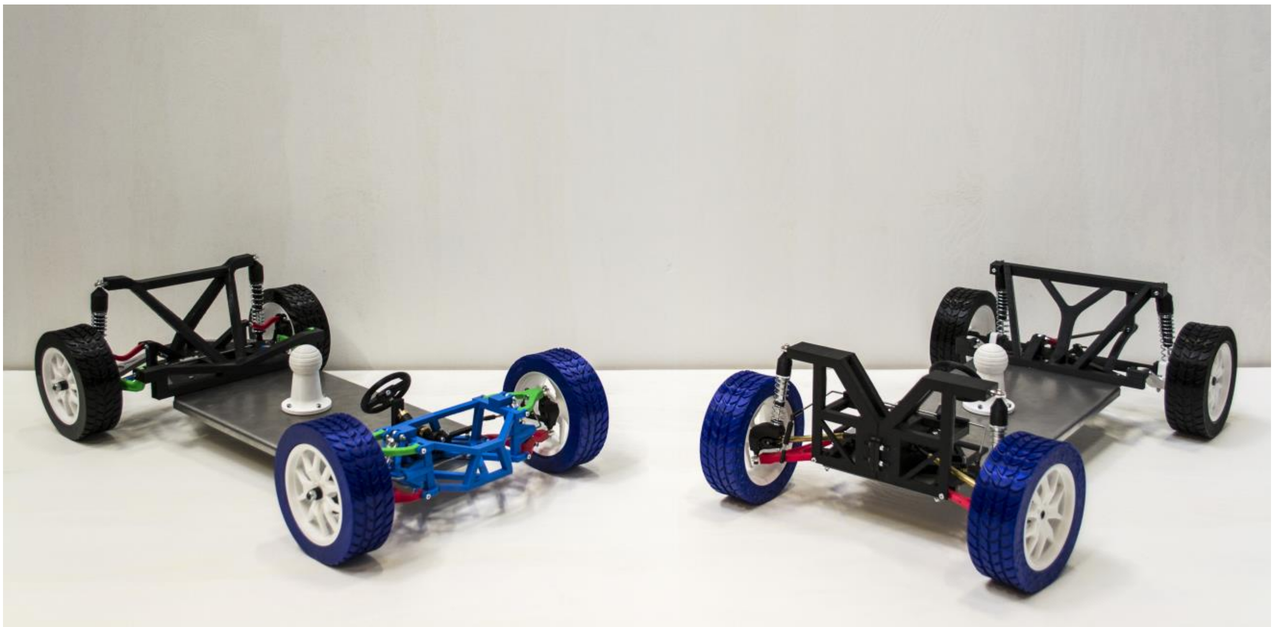


Design and 3D Printing of Down-scaled Passenger Vehicle Chassis

Bachelor's Thesis in Mechanical Engineering

Anton Öhammar, Jacob Grundén, Jon Jaleby, Kristofer Arvidsson, Oskar Hellsten, Simon Medbo

Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

BACHELOR’S THESIS IN MECHANICAL ENGINEERING

Design and 3D Printing of Down-scaled Passenger Vehicle Chassis

Anton Öhammar, Jacob Grundén, Jon Jaleby, Kristofer Arvidsson, Oskar Hellsten, Simon Medbo

Department of Applied Mechanics
Division of Vehicle Engineering and Autonomous Systems
Vehicle Dynamics Group
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

Design and 3D Printing of Down-scaled Passenger Vehicle Chassis
Anton Öhammar, Jacob Grundén, Jon Jaleby, Kristofer Arvidsson, Oskar Hellsten, Simon Medbo

Bachelor's Thesis 2017:04
ISSN 1654-4676
Department of Applied Mechanics
Division of Vehicle Engineering and Autonomous Systems
Vehicle Dynamics Group
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: + 46 (0)31-772 1000

Examiner: Bengt Jacobson bengt.jacobson@chalmers.se
Supervisor: Ingemar Johansson ingemar.johansson@cevt.se

Department of Applied Mechanics
Göteborg, Sweden 2017-05-12

Design and 3D Printing of Down-scaled Passenger Vehicle Chassis
Bachelor's thesis in Mechanical Engineering
Anton Öhammar, Jacob Grundén, Jon Jaleby, Kristofer Arvidsson, Oskar Hellsten, Simon Medbo
Department of Applied Mechanics
Division of Vehicle Engineering and Autonomous Systems
Vehicle Dynamics Group
Chalmers University of Technology

Abstract

To close the gap between the automotive industry and newly graduated engineers the graduates need to have a better understanding for how their design affects the end result. This thesis focuses on producing 3D printed suspension scale models with correct kinematics to help students better understand how different types of suspensions behave. The suspension types chosen for the models are MacPherson and double wishbone for the front suspensions and trailing arm multi link suspension and a live axle for the rear. These models were designed in Catia V5 and then 3D printed in a scale of 1:5 and assembled together with some off-the-shelf components such as coilover dampers and ball joints. The models for the MacPherson and the trailing arm are based on Chalmers suspension setups while the live axle is based on modified values of the trailing arm. The double wishbone however is primarily based on given requirements and guidelines from the Chalmers suspension setup. The two front suspensions feature functioning steering and the MacPherson one also has a differential to illustrate the function of having a propulsion system. These suspensions are mounted on two vehicle chassis where the front suspensions can be swapped so all the combinations can be shown. The curves for camber gain and bump steer are plotted for the MacPherson and double wishbone suspensions. The 3D printed models have similar kinematics to the original models. Because of the limited accuracy of 3D printing there is some play in the suspension components reducing the similarity between the curves and the real behavior when manipulating the suspensions by hand.

Keywords: vehicle suspension, 3D printing, kinematics, chassis, vehicle dynamics

Acknowledgements

We would like to say a special thanks to our supervisor Ingemar Johansson for his outstanding work and help during this project. Without his expert knowledge and will to help we would not have come as far nor have the understanding that we today have.

Thank you Ingemar

Preface

This thesis was done as a bachelor thesis at the Mechanical engineering programme of Chalmers University of Technology, Gothenburg. The work was carried out between January and May 2017.

Gothenburg May 2017

Anton Öhammar, Jacob Grundén, Jon Jaleby, Kristofer Arvidsson, Oskar Hellsten, Simon Medbo

Abbreviations

CAD	-	Computer Aided Design
3D	-	3 Dimensional
NVH	-	Noise, Vibrations and Harshness
PLA	-	Polylactide
CC	-	Centre-Centre
RC	-	Radio Controlled

Contents

1 Chapter One - Introduction

1.1 Background

The car industry of today is evolving and more rapid development cycles are now becoming a requirement to keep up with the competition in the industry. That is why the newly graduated engineers of tomorrow need to be even more prepared for the requirements of the industry. Therefore the universities need to close the gap between graduates and the industry. A step in this direction is to create the possibility for graduates to have a better understanding of what their design will result in. Since vehicle dynamics is a subject directly related to an end product it is essential to have great knowledge of how different suspensions work and how the theory behind it applies.

Bengt Jacobson, Professor in Vehicle Dynamics and leader of the Vehicle Dynamics group, teaches the course of Vehicle Dynamics at the Master's Programme of Automotive Engineering at Chalmers University of Technology. He has identified a need to explain suspension concepts with downscaled models. This is not easy accessed and with this as a foundation, a bachelor thesis was formed alongside Ingemar Johansson, Professor of the practice at Chalmers University of Technology, around the problem with a focus on students planning a future within the automotive industry. The final result of this thesis is to have created educational material for further use at the course of Vehicle Dynamics. How these models will be integrated in the education is not included in this thesis and will instead be a later task for Bengt Jacobson.

1.2 Purpose With the Project

The course of Vehicle Dynamics explains the dynamic and theory behind wheel suspensions. Very quickly this becomes a rather complex subject and requires a lot of material and time to fully explain the basic knowledge these concepts rely upon. To speed up the process functional models of different wheel suspension would benefit the education and the students would easier accept further complexity.

1.3 Envisioned Solution

One solution is to build down-scaled models that can function to enhance the education. The two models should describe and show two different types of suspensions each, a total of four different suspensions. These are MacPherson strut and double wishbone in the front with a multi link and live axle suspension in the rear. These models should be able to highlight different characteristics of suspensions such as camber, ackerman etc. This will thoroughly be explained later in this report.

1.4 Deliverables

The deliverables are the following:

- A front MacPherson suspension model in scale 1:5 with differential and steering
- A front Double wishbone suspension model in scale 1:5 with steering
- A rear Multi link suspension model in scale 1:5
- A rear Live axle suspension model in scale 1:5 without anti-roll bar
- Two vehicle chassis with interchangeable front suspensions

1.5 Prerequisites

As a base for this project Chalmers suspension setups are given with associated hardpoints and guideline values for camber, caster, kingpin, toe-in/out etc. Expert knowledge from Ingemar Johansson in suspension design engineering is used to break down the information and convert it into usable information. The Chalmers suspensions hardpoints for a

MacPherson and a multi-link suspension are available for reengineering. For the double wishbone only guideline values, listed in chapter 3.3 are available. Based on this, the suspensions has to be redesigned and developed to fulfill the given guidelines. Throughout the report two reference suspensions models hardpoints are used, Chalmers MacPherson suspension model and Chalmers Trailing arm suspension model. These hardpoints can be found in appendix A.1 and appendix A.3 respectively.

1.6 Limitations

To restrict the project from becoming too extensive, the work will be focused on suspensions for passenger cars. More specifically four different suspension setups, namely a MacPherson and a double wishbone for the front and a trailing arm and a live axle suspension in the rear. The focus will be on the suspension setup and therefore the propulsion system of the car will not be considered on three of the four setups. It will however be implemented on the MacPherson suspension model.

The models will be 3D printed which brings in limitations due to the method itself and the machines used. What the available 3D printers can handle will limit the size of the models and components such as joints will be purchased due to difficulty of printing. For 3D printing there are three printers available with a limited range of different colours on the printing material. Each printer can be booked for 24 hours per booking by any student that is licensed to operate the 3D printers. Furthermore the project also has a budget of 4000 SEK which needs to be taken into consideration when purchasing parts and manufacturing the models.

As specified by the project there is also a time limit which spans from 2017-01-17 to 2017-05-23. Due to the short timeframe and large scope of the project no calculations of the strength of the printed parts will be made. The design will instead be based on general engineering methods such as the use of trusses and no sharp transitions.

1.7 Method

The first step of the project is to gain an understanding of how different suspension types work by studying literature and other sources. With this as a starting point, the next step is to design four different suspension types in CAD and the CAD software will be Catia V5. The design process will be based on geometries from the existing Chalmers suspensions to ensure that the critical points are positioned in the correct places to get a realistic behavior from the physical models. Consulting with the prototype laboratory during the design process will ensure that the CAD models are suitable for printing in the 3D printers available at Chalmers. When the design process has resulted in finished CAD models of the different suspension types, they will be tested in computer simulations to compare the behavior to real world suspensions. If the behavior in the simulations is acceptable, the models will be 3D printed and assembled. After this the models will be measured to ensure that the design criterias are met.

1.8 Participants and Function in the Project

The thesis is carried out by a team of six students, Anton Öhammar, Jacob Grundén, Jon Jaleby, Kristofer Arvidsson, Oskar Hellsten and Simon Medbo. The members of the group study mechanical engineering and aspire to continue a master in automotive engineering. The functions through the project are divided to create a varied workload where all members get to test the various roles needed in this project. E.g. the position of administrator is a rotated post and every second week the role switches to another member. The work has been carried out as a team work and the team has been able to develop a very efficient work process.

2 Chapter Two – Background Theory

2.1 Introduction to Vehicle Technology – Wheel Suspension Engineering

A car consists of many different systems and parts. Body, body structure, chassis - including suspensions, and propulsion system. The suspension is the connection between the body structure and the wheels and allows relative movement between them. This means that it affects areas such as ride comfort and road grip. If a car were to have no suspension and the wheels would be connected directly to the body structure, problems would emerge. Mainly that the imperfections in the road surface would be transferred over with only minor isolation from the tyres to the body structure. This would result in an extremely stiff and uncomfortable ride but also in excessive load on the body structure which causes fatigue damage. This means that the suspension needs to isolate the chassis from movement and forces generated by the wheels as they follow the road. Another issue to arise is that the wheels would lose contact with the road causing there to be less grip when the road surface is not perfectly flat and even. With a suspension however the wheels can follow the surface much better due to the allowance of vertical movement relative to the body.

2.2 Wheel Suspension Parts

To be able to accomplish the tasks given to the suspension it needs some specific components. It needs some form of linkage, springs, dampers along other components. There are also additional components to increase comfort, grip and prevent damage. Under this chapter all these components are described.

2.2.1 Spring

A springs purpose is to give the desired wheel spring rate to control the wheel. Springs are often made out of spring steel and can be found in various sizes and shapes, with the most common version being coil springs. The springs are found on the front and rear suspensions, often mounted together with a hydraulic shock absorber but also mounted separately.

2.2.2 Shock Absorber

Is normally a hydraulic system designed to dampen and absorb shock impulses. The kinetic energy is converted to heat through the oil. They are found on both front and rear suspensions of a car.

2.2.3 Coilover Damper

A coilover damper is a combination of a shock absorber and a spring mounted together. It integrates both the shock absorber and the spring in an effective way. A double wishbone suspension is an example of where you can find a coilover damper.

2.2.4 Strut

A strut is essentially the same thing as a coilover except for the fact that it has an axial bearing at the top to allow for it to rotate around its own centre axis while taking forces perpendicular to it. It is found in a MacPherson suspension.

2.2.5 Anti-roll Bar

An anti-roll bar, also known as sway bar, is used to give roll stiffness to the chassis and decrease the body roll of the vehicle and in that way improve the vehicle's handling. It is connected to the suspension on both sides of the vehicle and improves handling in fast cornering. The bar itself is connected to the suspension via drop links which are smaller links with ball joints in each end. The anti roll-bar is e.g. used to tune the under- and oversteer balance in the car.

2.2.6 Upright

The upright carries the wheel bearing with the control arms and links. It is connected to the subframe or body structure. The MacPherson upright is connected to the body structure with the lower control arm and the strut. The structural design of the upright is important as it is a saftey critical part. An upright is found on a variety of different suspensions and its design depends on the suspension type and design.

2.2.7 Control Arm and Link

Control arms joins the upright and the body structure together and has one degree of freedom. The control arm has bushings or ball joints in each anchor point to enable some movements. Depending on which type of suspension and car model the control arm is mounted on it can have a wide spectrum of different designs, but still with the same function. Links have a similar function to control arms but with two degrees of freedom.

2.2.8 Bushing

Bushings are used to isolate vibrations to improve ride comfort and NVH as well as to enable links to move easily when forces are applied. Bushing are usually made from rubber containing a steel axle in the middle and one as housing to get stiffness and rigid mounting points. It can be found in various sizes and stiffnesses depending on where and how it is used. The stiffness rates in the bushings are giving the elasto kinematic performance in the suspensions.

2.2.9 Ball Joint

A ball joint consist of a bearing stud and socket enclosed in a casing and is made in steel. Protective sealing often made out of rubber prevents dirt from getting in the joint assembly. To achieve smooth movement the joint has lubrication sealed in the joint which also decreases the friction. Modern ball joints are lubricated for life and has no need of replacing old lubricant. The design allows free smooth movement in two planes at the same time which is desired in several parts of a suspension.

2.2.10 Subframe

The subframe is a structural component that carries components of the car and gives structural stiffness. Suspensions, drivetrain and the engine are bolted together with bushings to the subframe. It is found in both the front and the rear of a vehicle. It is often used to build a front and rear module as sub assembly, which is preferred in the final assembly of the car.

2.3 Terms and Definitions – Wheel Suspension Engineering

In automotive engineering many specific terms are used and regarding suspension there is no exception. Examples of what they describe are different angles, phenomenons or other useful information.

2.3.1 Coordinate System and Forces

The coordinate system for the Chalmers suspensions are defined as a cartesian coordinate reference system which can be seen in figure 2-1. Forces parallel the x-axis of a vehicle are known as longitudinal forces while forces parallel to the y-axis instead are known as lateral forces. Vertical forces are those parallel to the z-axis of a vehicle.

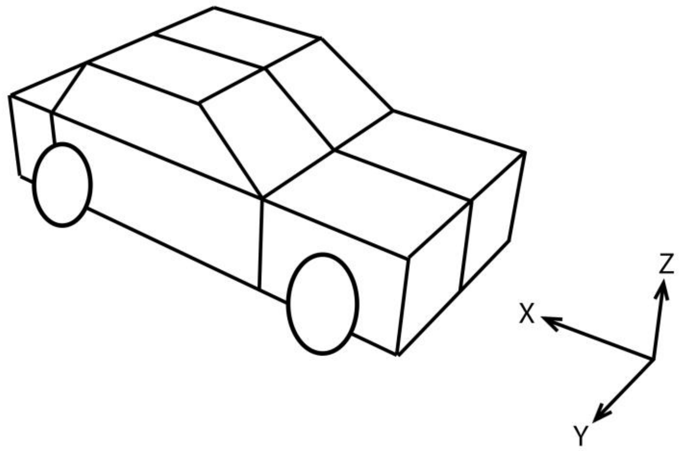


Figure 2-1 Cartesian coordinate system

2.3.2 Hardpoints

A hardpoint is a coordinate in a 3D dimensional space defining a position for a point of interest for a component in the suspension.

2.3.3 Body Roll

When a vehicle corners its weight will be pushed in the outward direction due to mass inertia which causes the vehicle to roll.

2.3.4 Roll Centre

The theoretical points which a vehicle can roll around without any lateral movement is defined as its roll centre. Each axle has its own roll centre meaning a normal car it has one in the front and one in the rear. The exact position of the roll centre is defined differently for each type of suspension geometry. Connecting these two points will give an axis known as the roll axis which will be where the vehicle will roll around. This means that any transverse force applied to this axis will not cause any roll. The position of the roll centre affects multiple areas but one of the major areas is handling. [1]

2.3.5 Dive and Squat

During acceleration and retardation a vehicle will pitch, meaning that it will rotate around the y-axis. During retardation the vehicle will dive meaning that the body will pitch forward and the front end will go lower while the rear end lifts up. Squat is similar to dive but the body pitches backwards instead of forward and it occurs during acceleration instead of retardation. This means the front end will lift up while the back end lowers down. [2]

2.3.6 Jounce and Rebound

When speaking of wheel travel the terms jounce and rebound are used. Jounce, also known as bump, refers to the compression of the shock absorber and spring. This means that the wheel travels upwards while rebound refers to the opposite. [3]

2.3.7 Bump Stop

To prevent the coils from reaching full bump which can damage the body, a bump stop is used. This means it decides the maximum wheel travel in bump. It is an elastic stop which can be mounted in a few different locations, for example in the coil springs. By absorbing large forces over a short distance it can limit the bump travel and is also important for isolation. [3]

2.3.8 Contact Patch

A tyre on a vehicle is deformed by the vertical forces being applied to it. Because of this it does not only make contact to the road in a single point or line but instead on a surface of the tyre. This surface or area is called the contact patch of the tyre. [4]

2.3.9 Ground Clearance

Ground clearance or ride height is the distance between the lowest part of the chassis of a vehicle and the ground. On SUVs this is typically a higher value compared to a regular sedan and then a sports car usually have an even lower ground clearance. Ground clearance affects the height of the centre of gravity which in turn affects how much the vehicle will roll and its handling. [5]

2.3.10 Bump Steer

Bump steer is the unintentional changes in the orientation of the wheels due to vertical travel of it. Since it causes unintentional steering changes it is normal to try and remove bump steer effects. [4]

2.3.11 Camber

The camber angle is the angle between a plane perpendicular to the ground and the wheel seen from the front or back. Camber can be negative, neutral or positive with neutral meaning 0 degrees between the line and wheel. Negative camber is when the top of the wheel is leaning inwards and positive when it is leaning outwards. [1]

2.3.12 Caster

In a side view the angle between the steering axis and the vertical axis through the wheel centre is called caster. It is defined as positive when the top of the steering axis is further back than the bottom. Both handling and steering are areas affected by the caster angle. [1]

2.3.13 Toe

Toe is the angle of the wheels with regards to the centerline of the vehicle when viewed from above. Toe can be in toe-in, toe-out or zero-toe settings and when wheels are set to toe-in they point inwards to the centerline. Zero-toe means they are pointing straight ahead along the centerline and toe-out that the wheels are pointing outwards. A negative toe angle represents toe-in and a positive toe-out. Changing the toe setting will have an impact on tyre wear, cornering and also the rolling resistance of the tyre. [2]

2.3.14 Kingpin

Kingpin is the steering axis which is defined as a line through the upper and lower ball joints in the front suspension. Alongside Kingpin there is also Kingpin inclination which is similar to the caster angle. Instead of being a view from the side like the caster it is viewed from the front and is the angle between the vertical plane and the Kingpin. [2]

2.3.15 Ground Offset

The ground offset, or scrub radius is defined as the distance between the kingpin and the centre of a wheels contact patch where they intersect with the road surface. This when viewed from the front. This distance is defined as positive when the kingpin intersects with the ground closer to the centerline of the car than the wheels centerline does. The steering is one of the areas affected by the ground offset. [1]

2.3.16 Ackerman

Ackerman is a steering geometry with basis in wanting all wheels to roll around a common point during a turn. To accomplish this with the rear wheels not turning the inner front wheel has to turn more, at a bigger angle, compared to the outer front wheel. This means that the inner and outer wheel has different turning radius. Using an Ackerman geometry is advantageous because it reduces the need for the tyres to slip when cornering [6] and reduces the lateral forces in the steering wheel. The amount of ackermann can be measured in ackermann percentage. 100% ackermann is when both wheels are traveling at an angle that makes the two circles concentric and 0% ackermann is when both wheels are traveling with the same steering angle. The percentage can be calculated with the following formula:

$$do = steering\ angle\ outer\ wheel$$

$$di = steering\ angle\ inner\ wheel$$

$$w = wheelbase$$

$$t = track\ width$$

$$dao = \frac{\tan^{-1}(\tan(di))}{(1 - \tan(di) * \frac{t}{w})} = Ackermann\ outer\ wheel\ steer\ angle$$

$$Ackermann\ percentage = \frac{(di - do)}{(di - dao)} * 100$$

The ackermann percentage is normally calculated at 20 degrees steering angle on the inner wheel. [5]

2.3.17 Overtravel

Wheel travel is designed to give the acquired wheel travel for normal use, overtravel due to misuse loads is more than this designed wheel travel. Overtravel needs to be considered for packaging reasons. [5]

2.3.18 Rack travel

Rack travel is the axial distance the steering rack travels along the y-axis when the wheels are turned.

2.3.19 Load Conditions

A vehicles weight can be defined in multiple different load conditions. Some of these weights are: dry weight, wet weight, complete vehicle shipping mass and kerb weight. Complete vehicle shipping mass is a vehicle's mass with all electrical and auxiliary equipment necessary for normal operation. This alongside standard or optional equipment from the manufacturer which should be specified in a list. The definition of kerb weight form [7] is the complete shipping mass plus the car being filled with all fluids namely lubricants, coolant, washer fluid and a full tank of fuel, but no passengers or cargo. The +2 stands for that there are two occupants included in the kerb weight and each occupant's weight is 75 kilograms. The kerb +2 weight is the weight where the suspensions are designed.

2.3.20 Wheelbase and Track Width

The wheelbase is the length between the centre of the rear and front wheel on the same side of the car. Furthermore the track width is the length between the centre of the contact patch for each wheel on the same axis.

2.4 Specific Information About the Wheel Suspensions

Since this project covers four different types of suspension it is self-explanatory that they can not all work in the same way. This section will distinguish these differences. Three of the four suspensions are independent which means that each wheel can move independently from the other. The last one, the live axle, is a dependent suspension which on the contrary means that the wheels can not move independently from one another but instead the second wheel is affected by first wheel's movement.

2.4.1 Front Suspension – MacPherson

A MacPherson suspension is commonly used in the front suspension of modern cars due to its simplicity and low manufacturing cost. The disadvantage is that it usually gives lower quality of ride and not as good handling of the car because it gives less freedom to choose camber change and roll centre. Since this is not as important for consumer cars it is a good choice of suspension setup.

The suspension consists of a lower control arm which is a linkage between the subframe and the upright and a strut which connects the upright to the body structure. This can all be seen in figure 2-2. The anti-roll bar can be attached to the shock absorber or the control arm. There is also a steering connection and there can be a drive shaft connection to the upright and the wheel respectively.

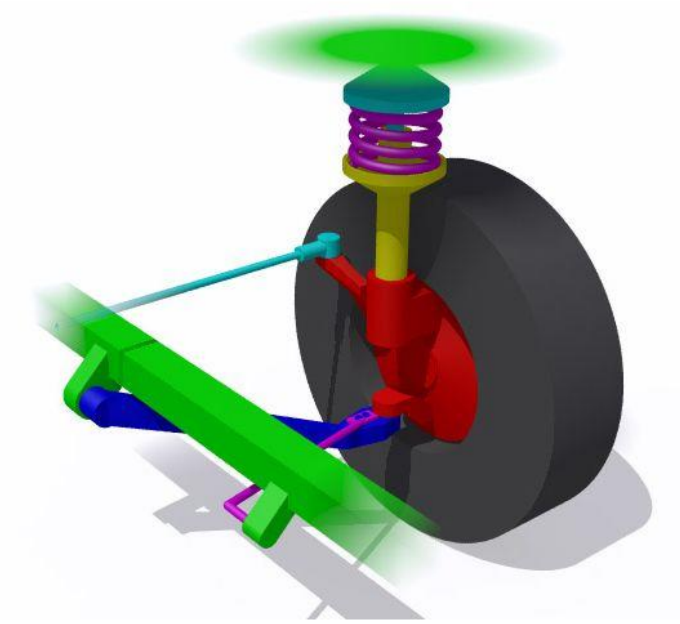


Figure 2-2 Mcpherson strut. From [8]. CC-BY-SA.

As the control arm is attached to the subframe at two points it allows for one degree of freedom, rotation around the x-axis. With the help of bushings the control arm also has the possibility to rotate slightly around the z-axis to withstand the longitudinal forces. A ball joint connects the control arm and the upright together to enable the wheels to turn. The strut is fixed to the upright and allows for some movement by a bushing and an axial bearing at the top mount to cooperate with the change in camber and turning of the wheels. The bushing at the top mount isolates for ride comfort and NVH. Note that the strut takes pitch and roll bending moments, thanks to a long piston inside the damper.

2.4.2 Front Suspension – Double Wishbone

A double wishbone suspension is usually found as a front suspension in sports and race cars but can also be used for a rear suspension. It is used on performance vehicles due to its large amount of free parameters meaning it has great adjustability and can be optimized for the vehicle in question.

