare the car for the different events. The understanding of the behaviour is also fundamental to design the next prototype to mitigate the flaws observed during the competitions and testing and correct any unwanted tendencies.

1.4 OBJECTIVES

With this background understood, the objective of the project can be defined as the design and integration of a new front suspension system which will be able to control independently the vertical (longitudinal) dynamics of the front axle from the lateral dynamics, with the aim being the improvement of the cornering dynamics of the vehicle during all the stages: braking/corner entry, steady state cornering and accelerating/corner exit. This design should also be easy to manufacture and assemble, as well as be easy to access to change the parameters, finally the design should try to lower the height of the centre of mass of the vehicle.

This main objective can be subdivided into sub-objectives such that they can be analysed individually, as well as given priority in regards of their importance:

- The design must comply with the ruleset defined by the Formula Student Competition.

- The design must be able to integrate with the current front suspension geometry without compromising its design parameters.

- The design must be able to fully decouple the heave from the roll movements on the front axle.

- The design must be able to adapt to different stiffnesses for both heave and roll to achieve the desired dynamic objectives.

- The design should be easy to manufacture and assembly, as well as accessible for set up changes.

- The design should lower the centre of mass of the vehicle by lowering or eliminating components.

These objectives, together with the main design objectives for the whole prototype, define the design decisions taken during the development of the project.

1.5 METHODOLOGY AND RESOURCES

The design methodology followed during this project has followed the following steps:

1. Conceptual design: This first step consists mainly in the understanding of the concept of a decoupled suspension and its advantages and disadvantages. During this phase, the main objectives of the project were defined, and the concept was evaluated according to these objectives.

2. Study and planning: During this step, the main research for the correct design for a suspension system was done, which helped plan the resources to be used during the project as well as the necessary calculations and analysis that needed to be performed. This phase was fundamental in defining the scope of the project, maintaining it concise.

3. Design phase: This phase consisted of the physical design and mathematical modelling of the new suspension system together with its manufacturing process. This phase was completely guided by the main objectives of the project. Making every design decision defined by the main objectives helped streamline the design process and keep unnecessary features out of the final design.

4. Validation: Throughout this final phase, the design was completely validated through simulations and calculations. During this phase, the design was also finalised by optimizing it as much as possible to try and achieve the design objectives described.

The resources used on this thesis were:

MATLAB: Generally used in the engineering filed, it is a programming language which allows both numeric as well as symbolic computing, data manipulation and representation, as well as many other features. For this project it has been used to for the mathematical dynamic modelling of the vehicle, as well as for the optimization and analysis of the geometry.

Excel: Mostly used as a spreadsheet program, it is a great tool for data visualization, and organization, as well as for the automation of calculations. For this project it has been used for the organization of the results, as well as for data visualization.

Solid Works: A general use CAD (Computer aided design) program with the ability to perform FEA (Finite element analysis) as well as CAM (Computer aided manufacturing) and many other functions. For this project it has been used extensively to both design all the components, as well as perform FEM analysis to validate the structural integrity of all the components.

Chapter 2. STATE OF THE ART

2.1 INTRODUCTION TO VEHICLE DYNAMICS

Vehicle dynamics is a field of engineering which studies the dynamic behaviour of any vehicle, and the effects different actions have on them. For this thesis when referring to vehicle dynamics we will be specifically referring to the field related to automobiles (with four wheels in two axles).

To fully comprehend the importance of vehicle dynamics for the design of a racing car, first one must understand the main goal for any racing vehicle, which is to be the fastest vehicle on that competition at that track. To be able to have the fastest vehicle, that vehicle must be able to maximize its acceleration on any direction the track needs and with the maximum amount of control, and that is how performance is obtained.

Vehicle dynamics studies the principles behind the control, stability, handling, and general motion of the vehicle. Through the understanding of this field, the design and optimization of the vehicle can be carried out with confidence, as this field defines the maximum acceleration available at any point in any axis, and therefore, through the correct understanding of this field, the performance of the vehicle can be maximized.

The behaviour of any vehicle can be understood using vehicle dynamics to study all the factors that influence the vehicle. These factors can be both driven by the design such as: tyre choice, suspension geometry, suspension stiffness, etc. As well as be external factors such as track temperature, air density, track smoothness, etc. By correctly designing the factors which depend on the designer and accounting for the ones which are external, a better performing vehicle can be designed for any application.

Regarding to automobiles, the field of vehicle dynamics can be divided into three main factors which can be designed to affect vehicle performance, which are:

- Tyre mechanics: The behaviour of the tyres is the most important factor to understand in vehicle dynamics, as it is through the tyres that the vehicle can achieve the grip needed to both control and accelerate the vehicle. The study of tyre mechanics involves understanding how the tyre grips, as well as how the grip is affected by any factor which affect the tyre for example: relative position with the ground, temperature, stiffness, etc.

- Kinematics: The kinematics of a vehicle define the relative movement of the wheels and the vehicle without considering the forces involved. It defines the possible positions of the tyres, as well as the possible compression and compression rates of the spring damper systems. It defines the steering geometry and the different centres of movement for the vehicle.

- Load control: Due to the accelerations, the different loads of the vehicle move their point of application, this can affect the performance of the vehicle and its stability. This control can be made by the driver using their controls: accelerator, brakes, steering wheel, etc. The control can also be applied to these devices using external active systems such as anti-lock braking systems (ABS) or traction control (TC). Finally, there is also the control of the load transfer of the vehicle by the springs and dampers of the suspension system which can either active or passively affect the load transferred to each tyre and control the transients to better stabilize the car.

To understand the field of vehicle dynamics, it is fundamental to understand a vehicles response to the different external forces it can be subjected to, as well as the basics of how tyres grip.

To understand the basics behind the way tyre grips the road there are three basic principles to know. The first one is that the coefficient of friction between the tyre and the road depends on many factors, one of the most important when discussing vehicle design is how the tyre surface contacts the road, this means that the relative position of the tyre with the ground greatly affects the performance of the tyre. The second one is that the maximum tangential force that a tyre can transmit to the road is directly proportional to vertical (normal) load it transmits to the road. The final one, and the most difficult to understand, is that the mechanism by which the tyre can grip during cornering is not only defined by the coefficient of friction and the normal load transferred but it is also dependant in the characteristics of the tyre and how it deforms, this deformation is known as tyre slip. The tyre deforms in such a way that the contact surface with the road follows the vehicles path whereas the rest of the tyre doesn't this difference in angle is known as slip angle. This deformation presents a resistance which helps increase the amount of lateral acceleration the tyre can support, the amount of resistance is also dependant on many external factors, but tyre position and load are also fundamental factors in defining this resistance. [3],[4],[5],[6]

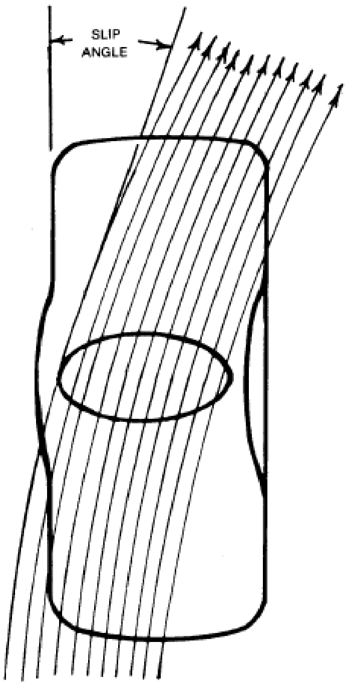

Figure 6. Illustration of slip angle. [4]

Firstly, we will define the axes and reference planes for the vehicle. The longitudinal axis (x axis) is defined as parallel to the floor of the vehicle and perpendicular to the rear axle. The lateral axis (y axis) is defined as parallel to the floor of the vehicle and to the rear axle. The vertical axis (z axis) is defined as perpendicular to the floor plane of the vehicle, generally the origin of these axis is defined in the centre of mass of the vehicle.

Once we have defined these axes, we can approach the concept of longitudinal and lateral load transfer.

Load transfer is the transfer of the load carried by a tyre due to variations in the movement of the vehicle. In this thesis we will discuss the longitudinal load transfer and the lateral. These are mainly caused by the balancing of the forces on the tyres due to variations in its acceleration, which in turn affect the vehicles behaviour like forces acting on its centre of gravity.

On the longitudinal axis when accelerating, this load transfer increases the front axle load which increases the front axle grip and the rear decreases. On the lateral axis, the load transfer increases the load on the outer tyres, increasing the grip on that side while decreasing it on the other. All these load transfers are greatly influenced by the vehicle's dimensions, centre of mass position, static weight distribution, kinematics and springs and dampers. To increase performance to the vehicle, downforce generating elements can be added which could also influence these load transfers, by creating more load at different speeds.

The general formula for steady state load transfer during continuous acceleration for a perfectly rigid vehicle with no springs or dampers depends only on four factors: the total mass of the vehicle; the continuous acceleration the vehicle is subjected to; the height of the centre of mass and the distance between the part that increases load and the part that decreases (for longitudinal transfer this distance is the wheelbase and for the lateral it is the track). In general, the formula can be written as:

$$\Delta W = m \times \frac{a \times h_c}{l}$$

[5]

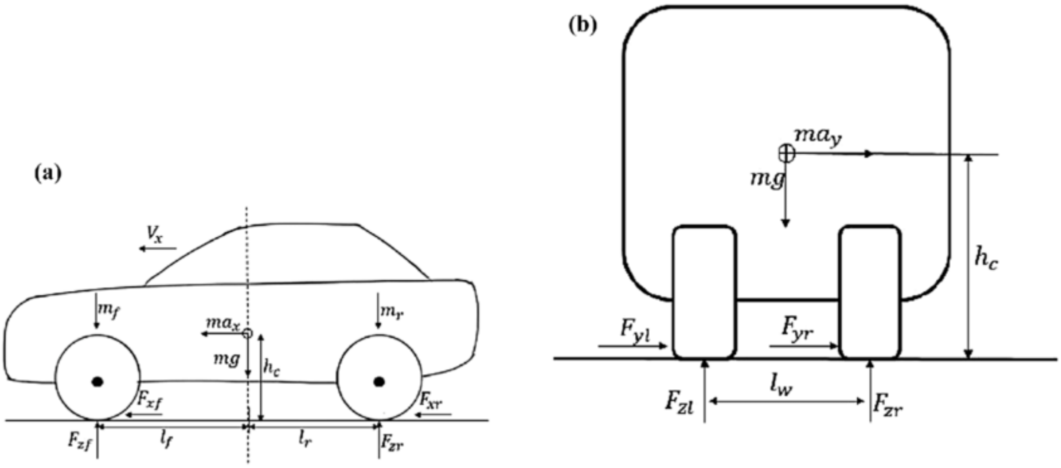

Figure 7. Free body diagram describing the forces involved in load transfer. a) longitudinal and b) lateral. [7]

These load transfers can be affected and controlled by the suspension system. This control of the dynamic behaviour of the vehicle can be obtained through the general kinematic and geometric design, or in a more adjustable manner incorporating elastic and energy dissipating elements between the axes.

When incorporating elastic components into the model, the result is the appearance of a rotational movement which accompanies the load transfer. Depending on in which axis they appear these movements are:

- **Pitch:** Rotational movement around the lateral (y) axis. It is caused by longitudinal acceleration (positive or negative) it causes the front and rear axles to move vertically in opposite directions, with the one where the load is increases rising and the other lowering. This vertical movement of the complete axle is known as heave.

- **Roll:** Rotational movement around the longitudinal (x) axis. It is caused by a lateral acceleration (such as the centripetal acceleration suffered during cornering) it causes the inner and outer tyres to move vertically in opposite directions, with the outer tyres rising and the inner tyres lowering.

- **Yaw:** Rotational movement around the vertical (z) axis. It is caused by the difference in grip from the inner and outer tyres and causes the front and rear axles to move laterally at different rates or directions.

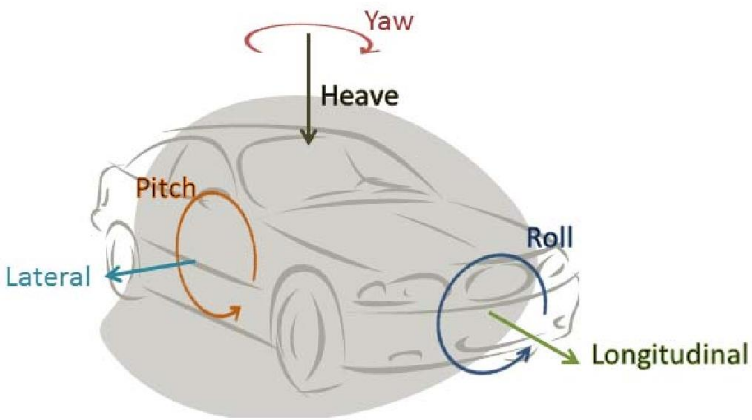

Figure 8. Diagram of the different rotational movements defined. [9]

2.2 SUSPENSION

2.2.1 SUSPENSION DESIGN

The suspension system in a vehicle is responsible of ensuring and controlling the contact between the vehicle and the road. Through the correct design of the suspension therefore, the dynamics of the vehicle can be controlled, which in turn affects not only the performance of the vehicle, but also its stability, comfortability and safety.

The design of a suspension system can be divided into three distinct parts which are greatly interconnected. These are: the selection of the architecture, the definition of the geometry and kinematic behaviour, and the design of the dynamic control of the vehicle.

Firstly, we will discuss the definition of the suspension architecture. Suspension architecture refers to the different types of mechanical layouts a suspension system can have. There are many types of suspension layouts, which in turn all have some advantages and disadvantages. The three main systems used today are:

- **The Solid Axle suspension:** Consisting of a solid axle joining both wheels, making the individual wheel positions dependent on the other wheel of the axle, and the position the wheels independent form the position of the vehicle. This is a very robust and cost-effective solution.

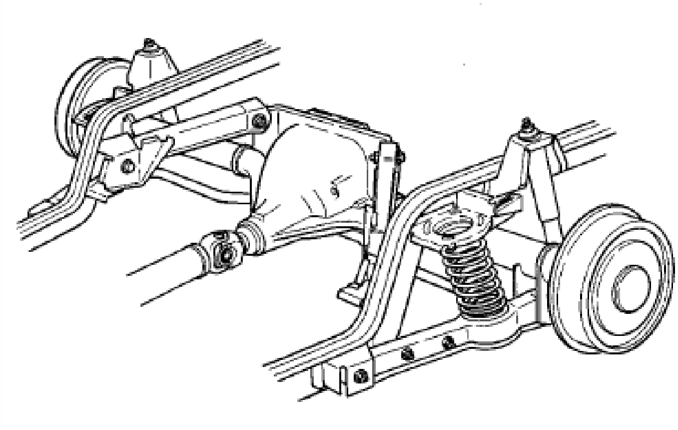

Figure 10. Solid axle for link suspension. [5]

- **The MacPherson Strut suspension:** consisting of a lower control arm which ties the wheel and upright to the car and the spring damper system acting directly to the upright. This is also a great cost-effective solution which achieves wheel independence, while not occupying much space.

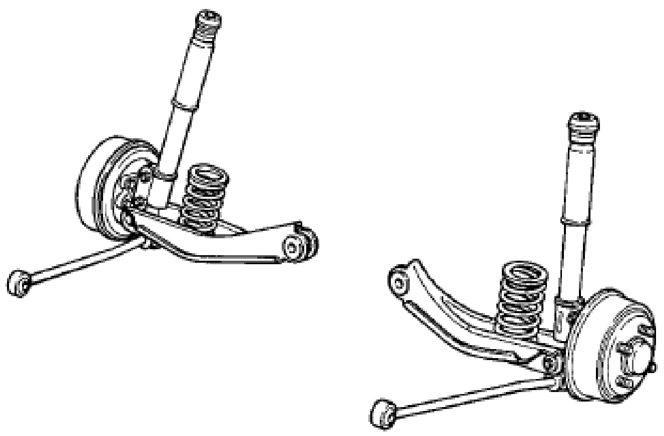

Figure 11. MacPherson Strut suspension. [5]

- The double wishbone suspension: the most complicated of the suspension systems discussed, but also the most effective at the control of the wheel position, while also offering the most degrees of freedom to achieve the desired design. This architecture is the one chosen for the Formula Student prototype.

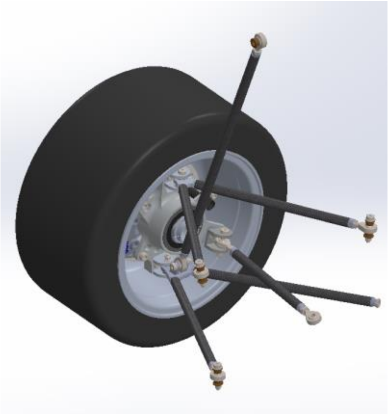

Figure 12. Double Wishbone suspension of the current IFS-06 Prototype

The advantages of using this more complicated design are mainly: the better control of the wheel path during suspension travel, which allows for a better control of the tyre behaviour, resulting in potentially a better performance on track; the improved stiffness of the setup, which in turn makes the suspension less compliant and therefore ensures the designed path is correctly followed under load. These advantages of the double wishbone setup are widely known, as can be observed when analysing high performing vehicles, as well as most Formula Student teams.

