MASTER

Design for a Formula Student race car

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Master's Thesis

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Dutch summary

Achtergrond informatie
Sinds zeven jaar wordt in Engeland de Formula Student competitie gehouden. Voor 2004 is het doel gesteld door een groep studenten van de Technische Universiteit Eindhoven om daaraan mee te doen. In 2003 is het Formula Student Racing Team Eindhoven opgericht om mee te doen met deze wedstrijd voor technische universiteiten van over de hele wereld. Om mee te kunnen doen met deze wedstrijd moet een ontwerp voor een éénpersoons race auto gemaakt worden, gebaseerd op een Formule 1 auto, maar dan kleiner en met minder vermogen. Het doel voor het team is om in 2004 met de Formula student competitie mee te doen met een ontwerp, dat is gebaseerd op de door de organisatie opgestelde regels, en niet met een rijdende auto. Voor het jaar 2005 is het doel om met een auto mee te doen in klasse 1.

Opdracht
De opzet van de ontwerpcompetitie is dat een productiebedrijf een team heeft gestrikt om een prototype auto te bouwen, welke door hen geëvalueerd zal worden. Deze auto moet een goede prestatie hebben met betrekking tot het accelereren, het remmen en het rijgedrag van de auto. Deze moet goedkoop, goed onderhoudbaar en betrouwbaar zijn, omdat het productiebedrijf van plan is om 1000 auto's in één jaar te produceren voor een prijs onder €21000. De uitdaging is om een ontwerp voor een prototype te maken, dat zo goed mogelijk aan deze eisen en wensen voldoet.

Doel
Het doel voor dit project is om een ontwerp te maken voor een Formula Student auto, rekening houdend met de gewenste rijeigenschappen en de regels opgesteld door de Formula Student organisatie. Daarvoor wordt de auto onderverdeeld in drie aparte delen, namelijk het chassis, de ophanging en de aandrijflijn. De stuurinrichting, het remsysteem en de aanpassingen aan het motorblok om deze geschikt te maken voor de Formula Student competitie zullen ook een deel van het project vormen.

Conclusies
Het ontwerp voor de Formula Student auto is onderverdeeld in drie aparte delen. Voor het chassis zal een aluminium sandwich structuur worden gebruikt om een licht en stijf chassis te vormen. De sandwich panelen zullen onderling verbonden worden met behulp van aluminium spanten. In dit chassis zullen verstijvingen worden gebruikt om de benodigde veiligheid voor de coureur te garanderen. Voor de wielophanging zullen triangels met ongelijke lengtes worden gebruikt om de vereiste wieluitslag en het gewenste rijgedrag op korte en krappe circuits te bieden. Een Suzuki 600 cc motorblok, dat normaal voor een motorfiets wordt gebruikt, zal worden gebruikt voor de aandrijving van de auto. Voor de overbrenging van het motorvermogen naar de achterwielen zal een ketting en een kettingwiel worden gebruikt. Dit kettingwiel zal op het differentieel worden gemonteerd, welke nodig is om het motorvermogen over de achter wielen te verdelen, rekening houdend met de grip van de achterwielen.

Aanbeveling
Om in de toekomst een competitieve auto te hebben zal het gehele ontwerp opnieuw bekeken moeten worden met betrekking tot de losse onderdelen, omdat niet alle onderdelen, door gebrek aan tijd, doorgerekend en geoptimaliseerd zijn.

English summary

Background
For the last seven years the Formula Student competition has been held in England. The aim for the Technische Universiteit Eindhoven was to compete in the 2004 event. In 2003 the Formula Student Racing Team Eindhoven (FSRTE) has been established to participate in this competition for technical universities from all over the world. To compete in this event a design had to be made for a single seated race car based on a Formula 1 car, but smaller and less powerful. The aim for the team in 2004 was to compete with a design for a race car, based on the regulations set by the organization, and not with a drivable car. For 2005 the aim is to compete with a car in class 1.

Objective
For the purpose of the design competition, the teams have to assume that a manufacturing firm has engaged them to produce a prototype car for evaluation. This car should have a very high performance in terms of its acceleration, braking and handling qualities. The car has to be low in cost, easy to maintain and reliable, because the manufacturing firm is planning to produce 1000 cars each year at a cost below €21.000. The challenge is to design a prototype car that best meets these objectives.

General aim
The aim for this project is to make a design for a Formula Student race car, considering the desired car handling and the regulations determined by the Formula Student organization. Therefore the car will be divided into three separate parts: the chassis, the suspension and the drive train. The steering mechanism, the braking system and the modifications needed to make the engine suitable for the Formula Student competition will also be part of the project.

Conclusions
The design for the Formula Student car has been divided in three main parts. For the chassis an aluminium sandwich structure will be used to create a light and stiff chassis. The sandwich panels will be connected using solid aluminium frames. In the chassis extra reinforcements will be used to provide the necessary safety for the driver. For the suspension an unequal length double wishbone structure will be used to provide the necessary wheel travel and to improve the handling of the car on short and tight circuits. A Suzuki 600 cc engine, which is normally used for motorbikes, will be used. For the transmission a chain and a chain wheel will be used. This chain wheel will be mounted on a differential, which will be used to divide the engine power over the two rear wheels depending on the grip of the tires.

Recommendation
To make the car competitive, the overall design has to be elaborated upon the designed parts, because not all parts have been calculated and optimized thoroughly, due to the lack of time.

Preface

In the beginning of November 2003 I started a graduation project to finish a Mechanical Engineering course in the Constructions and Mechanisms group at the Technische Universiteit Eindhoven. This project has been an assignment for the Formula Student Racing Team Eindhoven. In the beginning of the 2003/2004 academic year six students, who liked to compete in a racing competition for technical universities from all over the world, have established a team. The competition, in which they wanted to compete, is called the Formula Student competition. This competition has been set up for students of technical universities from all over the world to design, build, develop, market and compete as a team with a small single seated racing car.
This single seated race car has to be built in Formula 1 style, but the car will be smaller and less powerful. The competition has been divided in three different categories: teams with a completed car, which is able to drive (class 1), teams with only some parts that already have been built (class 2) and teams with only a design for the car (class 3). For the 2004 competition the Technische Universiteit Eindhoven competed in the class 3, but the objective for the 2005 competition is to compete in class 1. For competing in class 3 a design for a Formula Student car had to be made according the rules, which have been set by the organization. Because this is the first year that the team participated in the competition, the design had to be made from scratch, without any foreknowledge. To coordinate the design of the different parts of the racing car I've been appointed to be the head designer for the team.

In the beginning of the project the car, has been divided in three different parts: the chassis, the suspension and the drive train. The aim was to design these parts sequentially for a stepwise design of the car. Later on it became clear that the interaction between the different parts causes problems when the design will be made sequentially. The design for the chassis requires other choices than the design for the suspension, which resulted in a lot of compromises. Therefore in the final stage of the project the car has been considered as one entity to prevent mutual conflicts between different parts, when designing those separately. This method hasn't been very orderly, but this is the only way to design a complete race car, because the final design for a race car is one big compromise.

During the project Mr Nick Rosielle has been my tutor as head of the Constructions and Mechanisms group. His knowledge has been very useful for making the design choices for the chassis, the suspension and the drive train. Therefore I want to thank him. I also want to thank Mr Eef Reker for his support during the project and his assistance by building the mock-up for the chassis. Furthermore I like to thank the students in the Constructions and Mechanisms group, which have been very helpful during the Monday sessions.
I want to thank Mr T. Serne and Mr P. Brinkgreve for their assistance during the project regarding the handling of the car and for their technical knowledge about designing a car. Last but not least I want to thank the team members of the Formula Student Racing Team Eindhoven for the mutual cooperation and for the pleasant trip to England for competing in the 2004 Formula Student event, achieving a fifth place in class 3.

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Chapter 1: Introduction

1.1: Project background

The Formula Student competition is a competition in which technical universities from all over the world will be competing. This competition has been set up for university students to design, build, develop, market and compete as a team with a small single seated racing car. First of all a short overview of the Formula Student history will be given.

The United States started running their Formula SAE programme in 1981. In 1998 two US cars and two UK cars competed in a demonstration UK Event. The initiative was considered to be very worthwhile in providing students with excellent learning opportunities and practical skills, which resulted in the UK Formula Student competition. Formula Student is different from Formula SAE in that it is designed to be a progressive learning exercise throughout a three or four year academic course. However, the same rules are used for both Formula Student and Formula SAE (with some small changes), which means that student teams can enter their cars in the Formula SAE in the US, Formula Student and Formula SAE in Australia. Formula Student provides students with a real-life exercise in design and manufacture and the business elements of automotive engineering. The competition has been divided in three different categories: teams with a completed car, teams with some finished parts and teams that only made a design for a car. For the 2004 event the Technische Universiteit Eindhoven aimed to compete in the last category.
In 2004 Formula Student was held for the seventh time and it attracted 66 entries from different universities from 19 different countries. These entries are mainly from the UK, but also from the mainland of Europe and from North America, Asia and Australia.

1.2: Objective

For the purpose of the competition, the teams have to assume that a manufacturing firm has engaged them to produce a prototype car for evaluation. The intended sales market is the non professional weekend autocross, hill climb or sprint racer. Therefore, the car must have very high performance in terms of its acceleration, braking and handling qualities. The car must be low in cost, easy to maintain and reliable. In addition the car's marketability is enhanced by other factors such as aesthetics, comfort and use of common parts. The manufacturing firm is planning to produce 1000 cars per year at a cost below €21.000. The challenge is to design and fabricate a prototype car that best meets these objectives. Each design will be compared with other competing designs to determine the best overall car. It should be stressed that it is not simply the fastest car that wins; the teams should balance speed with safety, reliability, cost and good handling qualities.

General aim
The aim for this project is to make a design for a Formula Student race car, considering the desired car handling and the regulations determined by the Formula Student organization. Therefore the car will be divided into three separate parts: the chassis, the suspension and the drive train. The steering mechanism, the braking system and the adjustments needed to make the engine suitable for the Formula Student competition will also be a part of the project.

1.3: Method

Starting with the project first of all a survey has been made for the teams participated in the 2003 competition to examine the features for a competitive car. By making some rough calculations in the first stages of the project the behaviour for a car with those features could be predicted. The results of those calculations have been used for the chassis design to estimate the size of the forces working on the suspension. These forces have to be guided into the chassis. Therefore the suspension layout, the dimensions and the position of the wishbones, had to be determined simultaneously with the chassis design. For the suspension layout the desired car behaviour has been determined regarding the layout of the track that has to be driven. The endurance track and the velocity curve for that race have been given in appendix A. During the design of the chassis the parts for the drive train has already been considered too, because the engine had to fit in the chassis and the chain wheel dimensions, to create the transmission from the engine towards the differential, have determined the shape and size for the rear part of the chassis. Later on the brake system and the steering mechanism have been integrated in the design of the car.

In the first phase of the project the separate parts had been designed sequentially, but when these parts had to be joined together to complete the design, they had to be adjusted to be compatible. The more extensive the design of the car the more interactions between the separate parts. Therefore the chassis had to be redesigned and adjusted a couple of times, to keep the parts coherent. This led to the conclusion that the final design for a race car is one big compromise.

1.4: Structure of report

The report has been build up in eight chapters. In the first chapter a short introduction and the design objectives for the project have been given, which will lead to a couple of assumptions and some calculations, given in chapter 2, to determine the main dimensions and the size of the forces, which will act on the race car during driving. In chapter 3 the chassis design has been described, based on the assumptions and forces calculated in chapter 2. Chapter 4 describes the design aspects, which have to be taken into account when designing an independent suspension. These aspects have been translated into a design for the suspension, given in chapter 5. The drive train for the Formula Student car will be described in chapter 6 and the required braking and steering mechanisms have been clarified in chapter 7. From chapter 3 till chapter 7 the overall design for the Formula Student car has been described and from that part of the report conclusions are drawn and recommendations are made, which have been described in chapter 8.

At some points in the report terms, or parts, have already been used before they have been described. This is inevitable, because there are a lot of interactions between the different parts of the car. Therefore the structure of the report is not the exact sequence in which the different parts have been designed. To give the report a logical structure the sequence described in this paragraph has been used.

Chapter 2: Analysis car properties

Creating a competitive car for the first year in the Formula Student competition is very difficult when starting from scratch. First of all some research has to be done by looking at other teams and the features of their cars. With the results from the 2003 competition the best cars suitable for the Formula Student competition can be determined. The teams have to do several different tests to prove that their car not only drives very fast, but also has a good handling, which is very important for the tortuous tracks. A car with a fast cornering technique and high acceleration possibilities will be competitive in the race. Regarding the results of the different separate events the features of a possible winning car can be determined by copying the features of those competitive cars.
For the design of the chassis it is important that some boundary conditions will be regarded to demarcate the application for the Formula Student car.

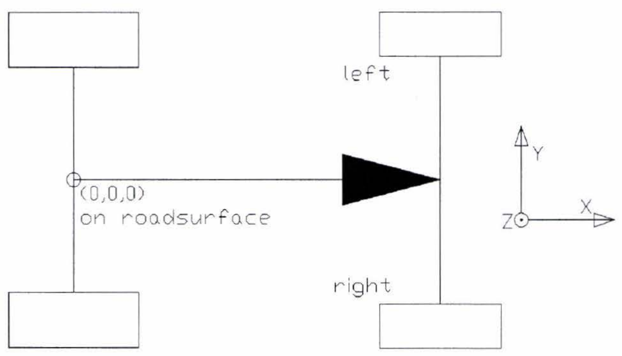
One of the most important features of the Formula Student car is the wheelbase. According to the rules a minimal wheelbase of 1525 mm has to be used for the car. Regarding the results of 2003 and the cars that have scored a lot of points in the endurance race and the autocross, a short wheelbase is favourable. For slalom the width of the car is important too and especially the front and the rear track of the car. Regarding the features of the best cars in 2003 the rear track of the car will be chosen smaller than the front track. Considering the fact that the rear wheels will be wider than the front wheels the width of the car measured over the front and rear wheels will be the same.

Without any knowledge of the dimensions and the properties of the car a realistic design for a Formula Student car can't be made. The dynamic forces acting on the suspension or the stationary forces acting on the car itself have to be predicted for a reliable design for the suspension and the chassis. Therefore some calculations will be made to estimate the size of these forces. With these forces the behaviour of the car can be estimated. The necessary assumptions for these calculations will be described in paragraph 2.1.

2.1: Assumptions

Theoretically the behaviour of the car can be predicted. Here to several calculations can be made for the behaviour of the car during acceleration, braking and cornering. These three actions will be executed sequentially when driving a race car. For making those calculations first of all an approximation for the total mass of the car has to be made. The position of that mass in the car has to be estimated too, to determine the position of the overall centre of gravity. The overall centre of gravity is one of the most important features for a race car, because the accelerating-, braking- and cornering-forces acting on the car will apply in the centre of gravity. These assumptions and other important features are shown in table 2.1.

Mass	300 kg	Mass front wheel	15 kg
Weight distribution	45/55 front/rear	Mass rear wheel	20 kg
x-position CoG	720 mm	Sprung mass	230 kg
y-position CoG	0		
z-position CoG	330 mm	Scrub radius	0 mm
		Trail	30 mm
Wheelbase	1600 mm	Maximum body roll	5 degrees
Track front	1250 mm	Standard toe	2 mm
Track rear	1200 mm	Standard camber	-0,5 degrees
Front tires	7.0/20.0-13 Avon 3 Ply Pro-Series	z-position lower pivot point upright	152,5 mm
Rear tires	8.2/22.0-13 Avon 3 Ply Pro-Series	z-position upper pivot point upright	377,5 mm
Rim width front	6 inch		
Rim width rear	8 inch	Minimum turn radius	3500 mm
Diameter front tires	530 mm	Driver length	1880 mm
Diameter rear tires	567 mm	Steering	neutral Ackermann
Stiffness tires (C_z)	140 N/mm	Brake balance	75/25 front/rear



The terms used in this table will be explained in the following chapters of the report.

Table 2.1: Assumptions

The tires are one of the most important parts of the car, because the generated forces by the tires during driving determine the handling of the car. Therefore a choice has to be made, which tires will be used for the Formula Student car. These tires will be described in paragraph 2.2.

2.2: Tires

Tires can resist side forces during cornering. Using the tire characteristics, the friction coefficient between the tire and the road can be determined. The tires that will be used are Avon 3 Ply Pro-Series. The choice has been made to use different tire sizes for the front wheels and the rear wheels. The weight distribution is 45/55, which means that the static load on the rear wheels will be 165 kg (55% of 300 kg) and on the front wheels 135 kg (45% of 300 kg). Therefore the rear wheels will be chosen wider than the front wheels; the front wheels 6 inch wide and the rear wheels 8 inch wide. These 8 inch tires can resist higher forces than the 6 inch tires, because the contact patch with the road is larger. The diameter of the rim for the front and the rear wheels is exactly the same: 13 inch, but the width of the rim is different: respectively 6 and 8 inch. The height of the tire is 55% of the tire width. This will result in a front tire diameter of 530 mm and a rear tire diameter of 567 mm.

The friction coefficient for the tires will be calculated using the tire characteristics. This friction coefficient determines the maximum lateral acceleration of the car. Therefore the maximum cornering force is necessary, which can be determined using the tire characteristics for the front tires in figure 2.1.

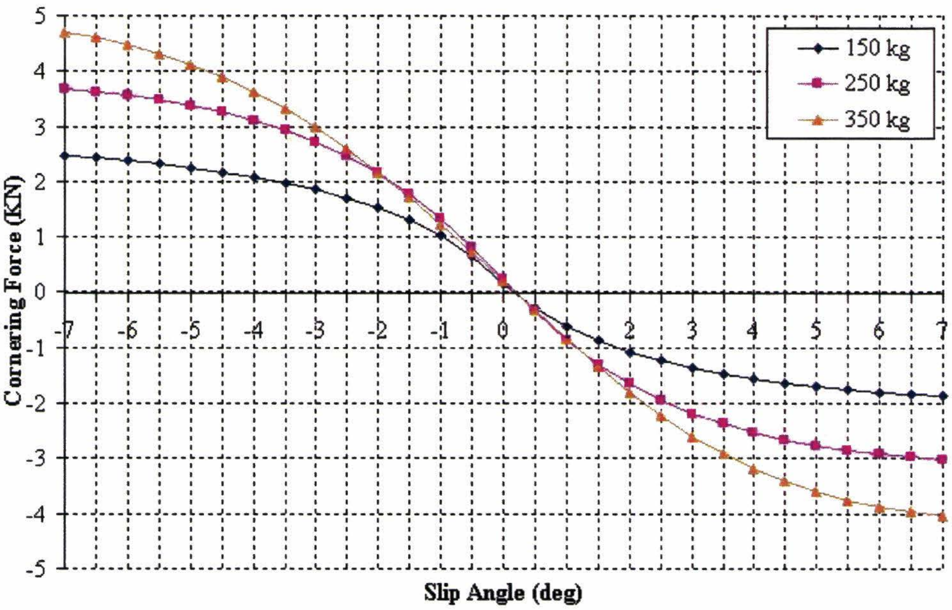


Figure 2.1: Front tire characteristics

Using the characteristics in figure 2.1 the friction coefficient can be calculated. This coefficient determines the size for the negative acceleration that can be achieved during braking and the lateral acceleration during cornering. For the longitudinal acceleration the rear tire characteristics have to be used.

Assuming that a worst case scenario will happen in a bend, the two inner wheels will lift off the ground, the outer front wheel has to support 135 kg. This means that the cornering force that can be generated by the tire will be 1700 N (Extrapolating the three lines in figure 2.1 using a slip angle of 7 degrees and a mass of 135 kg). The friction coefficient can be calculated using the maximum cornering force and the normal force at the front tire, as described in equation 2.1.

(Equation 2.1)

$$\mu_{front} = \frac{F_{corner}}{F_{n,front}}$$

$$\mu_{front} = \frac{F_{corner}}{m_{front} \cdot g} = \frac{1700}{135 \cdot 9{,}81} \approx 1{,}3$$

For the friction coefficient for the rear tires the tire characteristics in appendix B will be used. These characteristics represent the maximum cornering force for the rear tires, which will be 2500 N using 165 kg for the mass on the rear wheels.

(Equation 2.2)

$$\mu_{rear} = \frac{F_{corner}}{F_{n,rear}}$$

$$\mu_{rear} = \frac{F_{corner}}{m_{rear} \cdot g} = \frac{2500}{165 \cdot 9{,}81} \approx 1{,}5$$

The results from the calculation for the friction coefficient is that braking and cornering will be done with 1,3g and accelerating can be done with 1,5g. The lateral and longitudinal acceleration can be recorded in a friction boundary. The friction boundary of a tire is basically a vehicles performance envelope. It's expressed in lateral, accelerating and braking g's. When graphed, the friction boundary looks like an egg with the lateral g's on the x-axis and the longitudinal g's on the y-axis. This friction boundary is shown in figure 2.2.

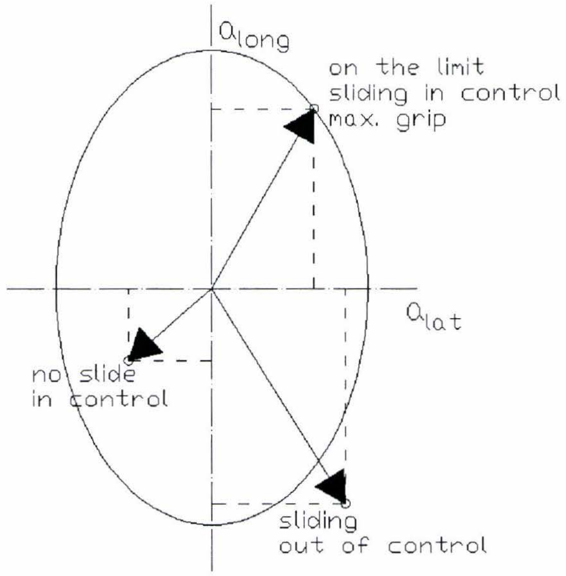


Figure 2.2: Friction boundary

The arrows in figure 2.2 represent resultant accelerations. A maximum longitudinal and lateral force can be generated, but not simultaneously. When a combination of longitudinal and lateral forces occurs, only that combination that will remain within the friction boundary will keep the car on the track. Using the friction coefficients calculated in this paragraph the load transfer during driving can be determined.

2.3: Load transfer

Using the friction coefficients for the front and the rear tires the dynamic load transfer in the car can be calculated. When a car is standing still or when the car is driving at a constant velocity (acceleration is zero) only static forces will be applied to the tires. A longitudinal or lateral acceleration (acceleration non-equal to zero) will cause a dynamic load transfer in the car. First of al the longitudinal load transfer will be explained in paragraph 2.3.1.

2.3.1: Longitudinal load transfer

A longitudinal acceleration will result in a point force in the centre of gravity of the car. The centre of gravity is situated above the road surface, which will cause a torque within the car, because the brake and acceleration forces will apply at the contact patch from the tires with the ground. This torque, the force in the centre of gravity multiplied with the height of the centre of gravity, will cause a dynamic load transfer between the front and the rear part of the car. During braking the inertia of the car will cause a forward load transfer and during acceleration this load transfer will be rearward.

The load transfer will cause a weight distribution different than assumed in table 2.1. The weight distribution on the four wheels is the sum of the static load and the dynamic load transfer.

The static load on the front axles and rear axles is respectively 135 and 165 kg. Figure 2.3 represents a side view of the car and it shows the forces working on the car during acceleration.

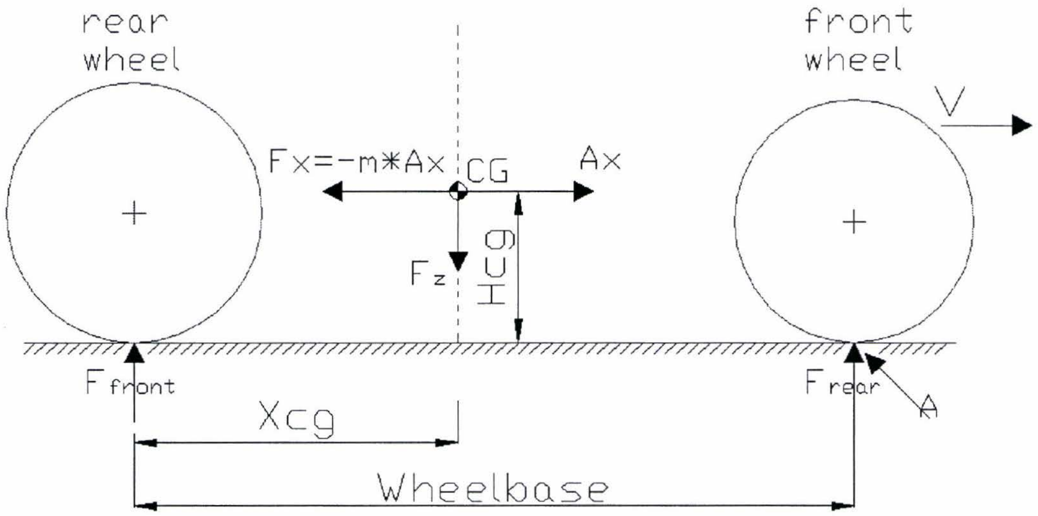


Figure 2.3: Longitudinal load transfer

In figure 2.3 the term Ax for the acceleration represents the amount of g's during acceleration, which is 1,5. ΔLx is the difference between the load on the front wheels and the load on the rear wheels. The calculation of the rearward load transfer during acceleration will be made using the equilibrium of moments around point A (in figure 2.3) in equation 2.3.

$$\Sigma M_A = 0 \qquad \text{(Equation 2.3)}$$

$$\Delta L_X \cdot Wheelbase - F_X \cdot H_{CG} = 0$$

$$\Delta L_X = \frac{H_{CG}}{Wheelbase} \cdot m \cdot A_X = \frac{330}{1600} \cdot 300 \cdot 1{,}5 = 93[kg]$$

Braking will be done with a negative acceleration of 1,3g. This means that the longitudinal load transfer from the rear to the front will be a factor 1,3/1,5 = 0,87 lower. This will be a forward load transfer of 80 kg. During a positive acceleration with 1,5g the weight distribution won't be respectively 135 kg and 165 kg for the front and the rear axle, but 135 - 93 = 42 kg and 165 + 93 = 258 kg. During a negative acceleration with 1,3g the weight distribution will be 135 + 80 = 215 kg for the front and 165 – 80 = 85 kg for the rear axle.