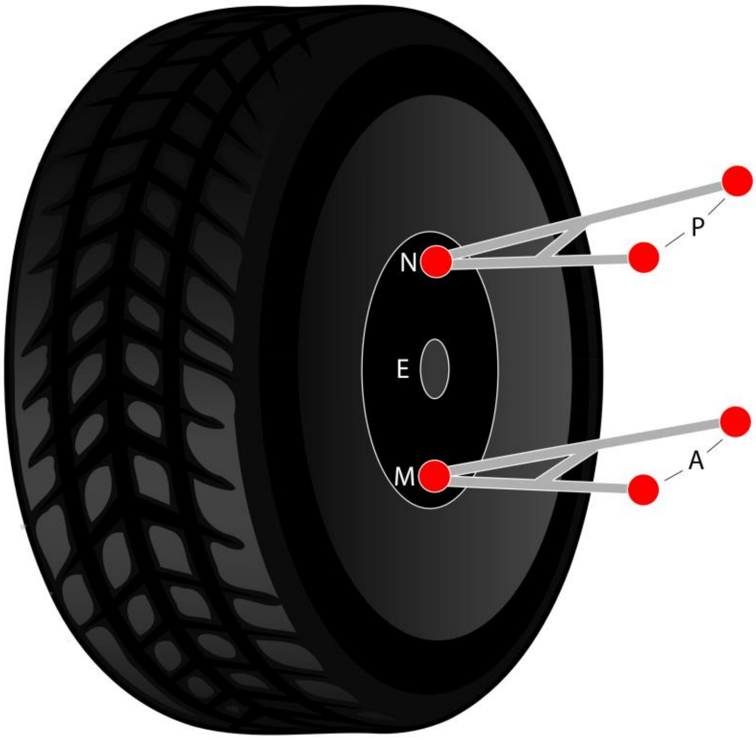


Figure 2-3 Illustration of a double wishbone suspension. From [9]. CC-BY-SA.

A double wishbone suspension, illustrated in figure 2-3, consists of two control arms called wishbones, one upper (N,P) and one lower (M,A), and an upright (E). The control arms act as a connection between the body structure and the upright. The two inner ends are attached to the body structure via bushings and the outer to the upright via a ball joint. These control arms can take various shapes for example an A or an L shape. The shock absorber and coil spring are mounted between one of the wishbones and the body structure. The upright is mounted between the two control arms outer ends and connects them to the wheel. It has space in the middle which can be used for mounting of an axle if propulsion on the axle is desired and an extension on one side for mounting of the steering connection.

2.4.3 Rear Suspension – Trailing Arm

This type of suspension, sometimes referred to as "multi-link" due to the multiple number of links connecting to the upright, is found in the rear of commercial cars. The difference from a regular multi-link is that one or more links are connected between upright and a pivot point while being perpendicular to and in front of the rear axle. This suspension is often used due to the simplicity to setup the suspension with the correct camber, toe etc.

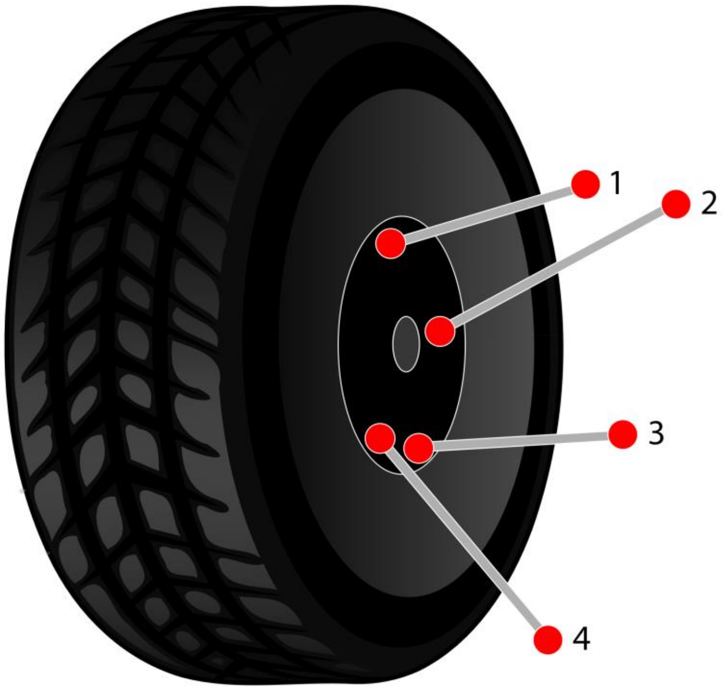


Figure 2-4 Illustration of a trailing arm suspension. From [9]. CC-BY-SA.

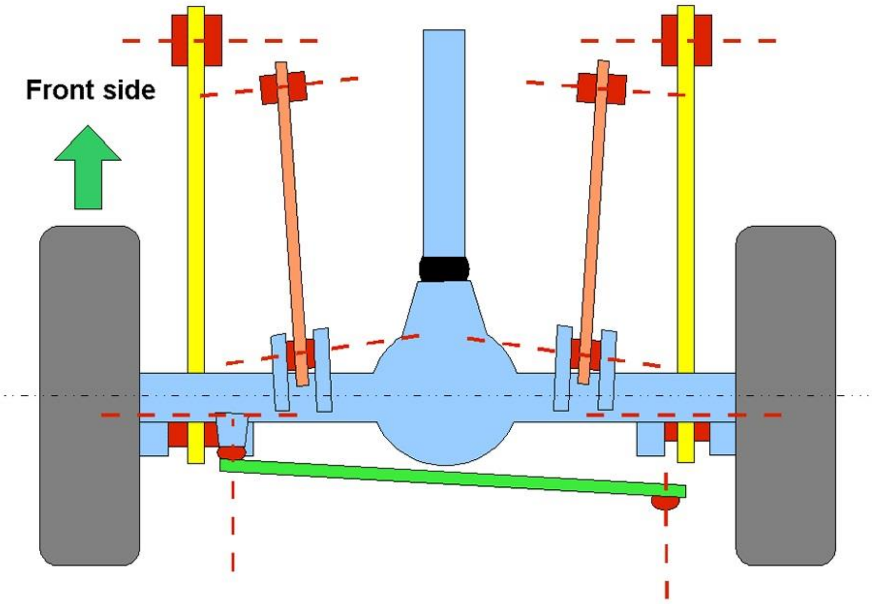
As you can tell from figure 2-4, there are four links connecting the upright to the subframe. The links are called:

- Camber link (1)
- Tie blade (2)
- Toe link (3)
- Spring link (4)

All these together allows for movement along the vertical axis. As with the other suspensions there are also bushings involved to allow for a slight movement in the other directions. The perpendicular link in figure 2-4 called tie blade, has an extra large bushing since this alone has to allow the suspension to smoothly handle longitudinal loads for ride comfort and isolates against NVH.

2.4.4 Rear Suspension – Live Axle

One simple and effective solution for a rear suspension is to use a live axle which is a straight axle going from one wheel to the other. This suspension is cost efficient, robust and easy to repair. There are however also disadvantages to this type of suspension, for example it does not offer as much adjustment as a multi-link suspension. It also takes up a lot of space above the axle itself as it needs to be clear to allow bump travel. Because of the axle running from one wheel to the other it is also heavy. Typical uses for this type of rear suspension is in heavy-duty and commercial trucks, off-road vehicles and previously in passenger cars.



Figur 2-5 Axle - 5 Link rigid 03. From [10]. CC-BY- SA.

A live axle suspension has coil springs and shock absorbers like most suspensions but can also be configured with leaf springs. These coil springs and shock absorbers are mounted from the body structure to the rear end of a lower support arm but exactly where and how can vary. These lower support arms, yellow in figure 2-5, are longitudinally mounted on either side of the vehicle. These are in the forward facing part connected to the chassis with the use of bushings. This allows for rotation around the y-axis in the mounting point. Towards the rear they are not just connected to the springs and shock absorbers but also to the main axle. The main axle is the one running from one wheel to the other and connecting the whole of this rear suspension. From one side of the axle up to the body structure on the opposite side of the vehicle runs a Panhard rod, marked as green in the figure. The purpose of the rod is to restrict the lateral movement of the axle. Mounted between the body structure and the axle there is also some form of additional control or stabiliser arms, often also an anti-roll bar. Both design and placement of these arms can vary between different vehicles to suit packaging.

3 Chapter Three – Design Engineering

3.1 Off-the-shelf Components

Since not all the components of the scale 1:5 suspensions could be 3D printed some had to be purchased. Ball joints were one such component and three different kinds were selected for use. One is a metal joint and the others plastic. Two of these have M3 sized holes where the plastic M3 ones has a flat surface on one side of the ball joint and are smaller in size than the metal ones. The planned use for the plastic M3 ball joints were the steering and to use the metal ones for everything else except the anti-roll bar drop links. For the drop links smaller M2 plastic ball joints were chosen.
The data for the metal ball joints are as follows:

- Shaft diameter: 6.5 mm
- Shaft length: 16 mm
- Ball centerline to edge: 7 mm
- Ball diameter: 7 mm
- Joint width: 12 mm

Other components which needed to be purchased were the springs and the shock absorbers. The available components were from RC cars and they were not springs and shock absorbers separately but instead coilover dampers. Due to the lengths and stiffnesses desired the options were limited and lengths could not be perfectly matched. Because the coilover dampers on RC cars are proportionally longer than on normal cars, smaller scales ones had to be used. For the rear suspensions scale 1:8 coilover dampers were chosen and for the front two different lengths of coilover dampers in scale 1:10 were chosen. On all the coilover dampers there are a ball joints in the lower mounting point and the possibility of adjusting the tension of the springs.

For the planned propulsion system a few components were needed. The core component was a differential, which was found in scale 1:8. Two axle shafts that keep the wheel in place and enables propulsion was purchased. A drive shaft in scale 1:8 that matches the axle shafts on the differential and by the wheel was purchased to link the wheel and differential together. For the suspensions that does not have propulsion machined axles were purchased. To make sure that the axles are aligned in the upright and can easily rotate, two bearings are added in each upright. Bearings were added to all the suspensions and therefore 16 bearings with the dimension 8x12x3.5 mm was purchased.

3.2 MacPherson Suspension

The front MacPherson suspension was based on an existing suspension. Hardpoints and principle design of the Chalmers MacPherson suspension model has been used as the basis for the design of the downscaled suspension model.

3.2.1 Subframe

The subframe had to be designed with a few alterations to what is considered to be a normal subframe as it has to include a upper strut mount. The strut had to be switched to a coilover damper as no struts were available for purchase. The differential required a housing to be held in place and the result can be viewed in figure 3-1. As the differential used is prpportionally larger than in a normal car it requires a lot more space and the consequence was packaging problems.

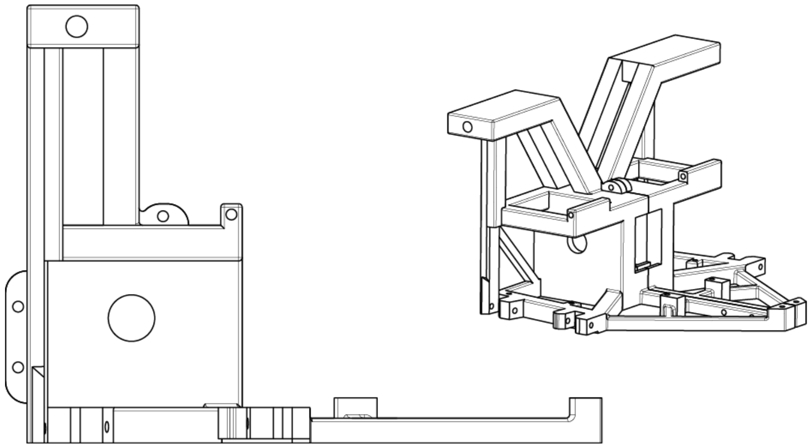


Figure 3-1 Macpherson suspension subframe

The procedure for designing the subframe started out with defining all the relevant hardpoints in CAD to ensure that the geometry would be correct. From this point an initial design was delivered to fit the control arms, the upper coilover damper mount and the differential. Because of the material being PLA the initial design was highly over dimensioned and the later iterations became cleaner and neater. Unfortunately, it was difficult to find coilover dampers in the correct dimensions for the model and smaller ones were the only ones available. To compensate for this the upper coilover damper mount was lowered while still being in the right angel to give the correct travel of the wheel. When lowering the damper mount the jounce and rebound measurements of 80 and 110 mm were ensured to still be correct. The steering rack was forced back from its standard position. The solution to this is further explain under chapter 3.8.

Traditionally, a MacPherson suspension is designed with the anti-roll bar connected to the shock absorber. It was discovered that attaching an anti-roll bar to the shock absorber would be very difficult and instead it was attach to the lower control arm. By doing this the anti-roll bar could be attached on the top of the differential housing.

3.2.2 Upright

To get the base for the upright the relevant hardpoints from the Chalmers MacPherson suspension were put up in a 3D CAD space. The shape was created by surrounding these points with material and connecting these. This gave a main body with space for the hole to fit the axle, a structure for mounting the coilover damper, a place for attaching the lower control arm and an arm for the steering connection. To be able to mount the coilover damper in the correct angle a hole through the upper part of the upright was needed. This was made by adding the upper hardpoint for the coilover damper and draw a line between this and the lower coilover damper point. To make sure the coilover damper mounting was well secure a slot for a screw-nut was also added. Additionally after testing a guard rail was added to keep the spring from moving around. The connection to the lower control arm was made through a ball joint. To allow this ball joint to have full movement a spacer with a conical shape was added on the bottom. For attachment of the ball joint a screw was used which meant another slot for the nut was made. For the bearings a ledge was added in the centre of the hole for the axle this to hold them in place. To allow the axle to rotate freely and be mounted correctly space was added around the hole on the inside of the upright. The upright can be seen in figure 3-2.

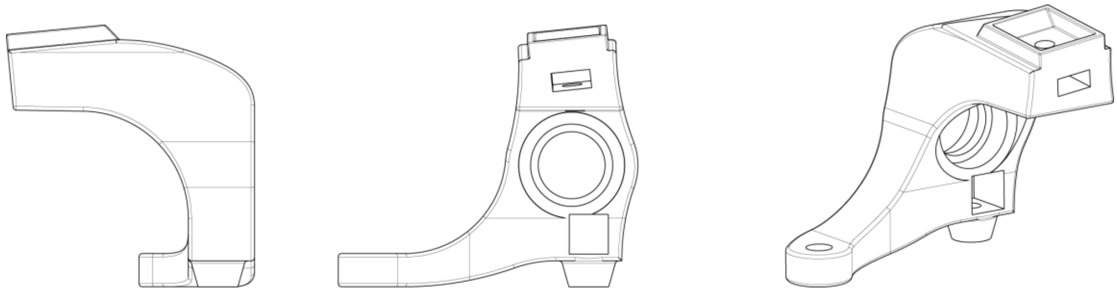


Figure 3-2 MacPherson upright

3.2.3 Lower Control Arm

The lower control arm was designed by inputting the correct hard points in Catia V5 and by visually analysing how the original control arm was designed. The three hard points forces the design to become a L-shaped control arm and with the initial design finished a first version was printed. With this version discoveries was made such as too tight fit for e.g. the connection to the subframe unabling the control arm to move properly. The anti-roll bar attachment was later in the project decided to be fitted on the control arm and the attachment point was put as far out on the control arm as possible to make the anti-roll bar as effective as possible. The end result can be shown in figure 3-3.

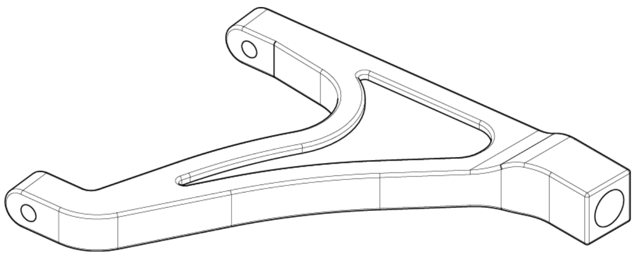


Figure 3-3 MacPherson lower control arm

3.2.4 Coilover Damper Mount

Since the coilover damper is an off-the-shelf component and designed for a RC car it was not the exact length and lacked a ball joint in the top. To solve the problem of length the coilover damper mount in the subframe was lowered. Since the ball joint was missing on the coilover damper a solution was to create a mount on top of the damper. It also has an attachment for a ball joint in horizontal position that is mounted together on the subframe. This design, seen in figure 3-4, enabled both the motion to be as correct as possible and was easy to manufacture.

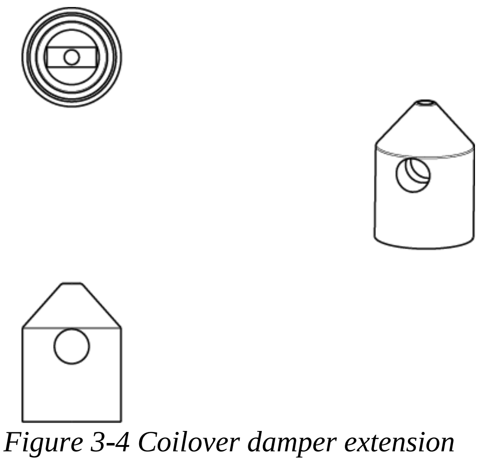


Figure 3-4 Coilover damper extension

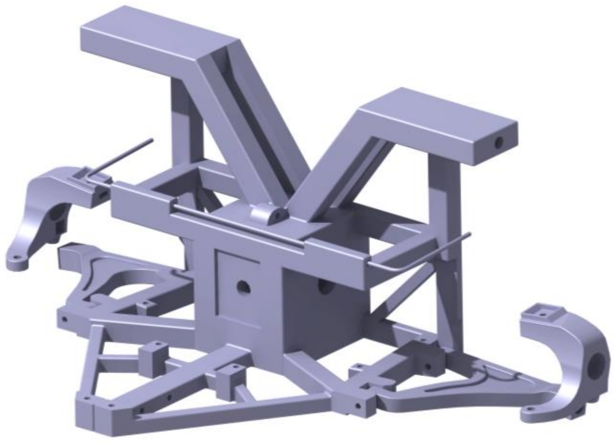
3.2.5 Anti-roll Bar

The anti-roll bar was the last addition to the assembly. According to the Chalmers MacPherson suspension the anti-roll bar was attached to the strut and connected via a drop link. Due to the design of the purchased coilover dampers this design was not possible. The bar was instead attached to the lower control arm and to make it possible to mount it was split into two parts and joint with a small coupling.

3.2.6 Assembly

The largest issue was, as for every part of this project, that bushings are not implemented in the models. This creates troubles for the assembly as it can only be minorly constrained or it will be locked. Thanks to the simplicity of the MacPherson the assembly could be moved around enough to ensure that the movements look correct. The differential created another complexity because there were no CAD files included in the purchase and due to time limitations it was not recreated in CAD. Therefore only measurements could be the deciding factor when making a decision to if the assembly would function or not.

With the assembly tool in Catia V5 there was a major fault discovered as the steering clashed with the subframe in the initial design. This was redesigned to allow the movement of the steering rack and due to this the anti-roll bar had to be moved together with the supporting structure of the coilover damper mount. The final assembly, without the steering components, coilover dampers, drop links and propulsion, can be viewed in figure 3-5 as a render.



Figur 3-5 MacPherson assembly render

3.3 Double Wishbone Suspension

The double wishbone suspension was unlike the other three suspensions designed in this project in one major manner. The geometry was entirely designed from scratch based on design requirements from the task. The design process was divided into five different steps. The first step was to set the basic dimensions of the suspension so it suits regular passenger car suspensions, for example wheelbase and track width. The second step was to set hardpoints for the wishbones and the upright, to get the right characteristics of the suspension. After that travel and steering was tuned by setting hardpoints for the coilover dampers and steering. With all critical hardpoints known a subframe, upright and wishbones were designed in CAD. The last step was to design an anti-roll bar and package it with respect to restrictions from other components.

3.3.1 Design Requirements

The following requirements were given for the double wishbone suspension. The data was given in kerb +2 and full scale.

Dimensions:

- Camber = -0.5 degrees, +/- 0.1 degrees tolerance
- Toe in = 0 degrees, +/- 0.1 degrees tolerance
- Lower wishbone angle = 0 degrees, +/- 0.1 degrees tolerance
- Camber gain = 21 degrees/meter, +/- 1 degrees tolerance
- Caster = 0 degrees, +/- 0.1 degrees tolerance
- Ground offset = 10 mm , +/- 0.1 mm
- Roll centre = 60 mm, +/- 1 mm
- Wheel offset = 52 mm, +/- 1 mm
- Bump steer = 15 degrees/meter, +/- 1 degrees tolerance
- Travel = -110 +80 mm, +/- 3 mm
- Wheel = 235/50 R19 8.0x19x52
- Static loaded radius = 85% of tyre profile
- Ackermann percentage at 20 degrees steering angle at inner wheel = 10-20%

General input:

- Package the outer control arm ball joints as wide as possible in z-direction
- Let the steering link be parallel to lower control arm
- Symmetry about the centre of the wheel in the x-direction

3.3.2 Basic Dimensions

The right characteristics can be achieved independent of the models size as long as the relations between all parts are correct. Some different dimensions can more or less be set without affecting the characteristics, if the design thereafter is made with respect to these dimensions. The dimensions are: subframe lower width, subframe upper and lower depth, lower wishbone length and spread of outer wishbone ball joints in the z-direction. To obtain realistic dimensions, known hardpoints from the Chalmers Macpherson suspension was used and scaled to 1:5. The lower MacPherson control arm dimension in the z-direction rounded up to closest integer led to 80 mm lower wishbone (Q, in figure 3-6) length in y-direction.

With the lower wishbone length and MacPherson track known alongside the horizontal lower wishbone requirement the subframe (S) lower width could be calculated to 147.6 mm in y-direction. The measurement between outer wishbone ball joints (M,N) is set so that there won't be packing problems under any circumstances. Distance from centre of ball joint to ball joint shell was unknown at this stage of the development so an extreme case was estimated to 6 mm as scaled dimension. Distance from ball joint centre to rim was set to 14 mm according to the Chalmers Macpherson suspension, the inner rim distance was 96.5 mm. That gave a spread between outer wishbone ball joints (M,N) of 56.4 mm. At last the width of upper and lower wishbone was considered, x-direction. It was not a critical measurement so the upper wishbone (R) was set to the same as the Chalmers Macpherson suspension lower wishbone (Q) and the lower to the same dimension plus 10 mm to add extra stiffness.

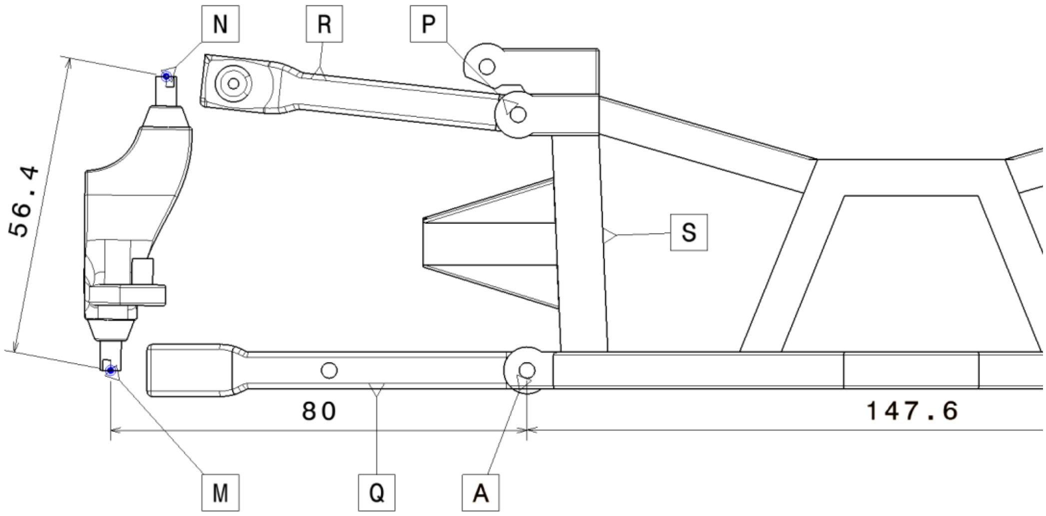
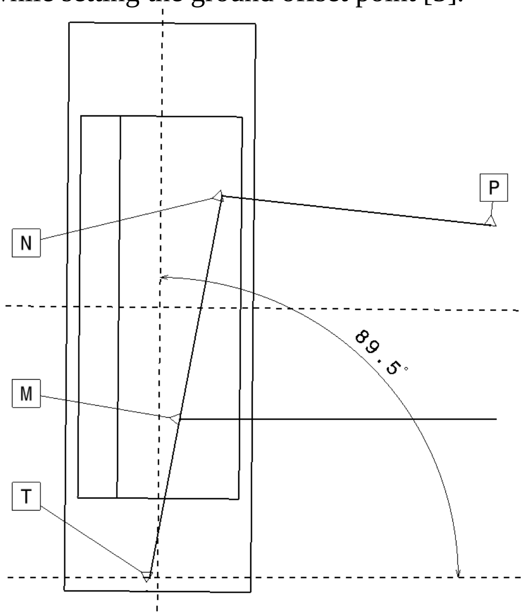


Figure 3-6 Basic dimensions [mm]

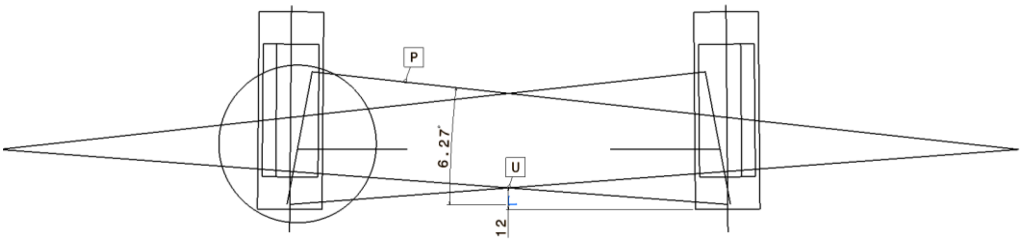
3.3.3 Wishbones and Upright Hardpoints

To obtain the right camber, caster, roll centre and ground offset, hardpoints for wishbones and upright were calculated. The design was done in a 2D sketch using Catia V5. Known basic dimensions were designed along with tyre and rim. To acquire the desired camber in design position the wheel was tilted 0.5 degrees negative while setting all hardpoints. Thereafter the upper wishbone outer ball-joint (N, in figure 3-7) hardpoint was set by designing a straight line through the lower ball joint (M) and ground offset point (T). Along that line the upper wishbone outer ball-joint hardpoint (N) was set. It was placed with the desired spread of 56.4 mm between the outer wishbone ball joints (M,N). The scrub radius point (T) was placed at the static load radius with an offset of 2 mm from wheel centre in y-direction. The deformation of the tire was also taken into consideration by subtracting 15% of the wheel profile radial while setting the ground offset point [5].



Figur 3-7 Upright hardpoints

Up until this stage the hardpoints that was set, in some words only determined static properties. The dynamic properties of the suspension require more advanced calculations and was based on the following method. The upper wishbone inner hardpoint (P, in figure 3-8) was set by using two restrictions, the given roll centre (U) and camber gain. The upper wishbone was extended with a line and a line was also drawn from where the wheel centre intersected with the ground. The lines was drawn out until they intersected with each other on the left and right side of the wheels. The roll centre (U) was defined by the point where the two lines from the ground offset intersect each other [4]. This point was therefore placed 12 mm above the ground according to the given data. That makes the made up geometry constrained regarding angels. The upper wishbone (R) got an angle of 6.3 degrees to the horizontal plane, as long as that angle and the so far set hardpoints are kept the correct roll centre will be achieved [11].



Figur 3-8 Roll centre design

The camber gain was then set by placing the upper wishbone inner hardpoint (P, in figure 3-9) at a calculated position along the 6.3 degrees construction line. This position was obtained by using circles, two circles had their radius set by the upper (R) and lower (Q) wishbone according to [4]. The change in y-direction after a certain change in z-direction could be obtained by following the border of the circles. The third circle (C3) was given the same radius as the length of the upright (E1), the centre of the circle was moved 4 mm in the z-direction and the radius line (E2) was connected to the upper circle (C2). The camber gain after 4 mm travel was then given by the angle difference between the upright (E1) and the radius line (E2). The length of the upper wishbone was adjusted until 0.46 degrees angle was obtained/4 mm travel.