The next point to discuss in suspension design is the kinematic design, also known as the suspension geometry which is the part which defines the tyre path and position and therefore is fundamental in defining how the vehicle grips the road. To correctly understand this point, we must firstly define some concepts regarding the geometry.

- Camber: Defined as the angle Between the tilted wheel plane and the vertical. It is crucial in defining the behaviour of the contact between the tyre and the road. It is defined positive, when the top of the wheel leans away from the vehicle. Usually, racing cars have negative camber.

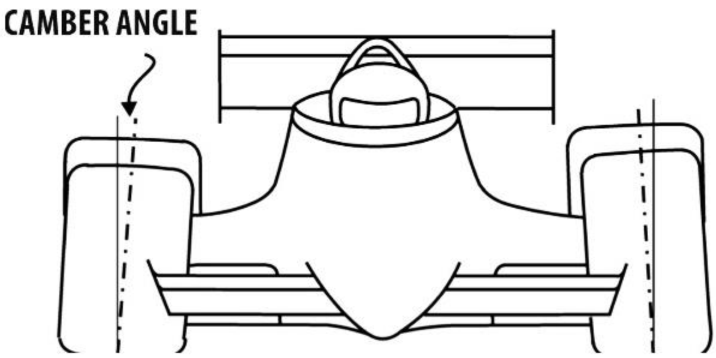

Figure 13. Diagram showing the negative camber angle. [10]

The way camber affects the tyre grip is not only by modifying the contact patch of the tyre with the road, but also by the lateral deformation of the trail, acting similar to the slip angle, by having the force known as camber force acting on the direction of the tilt. This force, similar to the force produced by the slip angle increases with the angle until a certain drop off point where it stops. It is also important to note that as this force acts in different directions on the inside and outside tyres, the increase in the grip of one tyre corresponds with the decrease of the other.

Understanding this, one can now understand the importance of the camber gain defined by the way the wheel travels through the suspension travel.

- Toe: Defined as the angle between the wheel plane and the longitudinal plane of the vehicle. This angle is crucial in defining the handling of the car, as it directly affects the steering input of the vehicle. It is defined as toe-in when the fronts of the tyres are leaning more to the centre plane of the vehicle, and toe-out when the opposite occurs. Usually, cars have toe-in in the front tyres and no toe on the rears.

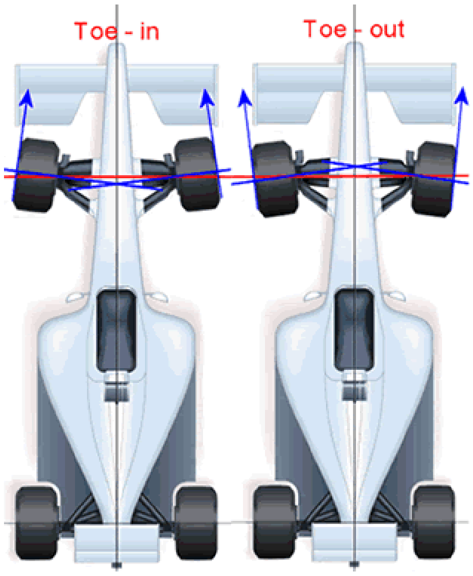

Figure 14. Diagram showing the toe angles. [11]

The way toe affects the handling behaviour of the car is increasing or decreasing the static slip angle of the tyres. Excessive toe in any direction causes the tyre to scrub and therefore to increase the tyre wear and decrease grip. Toe-in is used generally to increase straight line stability, reducing responsiveness in turn in, whereas toe out generally is used to improve turn in capabilities, helping the vehicle scrub initially the tyres less. Toe generally doesn't change very much with the suspension travel, but the steering input can affect the amount of toe on the front tyres.

- Caster and trail: Caster is defined as the angle as seen from the side between the central plane of the tyre and the axis defined by the mounting points of the upright known as the steering or kingpin axis. It affects the size and location of the contact patch of the tyre. The distance between the intersection of the kingpin axis with the ground plane and the centre axis of the wheel is defined as the mechanical trail and it affects the steering input force needed.

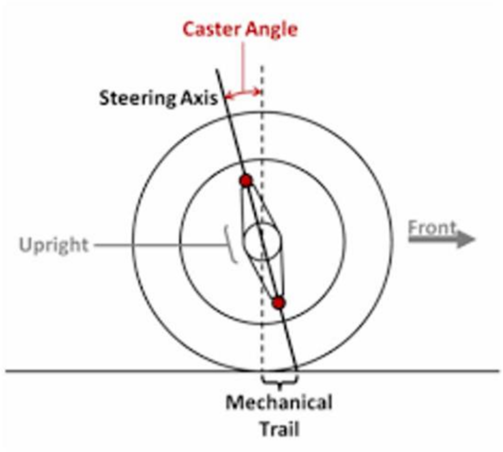

Figure 15. Diagram defining mechanical trail and caster angle. [12]

The caster angle generally affects the steering feel and responsiveness, as well as the self-ailing moment of the tyre. The trail is a result of the caster angle and is a parameter that is mostly used for the calculations of the steering force needed to achieve different turns at different speeds.

- KPI and scrub radius: The KPI (Kingpin Inclination) is defined as the angle as seen from the front plane of the vehicle between the kingpin axis and the vertical axis of the wheel. The distance from where the kingpin axis intersects the ground to the intersection of the wheels vertical central plane with the ground is known as the scrub radius.

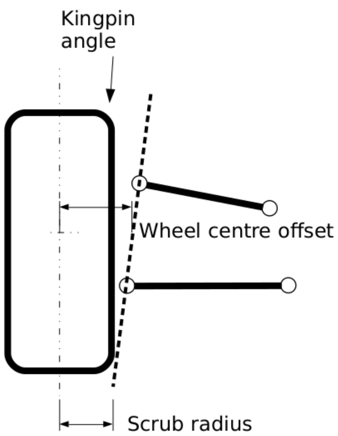

Figure 16. Diagram showing the KPI and scrub radius.[13]

These two factors affect the way the camber changes with suspension travel and steering input, and also the stability and grip on the tyres, and also the general feel and strength needed for steering.

The final point to discuss in suspension design is the design of the dynamic control of the vehicle, specifically the dynamic control of the suspension. The control of the vehicle's suspension can have active or passive elements, and the different controls can be interconnected or completely independent.

Suspensions with active controls have systems which are actively controlled by an external logic programmed to make the car behave in a predetermined manner reacting to the inputs of other sensors and data programmed. These systems are greatly complicated but can achieve a near perfect control of the vehicle's behaviour. The major drawback of these systems is the complexity of the system they require to function correctly as well as the need for precise and quick actuators which can increase the weight of the vehicle and the system.

Passive suspension systems are those in which the dynamic control of the suspension of the vehicle is achieved using only energy accumulating and energy dissipating elements. These types of systems are the most common as they generally are much less complex and don't rely on external power to function.

These passive systems are generally connected to each individual wheel, controlling the dynamic behaviour of the wheel by adding a spring and a damper to correctly control the time and frequency response of the tyre. This individual control is generally seen as too simple, as the vehicle has more modes of movement than a single wheel hop.

The next step to take is to interconnect each spring system of the same axle so that they can together also control the lateral load transfer on each axle during cornering. This is where anti-roll bars come into play. These devices connect both sides of the suspension system on an axle to add a factor to affect the load transfer between the sides.

Figure 17. The rear u-shape ARB of the ISC's IFS-06

The next step would be to completely decouple the vertical movement of an axle and the lateral load transfer, which makes the system able to independently control the behaviour of the vehicle during two very different modes, this heave-roll suspension is generally the norm in high-ranking Formula Student teams.

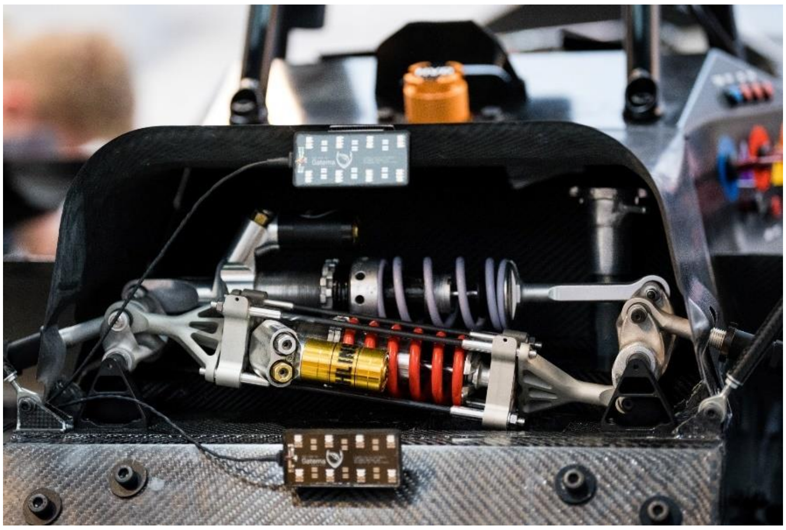

Figure 18. The roll-heave decoupled suspension system on TU Brno's prototype. [14]

The final step in a passive suspension to achieve the best dynamic control would be to interconnect the axis and the wheels in a diagonal and then all the modes of movement would be independently controlled. This fully decoupled passive suspension has been achieved in Formula Student only by one team in the world.

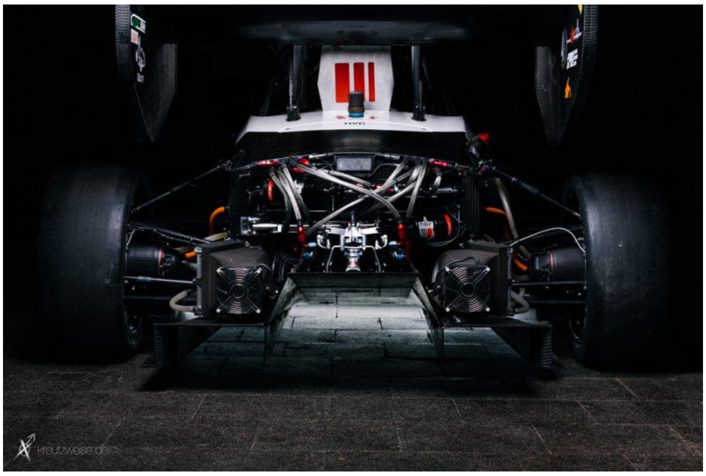

Figure 19. The fully decoupled hydraulic suspension of AMZ's Formula student prototype. [15]

2.2.2 ROLL AND HEAVE CONTROL

Now that the basics of vehicle dynamics and suspension design and the operating principles of a suspension system have been explained, we can understand the problems presented by both the heave and pitch movements. Firstly, we will discuss the problems with the vertical movement of the axle and the possibilities to better control it during different types of movement and then, the rolling motion's problems.

Heave, defined as the vertical movement of the axle, needs to be considered when designing a suspension system as it greatly affects the wheel positions for the axles during longitudinal acceleration. Other problems that excessive pitch can cause are bottoming out diffusers or front wings, or overloading the front tyres under braking and locking up.

This movement can occur in one of two mechanisms, the first one is a perturbance in the road affecting equally both tyres at the same time, and the second one is due to the acceleration in the longitudinal axis. For the first example, the spring damper system of the vehicle's suspension should be able to quickly react and absorb the perturbance if tuned correctly. For the second one, the control of the vertical motion can be controlled by many mechanisms:

- **Anti-dive/Anti-squat geometry:** "Anti-pitch" suspension geometry design is a way to design the suspension mounting points to move the pitching centre to reduce the pitching movement on the vehicle, therefore reducing the total movement of the axles. The problem with having too much anti pith geometry is the loss of control of the longitudinal weight transfer during longitudinal acceleration, as well as the high stiffness of the vehicle during acceleration and deceleration.

- **Multi-rate springs:** When using multi-rate or progressive springs on a vehicle, one can affect the characteristics of how it reacts to different inputs. By using multi rate springs, one can have a higher stiffness to take care of the more violent vertical movements due to road perturbances while maintaining a different stiffness for the control of the less aggressive heave during pitch.

- **Mode decoupling:** When using a mode decoupled suspension one can design the control of the different movement modes differently by selecting different spring rates and dampening coefficients.

Roll is defined as the rotation of the vehicle in the longitudinal axis. The control of the roll of the vehicle is a fundamental part of the design of a suspension system as roll is the main responsible for wheel movement during steady state cornering. This means that in order to obtain the maximum cornering performance of a vehicle, its roll control must be optimized for cornering. Other problems with excessive roll are the destabilization of the vehicle or modification to the aerodynamic platform.

Roll control is traditionally done by the design of anti-roll bars which in turn define the roll stiffness of the vehicle, as well as control the lateral load transfer during cornering. This design is mainly good but has two main drawbacks that are the influence of individual corner stiffnesses and the inability to independently dampen the roll response. To optimize this design one can, try to decouple the individual corner stiffnesses from roll stiffness as well as incorporate a dampening element into that decoupled system.

2.2.3 IFS-06

The IFS-06, the sixth prototype developed by the ISC Formula Student team. Designed for the 2023-2024 season, it is the basis for the design of this project. The prototype is a single motor electric prototype with a steel tube space frame chassis.

The cars suspension system has a double wishbone pushrod setup in both axles with a z-type anti roll bar in the front axle and a u-type in the rear axle.

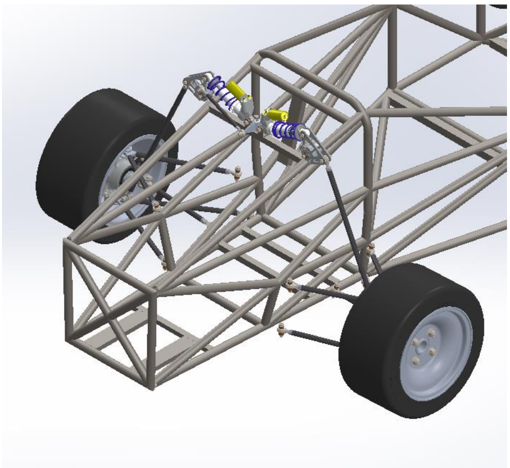

Figure 20. Close up of the ISF-06 front suspension

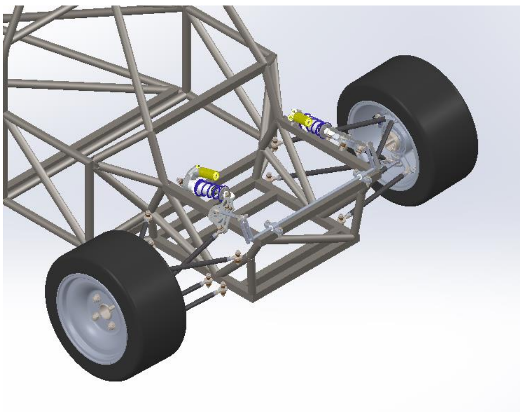

Figure 21. Close up of the IFS-06 rear suspension

Its design metrics were:

- Better camber control, with a static camber of 2.5 and 2 degrees for the front and rear axles.

- Integration of an anti-dive of 30% and an anti-squat of 20%.

- Reduction of the wheelbase and track to 1570 and 1200 mm respectively.

- Optimization of the Motion ratio to 1.2 in both front and back.

- Minimization of roll centre migration during cornering.

- Better control of steering forces.

- Change to Hoosier R20 Tyres.

- Turning circle reduction to 3.9m.

- Incorporation of a 100% Ackerman steering.

- Better roll control on the rear axle by reducing roll stiffness and roll gradient.

This design will serve as the basis for the new decoupled setup designed in this thesis.

Chapter 3. DESIGN

3.1 DEFINITION OF DESIGN OBJECTIVES

The main objective of the project is, as stated in the introduction: The design and integration of a new front suspension system which will be able to control independently the vertical (longitudinal) dynamics of the front axle from the lateral dynamics, with the aim being the improvement of the cornering dynamics of the vehicle during all the stages: braking/corner entry, steady state cornering and accelerating/corner exit. This design should also be easy to manufacture and assemble, as well as be easy to access to change the parameters, finally the design should try to lower the height of the centre of mass of the vehicle.

This main objective of the project can be projected into the specific design considering various points of view of what the expected outcome is. This can be divided into three different and distinct types of objectives: The team’s objectives, the physical objectives and finally the main performance objectives.

3.1.1 TEAM'S OBJECTIVES

To begin describing the objectives it is fundamental to understand the team’s objectives, as this project is done with the ISC Formula Student racing team. This means that the team’s objectives should be considered in every design decision to act as a method of validation these decisions.