When a car is cornering the longitudinal load transfer is minimal, because the longitudinal acceleration won't be high. The lateral acceleration will cause a lateral load transfer, which will be described in paragraph 2.3.2.

2.3.2: Lateral load transfer

When cornering the outer side of the car will become much heavier than the inner side, due to the lateral load transfer. The distance between the centre of gravity and the road surface will cause this lateral load transfer. For the lateral load transfer during cornering the same method can be used as for the longitudinal load transfer. The track widths have been determined to be different for the front and the rear of the car and the static load on the rear wheels is bigger than for the front wheels. This will result in a different load transfer for the front side than for the rear side. These load transfers can be calculated using the equilibrium of moments around point B in figure 2.4.

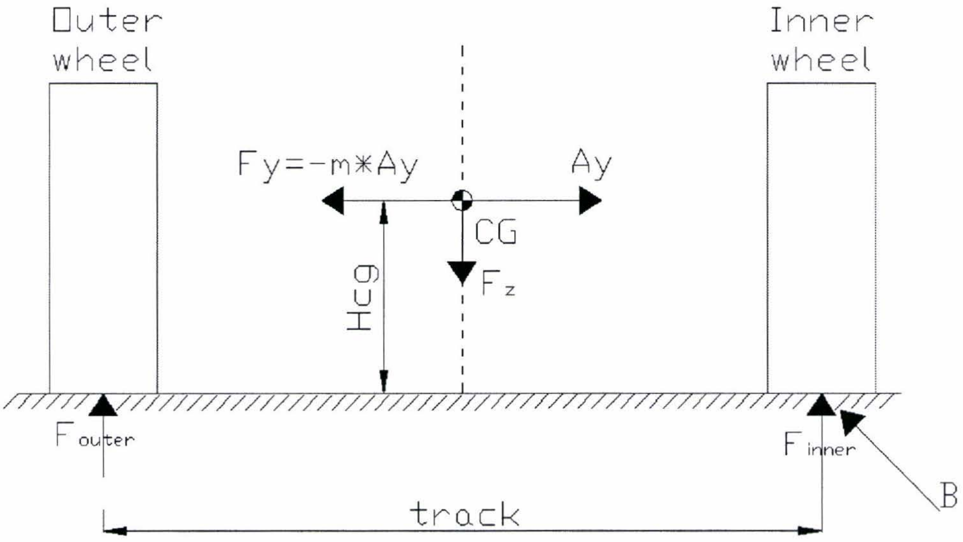


Figure 2.4: Lateral load transfer

In figure 2.4 the term Ay represents the amount of g's in lateral direction and ΔLy is the difference between the load on the inner and the outer wheel. Calculating the equilibrium of moments around point B, using equation 2.4, the lateral load transfer can be calculated.

$$\Sigma M_B = 0 \qquad \text{(Equation 2.4)}$$

$$\Delta L_Y \cdot track - F_Y \cdot H_{CG} = 0$$

$$\Delta L_{Y,front} = \frac{m_{front} \cdot A_Y \cdot H_{CG}}{track_{front}} = \frac{135 \cdot 1{,}3 \cdot 330}{1250} = 46[kg]$$

$$\Delta L_{Y,rear} = \frac{m_{rear} \cdot A_Y \cdot H_{CG}}{track_{rear}} = \frac{165 \cdot 1{,}3 \cdot 330}{1200} = 59[kg]$$

The two front wheels have to support 135 kg in static condition, which means that each wheel has to support 67,5 kg. During cornering with 1,3g the outer wheel has to support 67,5 + 46 = 113,5 kg and the inner wheel 67,5 – 46 = 21,5 kg. When driving an opposite bend the lateral load transfer will be exactly the same, but opposite, which means a load of 113,5 kg on the right wheel and 21,5 kg on the left wheel.
The two rear wheels have to support 165 kg, which means that each wheel has to support a static load of 82,5 kg. A lateral load transfer of 59 kg will mean that the outer rear wheel has to support a load of 82,5 + 59 = 141,5 kg and the inner rear wheel 82,5 – 59 = 23,5 kg.

During a lateral acceleration the chassis will start to roll towards the outer wheels. This will cause a change in y-position of the centre of gravity, which will result in a lateral load transfer too. This effect will be explained in paragraph 2.3.3.

2.3.3: Load transfer due to body roll

As described in paragraph 2.3.2 in a bend a lateral load transfer will be generated from the inner wheel to the outer wheel. Not only because of the change in load distribution, but also because of the roll of the car. During roll the position of the centre of gravity is changing. It will move towards the outer wheel, because the chassis with the driver will rotate about the roll axis of the car. This axis will be generated by the front and the rear roll centre, which will be explained furthermore in chapter 4. Figure 2.5 shows the displacement of the centre of gravity assuming a body roll of 5 degrees. The centre of gravity height has been estimated at 330 mm and the roll centre, which will be determined in chapter 4, has been placed 94,6 mm above the ground.

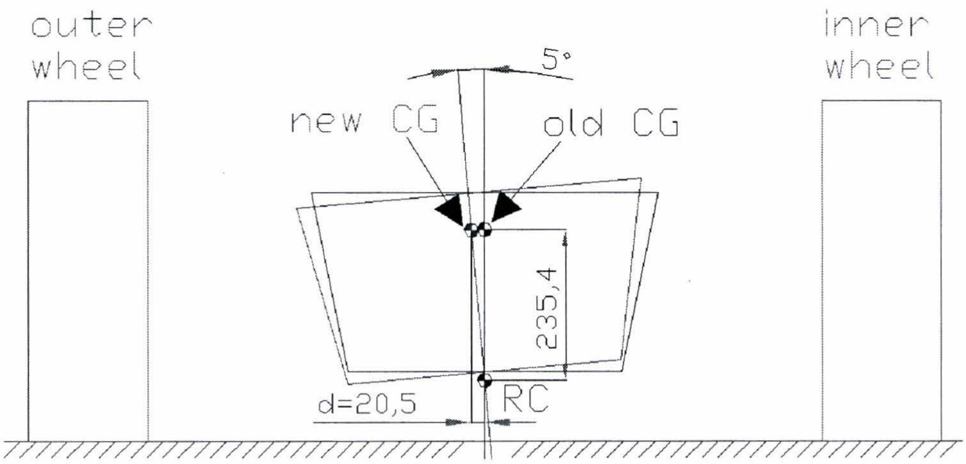


Figure 2.5: Lateral load transfer due to body roll

In figure 2.5 is shown that the displacement of the centre of gravity with a body roll angle of 5 degrees will be 20,5 mm. This will result in a higher load on the outer wheels, because the mass distribution is not symmetric anymore. The load transfer due to body roll for the Formula Student car is calculated, using the front track and the total mass, with equation 2.5.

$$\Delta L_{roll} = \frac{m \cdot (0{,}5 \cdot T_{front} + d)}{T_{front}} - 0{,}5 \cdot m \qquad \text{(Equation 2.5)}$$

$$\Delta L_{roll} = \frac{300 \cdot (625 + 20{,}5)}{1250} - 150 \approx 5[kg]$$

From equation 2.5 can be concluded that the bigger the roll angle, the bigger the change in y-position for the centre of gravity. An increase of parameter d will result in a higher load transfer due to body roll (ΔL_{roll}). Compared with the load transfer, due to lateral acceleration, the weight transfer due to body roll is small, but it cannot be ignored, because camber change will exist due to body roll while cornering. This will be explained in chapter 4.

To reduce the load transfer due to body roll stiffer springs could be used to increase the roll stiffness and to decrease the roll angle. The suspension layout can be changed to raise the position of the roll centre so that the distance between the roll centre and the centre of gravity will be reduced to minimize the arm for the roll moment. Lowering the overall centre of gravity will be the best effort, because then the load transfer due to lateral and longitudinal acceleration will be reduced as well, which will result in better road stability.

The load distribution in the car due to the longitudinal and lateral load transfer for each wheel is shown in appendix C.

The forces on the tires during driving depend on the longitudinal and lateral acceleration of the car. These forces have to be guided through the suspension into the chassis and will be described and calculated in paragraph 2.4.

2.4: Forces on A-arms

The position of the lower and the upper pivot points of the A-arms have been chosen as low and as high as possible in the wheels. The highest forces will be guided through the lower A-arm into the chassis. The bigger the distance between the road and the lower A-arm the bigger the forces through this A-arm. Therefore the choice has been made to choose the position of the pivot point for the lower A-arm that it will just fit in the wheel, preventing contact between the lower A-arm and the wheel during cornering. The position of the upper pivot point is not that important, because only a small part of the total forces on the wheels will be guided through the upper A-arm. However it is important to create an upright as long as possible, which will result in an upper pivot high in the wheel. Figure 2.6 and equation 2.6 and 2.7 determine the ratio between the forces in the upper and lower A-arms.

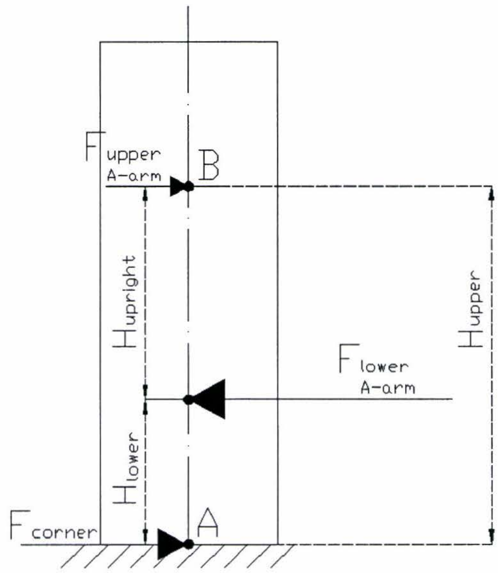


Figure 2.6: Forces on A-arms

$$\Sigma M_A = 0$$

$$F_{lower\_A-arm} \cdot H_{lower} - F_{upper\_A-arm} \cdot H_{upper} = 0 \qquad \text{(Equation 2.6)}$$

$$F_{upper\_A-arm} = \frac{H_{lower}}{H_{upper}} \cdot F_{lower\_A-arm}$$

$$\Sigma M_B = 0 \qquad \text{(Equation 2.7)}$$

$$F_{corner} \cdot H_{upper} - F_{lower\_A-arm} \cdot H_{upright} = 0$$

$$F_{lower\_A-arm} = \frac{H_{upper}}{H_{upright}} \cdot F_{corner}$$

Equation 2.6 and 2.7 show that the forces on the A-arms depend on the ratio between the height of the upright and the position of the pivot points regarding the road surface. Considering the fact that the z-position of the lower pivot point is 152,5 mm and the upper

pivot point 377,5 mm (the height of the upright is 225 mm) the ratio between the lower and upper A-arm force will be approximately 2,5 (using equation 2.6).
The force on the lower A-arm depends on the cornering force. The ratio between those two forces is approximately 1,68 (using equation 2.7). The ratio between the force on the upper A-arm and the cornering force is 1,68 / 2,5 = 0,68.
The cornering force for the outer front wheel can be calculated using equation 2.8, considering the fact that the front wheel not only has to support the static load but also the dynamic load, which will be higher due to the lateral load transfer. The mass that will be used for the calculation is the static plus the dynamic load on the outer front wheel. This will be 67,5 + 46 + 5 = 118,5 kg.

$$F_{corner} = \mu_{front} \cdot F_{N,corner} \qquad \text{(Equation 2.8)}$$

$$F_{corner} = \mu_{front} \cdot m_{corner} \cdot g = 1{,}3 \cdot 118{,}5 \cdot 9{,}81 = 1511[N]$$

Using this cornering force for the left front wheel the forces on the upper A-arm and the lower A-arm can be determined.

$$F_{upper\_A-arm} = 0{,}68 \cdot F_{corner} = 0{,}68 \cdot 1511 = 1027[N]$$

$$F_{lower\_A-arm} = F_{upper\_A-arm} + F_{corner} = 1027 + 1511 = 2538[N]$$

These forces will be applied on the left part of the front suspension during cornering with 1,3g. All the forces on the A-arms can now be calculated, because during braking and accelerating only the size and the direction of the forces at the tires will change. These forces are shown in appendix D for each tire. The ratio between these forces will always be the same, because the layout and the position of the upright have been chosen. All these forces working on the A-arms of the suspension, during acceleration, braking and cornering, have been calculated in appendix E.

For these calculations the assumption has been made that the load on the front wheel is placed at the same height as the load on the rear wheels, at the height of the overall centre of gravity. Furthermore the combination of lateral and longitudinal forces will change the situation, because only steady bends with a constant lateral acceleration have been analyzed.
Braking has to be done before a bend. When a driver doesn't brake before but in a bend a combination of longitudinal and lateral load distribution will occur. Considering the fact that the lateral forces in a steady bend will be as high as the tires can generate, extra longitudinal forces, due to braking, will result in sliding of the car. This sliding will be explained in paragraph 2.5

2.5: Car behaviour

The behaviour of a car depends on the type of steer that has been established. Especially the behaviour in a bend is important to be good, because in the Formula Student competition cornering is very important. The type of steer can be divided in three different set ups: understeer, neutral steer and oversteer. First of all these terms will be explained.

Understeer appears when the front wheels of the car (steering wheels) lose traction while cornering and the car will be pushed out of the normal intended cornering line. This will cause the car to head towards the outside of a corner. To overcome understeer for a rear wheel driven car like the Formula Student car it is desirable to gain more grip at the front wheels.

Slowing down, less braking and less steering are the possibilities for a driver to cope with understeer. The front wheels have to be slowed down enough to regain traction, but when the speed is too high continuous braking won't be the solution, because a locked/stopped wheel has less traction than a turning/rolling wheel. Less steering can be a solution, because less steering means more traction and after getting more grip the required steering action can be made again.

Neutral steer is the fastest way to round a bend, when all four wheels slide evenly. Since the total traction of each tire is being used, the entire available grip for the tires will be used at the contact patch with the ground. This is called drifting. Neutral drive is the fastest way to round a corner most of the time, but the disadvantage of neutral steering is that it is the hardest handling mode to achieve for a suspension tuner.

Oversteer appears when the rear wheels lose traction in a bend and head towards the outside of the bend forcing the front of the car to point towards the inside of the bend. This normally occurs due to the rear tires losing traction. When turning into a bend usually the front of the car will carry on as usual, while the rear end of the car swings around to the front of the car. Steering towards the outside of the bend will help to overcome oversteer.

When a car is entering a corner, light understeer is needed to provide the stability while the driver is easing off the brakes and building up cornering force. In mid corner, neutral steer is needed. In the exit phase, a slight oversteer would be preferable as it helps in tightening the path, especially if the car is driving the bend with a "slow in – fast out" style. However, the degree of oversteer must be progressive and easily controllable by applying and easing throttle. This is called power oversteer. Without power oversteer, the throttle has to be eased or the car will run wide out of the corner.
To determine whether a car is under steered or over steered the slip angles for the front and the rear tires are important. An overview is shown in figure 2.7.

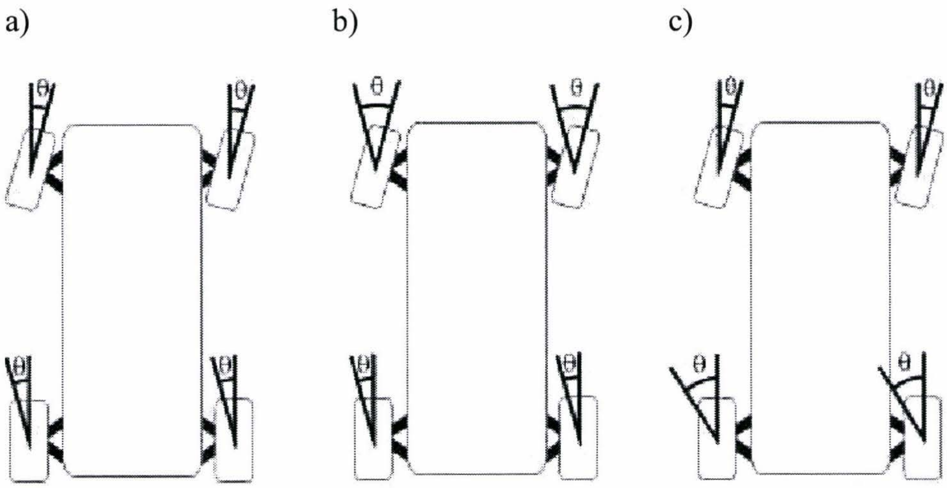


Figure 2.7: Slip angles a) neutral steer b) understeer c) oversteer

The difference between the front and the rear angles determine the behaviour of the car.

- Neutral steer (figure 2.7a) : Front Slip Angle = Rear Slip Angle
- Understeer (figure 2.7b) : Front Slip Angle > Rear Slip Angle
- Oversteer (figure 2.7c) : Front Slip Angle < Rear Slip Angle

The slip angle is the difference in which the wheels of a car are pointed, versus the angle that the tires contact patch is placed on the road. The main thing that affects slip angle is the manipulation of the individual load placed on each wheel while cornering. A rear wheel driven car has more weight on the rear wheels, which result in higher slip angles for the rear wheels. This will cause understeer. The weight distribution between the front and the rear wheels won't be more than a difference of 40/60 percent. Conversely the same will be for a front wheel driven car, where most of the weight has to be supported by the front wheels. This will develop oversteer. That is also a reason why a mid engine car with equally loaded tires will be more or less neutral. Slip angles, weight distribution and the Polar Moment of Inertia (PMI) are the main factors, which determine the handling of a vehicle.

PMI describes the mass distribution along the x-direction of the car. Low PMI cars have most of their mass around the centre of gravity, the centre part of the car. In high PMI cars the mass of the heavy parts have been placed far away from the centre of gravity. For low PMI cars it is easier to achieve a neutral balance, because of the centralized masses. High PMI cars tend to oversteer or understeer. For the Formula Student car a low PMI is desired. This means that the heaviest parts of the car, the driver and the engine, have to be placed near the centre of gravity. The mass of the wheels will have a great influence on the PMI, because the distance between the centre of gravity and the centre points of the wheels is relatively large. Therefore it is important to keep the wheel mass low, which will improve the handling of the car.

The set up of the car has to be determined by the driver and the suspension tuner. Some drivers like to ride with a little bit of understeer, but other drivers prefer little oversteer. In combination with the suspension tuner the setting has to be determined. Therefore the suspension has to be adjustable when driving with different drivers. The behaviour of the car in slalom can be determined using the yaw frequency of the car.
It is important that the Formula Student car won't have too much under- or oversteer, because for tortuous tracks, which are common in the Formula Student competition, these effects will cause a lot of problems, especially for slalom.

2.6: Slalom properties

The handling of the car, for example during slalom, can be expressed in the yaw frequency. This is the rotation around the z-axis. This rotation around the z-axis depends on the moment of inertia of the total car. For a rough estimation for the moment of inertia for the Formula Student car it has been divided in five parts: four separate wheels (the unsprung mass) and the chassis with the driver and the other parts as one piece. The total mass will be 300 kg. A complete wheel will be estimated at 15 kg, considering the fact that the rear wheels will be bigger, but they doesn't contain a brake disc and calliper. For the centre body of the car this will mean 240 kg including the driver. To compute roughly the shape of the central part of the car a rectangular box of 1600x500x500 mm will be used. For the yaw frequency the first aspect that has to be calculated is the moment of inertia for the five separate parts, starting with the rectangular box. The inertia of a rectangular box can be calculated using equation 2.9.

$$J_{box} = \frac{1}{12} \cdot m_{box} \cdot (l_{box}{}^2 + w_{box}{}^2) \qquad \text{(Equation 2.9)}$$

$$J_{box} = \frac{1}{12} \cdot 240 \cdot (1{,}6^2 + 0{,}5^2) = 56{,}2[kgm^2]$$

The mass of the wheels is concentrated in the centre of the wheels. To determine the moment of inertia for one wheel the distance between the centre of gravity and the centres of the wheels will be necessary.

$$r_{front} = \sqrt{(0{,}55 \cdot Wheelbase)^2 + (0{,}5 \cdot T_{front})^2} = \sqrt{0{,}88^2 + 0{,}625^2} = 1{,}08[m]$$

$$r_{rear} = \sqrt{(0{,}45 \cdot Wheelbase)^2 + (0{,}5 \cdot T_{rear})^2} = \sqrt{0{,}72^2 + 0{,}6^2} = 0{,}94[m]$$

r_{front} represents the distance to the front wheels and r_{rear} the distance to the rear wheels. For the calculation of the moments of inertia for the wheels, equation 2.10, the mass of the wheels will be multiplied by the square of the distance towards the centre of gravity.

$$J_{front} = m_{front} \cdot r_{front}^{\ 2} = 15 \cdot 1{,}08^2 = 17{,}5[kgm^2] \qquad \text{(Equation 2.10)}$$

$$J_{rear} = m_{rear} \cdot r_{rear}^{\ 2} = 15 \cdot 0{,}94^2 = 13{,}3[kgm^2]$$

For the total moment of inertia, the inertia of two front wheels, two rear wheels and the rectangular box will be added.

$$J_{total} = J_{box} + 2 \cdot J_{front} + 2 \cdot J_{rear} = 56{,}2 + 2 \cdot 17{,}5 + 2 \cdot 13{,}3 = 117{,}8[kgm^2]$$

Furthermore the stiffness for the total car has to be calculated. This will be done using equation 2.11. For this calculation the front (C_{front}) and the rear tire-stiffness (C_{rear}) have been chosen to be equal: 140 N/mm.

$$K_T = a^2 \cdot (2 \cdot C_{front}) + b^2 \cdot (2 \cdot C_{rear}) \qquad \text{(Equation 2.11)}$$

$$K_T = 0{,}88^2 \cdot (2 \cdot 1{,}4 \cdot 10^5) + 0{,}72^2 \cdot (2 \cdot 1{,}4 \cdot 10^5) = 3{,}62 \cdot 10^5 [N/m]$$

The terms a and b respectively represent the distance between the centre of gravity and the front and the rear axle. The square root of the stiffness divided by the total moment of inertia, as described in equation 2.12, will determine the yaw frequency.

$$\omega_{yaw} = \sqrt{\frac{K_T}{J_{total}}} \qquad \text{(Equation 2.12)}$$

$$\omega_{yaw} = \sqrt{\frac{3{,}62 \cdot 10^5}{117{,}8}} = 55{,}4[rad/s]$$

$$f_{yaw} = \frac{\omega_{yaw}}{2\pi} = 8{,}8[Hz]$$

The yaw frequency for the car has been calculated at 8,8 Hz. Comparing this result with the yaw frequency for a normal street car, between 4 and 7 Hz, the handling of the Formula Student car will be better, which is desirable regarding the tight tracks that have to be driven. A high yaw frequency will mean that the switch between a left and a right bend can be made very quickly.

The desired behaviour has been described in this chapter. This behaviour has to be established for the Formula Student car. Therefore the chassis, the suspension and the drive train have to be designed. First of all the chassis design will be described in chapter 3.

Chapter 3: Chassis

For a car that has to drive at a high speed, like a race car, the aerodynamics is very important to reduce the air resistance. Regarding the relatively low speed for the Formula Student car, although it is a race car, the aerodynamics has been neglected during the design of the chassis, because the results from cars in 2003 indicate that the maximum speed on the track during the different events won't exceed 110 km/h. This is the result of the huge amount of bends and short straights. Therefore aerodynamics will not be important for the Formula Student car, because aerodynamics will only start to work at speeds above 100 km/h. For the design of a race car wings are used to create down force for the car, but these wings are, just like the aerodynamics, only important at high velocity. Therefore in the design of the chassis these wings won't be used.
For the Formula Student competition the chassis has to meet a lot of rules, regarding dimensions and safety. These requirements will be described in paragraph 3.1.

3.1: Chassis requirements

There are a lot of rules to cope with considering the chassis. Most of these rules have been made regarding the safety for the driver during the different events. First of all the function of the chassis has to be regarded. Therefore some different questions have been formulated:

- What is the function of the chassis in reference to the whole car?
- Which persons have to be able to sit in the car?
- Which safety aspects have to been taken care of?
- Which materials will be allowed according the requirements?
- What has to be the comfort during driving?

Answering these questions will determine the shape and the design of the chassis.

First of all it is important to regard the function of the chassis when over viewing the whole car. For the shape and the layout of the chassis it is important that the tire forces will be guided nicely through the suspension into the chassis. Therefore the connection points from the suspension with the chassis will be the most important points in the chassis, so that these points have to be designed very stiff and strong. Furthermore the chassis has to provide the necessary space for the driver and the other parts that have to be mounted and placed into the chassis.

For the Formula Student contest four different drivers have to drive the car during the different tests. For the construction of the car it will be useful to select four drivers with a comparable length. This means that the position of the steering wheel and the foot pedals won't have to be adjusted during a race, when a driver change has to be made. According the requirements a 95th percentile male has to be able to sit in the car. The space in the front part of the chassis will be as big as required for that person. A 95th percentile male represents a driver with a length of 1,86 m. During the races a helmet has to be worn, which implies that the overall length of the tallest driver will be 1,90 m.

For the Formula Student contest the safety aspect of the car is very important. The car has to sustain different worst cases. The driver has to be protected for a roll over of the car, a front impact and a side impact. That is why a roll hoop is obligatory to provide a head clearance for the driver during a roll over. This hoop has to be supported either to the front or the back of the hoop using braces. For the front impact a space of 150 mm before the feet has to be constructed containing a crash box, which has to absorb the energy of a frontal impact. The side impact protection has to absorb the energy of a side impact. A last safety aspect is that the driver should be able to be out of the car within five seconds. Therefore the steering wheel has to be easily removable to create space for the legs of the driver, when getting out of the car.