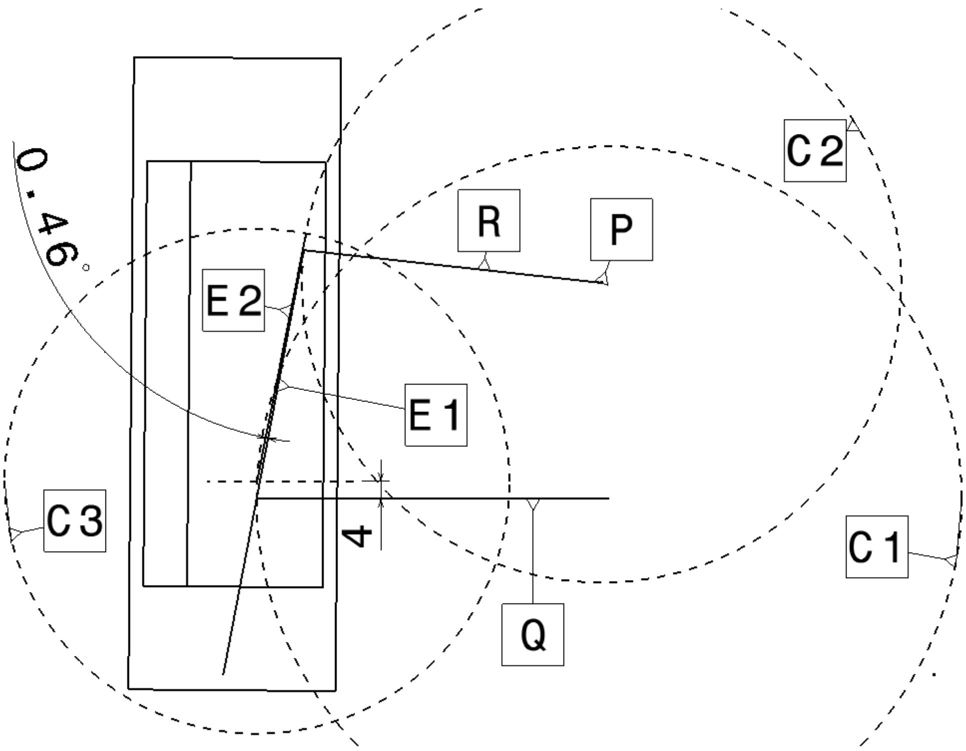


Figure 3-9 Camber gain

3.3.4 Coilover Damper Hardpoints

The correct travel was obtained by setting hardpoints for the coilover damper (D, in figure 3-10). The desired travel of +80 mm and -110 mm on a full scale vehicle which gave a total travel of 38 mm (1) in scale 1:5. The coilover damper that was used has a travel of 18 mm that gives a needed gear ratio of 2.1:1 (2) to acquire the accurate travel at the wheel centre. Some simplifications were made in the calculations. The angle of the coilover damper was not taken into account and the travel was said to be solely determined by the movement of the lower wishbone movement around its inner attachment and the extended lever that the upright (E) and rim contributes with, this simplification contributed to the desired extra travel and was therefore motivated. It is good to have extra travel when it comes to visualization and education. The total length of moving lever was then 82 mm.

The simplifications made it possible to let the attachment position of the coilover damper along the lever be used to get the right travel at wheel centre, a ratio of 2.1 gives that the coilover damper lower attachment (B) needed to be placed at most 47.6% (3) from the lower wishbone inner attachment (A). With a lever of 82 mm the coilover damper lower attachment (B) needed to be placed at most 39 mm (4) from the lower inner wishbone attachment (A), one millimeter was subtracted from the required measurement to be conservative and on the right side of the limit. That gave a final placement for the coilover damper lower end (B) of 38 mm (5) from the lower inner wishbone attachment (A).

The upper coilover damper attachment (C) was placed so that the total travel of the coilover damper gave 16 mm compression and 22 mm extension which correlated to the relationship with the desired travel of +80 mm and -110 mm on a full scale vehicle (6). Fully extended the coilover damper is 78 mm, that gave a compressed length of 65.6 mm (7) in kerb +2 design position. The coilover damper was centered in y-direction within the v-shaped area of the upper wishbone. That gave a y-position for the coilover damper upper attachment, 6 mm in the y-direction from the upper wishbone inner attachment. From the compressed design position of the coilover damper (7) and after centering the coilover damper, the upper attachment (C) position in the z-direction was set to 9.2 mm from the upper wishbone inner attachment. The hardpoints that were set in the previous steps were generated in a design environment and was verified as correct in a later stage, first through assembling in CAD and then physical testing. There was no need for reengineering.

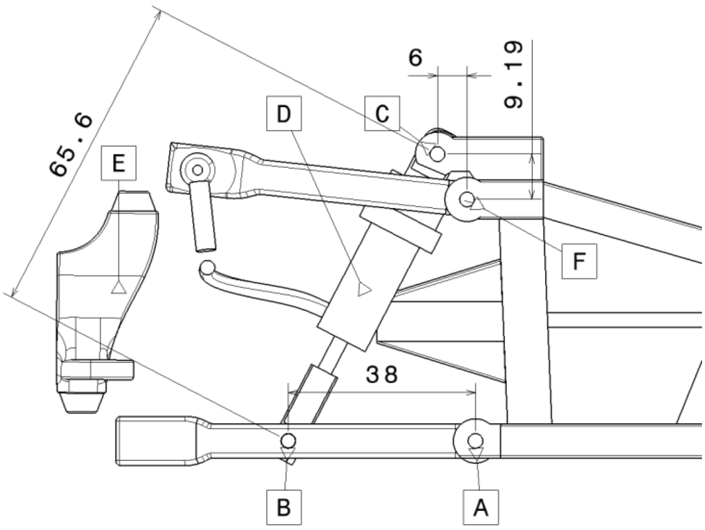


Figure 3-10 Coilover damper hardpoints [mm]

(1) $(80 + 110) * 0.2 = 38\,mm$
(2) $38/18 = 2.1$
(3) $1/2.1 = 0.476$
(4) $0.476 * 82 = 39\,mm$
(5) $39 - 1 = 38\,mm$
(6) $\left(\frac{80}{190}\right) * 38 = 16 \quad \left(\frac{110}{190}\right) * 38 = 22\,mm$
(7) $76 - \frac{110}{190} * 18 = 65.6\,mm$

3.3.5 Steering Hardpoints

The main steering properties that are affected by the steering hardpoints are Ackermann and bump steer. The same set of dimensions affects both of these properties. Therefore an iterative process was needed to achieve numbers on those two properties that were within the requirements. The starting point when setting these points was the upright from the Chalmers MacPherson suspension model. This upright has a CC offset from the kingpin axle (G, in figure 3-11) to the outer tie rod ball joint (I) of 31.2 mm and a distance of 2.1 mm in the y-direction. Those dimensions were used and the position in the z-direction was determined by the inner tie rod ball joint (J) and the criteria that the tie rod should be level to the lower wishbone.

The steering rack (H) was designed with 18 mm offset in the z-direction from the bottom to the tie rod ball joint (J), the bottom is the surface that will be mounted flush to the subframe. From the flat attachment surface of the subframe there was an additional 3.5 mm of offset to the lower wishbone inner attachment (A) that was used as reference when setting the steering hardpoints. That made a total of 21.5 mm from the lower wishbone inner attachment (A) to the inner tie rod ball joint (J) in the z-direction. To meet the requirement of parallel tie rod and lower wishbone, the outer tie rod ball joint (I) was placed 21.5 mm from the lower outer wishbone ball joint in the z-direction. Thereby base dimensions in all degrees of freedom were given for the outer tie rod ball joint (I).

The inner tie rod ball joint (J) already had a given position in the z-direction, the position and travel in the y-direction is given by the Chalmers MacPherson suspension model. 73.8 mm from the centre of the car and a travel of 16.5 mm in each direction. The position in the x-direction was at first set so the tie rod is parallel with the y-axis. Thereafter the inner tie rod ball joints was moved in x and z direction until the right bump steer and ackermann was achieved [4]. By changing the position in z-direction the bump steer was mainly affected. By moving it in x-direction the ackermann was mainly affected. The final x-direction position was 41.5 mm from wheel centre and the final z-position was 21.5 mm from the lower inner wishbone attachment which was the same as the original position from the steering rack height.

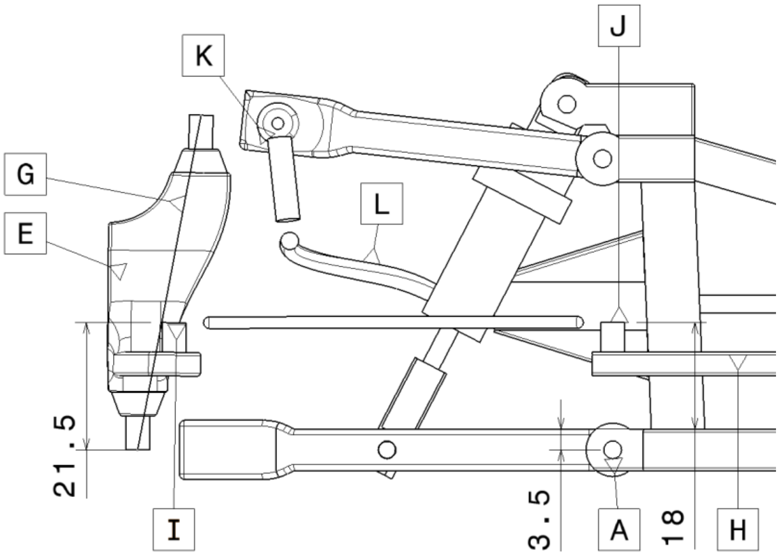


Figure 3-11 Steering hardpoints [mm]

3.3.6 Anti-roll Bar

The purpose of the anti-roll bar on the model was to show the basic functions. The design was therefore only based on packaging limitations. The drop link (K, in figure 3-11) lower end moves towards the upright and the front of the vehicle while the wheel travels due to the upper wishbone rotation centre. Therefore the drop link (K) lower end was tilted towards the rear and centre of the car in design position to counteract the unintentional movement. The drop link and outer end of the anti-roll bar was placed as far as possible from the centre of the car and the two ends were combined by a U shaped anti-roll bar (L) supported by bushings in the subframe according to [11]. The anti-roll bar was designed with a coupling in the centre to make it possible to detach it when desired.

3.3.7 Assembly

During the whole design process parts were designed in CAD immediately after hardpoints that were linked to the specific parts were calculated. It was done to be able to assemble the parts in CAD and make sure that they work together as expected before setting other hardpoints that could be affected by limitations detected in the assembly, but also to avoid manufacturing faulty parts. The part design was done by connecting hardpoints of each part in a esthetic and mechanically necessary way but without concern of stiffness. The complete assembly of the suspension can be seen in figure 3-12, all designed parts are included except from the premanufactured ball joints and bearings.

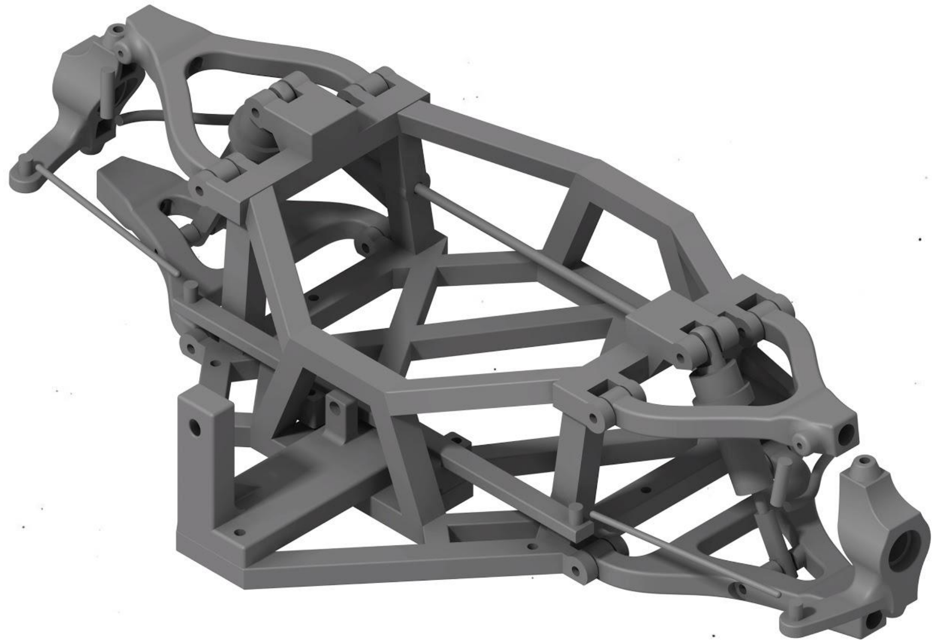


Figure 3-12 Render of double wishbone assembly

3.4 Trailing Arm Suspension

The trailing arm suspension was the other suspension being reengineered from an existing setup. Hardpoints were given from the Chalmers Trailing Arm suspension. The main task was therefore to connect those hardpoints by both the different links and a subframe.

3.4.1 Links

The links were designed one by one. With the coordinates for each point put into Catia V5 the shape between them were drawn by the usual looks of the parts to get the the model to be as realistic as possible according to the Chalmers suspension. They were all made planar and straight except for the tie blade and the camber link. The tie blade, figure 3-15, was made with a slight bend to allow for it to stress the point on the subframe in a longitudinal direction without to much rotation when such forces are applied. This shape also makes it rotate well when the wheel is traveling in a vertical direction. The camber link, figure 3-13, was designed with an “s-shape” to clear the side member, which is a part that supports the side structure of the car and adds strength, as the problem of packaging everything in the space available is a common problem. The spring link, figure 3-16, was designed as a flat piece with a circular platform in the middle where the spring would sit on the real size car without a coilover damper. It was also designed with a slot where the coilover damper could be slid into and secured with a screw. The toe link can be seen in figure 3-14.

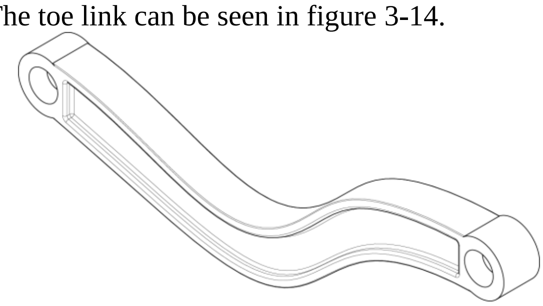


Figure 3-13 Camber link

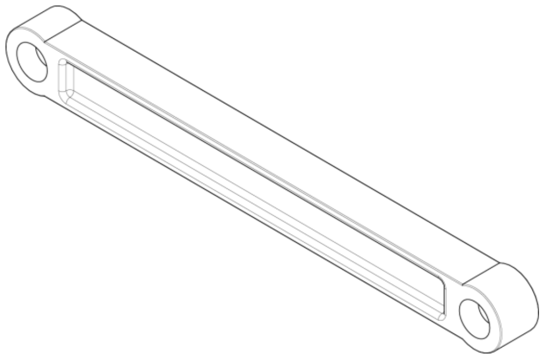


Figure 3-14 Toe link

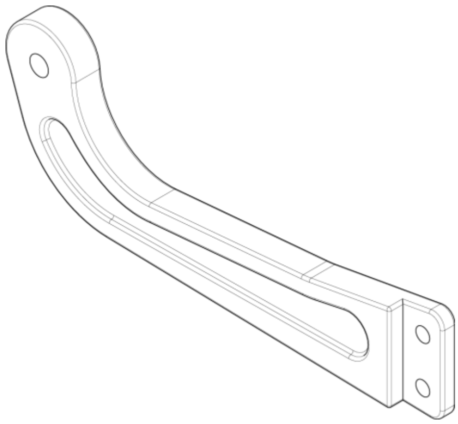


Figure 3-15 Tie blade

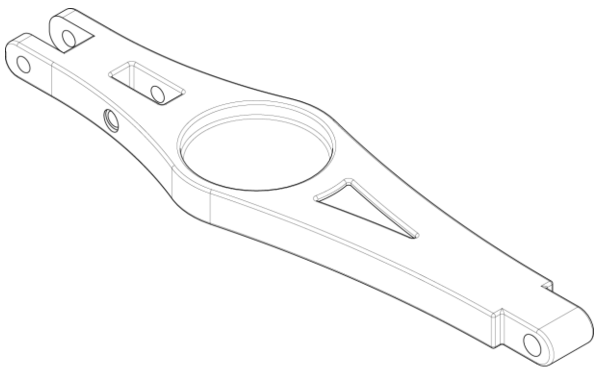


Figure 3-16 Spring link

The first drafts were mostly kept in the final design except for the holes which were redesigned with a conical shape, which represents a bushing, to allow for them to twist as the wheel suspension would get over constraint otherwise.

3.4.2 Subframe

With the inner hardpoints used to design the links, the subframe could also be designed as it uses exactly the same ones. These were put into a 3D space and connected in a manner that would take as little space, and plastic from the 3D printer as possible, but still be stiff enough to not get any fractures. As stated in the project there were no calculations done to verify the strength of the frame, but they were designed in a way that reminds of a truss, which is known among mechanical engineers to be a durable design. As this part was the first one to be test-printed there was a difference in the design since the strength of the PLA was at that point unknown. After some testing it was found to be too little play between the subframe and the links. It was also thicker than necessary and therefore a lot of material was removed before the next print to save material and lower the time in the printer.

To get the correct percentage of jounce and rebound, the hardpoint for the upper mount of the coilover had to be moved 2.3 mm upwards along its central axis. As the hardpoint stayed on the axis, the movement of the suspension would not be affected. The final design of the subframe can be seen in figure 3-17.

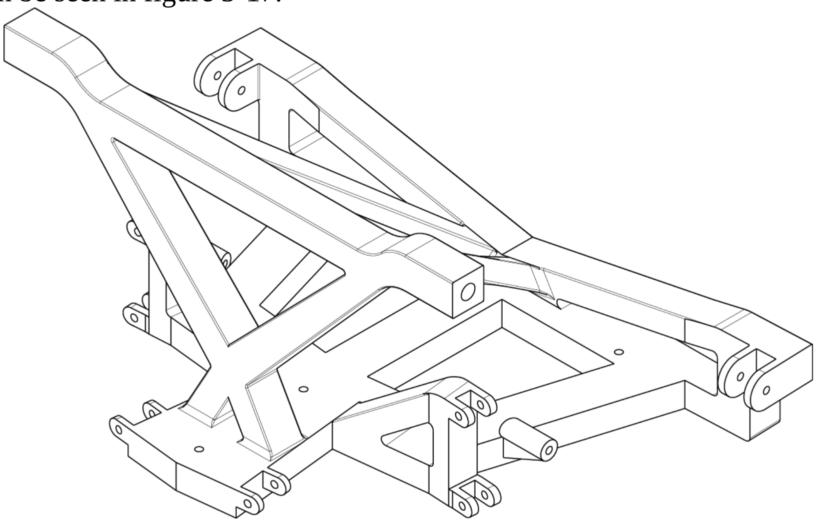


Figure 3-17 Subframe trailing arm

3.4.3 Upright

Based on the outer hardpoints of the links the upright, figure 3-18, was created. The hardpoints were placed in the 3D space in Catia V5 and linked together to a straight block. Refinement of the shape was made and attachment points were created for the different links. As could be seen in an assembly of the whole suspension in Catia V5, there were problems with getting everything to fit in the rim. There were also a few brackets for the links that were under dimensioned and therefore broke. Because of this a redesign of the upright was made. After 3D printing the prototype and mounting it together with all components, areas in need of improvement were detected. Improvements such as removing material that was blocking the links from proper movement was done. There were also slots made to fit the bearings for the wheel axle.

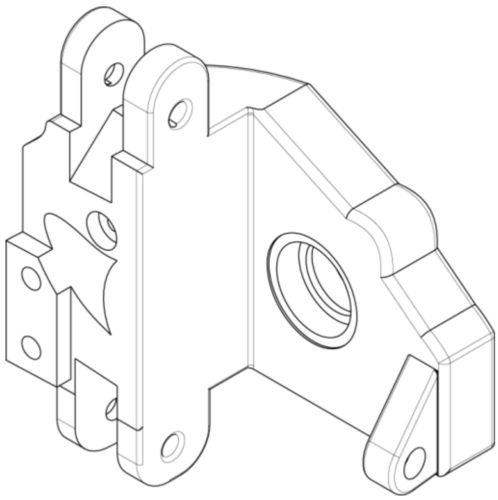


Figure 3-18 Upright trailing arm

3.4.4 Anti-roll Bar

With gathered knowledge of the function of an anti-roll bar it was designed based on available space, hardpoints and maximum functionality. The drop link was made out of two ball joints and the anti-roll bar itself out of a 2 mm steel rod. The rod was machined by cutting two parts to the desired length, then threading the outer sides on each of them. To get the correct shape they were bent and then linked together by a coupling, see figure 3-19, which had to be mounted with an offset to fit to the subframe. The coupling makes it possible to switch between having an anti-roll bar or not. To make it work properly was shown to be a repetitive processes and that was where the advantage of rapid prototyping was useful.

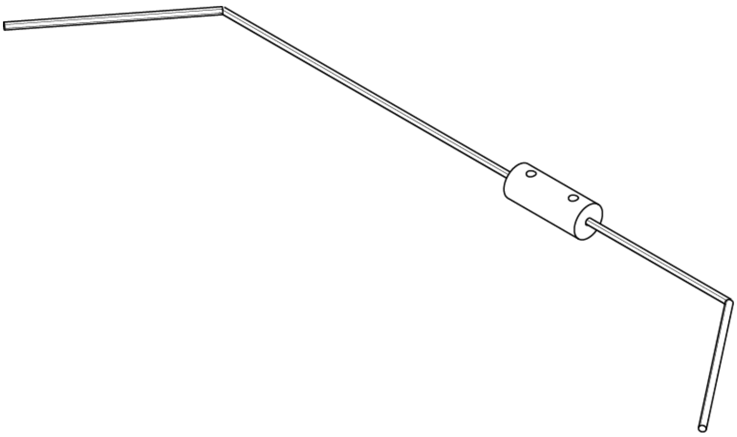


Figure 3-19 Anti-roll bar trailing arm

3.4.5 Assembly

The last step was just a matter of deciding how the connections were to be made in Catia V5 and making sure that all parts would fit together in the design position. Most of them were screws either making it fully constraint or allowing rotation around one axis. All the parts were connected with 1-2 mm of width difference to allow for some play since this was a requirement as the suspension can't move without it. Shock absorbers that fitted perfectly to the hardpoints were impossible to find, and the most suitable ones were a bit too short, therefore an extender was designed. It was made to offset the upper hardpoint 16.25 mm downwards along the shock absorber centre line and was designed as described in the MacPherson chapter (3.2). A rendering of the assembly can be viewed in figure 3-20.

Figure 3-20 Render of trailing arm suspension

3.5 Live Axle Suspension

Since there were no existing live rear axle information to begin with the base was taken from the trailing arm suspension. The existing information, such as hardpoints, from the trailing arm was combined with photos of a live axle suspension setup from a Volvo 940 and a schematic image of another live axle version, see figure 3-21. This was done to be able to reengineer a new live axle suspension with trustworthy accuracy and correctness, but due to the lack of information the suspension is not entirely correct.

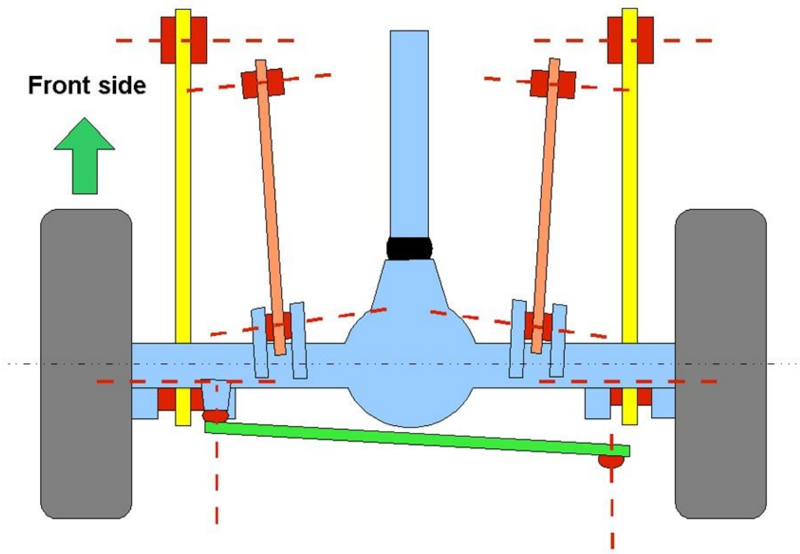


Figure 3-21 Axle - 5 Link rigid 03. From [11]. CC-BY- SA.

3.5.1 Axle

Since the axle itself is the main part of a live axle suspension that was the starting point. From the hardpoints of the Chalmers trailing arm suspension the y-coordinate for the wheel centre was given and with it the length of the axle could be calculated as:

$$\mathrm{L} = 2 * \frac{793.2}{5} = 317.3 \text{ mm}$$

This gives the length to be 317.3 mm in scale 1:5, but since that was too long for the 3D printer available it was split in half. In the centre of the axle a sphere was placed to represent the rear final gear. This was instead of an actual final gear since the model will not have a propulsion system on this axle. On each end of the axle mounting for the wheels was required. This was done with the help of two bearings so the diameter at each end needed to be increased to accommodate them. It also needed to have space for a screw to be screwed through the bearings, with a slot on the inner side to allow mounting of a screw-nut. Furthermore the panhard rod needed some form of mounting which was made as a hole through the axle and a spacer, similar to what is seen in figure 3-21. The spacer was needed to stay clear of the thought final gear as the axle moves vertically. The final part of the axle was the mounting for the additional stabiliser arms. This mount was made to accommodate ball joints so that the axle would not be locked in any direction except revolving around its own centre line. An additional guard on each side for the support arms had to be added so that they would not move around along the axle. The final design can be seen in figure 3-22.

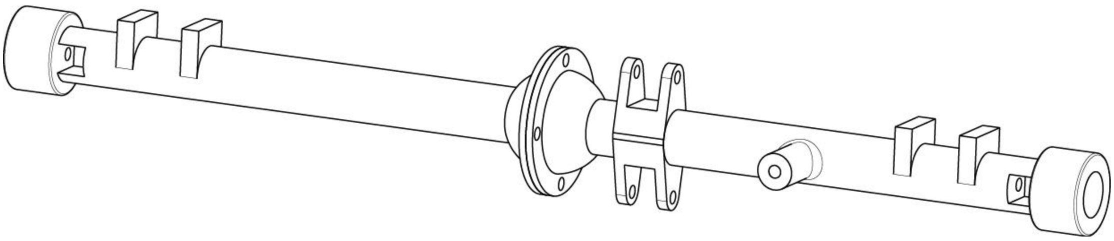


Figure 3-22 Axle

3.5.2 Support Arm

Close to the end on both sides a support arm, see figure 3-23, is mounted to the axle and connects it to the coilover damper. The support arm has no exact equivalent on the trailing arm but the hardpoints for the tie blade were used to get an approximation of the length and height difference. This gave the basic dimensions and made it possible to move on and create a mounting point for the coilover damper. Most live axle suspensions uses separate springs and shock absorbers and therefore two mounting points. With the coilover damper purchased only one mounting point was needed and the one where the springs are originally mounted were chosen. Since the bought coilovers had a ball joint in the lower mounting point all that was needed was a vertical slot to accommodate the ball joint and a horizontal hole for a screw.