The first and most important objective to have in mind is that this project is part of the design phase for the IFS-07, next year’s prototype, which has not been designed yet. This means that even though the physical design has been done based on the chassis of the IFS-06, it should be able to adapt to next year’s design and therefore, the most important features should be parametrically designed, and easily changed to accommodate possible changes to the design of both the chassis and the suspension geometry.

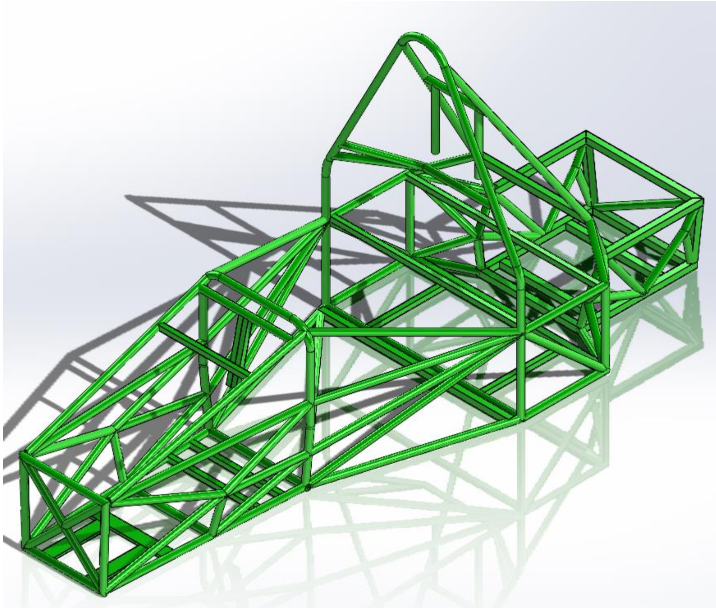

Figure 22. CAD model of the current IFS-06 chassis, the basis for the design

The other general team objectives are more directed to the prototype and the team as a whole. These general objectives should be taken into account when designing but should not be the main drivers of the design. The objectives can be divided into the season objectives, which are to improve the total team score on the competitions which were visited last year. This can be achieved by a better understanding of how we can obtain better scores on the different events and approach the competitions more as the most important parts of the season and not the way to end the season, as well as obtaining a better performance in both dynamic and static events. The other part of the team's objectives are the prototype's objectives, which have been set to mainly improve the understanding, verification and design validation of the vehicle by both more testing and a more thorough understanding of our design philosophy and manufacturing methods. The other main objective for the prototype is trying to reduce the centre of mass height, together with the total mass of the vehicle and can be approached by all departments by trying to simplify components and systems.

Based on these objectives, the conclusions we can obtain for the design are that it should be easily modified to be adapted to the possible different chassis and suspension geometry, it should try to lower its centre of mass, and it should not only allow the team to better understand the car but also it should allow the car to achieve a better on track performance.

3.1.2 PHYSICAL OBJECTIVES

The physical objectives of the design are mainly based on three things, the kinematics of the suspension, the dynamics of the suspension, and finally the physical constraints defined by both the existing design and the ruleset defined by the competition.

Firstly, we will discuss the kinematic objectives of the new design. This new design should be able to incorporate into any suspension geometry without altering the desired kinematic characteristics, this means that the suspension with and without the roll heave system should respond kinematically in an identical manner. To correctly validate this, the geometries should be analysed in kinematic design programs and the camber gain, toe increase and general wheel position during all of its travel should be identical independently of the design utilized. Also, regarding kinematics, the objective for the motion ratio will be set to 1.2 to better utilize the spring-damper system. The last point regarding kinematic design is the importance of maintaining a common plane of movement for both sides of the suspension, which will define the plane where the roll-heave system will be mounted. The failure to obtain the same plane will result in the appearance of unwanted forces which would affect the design.

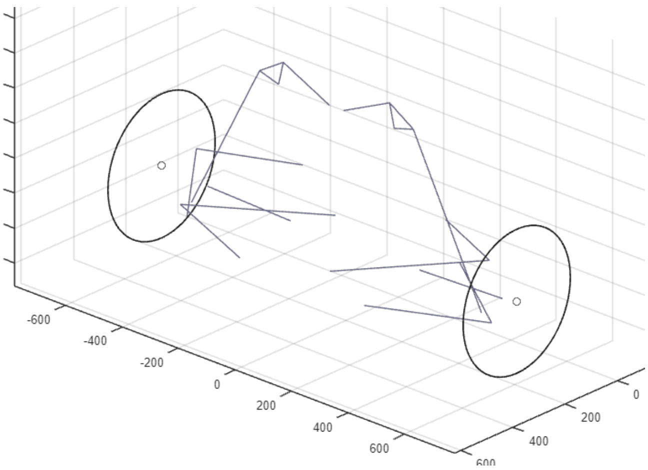

Figure 23. The IFS-06 Front suspension in the Kinematic 3D Analysis MATLAB app

The dynamic objectives of the design are: Firstly, and most importantly to be able to produce a design which can effectively decouple the heave form the roll to better control each movement individually. Once this general design is achieved, the mathematical modelling of the system as an individual axle to study the effects of the different rates on both heave and roll should be obtained. Through this model the real roll and heave stiffnesses of the vehicle should be parametrically obtained. Based on these objectives, the testing campaign for the vehicle should be carried out and the different responses and models verified. Some general objectives for the system are natural heave frequencies higher than 2 Hz to achieve a better handling car and a roll gradient around 1 deg./g. [3],[4],[5]

Finally, we will discuss the physical objectives, these can be divided into two groups:

The objective to reutilize the most possible amount of pre-existing hardware for both assembly and design. This means to be able to design the system in such a way that the way it is assembled is similar to the rest of the systems in the vehicle utilizing mainly M6 or M5 shoulder screws with nylon locking nuts. And designing the systems in such a way that supports and other machined or bought hardware can be reused. This facilitates and cheapens the switch to the new system, as the hardware needed doesn't change. The other main component which can be brought from the previous design is the spring-damper element, this would also reduce the costs of the design as well as maintain the mounting hardware common to both designs.

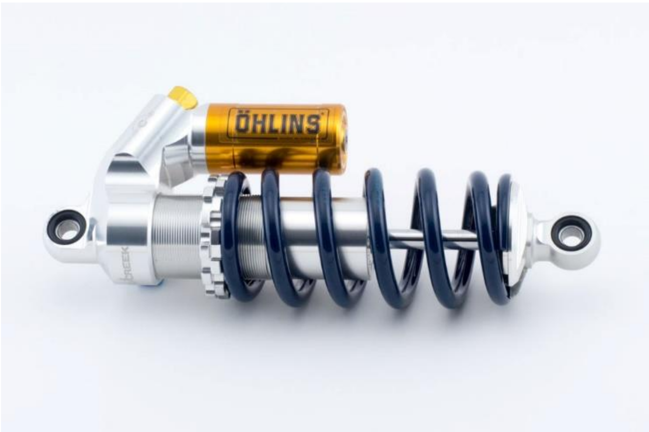

Figure 24. The current spring-damper element

The objective to fit in the design envelope defined by both the ruleset constraints and the physical constraints too. This design envelope will be further defined when describing the physical design phase. The design also should be defined with the ease to assemble, and modify parameters in mind, so when working on the car it is not an inconvenience to alter suspension parameters. This design objective is fundamental, as easy to change parameters are fundamental for the team to correctly understand and validate the design by modifying the behaviour and correlating it to the model.

3.1.3 LAP TIME OBJECTIVES

The lap time objectives have been mainly taken into account for the main dynamic response design of the vehicle. The lap times analysed were last year's Formula Student Spain Skidpad times, which before penalties the best time was 5.6s. Analysing the lap, high levels of oversteer produced by a very stiff rear ARB were observed, based on this analysis, the lap time objective of 5.4s was defined. This time in the skid pad event can be directly corelated to the lateral acceleration the vehicle can withstand, for the objective time this acceleration is equal to 1.26 g of lateral acceleration.

Figure 25. Image of the car oversteering at last year's Formula Student Spain Skidpad event

Setting this acceleration as the objective, the dynamic behaviour of the car can be designed to achieve its maximum performance at that lateral load.

Skidpad time was the selected lap time as it is the event where the most important factor for vehicle performance is the lateral dynamics of the vehicle. This specific Skidpad time also showed the limits of the performance of the vehicle, as during the following attempts, the prototype spun out. Therefore, this objective time tries to address those problems by optimizing the lateral dynamics if the vehicle for those accelerations. This means that the objective for the design of the vehicles is to achieve the optimum camber at the objective acceleration to be able to maximize the tyre grip. This camber objective is set to around 3 degrees, understanding this amount of camber to be the optimum for the tyre from studies conducted last year. Considering the static camber of the wheels and the camber gain of the front axle the roll angle defined is around 1.2 degrees to obtain the desired camber, taking our objective acceleration we can define a front axle roll stiffness of around 1 degree per g.

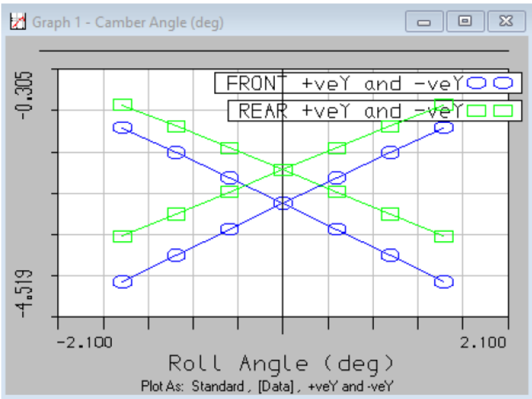

Figure 26. Camber gain during roll of the current suspension geometry

Another notable point to understand behind this lap time objective is that the suspension system has already been modified by introducing a softer rear anti roll bar, so the lap time might have already been lowered, but it is currently untested. The objective therefore should also be met by a parametric design of the dynamic response of the vehicle.

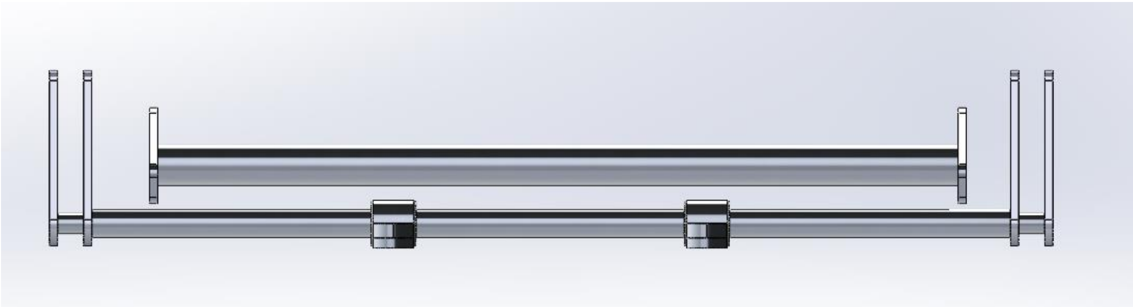

Figure 27. Diameter comparison between the old (upper) and the new (lower) rear ARB

3.2 DYNAMIC RESPONSE DESIGN

3.2.1 DESIGN CONSIDERATIONS

To define the dynamic response of the vehicle, first we must define the dynamic system equivalent to the model. This means that the total vehicle behaviour will be represented by the relative movement of its tyres and how they respond to different perturbances.

To be able to simplify the diagrams, the system will be modelled as a combination of two models, one for each mode of movement which work independently.

Firstly, we will discuss the heave system. In heave, both wheels simultaneously move, and this movement is supported by the compression of the heave spring-damper system. Therefore, the modelling of this movement can be simplified to a two degrees of freedom model where the sprung mass of the front axis is supported by the heave spring damper system which connects it with the unsprung mass which in turn is connected directly to the road through the wheels which are modelled as a spring with its total rate being the parallel of an individual vertical wheel rate.

Through this model, we can not only study the general behaviour of the front axle of the vehicle while on heave but also the response of the wheel hop effect, a resulting factor of the unsprung mass of the vehicle having a different response to the main sprung mass.

Through the simplification of this model, the complete heave behaviour of the front axle can be studied in the same manner to the quarter car model that is greatly studied.

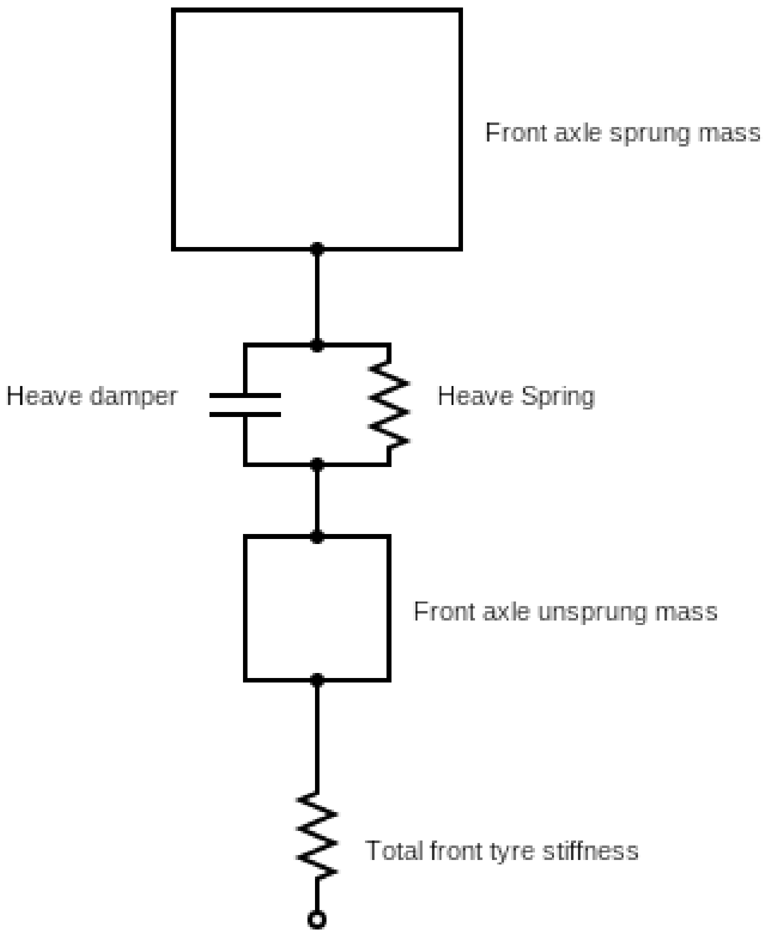

Figure 28. Front heave Dynamic model

For the roll movement on the front axle, the system has been modelled differently, as the focus of the roll control in not understanding and controlling perturbances or very sudden violent loads, this axle can be modelled as a torsional spring equivalent to the torsional stiffness of the vehicle, and the damper can be modelled as an equivalent damper to the one design. The mathematical formulation for these equivalences will be further described.

This approach as a system with a single degree of freedom for the roll movement helps better understand how the different spring rate used in the roll spring-damper system directly affects the roll stiffness of the axle. Other important property of this model to understand is that it describes roll as the difference in position of both wheel centres of the same axle.

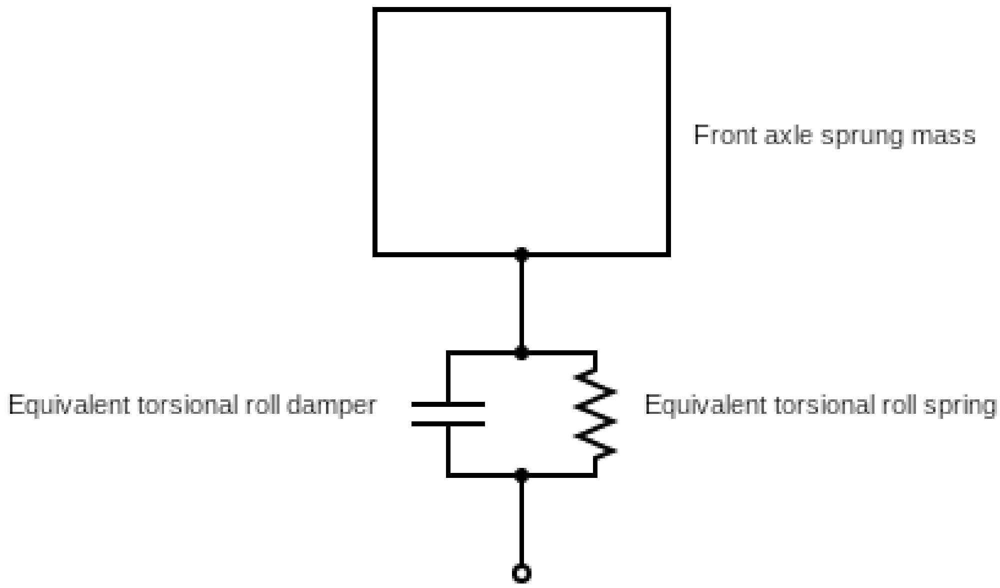

Figure 29. Front roll Dynamic model

3.2.2 DYNAMIC RESPONSE DESIGN

The dynamic response was designed and optimized using MATLAB. All the scripts are based on firstly obtaining the correct stiffnesses and damping factors on the wheels (utilizing the different installation ratios to obtain the correct "real distance factors"), then by simulating an applied perturbance either solving the differential equation system for the system with multiple degrees of freedom or the simple equation and obtaining the graphical response and analysing it.