According the rules for the Formula Student car the main hoop and its braces has to be constructed using a steel tube. Furthermore the other parts of the chassis can be constructed either out of steel or other materials. Therefore the factor E*I of the used materials will be normative.

The comfort of the driver won't be very important. The time that the drivers have to be in the car is not that long that comfort will be necessary, because most tests will take not more than one minute. Only the endurance test will take up to half an hour for two drivers. This means that one driver has to sit for about 15 minutes in the car at racing speed. During this period the driver has to be able to drive the car as fast as possible, which means that the driver must not be hampered during those 15 minutes to achieve the best performance.

During the design of the chassis the mounting of different parts in and onto the chassis will be taken into account, so that those parts will fit into the chassis in a later phase of the overall design. These connections have to be made onto the stiffest and strongest points in the chassis. Therefore the used structure used and material for the chassis will be described to determine the stiffest and strongest points in the chassis.

3.2: Chassis structure

Normally for a race car a tube frame is used to create a strong and stiff structure for the protection of the driver. Around this structure a well shaped and light plate work will be constructed to improve the aesthetics of the car. A disadvantage of a welded tube frame is that the shape is not steady due to the heat and the deformations during welding.
A different technique is to make a monocoque, which will be made using light and strong materials, like carbon fibre. For a first year's team the use of a monocoque is very difficult, because the shape has to be directly right using this technique. When the layout of the suspension is proved to be right a monocoque can be used. When during tests something has been detected that has been done wrong, only small adjustments can be made, otherwise a new monocoque has to be built. Another disadvantage is the high costs for a monocoque, because of the high production and machine costs.

An important requirement for the Formula Student car is that costs for one car should be below € 21.000, regarding the fact that the car has to able to be built in a series of 1000. Therefore it has to be low cost and easy to build. To keep the production process easy and cheap the choice has been made to make the chassis of aluminium sandwich panels. The use of these panels will be explained in paragraph 3.2.1.

3.2.1: Laminated sandwich structure

The choice has been made to build the chassis of aluminium sandwich panels. This type of construction is mainly used in air and space vehicles. The advantage of a sandwich construction is a low weight and a high strength/stiffness for the chassis. An aluminium sandwich structure will be made with three different elements.

1) two rigid, thin, high strength skins
2) one thick, low density core
3) an adhesive attachment, which forces the core and facings to act as a continuous structure.

The skins of a sandwich panel act similarly to the flanges of an I-beam by taking the bending loads; one skin in compression and the other in tension. The core resists the shear loads and increases the bending stiffness of the structure by spreading the skins apart. The stiffness of the sandwich construction is adjustable in two ways, changing the thickness of the outer skins or changing the thickness of the core.

The advantage of the construction method is that the cost will be relatively low. The shape of the panels will be flat so that the production of the panels will be very easy, just gluing the different layers onto each other, using epoxy glue. The materials that will be needed are aluminium and a core material. For a light and stiff sandwich structure foam between two aluminium skins is very suitable and the production and the material costs will be low. For the core the choice has been made to use polyurethane. The only thing that has to be determined is the thickness of the different layers of the sandwich panels. The thickness of the core will determine the strength of the sandwich panel. To make a choice for the thickness of the core figure 3.1 has to be considered.

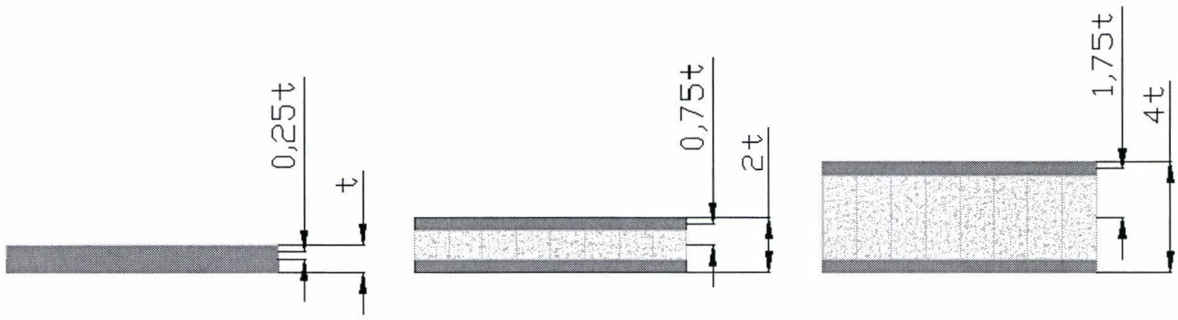


Figure 3.1: Core thickness influence

The mass inertia of the sandwich panel depends on the outer fibre distance, the distance between the centre lines of the aluminium skins and the centre line of the sandwich panel. The mass inertia coheres with the square of the outer fibre. For the total mass inertia only the aluminium skins has been considered, because the foam won't affect the total mass inertia, while the foam has been placed on the centre line of the sandwich panel. The mass inertia can be calculated with equation 3.1, using the thickness and the width of the aluminium skins.

$$I_{sandwich} = 2 \cdot t_{alu} \cdot w_{alu} \cdot e^2 \quad \text{(Equation 3.1)}$$

For the comparison between the three configurations in figure 3.1 the t_{alu} and w_{alu} won't be changed, only the outer fibre distance. The comparison between different core thicknesses in figure 3.1, considering the moment of inertia, has been made in table 3.1.

	t	2t	4t
Outer fibre (e)	$0{,}25 \cdot t$	$0{,}75 \cdot t$	$1{,}75 \cdot t$
e^2	$0{,}0625 \cdot t^2$	$0{,}5625 \cdot t^2$	$3{,}0625 \cdot t^2$
Relative values	1	9	49

Table 3.1: Comparison core thicknesses

From table 3.1 the conclusion can be made that the thicker the core material, the higher the mass inertia of the sandwich panel. The mass won't rise significant, because the foam has a density of 35 kg/m^3 (aluminium 2700 kg/m^3). For the sandwich structure this means that only 1,3% of the total weight will be foam and the rest is aluminium, considering the case with the total thickness of 2t (the foam volume is equal to the aluminium volume). For the stiffness of the total chassis the thickness of the core should be as high as possible, but a thick sandwich panel will result in a bulky and unattractive look for the chassis. Therefore the choice has been made to create a sandwich panel with aluminium skins with a thickness of 1 mm and a core thickness of 10 mm. This will result in a wall thickness of 12 mm. In places where less stiffness is necessary the thickness of the aluminium skins will be reduced till 0,5 mm. This has to be determined later on using Finite Element Analysis (FEA).
The aluminium sandwich panels have to be connected. Therefore aluminium frames will be used. The layout and the function of these frames will be explained in paragraph 3.2.2.

3.2.2: Frame structure

The sandwich panels have to be connected to each other to create the shape of the chassis. Therefore aluminium frames will be used. These aluminium frames has to be designed regarding a couple of requirements.
First of all the connection between the frames and the sandwich panels has to be easy and light weighted, because the use of fasteners means extra weight.
Second the frames have to represent the stiffest points of the total chassis, because the sandwich panels will not be suitable to connect the suspension onto. The forces from the wheels through the suspension into the chassis will be the biggest forces on the chassis. Therefore a solid connection between the suspension and the chassis will be very important, considering the total car stiffness. When connecting the suspension onto the sandwich panels only, the forces will cause a deformation of these panels and the forces won't be guided properly into the chassis.
At last the make ability of the frames has to be easy, because it has to be suitable for the construction of 1000 cars.

The stiffness of the frames will be guaranteed by making the frames in one piece. This will result in one rigid part to avoid internal tolerances in the frames and to reduce the weight, because the use of fasteners won't be necessary. Also the strength and stiffness will be ensured, because fewer fasteners mean less fatigue. These frames will be milled and sawed from an aluminium plate with a thickness of 25 mm. With finite element analysis and the position of the driver the shape of the aluminium frames will be determined. These analyses will be useful to minimize the weight of the frames, to reduce the overall mass of the car. Removing excessive material won't affect the total stiffness of the frames.

The shape of the frames will be determined, using the shape and the position of the driver. The position of the driver will be described in paragraph 3.2.3.

3.2.3: Driver position

The dimensions of the driver have been determined by the rules for the Formula Student competition. A 95th percentile male has to be able to drive the car, which will be a person that has a length of 1,88 m. The position of the driver has been determined by testing the angle of the back of the driver and the position of the arms and legs. The most important dimensions for the driver has been drawn in figure 3.2a and the position for the driver to overview the road and to steer comfortably has been drawn in figure 3.2b.

a) b)

Figure 3.2: Driver a) 95th percentile male measurements b) driver position

Figure 3.2b shows a simplified 3D model for a 95th percentile male, which has to be able to sit and drive in the car. In this position the driver should be able to drive 15 minutes without any physical problems. For the shape of the chassis the length of the legs and the position of the elbows will be important. The elbows will be the widest point of the driver to determine the width of the chassis. This position of the driver will determine the position for the frames.

3.2.4: Frame positions

Now the shape and the position of the driver are known the length towards the front of the car can be determined using the length of the driver's legs. These have to be placed upon the gas and the brake pedal and behind this set of pedals an empty space of 150 mm has to be created for the front impact protection. In this area no technical features are allowed, only some materials to improve the front impact protection.

Considering the fact that the driver has to be separated completely from the engine using a firewall, the seat frame has been designed. This seat frame will also be constructed using the sandwich technique. For the shape of the seat frame the position of the driver is very

important, because the back of the driver has to be supported by the seat frame; the driver will be attached to the seat frame using a 5 or 6 point restraint harness.
In the main frame a dashboard will be integrated and the steering axle has to be supported in this frame. Therefore the position of the main frame depends on the driver's hands. These hands will determine the position of the steering wheel and the steering axle.

Using these requirements the frame positions, regarding the position of the driver and the engine, have been determined. These positions are shown in figure 3.3.

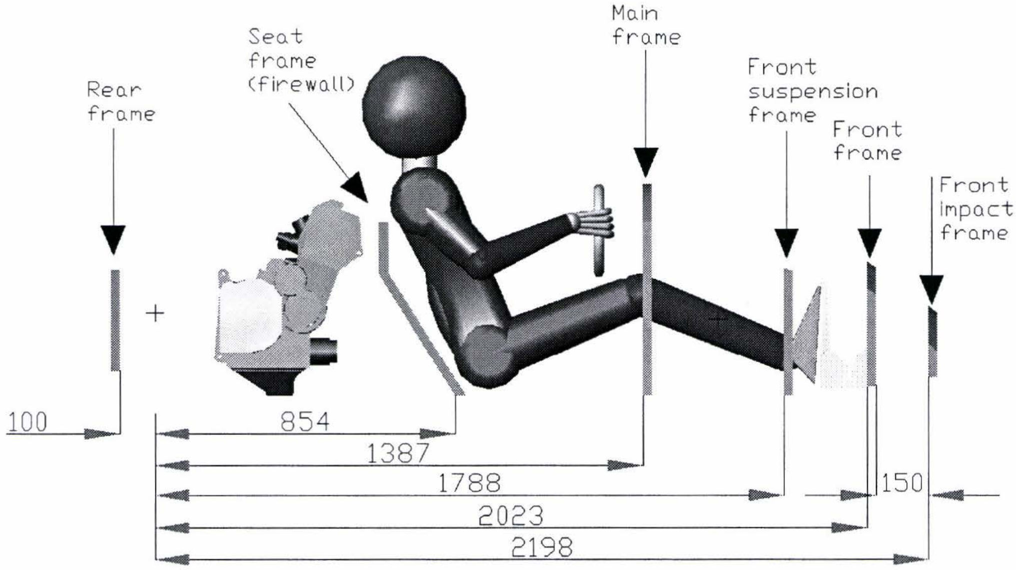


Figure 3.3: Frame positions

For the rear frame, which will be 100 mm behind the rear axle, this position has been determined using the dimensions of the engine and the chain wheel for the transmission towards the rear wheels. The front frame and the front impact frame have been placed with a mutual distance of 150 mm for the required crush zone regarding the necessary space for the pedals. The frame around the feet of the driver is placed at that position for the desired symmetry for the front suspension, regarding the fact that the centre of the front tire will be right between that frame and the main frame. This frame is called the front suspension frame.

Now the different frame positions have been determined their shapes will be determined according the function and the position regarding the driver.

3.2.5: Frames design

The shape of the frames will be determined by the function of those frames. The seat frame has to be a fire wall towards the engine, which will result in a closed frame. This seat frame will be made using the sandwich structure. Two skins of 1 mm aluminium and a polyurethane core with a thickness of 23 mm will form the 25 mm thick seat frame. The widest point of the chassis will be at the seat frame, which means that the seat frame is relatively big. Using a sandwich structure the mass will be lower, than using a solid aluminium frame. Sandwich will be sufficient, because the suspension won't be attached to the seat frame. The bend in the frame has been made to provide more space in the rear part of the chassis for the engine. The seat frame is shown in figure 3.4.

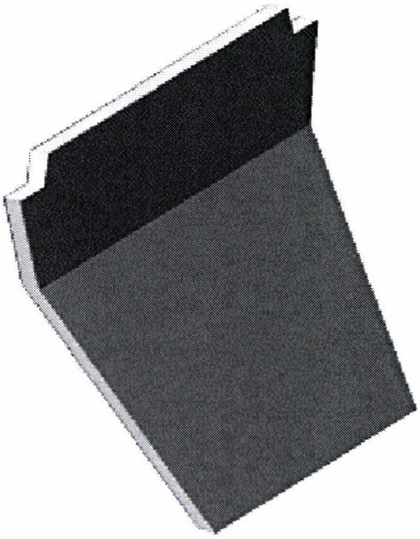


Figure 3.4: Seat frame

For the main frame this will be different because the driver has to sit right through that frame. The shape of the main frame is determined by the position of the driver's knees. The position of the knees determines the height of the dashboard and the width of the main frame. The lower part of this frame will be a frame structure from solid aluminium □ 25x25 mm bar aluminium. This main frame, the front frame and the front impact frame will have approximately the same shape, only the size is different. These frames are shown in figure 3.5a, 3.5b en 3.5c.

a) b) c)

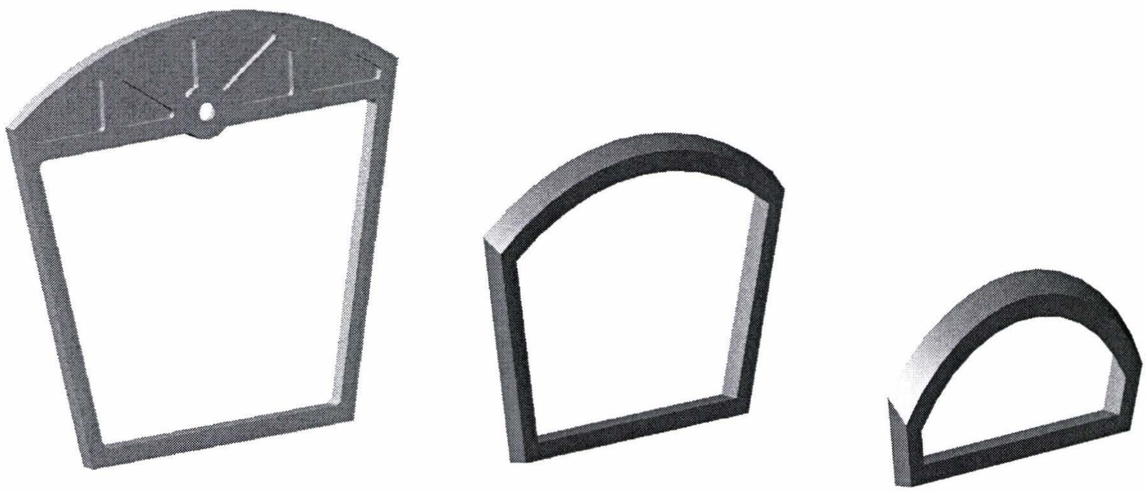


Figure 3.5: Curved frames a) main frame b) front frame c) front impact frame

The curve in the frames has been made to improve the aesthetics for the chassis. Over these curved frames a nicely shaped hood will be mounted. For the front suspension frame and the rear frame a different principle is used. The front suspension frame consists three square beams, which are made out of one piece to lose the fasteners and to create extra stiffness for the chassis. The rear frame will be a solid plate of 25 mm thick aluminium with a milled slot for the chain wheel and some triangular shapes, so that a frame work will remain consisting square beams of 25x25 mm. This will reduce the weight of the frame, but the holes won't affect the stiffness of the frame very much. The rear frame will be used to connect the rear suspension onto, so this frame has to guide the forces from the suspension into the chassis. The shape of the front suspension frame and the rear frame are shown in figure 3.6a and 3.6b.

a) b)

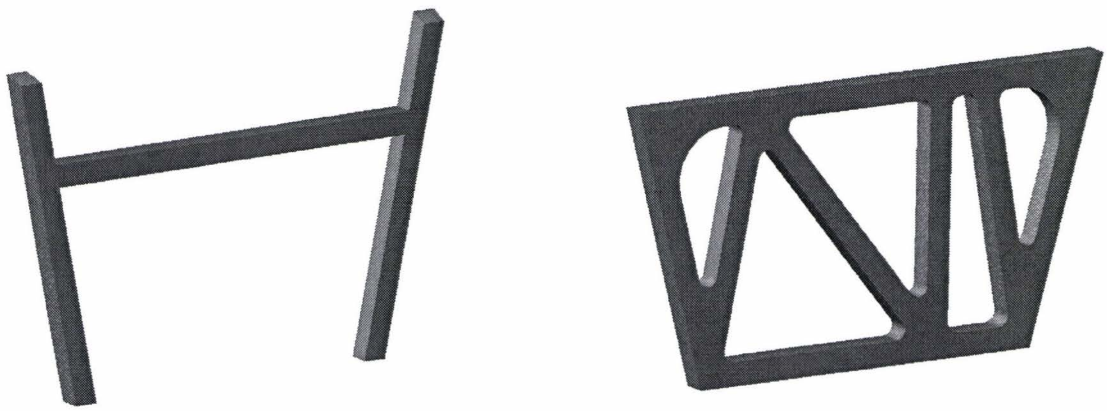


Figure 3.6: Straight frames a) front suspension frame b) rear frame

The main frame, the rear frame and the front suspension frame will be used for the connection with the suspension. The stresses in these frames have been determined using point forces, which are shown in appendix F.

The two side panels and the bottom panel have to be connected with the frames. This connection will be made using a small slot in the frames to fixate the inner skin of the sandwich panels. The outer skin of the sandwich panel will be used as the base for the chassis (figure 3.7, step 1). These skins will run from the front to the rear of the chassis made in one piece. The polyurethane core will be glued onto the outer skin at the proper position, regarding the final frame position (step 2). From the inside the inner skin of the sandwich panels will be glued upon the polyurethane to complete the sandwich construction (step 3). After that the frames will be placed so that the inner skin will fit into the slots of the frames and then the frames will be connected onto the outer skins of the sandwich panels (step 4 and 5). From the other side of the frame the foam and the inner skin will be placed for the following chassis section (step 6 and 7). This will result in a phased construction for the chassis. The connection as described is shown in figure 3.7.

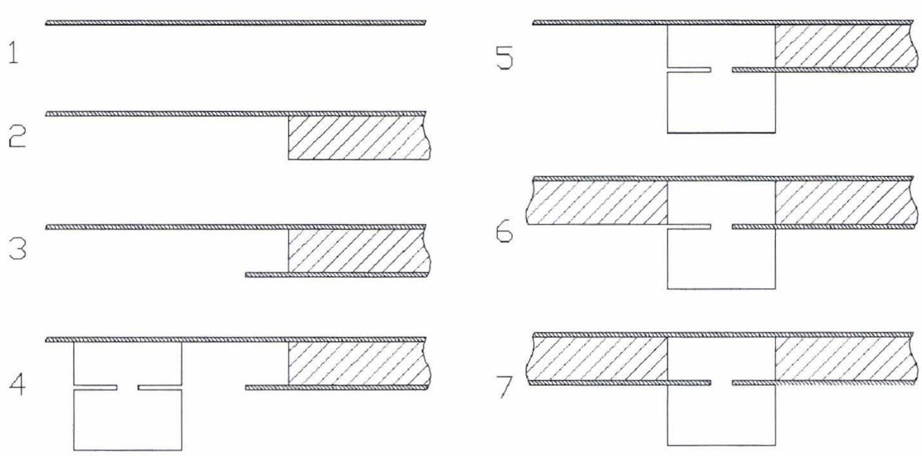


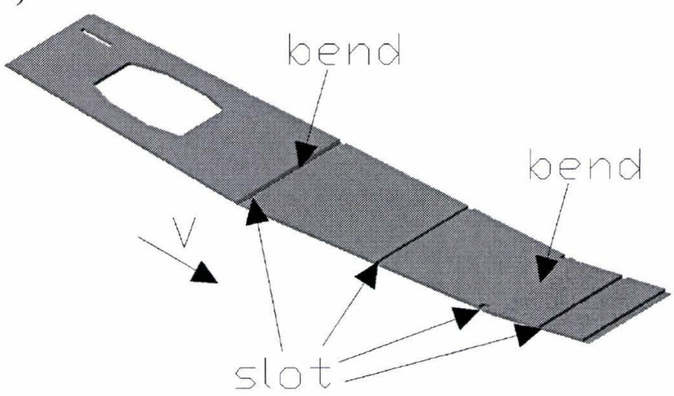
Figure 3.7: Connection between sandwich panel and frame

This method will be used for both the side panels and the bottom panel. The sandwich panels and the frames will form the chassis. Now the shape of the frames and the connection between the frames and the sandwich panels is known, the shape of the side panels and the bottom panel will be determined.

3.2.6: Sandwich panels design

The choice has been made to create an easy and straight shape of the chassis. The make ability of the sandwich panels is very good, because they are all flat. The bottom sandwich panel tapers towards the front and the rear of the car, so that the widest point of the chassis will be at the height of the seat frame. Towards the front and the rear this panel contains ascending slopes. Therefore the bottom panel has to be bended at two places. The shape of this bottom panel is shown in figure 3.8a. In the front view of the car the side panels taper towards the bottom panel, because the widest point of the chassis will be at the height of the elbows of the driver. The maximum height of the side panel is at the position of the driver to create as much protection as possible. Towards the front and the rear of the chassis these panels descend. The shape of one side panel, the left one, is shown in figure 3.8b.

a)



b)

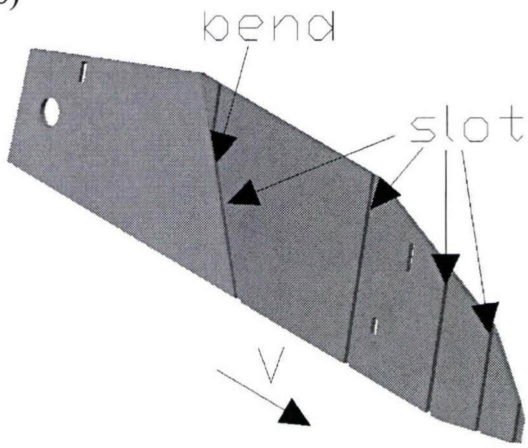


Figure 3.8: Sandwich panels a) bottom panel b) side panel

The slots in the bottom and the side panel have been made to provide the necessary space for the frames, which will be connected onto the outer skin of the sandwich, as described in paragraph 3.2.5. In the bottom panel a hole has been made to provide the space needed to place the engine as low as possible, considering the ground clearance of 50 mm.

Now the main shape of the chassis has been determined regarding the position of the driver and the engine reinforcements will be added to the chassis to provide extra stiffness and safety. These reinforcements will be described in paragraph 3.3.

3.3: Chassis reinforcements

To improve the stiffness of the chassis some extra parts will be used in the chassis. To use the extra material as good as possible these parts will be designed with a double function. According the rules for the Formula Student competition the safety aspect is very important.

Therefore a couple of rules have been set regarding the safety of the driver. For a frontal collision the car should have a front impact protection.

3.3.1: Front impact protection

For the Formula Student car a front impact protection is very important. Therefore a front impact box will be used to absorb the energy during a frontal collision with another car. A collision is possible during the endurance event, because multiple cars will be simultaneously on the track. For the amount of energy that has to be dissipated the assumption has been made that an other car with a mass of 300 kg and a velocity of 20 m/s will collide. The energy, which has to be dissipated during a frontal impact, will be calculated according equation 3.2.

$$E_{impact} = \frac{1}{2} \cdot m_{car} \cdot V_{car}^{\ 2} \qquad \text{(Equation 3.2)}$$

$$E_{impact} = \frac{1}{2} \cdot 300 \cdot 20^2 = 60[kJ]$$

To dissipate this amount of energy a box will be constructed in the empty space between the front frame and the front impact frame. Aluminium honeycomb is very useful for that application, because it will be able to dissipate a lot of energy. A second reason for using this material is that the honeycomb is very light and stiff as well. An example for an aluminium honeycomb sandwich panel is shown in figure 3.9.