The next part was mounting the axle itself to the support arm. The axle needs to be able to rotate around the x-axis of the car and because the arm will move vertically special mounting was needed. This was done by narrowing and rounding the arm where the axle is to be mounted. A similar design with the rounding was done on a complementary piece to be able to strap the axle down onto the support arm. This design should allow for more movement and not cause the suspension to be completely rigid. Since there are no bushings the holes to mount the support arm to the subframe were made with a conical shape to also allow for some movement.

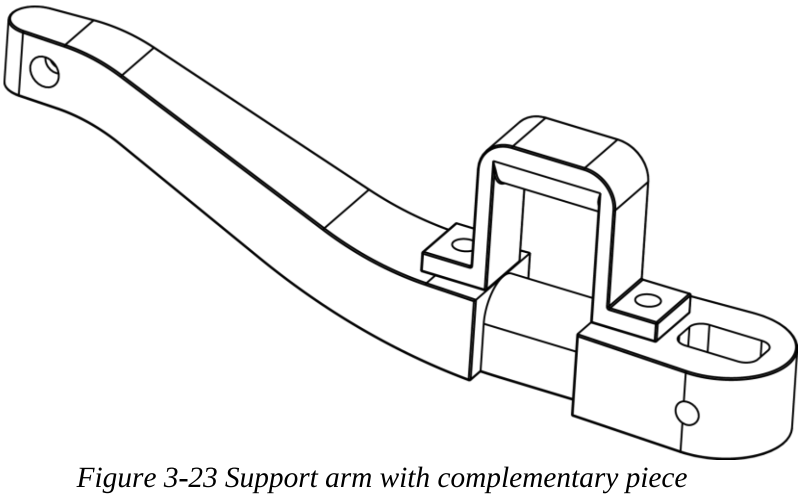


Figure 3-23 Support arm with complementary piece

3.5.3 Panhard Rod and Stabiliser Arms

Both the stabiliser arms, figure 3-24, and the Panhard rod, figure 3-25, were made as straight round rods in appropriate lengths. Since these are screwed in with ball joints all ends needed to be made with a hole to accommodate them. The holes needed to be 16 mm deep and have a diameter of 6.5 mm so that the ball joints would fit. Instead of having the entire Panhard rod having a uniform thickness the middle part as made thinner to not waste material. The stabiliser arms however were made uniform because they are much shorter and does not have much space left in the middle section.

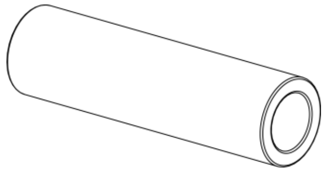


Figure 3-24 Stabiliser arm

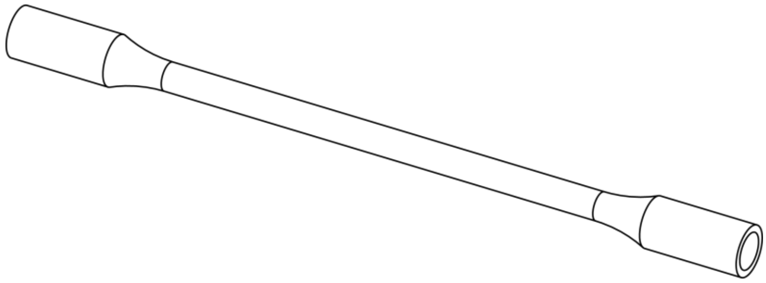


Figure 3-25 Panhard rod

3.5.4 Subframe

To connect and hold all components in place a subframe, figure 3-26, was needed. The first step was placing all the points connecting the different components in the 3D CAD space. The points were found from knowing all dimensions of the different components and how and where they were connected. Next step was to create the mounting solutions which differs depending on the type of connection used. For the support arm a simple hole to mount with a screw and for the Panhard rod as well as for the stabiliser arms similar solution but able to accommodate a ball joint. What remained after this was to connect all the mounts together without obstructing the travel of the axle and at the same time not using an unnecessary amount of material.

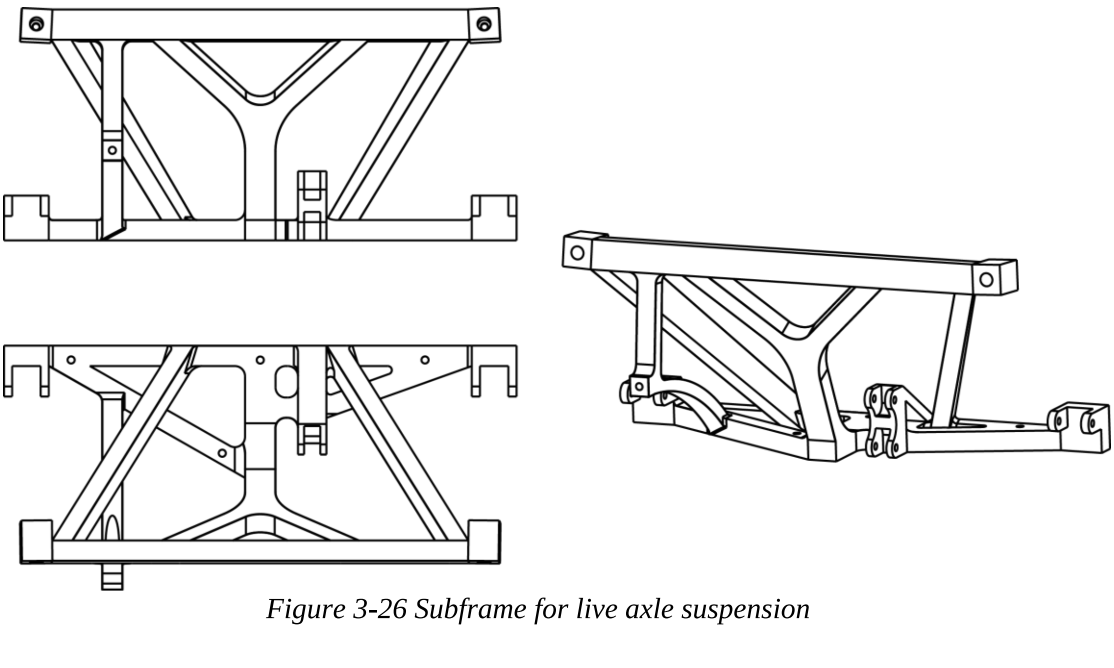


Figure 3-26 Subframe for live axle suspension

3.5.5 Assembling

When assembling the suspension the screws were implemented as axial constraints so the only allowed movements were along the axis and rotation around it. This was limiting in the sense that the bushings which could be mounted in these position normally are not compensated for at all. The compensations made with conical holes as well as larger holes would not help either. For ball joints a distance constraint was set so that the rotation possibilities were kept. This solutions allows more than the joints do but when the movements are normal it should give a fair representation. The final assembly can be seen as a render in figure 3-27.

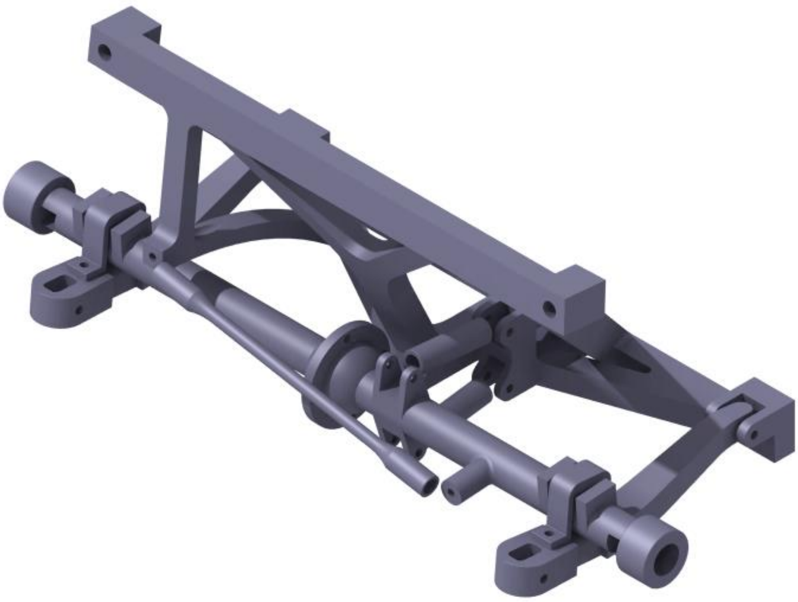


Figure 3-27 Render of live axle suspension

3.6 Chassis

To get the correct kinematics out of the front and rear suspensions when applying force they had to be linked together. This was naturally done with a chassis, to which the subframes were attached. It was important that the chassis were stiff to not affect the behavior of the suspensions in a way that conflicts with the kinematics, as this is desired in this model.

The main target was to fit the chassis to the wheelbase and the subframes. The wheelbase was a given value at $\frac{2735}{5} = 547\ mm$ and for the subframes it was just a matter of fitting to their different designs. This was done by taking the measurements of the different angles and points and transfer them to the chassis. Matching holes were made to allow for a firm fitment between the parts.

One of the requirements from the task was to develop two different chassis to get all the different combinations and have them working at the same time. Because of the similarity of the two front suspensions, MacPherson and double wishbone, it was chosen that those were the ones that were going to be interchangeable. The other two, trailing arm and live axle, were permanently attached to the rear of the two chassis, which can be viewed in figure 3-28 and 3-29.

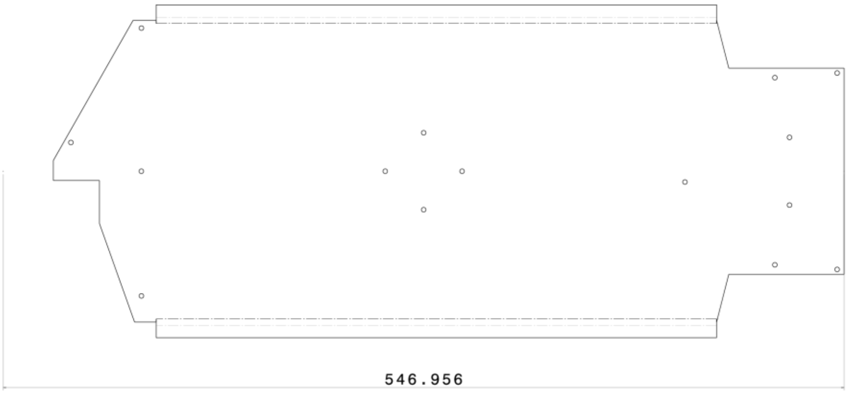


Figure 3-28 Chassis for live axle

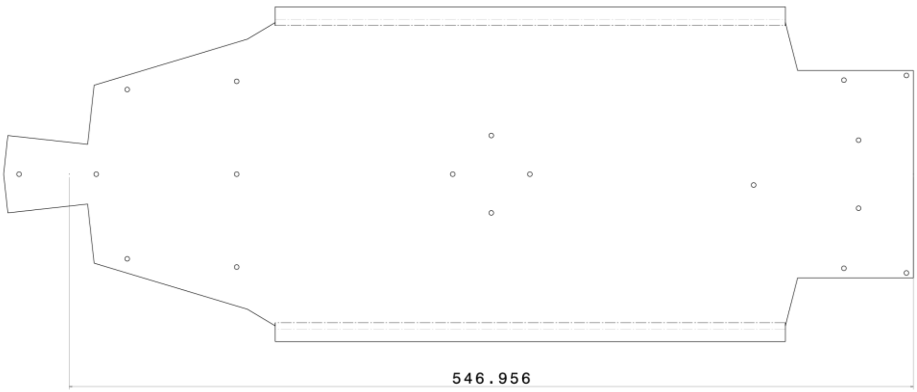


Figure 3-29 Chassis for trailing arm

The workbench “generative sheetmetal design” was used in Catia V5 which has built in features that makes the part fit for bending without having to design the bend radius manually. The sides of the chassis were designed with an edge bent downwards to make the design stiffer in bending, as with cold processing there are large strains building which increases the dislocation density and therefore makes it harden, according to [12]. As the chassis were supposed to be cut by waterjet they had to be manufactured from a flat sheet of 2 mm steel. This was where the built in features came in handy. From the 3D-model with walls bent downwards there was a simple way to make it lay flat with the walls bent outwards and the extra material automatically added needed to make the bend possible and still get the desired measurements. There were also a number of holes made in the centre to form a screw joint between a handle and the chassis. This handle was designed to make it easier to apply a force in the desired direction without having to use two hands. Renders of the two chassis can be seen in figure 3-30 and 3-31.

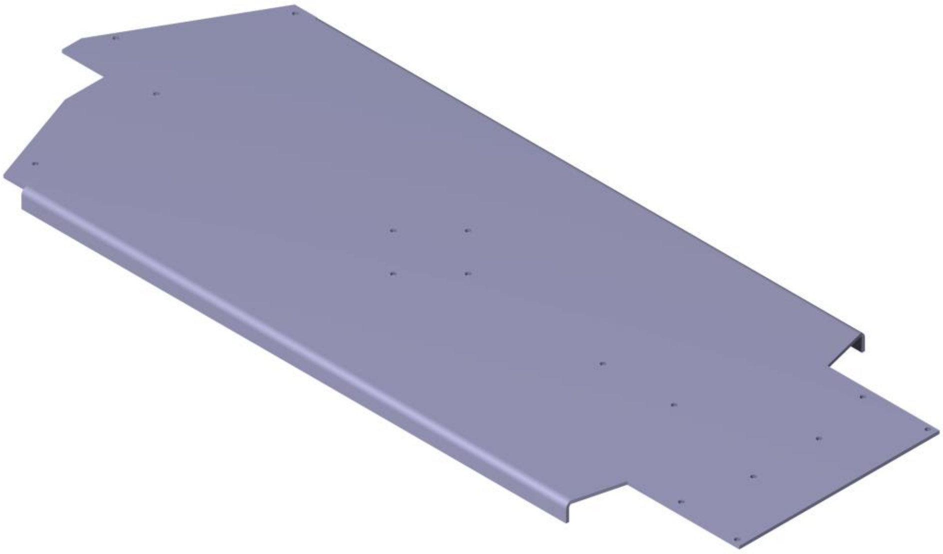


Figure 3-30 Render of chassis for live axle

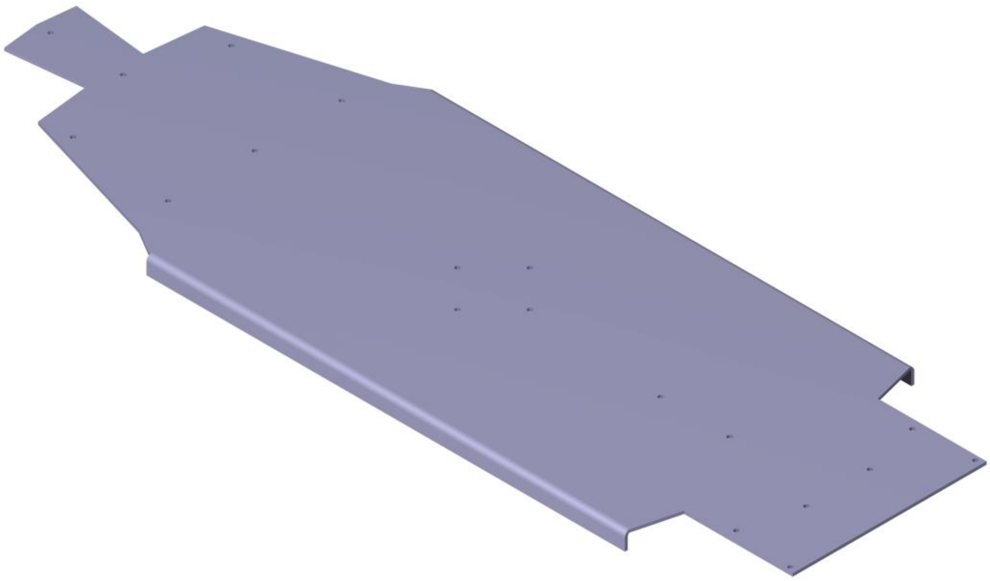


Figure 3-31 Render of chassis for trailing arm

Because of limitations in the bending machine in the prototype lab, the width of the chassis had to be reduced to 200 mm, whereas a more realistic width would have been 360 mm to match the real suspension.

3.7 Steering

In order for the model to be able to demonstrate how turning the wheels affects the behaviour of the suspension a simple steering mechanism was needed. The steering mechanism chosen was a rack and pinion system since it is used in most passenger cars today.

3.7.1 Subframe

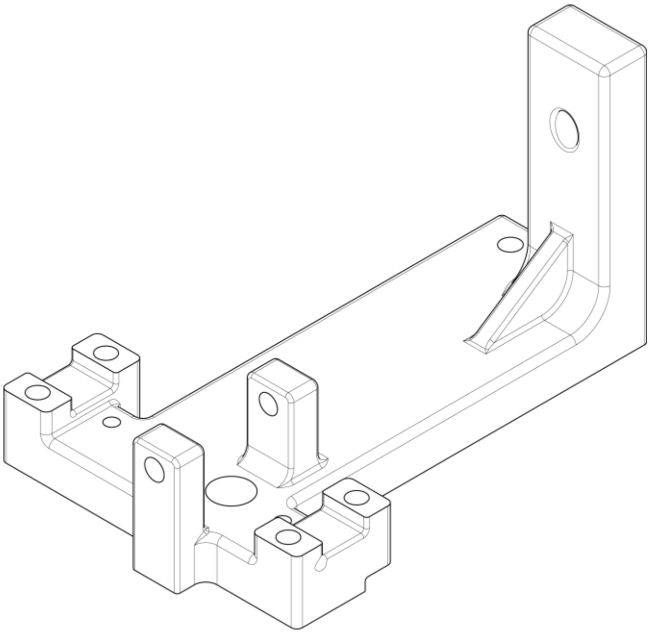


Figure 3-32 Steering subframe

The starting point for the design engineering process of the steering linkage was designing the subframe that would hold all the components in place. Because the suspension models are modular, it was important to ensure that the steering subframe could be mounted on either of the two front suspension modules. This was done by deciding on a flat profile for the front and underside that could be positioned behind the suspension subframes on a small platform sticking out from the back of the suspension subframes. The subframe was designed to have two y-shaped stands with mounting brackets on top to support the steering rack and allow the steering rack to slide from side to side. To support the steering shaft and pinion, three supporting towers with holes for the steering rack and pinion axle to pass through. Because the rack and pinion are positioned horizontally but the steering rack usually is not, a universal joint was needed to transfer the rotation of the steering rack to the pinion. The U-joint consists of two so called yokes which are connected by a cross which allows rotation in in any direction. The U-joint was designed to be as small as possible but still be able to clamp onto a 3 mm shaft.

The U-joints turned out to be far too small for the available 3D printers but unlike the rack and pinion the U-joints were not parametrically designed, so making them larger would take a lot of time. Because of the difficulty of redesigning them it was decided to order off-the-shelf U-joints to save time and ensure proper operation. The available U-joints were much larger than the designed ones and that caused clearance issues between the outer edge of the U-joint and the rear support tower. To address this the rear support tower was made one millimeter slimmer.

During test assembly it was discovered that the pinion axle did not have enough support which caused it to twist along the y- and z-axis and reducing the precision and feel in the steering. The twisting was eliminated by adding a third support tower in front of the steering rack at the very front of the steering subframe. The finished design of the steering subframe can be seen in figure 3-32.

3.7.2 Steering Rack and Pinion Gear

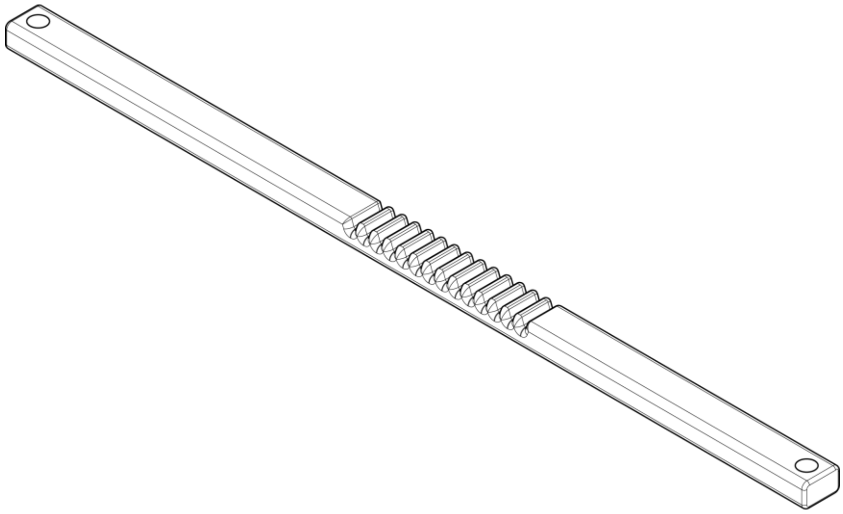


Figure 3-33 Steering rack for double wishbone suspension

To transfer the rotational movement of the steering wheel to the wheels of the car a rack and pinion is needed. The rack for the double wishbone suspension can be seen in figure 3-33. The pinion was designed by first setting the module to 0.5 mm and the number of teeth to 14. This gave a good compromise between being printable but at the same time not resulting in a gear ratio that is too high compared to the real car. The pressure angle was chosen to be 20 degrees as this is the most common pressure angle in gears. These values were the only ones needed to calculate the remaining dimensions of the gear. The following equations were used to design the pinion.

$$Number\ of\ teeth\ z = 14$$
$$Module\ m = 0.5\ mm$$
$$Pitch\ circle\ radius = \frac{mz}{2} = 3.5\ mm$$
$$Addendum\ circle\ radius = \frac{mz}{2} + m = 4\ mm$$
$$Dedendum\ circle\ radius = \frac{mz}{2} - 1.25m = 2.875\ mm$$

The rack was based on the Swedish standard SMS 296 [13] which can be seen in figure 3-34. It has a pressure angle of 20 degrees which matches that of the gear that was previously designed. In order for two gears to mesh properly, they need to have the same pressure angle and module so the rack was given the same module and pressure angle as the pinion. The minimum length of the steering rack was given by the requirement to allow for 16.5 mm of rack travel to either side. This meant that the geared part of the rack had to be at least 33.0 mm long. The distance between the two inner ball joints was given by the hardpoints and was 148 mm. The total length of the steering rack was required to allow for the inner ball joints to be mounted 148 mm apart. The height of the steering rack was not a given dimension so it could be decided freely. It was set to 4.5 mm as this was considered to be thick enough since the steering rack would not bear any load in the z-direction. The position of the ball joints in the z-direction is critical and is given by the hardpoints. This position had to be matched by adjusting the z-position of the steering subframe relative to the suspension subframe. The width of the steering rack was also unrestricted but was chosen to be 7 mm to give sufficient margin for the holes for the ball joint mounting screws.

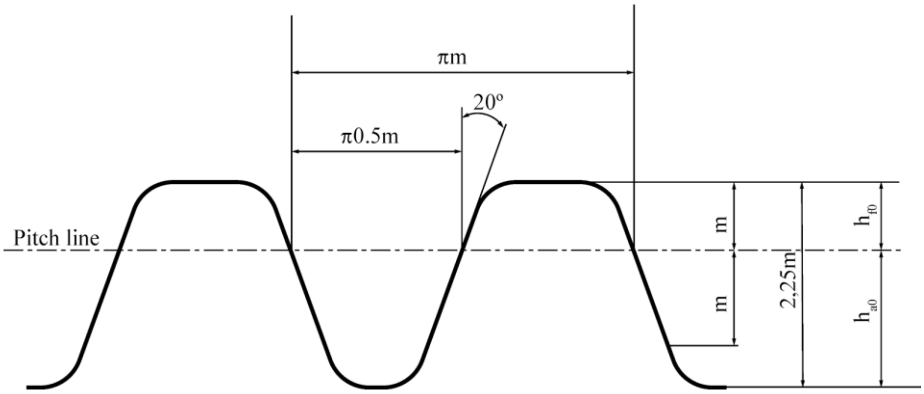


Figure 3-34 SMS 296 Reference Profile

The pinion and rack need to be mounted with the correct spacing to make sure that they mesh correctly. This distance is given by the formula $a = Do2 + \frac{Do1}{2}$[14] where Do1 is the pitch circle of the pinion and Do2 is the distance from the base of the steering rack to the pitch line which can be seen in figure 3-35.

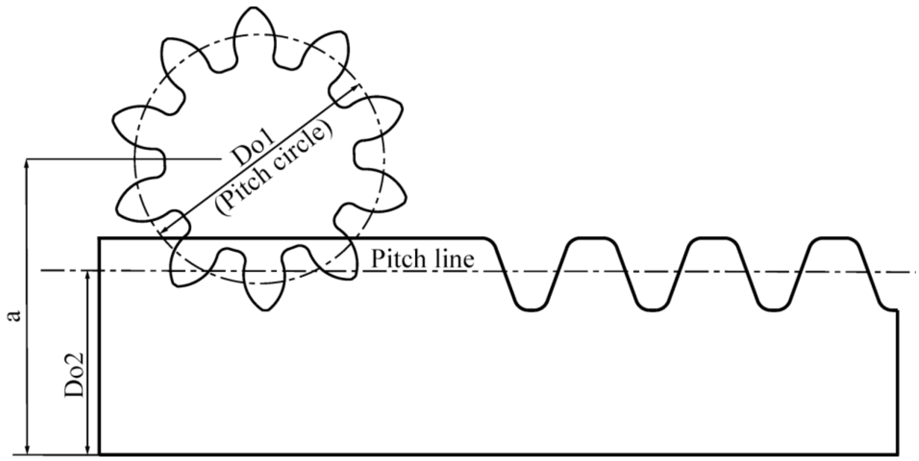


Figure 3-35 Axle distance for rack and pinion

Since the model with MacPherson struts also has a differential that occupies most of the room where the steering rack normally would be mounted the steering rack had to be moved along the y-axis to clear the differential housing. The problem with moving the steering rack is that when the inner tie rod ball joint moves it changes the steering dynamics i.e bump steer and ackerman. To avoid this, the steering rack was designed to have “arms” extending forward so that the inner tie rod ball joints still can be mounted in the specified locations (See figure 3-36). Changing the length of these arms does not affect the steering geometry so it can be adapted to fit different suspensions.

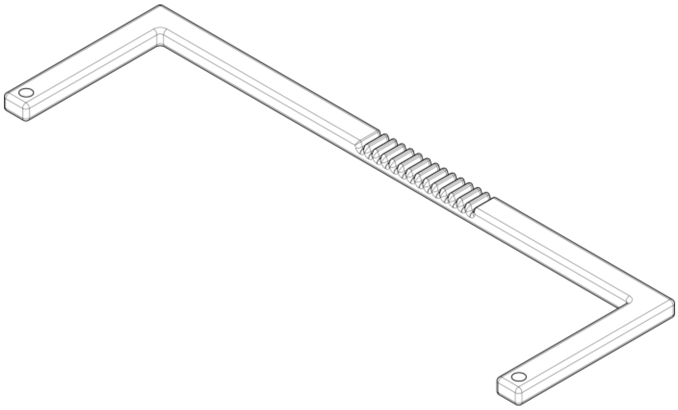


Figure 3-36 MacPherson steering rack

After the first iteration of the rack and pinion was 3D printed it was clear that that the 3D printer was not able to print such fine teeth as those on the rack and pinion. To overcome this the number of teeth on the pinion was reduced from 14 to 10 and the module was increased from 0.5 to 0.8 mm on both the rack and pinion to make the teeth larger.