The heave model has four main parameters which are: front sprung mass, front unsprung mass, front heave stiffness, front tyre stiffness. Utilizing these parameters to calculate the dynamic response of the vehicle to a movement. The code used was this:

```
clear all
clc

% Variables
m = 330; % [kg] total mass
mfu = 15; % [kg] unsprung front mass
mf = 100-56; % [%] total front mass percentage
ir_h = 1.2; % instalation ratio for the front  heave spring
kw = 110000*2; % [N/m] tyre stiffness
kfhs = 43780*2; % [N/m] heave spring stiffness
cfhd = 1000*2; % [N*s/m] heave damper damping coeficient



kfhw = kfhs*ir_h^2; % damping coeficient in tyre

cfhw = cfhd*ir_h^2; % damping in the tyre

mfs = m*mf/100-mfu; % frunt sprung mass

% Mass matrix
M = [mfu , 0;
    0 , mfs];
% Stiffness matrix
K = [kw+kfhw,-kfhw;
    -kfhw,kfhw];
% Damping matrix
C = [cfhw,-cfhw;
    -cfhw,cfhw];

% definition of oscilation modes and frequencies
[Phi,omega2] = eig(K,M);

ti = 0:pi/1000:2;
ti_2 = 0:pi/10000:pi/10;
pr_1 = Phi(1,1)*sin(ti*sqrt(omega2(1,1))+3*pi/2);
pc_1 = Phi(2,1)*sin(ti*sqrt(omega2(1,1))+3*pi/2);
pr_2 = Phi(1,2)*sin(ti_2*sqrt(omega2(2,2))+3*pi/2);
pc_2 = Phi(2,2)*sin(ti_2*sqrt(omega2(2,2))+3*pi/2);

figure
plot(ti,pr_1,'r'),hold on
plot(ti,pc_1,'g'),hold on
plot(ti,(pr_1+pc_1),'b')

figure
plot(ti_2,pr_2,'r'),hold on
plot(ti_2,pc_2,'g'),hold on
plot(ti_2,(pr_2+pc_2),'b')

% mode equivalent calculations
m1 =transpose(Phi(:,1))*M*Phi(:,1);
m2 =transpose(Phi(:,2))*M*Phi(:,2);
```

```
k1 =transpose(Phi(:,1))*K*Phi(:,1);
k2 =transpose(Phi(:,2))*K*Phi(:,2);
c1 =transpose(Phi(:,1))*C*Phi(:,1);
c2 =transpose(Phi(:,2))*C*Phi(:,2);

%modal contributions
rang_t = [0. 2];
d0 = [0.0101325 0.];
solu_1 = ode45(@(t,y)SDoF(t,y,m1,k1,c1),rang_t,d0);
v_t = (0:0.01:2);
y_1 = deval(solu_1,v_t);
figure
plot(v_t,y_1(1,:)), hold on

d0_2 = [0.1 0.];
solu_2 = ode45(@(t,y)SDoF(t,y,m2,k2,c2),rang_t,d0_2);
y_2 = deval(solu_2,v_t);
v_t_2 = (0:pi/2000:pi/10);
figure
plot(v_t_2,y_2(1,:)), hold on

y_fin = y_1+y_2;
figure
plot(v_t,y_fin(1,:))

figure
plot(v_t,y_1(1,:)), hold on
plot(ti,(pr_1+pc_1))

frechz=(sqrt(omega2))/(2*pi)

function der = SDoF(t,y,m,k,c)
der(1,1) = y(2);
der(2,1) = (1/m)*( - c*y(2)-k*y(1));
end
```

And the resulting parameters obtained this simulation graph.

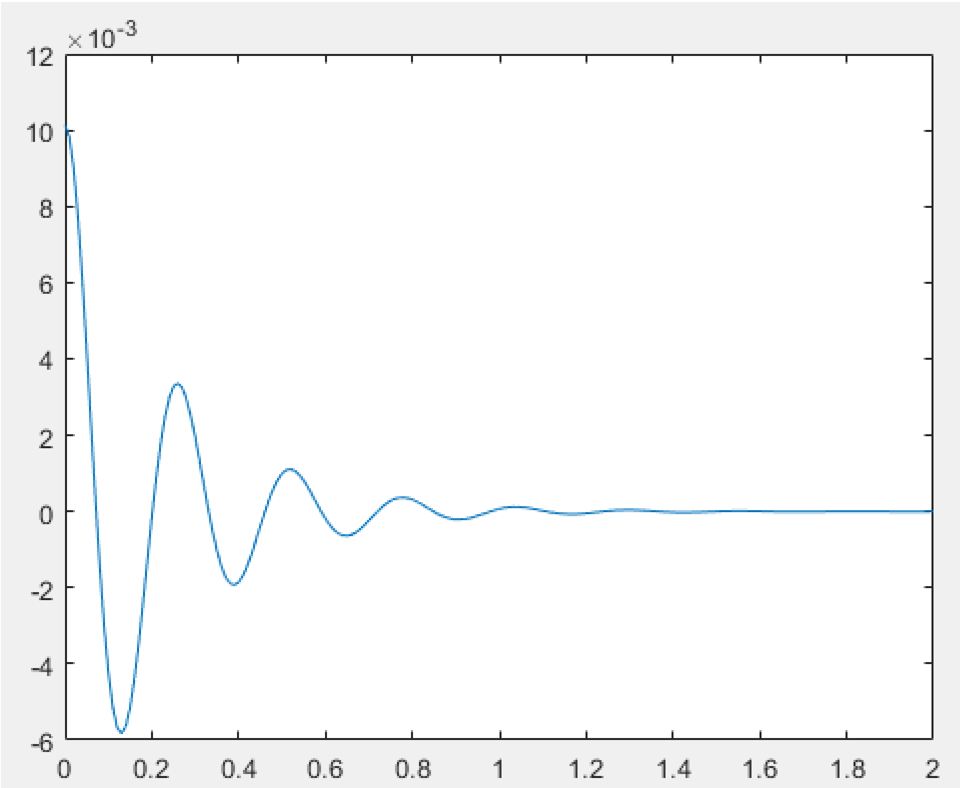

Figure 30. Graph simulating a 1 cm disturbance in heave

In this graph we see a simulated heave movement of 0.01m which is completely controlled in under 1.5 seconds with the second peak having a reduction of in amplitude of over 50%. This control was designed with the dampening of the dampers set to the medium position (1000 Ns/ m) and the stiffness of the spring being set to 43780 N/m. With these parameters the resulting frequencies for the movements were 3.91 Hz for the vertical movement and 24.36 Hz for the wheel hop, these frequencies are well within the desired range, and, as seen by the graph help the system act fast enough. This design is also coherent with the whole car as the rear axis has a higher main frequency of 4.2 Hz which is desired to better settle the whole car.

This code, as previously describes firstly obtains the wheel rates of the suspension and then by assembling the different mass, stiffness and damping matrixes obtains the dynamic response of the system to the simulated perturbances.

Regarding the roll control, Firstly, the equivalent front roll stiffness based on the objective of the front roll gradient of 1 degree per g was calculated using this formula

$\frac{\phi}{A_Y} = \frac{W_F \times H}{K_{\phi F}}$ [3] where $\frac{\phi}{A_Y}$ refers to the front roll gradient, H to the front weight and W_F to the total front stiffness in the wheels, obtaining a front stiffness in the wheels of 8854.8 rad/N which corresponds to a total front stiffness for the roll spring of about 10000N/m

The simulation for the validation of the roll control was done comparing the behaviour of the front axle with different roll gradients while subjected to an acceleration of 1.26g with a damping factor of 0.7. The amount of total roll of the body was the parameter observed, which we wanted to be around 1.2 degrees to obtain the desired camber gain.

The code used was this:

```
close all
clear all
clc

% Definition of variables
rpg_old = 1/0.6; %roll gradient old
rpg_new = 1/1; %roll gradient new
seta = 0.7; % damping
m = 280; % mass
a =1.26; % acceleration
hcg = 0.3; % com height
f_eq = a;

k1 = rpg_old;
k2 =rpg_new;

t0 = 0.;
tf = 200;
rang_t = [t0 tf];
d0 = [0. 0.];



aux_t = (t0:0.001:tf)';
aux_f = ones(size(aux_t,1),1)*f_eq;



solu = ode45(@(t,y)SDoF(t,y,m,k1,seta,aux_t,aux_f),rang_t,d0);
```

```
solu2 = ode45(@(t,y)SDoF(t,y,m,k2,seta,aux_t,aux_f),rang_t,d0);

v_t = (t0:0.01:tf);
y = deval(solu,v_t);
y2 = deval(solu2,v_t);

figure
plot(v_t/9.81,y(1,:)), hold on
plot(v_t/9.81,y2(1,:)), hold on
ylabel('degrees of roll angle gained')
xlabel('time under acceleration = 1.26g')
legend('old ARB','new ARB')


function der = SDoF(t,y,m,k,seta,aux_t,aux_f); omega_n = sqrt(k/m);
c_crit = 2*omega_n*m;
c = seta*c_crit;
f = interp1(aux_t,aux_f,t);

der(1,1) = y(2);
der(2,1) = (1/m)*(f - c*y(2)-k*y(1));
end
```

And the parameters observed were these:

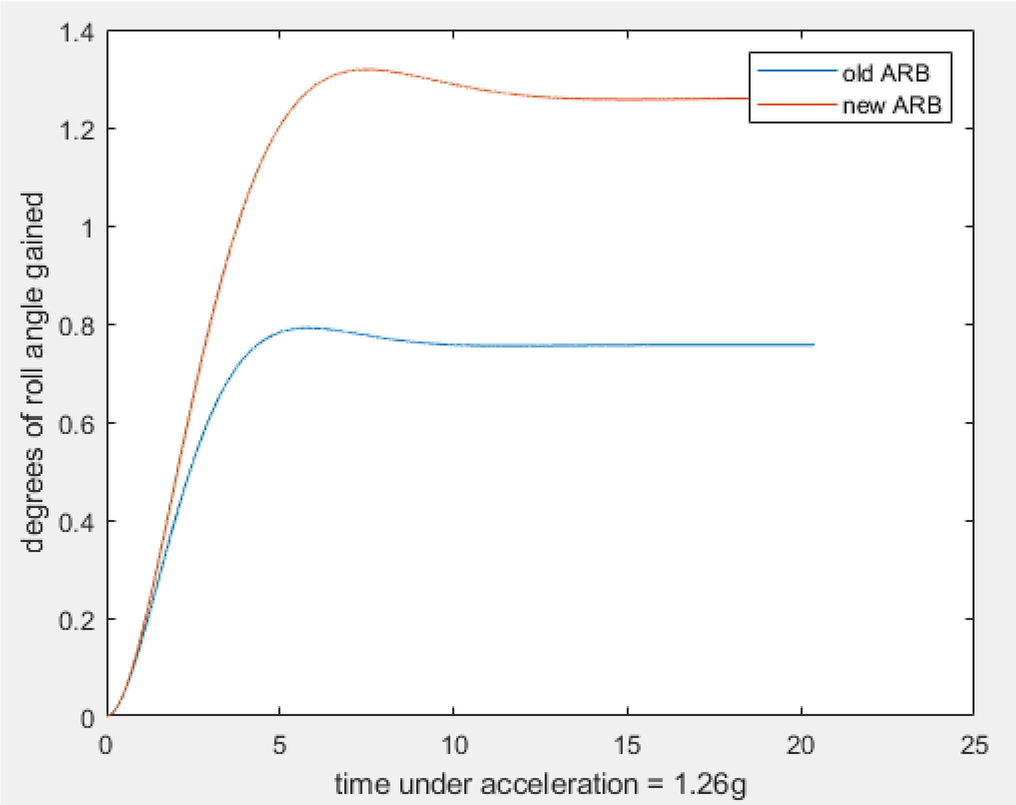

Figure 31. Graph simulating a 1.26 g lateral acceleration for roll

This resulting graph shows the difference in behaviour between the previous setup with a gradient of 0.6 degrees per g and the current one with a gradient of 1 degree per g. In the graph one can how even though the new more flexible system is slower, it better conforms to achieving the desired objective of 1.2 degrees of roll for the 1.26 g turn. This in turn means that the correct choice of the stiffness of the roll spring is around the 10000 N/m value.

3.3 PHYSICAL DESIGN

3.3.1 PHYSICAL DESIGN CONSTRAINTS

The constraints due to ruleset, the ruleset defines areas where no parts are allowed to go. For the design of a suspension system on the front axle the most crucial area is the inside of the cockpit. The ruleset states that the cockpit must have a free internal cross section defined by a template in which no components must intrude. This rule constraints the possible design for a suspension to fit through the cockpit to a smaller design area.

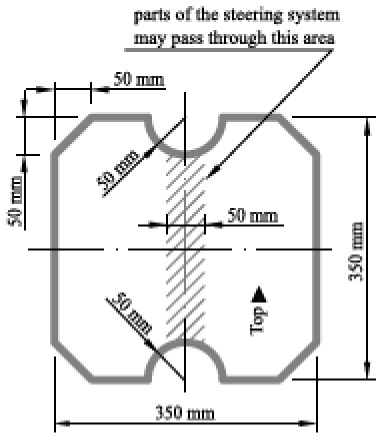

Figure 32. Internal cockpit cross section template

The constraints due to the base design, the chassis in which the design is being based has been taken as a hard point allowing only changes to the chassis structures which were designed to house pre-existing suspension components. This coupled with the decision to design the system based on the previous geometry defines the possible spaces where a system like this can be housed. Through looking at the possible designs, a final design template was obtained which considered both the ruleset and previous design and defined the zone inside the cockpit where the system could be housed.

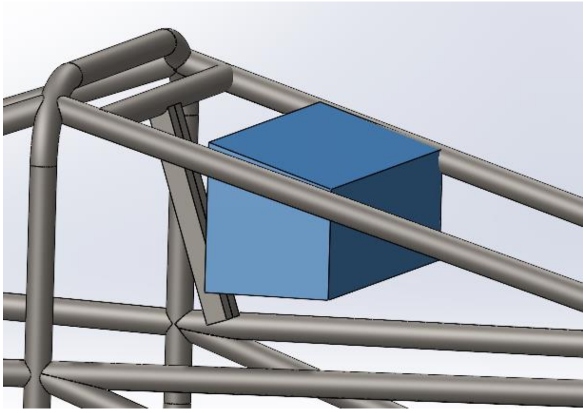

Figure 33. Design template for the front suspension system into the cockpit

For the final design constraints, the current materials as well as the fabrication methods we have available have been taken into account. The first is that we already have great spring-damper elements which can be incorporated into the design, secondly is that the fabrication technologies we have available are reduced, as we don't have access to more modern methods cush as additive manufacturing for metal. Multi axis CNC machining is the most versatile method we have access to, but it is still needed to be sourced to an outside machining shop, if we wanted to do the fabrication in house, it would need to be either water cut or very simple machining.

3.3.2 PHYSICAL DESIGN OF THE COMPONENTS

To begin defining the physical design, an overview of the general design will be made. The general design for a roll-heave decoupled front suspension consist of three main parts: The rockers, the heave spring and the roll spring.

The general overview of the design consists in the separation of the roll spring and the heave spring so that they are actuated only during their respective movements. For roll that is when both tyres move opposite to each other and for heave when both tyres rise or fall simultaneously. To decouple the roll from the heave, a system can be designed that compresses the heave spring only when simultaneous movement of both tyres occurs. Such a design can be achieved by connecting the spring the top sides of both of the rockers together so when the tyres rise from both sides simultaneously the spring is compressed. To decouple the roll control, a similar design can be implemented. The top side of one rocker will be connected to the bottom side of the other rocker by a spring so when the tyres move in opposition, the spring is actuated upon.

The following diagrams illustrate how the general design works.

Figure 34. Diagram showing the compression of the heave spring and no reaction of the roll spring when both tyres move equally. [16]

Figure 35. Diagram showing the compression of the roll spring and no reaction of the heave spring when both tyres move opposite. [16]

As it can be seen this system can effectively control independently roll and heave. The drawbacks of the system are in general two: First the need for a spring that works on both compression and elongation for roll control and the second one is the inability to control individually wheel perturbation.

For the first drawback, a “double spring” system can be designed so that the spring can act in compression on both sides of roll. This can be achieved by enclosing the spring-damper element in a device which can compresses the element independently of whether its sides move toward or apart from each other.

For the second drawback, the individual perturbation cannot be independently controlled due to the interconnection between both sides of the vehicle. This drawback is also common in most suspensions and could only be addressed either by completely disconnecting the tyres form each other, which would result in the loss of roll and heave control and therefore is much worse for the control of the vehicle; of by the designing of a fully decoupled suspension which involves the fabrication of a hydraulic system to be able to fully independently control each tyre as well as each movement.