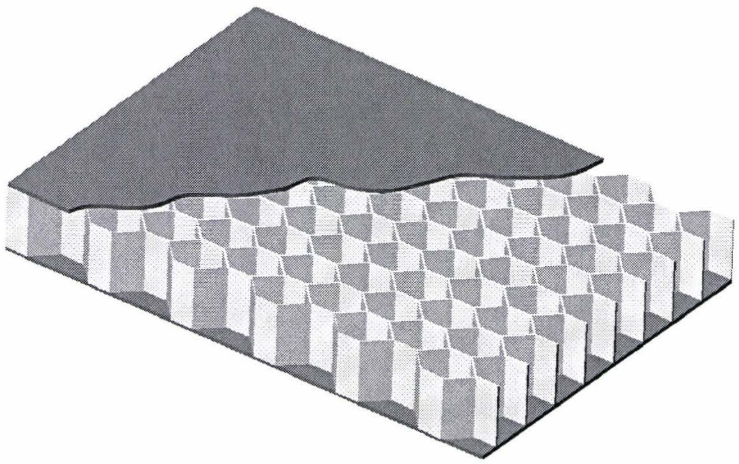


Figure 3.9: Aluminium honeycomb sandwich panel

For the front impact box multiple layers of the sandwich panel shown in figure 3.9 will be used. These layers will be glued onto each other and they will be placed with the honeycomb structure pointed towards the front of the car. The impact box will fill the whole 150 mm of the crush zone and will have a variable width and height, so that the space between the front frame and the front impact frame will be filled completely. Then the minimal obliged frontal area with a width of 200 mm and a height of 100 mm will be achieved. The total weight of this box will be 280 g considering a density of 60 kg/m^3. This box will be placed at the position shown in figure 3.10

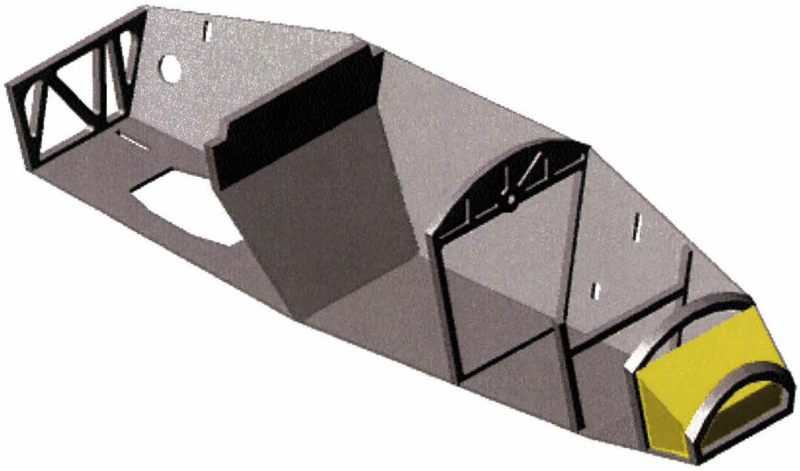


Figure 3.10: Front impact protection box

Just as for the front impact protection there are requirements for a side impact. There is also a possibility that a car will drive into the side of the car, between the front and the rear wheel. Therefore a side impact protection will be integrated in the chassis.

3.3.2: Side impact protection

The amount of energy that has to be dissipated during a side impact will be the same as for the front impact energy: 60 kJ for a car of 300 kg with a speed of 20 m/s. In appendix G a side impact analysis has been made, using multiple point forces on the side panel of the chassis. Therefore two side protection boxes will be made between the main frame and the seat frame. The height of these boxes will be 200 mm from the upper skin of the bottom panel and will have a width of 55 mm at the seat frame. This box will have a width of 43 mm at the side of the main frame, due to the tapered shape of the chassis. The width will be less towards the bottom panel, because the side panels will be tapered. The inner side of the side impact boxes will be straight. The shape of those side impact boxes is shown in figure 3.11.

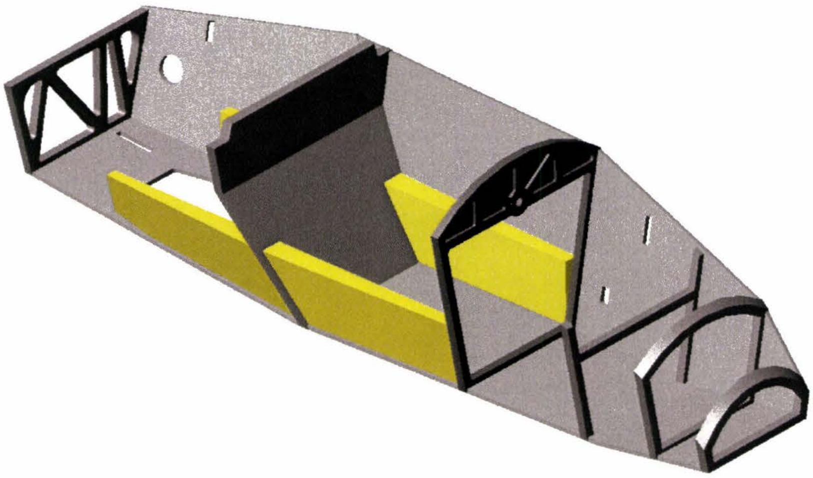


Figure 3.11: Side impact protection boxes

The side impact boxes will be made of the same sandwich material that will be used for the bottom and the side panel. The will continue behind the seat frame. This will be done to provide a higher stiffness for the side panels at the back of the car. Onto the rear side panels the front members of the rear suspension will be connected. The side impact boxes will provide extra stiffness for the connection points from the suspension with the chassis. Using extra boxes on each side of the car the stiffness for the rear suspension will be guaranteed. To avoid a complete open box structure for the rear side of the chassis a tube will be constructed over the engine from the left to the right side panel.

The shape of the chassis is until now an open box, which is not preferable regarding the torsion stiffness of the chassis as a whole. To improve the torsion stiffness some extra parts will be mounted into the chassis.

3.3.3: Chassis torsion stiffness

The torsion stiffness of the chassis is relatively low, because the chassis is, until now, an open box structure. To raise the stiffness the open box has to be closed. Not for all parts of the car will that be possible because the driver has to be able to get in the car and at the back of the car various parts and assemblies such as the engine, have to be mounted in the chassis. The front part of the car however will be closed. Above the legs of the driver a sloped sandwich panel will be used to close the front part of the open box to gain torsion stiffness. This laminated sandwich structure will be just like the side panels: a panel with a thickness of 12 mm (1-10-1). For the mounting of different parts above the legs of the driver: the steering system, the suspension and the anti-roll mechanism a second sloped sandwich panel will be placed under the first one. The distance between these two panels has been chosen to provide enough space for the front suspension parts. These panels will provide extra stiffness for the connection points of the suspension with the chassis too. These two panels are shown in figure 3.12.

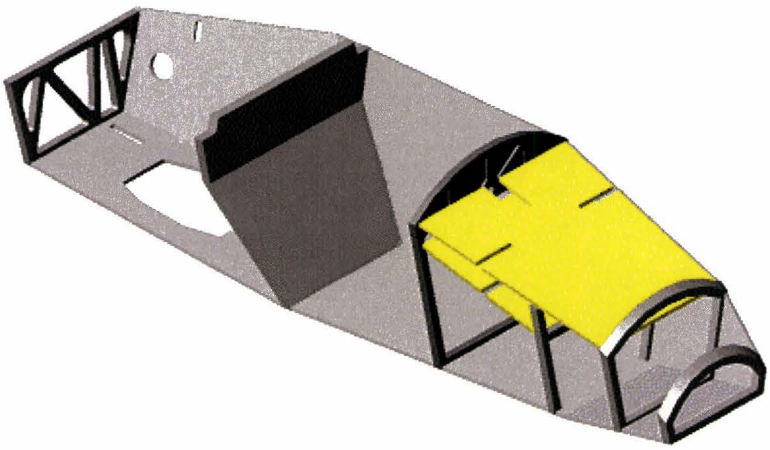


Figure 3.12: Sloped panels

The slots in the panels have been made for the front suspension and for the steering mechanism. For more torsion stiffness for the open part of the chassis, in the section where the driver will be seated, two tunnels above the driver's elbows will be placed. These tunnels will be mounted onto the side panels between the main frame and the seat frame. To create a double function for those tunnels the tunnels will be kept hollow to guide wires and cables from the front of the car to the rear and vice versa. The position of those tunnels is shown in figure 3.13.

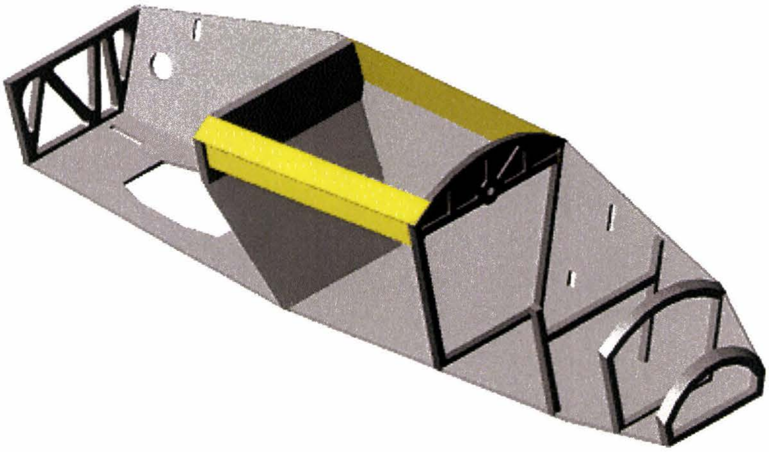


Figure 3.13: Torsion tunnels

These tunnels have been made of aluminium. The outer skin of the side sandwich panel will be extended towards the inner skin of the sandwich panel, to create the tunnel. The wires and cables will be guided through hollow tubes, which will be placed in those tunnels. In the main frame and the seat frame a hole has to be made to provide space for the cables and wires. After placing the tubes the tunnel will be filled with polyurethane foam to fill the empty spaces, which increases the stiffness and the strength of the tunnels. These tunnels will increase the torsion stiffness for the chassis and will provide extra safety for the driver. Now the shape of the chassis has been presented, the stresses in the chassis during driving can be determined. This has been done in appendix H.

To provide more safety for the driver a roll hoop has to be integrated in the chassis, which has to provide the safety of the driver during a roll over. The shape and the construction of the roll hoop will be explained in paragraph 3.3.4.

3.3.4: Roll hoop

According the requirements a roll hoop has to be constructed within the chassis, so that the driver of the Formula Student car will be protected for a roll over. This roll hoop has to be constructed using steel pipe with a diameter of 25,4 mm (1 inch) and a minimal wall thickness of 2,1 mm. According the requirements the roll hoop has to provide a head clearance of 51 mm (2 inches). This will be measured connecting the highest point of the main frame, with the integrated front hoop (the curved shape), and the top of the roll hoop. The line between these two points has to provide this head clearance.
The roll hoop will be connected onto the bottom panel and the seat frame. These connections will be made using bolts and nuts. Therefore aluminium inserts in the seat frame and steel inserts in the roll hoop will be used to make a rigid connection. This connection between the roll hoop and the seat frame is shown in figure 3.14.

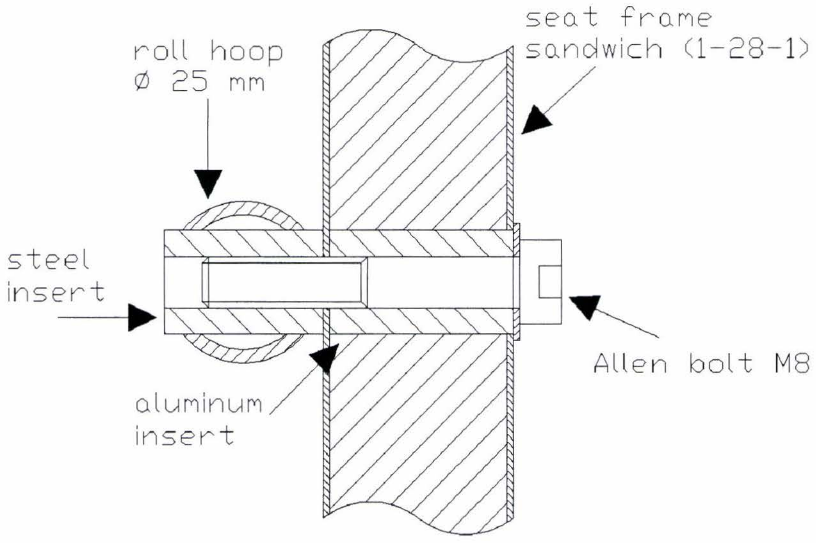


Figure 3.14: Connection between roll hoop and seat frame

To support the roll hoop two braces will be used. These braces will be made of steel pipe with a diameter of 25,4 mm (1 inch) and a wall thickness of 1,65 mm, which is the minimum required thickness. These braces will be welded to the roll hoop 75 mm below the top of the hoop. This is according the rules, which require a maximum distance of 160 mm. The braces and the roll hoop will be removable together for keeping the possibility to remove the engine, when a reparation or adjustment for the engine will be necessary. Therefore bolts and nuts will be used to connect the braces to the rear frame. For these connections steel inserts in the braces will be used to obtain a rigid connection. The shape of the roll hoop and the braces is shown in figure 3.15.

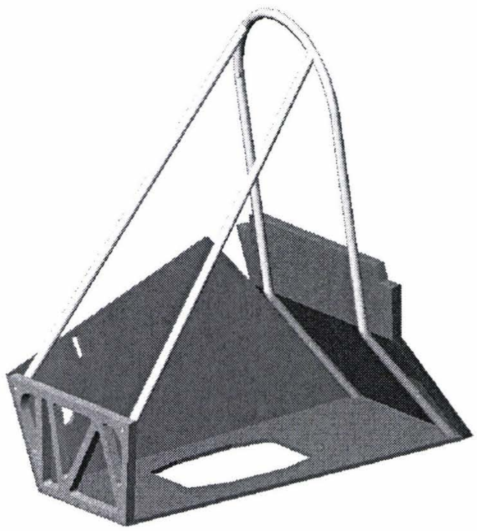


Figure 3.15: Main roll hoop and braces

The height of the roll hoop has been determined using the fact that the position of the roll hoop has been chosen to be right behind the seat frame. Therefore the roll hoop will make exact the same angle as the seat frame. Where the bend has been made in the seat frame, the roll hoop will make that bend too. The required head clearance has been provided, which is shown in appendix J. The bracing for the front hoop (the upper part of the main frame) has been integrated in the chassis, because the upper sloped panel above the legs of the driver will be used as bracing. This demonstrates the double function of this panel.

Now all the parts of the chassis have been designed, the shape and the layout of the total chassis can be determined. Assembling all these parts the chassis shown in figure 3.16 will be formed.

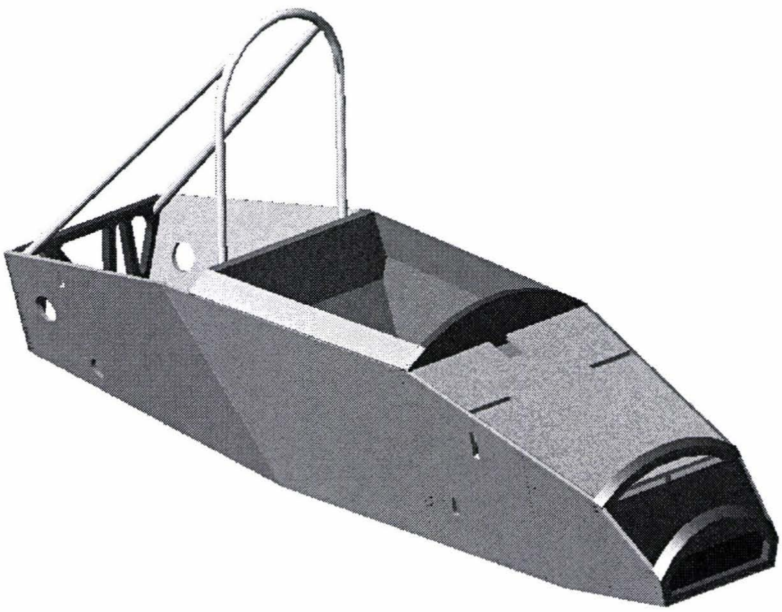


Figure 3.16: Total chassis

The shape of the chassis, shown in figure 3.16, is very angular without any nice curvy lines, which are common in race cars. Also some parts of the front suspension will be mounted upon the upper sloped panel, so that these parts will be vulnerable. Therefore a nice shaped light weighted hood will be attached to the front part of the car to cover these parts and to add some aesthetics to the car. This hood will be shaped according the form of the upper part of the main-, the front- and the front impact frame.

The chassis shown in figure 3.16 is the base structure to which all other parts will be attached. The main parts that have to be integrated into the chassis will be the engine and the differential. The design for the engine suspension will be described in paragraph 3.4.

3.4: Engine suspension

The most important part that has to be placed into the chassis will be the engine. The Suzuki GSX-R 600 engine will be used for the Formula Student car, because it is in his class one of the most powerful engines, below the 610 cc. Further specifications for this engine will be given in chapter 6. This engine contains eight suspension points, which means that a couple of hard points can be used for the mounting into the chassis. These points are shown in figure 3.17a and 3.17b.

a) b)

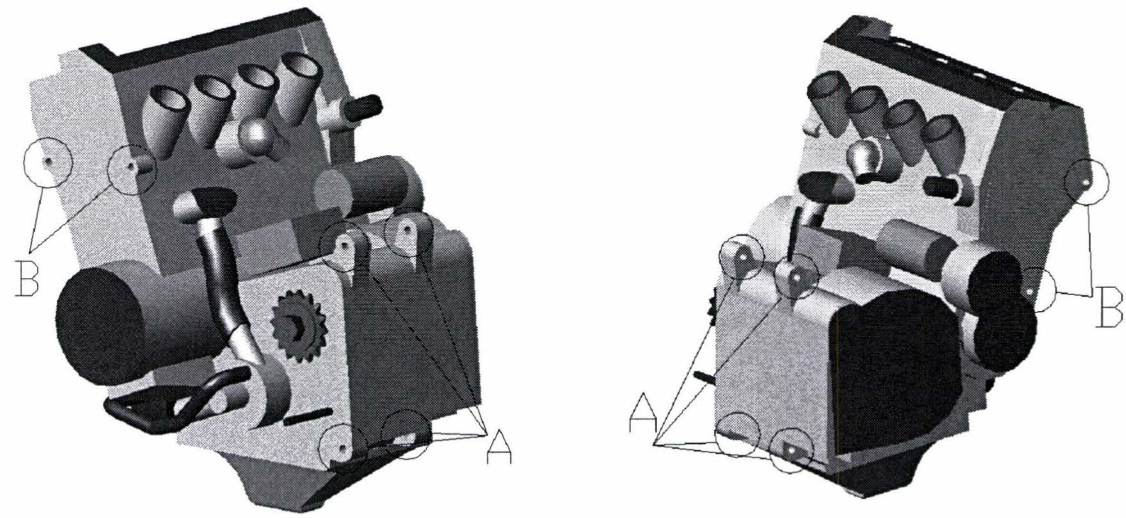


Figure 3.17: CAD layout Suzuki GSX-R 600 a) left side b) right side

For the placement of the engine in the chassis the hard points in figure 3.17a and 3.17b will be necessary. These points are situated at the side of the gearbox (A) and at front side of the engine where the cylinders have been placed (B). The four hard points at the backside of the engine will be suited to use two plates towards the rear frame and the bottom panel. One plate will be mounted at the left side of the engine using the two left hard points. The second plate will be used to connect the two hard points at the right side of the engine to the chassis. The only problem is that the hard points at both sides of the engine don't lie in the same plane. Therefore bushes will be used at the engine side to fill the difference of 17 mm on either side. These bushes will have a length of 12 mm so that one side of the plate has to countersink onto the hard points of the engine. This will improve the stiffness of the connection because the length of the bushes will be smaller. Allen bolts M10 will be used at the side of the engine, because the engine suspension points have M10 thread. Towards the rear frame Allen bolts M8 will be used to connect the aluminium plates to the frame. These two plates are shown in figure 3.18.

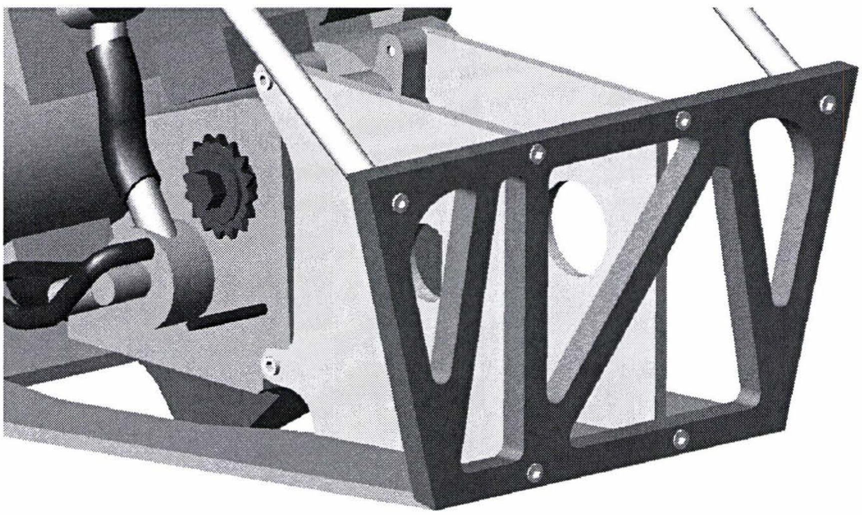


Figure 3.18: Aluminium plates for engine suspension

The plates will be made of aluminium to keep the weight as low as possible in comparison with steel. The shape has to be milled of a solid plate. The holes in the plates have been made for the differential, the drive shafts and the bearings, which will be mounted in these plates. The bearings will be mounted in the plates so that the forces on the bearings will go right into these plates and into the engine.

At the left side of the engine the driven shaft from the gearbox is placed. Therefore the chain wheel for the transmission will be mounted on the left side of the car, which means that the biggest forces will be applied to the left aluminium plate. Therefore the left plate will have a thickness of 17 mm and the right plate 12 mm. The thickness of the two plates various locally. Around the hole for the bearings and at the connection points with the engine and the chassis, the plate has to have the full thickness, but the thickness can be less in the other parts. This means that weight saving milling can be done to keep the weight of the two aluminium plates as low as possible.

The two aluminium plates will only fix the x- and z-movement of the engine. For the fixation of the lateral movement of the engine a third plate will be used, which will be mounted on the backside of the engine and will be connected to the bottom panel. The width of that plate will be the distance between the two first mentioned aluminium plates.

For the connection of the engine towards the front of the car, the seat frame and the roll hoop will be used. The engine contains four hard points to make this connection, on either side two. The position of those two points is different on each side, so that for both sides different methods will be used to connect the engine with the roll hoop. This is shown in figure 3.19a and 3.19b.

a) b)

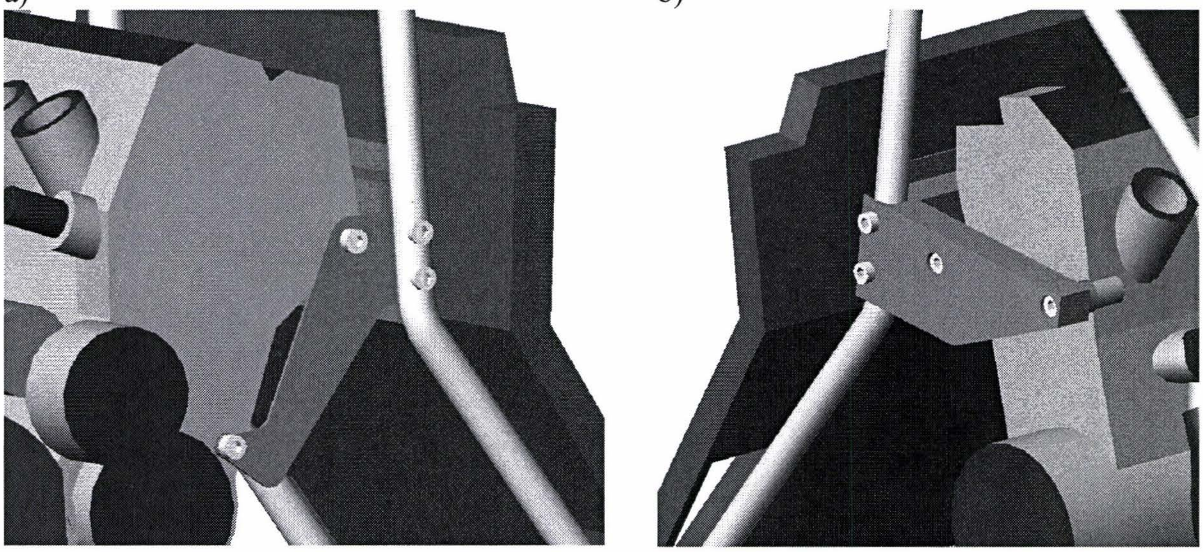


Figure 3.19: Connection engine to roll hoop a) right side b) left side

For the right side of the engine (figure 3.19a) an aluminium plate will be used, which will be connected to the engine with two Allen bolts M10 and connected to the roll hoop using bolts and nuts through the roll hoop and the aluminium plate. Therefore a steel insert will be placed in the hoop. At the left side (figure 3.19b) a box profile will be used to connect the engine to the roll hoop. This box will be connected to the engine using two Allen bolts M10, which will be countersunk into the box using two bushes between the outer skins of the box. The box is 80x30x2 mm, which means that the two skins of the box will go on either side along the roll hoop (25,4 mm diameter). A hole will run through the hoop and the box for a connection with bolts and nuts. Therefore a steel insert will be placed in the hoop to create a rigid connection. The box has been used, because two plates only won't fixate the sixth degree of freedom, the lateral movement of the engine. Using a box the engine position will be fixed.

To avoid the rotation of the engine around the connection points with the roll hoop, which can cause a twist of the roll hoop, an omega profile will be mounted onto the seat frame between the two members of the roll hoop, the seat frame will be used to close the box. The flanges will be attached on the seat frame using five rivets at the under and upper side of the box. This box profile mounted onto the backside of the seat frame is shown in figure 3.20.

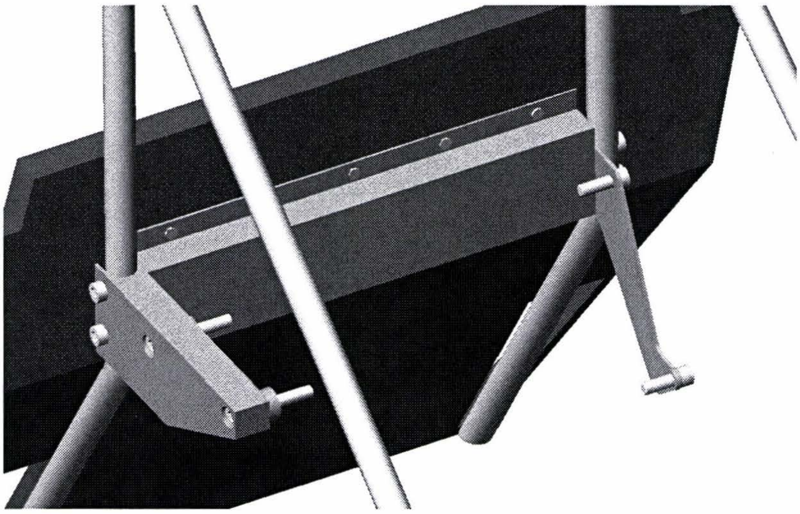


Figure 3.20: Box profile onto seat frame

Furthermore a jacking point has to be provided to the chassis. The function and the shape of the jacking point will be described in paragraph 3.5.