3.7.3 Pressure Plate

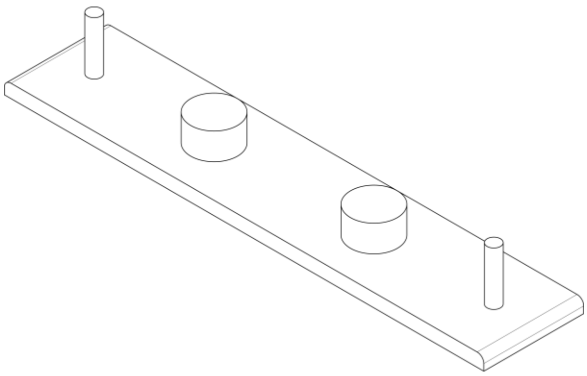


Figure 3-37 Pressure plate viewed from the bottom

Because of the limited accuracy of 3D printing and the tight tolerances needed for the rack and pinion to mesh properly it was decided to make the rack spring loaded to minimize the risk of the gears slipping. This was done by designing a pressure plate to be placed under the steering rack. To assure that the pressure plate would not move in any other direction than up or down, guiding pins were added to the underside of the pressure plate. Holes for the guiding pins were made in the corresponding locations on the steering subframe. The steering subframe did also get holes where two small compression springs could be placed and push up on the pressure plate. When the steering was assembled on the double wishbone suspension the steering rack would not move smoothly and the problem was derived to the spring mechanism pressing up too hard on the rack causing the rack and pinion to bind. When the steering was tested without the springs and pressure plate it worked perfectly so the spring mechanism was abandoned for the double wishbone suspension. It did however work on the MacPherson suspension. The finished design of the pressure plate can be seen in figure 3-37.

3.7.4 Steering Wheel

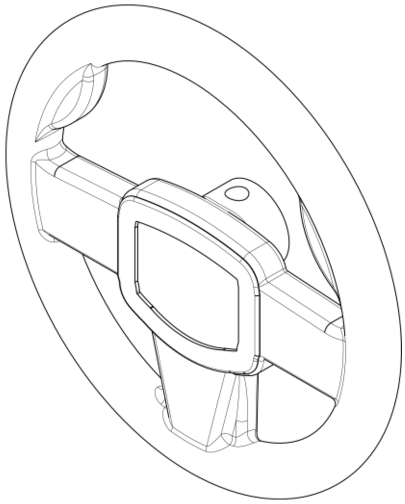


Figure 3-38 Steering wheel

To make it easier to turn the wheels a steering wheel was needed. The steering wheel was designed with a sporty look to fit the characteristics of the suspension models. It has an outside diameter of 67 mm which equates to 335 mm in full scale and is close to a typical steering wheel diameter. A hole was made in the rear of the steering wheel for the 4 mm steering shaft to slide into. To secure the steering wheel to the steering shaft a hole for a 3 mm screw was made perpendicular to the steering shaft. The steering wheel can be seen in figure 3-38.

3.8 Tyre and Rim

To be able to see the different angles during steering and travel of the suspension wheels were needed. The dimensions of the tyre and rim was based on a common 19 inch wheel with tyre dimensions 235/50 R19 and rim dimensions 8.0x19x52. These dimensions have been converted to the scale 1:5 to fit the model. The number 235 is the value of the width of the tyre in millimeter and 50 gives the proportion of the height of the tyre based on the width of the tyre. So the height of the tyre is 50 percent of 235 mm, which is 117.5 mm. R19 stands for the size of the tyre which is 19 inch in diameter. In the following designation for the rim 8.0x19x52 the first number indicates the width of the rim, in this case 8 inch. The second number indicates the diameter of the rim and is 19 inch. And the third number indicates where the mounting point of the rim is. Wheel centre (offset 52 mm) is the point in the centre of the rim where it will be attached to on the hub. This point positions the wheel in y-direction to give the intended steering and suspension geometry. Rim and tyre can be viewed in figure 3-39.

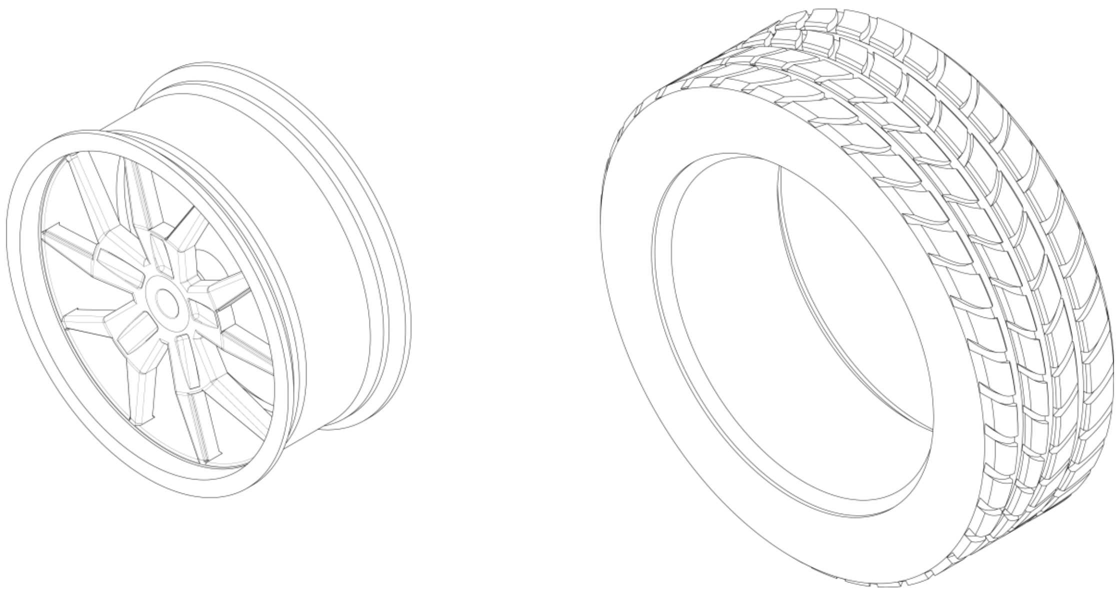


Figure 3-39 To the left is the rim and to the right the tyre

During the design phase parameters as strength and possibility to 3D print was considered. A redesign of the centerpiece of the rim had to be made to enable mounting of the drive shaft and get drive to the wheel. A standard pattern has been done on the tyre to symbolise and get a realistic feeling. The tyre and tre pattern was not in any way tested for grip it was only tested to work as a standing model. To save time and get as few different components as possible the design of the tyre was made symmetrical to enable usage on both sides. To make the rim more convenient to print and eliminate a lot of support material a 45 degree angle was made on the inside of the rims sharp edges that keeps the tyre from sliding off. This modification was also made on all sharp edges on the tyre to improve the surface of the printed tyre. When printing in a softer and elastic plastic material the 3D printer has a hard time printing sharp edges because it rips loose the threads it is making. A major improvement of the surface of the tyre was seen when the modification was applied. The final wheel can be seen as a render in figure 3-40.

Figure 3-40 Render of tyre and rim mounted together

3.9 Simulation

The simulations carried out were mainly done in Catia V5. For simple geometry changes it is beneficial to use Catia V5 and easy to access since it is the workbench where the majority of every suspension and detail has been designed. Through Catia V5, clashes of assemblies can be detected and end point values for suspension travel can be verified.

The shortcoming of Catia V5 is that it is not effective for analysing more than one data point. To solve this issue a software called “Optimum Kinematics” was used and with this software simulations can be done for different scenarios. By importing the hardpoints of the suspension and defining wheels and static characteristics a simulation can be made. Simulations can then be done for heave (bump), steering or roll. Defining a scenario, e.g. heave, the software will simulate the travel of the chassis vertically to the limitations defined by the user. What is important here is that when the chassis travels a given amount of millimeters upwards it represents the suspension traveling downwards. By doing this appropriate graphs could be drawn for camber gain and bump steer. Unfortunately, the software did not support the rear suspensions due to the limitations of the licens. Because of this, simulations have only been done for the two front suspensions.

3.10 Manufacturing Parts

3.10.1 3D Printing

Before starting the printer, files from Catia V5 were saved as stl files and prepared in the software “Simplified 3D” or “Makerbot Print”. The program inserted supports and rafts to the product if needed and converted it into a file format that could be read by the printer. The desired filament material and colour was loaded in the 3D printer and preheated to the working temperature. The 3D printers used were a Makerbot Replicator 2, a Makerbot Z18 and a ZYYX+ and are located in the course lab at Chalmers. In figure 3-41 the Makerbot replicator 2 can be seen and the ZYYX+ in figure 3-42.

When the previously described steps were finished the printer was started and left to print the desired part. It could take anywhere between 15 minutes and 48 hours depending on the size of the parts and the amount of plastic needed. Common problems that could occur when 3D printing were filament rolls that got stuck, lack of filament on the roll which made the print cancel, filament that got stuck in the extruder and nozzle, the part coming loose from the build plate it is printing on, defects because of loose supports and the extruder getting out of position.

3D printing is an iterative process where you have to test many times to succeed and get high quality products. Every part is unique and you never know if the print will work out as intended. Depending on the size of the part different 3D printers need be used.
The material used in the printers is PLA plastic and is preferably used instead of ABS plastic which is toxic. For the tyres NinjaFlex TPE plastic is used to get the softness needed to imitate a real tyre.

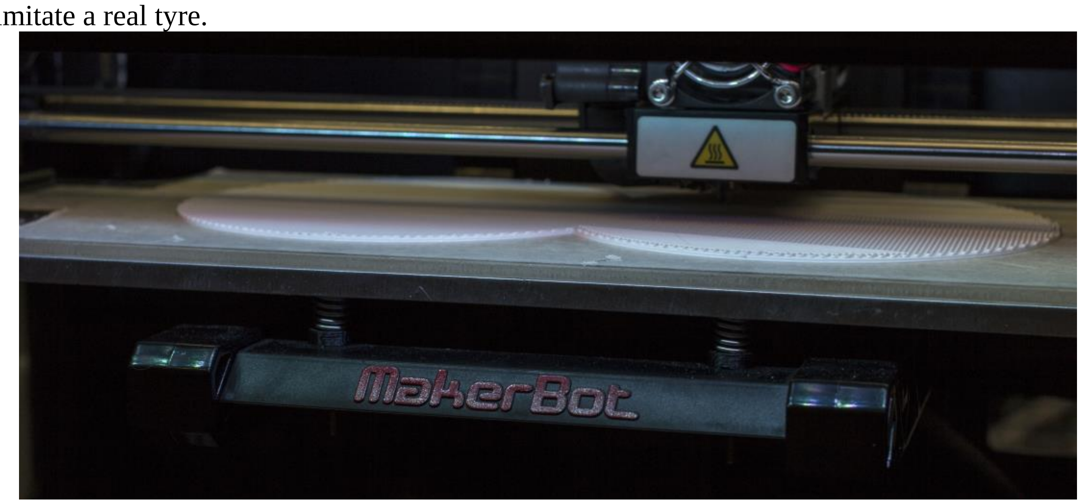


Figure 3-41 Makerbot replicator 2 during print

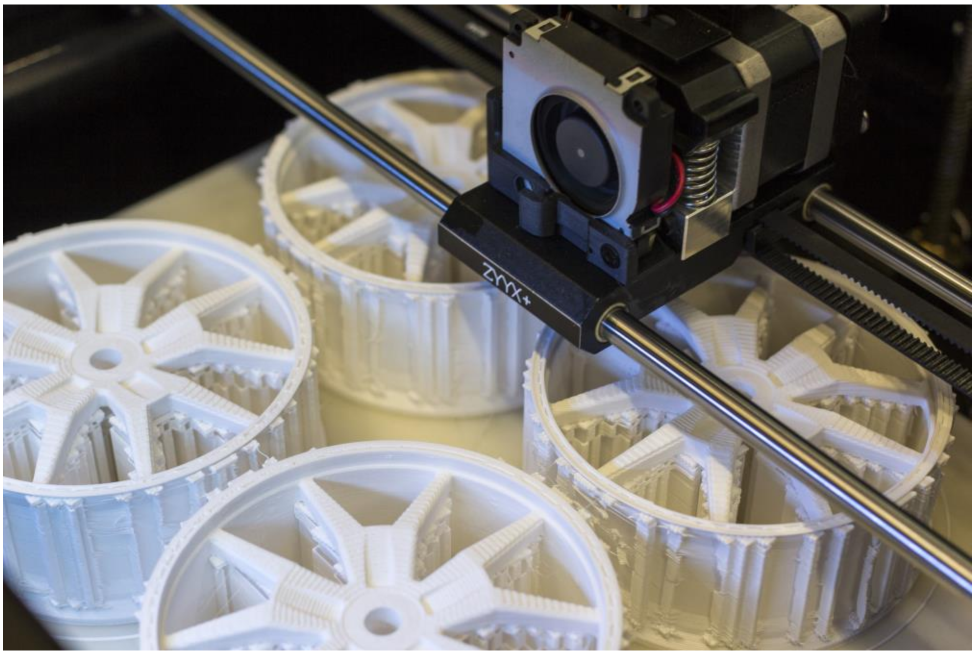


Figure 3-42 ZYYX+ during print

3.10.2 Machining Anti-roll Bar

From the anti-roll bars designed in Catia V5 drawings were printed and brought down to the workshop of Chalmers. They were based on a 2 mm silver steel rod that was bent to the desired shapes. The anti-roll bar consist of three parts: two rods and one cylinder with stop screws in it. The cylinder with the stop screws is used to link the rods together. When unscrewing one of the stop screws, they disconnect from each other and the anti-roll bar is out of function.

3.10.3 Machining Chassis

Based on Catia V5 files the chassis with its holes were water jetted from a two millimeter steel plate in the workshop of Chalmers. All sharp edges of the plate were rounded off to improve the mobility of the vehicle without causing any wounds. The side edges were bent 90 degrees to get easy access to the joystick and get good stability of the body structure.

3.10.4 Assembling/Post Processing

When the parts were manufactured they were assembled with bolts and screws. Choice of method is based on that some parts are supposed to be interchangeable and some not. As the plastic shrinks a few percent, around 3-5% depending of the size of the part and which printer that was used, most of the holes had to be drilled up to the needed dimension. This was especially a problem with mounting the subframes to the chassis as the precision of the waterjet is almost perfect, which the 3D printers are not.

4 Chapter Four – Results

In this chapter all results will be presented and all measured values can be found in Appendix B.

4.1 Static

Static photos of the entire models can be seen in figure 4-1 and 4-2.

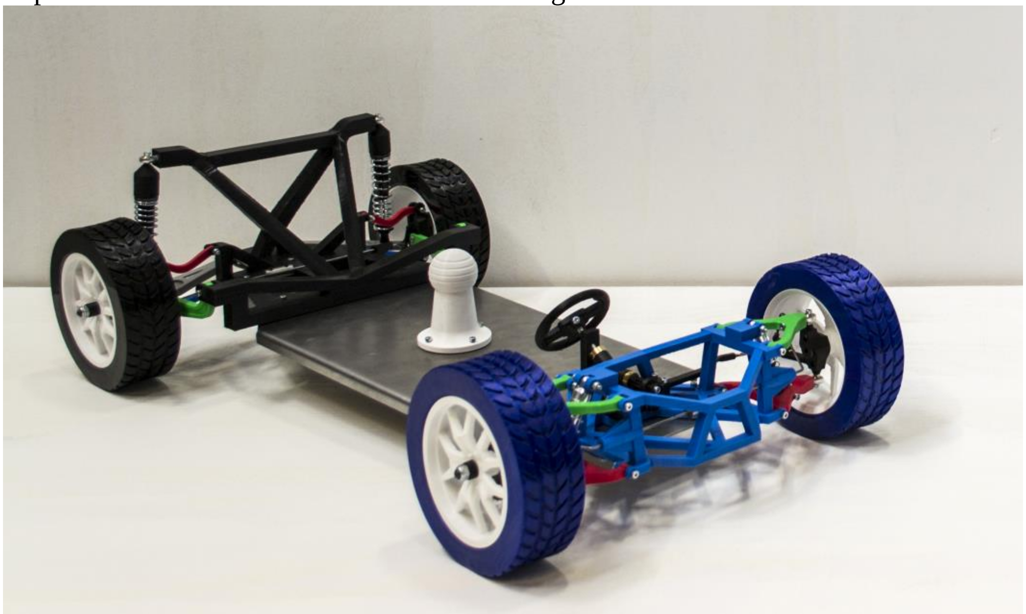


Figure 4-1 Chassis with trailing arm and double wishbone mounted

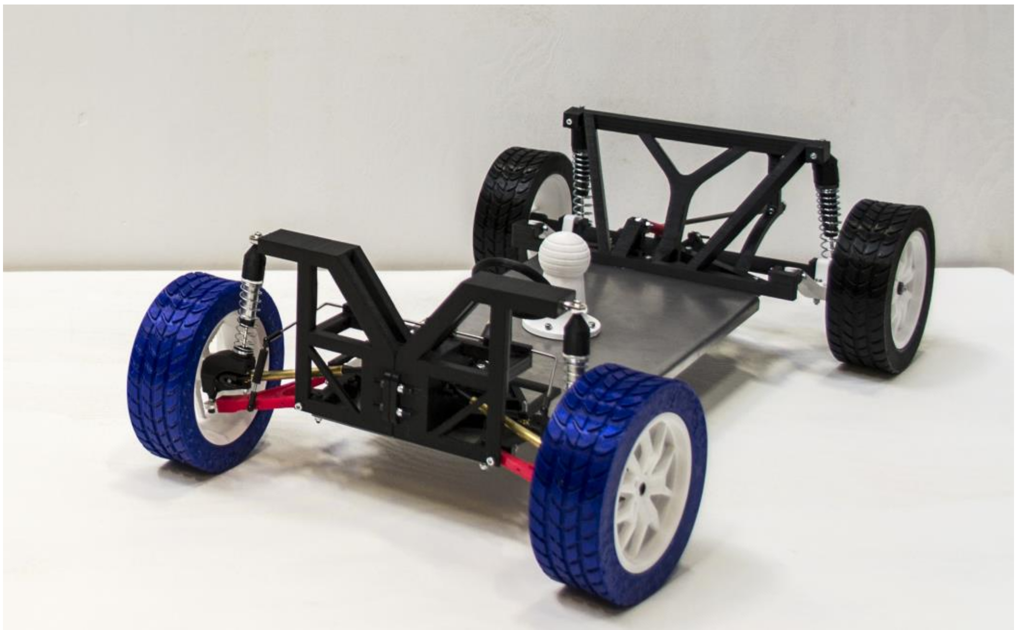


Figure 4-2 Chassis with live axle and MacPherson mounted

4.1.1 MacPherson

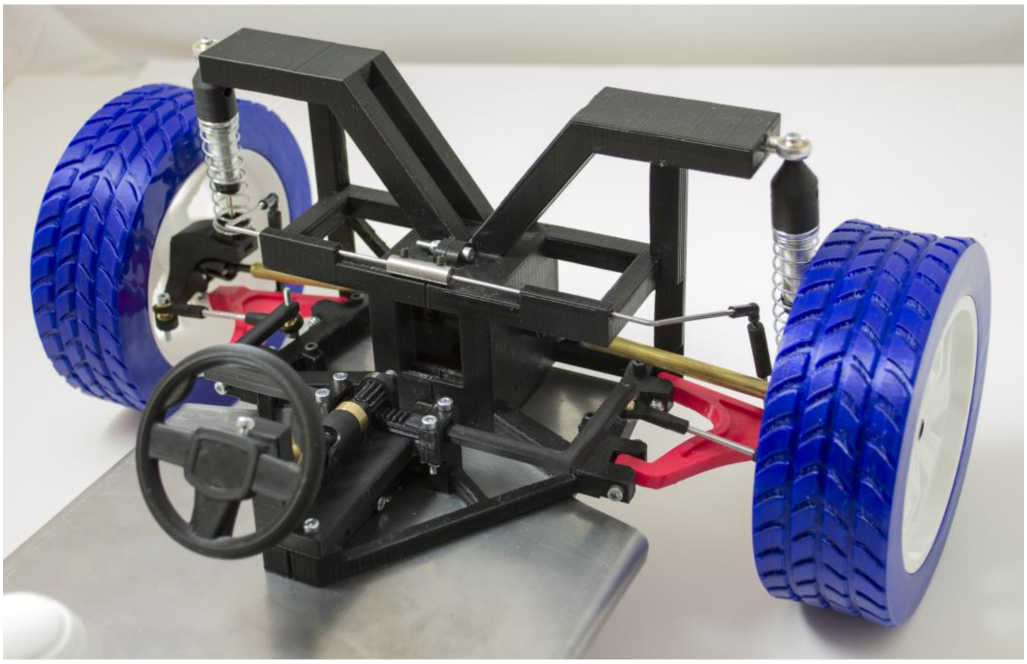


Figure 4-3 Picture of MacPherson suspension

The result of the MacPherson strut suspension model in scale 1:5 is shown in figure 4-3. A colour code with two different colours keeps the components of the suspension apart, which can be read in table 1. The control arm is coloured red and both the upright and subframe is coloured black and can be seen in figure 4-3. A black steering rack can be found in the rear centre of the subframe and enables steering on the suspension from the attached steering wheel. At the rear of the subframe a knob is found that enables propulsion on both front wheels. On the model there are two coilover dampers, one anti-roll bar and two 12 mm wheel bearings on each upright. To enable steering and travel two ball joints links each upright and steering rack together. The joint point for the anti-roll bar is located on the contral arm. One ball joint links the upright and control arm together to get the right movement. The upper part of the coilover dampers are mounted together with the subframe with a ball joint that enables the sought after movement.

Table 1 Colour and corresponding part - MacPherson

Colour	Part
Black	Upright
Red	Lower control arm
Black	Subframe

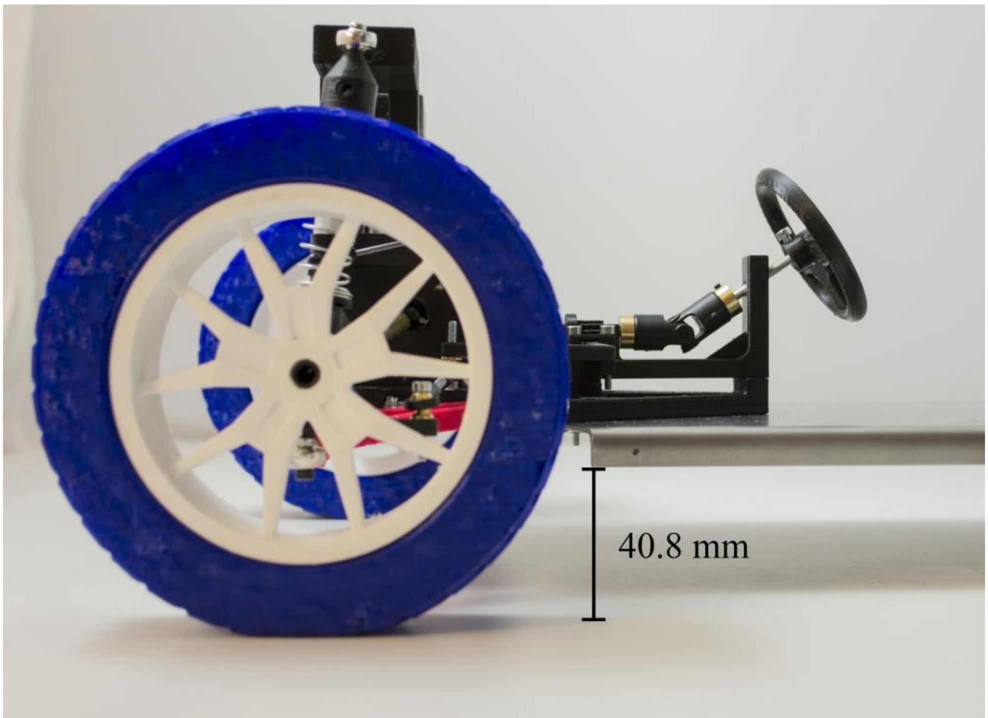


Figure 4-4 Ride height measurement

In figure 4-4 the ride height is measured to 40.8 mm and in figure 4-5 the track width can be seen to be 318 mm. The camber can be seen in figure 4-6 in static kerb +2 position of the model and is -0.5 degrees.

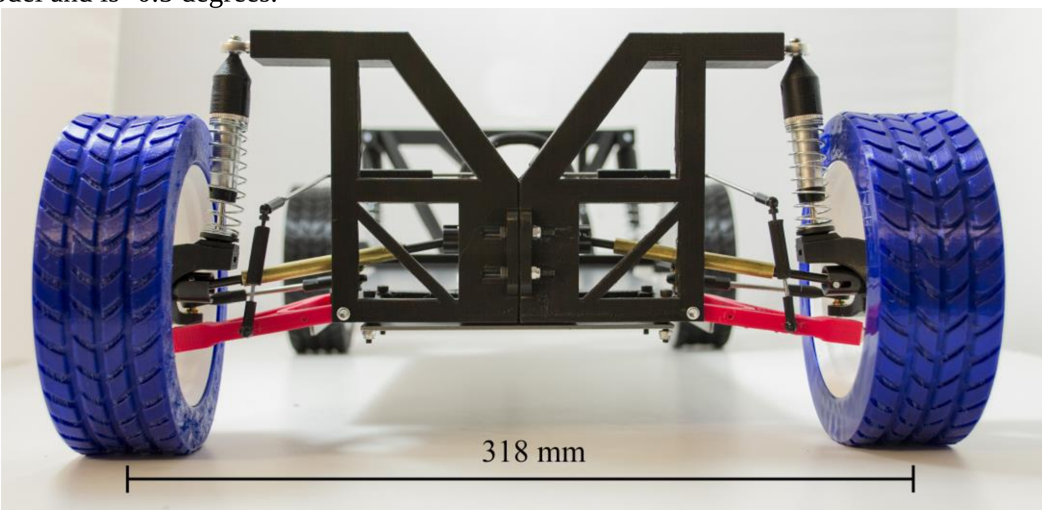


Figure 4-5 Measurement of track width

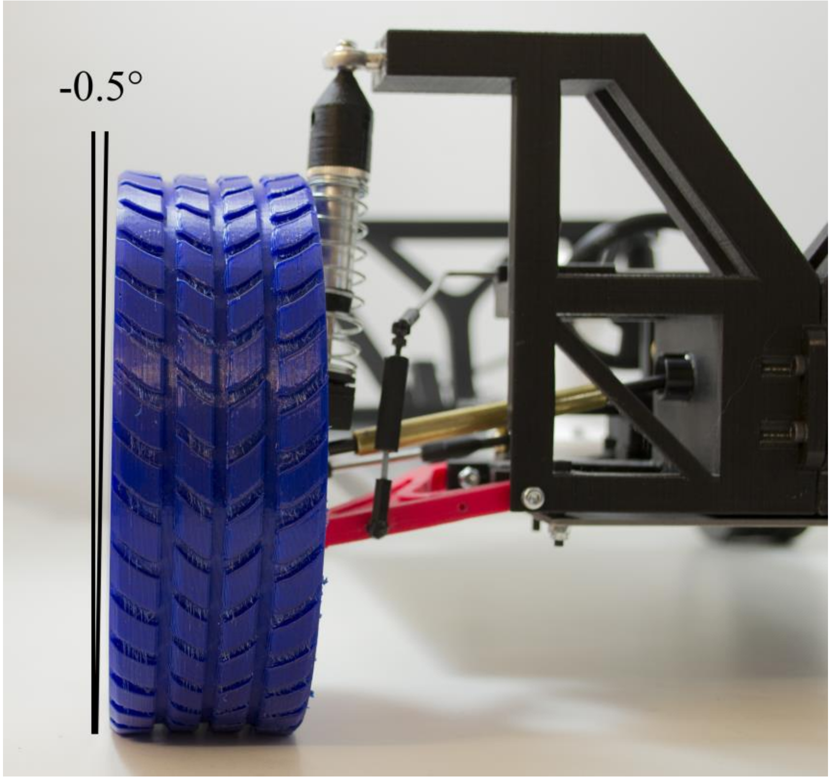


Figure 4-6 Measurement of static camber

4.1.2 Double Wishbone

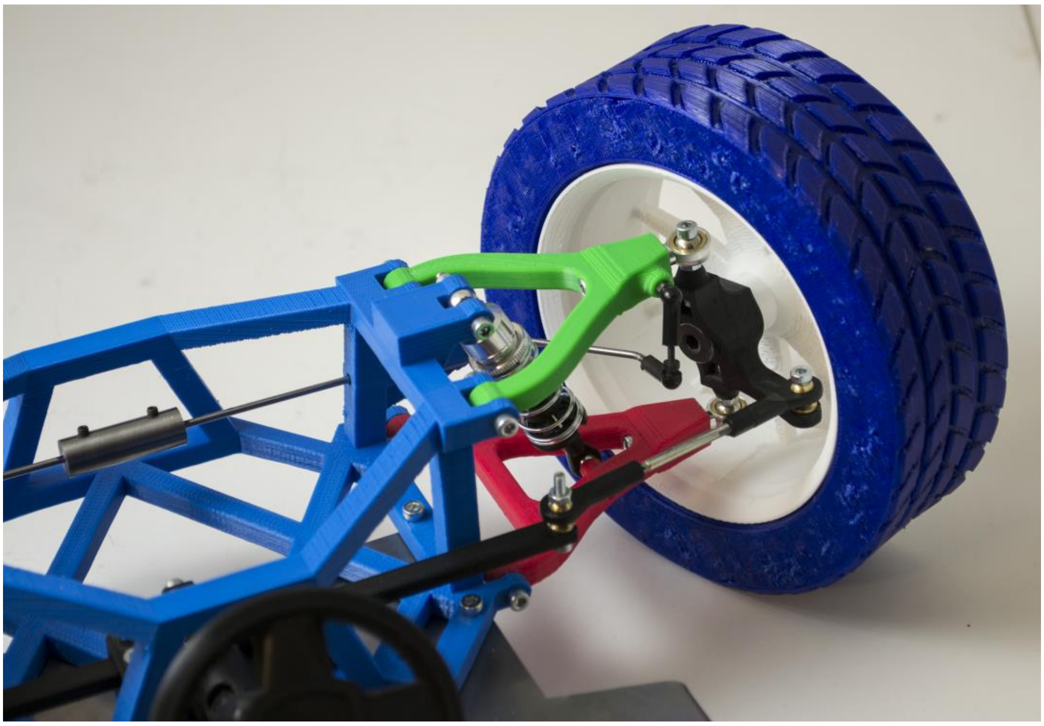


Figure 4-7 Picture of double wishbone suspension

The double wishbone suspension can be seen in figure 4-7. It consists of four colour coded parts, two coilover damper and a detachable anti-roll bar. The colour coded parts and their respective colour can be seen in table 2. The upright is black and it carries two 12 mm wheel bearings that are separated in the y-direction and gives stability to the wheel axis. The fixation of the upright is handled by two ball joints that enables both travel and steering. The ball joints are placed along the kingpin axis with a spread of 57.5 mm. Two wishbones are connected to the upright via the ball joints. The upper wishbone is green, it holds the connection to the anti-roll bar, and measures 68 mm in the y-direction between its hardpoints. The lower wishbone which is red, holds the connection to the coilover damper and measures 80 mm in the y-direction between its hardpoints. Both wishbones are symmetric around the wheel axis and has two attachment points in the subframe to handle forces in the x-direction. The fourth part is the blue subframe that holds attachments for both wishbones, the coilover damper and steering rack. The fifth and last main part is the black steering rack, it is fitted behind the wheel centre in the subframe and is linked to the uprights via ball joint mounted tie rods.