The final design for the roll-heave system was designed to be able to fit inside the design envelope inside the cockpit due to this design reducing the maximum heigh of the centre of mass, one of the main objectives of the prototype. In order to fit inside the envelope, the design needed to fit the pushrods through the chassis, as well as eliminate the bar currently in place to hold the front ARB and spring-damper elements.

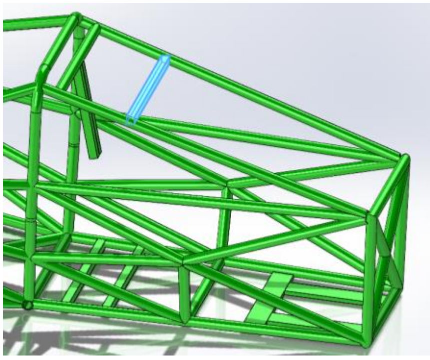

Figure 36. Image of the CAD of the chassis with the bar to be deleted highlighted

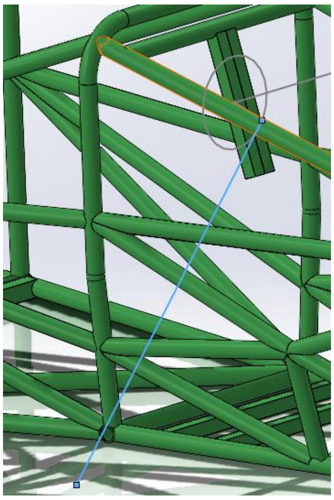

Figure 37. Image of the CAD of the chassis with the position of the pushrod highlighted

3.3.2.1 Rockers

The rocker or bellcrank is the component designed to transmit the wheels motion into the compression or elongation of the spring. It consists of a pivot point, and three arms. One arm where the pushrod actuates, one for the actuation of the heave spring and one for the actuation of the roll spring.

For the correct design of the rockers, an analysis of the motion ratio is needed to determine the correct position of the pivot point, as well as the relative lengths of the different arms. The desired Motion ration for heave was of 1.2 and for roll was of 0.8 to obtain the desired stiffnesses on the wheels. In order to obtain these parameters a study was made on the possible position of the rocker and the relative length of the arms to optimize during all of the travel the motion ratios. This analysis was performed in an app designed in MATLAB which calculates the dynamic motion ratio of the suspension. The results were these two graphs showing the motion ratio for the suspension though all of its travel. The first one being the roll spring and the second being the heave spring.

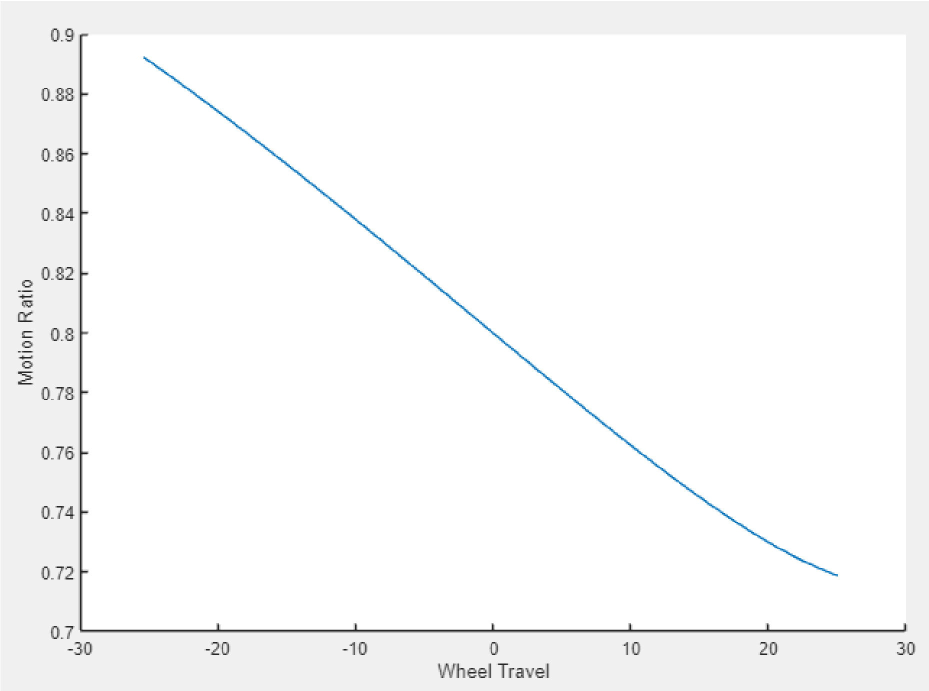

Figure 38. Dynamic motion ratio for the roll spring

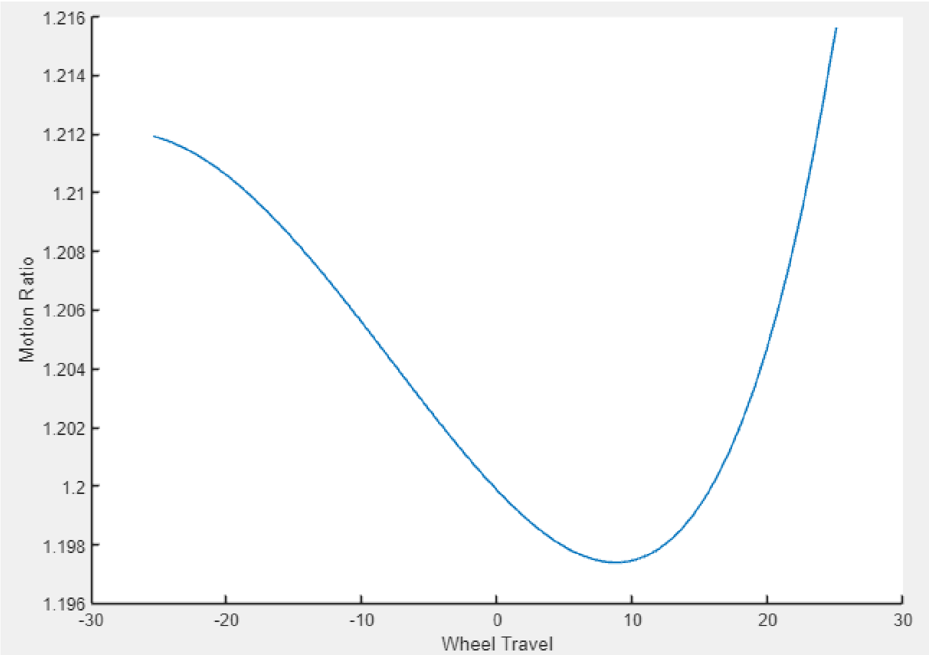

Figure 39. Dynamic motion ratio for the heave spring

As it can be seen in both tables the motion ratio objective is achieved with an error of less than 0.1 in the roll ratio, with the heave ratio being practically continuous throughout the whole travel.

Form this optimization of the motion ratios, the position of the points which define the rocker were obtained.

	x	y	z
Pivot point position	906.6	161.0	488.6
Heave arm length	81.1 mm		
Roll arm length	50 mm		
Pushrod arm length	65.5 mm		
Pushrod-Heave angle	71.21°		
Pushrod-Roll angle	37.6°		

Table 1. Definition of the rocker pivot point and geometry

It is important to note that the points of the rocker pivot point are obtained in reference to the rest of the geometry of the vehicle as defined in the CAD model, and the Pushrod-Roll angle refers to the side where the roll arm is in the same quadrant of the rocker as the pushrod arm.

Once the geometric definition of the rockers explained, the physical design can be approached. The design was done in SolidWorks and the total length of the rocker was determined with the finalised design in order to maintain both springs separated.

The design consists of two rockers to correctly actuate the heave and roll springs.

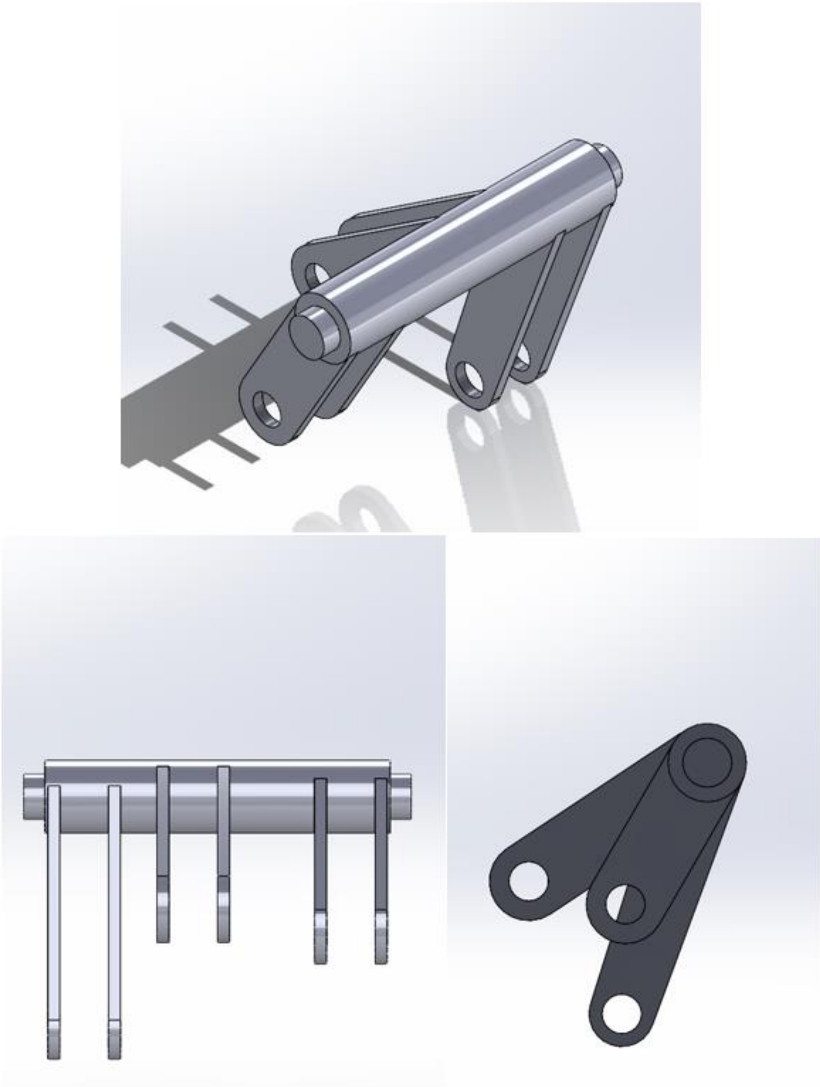

Figure 40. Font, Side and Isometric view of the rocker for the right side of the vehicle

The design of the right rocker, was mainly straightforward, taking knowledge form the experience gained in the previous prototypes and designing the arms of 4 mm thick aluminium. And the sections with the reduced diameter have been designed to house the bearings for the correct functioning of the system.

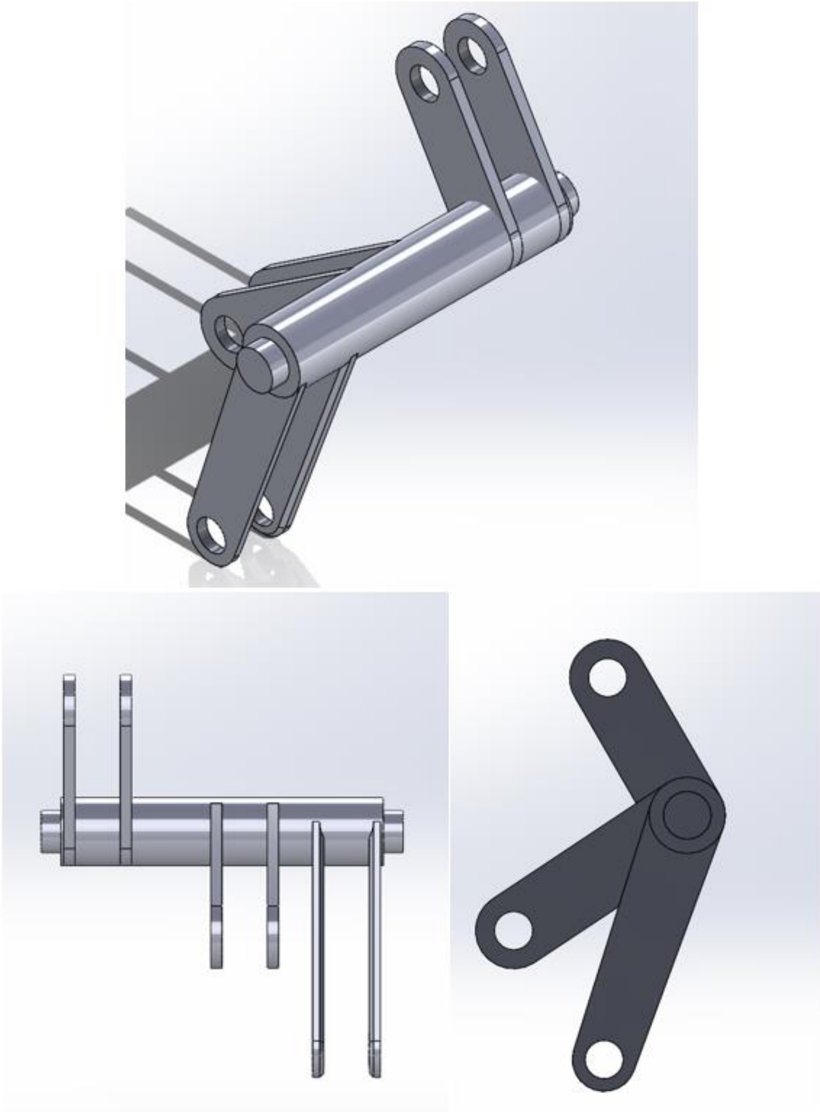

Figure 41. Font, Side and Isometric view of the rocker for the left side of the vehicle

The design of the left rocker not only took the same main cues from the right side, but it also had to be adapted to fit the rear mounting point of the roll damper. This was achieved by incorporating a chamfered section.

The material of the rockers will be 7075 aluminium, as it is an alloy which, not only has great mechanical properties but also a reduces density and is easy to machine. The fabrication method for these main components will be muti axis CNC machining, due to the complicated structures of the arms.

3.3.2.2 Rocker mounting points.

Once the rockers were designed, the mounting brackets for them were the main focus. Taking into account the need for a bearing, one was selected using the web applet designed by SKF bearing selector and looking of a bearing that fitted the parameters of both size and function, with the function being a rotating shaft with a negligible axial load. Once the SKF bearing was obtained, the NTN equivalent was searched, as NTN is one of the sponsors of the ISC Formula Student racing team. The final Bearing selected was the NTN 6902LLU/5K bearing, a single file ball bearing with radial contact and a steel cage with seals on both sides.

Figure 42. Cut view of the NTN 6902LLU/5K bearing

Once the bearing selection was completed, the designed for the Rocker mounting points was finalized. Selecting shouldered M5 bolts for mounting with the design incorporating space for the rounded heads.

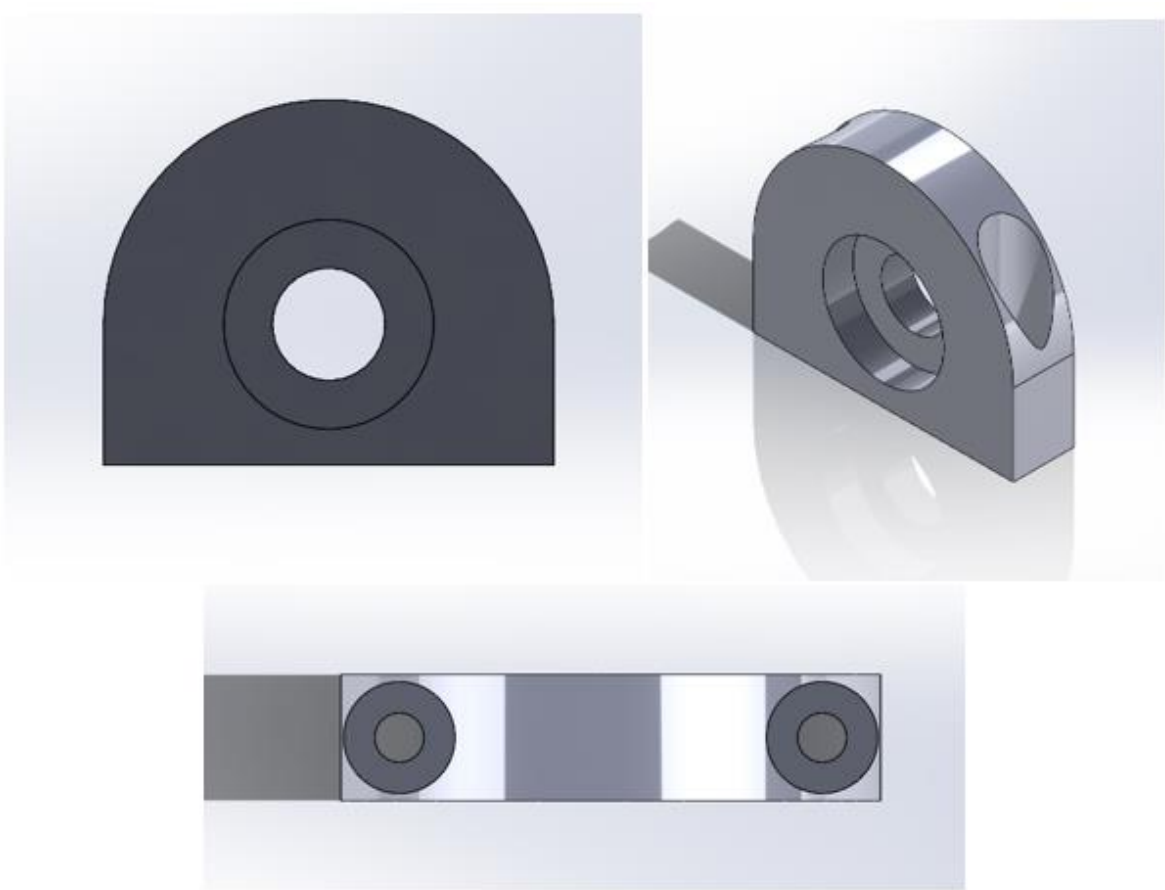

Figure 43. Front, isometric and top views of the rocker mounting point design

Consisting of a bearing housing, and two holes to accommodate screws to mount to the mounting points of the chassis, the design was completely simplified. To achieve a geometry which could be obtained by simple manual machining.