3.5: Jacking point

To keep the possibility to quickly move the car when the driveline is stuck, due to a failure, a jacking point has to be provided at the rear of the car. This jacking point provides a quick jack for lifting the car at the back side so that the rear wheels will be off the ground. Therefore the jacking point has to be capable to support the weight of the car. According the requirements the jacking point has to be oriented horizontally and perpendicular to the centreline of the car and it should be exposed around the lower 180 degrees of its circumference. The jacking point has to have a minimum length of 300 mm and has to be a tube, made of steel or aluminium, with an outer diameter of 25,4 mm (1 inch). The jacking point is shown in figure 3.21.

Figure 3.21: Jacking point

In figure 3.21 the position of the jacking point is determined, namely underneath the rear frame. The tube has been made of aluminium with a wall thickness of 2,1 mm. The connection of the aluminium tube with the rear frame will be made using two aluminium inserts in the tube, to create a solid connection with two Allen bolts M8. Therefore thread will be tapped in the lower member of the rear frame. This position of the jacking point provides a ground clearance of 95 mm at the back of the car.

Chapter 4: Suspension principles

One of the main and most complicated parts of the Formula Student car is the suspension. The front and the rear suspension determine the behaviour of the car during the three most important actions: accelerating, braking and cornering. The behaviour of the car, and especially the behaviour of the chassis with the driver and the other parts mounted in the chassis, is important for the feeling of the car by the driver. A car can be driven to the limit when it responds exactly to the input of the driver. Therefore the suspension of the car is very important, because this part of the car connects the road to the chassis. For the handling of the car it's important that the suspension will be designed well, so that the car responds to the input from the driver. For the different races it's important to have the possibility to adjust the suspension. The set up of the car has to be changeable for the different tests.
The suspension of the car is a very important part. The suspension has to transmit the input in the steering wheel onto the tires and the suspension has to transmit the input of the road surface on the tires into the chassis. Therefore the design of the suspension is essential for the handling of the car. For the design of the front and rear suspension several parameters will be important like camber, caster, KPI and the roll centres. These different terms will be explained in this chapter.

4.1: Suspension design parameters

The movement of the chassis in bends and on the straight ends depends on the position of the roll centres and the dive centre. The two roll centres determine the roll axis around which the chassis with the driver and the engine will rotate. This means that the position of the roll and the dive centres is important and the placing of those centres is not just guessing. For the position of the roll centres the lateral forces on the car are very important.

4.1.1: Roll moment

To determine the position of the roll centres the first thing that has to be considered is that torsion in the chassis has to be prevented. The lateral forces on the rear and the front of the car are different, which can result in torsion on the chassis. The lateral forces apply in the centre of gravity and the chassis rotates around the roll axis. The roll axis of a car is the axis that will be determined by the front and the rear roll centre. Connecting these two points will create that axis. The distance between the height of the centre of gravity and the height of the roll centres will cause a roll moment. For preventing torsion in the chassis the front roll moment has to be equal to the rear roll moment. These moments can be calculated using the lateral forces (F_Y) and the height of the roll centres and the centre of gravity. These parameters are shown in figure 4.1, which represents the front and the rear roll moment. The section at the front axle has been drawn, but the principle for the front and the rear roll moment is the same. Using equation 4.1, the equilibrium of moments, the relation between the front and the rear roll centre can be determined.

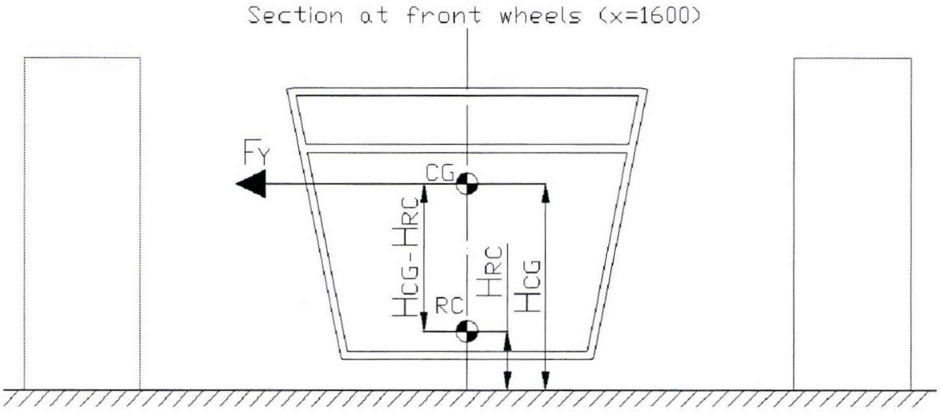


Figure 4.1: Roll moment

$$M_{roll,front} = M_{roll,rear} \qquad \text{(Equation 4.1)}$$

$$F_{Y,front} \cdot (H_{CG} - H_{RC,front}) = F_{Y,rear} \cdot (H_{CG} - H_{RC,rear})$$

$$m_{front} \cdot g \cdot \mu \cdot (H_{CG} - H_{RC,front}) = m_{rear} \cdot g \cdot \mu \cdot (H_{CG} - H_{RC,rear})$$

$$H_{RC,rear} = H_{CG} - \frac{m_{front}}{m_{rear}} \cdot (H_{CG} - H_{RC,front}) = 0{,}182 \cdot H_{CG} + 0{,}818 \cdot H_{RC,front}$$

From this calculation the conclusion can be made that the height of the rear roll centre ($H_{RC,rear}$) depends on the front roll centre height ($H_{RC,front}$) and the height of the centre of gravity (H_{CG}).

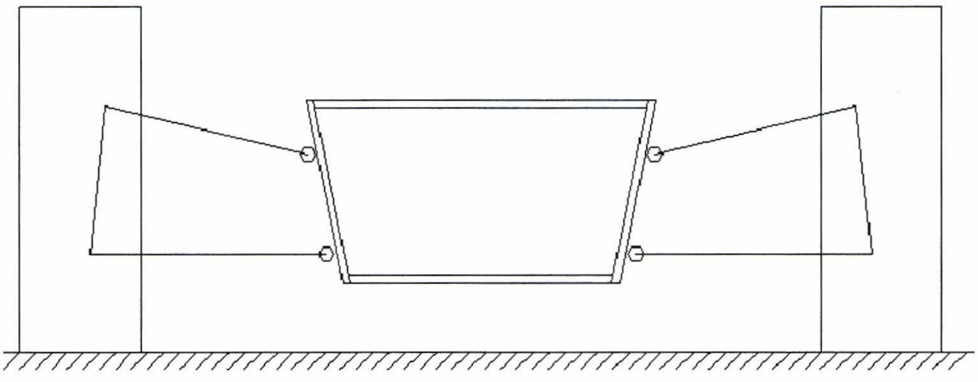
4.1.2: Front roll centre

The position for one roll centre has to be determined using the layout of the A-arms of the suspension. This will be done for the front suspension. For the front suspension the layout will be made assuming that both the outer and the inner wheel will stay flat on the ground during cornering. Using the vertical section of the front suspension at the front wheels the front roll centre can be constructed.

Before the front suspension can be designed the shape of the chassis had been made. The vertical section of the chassis over the front axle shows an upside down trapezoid, which means that the upper A-arms will be shorter, because the smaller distance between the chassis and the upright. Two A-arms for each wheel will be used to form the suspension. The upright is the part of the suspension that connects the two A-arms with the wheel.

In a bend the chassis will rotate around the roll axis, which results in a rotation of the A-arms. The tires will stay flat on the ground, which means that the tires will scrub over the ground and the track width will become smaller (couple of mm). This rotation of the A-arms is shown comparing figure 4.2a, which represent a car on a straight end, and figure 4.2b, for a car in a bend with 5 degrees roll.

a)



b)

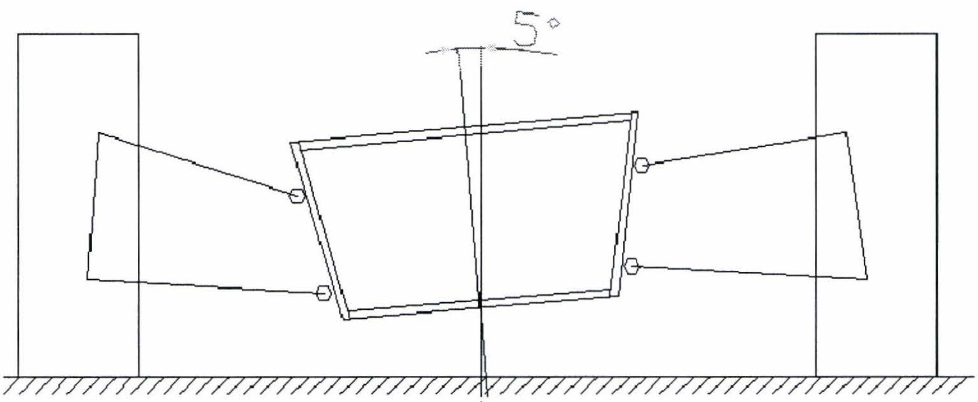


Figure 4.2: Chassis rotation a) straight ahead b) 5 degrees body roll

Comparing figure 4.2a en 4.2b it can be seen that both wheels stay flat on the ground when going through a bend. The track width will become smaller, because the wheels will be pulled together, due to the rotation of the A-arms around the pivot points on the chassis. The lines from the chassis to the tires represent the A-arms. The configuration shown in figure 4.2 provides the desired tire behaviour. Now the angles for the A-arms are known, the roll centre for this suspension layout can be determined. First of all the position of the front roll centre will be determined. The construction of the front roll centre is shown in figure 4.3.

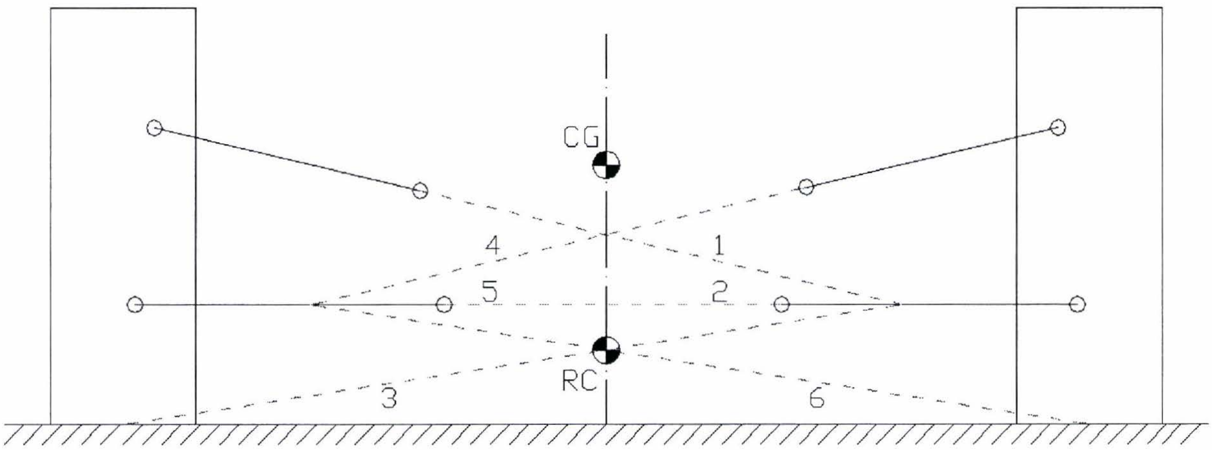


Figure 4.3: Construction front roll centre

When the lines, which represent the left upper and lower A-arm (1+2), will be extended until they cross an intersection has been found. From the middle point of the contact patch of the left tire towards the intersection for the two A-arms the third line (3) will be drawn. When these lines will also be drawn in the opposite way (4+5+6), the roll centre has been found in the intersection between line 3 and 6. The height of the front roll centre is 94,6 mm above the ground surface in this situation.

4.1.3: Rear roll centre

To determine the roll of the car around the roll axis, the position of the rear roll centre has to be determined. This position depends, according equation 4.1, on the height of the centre of gravity and the height of the front roll centre. Using this equation the height of the rear roll centre will be determined.

$$H_{RC,rear} = 0{,}182 \cdot H_{CG} + 0{,}818 \cdot H_{RC,front}$$

$$H_{RC,rear} = 0{,}182 \cdot 330 + 0{,}818 \cdot 94{,}6 = 137{,}4[mm]$$

Figure 4.4 represents the section of the rear suspension over the rear axle. The A-arms have been constructed using the height of the rear roll centre.

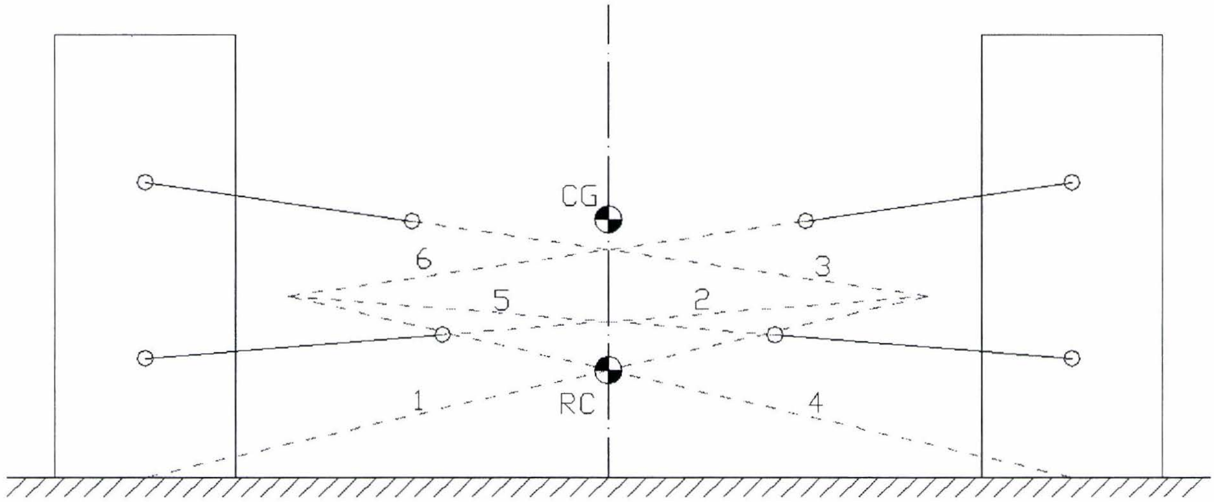


Figure 4.4: Construction rear roll centre

The construction for the rear roll centre is equal to the front roll centre. The only thing that is different is the line sequence. First of all line 1 has been drawn, which establishes the height of the rear roll centre. After that line 2 and 3 have been constructed so that the A-arm angles won't be too extreme. Lines 4, 5 and 6 have been constructed in the opposite way. The position of the pivot points of the upper and lower A-arm in the upright have been placed in a vertical line at the same height (z-position) as the pivot points for the front suspension. The position of the pivot points for the upper and lower A-arm at the chassis depend on the shape of the chassis, which causes an upper A-arm that will be shorter than the lower A-arm, just as for the front suspension. The two roll centre positions have established the lateral movement for the chassis. For the longitudinal movement of the chassis the dive centre position has to be determined.

4.1.4: Dive centre

For the movement of the car in longitudinal direction the dive centre of the car is important. This is the point about which the car rotates when braking or accelerating. For the construction of the dive centre the connection points of the front and rear suspension onto the chassis and the position of the centre of gravity are important. The desirable movement of the car in longitudinal direction depends on the wishes of the driver. Some drivers like it when the car doesn't dive at all, which means that more or less throttle won't cause a movement of the chassis. For the Formula Student car the dive centre will be placed so that the front part of the car drops during braking and will lift while accelerating. This movement is equal to a normal street car.

The amount of dive depends on the rules. A requirement for the car is that it has a wheel travel of minimal 2 inches (± 51 mm). This means that the springs, which will be used, have to guarantee this required wheel travel. The assumption has been made that, due to the weight of the car and the weight of the driver in full driving condition: wearing a helmet, a suit, gloves and shoes, the static compression will be one third of the total wheel travel, which means 17 mm. During dive another 25,5 mm of extra compression will be realized, so that the remaining 8,5 mm will be a buffer for an extreme situation like a bump during braking. The 25,5 mm dive will be generated by placing the dive centre in front of and below the centre of gravity.
The longitudinal forces will apply in the centre of gravity and the height difference between the dive centre and the centre of gravity will cause a dive-moment around the dive centre. The construction of the dive centre resembles the construction of the two roll centres. This construction is shown in figure 4.5, a side view of the car.

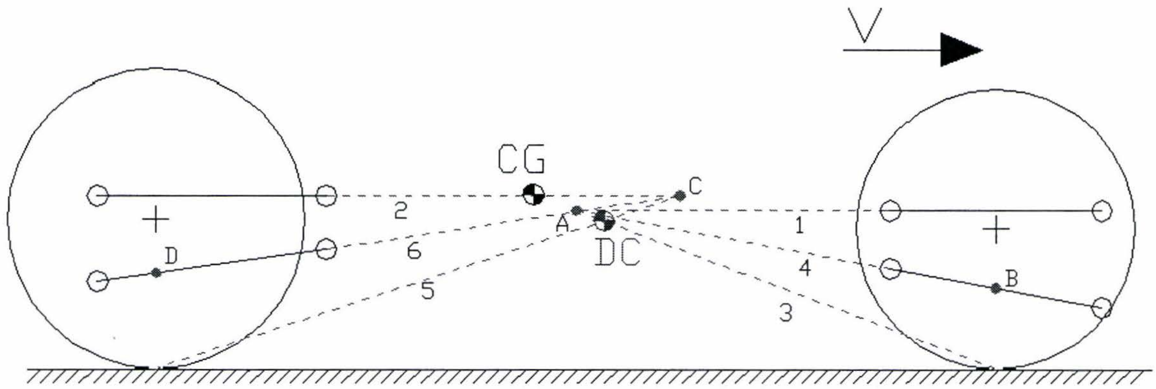


Figure 4.5: Construction dive centre

In figure 4.5 first of all two lines (1 and 2) have been drawn, which connect the two pivot points of the upper A-arm with the chassis for the front and the rear suspension. These two lines have been chosen horizontally. This has been done regarding the air stream along the chassis. This is only important for high speed cars regarding the aerodynamics, but for the Formula Student car this is a good starting point. The second assumption that has been made is that the dive centre has to be placed on the line (3) from the front wheel to the centre of gravity. Lines 1 and 3 generate point A, which leads to the next line (4) for the angle of the lower A-arm for the front suspension. This line is drawn through point A en the point that has been determined by the construction of the front roll centre (B). This point represents the

section of the lower A-arm at the height of the front axle. The height of the upper A-arm has been determined in the same way, using the section of the front roll centre.
The next line (5) determines the required position of the dive centre. The position of this line results in a dive centre that will be 133 ⅓ mm in front and 50 mm below the centre of gravity right on the line from the front wheel towards the centre of gravity. The last line (6) determines the angle of the lower A-arm for the rear suspension, using the intersection of line 2 and 5 (point C) and the point (D) that represents the section of the lower A-arm for the rear suspension.

Using the position of the roll centres and the dive centre the movement of the chassis in lateral and longitudinal direction has been established. For the desired handling of the car some alignments have to be made for the suspension, which have to result in a drivable car.

4.2: Suspension alignments

For the suspension several parameters are important to create a car set-up so that the car will be able to drive smoothly around short and tight tracks. Therefore the position for the front and rear wheels is very important. The wheels have to be placed under different angles for the desired steering conditions. The function and the set up for these different angles will be described in this paragraph. First of all the angle of the wheels in top view will be described in paragraph 4.2.1.

4.2.1: Toe

The amount that the wheels are pointed in or out, in top view, is respectively called toe in or toe out. The size indication for this parameter will often be given in millimetres, the difference between the track widths as measured at the leading and trailing edges of the rims. A little change can make huge differences for the handling of the car. When a rear wheel driven car, like the Formula Student car, moves forward the front part of the car tends to move back generating toe out. Therefore a rear wheel driven car will have toe in, to compensate this phenomenon and to keep the wheels straight during driving. If a rear wheel driven car has toe out it often tends to wander over the road. The rear wheels will have a little toe out, because the power, generated by the engine and transmitted by the drive shafts towards the rear wheels, will pull the rear tires in a straight line with the driving direction, using 1 – 3 mm toe out.
Just as for the angle in top view, the angle for the wheels in front view has to be determined This will be the next parameter that has to be established.

4.2.2: Camber

Camber is the angle of the wheel relative to the z-axis, as viewed from the front or the rear of the car. If the wheel leans in towards the chassis, it has negative camber; if it leans away from the car, it has positive camber. These two different types of camber are shown in figure 4.6.

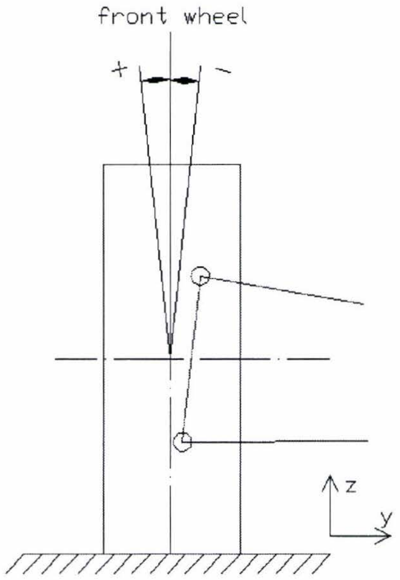


Figure 4.6: Camber

Camber is important for the contact patch between the tire and the road. The cornering force developed by a tire is highly dependent on its angle relative to the road surface, and so camber has a major effect on the road holding of a car. A tire develops its maximum cornering force at a small negative camber angle, typically around negative 0,5 degrees. This fact is due to the contribution of camber thrust, which is an additional lateral force generated by elastic deformation as the tread rubber pulls through the tire/road interface (the contact patch). For the front tires of the Formula Student car camber will be established at negative 0,5 degrees.

Camber and toe are static adjustments to the position of the wheels. The dynamic behaviour of the wheels will be determined by the layout of the suspension. First of all the desired wheel movement will be considered for the handling of the car. Therefore the term Ackermann will be explained in paragraph 4.2.3.

4.2.3: Ackermann

Ackermann steering geometry is a geometric arrangement of linkages in the suspension layout of the car. This has been designed to solve the problem for the wheels on the inside and outside of a bend needing to trace out circles of different radii. When a vehicle is steered, it follows a path that is part of the circumference of its turning circle, which, at low speed, will have a centre point somewhere along a line, extended from the axis of the rear axle.
When the front wheels of a vehicle are steered away from the straight ahead position the design of the steering linkage will determine whether the wheels stay parallel or that one wheel steers more than the other. For the steering response of the wheels three different situations can be considered. These three situations are shown in figure 4.7.

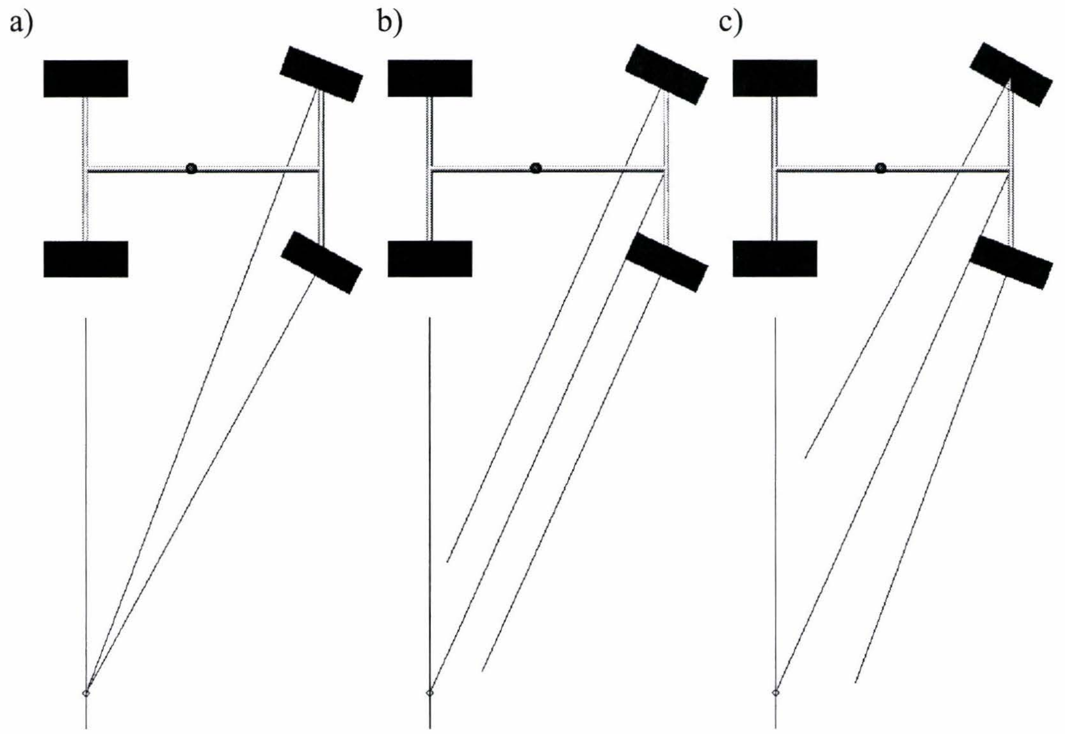


Figure 4.7: Steering geometry a) neutral Ackermann b) parallel c) reverse Ackermann

In figure 4.7a neutral Ackermann is shown, which means that there has been chosen for a larger steering angle for the inner front wheel than for the outer front wheel. All the lines perpendicular to the wheels will intersect in one point. Figure 4.7b shows that the steering angle for both wheels is exactly the same, which is called parallel steering. In figure 4.7c the last situation has been shown. In this case the inner wheel has a smaller steering angle than the outer wheel, which is called reverse Ackermann. For the Formula Student car the choice has been made to steer with neutral Ackermann, which implies that the inner wheel has to steer more than the outer wheel. This difference will be caused by adjusting the linkage of the tie rods of both wheels that the angle difference will be created.