Table 2 Colour and corresponding part – Double wishbone

Colour	Part
Black	Upright
Red	Lower wishbone
Green	Upper wishbone
Subframe	Blue

The following static results were measured on the scale model in kerb +2:

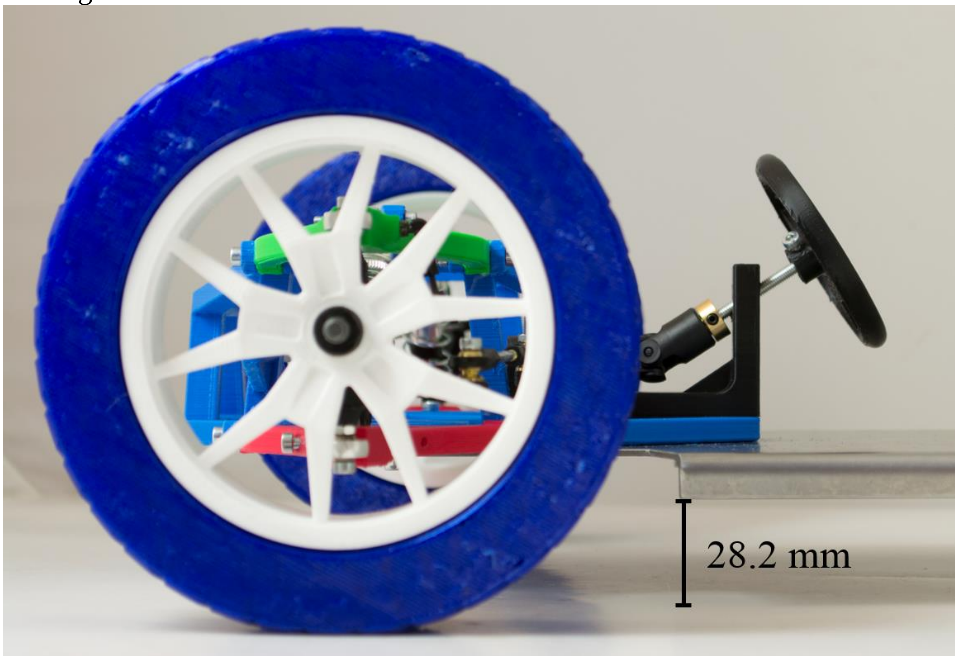


Figure 4-8 Ride height on double wishbone suspension

The ride height in the design position is measured to 28.2 mm from the lowest part of the chassis and can be seen in figure 4-8.

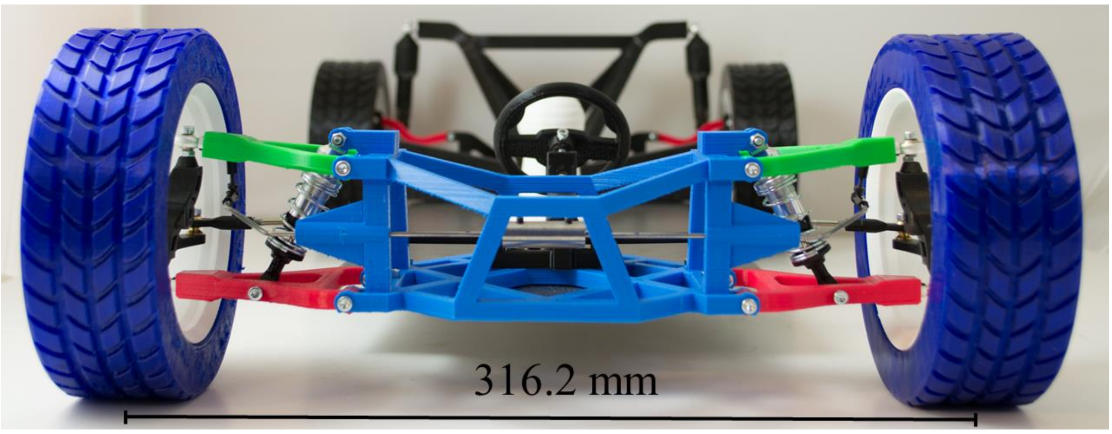


Figure 4-9 Track width on double wishbone suspension

The track width is 316.2 mm as seen in figure 4-9.

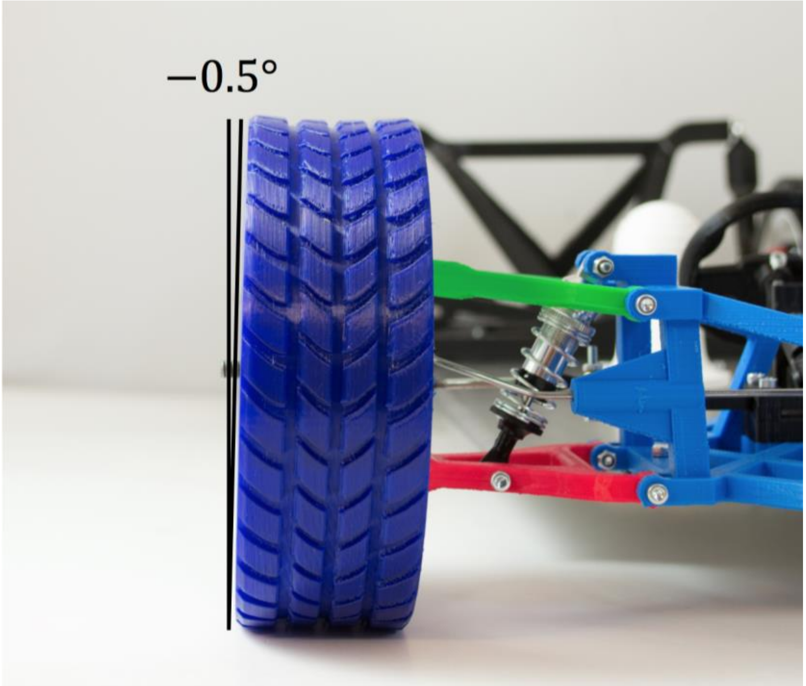


Figure 4-10 Camber angle on double wishbone suspension

As shown in figure 4-10 the camber angle in the design position is -0.5 degrees.

Final static kerb +2 dimensions of the double wishbone design are listed below:

- Ride height: 28.5 mm
- Track: 317.7 mm
- Camber: -0.5 degrees
- Caster: 0 degrees
- Toe: 0 degrees
- Roll centre: 12 mm
- Ground offset: 2 mm

4.1.3 Trailing Arm

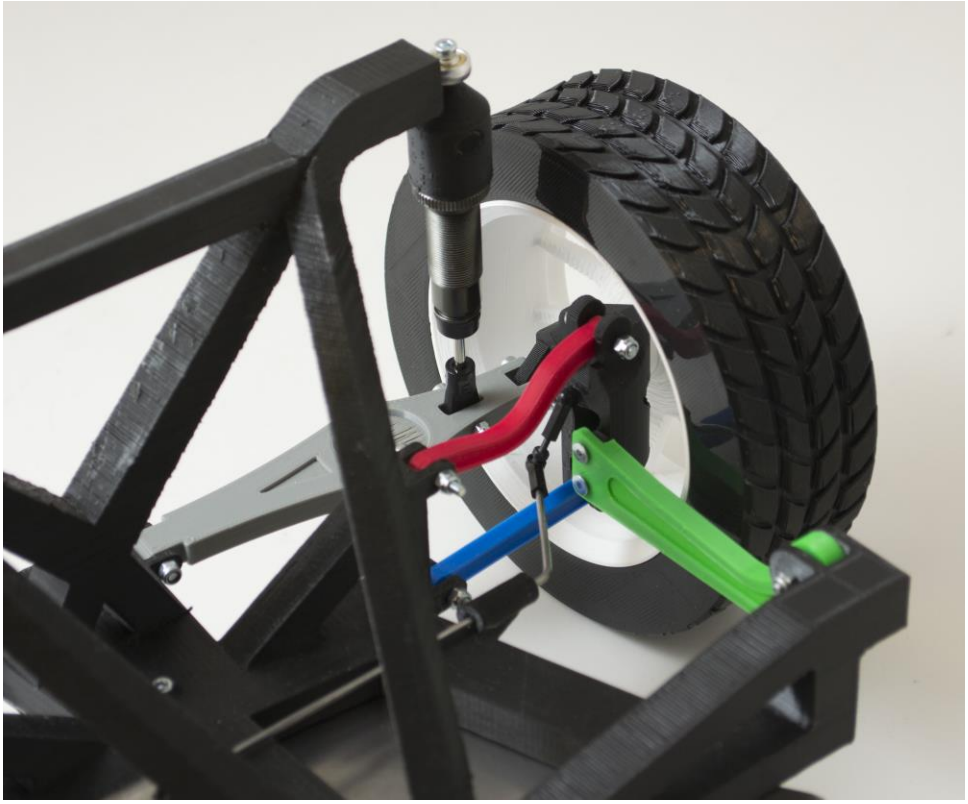


Figure 4-11 Picture of trailing arm

The resulting 1:5 scale model of the trailing arm suspension is shown in figure 4-11. There are six different colour coded parts and they are described as shown in table 3.

Table 3 Colour and correspnding part – Trailing arm

Colour	Part
Black	Upright
Grey	Spring link
Blue	Toe link
Red	Camber link
Green	Tie blade
Black	Subframe

The model also has two coilover dampers, two 12 mm wheel bearings per upright and an anti-roll bar. The top part of the coilover is attached to the subframe by a ball joint to allow for movement in the desired directions and bottom is mounted by a screw. As all the other links only needs rotation they are attached by screws.

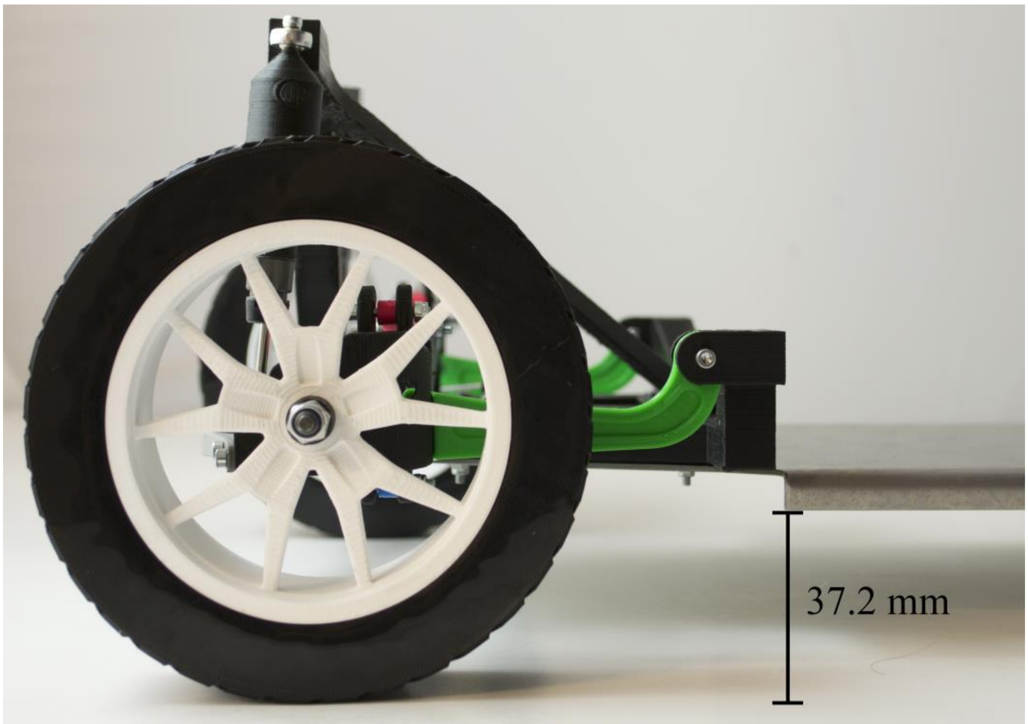


Figure 4-12 Ride height on trailing arm suspension

The resulting ride height in design position is measured to 37.2 mm from the lowest part of the chassis at the rear to the ground, figure 4-12.

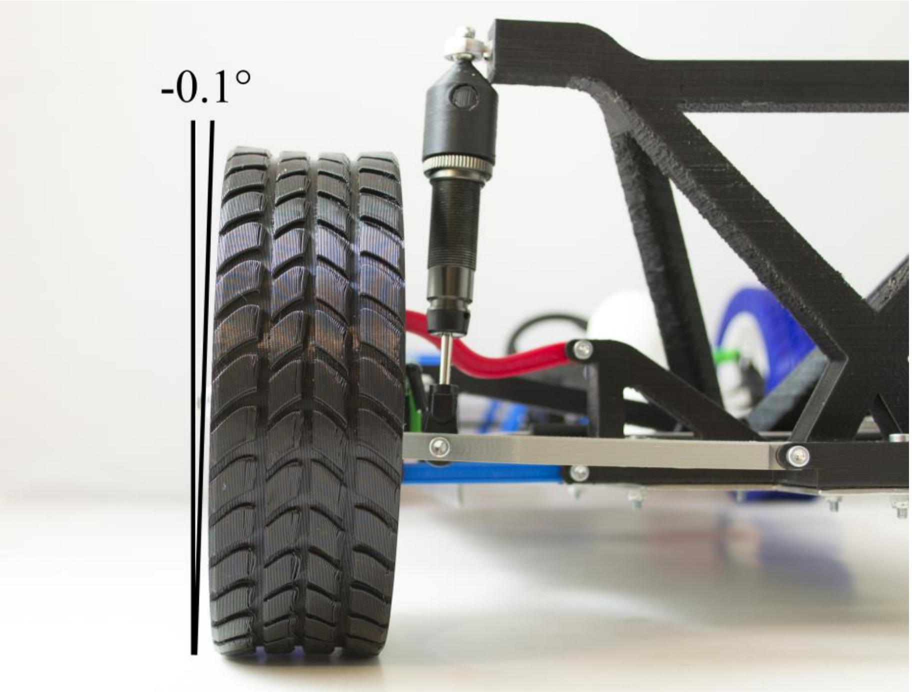


Figure 4-13 Camber angle in design position on trailing arm suspension

The camber angle is -0.1 degrees for the wheel in the design position nad can be seen in figure 4-13.

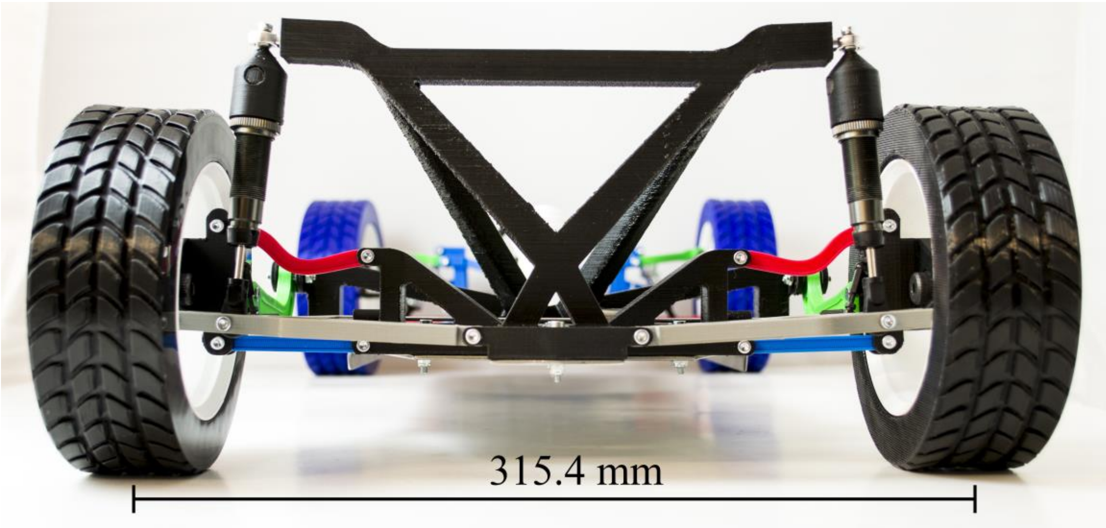


Figure 4-14 Track width on trailing arm suspension

The track width is measured to 315.4 mm

4.1.4 Live Axle

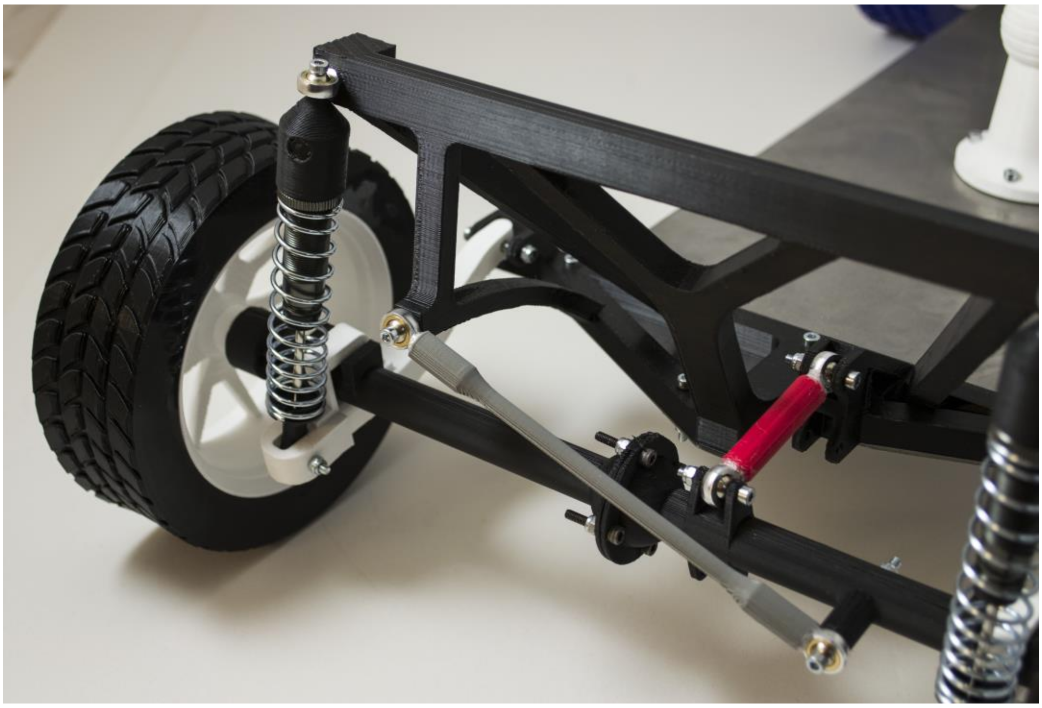


Figure 4-15 Picture of live axle suspension

In figure 4-15 the live axle suspension model in scale 1:5 is shown. The 3D printed colour coded suspension parts and corresponding colours can be found in table 4. Except for those parts it also consists of two coilover dampers with 3D printed extensions. To keep the wheels in place two bearing separated in the y-direction by a spacer is placed on each end of the axle.

Table 4 Colour and corresponding part – Live axle

Colour	Part
Black	Axle
White	Support arms + complementary piece
Grey	Panhard rod
Red	Stabiliser bars
Black	Subframe

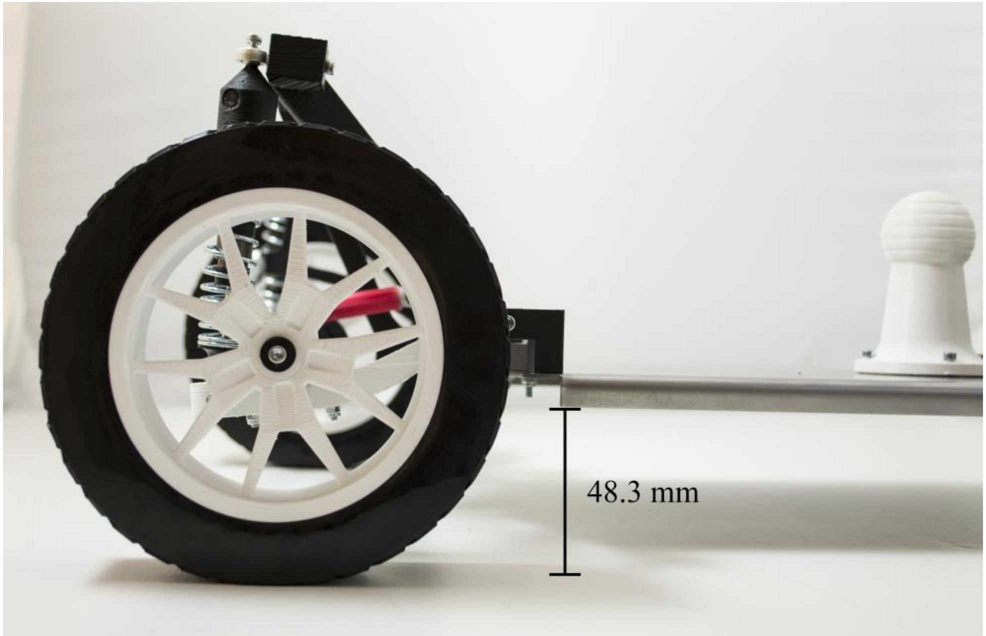


Figure 4-16 Ride height on live axle

The ride height, which can be seen in figure 4-16, is in the design position 48.3 mm measured from the lowest point on the chassis to the ground where the wheels touch.

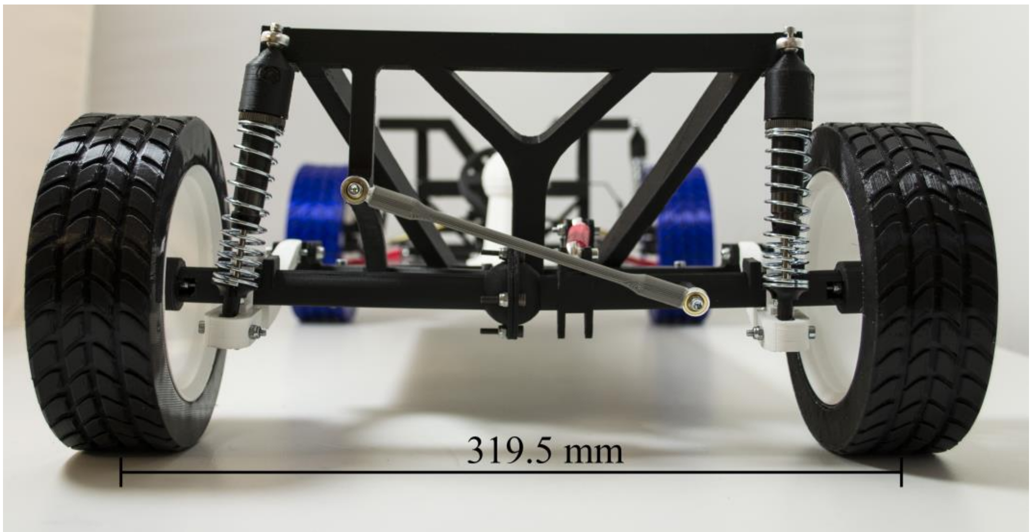


Figure 4-17 Track width of live axle

The track width is measured to 319.5 mm and can be seen in figure 4-17.

4.2 Kinematics

All graphs in this chapter shows values for a full scale model as these values are more familiar and useful in comparison.