3.3.2.3 "Double spring" mechanism

The next step for the design of the new suspension was to design a “double spring” device which can compresses the element independently of whether its sides move toward or apart from each other.

For this design, two main parts were designed. Firstly, the mounting of the damper and secondly, the mounting points for the rocker. The design consists of the compression of the spring between the two mounting points of the damper by attaching one sides rocker mounting to the others damper mounting so that the rod connecting the two mounting points is only locked at the ends, allowing the mounting points to move towards each other but not away from each other, as the connections are made in a diagonal, both sides can compress when the rocker compresses the device and when the rocker movement elongates the device, the connecting rods compress the damper mounting points together. The mechanism is visually explained by the following diagrams.

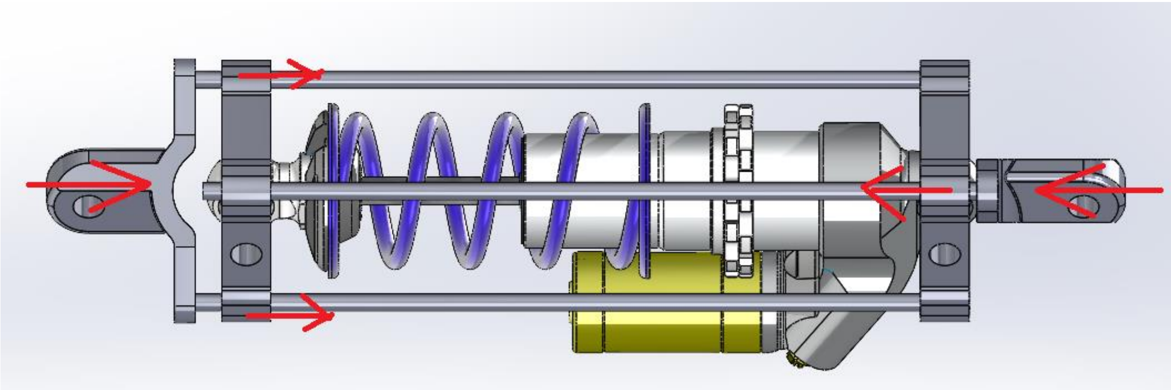

Figure 44. Diagram showing the compression of the "double spring" device and the compression of the spring

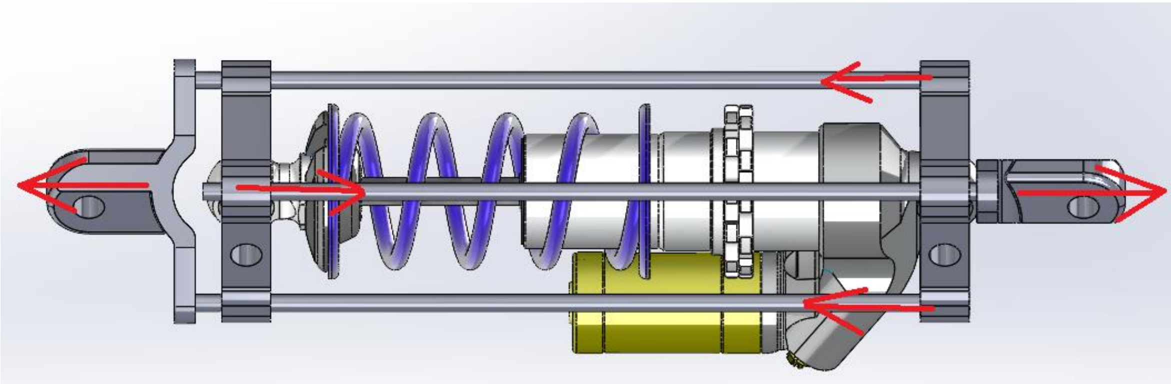

Figure 45. Diagram showing the elongation of the "double spring" device and the compression of the spring

The individual parts of the system were designed to be machined in 7075 aluminium. The rocker mountings were designed to be machined by a multi axis CNC machine to be able to achieve the complicated geometries, and the damper mounting points were designed in a simpler manner to be able to be hand machined, even though a further version could be designed, focused of reducing the weight.

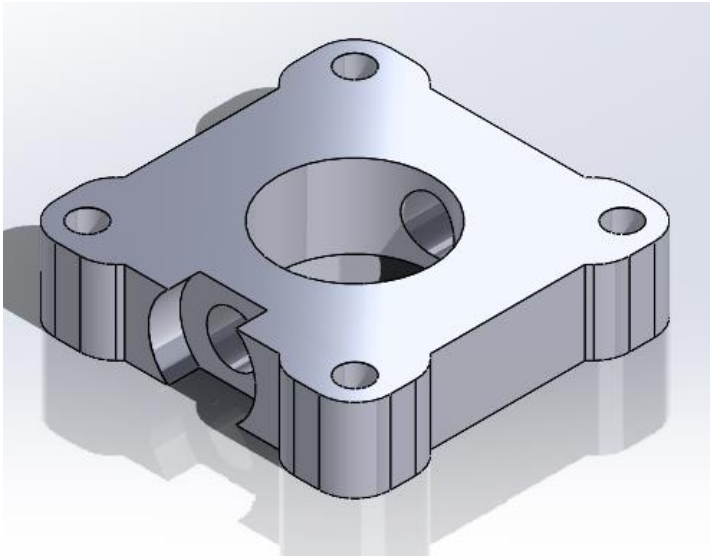

Figure 46. Image of the damper mounting CAD

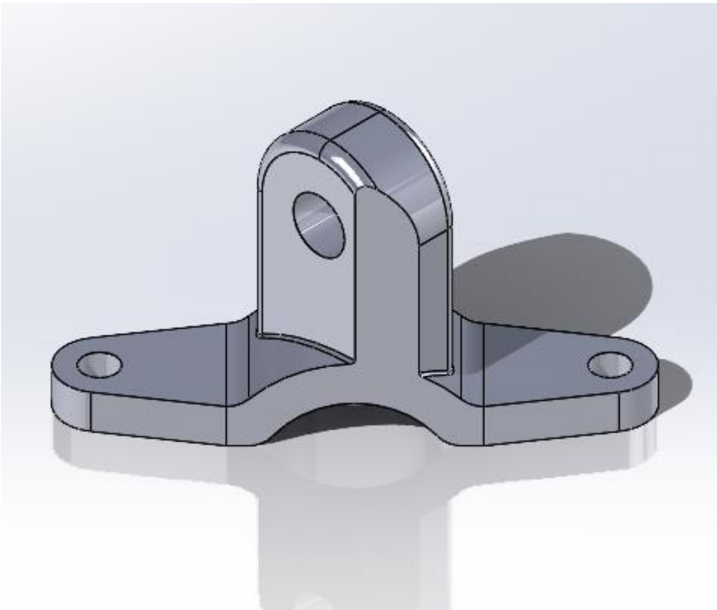

Figure 47. Image of the rocker mounting CAD

3.3.2.4 Heave spring extension

By reusing the spring-damper elements, the static length of the element must be altered to obtain the desired design. For this reason, a new mounting point for the top of the element must be designed to achieve this static length. As this length depends mainly on the overall geometric design, a way to alter the length of the system was implemented by the addition of shims between the elements to increase the static length.

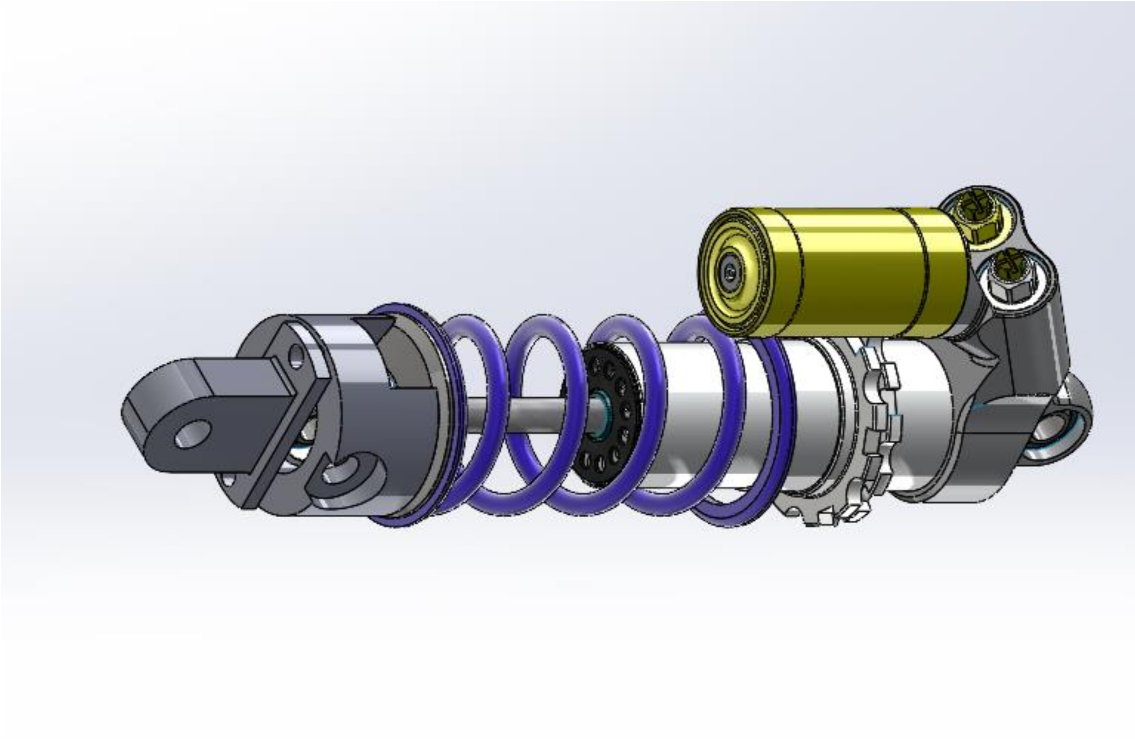

Figure 48. Image of the CAD model of the general design of the heave spring extension mounted on the spring damper system

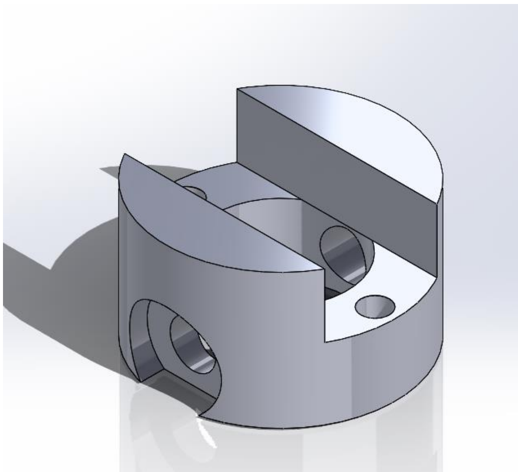

Figure 49. Image of the CAD model of the mounting point for the damper

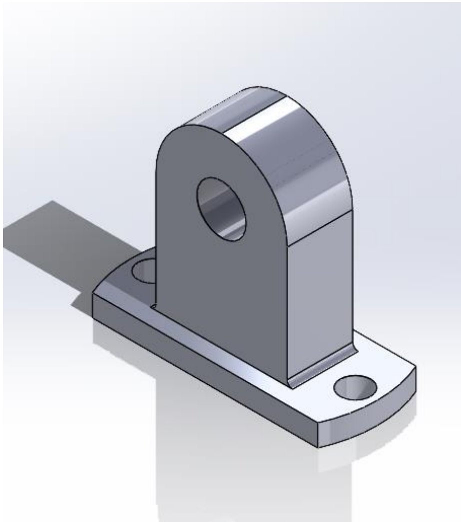

Figure 50. Image of the CAD model of the mounting point for the rocker

3.3.3 FINAL COMPLETE DESIGN

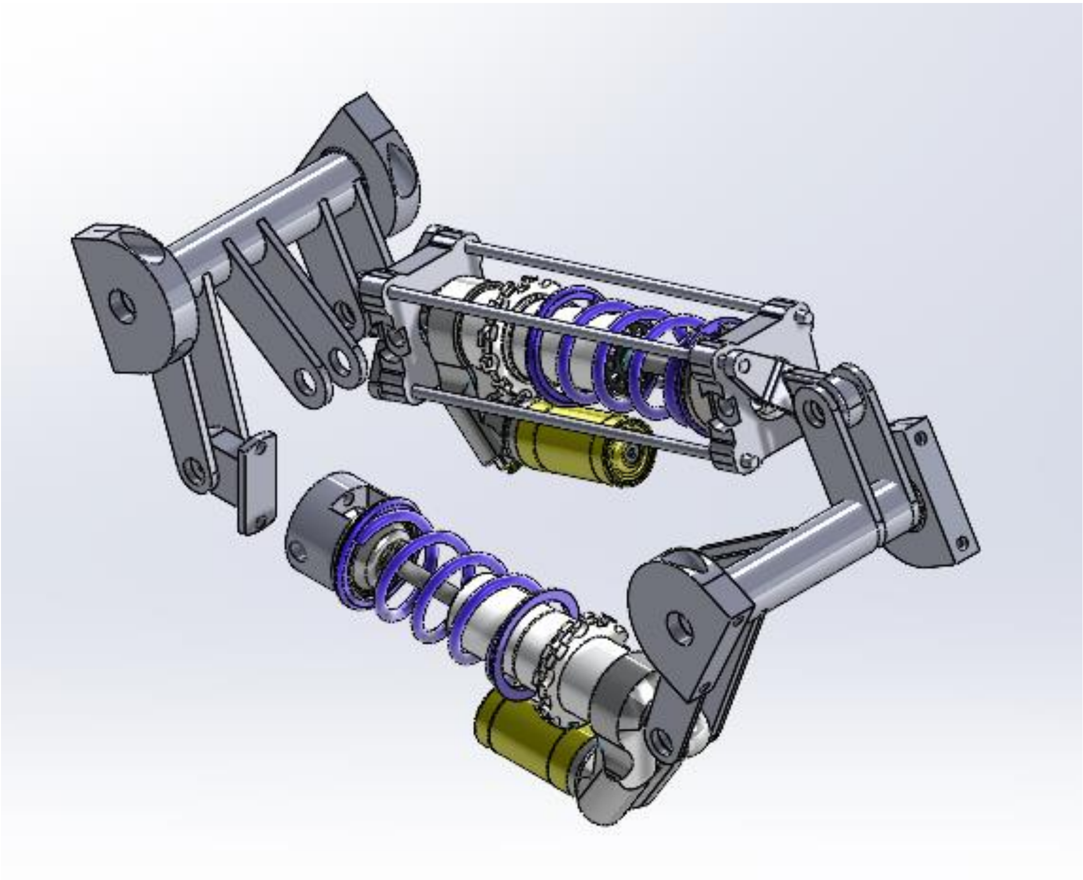

Figure 51. Image of the CAD model of the final design assembled

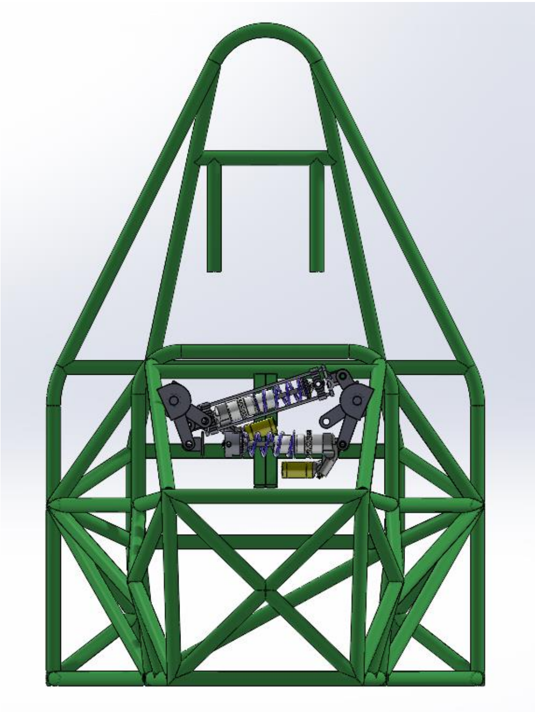

Figure 52. Image of the CAD model of the front plane of the design mounted into the chassis