Considering the narrow and short bends that have to be driven during the Formula Student competition the choice has been made to create a steering layout that at near zero speed the car has to be able to drive a bend of 180 degrees with an outer diameter of 9 m. This will result in a desired driving radius of 3,5 m, because the width of the car has to be subtracted from those 9 meters. Using these parameters and considering the fact that the car will drive with neutral Ackermann the desired steering angles can be determined. The angles that will be necessary to achieve neutral Ackermann have been calculated in appendix K. These calculated angles are shown in figure 4.8.

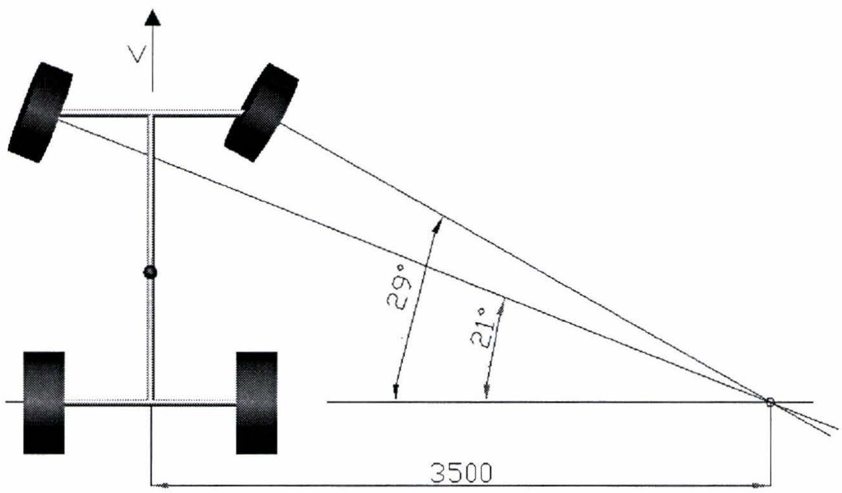


Figure 4.8: Neutral Ackermann wheel angles

Figure 4.8 shows that the inner wheel has to steer more (29 degrees) than the outer wheel (21 degrees). When turning in the other direction the angles of the different wheels change. This means that both front wheels will have an angle range of 50 (29 + 21) degrees. The difference between the wheel angles of the front wheels in the most extreme position is called the Ackermann angle, which will be 8 degrees in this case. This Ackermann angle is not constant but varies with the turning radius. The Ackermann factor is the ratio between the actual angle between the front wheels and the full Ackermann angle as shown in figure 4.8. For parallel steering the Ackermann factor is zero. When Ackermann geometry is fully implemented the Ackermann factor is 100%.
A high Ackermann factor is useful in taking tight corners at low speed. At higher speeds its usefulness is dubious. In fact, during high-speed cornering the dynamic effects compensate for the Ackermann effect. Because the Formula Student car has to make tight corners at a relative low speed, compared to the possibilities of the car, neutral Ackermann steering has been established.

In figure 4.8 it's visible that the front wheels not only have to make an angle in top view of the car during steering. Also in front view of the car a little angle (about 5 degrees) has to be made so that the lines perpendicular to the wheels not only intersect, but especially intersect on the road surface. This required steering movement has to be obtained by the construction of the upright with the right KPI- and caster angles. These two parameters will be described in the paragraph 4.2.4 and 4.2.5.

4.2.4: Kingpin inclination

The following choice that has been made is the Kingpin Inclination (KPI) by using the required steering angles for the two front wheels. KPI is the angle described by a line drawn down through the upper and lower pivot points of the upright, viewed from the front of the car. Extended to ground level, the distance from here to the tire centre-line at ground level is the scrub radius. For the Formula Student car the Kingpin-axis and the tire centre-line intersect at ground level. This will give both lightness of steering feel and virtually no kickback through the steering wheel when hitting bumps. This is called centre point steering. Centre point steering eases the steering, because the friction between the road and the tire, during cornering (when the wheel rotates), will be lower then when a positive or negative scrub radius will be used. Also the tires will lift more during cornering when using a positive or negative scrub radius. This will result in a higher steering resistance and the contact patch with the road will be smaller. Furthermore centre point steering has been established to avoid steer torques for the driver. If the driving or braking forces are different on the left and right wheels, a steering torque will be felt by the driver. When using zero scrub radius, centre point steering, there is no moment arm for the drive or brake forces to generate torque about the kingpin. Considering Ackermann and the requirement to drive with centre point steering the KPI-axis can be constructed. This leads to the configuration in figure 4.9. Using equation 4.2 the KPI-angle has been calculated.

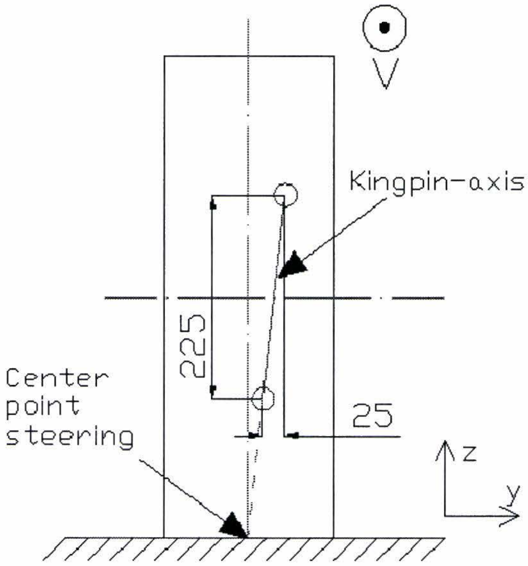


Figure 4.9: Kingpin inclination angle

$$KPI\_angle = \arctan\left(\frac{y_{upper} - y_{lower}}{H_{upright}}\right) \qquad \text{(Equation 4.2)}$$

$$KPI\_angle = \arctan\left(\frac{25}{225}\right) = 6{,}3°$$

Figure 4.9, the front view of the right front tire, and equation 4.2 show that the KPI angle is 6,3 degrees. The two circles represent the two pivot points of the upper and lower A-arm with a vertical distance in z-direction of 225 mm and a horizontal distance in y-direction of 25 mm. For the stability of the car the next parameter that has to be considered is the caster-angle.

4.2.5: Caster

The KPI-angle is not the only angle, which has to be considered for the front wheels. Just as for the front view there is an angle between the pivot points of the A-arms in side view. This angle is called the caster angle. Caster has to provide the steering stability, which will keep the front wheels in the straight-ahead position and also assists in straightening up the wheels after a bend. This angle causes the wheel to rise and fall with steer, to create Ackermann. Furthermore positive caster creates mechanical trail which is used for rear driven vehicles to create a self aligning of the two front wheels. The more trail is set the larger the moment arm for the tire side force, which will act on the kingpin axis. Also, the greater the caster angle the greater the horizontal offset between the pivot points in the upright. This will then elongate

the contact patch and the loadings too much on the front or rear of the tire and will lead to sensitivity in the tire to minor imperfections in the track surface.
Therefore it's important that the caster angle and the trail are considered very well. For the Formula Student car the trail has been set at 35 mm and the other choice which has been made is that the upright axis is going through the centre of the wheel. This results in a configuration that is shown in figure 4.10, where the front wheel of the car has been drawn. With equation 4.3 the caster angle has been calculated.

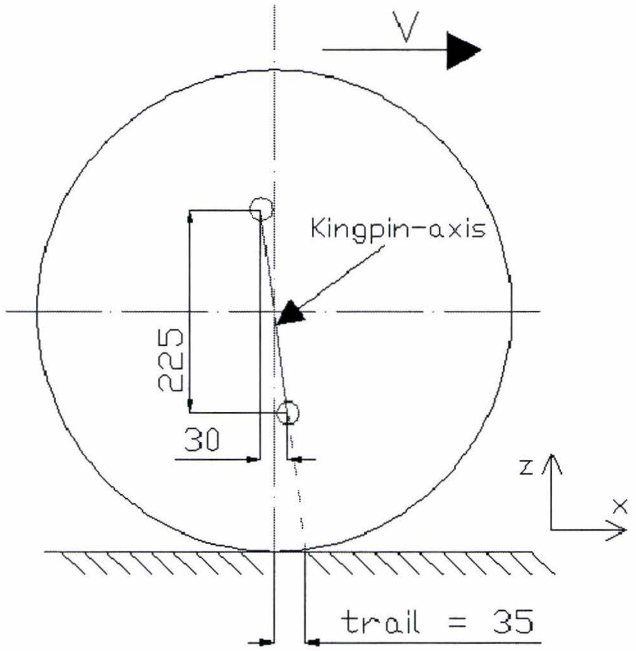


Figure 4.10: Caster angle

$$caster\_angle = \arctan\left(\frac{x_{upper} - x_{lower}}{H_{upright}}\right) \quad \text{(Equation 4.3)}$$

$$caster\_angle = \arctan\left(\frac{30}{225}\right) = 7{,}6°$$

Considering the trail of 35 mm and the distance between the lower and upper pivot point of the A-arms (in figure 4.10 the little circles) a caster angle of 7,6 degrees has been created. This causes a difference in x-direction of the pivot points of the A-arms of 30 mm.
The angles for the wheel and the kingpin axis have all been determined now and these angles will cause a camber change, because steering will be done according Ackermann. This camber change is shown in appendix L.

For the layout of the suspension one other alignment has to be considered. In the next paragraph the location of the tie rod will be determined to prevent bump steer.

4.2.6: Bump steer

The tie rod that will be used to transmit the input at the steering wheel in to the upright has to be designed to prevent bump steer. Bump steer is when the left or right wheels steer themselves without input from the steering wheel. The undesirable steering is caused by bumps in the track interacting with improper length or angle of the suspension and steering linkages. The tie rod is connecting the upright with the steering mechanism. For the position of the tie rod different choices have to be made to create the different steering angles for the two front wheels, considering neutral Ackermann. The tie rod position in the wheel will be integrated in the upright. Therefore it is important that this position is constructed considering bump steer. First of all the position of the steering mechanism has to be determined. The steering mechanism can be positioned in two different ways shown in figure 4.11a and 4.11b.

a) b)

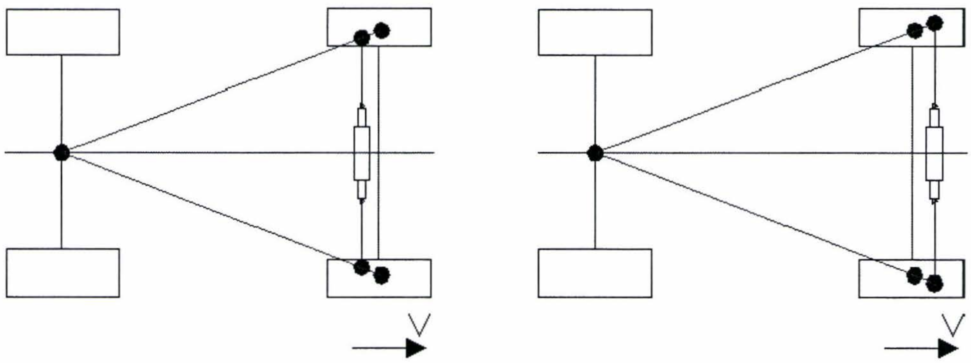


Figure 4.11: Position steering mechanism: a) behind front axle b) in front of front axle

To make a choice some parameters had to be taken into account. First of all the amount of space that is available will be looked at. Considering the option with the placement in front of the front axle a very deep rim (a large negative offset) has to be used, considering the choice of the pivot points in the upright. Because in the front wheel different parts for the brake system have to be placed, such as brake disc and brake calliper, the space in the rim is small. The second point is that when the wheels will be steered into a tight bend the tie rods will interfere with the rim. Therefore the choice has been made to place the steering mechanism behind the front axle. Considering the position of the driver's legs this option will be better too, because the desired position of the steering mechanism in the chassis is underneath the driver's legs. When placing the steering mechanism behind the front axle, the legs of the driver will be higher, in side view, at that point in the chassis. The position before the front axle will cause interference of the tie rod with the driver's ankles.

Considering the fact that for the steering of the car there has been chosen to drive with neutral Ackermann the connection point of the tie rod with the upright has been determined. This is shown in figure 4.12, where three different ways for the movement of the front wheels are shown.

a) b) c)

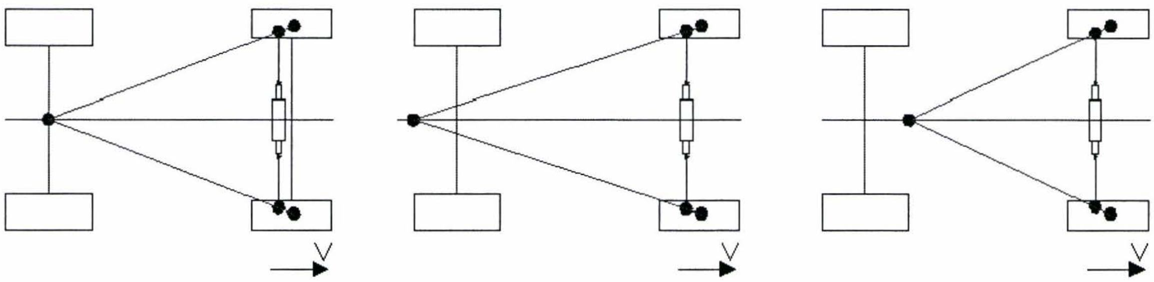


Figure 4.12: Pivot point tie rod in upright: a) neutral - b) less - c) more Ackermann steering

Figure 4.12a shows driving with neutral Ackermann. This is defined by angling the steering arms so that a line drawn between both the kingpin and steering arm pivot points intersects in the middle of the centre line of the rear axle. As this gives neutral Ackerman steering geometry, there is no toe angle change on the inside wheel. Figure 4.12b shows less Ackerman angle, which involves adjusting the angle of the pivot points on the steering arms so that the point of intersection is behind the centre line of the rear axle. This steering geometry achieves a reduced amount of angular inequality of the turned wheels, which results in the inside wheel trying to follow a larger diameter circle than it actually does. Figure 4.12c shows more Ackerman angle, which involves adjusting the angle of the pivot points on the steering arms so that the point of intersection is in front of the centre line of the rear axle. This steering geometry achieves greater angular inequality of the turned wheels, which results in the inside wheel trying to follow a smaller diameter circle than it actually does.

Another choice which has to be made is the height, in side view of the car, of the pivot point of the tie rod with the upright. The distance in z-direction between the lower and upper pivot point is 225 mm, so there is quite a range to connect the tie rod. For that choice the forces on the tires had to be considered. When a driver is trying to make a bend, side forces will be building up at the contact patch of the tires with the road. These forces will create a moment on the steering axle, which can be calculated by determining the maximal side forces and multiply this with the height of the connection point with the tie rod. The higher this point the bigger the moment. Therefore the choice has been made to place the tie rod as low as possible in the upright, considering the available space and the position of the lower A-arm, because interference between different parts has to be prevented.

The position of the tie rod in the upright has now been determined. For the tie rod the position where it enters in the chassis is very important too. When driving, it is desirable that the movement of the tires won't cause a steering movement of the tires. For the driver bump steer is very irritating because of the extremely fast corrections that have to be made during driving. Therefore the choice has been made to construct the tie rod with bump steer prevention. The method to construct the tie rod to prevent bump steer is shown in figure 4.13, where one side of the section of the front wheels at the height of the front axle has been drawn.

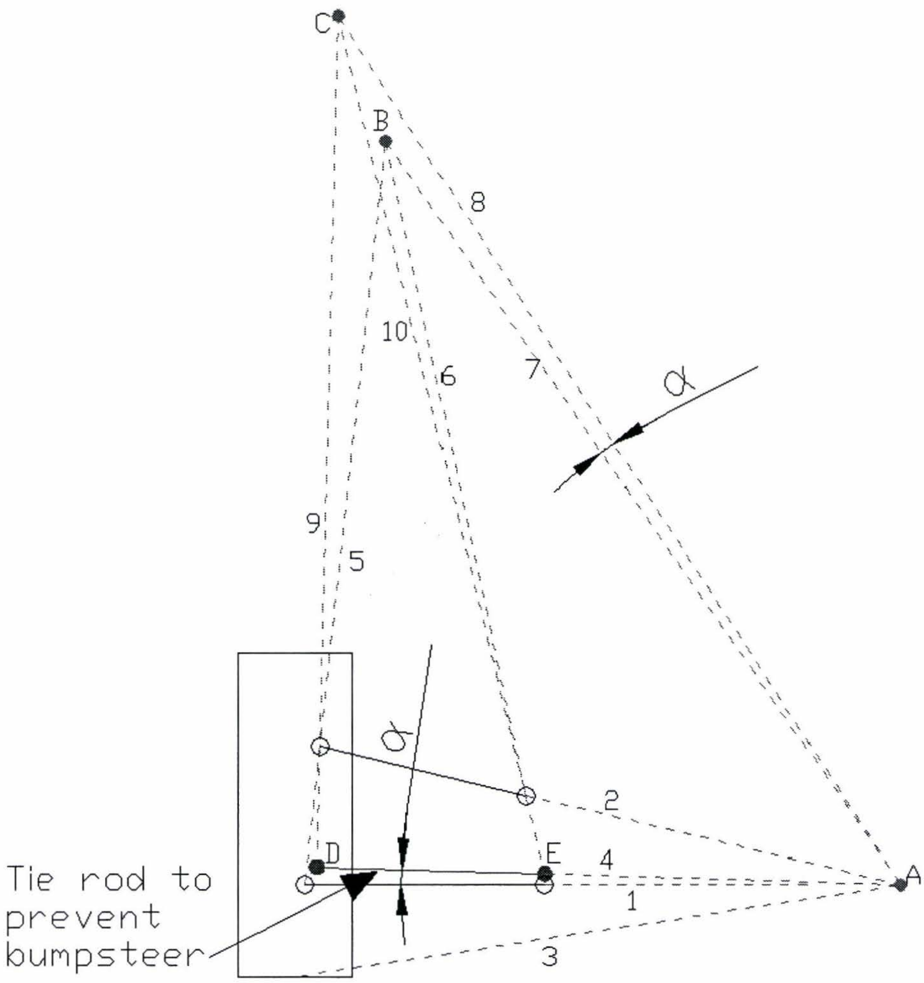


Figure 4.13: Construction front tie rod

The construction of the tie rod is based on the construction of the roll centre. First of all point A is determined by drawing three lines (1, 2 and 3). From the connection point of the upright with the tie rod, point D, a line (4) has been drawn to point A. The king pin axis (5) and the line through the pivot points of the A-arms on the chassis (6) create intersection point B. After that a line (7) will connect point A and B. The angle created by line 1 and 4 is exactly the same angle between line 7 and a new line (8). From point D a new line (9) through the upper pivot point of the kingpin axis is creating intersection point C. From this point the last line (10) has been drawn through the upper pivot point at the chassis to create an intersection between line 4 and 10, point E. The line between point D and E is the exact position of the tie rod to prevent bump steer.

In the section, drawn in figure 4.13, there are only two pivot points on the chassis. These points represent the section of the lower and the upper A-arm at the height of the front axle. This includes that the connection point of the tie rod with the chassis depends on the x-position of this point. This position determines the height of point E, because bump steer prevention has been made in 2D, but when the A-arms will be analysed in 3D it won't be lines anymore but planes. Regarding the fact that the steering mechanism will be placed 100 mm in front of the front axle the correction here fore will be 10 mm in z-direction. This means that the true pivot point of the tie rod at the chassis will lie 10 mm higher than drawn in figure 4.13.

The position of the pivot points on the chassis will be used to guide the forces from the suspension in to the chassis. Therefore the position of the centre lines of the A-arms and the chassis are important and will be regarded in paragraph 4.3.

4.3: Suspension centre lines

The positions of the wishbones are very important. The centre lines of the wishbones have to intersect in the pivot points of the upright to prevent moments in the A-arms. Furthermore the connection of the suspension with the chassis has to be considered too, because those points have to resist the highest forces in the chassis. The forces have to be guided properly into the chassis and this only can be done when the pivot points of the suspension on the chassis will be the points with the highest resistance against the forces from the suspension. Therefore the choice has been made to connect the suspension to the frames of the chassis, because they represent the hard points in the chassis. These hard points are shown in figure 4.14.

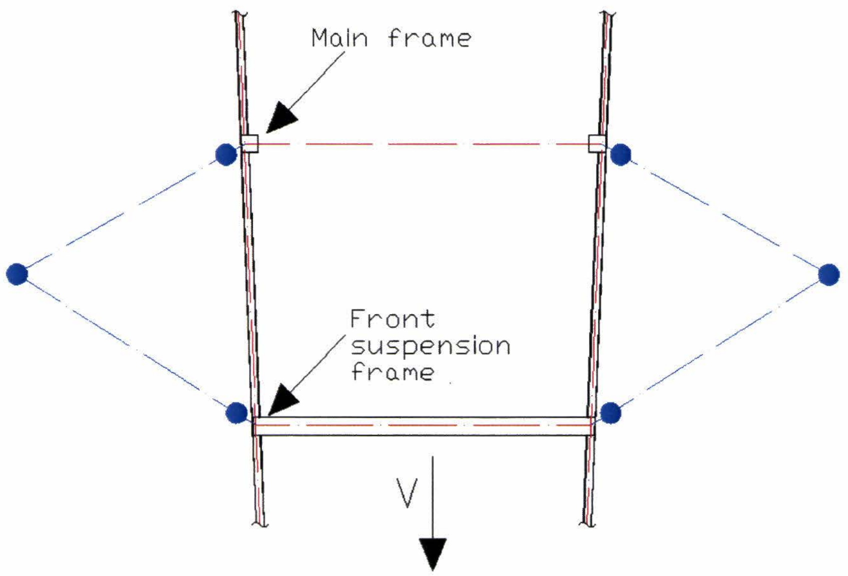


Figure 4.14: Centre lines suspension and chassis

Figure 4.14 shows the section of the chassis at the height of the pivot points of the upper A-arm for the front suspension and the chassis. Figure 4.14 not only shows that the centre lines of the tubes for the suspension (blue lines) intersect in the pivot points of the upright, but also that the centre lines of the suspension intersect with the centre lines of the frames (red lines). The intersection between the centre line of the tubes and the centre lines of the frames is placed on the centre lines of the sandwich panels. Using this method the forces will not only be guided into the frames, but the sandwich panels will also be used to guide the forces properly to the centre of the car.

Until now the choices for the position of the different suspension parts have been made. This has been done using only 2D lines and circles as pivot points. To make a real impression for the layout of the front suspension, these 2D impressions will be translated into a 3D layout for the suspension parts in chapter 5.

Chapter 5: Suspension layout

The principles used for the suspension have all been explained in chapter 4. The centre lines for the suspension have been determined, but now the centre lines have to be translated into reality, which means a three dimensional layout for the suspension. The construction for the different suspension parts will be explained in this chapter, starting with the A-arm construction and the connection between the upper and lower A-arm.

5.1: A-arm structure

The A-arm structure will be divided in different parts. The wishbones, the upright and the bearing are considered to be separate parts. An A-arm will be made of two separate rods, which will be connected rigidly. For these connections left- and right-handed screw thread will be used on either side of the rods. When using both left- and right-handed screw thread, the length will be adjustable in the range of a couple of millimetres. First of all the calculation and the construction of one wishbone will be explained and the connection of two wishbones.

5.1.1: Wishbone dimensions

The suspension will be made using aluminium tubes. The dimensions of these tubes will be determined using the forces from the tires to avoid buckling of the tubes. The highest force in the suspension will be generated in the lower A-arm of the rear suspension during 1,5g acceleration. This force attaches in the pivot point of the lower A-arm and will be resolved in two components, a tensile force (F_{tens}) and a pressure force (F_{pres}). This process is shown in figure 5.1.

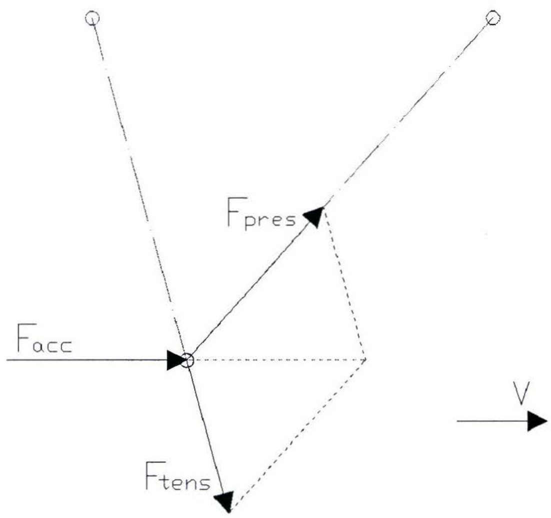


Figure 5.1: Forces on A-arm

Analysing all the different forces on all the A-arms of the total car (appendix E) the highest force in one A-arm will be about 4500 N in the rear lower A-arm during accelerating. The length of the longest member of this A-arm is 370 mm. This force and tube length will be used to determine the necessary moment of area in the tubes for the suspension. With equation 5.1 the minimal required moment of area can be determined, using the Young's modulus for aluminium of $7 \cdot 10^{10}$ N/m^2.

$$F_{buckle} = \frac{\pi^2 \cdot E \cdot I}{l_{buckle}^{\;2}} \Rightarrow I = \frac{F_{buckle} \cdot l_{buckle}^{\;2}}{\pi^2 \cdot E} \qquad \text{(Equation 5.1)}$$

$$I_{tube} = \frac{4500 \cdot 0{,}37^2}{\pi^2 \cdot 7 \cdot 10^{10}} = 8{,}92 \cdot 10^{-10} [m^4]$$

The dimensions for the tubes have been chosen to be an aluminium tube with an outer diameter of 14 mm and an inner diameter of 10 mm, using a wall thickness of 2 mm. The moment of area for this tube can be calculated using equation 5.2.

$$I_{tube} = \frac{\pi \cdot \left(d_{out}^{\;4} - d_{in}^{\;4}\right)}{64} \qquad \text{(Equation 5.2)}$$

$$I_{tube} = \frac{\pi \cdot \left(0{,}014^4 - 0{,}01^4\right)}{64} = 13{,}95 \cdot 10^{-10} [m^4]$$

A safety factor of 1,5 for the most extreme forces on the suspension will be reached, using this tube. The stresses in the suspension, using this aluminium tube have been analyzed in appendix M.