4.2.1 MacPherson

The first graph, figure 4-18 shows the camber gain when the suspension travels from 80 mm jounce to 110 mm rebound, which can be seen on the x-axis. On the y-axis the degrees of camber is shown and what can be noticed here is that the negative camber gain reaches its minimum at -1.2 degrees and then slowly increase to a final -1.0 degrees at 80 mm. In the rebound motion of the wheel the camber reaches a value of 1.3 degrees

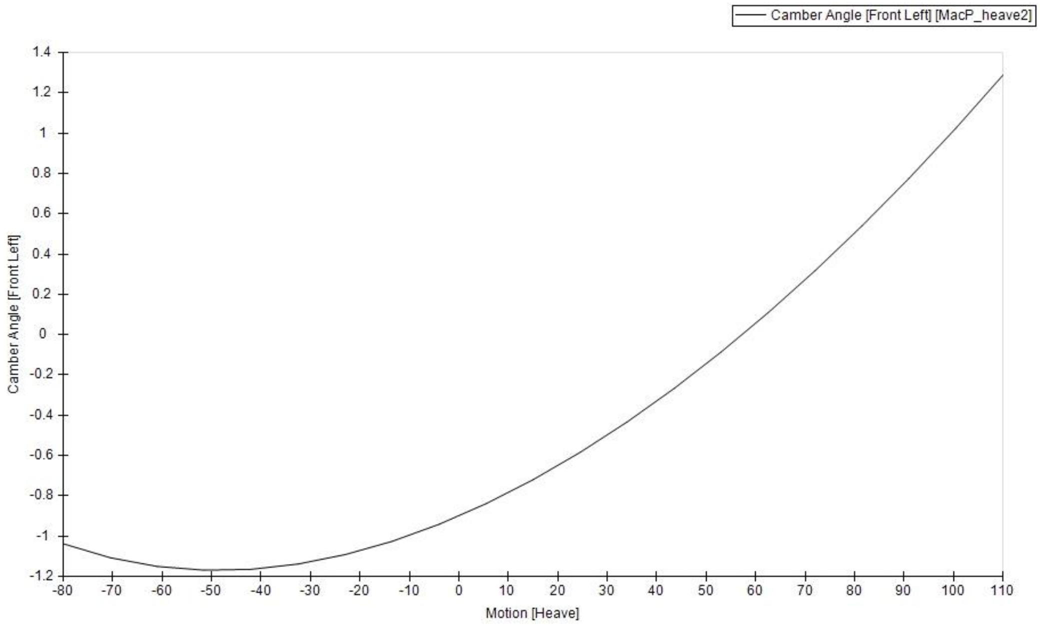


Figure 4-18 Camber gain, y-axis shows degrees and the x-axis shows travel in mm

Regarding the bump steer for MacPherson, seen in figure 4-19, the same x-axis as in figure 4-18 is used with the degrees of steering on the y-axis. As the wheels move up to its peak of 80 mm the steer angle is -3.3 degrees compared to the when the wheel moves down to 110 mm and reaches a value of 0.5 degrees.

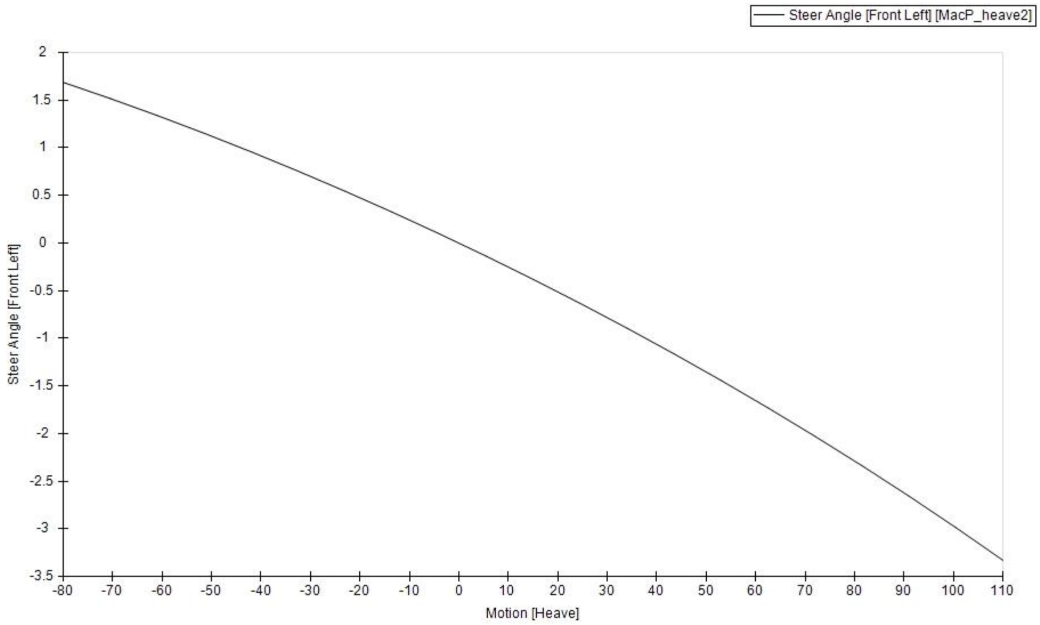


Figure 4-19 Graph of bump steer

The following kinematic results were measured on the scale model:

- The wheel travel is 11.8 mm rebound and 8.6 mm jounce with a total wheel travel of 20.4 mm.
- The camber gain is 1.3 degrees at full rebound and 0 degrees at full jounce -0.9.
- The maximum steering angle is 36.3 degrees for the inner wheel and 25.7 degrees for the outer wheel. Ackerman percentage is 28.3 % at full steering angle.

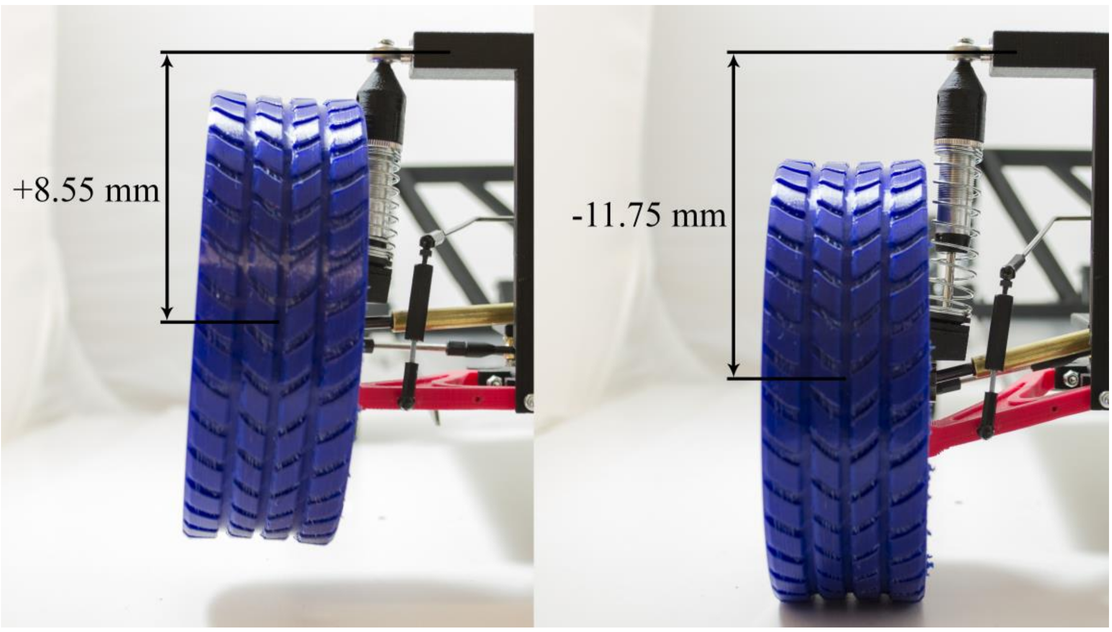


Figure 4-20 Wheel travel MacPherson

In figure 4-20 the travel of the wheel can be seen during full jounce to be 8.6 mm and during full rebound to be 11.8 mm.

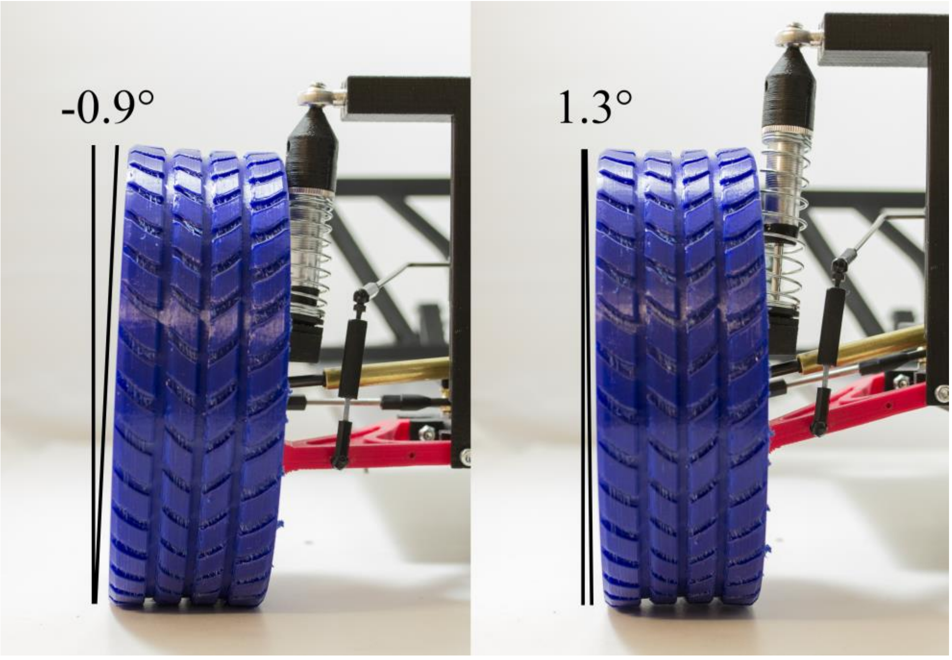


Figure 4-21 Camber angle in full jounce and full rebound

The camber angle in full jounce is -0.9 degrees and full rebound is 1.3 degrees, which can be seen in figure 4-21.

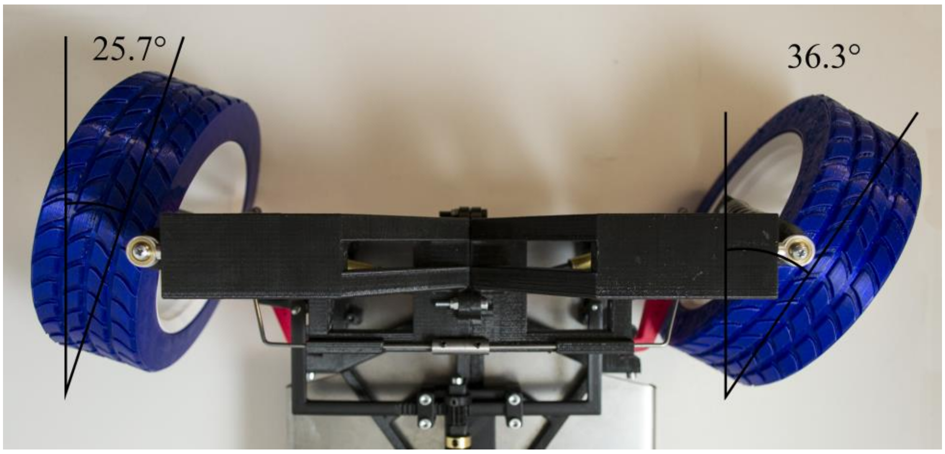


Figure 4-22 Showing of steering angle on outer and inner wheel

The steering angle on the inner wheel is 36.3 degrees and 25.7 degrees on the outer wheel when having full steer angle, which can be seen in figure 4-22.

4.2.2 Double Wishbone

Camber gain can be seen in figure 4-23. The x-axis shows the motion of the chassis and the y-axis the camber angle. When raising the chassis 110 mm the wheel travels -110 mm and the camber angle changes to 1.4 degrees. Positive wheel travel of 80 mm results in -2.4 degrees camber. The camber change is not symmetric around the design position, positive wheel travel results in greater camber change than negative wheel travel.

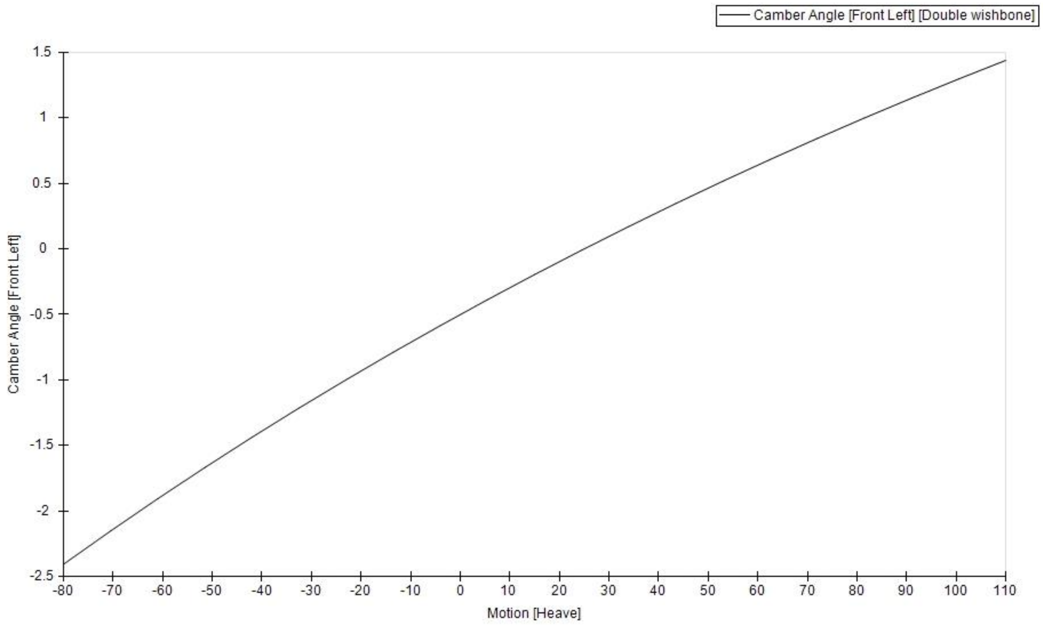


Figure 4-23 Camber

Bump steer can be seen in figure 4-24. The x-axis shows the motion of the chassis and the y-axis the steering angle change. When raising the chassis 110 mm the wheel travels -110 mm and the change of steering angle becomes 1.8 degrees. Positive wheel travel of 80 mm results in -1.2 degrees of changed steering angle. Negative steer angle change equals toe in.

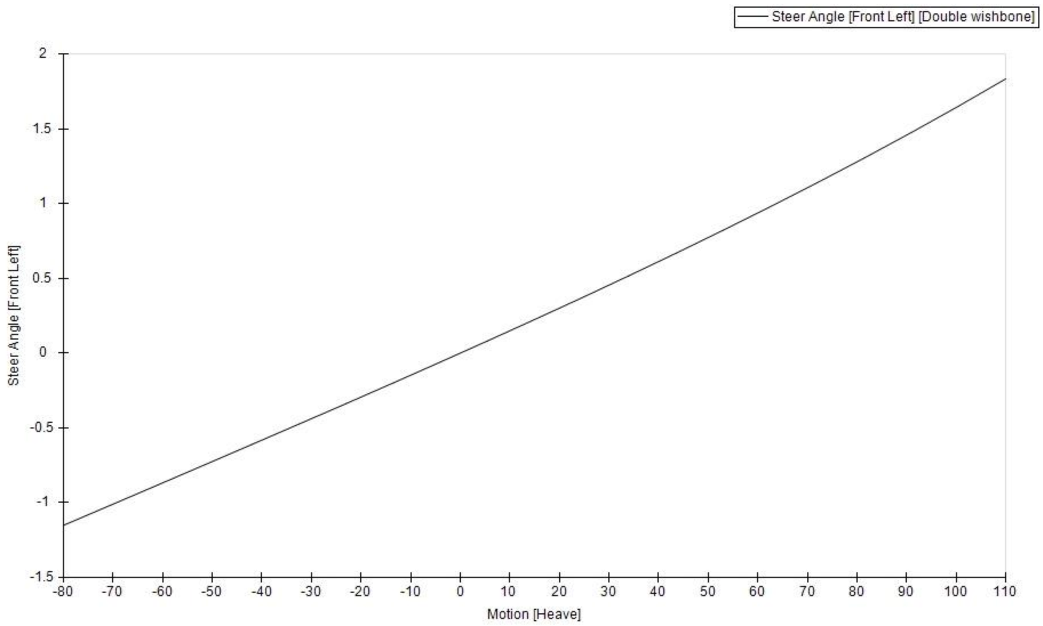


Figure 4-24 Steering angle

The following kinematic results were measured on the scale model:

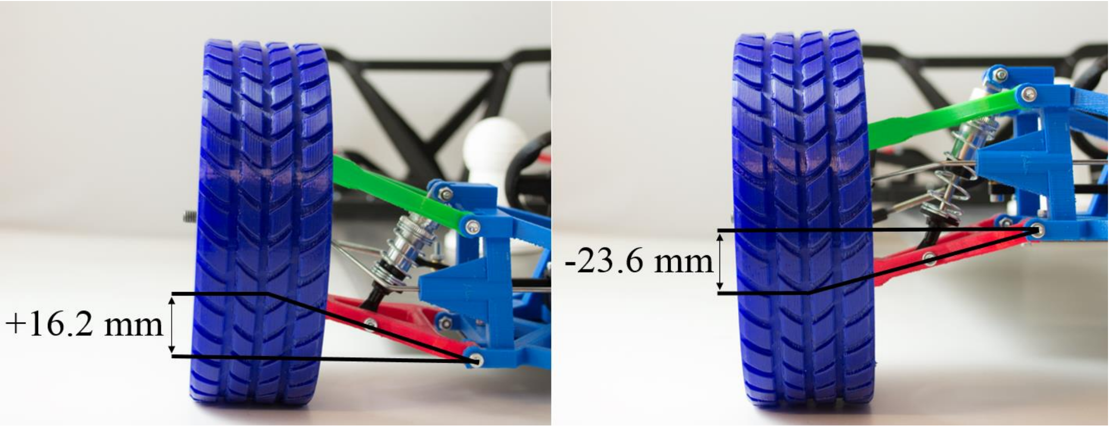


Figure 4-25 Wheel travel on double wishbone suspension in full jounce and rebound

Figure 4-25 shows the travel of the wheel. Full jounce is 16.2 mm and full rebound is 23.6 mm. Total travel measured from full jounce to full rebound is 39.8 mm.

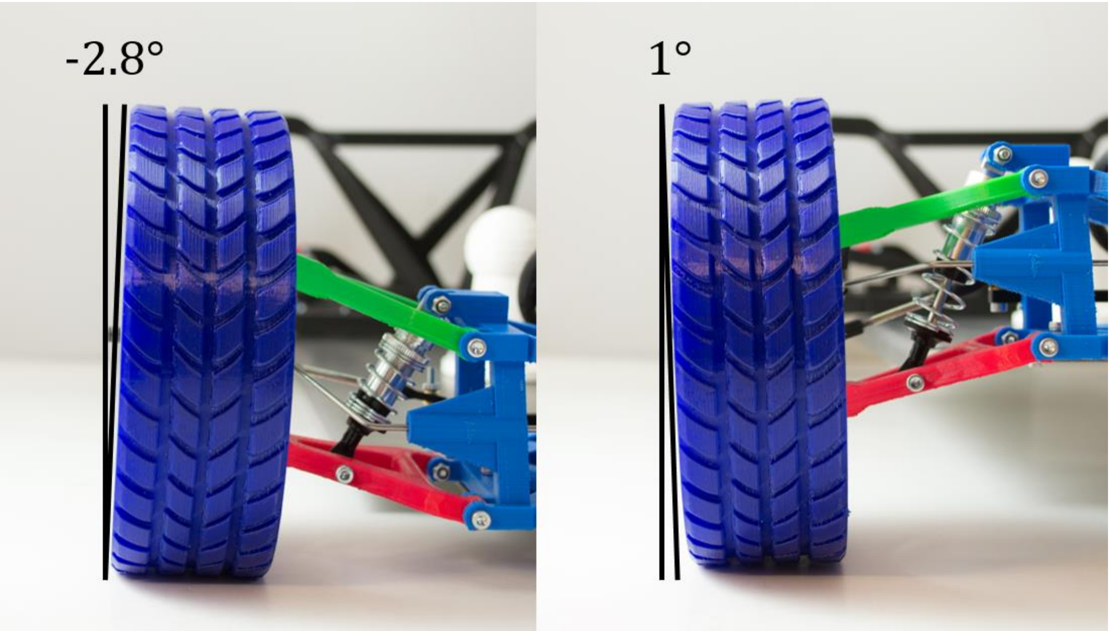


Figure 4-26 Camber angle on double wishbone suspension in full jounce and rebound

The camber angle in full jounce is -2.8 degrees and full full rebound is 1.0 degrees which can be seen in figure 4-26.

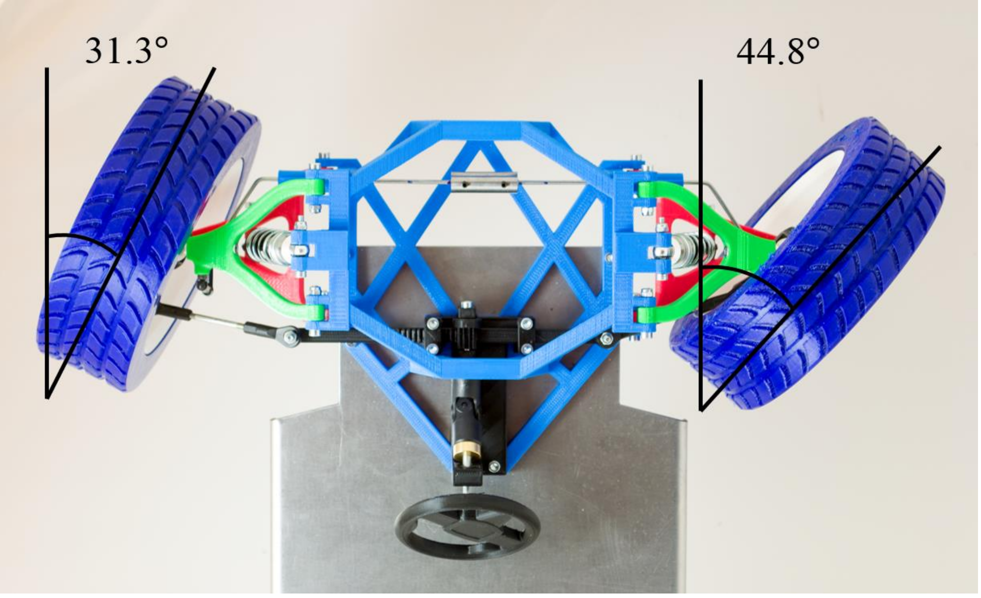


Figure 4-27 Steering angle on double wishbone suspension

The steering angle on the inner wheel is 44.8 degrees and 31.3 degrees on the outer wheel when the steering is at full lock. The angles can be seen in figure 4-27. Ackerman percentage is 31.1% at full steering angle.

4.2.3 Trailing Arm

As there are no softwares available to calculate the exact curves of the kinematics it is instead measured by hand and evaluated by its ability to visualize the correct movement.

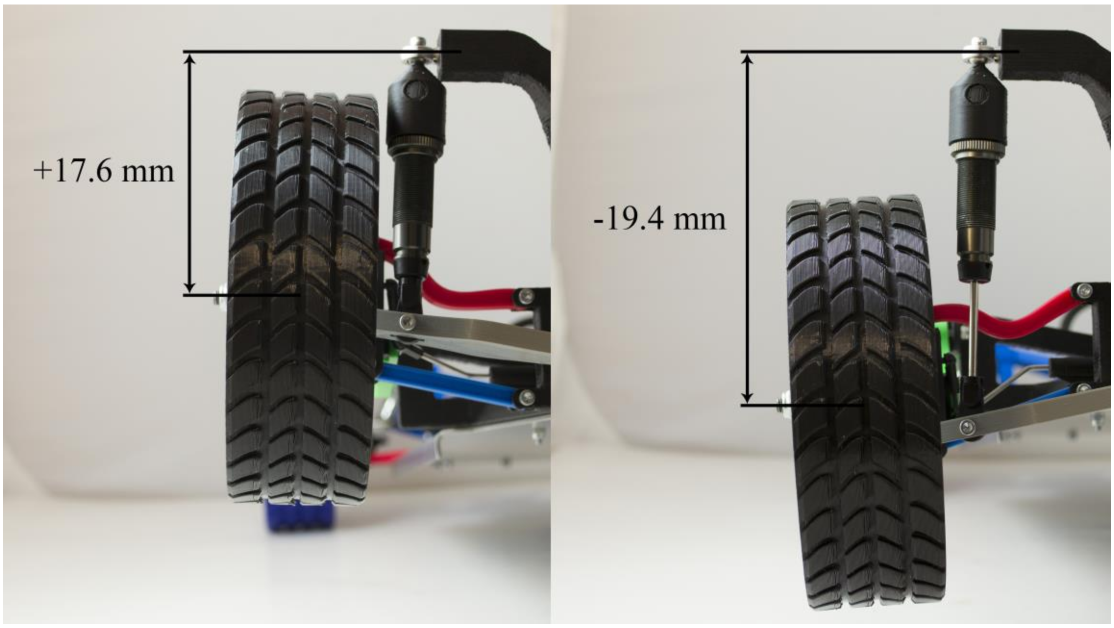


Figure 4-28 Wheel travel shown in full jounce and full rebound

The suspension has a wheel travel of 19.4 mm at full rebound and 17.6 mm at full jounce which gives a total of 37 mm wheel travel, to be seen in figure 4-28.

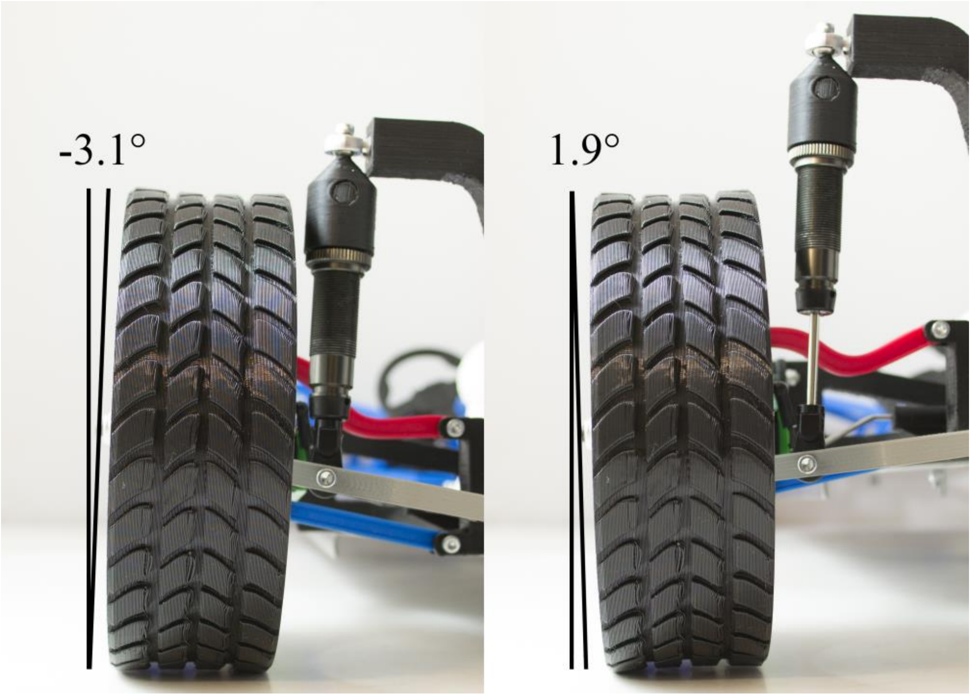


Figure 4-29 Camber in full jounce and full rebound

Figure 4-29 shows the suspension under full jounce and full rebound with the load being placed between the wheels. The camber is measured to -3.1 degrees at full jounce and 1.9 degrees at full rebound. With a total difference in angle of 5 degrees and a total travel of 37 mm the camber gain per meter in full scale is calculated to 27 degrees/meter.