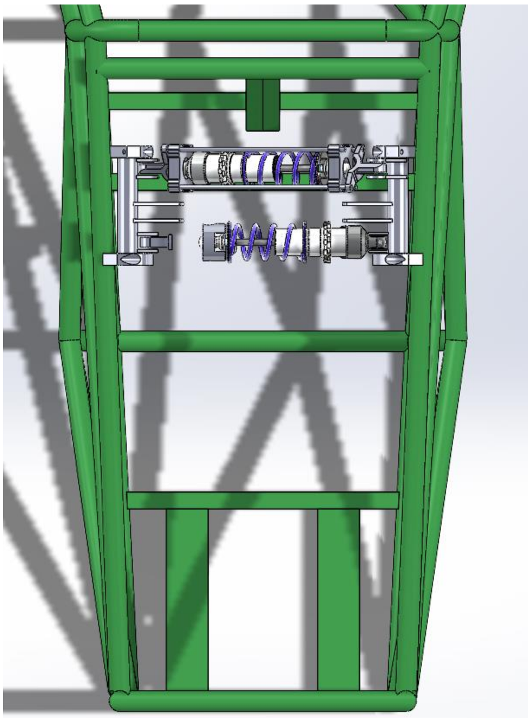

Figure 53. Image of the CAD model of the top plane of the design mounted into the chassis

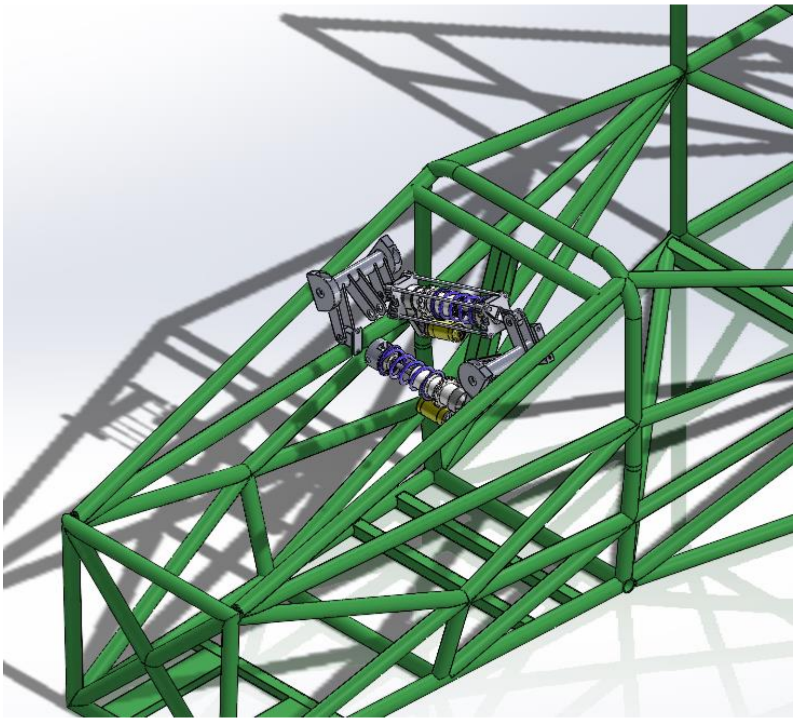

Figure 54. Image of the CAD model of the isometric view of the design mounted into the chassis

Chapter 4. DESIGN VALIDATION

4.1.1 FEA OVERVIEW

Finite element analysis (FEA) is a design tool used in engineering for the simulation of large complex elements that works by subdividing the main elements into smaller elements defined as a mesh. For the validation of the design of the rockers, Mechanical FEA has been used, which simulates fixtures and forces applied to elements designed by CAD. This simulation was carried out in SolidWorks, which includes a CAE (computer aided engineering) toolbox.

The quality of the simulations obtained by FEA can vary greatly, that is why the input of the simulation must be of the highest quality possible.

In order to carry out a simulation, the first element needed is the final design of the piece to be studied. This final design should try to avoid hard edges where possible, incorporating chamfers to gradually change the geometry, as FEA software often have problems calculating elements on hard edges. For the final preparation step for the designed part, the material properties should be selected.

Once the geometry and material are correctly defined, the Object will be meshed and separated into small elements. The size and shape of these elements is crucial for the final result of the analysis, as well as the time and resources it takes to run. In general, the higher number of elements leads to a better result, as well as a higher simulation time, this means that the elements should be smaller to be able to more accurately predict the behaviour of the part, but only in the main important parts.

The final step before carrying out the simulation is to define the boundary conditions. For a mechanical simulation as the ones carried out for this thesis, the boundary conditions must include the fixtures of the part, as well as the forces it will be subjected to. The correct calculation of the forces is a fundamental part of the process, as the unproper selection can lead to a catastrophic failure of the design.

4.1.2 DESIGN ADAPTATION

The design adaptation for FEA of the final rocker models mainly consisted of the rounding of the joint between the actuating arms and the main rotating cylinder for all the arms.

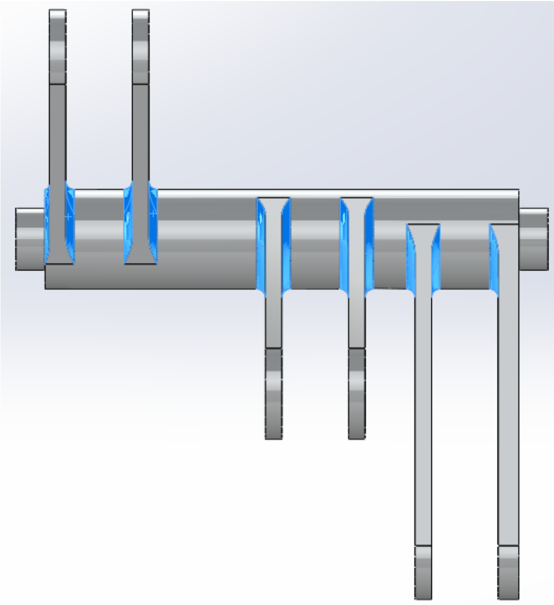

Figure 55. Image of the CAD model of the left-side rocker with the changes highlighted

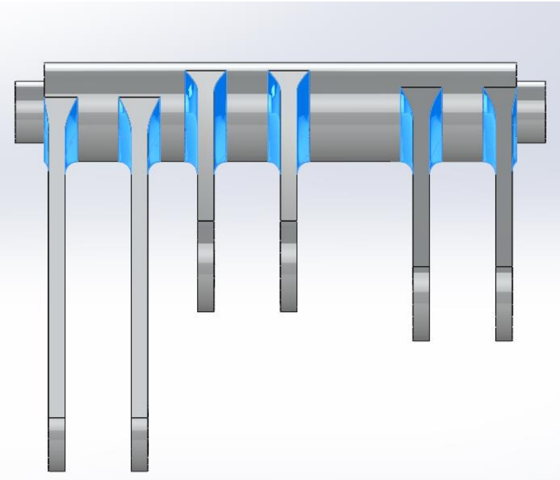

Figure 56. Image of the CAD model of the right-side rocker with the changes highlighted

4.1.3 MESHING

The meshing of the parts was done utilizing SolidWorks embedded meshing software. The first element meshed was the left-side rocker and based on the results observed form the first simulation, the parameters for the second simulation were changed to obtain a quicker less resource consuming result.

Study name	Case 1 (-Config 1-)
DetailsMesh type	Solid Mesh
Mesher Used	Curvature-based mesh
Jacobian points for High quality mesh	16 points
Mesh Control	Defined
Max Element Size	1 mm
Min Element Size	0,2 mm
Mesh quality	High
Total nodes	4981847
Total elements	3523738
Maximum Aspect Ratio	5,5438
Percentage of elements with Aspect Ratio < 3	99,8
Percentage of elements with Aspect Ratio > 10	0
Percentage of distorted elements	0
Number of distorted elements	0
Time to complete mesh(hh:mm:ss)	00:02:27

Table 2. The parameters of the mesh of the left-side rocker

Study name	Case 2 (-Config 2-)
DetailsMesh type	Solid Mesh
Mesher Used	Curvature-based mesh
Jacobian points for High quality mesh	16 points
Mesh Control	Defined
Max Element Size	1,5 mm
Min Element Size	0,25 mm
Mesh quality	High
Total nodes	2060901
Total elements	1441135
Maximum Aspect Ratio	81,98
Percentage of elements with Aspect Ratio < 3	99,7
Percentage of elements with Aspect Ratio > 10	0,00153
Percentage of distorted elements	0
Number of distorted elements	0
Time to complete mesh(hh:mm:ss)	00:00:49

Table 3. The parameters of the mesh of the right-side rocker

Based on the results obtained, it can easily be seen how both meshes have a great quality, with element counts of over a million elements in both and with less than 0.01% of the elements having an aspect ratio of over 10 and over 99.6% having an aspect ratio lower than 3. Which means the mesh correctly transitions between the elements without distorting. It is also important to note that the change in the mesh parameters for the second part resulted in a reduction of time of 66% with a similar overall mesh quality.

Another notable point is that the meshing of the critical points, defined by the unions between the actuating arms and the main cylinder, was controlled to different specifications to obtain a better-quality analysis on the most stressed parts. The parameters for these areas were:

Study name	Case 1 (-Config 1-)
Mesh type	Solid Mesh
Entities	12 edge(s), 11 face(s)
Units	mm
Size	0,15
Ratio	1,4
Identifier	1

Table 4. The parameters of the mesh of the controlled region of the left-side rocker

Study name	Case 2 (-Config 2-)
Mesh type	Solid Mesh
Entities	11 edge(s), 13 face(s)
Units	mm
Size	0,25
Ratio	1,4
Identifier	1

Table 5. The parameters of the mesh of the controlled region of the right-side rocker

As it can be observed in the tables, the parameters of the controlled regions were also altered to help with the resource load.

The final meshes obtained for the parts were:

Figure 57. Mesh of the left-side rocker

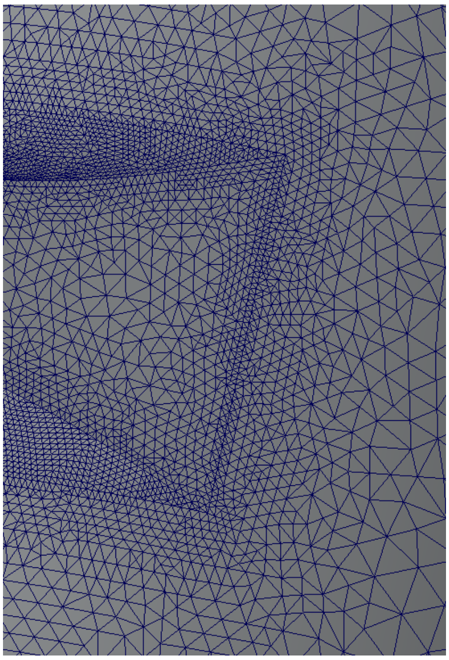

Figure 58. Closeup of the mesh of the controlled region of the left-side rocker

Figure 59. Mesh of the right-side rocker

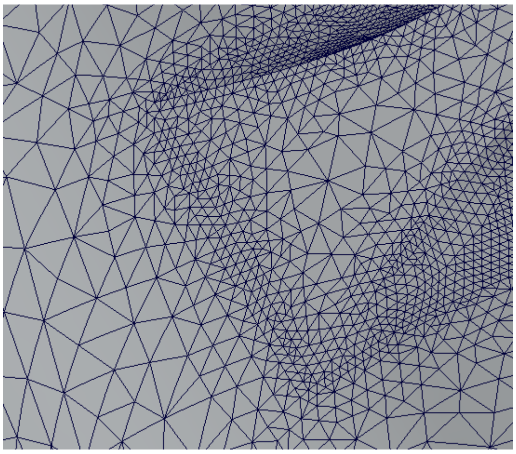

Figure 60. Closeup of the mesh of the controlled region of the right-side rocker

As it can be observed, the change in meshing on the second part allowed the total mesh to be less dense while maintaining the adequate size on the important regions, as well as a good aspect ratio.

4.1.4 LOAD AND FIXTURE APPLICATION

The loads selected, were obtained based on last year's analysis of the suspension utilizing a spreadsheet which calculates the maximum loads suffered by any bar of the suspension system. These values were taken as the maximum forces to be applied by the pushrod, and both springs, and are over dimensioned by a factor of about 1.2.

The fixture for the study was taken as the endpoints of the cylindrical section to allow possible internal elastic deformations.

4.1.5 ANALYSIS OF THE RESULTS

The results obtained from the simulations consisted of diagrams of the local deformations, stress and factor of safety.