5.1.2: Wishbone construction

The main part of a wishbone is the aluminium tube. These tubes have to be on one side connected to a rod end, which will provide the rotation of the A-arm in the pivot point with the chassis. The rod ends used will be M8. At the other side two tubes of one A-arm have to be connected to each other for the mounting in the upright. The construction used for a wishbone is shown in figure 5.2.

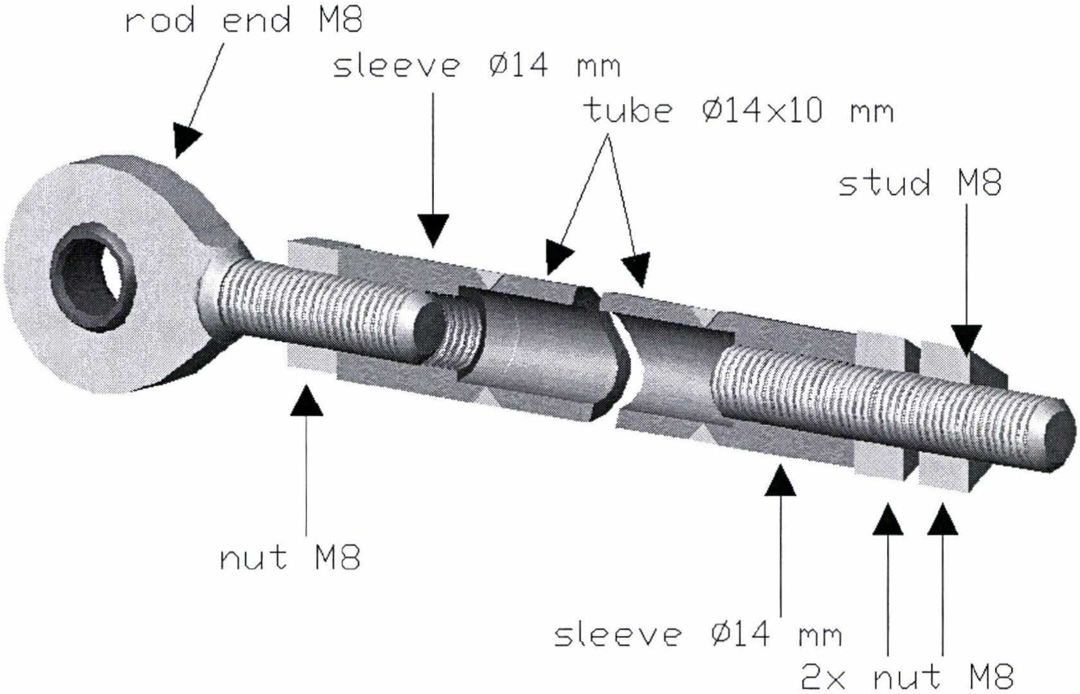


Figure 5.2: Section wishbone

Figure 5.2 shows the section of a tube. In reality these tubes will have a length close to 350 mm. The lengths of the wishbones are all different, because the angles in the suspension have all been designed slightly different. The tie rods and the pushrods, which will be described later, will have the same layout as the A-arms. The only difference is that these two parts will contain rod ends on both sides of the tube.

One side of the aluminium tube of the A-arm will be mounted onto the rod end and the other side has been provided with external thread. Onto the tubes aluminium bushes will be welded to create the internal thread on both sides. The rod end will be screwed into the tube on one side and will be kept in place using a nut. For a small adjustment to the length of the tubes, to make little adjustments to the suspension layout, this nut will be used to create a range of a few millimetres. By repositioning the rod end the length of one member of the suspension will be changed. For the tie rod and the push rod two end bushes, containing internal thread, will be welded onto the tube. On the other side of the A-arms a steel insert stud will be screwed into the bush and will be kept in place using two nuts. Using this stud two members of one A-arm can be connected to each other using an aluminium block.

The layout of the aluminium blocks for the upper and lower A-arm is different because of the different angles for the suspension. The front and the rear suspension contain different angles so that the holes with internally thread for the wishbones have to be made under different angles. This means that for the front and the rear suspension 8 different blocks will have to be made. For the upper A-arm a rod end will be used for the pivot point in the upright, which will be positioned using two nuts on either side of the aluminium block. Therefore a hole in the middle of the block has been drilled. The used principle for all the aluminium blocks is the same. These blocks are shown in figure 5.3 and 5.3b.

a) b)

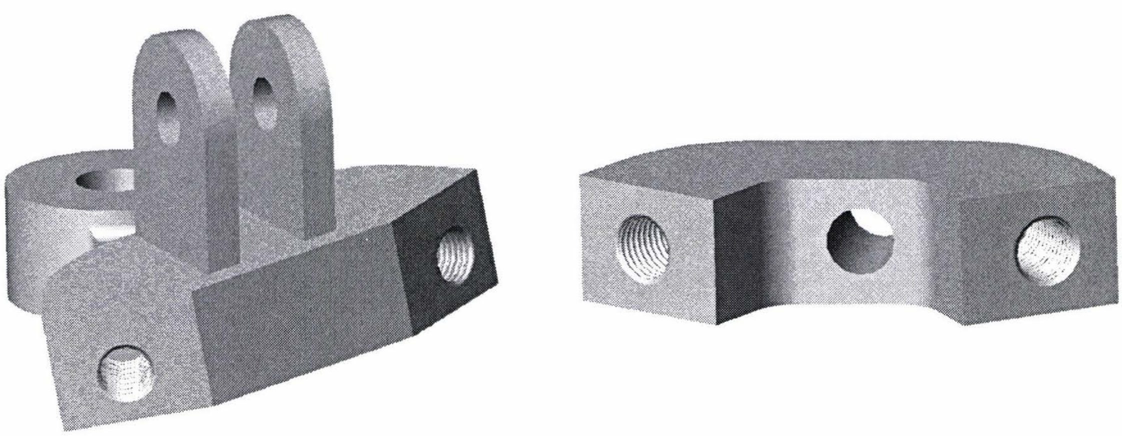


Figure 5.3: Aluminium blocks for A-arms a) lower A-arm b) upper A-arm

The two flanges on the aluminium block for the lower A-arm will provide the double shear for the pushrod. The centre line of the pushrod will intersect the centre lines of the members of the lower A-arm exactly in lower pivot point of the upright. This results in tension and compression loading of the tubes instead of bending. The upper aluminium block will be connected to the upright using a rod end, because the forces on the upper A-arm are radial only instead of axial and radial as for the lower A-arm.
The rod ends that will be used at the chassis side of the suspension have to be connected to the chassis. This connection will be explained in paragraph 5.1.3.

5.1.3: Connection A-arm to chassis

The A-arms have to be connected to the chassis. Rod ends will be used to create this connection. The rod ends provide the fixation of three translations. The three rotations will be kept free. The rod ends will be positioned in vertical direction, because of the motion on the suspension. The biggest movement of the suspension will be the rotation around the pivot points on the chassis. This rotation will be maximal 10 degrees. The available side rotation of the rod ends is only 12 degrees, which means that positioning the rod end horizontally will be critical when a worst case load will be generated. Therefore the rod ends will be positioned, which is shown in figure 5.4.

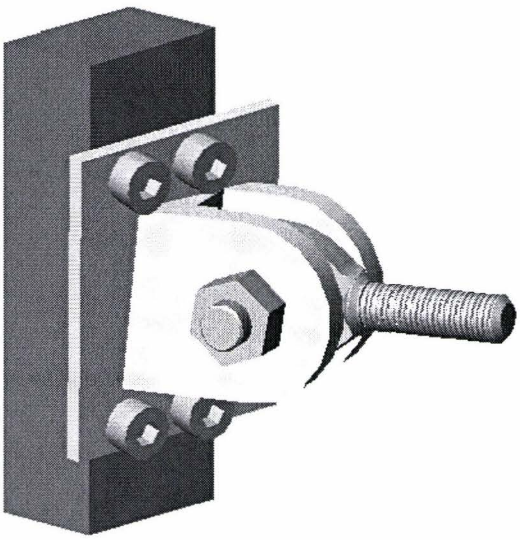


Figure 5.4: Connection between suspension and chassis

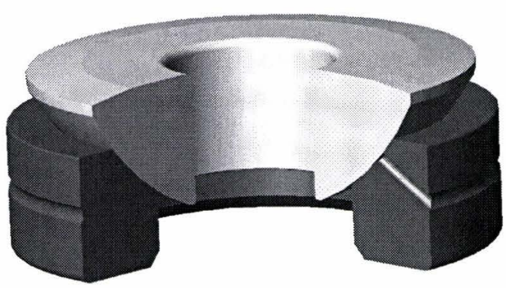
In figure 5.4 aluminium blocks are shown which will provide the connection of the rod ends to the chassis, especially the frames. For the front and the rear suspension eighteen different blocks will be used, regarding the angles of the A-arms. The outer layer of the sandwich panel will be mounted between the aluminium block and the frame. These blocks will be milled out of solid aluminium and will be bolted on to the frame using four M5 bolts. Therefore thread will be tapped into the frame. For the layout of the aluminium blocks the choice had been made to use a double sheer construction, which means that the rod ends will be locked up between two flanges of the aluminium block. The rod ends will be fixated in the aluminium blocks using M8 close tolerance bolts and nuts. These bolts have to fit perfectly in the rod ends so that there will be as little radial tolerance as possible.

The required motion of the A-arms at the chassis side has now been provided. At the other side of the A-arm, where the wishbones have been screwed in the aluminium blocks, an other type of bearing will be used to provide the required rotation for the A-arms.

5.1.4: Axial spherical plain bearings

A rod end will be used for the connection of the upper A-arm to the upright. Together with the pivot point of the tie rod in the upright this is the only point in the suspension layout where the rod end is placed horizontally, because of the lack of space in the upright. For the connection of the lower A-arm to the upright an other bearing will be used: an axial spherical plain bearing. The lower pivot point has to resist the vertical component of the forces from the pushrod. Therefore the use of a rod end won't do. When a rod end will be used horizontally for the lower pivot point in the upright the force of the pushrod will result in a load case which will damage the rod end, because a rod end can only resist high forces in plane. Forces out of plane will cause a wrong use of the rod end, which result in failure of the rod end. The axial spherical plain bearings can resist those forces out of plane. This type of bearing is shown in figure 5.5a and how the bearings will be integrated, regarding the aluminium block for the lower A-arm, is shown in figure 5.5b.

a)



b)

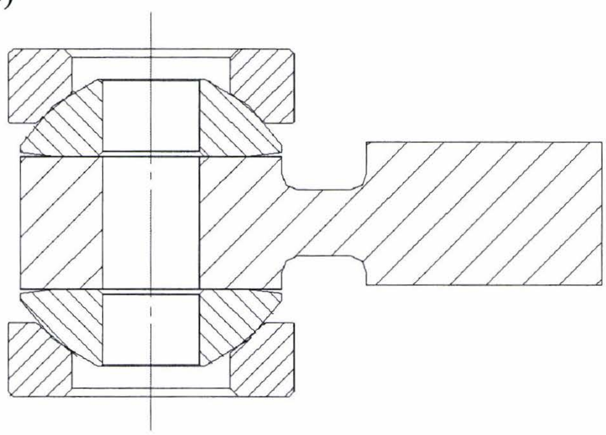


Figure 5.5: Axial spherical plain bearing a) layout bearing b) implementation bearings

Figure 5.5a shows that a lubricant will be used to take care of the lubrication of the bearing. Therefore a little canal has been made in the outer part of the bearing to guide the lubricant to the contact area between the two separate parts. This canal is shown in the right part of figure 5.5a. To provide the rotation for the aluminium block for the lower A-arm one pair of bearings will be used. The two inner-bearing cups will be mounted onto the aluminium block and the two outer-bearing cups will be kept in place in the upright. Therefore it's important that the clearance between the two parts of one bearing will be as low as possible.

The connection between the upper and lower A-arm will be made using uprights. These parts will be described in the paragraph 5.1.5.

5.1.5: Upright

For the suspension layout the position and the shape of the uprights are very important. These two features will be determined by the required function of the uprights. The uprights have to maintain the position of the pivot points of the upper and the lower A-arm. These points provide the KPI- and the caster-angle and determine the length of the members of the A-arms.

Furthermore the position of those two points determines the behaviour of the suspension during bump and droop and they establish the movement of the car during roll.
The other functions of the upright is to provide the pivot points with the tie rod and it has to fit the axle, so that the wheel and the brake disc (for the front upright) can rotate, with respect to the upright and the brake calliper. Therefore angular contact ball bearings will be used to provide the rotation of the wheel. These bearings have to be mounted within the upright or in the hub which will connect the wheel with the bearings or with the axle. This will be different for the front and the rear axle.

First of all the front uprights will be regarded. The difference between the front and the rear upright is that the front upright has to be able to rotate according the steer geometry provided by the layout of the front suspension. The rear upright may not steer. For the front upright this means that it has to be able to make a total angle of 50 degrees according the outer limits of the steering angles to provide neutral Ackermann. This means that the lower and upper A-arm have to be able to reach those extreme angles, clearing the rim

For the structure of the uprights the choice has been made to use a sort of sandwich construction. This will result in a relative light and stiff construction for the upright. Therefore a plate structure of aluminium has been made to provide the correct location of all the necessary pivot points, i.e. the A-arms, the tie rod and the push rod. For the pivot point of the lower A-arm a chamber has been made for the axial spherical plain bearings and the aluminium block. The shape of this chamber has been made using the required 50 degrees for the steering rotation. During this rotation the aluminium block has to be prevented from making contact with the upright under maximum steering angle. The plate structure for the upright has been designed with the centre lines for the plates intersecting as result. Figure 5.6, the front view of a part of the upright, shows a sample of this principle.

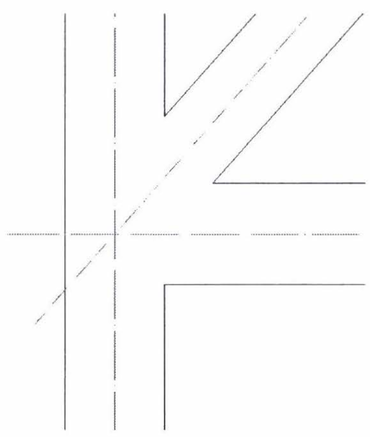


Figure 5.6: Plate structure principle

Using the principle of figure 5.6 the forces in the upright will be guided nicely into the structure and towards the axle. For the connection with the upper A-arm and the tie rod, flanges will be mounted upon the plate structure. In these flanges holes with a diameter of 8 mm will be made to provide the connection with the upper A-arm and the tie rod using bolts and nuts M8. These flanges have a thickness of 5 mm and a mutual distance of 10 mm, which has been chosen regarding the dimensions of the rod ends (thickness is 6 mm). The rod ends require a rotational movement, because of their rotation angle out of plane. The centre of the upright is a cylinder with a diameter of 60 mm provided with a drilled hole for the front axle. The design of the front upright is shown in figure 5.7a en 5.7b.

a) b)

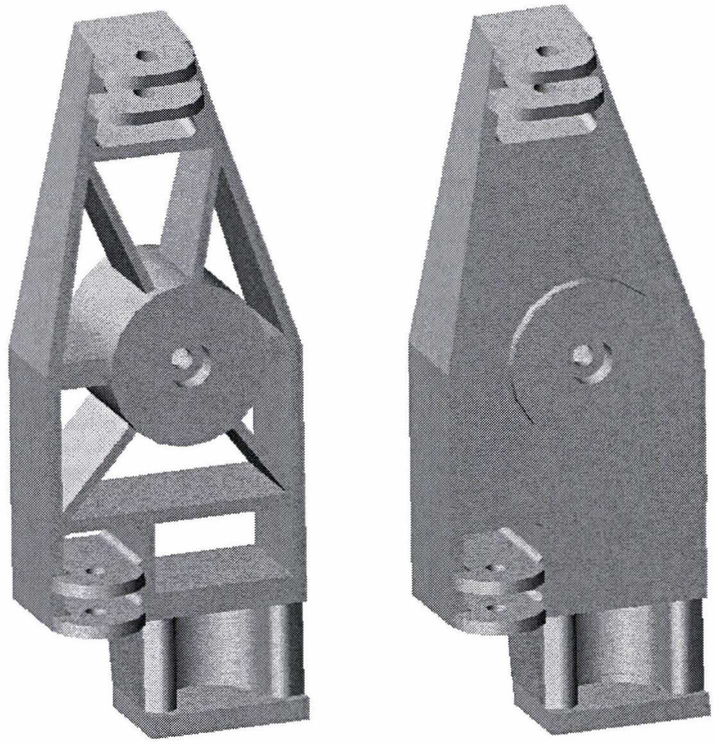


Figure 5.7: Front upright a) plate structure b) closed box structure

The plate structure will be covered at the front and the backside of the upright, as far as possible, to provide more torsion stiffness (figure 5.7b). This will be done using two aluminium skins with a thickness of 2 mm. These skins create a box structure in sandwich style. The lower part of the upright, the pivot of the lower A-arm, will be rounded, so that the aluminium block will not make contact with the upright. This block and the bearings will be fixed in the milled chamber using the rectangular block at the under side of the upright, which will be bolted onto the upright using four Allen bolts M5. The upper part of the upright, above the axle, has been chamfered at the rear side to create a weight reduction, because the removed part didn't tribute very much to the overall stiffness of the upright.
The pivot point of the tie rod has been placed just above the pivot of the lower A-arm to avoid mutual contact, but placing the tie rod as low as possible to minimize the forces in the tie rod. In this position bump steer has been prevented.

The rear upright has been designed in the same way as the front upright, which means that the same plate structure has been used to create the inside of the closed box. For the closing of the box two skins of aluminium with a thickness of 2 mm, just like the front uprights, will be used. The only difference with the front upright is that the tie rod for the rear suspension, which will be fixed within the rear suspension, unlike the front tie rod, has been placed at the height of the rear axle. It has been placed in such a way that bump steer is prevented for the rear suspension. The construction of the tie rod for the rear suspension is shown in appendix N. Figure 5.8a en 5.8b show the design of the rear upright.

a) b)

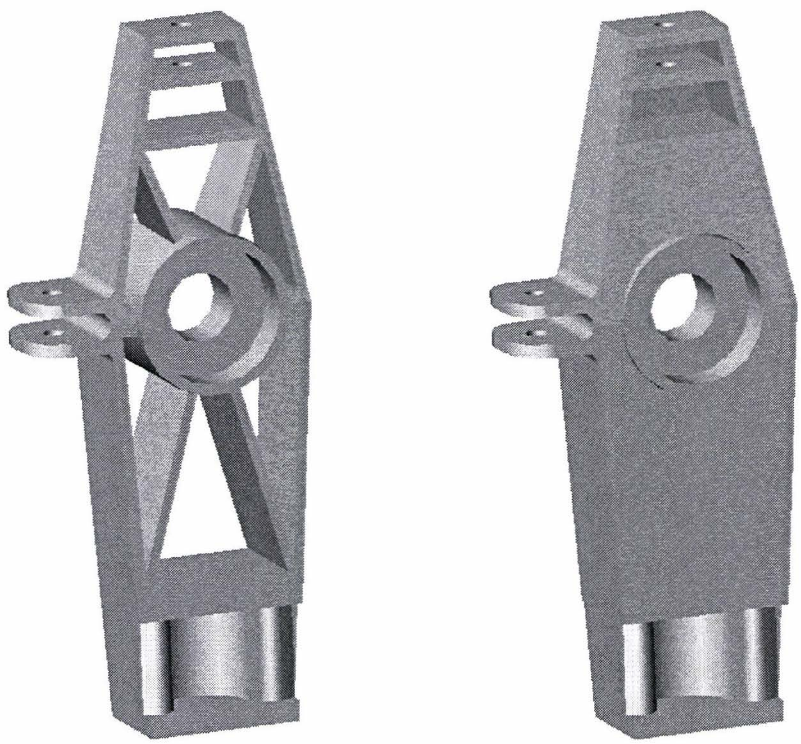


Figure 5.8: Rear upright a) plate structure b) closed box structure

The pivot point for the lower A-arm will be shaped in the same way as in the front upright. The part above the axle will be chamfered on the front and the back side to provide weight reduction. At the front and the backside a hole with a diameter of 52 mm and a depth of 15 mm has been made in the central cylinder with a diameter of 60 mm. This provides the necessary space for two angular contact ball bearings, which will be used to provide the rotation for the rear drive shaft, with respect to the upright. The necessary space for the aluminium block for the lower A-arm is provided just as for the front upright using the rounded edges, because the front member of the lower A-arm has been placed under a large angle towards the front of the car. This results in a sharp angle beneficial for traction. The closed box structure has been chosen, because the stress will be lower when using the same amount of material. This is shown in appendix P.

The mutual A-arm positions have been guaranteed using the uprights and the movement of the A-arm will be guaranteed using rod ends and axial spherical plain bearings. An other suspension part is the monoshock. This part will be used to provide the necessary wheel travel.

5.2: Monoshock

Normally for the suspension of a race car two coil over shock absorbers will be used, which will be connected to a push- or a pull-rod using a rocker. These shock absorbers will be placed vertically along the side of the chassis, on the in- or the outside, or horizontally below or above the legs of the driver. The shock absorbers will be connected to the chassis to support the forces, which will be guided in to the shock absorbers. The consequence of this

layout is that, when different forces act on the left and the right side of the car, the forces on the shock absorbers will cause torsion in the chassis.
When using a monoshock in horizontal position right between the left and the right rocker the monoshock will not rigidly be connected to the chassis. In that way zero roll stiffness is generated allowing this property to be set independently by a separate system, which will be explained in paragraph 5.3. This makes tuning the car much more straight forward and less "magic". To create the required wheel travel for the Formula Student car a monoshock has to be chosen with the right properties. The required properties will be explained in paragraph 5.2.1.

5.2.1: Monoshock requirements

The choice for a monoshock has been made considering the weight of one shock absorber, the compactness of one unit and the simplicity with respect to the adjustments of the suspension. Placing the monoshock between the two rockers on either side of the chassis makes it possible to avoid horizontal forces from the pushrods into the chassis. The forces from the right and the left pushrod compensate each other and will cause a zero resultant horizontal force.

For the monoshock a Fox 4 grip coil-over spring-damper will be used to provide the necessary wheel travel for both the front and the rear suspension. For the static impression one third of the wheel travel of the car will be used. Using the dimensions of the chassis and the position of the rockers a build in length of 380 mm is available to place the coil over spring damper. Therefore a damper with a length of 400 mm will be used, which requires a static impression of 20 mm for the used spring. This means that the movement of one pushrod, which has to convert the wheel travel of 17 mm under static impression, has to result in 10 mm spring travel. Adding both sides 20 mm will be reached. Using a linear spring-damper will cause a total spring travel of 30 mm when, during braking, the wheel travel at the front suspension will be 25,5 mm. For the last 8,5 mm of the wheel travel 5 mm on each side has to be left in the spring travel. This results in a total spring travel of 60 mm (20 + 30 + 10). This required spring travel will lead to the stiffness of the necessary spring. This stiffness has to be created in choosing the length of the spring (the spring will be shorter than the length of the damper) and the dimensions of the spring itself: the wire diameter and the diameter of the spring coil.

To convert the wheel travel into a stroke on the spring-damper a pushrod and a rocker will be used. The properties of those parts of the suspension will be explained in paragraph 5.2.2.

5.2.2: Pushrod and rocker

The pushrod has to connect the lower A-arm to the monoshock. Therefore a tube with the same section as the wishbones for the A-arms will be used. On one side the pushrod is connected to the flanges of the aluminium block of the lower A-arm using a M8 close tolerance bolt and nut and on the other side the pushrod will be connected to a rocker. This rocker has to transpose the movement of the pushrod, and thus the required wheel travel, on to the monoshock. The necessary rocker ratio will be determined by the position of the pivot points in the rocker. The rotation point for the rocker for the front suspension will be mounted under the lowest sloped sandwich panel above the legs of the driver. Therefore an aluminium block will be shaped which will be connected to the side panels and the lower sloped panel.

Using a needle bearing, the frictionless rotation of the rocker, which is about 25 degrees, will be guaranteed. A 2D layout of the pushrod and the rocker is shown in figure 5.9.

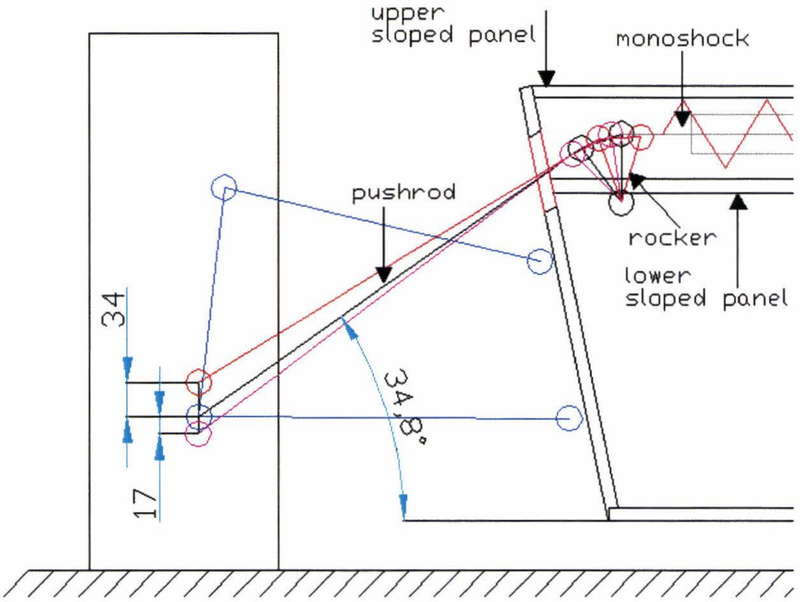


Figure 5.9: Pushrod and rocker motion

The ratio for the rocker is determined by the wheel travel required for the Formula Student car and the properties of the monoshock used. In figure 5.9 the three different situations are shown. The black line is the position of the pushrod under static load. The position of the rocker and its pivot points are shown in black too. For the maximal bump, 34 mm, the position of the pushrod and the rocker has been drawn in red and for the maximal droop, 17 mm, the situation has been drawn in purple. The monoshock will be sandwiched in the rocker just like the pushrod. This monoshock will be mounted between the two sloped panels above the legs of the driver.
The conversion from the wheel travel to the spring travel has been made using a pushrod and a rocker. The spring stiffness will be determined using the static and the dynamic forces on the front wheels. This will be done in paragraph 5.2.3 for both the front and the rear spring stiffness.