4.2.4 Live Axle

Similar to the trailing arm suspension there are no softwares available for the curves so the kinematics are illustrated with pictures instead. Figure 4-30 show full jounce and rebound movement. The wheel travel is 15.9 mm rebound and 11.5 mm jounce with a total wheel travel of 27.4 mm. The camber and toe does not change during wheel travel which can also be seen in figure 4-30.

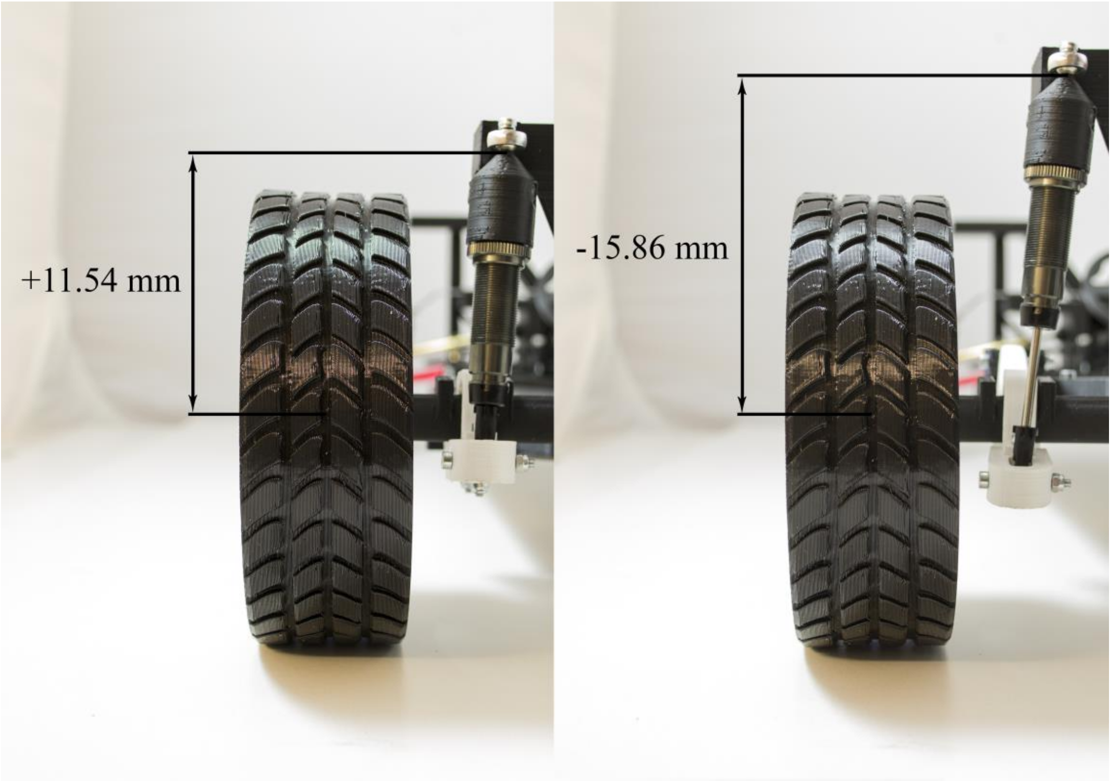


Figure 4-30 Full jounce and full rebound movement

5 Conclusions and Discussion

5.1 Conclusions and Fulfilment of Deliverables

All of the following objectives of this thesis have been met:

- A front MacPherson suspension model in scale 1:5 with differential and steering
- A front Double Wishbone suspension model in scale 1:5 with steering
- A rear Trailing arm suspension model in scale 1:5
- A rear Live axle suspension model in scale 1:5 without anti-roll bar
- Two vehicle chassis with interchangeable front suspensions

The following conclusions apply on all models. The spring and anti-roll bar stiffness is not properly scaled. Due to the lack of bushings the models have some play in the joints reducing the accuracy of the movements. This results in inaccurate and varying values for camber, toe etc. The stiffness of all parts is as expected not representative. Despite this the models have good illustrative capability and are therefore well suited as educational material. It has been very good for the understanding of suspension systems to work with the selected variants of suspensions. Due to the volume of work with the variants the time has not allowed for solving all details.

5.2 Future Work

Several areas can be improved to get a more realistic and correct model. Improvements such as adding bushings to links and control arms to get a more realistic movement and feeling should be done. This would reduce play in subframes and uprights. If bushings could be added realistically a better understanding of how they work could be achieved since bushings are moving and flexing. Enabling 3D printing in rubber will make it possible to implement bump stops which are a crucial part of the suspension that is often hard to understand. Redesigning the upper control arm on the double wishbone suspension so it can be moved sideways to see how caster, camber and toe is affected. It can not be done on most normal cars but it is preferable on the model to show how small movements of the control arm can make major effect on the setting of a car. Another aspect is to further develop the spring stiffness to better suit the scaled models.

By adding radio controlled propulsion and brakes it would be easier to simulate and show how anti-squat/dive works and see the difference between shaft torque and torque from the brakes and how it affects the vehicle. If the vehicle is radio controlled it will also be possible to see how the anti-roll bar affects the roll, a natural movement of the suspension which can be seen when going over rough terrain or obstacles. To get a better understanding of the importance and the different characteristics of the suspensions a rig could be made to easily demonstrate it. With the rig a systematic series of movement could be implemented and the movement of the different suspensions can easily be seen. In the existing model the anti-roll bar is too weak relative to the springs as it can not cope with lifting the opposite wheel and needs to be modified and calculated to give the right stiffness. To improve even more a change of material to spring steel would be preferable to get the right characteristics and function of the anti-roll bar.

The tyres should be developed further to get a more realistic stiffness and better grip if the models are to be radio controlled to enable the vehicle to roll, dive and squat correctly when moving. The existing tyres have a honeycomb structure inside that gives the load carrying capacity instead of air like a real tyre. A future development of the tyre stiffness can be made to get a more realistic behavior when the tyre is loaded. If the tyres instead are to be air filled a redesign of the rims is necessary to handle the pressure that will occur. Redesign of the rim would maybe be necessary if the vehicle would be radio controlled as large forces would occur when jumping, going over bumps and similar. Existing rims are designed for a static model where no large forces are applied. They may work for larger loads but field tests should be carried out to see if they manage the forces. The resolution and infill of the 3D printed rim is critical since it affects if the rim can stand large or nearly zero loads before breaking. Adding steering limiters on the steering rack will make sure that the rim does not hit the wishbone or the steering link hits the rim. This is needed to get the exact steering angle as specified by the given data.

Creating graphs for trailing arm and live axle on its movement is advantageous to be able to compare reality to the 1:5 model. By adding more graphs with different setups more conclusions of how the vehicle acts can be made. For example different camber or caster and their impact on the grip could be shown. This to link data to reality together and understand what the parameters do.

To get an even better feeling of how the car and the suspension moves a removable body could be added. This would show how the wheel moves up and down in the wheel well and give a perspective of how much it moves. A body would also improve the visibility of roll if the vehicle is moving. Another positive aspect is that it would better visualise how big the wheel envelope, meaning the volume created by tracing the movement of the wheel as it steers and travels, has to be to not get any interference.

5.3 Discussion

5.3.1 Plastic Shrinkage

Each part that is 3D printed has a shrinkage of 3-5% depending on which 3D printer that has been used. This could be critical for some parts and for some parts not. After some test prints knowledge was gained on how much components shrinks in each 3D printer and can be used for future components. Knowledge in how holes should be dimensioned to fit different standard screw sizes and similar. This to avoid drilling holes to make the screw fit. The crucial part to avoid is to go through the solid layers and get to the hollowness. If this is done a new part needs to be printed because the structural integrity of the hole will be lost. Even though the knowledge of shrinkage the printers varied a lot and some parts still shrunk too much.

5.3.2 Coilover Damper Mount Hardpoint

The suspensions have designs similar to real passenger cars, where especially the links and uprights look like they would on a full scale model. As the subframes are designed on freehand, they do not correspond to what the real subframes look like. One of the reasons, which is mutual for all suspensions, is that the hardpoint where the coilover damper or strut attaches to the subframe instead should be mounted to the body structure of the car. For obvious reasons that can not be done as there was not supposed to be any body structures manufactured. Regardless of the looks of the subframe, all hardpoints are downscaled and positioned in the correct place which gives the same characteristic as the Chalmers suspension.

5.3.3 Stiffness

In general it is hard to dimension the stiffness in the subframes, chassis, coilovers, struts and anti-roll bars when applied on downscaled models. Furthermore it was not in the scope of the project. When using off-the-shelf components the lack of detailed information makes choosing the correct components difficult. The stiffness of the spring on RC components is usually not specified more than as “soft”, “medium” or “hard” and therefore it is a gamble when making the choice. When off-the-shelf components were to be ordered no information about the chassis weight was known, so only approximations were made.

It is hard to scale down the forces that occur on separate parts on a suspension in full scale. This is because of the changes to other materials on the model. In this case the suspension components are made of plastic which is much weaker than the steel normally used on the full scale car. It is plausible to dimension bushings, springs and anti-roll bars through testing and calculations. Each 3D printed component has different stiffness depending on the infill and the resolution used for each part. When force is applied to get movement of the wheel the 3D printed components flex. To get every component to work perfectly and with the right stiffness is a time consuming challenge. For manufacturing bushings no suitable material has been found to get the right characteristics and stiffness. It is also a complex product to print, especially when the bushings are so small on the downscaled model.

There is also an issue when a calculated length that the coilover dampers need to be can not be matched because of limited types of RC car dampers available. The designed extenders are an unwanted part of the suspensions, but also the most fitting solution to the two front suspensions. The mounted components to MacPherson and double wishbone, have similar stiffness but are of different lengths. These are much softer than those mounted on both of the rear suspensions, trailing arm and live axle, which is an issue since that is not representative of a setup on a real car. To get them to match the front better, the springs had to be cut down and stretched to decrease the stiffness. Still there is a variation between the front and the rear making it uneven and a future solution would be to order or manufacture springs of the appropriate stiffness.

5.3.4 Anti-roll Bar

A silver steel, also known as high carbon steel, rod with a diameter of 2 mm is used for the anti-roll bars on the model. As found when testing the models, it would be preferable to use a larger diameter or treated silver steel rod so the anti-roll bar is stiffer relative to the springs to get a better representation of the function. As of now the movement of one wheel during bump gives almost no movement of the other wheel, which it should. No calculations were made on the stiffness and the material that was available at Chalmers is what was used to manufacture the anti-roll bars. If the stiffness of the original anti-roll bar was known it would still be hard to dimension it for the model because of the use of a different material as mentioned before. The function can still be shown but not as good as it is on real cars. However, for educational purpose the anti-roll bar still shows enough to create a perception of what the purpose is.

5.3.5 Play Between the Parts

As pointed out earlier the play in the attachments is necessary because the bushing function is not in place and would not move without it. This comes with the disadvantage of a loss of accuracy in the kinematics as the play gives no resistance or elasticity whereas the bushings do. One example is that the double wishbone suspension changes its wheel settings just depending on how it stands. In this project where there are many other things that affect the result, such as the plastic warping and shrinking in the printer, it is considered being a suitable simplification. The models still give a representative image of the full size suspensions.

Another downside of this is that the toe angle is very hard to secure with the loss of accuracy. Because of the very small angle of toe for those suspensions with toe, e.g. MacPherson should have 0.03 degrees of toe and with just a minor amount of play it quickly because hard to measure a correct angle. Since the toe is hard to measure this also affects the possibility of measuring bump steer. When measuring the bump steer the values cannot be secured to be accurate because the steering rack bent when friction force was created between ground and tyre after applying force in z-direction to the model. The combination of play and bent steering rack made the bump steer result unusable. Therefore the decision was made to not involve the bump steer in the result.

5.3.6 Colour coding

All 3D printed components are colour coded which helps for a clear and easy visualisation suitable for educational purposes as it can be seen from far away. For the most important parts the more visible colours were chosen. The design of the parts also help towards the visualisation and educational value as it is an open “skeleton” design with as few hidden parts as possible, which makes all components clearly viewable. The drawback of that is a design which could possibly be less resistant to large forces as there could be more material added if the visibility was not a priority. Under normal conditions they work as intended.

5.3.7 MacPherson

A parameter that affects the design is the differential that is integrated in the centre of the subframe. The design of the driveshafts is simplified compared to real driveshafts, but they still fulfill their purpose of driving the wheels when the differential is turned. The driveshafts are mounted to link the differential to the upright to enable propulsion. This is to simulate propulsion on the MacPherson suspension. Other parts as control arms and uprights are more similar to their full scale counterparts but with small modifications. These differences are often done to simplify the product and enable it to be 3D printed. The anti-roll bar was also simplified due to the problems with attaching it to the coilover damper. Because of this there is hard to reassure that the properties of the anti-roll bar is correct but the same principle was used to design it.

Since the differential requires an abnormal amount of space the packaging problem of the steering rack has been an issue. The solution is clever but due to the weakness of the 3D printed material the U-shaped steering rack is affected by a bending moment causing the steering joints between the rack and the upright to clash with the subframe. A solution to this would be to manufacture a stiffer steering rack with the design at hand. E.g. a steel rack would withstand the moment much better. Another solution would be to design a u-shaped support for the steering rack to ensure its position. Due to the steering not being assembled until the very last phase of the project, these problems were not noted before it was too late to make improvements. However, when giving full steering the inner wheel gives 36.3 degrees and the outer wheel is 25.7 degrees. This is less than the design values that was given but it still shows the function of the ackerman steering, which is desired.

The upright is also affected by the weakness of the material. The attachment of the coilover damper to the upright is constructed to withstand as much of lateral forces as possible with the limitations at hand. Unfortunately, the weakness of material forces the coilover to first move laterally before moving in its correct direction. This causes the camber travel of the wheel to not take the correct path and it differs to the calculated result. However, it does not harm the end purpose of this model since the negative camber is increased with the weak structure of the upright.

The ride height of the printed model is 40.8 mm and the given ride height of the full scaled car is just above 200 mm and with the factor of 5 in downscaling the result is acceptable. Similar results can be found when analyzing the track width that measured at 317.1 mm compared to the design target of 318.1 mm. The travel of the wheels are 8.6 mm in jounce compared to 11.8 mm in rebound of the measured model which is a total of 20.3 mm of travel. The range should however be a factor of two larger and this is because of the shorter coilover. A solution to this would be to find a better coilover damper and redesign the subframe. When it comes to the static camber it was measured to -0.5 degrees compared to the design goal of -0.9 degrees which met the goal fairly well. The camber gain travels from -0.9 to 1.3 degrees which matches the goal value better than the static camber. Even though the camber is not completly accurate the representation of camber is still there and it is possible to explain the effect of it.

5.3.8 Double Wishbone

The double wishbone suspension model unfortunately did not get the correct colour of the subframe, it is blue instead of black. The functionality is still the same but compared to the other suspensions it differs in colour coding which can be misleading. This is because of printing problem and lack of material.

The suspension was designed for 21 degrees of camber-gain/meter travel, as it would have in full scale which is a more recognizable value. When calculating the hardpoints that affected the camber gain the measurements were set in integer millimeter to get easy manageable numbers. This made the final design camber gain 20.2 degrees/meter. The measured camber gain on the model was calculated between full jounce and rebound, it resulted in 20.16 degrees/meter which is within the tolerance and more than good enough. Therefore no re engineering was needed. The static camber on the model was designed as -0.5 degrees and measured to -0.52 which is a good result. The camber results was overall better than expected and could be a coincidence when looking at the overall performance of the ball joints and joints.

When the hardpoints were tuned for the correct bump steer there were some difficulties to get the numbers to add up. Because of this, the bump steer was made larger as it only has a positive impact on the ability to visualise the bump steer. The final design result of 15.7 degrees/meter travel was in the upper span of the tolerance region but according to the group better than the exact requirement for visualisation reasons. The same principle was utilized during the ackermann design, the benchmark values are 10-20% and the value on the model is 15.2% at 20 degrees steer angle for the inner wheel which is over the minimum and enables good visualisation. The ackermann at 20 degrees steering angle on the inner wheel was not possible to calculate accurate due to play in the steering. The ackermann at full steer angle was calculated to 31.1%. The values on the actual model are not entirely correct but they are large enough to illustrate the ackermann steering geometry.

The amount of jounce and rebound is more than in the specification due to intentionally rounded up values, but the ratio between jounce and rebound is correct. In total the movement of the wheel centre on the model is 4.6% larger than the total downscaled travel given from the task. Having more total travel makes visualisation easier because the change of angles is bigger in the end positions. The extra 4.6% of travel is therefore considered as positive for the task of the model's purpose. The ride height did not have a requirement, it was therefore developed during the whole process with the aim of a lower front then rear. The final ride height on the model, 28.2 mm is at the smallest but it gives a lover front and is therefore considered as good.

The measurement that is most affected by the shrinkage of the plastic is of course the one that involves most stretch of parts, the track width. It was designed to be 317.7 mm but was measured to 316.2 mm. It was expected but could have been avoided by compensating for the shrinkage.

5.3.9 Trailing Arm

One specific design feature that differs from the others is that the tie blade had to be attached to the subframe instead of the body, where it should be. This gives an even larger and less representative subframe. As long as the spectator is aware of this it does not affect the understanding of the functionality of a trailing arm suspension.

The rear ride height of the Chalmers suspension is said to be 200 mm. In scale 1:5 this equals to 40 mm. With the model in design mode the height is measured to 37.2 mm from the lowest part of the chassis at the rear to the ground. Since the chassis only purpose in this case is to be a rigid connection between the front and rear suspensions, this was made to have an approximately correct ride height and is for this reason an acceptable value.

The travel specified for the Chalmers MacPherson suspension at the wheel is 38 mm and as the Chalmers trailing arm did not have that data specified that was also used when designing this suspension. Out of the 38 mm, 16.3 mm is jounce and 21.7 mm is rebound. The coilover used in the scale model has a travel of 33.3 mm. Because of the coilover being mounted further inwards on the spring link, which acts as a lever, the gearing gives a travel of 37 mm at the wheel, which is acceptable given the conditions. The jounce is measured to 17.6 mm and the rebound 19.4 mm. In Catia V5, the proportions between these values were the same as in the Chalmers MacPherson suspension. As there are no precision tools available to take these kind of measurements that could be the cause of the offset or otherwise because of the 3D printer.

The toe angle given by the Chalmers trailing arm suspension is 0.1 degrees inwards in design mode. Because of the precision of the 3D printers used, the rim and tyre varies very much under a revolution causing the toe to, at a certain position, be a factor 10 larger than the design value. As of this, the decision was made to not involve this result since it would not add value to the report.

A negative camber of -1 degree is specified and the measured one is -0.1 degree. This suffers from the same problem as the toe angle and could be fixed with a higher precision printer or of the self components. However it is hard to find the right tyre and rim as of the specified dimensions and design. As there is no data specified on the camber gain during rebound and jounce, the result is compared to the given value of the double wishbone. It is 21 degrees/meter in full scale and the measured value is 27 degrees/meter. How the camber gain of a double wishbone suspension relates to a trailing arm suspension is unknown, but there are in this case now drawbacks with a larger value as this only gives a better visualization. The comparison is also a bit unreliable as the given value is in a linear range close to the design position and the calculated one is for the full travel of the suspension.

The track width specified from the Chalmers suspension is 1586.4 mm or 317.3 mm in 1:5 scale. From the measurement taken the track width is 315.4 which aligns very well with the theoretical measure. The difference of about 2 mm comes from the already mentioned problem of the printed parts shrinking a few percents during cooling.

5.3.10 Live Axle

When applying forces to the wheels fairly realistic movement from the axle and links can be seen. One of the reason the movement is not fully realistic is because the suspension model is based on pictures and not hardpoints for an actual suspension. Because of this the stabiliser arms are not fully functional and only one can be mounted normally. When the other one is mounted it is under strain resulting in impaired movements. Even though these problems are there the base movement of the suspension is correct meaning it is still illustrative.

Another issue is that the axle stands too far to one side. This can be because of the Panhard rod being too short in combination with the coilover damper getting too long with the mounted extensions. The coilover damper and mount extension causes other problems as well. For one the wheel travel is too short and the ride height is too high which can be corrected with these components. Furthermore because of the way the wheels are mounted, with a spacer, the track width gets increased by the thickness of the spacers. This can be seen if a comparison between the calculated value of 317.3 mm and the measured value of 319.5 mm is made. This does not affect the suspension much and is therefore not an issue. The support arms with their complementary pieces are narrowed and rounded which is not the case in reality. This solution allows for more accurate movement which is more important for the model than a fully representative look. The fitting of the wheels is not representative either but is not a priority. For the same reason the Panhard rods subframe mount is odd.

5.3.11 Chassis

Two chassis with interchangeable front suspensions have been delivered. They have 3 mm screw holes to enable swapping between the two front suspensions to be able to make all four combinations possible. This works as intended, but if there was more time a faster and easier locking mechanism could have been developed. There is also a handle mounted on the chassis to make applying force to the vehicle and see the movement on the suspensions easier. Further development could be done to make the handle more ergonomic for carrying the model and applying wanted force. A calculation on where the handle should be placed could be done to get the right centre of gravity and the right distribution of force. This will be hard to accomplish because of the different stiffness on the dampers in the front and rear suspensions. The chassis does not resemble a real chassis as it only serves to connect the front and rear suspensions. A lot of free space is found on the structure, because of the needed size to fulfill dimensions and for future possibility to make it remote controlled if wanted.

6 References

[1] B. Hall, "Suspension systems and components", in *An introduction to Modern Vehicle Design*, J. Happian-Smith, Ed. Oxford, England: Butterworth-Heinemann, 2001, pp. 277-331.

[2] B. Jacobson et al, *Vehicle Dynamics: Compendium for Course MMF062*, Gothenburg, Sweden: Chalmers University of Technology, 2014.

[3] J. Reimpell, H. Stoll, J. W. Betzler, *"The Automotive Chassis: Engineering Principles"*, 2nd ed., Oxford, England: Butterworth-Heinemann, 2001. [Online]. Available: http://www.sciencedirect.com/science/book/9780750650540. Accessed: April 20, 2017.

[4] R. N. Jazar, "*Vehicle Dynamics: Theory and application*", 2nd ed., New York, NY, USA: Springer, 2014. [Online]. Available: https://link.springer.com/book/10.1007%2F978-1-4614-8544-5. Accessed: April 20, 2017.

[5] Ingemar Johansson, Professor of the practice, Chalmers University of Technology, CEVT

[6] D. Koladia, "Mathematical Model to Design Rack And Pinion Ackerman Steering Geometry", *International Journal of Scientific & Engineering Research*, vol. 5, issue 9, September, 2014. ISSN 2229-5518, [Online]. Available: http://www.ijser.org/researchpaper/Mathematical-Model-to-Design-Rack-And-Pinion-Ackerman-Steering-Geomtery.pdf. Accessed: April 20, 2017.

[7] *Road vehicles - Masses - Vocabulary and codes*, ISO1176, 1990.

[8] By Casito. "McPherson Strut", 2016.[Electronic image]. Available: https://commons.wikimedia.org/wiki/File:Mcpherson_strut.jpg. Accessed May 7, 2017.

[9] By Clker-Free-Vector-Images. "Tyre vector image", 2014.[Electronic image]. Available: https://pixabay.com/en/tyre-wheel-rubber-automobile-311665/. Accessed May 7, 2017.

[10] By Tennen-Gas. "Axle - 5 Link rigid 03", 2007.[Electronic image]. Available: https://commons.wikimedia.org/wiki/File:Axle_-_5_Link_rigid_03.gif. Accessed May 6, 2017.

[11] G. Genta and L. Morello, "Suspensions", in *The Automotive Chassis Components Design: Volume 1: Components Design*, Dordrecht: Springer, 2009, pp. 133-237.

[12] L. Hågeryd, S. Björklund and M. Lenner, *Modern produktionsteknik. Del 1*, 1st ed. Stockholm: Liber, 2002.

[13] M. Mägi, K Melkersson, "Kuggväxlar," in *Lärobok i maskinelement*, 1st ed. Gothenburg, Sweden: Chalmers University of Technology 2012.

[14] "Kuggstänger", *Mekanex Maskin AB*, 2017. [Online]. Available: http://www.mekanex.se/produkter/komponenter/kuggstanger/. Accessed: March 28, 2017.

Appendix A - Hardpoints

A.1 Chalmers MacPherson Suspension

Chalmers front suspension - MacPherson hardpoints
Values are given in mm

Name	X	Y	Z
Wheel centre (disc to rim)	1513,32	-845,793	247,864
Wheel centre (offset 52mm)	1513,32	-793,8	247,047
Front lower control arm mounting	1531,2	-373	161
Rear lower control arm mounting	1815	-408	168
Ball joint knuckle	1500,4	-767,7	126,5
Strut upper	1566,861	-589,724	815,471
Strut aligning point	1510,8	-649	285,717
Outer tie rod ball joint	1664,5	-732,3	205,5
Inner tie rod ball joint	1708	-370	223,5
Stabilizer to subframe pivot	1843	-300,2	220
Stabilizer to link	1587	-580	264
Stabiliser link to strut	1595	-592,3	540,2
Rebound		-110	
Jounce		80 (83 with overtravel)	
Overtravel		3	
Racktravel		82,6	

A.2 Double Wishbone Suspension

Front suspension - Double wishbone hardpoints
Values are given in mm
Scale 1:5

Name	X	Y	Z
Rear lower wishbone mounting	-33,5	73,8	0
Front lower wishbone mounting	33,5	73,8	0
Outer lower wishbone ball joint	0	153,8	0
Rear upper wishbone mounting	-28,5	75,511	49,076
Front upper wishbone mounting	28,5	75,511	49,076
Outer upper wishbone ball joint	0	143,113	56,43
Coilover damper upper mounting	0	81,511	58,268
Coilover damper lower mounting	0	111,8	0
Inner tie rod ball joint	-41,52	73,8	21,5
Outer tie rod ball joint	-31,005	147,647	21,448
Wheel centre (disc to rim)	0	158,775	28,304
Wheel centre (offset 52mm)	0	169,175	28,393

A.3 Chalmers Trailing Arm Suspension

Chalmers rear suspension - Trailing arm hardpoints
Values are given in mm

Name	X	Y	Z
Wheel centre (disc to rim)	4248	-845,2	228,9
Wheel centre (offset 52mm)	4248,1	-793,2	228
Tie blade to body	3718,1	-626,8	283,2
Tie blade to knuckle upper	4110,4	-642,4	264,9
Tie blade to knucle lower	4119	-642,7	174,9
Camber link inner	4141	-385,5	317,6
Camber link outer	4158	-689,4	359,4
Spring link inner	4406,5	-149,6	158
Spring link outer	4343	-725,3	167,5
Toe link inner	4141	-395	134,5
Toe link outer	4163	-685,3	122
Damper upper	4304,5	-572,8	706,6
Damper lower	4354,8	-618,1	162,5
Stabilizer to subframe pivot	4007,3	-419,5	188,6
Stabilizer to link	4218,6	-620	136,8
Stabar link to knuckle	4185,6	-640,8	276,4

Appendix B – Measured Results

Type	MacPherson	Double Wishbone	Trailing arm	Live axle
Track width [mm]	318.05	316.2	315.44	319.5
Ride height [mm]	40.75	28.2	37.2	48.34
Full jounce Camber [degrees]	-0.929	-2.8	-3.1	0
Static Camber [degrees]	-0.516	-0.52	-0.1	0
Full Rebound Camber [degrees]	1.34	1.03	1.9	0
Travel full jounce [mm]	8.55	16.15	17.55	11.54
Travel full Rebound [mm]	11.75	23.6	19.4	15.86
Steering Inner [degrees]	36.25	44.83	x	x
Steering Outer [degrees]	25.72	31.27	x	x
Ackermann % at full steer angle	28.3	31.13	x	x