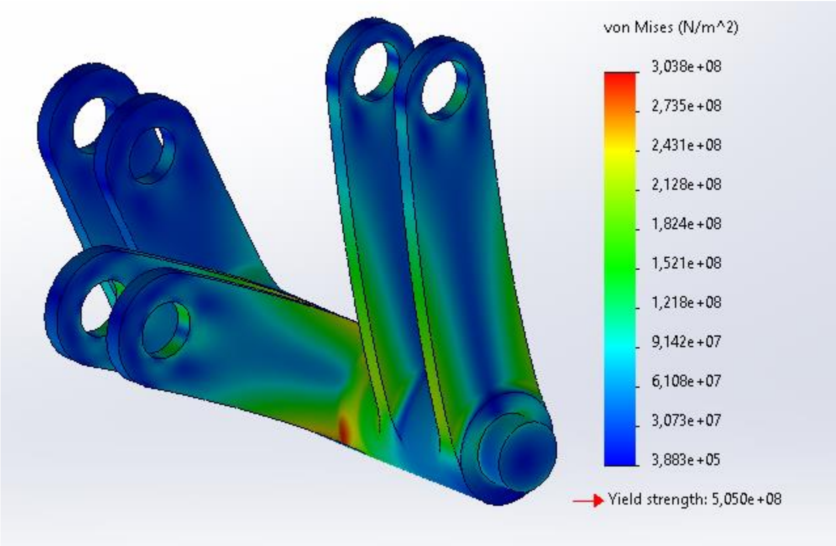

Figure 61. Image of the von Mieses stresses of the right-side rocker

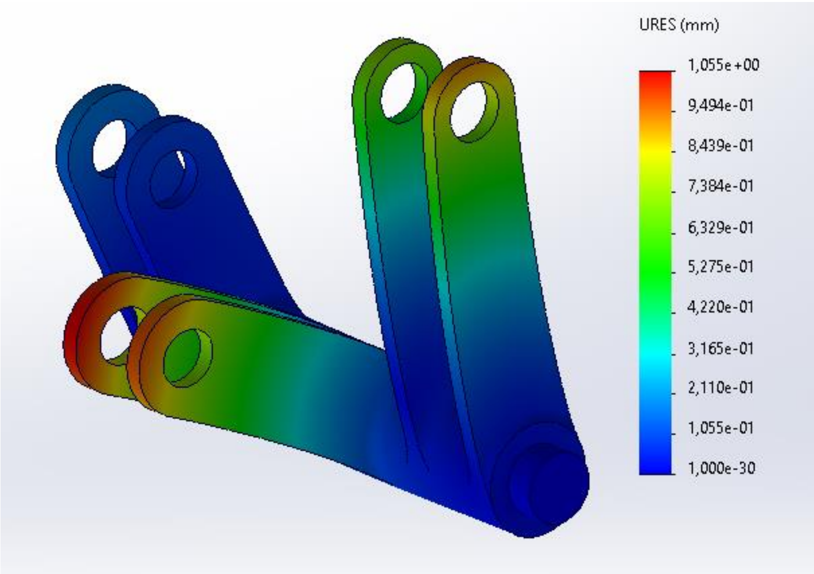

Figure 62. Image of the static deflection of the right-side rocker

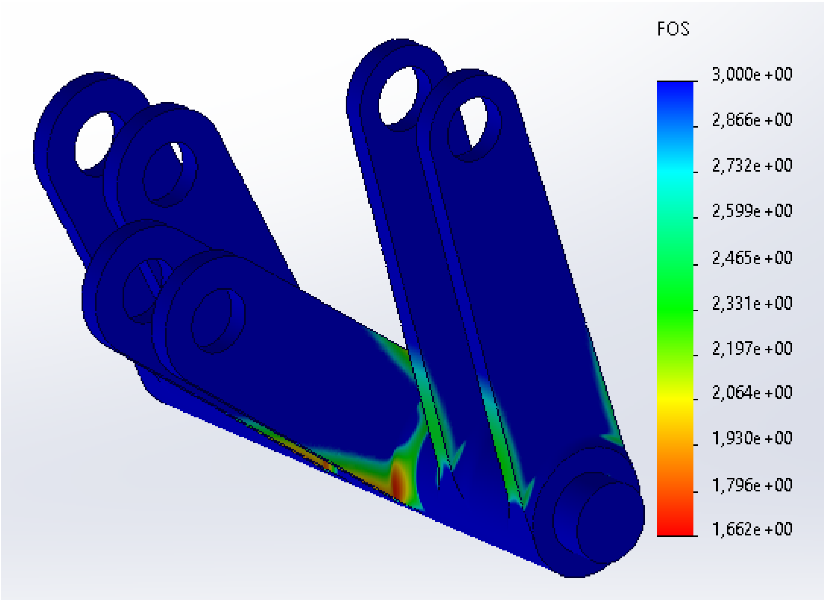

Figure 63. Image of the factor of safety of the right-side rocker

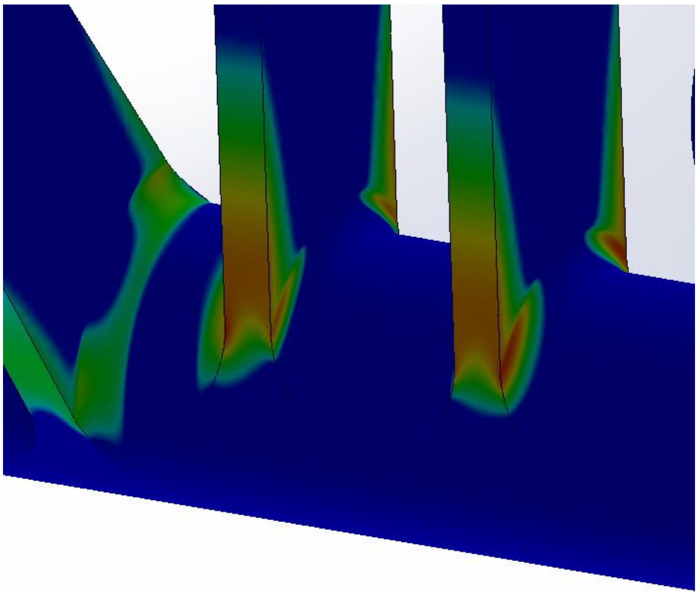

Figure 64. Detail of the most stressed part of the right-side rocker

In the simulation results, the right-side design has a minimum factor of safety of 1.6, which is within the design specifications. The other results show a maximum static deflection of 1 mm which is within the accepted length. It is also important to note that the point of maximum stress was found in the union between the actuating arms and the main body.

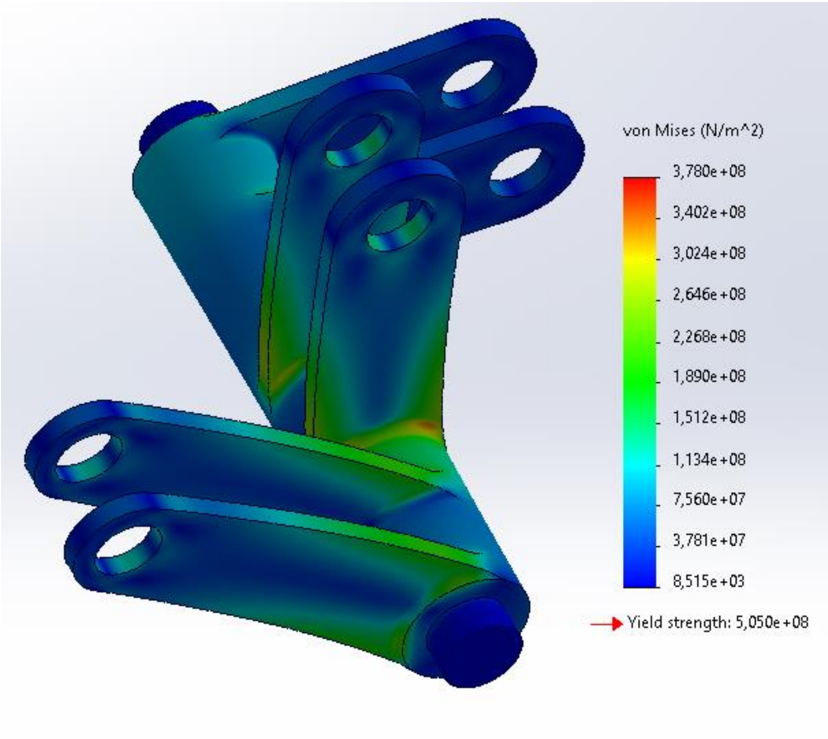

Figure 65. Image of the von Mieses stresses of the left-side rocker

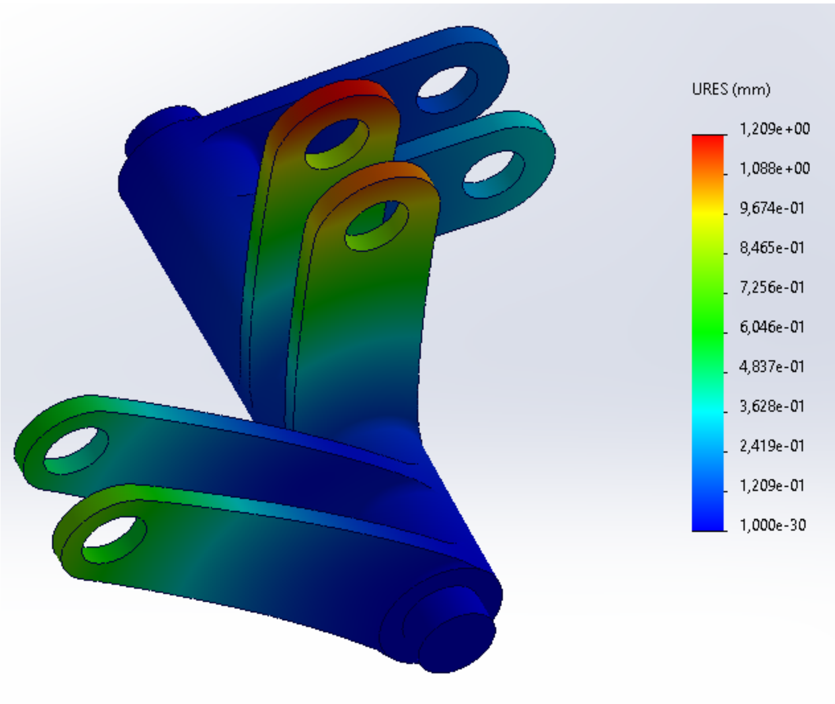

Figure 66. Image of the static deflection of the left-side rocker

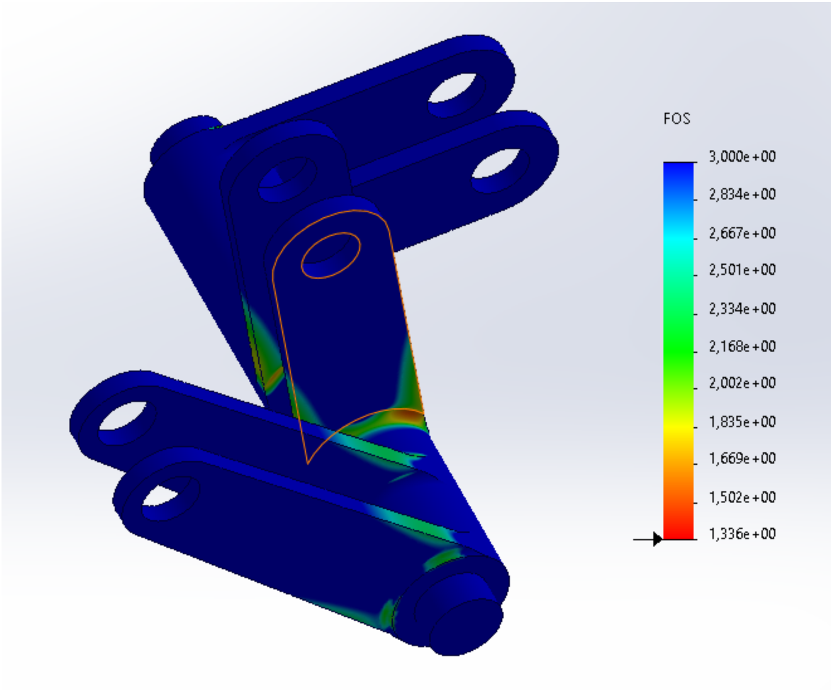

Figure 67. Image of the factor of safety of the left-side rocker

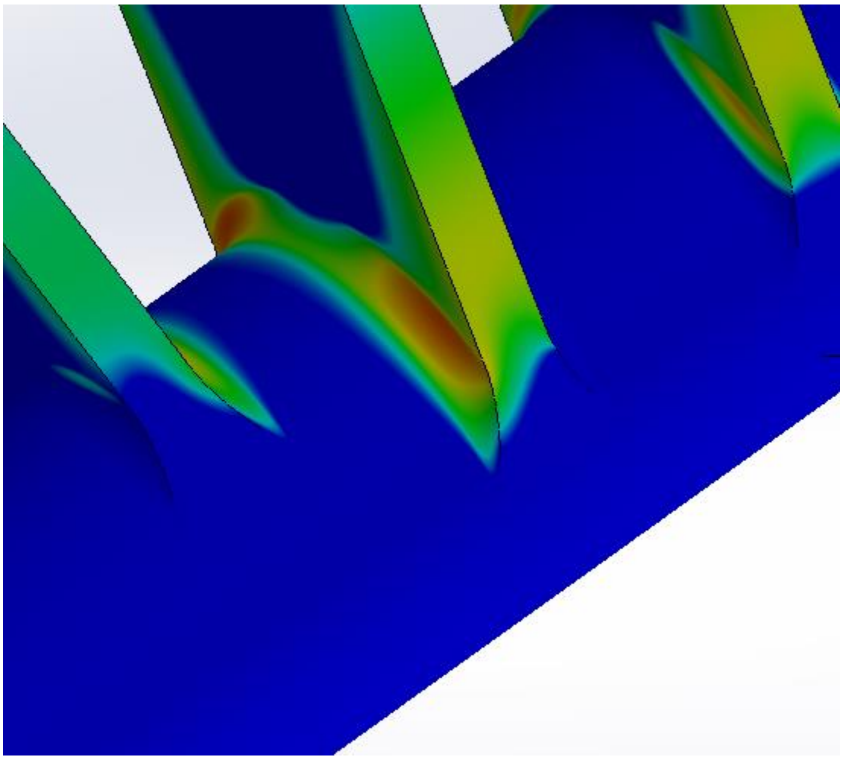

Figure 68. Detail of the most stressed part of the right-side rocker

As it can be seen in the simulation results, the left-side design has a minimum factor of safety of 1.3, which understanding the overestimation of the forces is well within the design specifications. The other results show a maximum static deflection of 1 mm which is within the accepted length. Similarly to the other side, point of maximum stress was found in the union between the actuating arms and the main body, as was expected, and therefore, the meshing was correctly done.

To conclude, we can comfortably state that the design will correctly perform in the desired manner without failure.

Chapter 5. CONCLUSION AND ROAD MAP

5.1 CONCLUSION

To conclude this thesis, the design for the roll-heave decoupled suspension can be taken as fully completed as the correct analysis and justification for the project has been completed. The understanding gained during the thesis will also prove fundamental for the integration of the system in future prototypes.

As previously defined, the objective of the thesis was the design and integration of a new front suspension system which will be able to control independently the vertical (longitudinal) dynamics of the front axle from the lateral dynamics, with the aim being the improvement of the cornering dynamics of the vehicle during all the stages: braking/corner entry, steady state cornering and accelerating/corner exit, with the consideration for the design being that it should be easy to manufacture and assemble, be easy to access to change the parameters and should try to lower the height of the centre of mass of the vehicle.

Looking at the final design, it can be seen how it achieves the objectives by being a completely decoupled system which will improve, as observed in the simulations, the behaviour of the vehicle in all the dynamics. The ease of manufacturing has been taken into account, as well as the ease to change the parameters and the lowering of the centre of mass.

The specific design requirements were also met, by achieving a parametric design which can easily be adapted to any possible changes in the team's chassis or suspension geometry with minimal design changes needed in order to be adapted, and the needed changes are already studied and documented in this thesis. The dynamic design objectives were also met, with the correct ride frequencies obtained, as well as the desired roll gradient to accommodate the design to the vehicle.

Moreover, the progress achieved by this thesis for the design of the IFS-07 prototype is not only achieved by the general design developed, but also by the study into the field of vehicle dynamics carried out, especially in regard to how does this field affect the overall design of a Formula Student prototype.

To sum up, it can be stated that the significant changes presented by this thesis are:

- Redesign of the front suspension to incorporate a roll-heave decoupled system.

- Better understanding and mathematical modelling of the dynamic behaviour of the vehicle.

- Improved dynamic behaviour of the vehicle by the correct selection of spring stiffnesses for both heave and roll.

- Better understanding of the kinematic behaviour of the vehicle's suspension.

Some of the limitations of the thesis include:

- **The decoupling of the rear suspension:** The rear suspension design was not done, due to the spatial constraints on the rear of the vehicle.

- **The dynamic modelling of the full vehicle:** The dynamic modelling of the vehicle was not achieved, due to the high complexity of the interactions between the systems.

- **The manufacturing of the suspension:** The manufacturing process of the suspension was not defined in detail, as well as was limited to some easier to use technologies.

5.2 PATH OF DEVELOPMENT FOR THE SUSPENSION

The future work for the team regarding the suspension system of the vehicle and the understanding of the vehicles dynamic behaviour is plenty. This work for the next seasons should include:

- Rear decupled suspension: With the design for the front axle done, the design for a roll-heave decoupled rear axle should be made when the spatial limitations are cleared. This design should help the vehicle, completely and independently control its roll and heave.

- Full dynamic model of the vehicle: With a first approach done in this thesis for the dynamic modelling, so be it a very simple independent model for the front axle, a total interconnected dynamic model for the vehicle should be made, to better understand the behaviours of the designs made before taking them to reality. This model should also be validated correctly with the real behaviour of the car.

- Verification and telemetry: The correct verification of the vehicle through the implementation of a correctly defined and calibrated telemetry system is fundamental to correct design and setup of the suspension system. This verification process should also help correlate the designed behaviour of the vehicle to the real behaviour of the vehicle.

- Understanding of the fabrication methods: The understanding of the fabrication methods will directly correlate to the precision and validity of the manufactured system. The main focus for this point should be the design and fabrication of a welding jig and new suspension hardpoints to be able to more correctly and accurately define the suspension geometry.

- General weight reduction of the system: The general wight reduction of the system, with a special focus in the reduction of the unsprung mass is essential for a better dynamic behaviour of the vehicle. It is important to note that the majority of the mass of the vehicle's suspension system is part of the unsprung mass, with the reduction of weight in both uprights and wheels being possible, this point should be a main focus.

-Lap time simulation: The justification of every design change on the vehicle should be tied with the effect that change has on the lap time. This is why a lap time simulation, with an especial emphasis on the dynamic behaviour of the vehicle is an essential tool for the design validation. This toll should also help understand better the setup changes, as well as be validated with the reality.

The changes here proposed are thought to show what the general design lines for the suspension and vehicle dynamics department should follow, taking the current state of the project.

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ANNEX I. SUSTAINABLE DEVELOPMENT GOALS

The project presented, aligns with the Sustainable Development Goals (SDG) such as:

- Goal 9: Industry innovation and infrastructure. The design of new more efficient suspension systems helps develop new innovations in the automotive industry; by introducing new and different ideas, the industry can develop these ideas into more efficient designs which can therefore be applied to the everyday use.

- Goal 11: Sustainable cities and communities. The study of new, better responding and better riding suspension systems can help make the cars safer for both occupants of the vehicle and bystanders, helping the vehicle maintain control in unstable situations therefore helping with the creation of safer, and therefore more sustainable, communities where automobiles can exist.

- Goal 13: Climate action. The study and understanding of the dynamics of any vehicle can help achieve better energy efficiency, this helps reduce the energy consumption of vehicles. The other way this project helps with the climate action is that by being part of an electric formula student team, one can understand the new emerging problems with the suspension systems of EVs, this knowledge can be applied to better understand these types of vehicles and how they can be approached either by big manufacturers, bringing less polluting vehicles to market or smaller racing series and how they can in a more performance oriented way transition to the greener electric mobility.´

Figure 69. SDG's