5.2.3: Spring-damper stiffness

For the calculation of the spring stiffness the required ratio between the wheel travel and the spring travel are important. With two different positions, the static impression and the static plus the dynamic impression during braking, the required stiffness can be determined. The normal force on one front tire is known during static impression. During braking this force will become higher because of the longitudinal load transfer towards the front wheels. These two different forces have to result in two different strokes on the spring-damper. When these two situations for the wheel travel will be displayed in a diagram the required stiffness at the wheels can be calculated by measuring the steepness of the line between those two points. Figure 5.10 shows these two points for the front suspension.

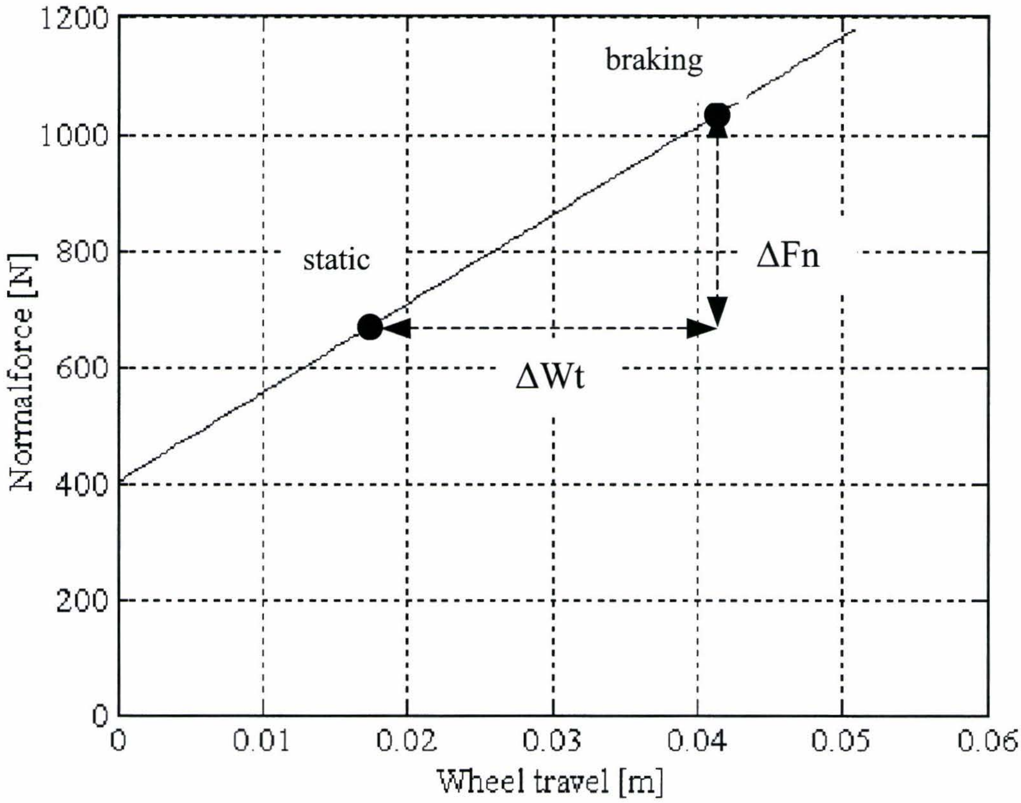


Figure 5.10: Front wheel stiffness

In figure 5.10 the stiffness can be calculated using the slope of the line. The line doesn't go through point (0,0) which means that a fore pressure on the wheels will be necessary. The required stiffness at the front wheels can be calculated using equation 5.3.

$$C_{wheel,front} = \frac{\Delta Fn}{\Delta Wt} = \frac{Fn_{brake} - Fn_{stat}}{Wt_{brake} - Wt_{stat}} \qquad \text{(Equation 5.3)}$$

$$C_{wheel,front} = \frac{1052 - 662}{0{,}0425 - 0{,}017} = 1{,}53 \cdot 10^4 [N/m]$$

The stiffness for the monoshock can be calculated by multiplying the stiffness at the wheels with the square of the ratio of the rocker, which connects the pushrod with the monoshock. The ratio of the rocker will be calculated using equation 5.4.

$$i_{rocker} = \frac{\Delta Wt}{\Delta St} = \frac{Wt_{brake} - Wt_{stat}}{St_{brake} - St_{stat}} \qquad \text{(Equation 5.4)}$$

$$i_{rocker} = \frac{0{,}0425 - 0{,}017}{0{,}025 - 0{,}01} = 1{,}7$$

The necessary spring stiffness will be the wheel stiffness multiplied with the square of the rocker ratio. Therefore equation 5.5 will be used.

$$C_{spring,front} = i_{rocker}^{\ 2} \cdot C_{wheel,front} \qquad \text{(Equation 5.5)}$$

$$C_{spring,front} = 1{,}7^2 \cdot 1{,}53 \cdot 10^4 = 4{,}42 \cdot 10^4 [N/m]$$

For the rear spring stiffness this calculation has been made in appendix Q, using the same method, which results in a spring stiffness of $C_{spring,rear} = 5{,}23 \cdot 10^4 [N/m]$. Now the required spring stiffness is known the dimensions of the spring can be calculated. The volume of the spring can be calculated according the amount of energy that has to be generated by the spring during braking. This will be calculated using equation 5.6.

$$E_{spring} = \frac{1}{2} \cdot C_{spring,front} \cdot \Delta St^2 \qquad \text{(Equation 5.6)}$$

The amount of energy that can be stored in a spring depends on the properties of the used materials and the volume of the spring. This energy can be calculated using equation 5.7.

$$E_{spring} = \frac{\mu_{spring} \cdot V_{spring} \cdot \tau_{steel}^{2}}{2 \cdot G_{steel}} \qquad \text{(Equation 5.7)}$$

Combining equation 5.6 and 5.7 will be equalled the necessary spring volume can be calculated using equation 5.8, considering $\tau_{steel} = 6{,}0 \cdot 10^8$ N/m^2 and $G_{steel} = 8{,}0 \cdot 10^{10}$ N/m^2.

$$\frac{1}{2} \cdot C_{spring,front} \cdot \Delta St^2 = \frac{\mu_{spring} \cdot V_{spring} \cdot \tau_{steel}^{2}}{2 \cdot G_{steel}} \qquad \text{(Equation 5.8)}$$

$$V_{spring} = \frac{C_{spring,front} \cdot (St_{brake} - St_{stat})^2 \cdot G_{steel}}{\mu_{spring} \cdot \tau_{steel}^{2}}$$

$$V_{spring} = \frac{4{,}42 \cdot 10^4 \cdot (0{,}025 - 0{,}01)^2 \cdot 8{,}0 \cdot 10^{10}}{0{,}5 \cdot \left(6{,}0 \cdot 10^8\right)^2} = 4{,}42 \cdot 10^{-6} [m^3]$$

To reach this required volume different parameters can be changed. The wire diameter, the length of the spring, the diameter of the used spring coils and the number of active coils are all parameters, which determine the stiffness and the volume of the spring. The material used, steel, has to be positioned in such a way that the required material volume and the required stiffness will be reached. The optimization for this process will be done using the fact that the minimal inner diameter of the used spring will be 50 mm to have enough space for the damper and the fully loaded spring length will be 120 mm with the desired spring travel of 60 mm. Using an optimization program to calculate compression springs, see appendix R, a spring with the properties shown in table 5.1 will be used for the front suspension.

Mean spring diameter	58,09 [mm]
Wire diameter	8 [mm]
Outer / inner spring diameter	66,09 / 50,09 [mm]
Number of active coils	4,5 [-]
Free spring length	160,51 [mm]
Spring stiffness	44,14·10^3 [N/m]
Spring weight	0,48 [kg]

Table 5.1: Front suspension spring properties

For the rear suspension the spring will be determined in the same way, because there will be used a monoshock too. Because the load on the rear tires is higher than on the front tires the stiffness of the spring has to be higher than the stiffness for the front suspension, which has been calculated in appendix Q. Using the same optimization program as for the front spring (appendix S) the spring with the properties shown in table 5.2 will be used for the rear suspension.

Mean spring diameter	54,49 [mm]
Wire diameter	8 [mm]
Outer / inner spring diameter	62,49 / 46,49 [mm]
Number of active coils	4,9 [-]
Free spring length	161,27 [mm]
Spring stiffness	$52{,}32 \cdot 10^3$ [N/m]
Spring weight	0,46 [kg]

Table 5.2: Rear suspension spring properties

For this spring the minimal inner diameter has been chosen to be 45 mm, with a fully loaded length of 120 mm. Considering the fact that the build in length for both the front and the rear monoshock will be 380 mm, the spring will only cover a part of that length. The type of monoshock that will be used is a coil-over spring-damper, which is shown in figure 5.11.

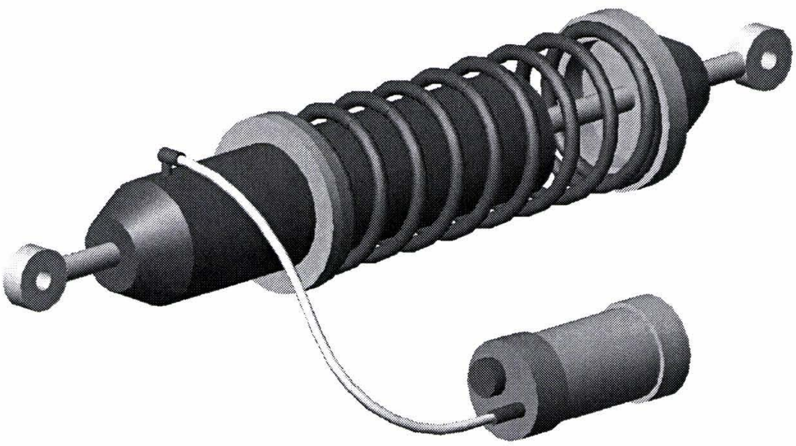


Figure 5.11: Coil-over spring-damper with oil reservoir

The prestress on the monoshock is adjustable, because the build in length of the spring is variable. Using the two screws at the left side of the spring, see figure 5.11, the prestress can be adjusted. Also the pressure of the fluid for the damper, which is stored in a separate reservoir, is changeable. The damping can be adjusted by adjusting valves in this reservoir. Between the two sloped panels, where the front monoshock will be placed, is not much space. Therefore it is handy that the reservoir is seperate. Above the monoshock an anti-roll mechanism will be placed to create an independent adjustment for the monoshock stiffness and the anti-roll stiffness. This mechanism will be described in paragraph 5.3.

5.3: Anti roll mechanism

The monoshock has been used to create the required wheel travel. This means that during braking the dive of the car will be suppressed using the monoshock. When a car drives through a bend, the movement of the car, rotation of the chassis around the roll axis, has to be suppressed too. This is called the anti-roll stiffness. By placing the monoshock horizontally between the two front rockers the roll of the car won't be hampered. This is preferably done using an individual anti-roll mechanism. For the adjustments to the anti-roll stiffness and the bump stiffness it is desirable to separate the two different mechanisms. Therefore a special anti-roll mechanism has been designed.

For the desired wheel travel and spring travel, the dimensions of the monoshock and spring have been determined. For the forces on the monoshock and the anti-roll mechanism it is desirable that both mechanisms be placed right between the rockers, so that the forces will be in one plane. For that reason the choice has been made to place the anti-roll mechanism above the monoshock. For the mounting on the chassis this means that the monoshock will be placed right between the two sloped sandwich panels above the legs of the driver and the anti-roll mechanism will be placed on top of the upper panel right above the monoshock. This means that the anti-roll mechanism can now also be placed right between the two rockers. Therefore the rockers have to be executed with four pivot points: one for the monoshock, the anti-roll mechanism, the pushrod and the chassis rotation point beneath the lower sloped panel. The rocker that has been designed for that purpose is shown in figure 5.12.

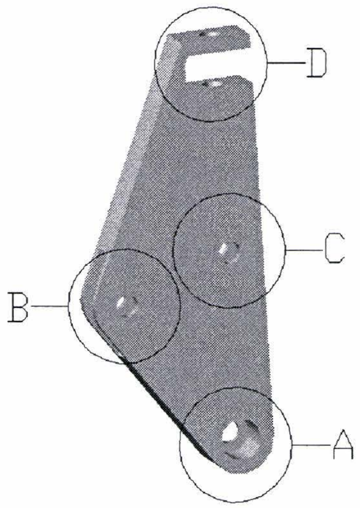


Figure 5.12: Front suspension rocker

The desired ratio between the pushrod and the monoshock of 1,7 will be determined by the distances between point A and B and the distance between point A and C, considering the angle of the pushrod (34,8 degrees). The position of point D has been determined by the diameter of the monoshock and the height of the anti-roll mechanism. For the anti-roll mechanism it's important that during bump it won't add extra bump stiffness to the stiffness of the monoshock and during roll it's important that the monoshock won't add any stiffness to the anti-roll stiffness. Therefore the choice for a rotation system has been made, which has been analyzed. The extreme positions during bump and roll have been drawn in appendix T. The stiffness for the anti-roll will be obtained from Belleville spring washers. This stiffness can easily be changed by adding, removing or inverting a spring. The mechanism used will be attached to the rocker in point D. Figure 5.13 shows the top view of the mechanism used.

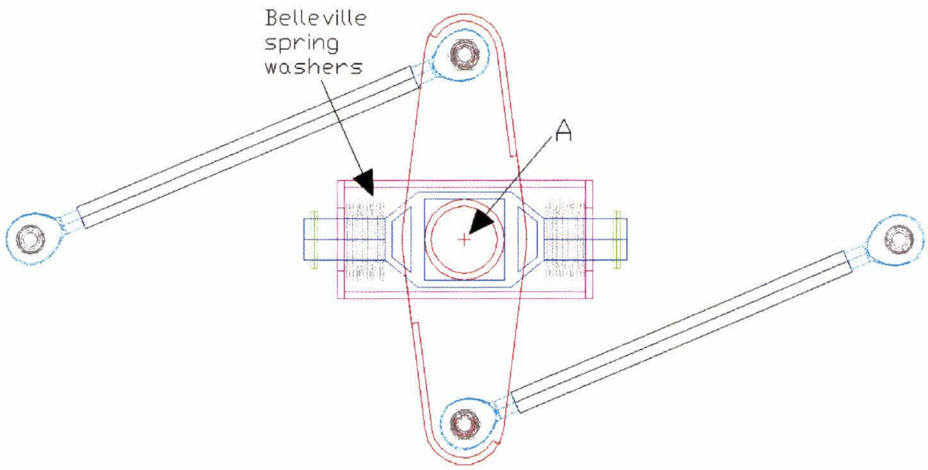


Figure 5.13: Working anti-roll mechanism

During bump the two tubes provided with two rod ends each will be pushed towards the centre of the car, which results in a rotation of the red part around point A. This rotation won't have any effect on the Belleville spring washers (grey parts), because the position of the blue part won't be changed. During roll the two tubes will be pushed in one direction, which results in a translation of the red and the blue part. This translation will be transmitted onto the springs. This stroke of the blue part creates the anti-roll stiffness. A 3D view of the anti-roll mechanism is shown in figure 5.14.

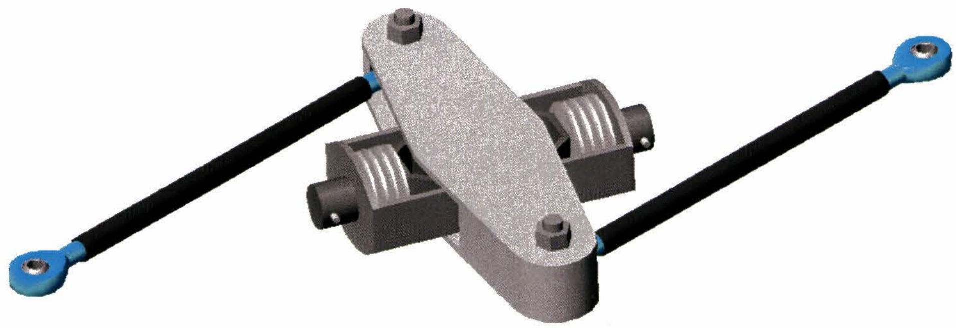


Figure 5.14: Anti-roll mechanism for front suspension

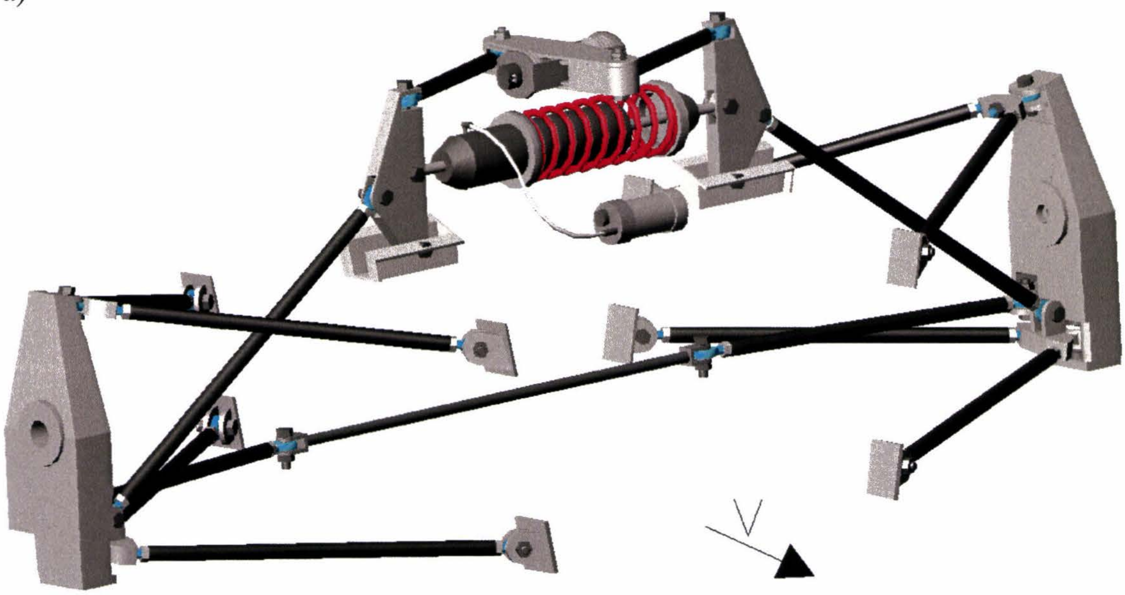
The bump and the roll stiffness have been separated. Only during heavy roll, more than a 10 mm stroke on the monoshock, bump and roll are not completely separated, because the pivot point of the monoshock in the rocker only can translate 10 mm to the outside of the car and 20 mm towards the centre of the car. During roll this can result in a stroke of 10 mm on the spring of the monoshock. Therefore the roll stiffness will be preset in such a way that this 10 mm won't be reached.

5.4: Overall suspension layout

The layout of the anti-roll mechanism for the rear suspension is completely the same, also the same shaped rockers, but the only difference is that the rear rockers will be placed under an angle of 21 degrees with the z-axis, to avoid contact between the rear pushrods and the drive shafts.

All the parts for the front and the rear suspension have been designed. When combining these parts figure 5.15a and figure 5.15b can be created.

a)



b)

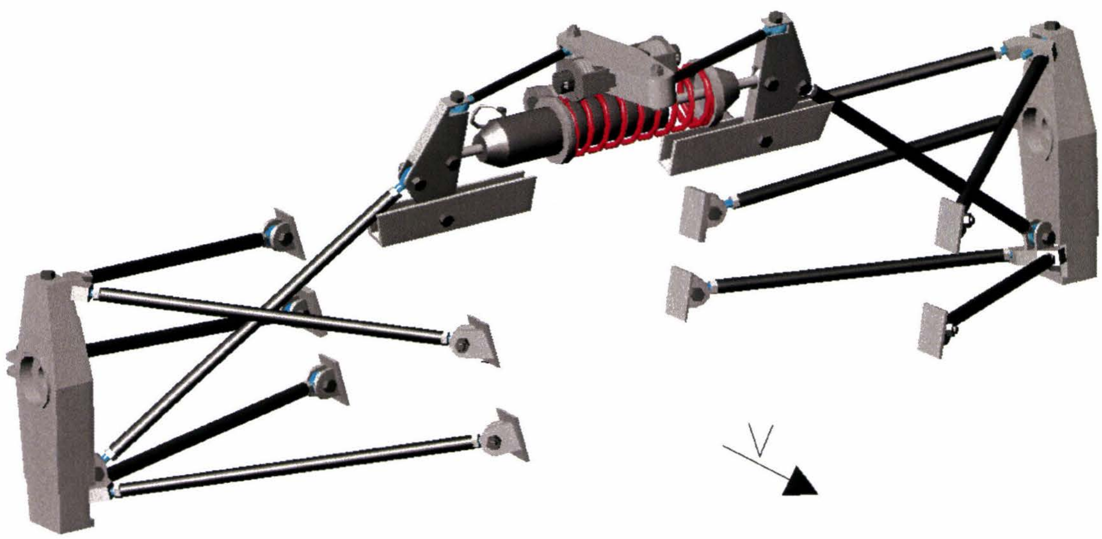


Figure 5.15: Suspension layout a) front suspension b) rear suspension

Considering figure 5.15a and figure 5.15b the layout for the front and the rear suspension differ at some points. The front tie rod is movable and the rear tie rod is fixed. The angles for the A-arms are all different but the principle for the layout of the A-arms is all the same. The uprights are different, because the bearings used for the front and the rear axle have been placed differently. Furthermore the principles are all the same but the layout of the parts for the front and the rear suspension are different, considering the angles.

After designing the suspension considering the shape of the chassis, the drive train of the car has to be integrated. During the design of the chassis the position of the engine has already been fixed, but the other parts for the drive train are still unknown. These parts and the layout of the drive train will be explained in chapter 6.

Chapter 6: Drive train

The Formula Student car drive train has to transmit the power of the engine onto the rear wheels, because the car will be rear wheel driven. The drive train will consist of an engine, a transmission from the engine towards a differential and drive shafts towards the rear wheels. The design of these parts and their connection will be described in this chapter. First of all the engine has to be chosen.

6.1: Engine

For the powering of the car an engine has to be used. According the requirements a swept volume of the engine is not allowed to exceed 610 cc. Therefore an engine with 610 cc or less will be used to power the Formula Student car. When selecting the engine the main focus is the reliability and the power. Therefore the choice has been made from a range of 600cc motorbike engines, because they represent the most powerful engine's allowed by the rules. Considering the power, torque and revolutions of a modern super sport engine, it will be slightly over dimensioned for the application for the Formula Student car. This will stimulate reliability.
To make the right choice for the engine several specifications of the different engines have to be compared. The choice for the engine will be made between five different engine types which have been compared in table 6.2. These engines have a lot of properties in common. These properties are shown in table 6.1.

Cycle		4-stroke
Cylinder line up		4 in-line
Cylinder capacity	cm^3	600
Transmission		6-speed
Starter		electric
Carburetion		injection
Cooling		liquid
Valve actuation		DOHC

Table 6.1: Engine properties

Brand		**Suzuki**	**Yamaha**	**Kawasaki**	**Honda**	**Triumph**
Type		GSX-R 600	YZF-R6	Ninja ZX-6 RR	CBR 600 RR	Daytona 600
Max. Power	kW	88,3	86	86	84	78
	at rpm	13000	13000	13000	13000	12750
Max. Torque	Nm	69,6	66,4	65	64	68
	at rpm	10800	12000	12000	11000	11000
Bore	mm	67	65,5	67	67	68
Stroke	mm	42,5	44,5	42,5	42,5	41,3

Table 6.2: Engine comparison

According table 6.2 the different engines have a lot of similarities like the cylinder line up, the carburetion and the cooling. For the Formula Student car the maximal power and the maximal torque of the engine will be an important issue, because the car has to accelerate frequently, when driving over the winding track. Regarding these parameters the Suzuki engine is the most powerful in its class compared to the other engine types. Therefore the Suzuki engine will be used for the Formula Student car. The choice for this engine has been eased as the supplier had been able to sponsor this engine. The engine is shown in figure 6.1.

Figure 6.1: Suzuki GSX-R600

The standard engine has been designed and built by professionals, which makes it well suited for its purpose. The air intake system, the exhaust and the cooling system will be changed to make the engine more suitable for the Formula Student competition. First the redesign of the air intake system will be explained in paragraph 6.1.1.

6.1.1: Air intake system

Considering the Formula Student car an important rule regarding the engine will be the obligatory air restriction. The air intake system has to contain a circular restriction with a maximum diameter of 20 mm. This restriction is required to limit the power capability for the engine. All the air must pass through this restriction, which has to be placed between the throttle and the engine. Therefore a new air intake system has to be designed, because the normal intake system that will be used for a motorcycle won't be allowed for the Formula Student car.

Due to the restriction that has to be used the intake has to be redesigned completely compared to the normal intake system that is used for motorcycle. The restrictor has to be a part of the air intake system, which will lead to the following sequence of: air filter – throttle – restrictor – diffuser – collector – runners – injectors – engine. The properties applied to these parts are shown in figure 6.2.

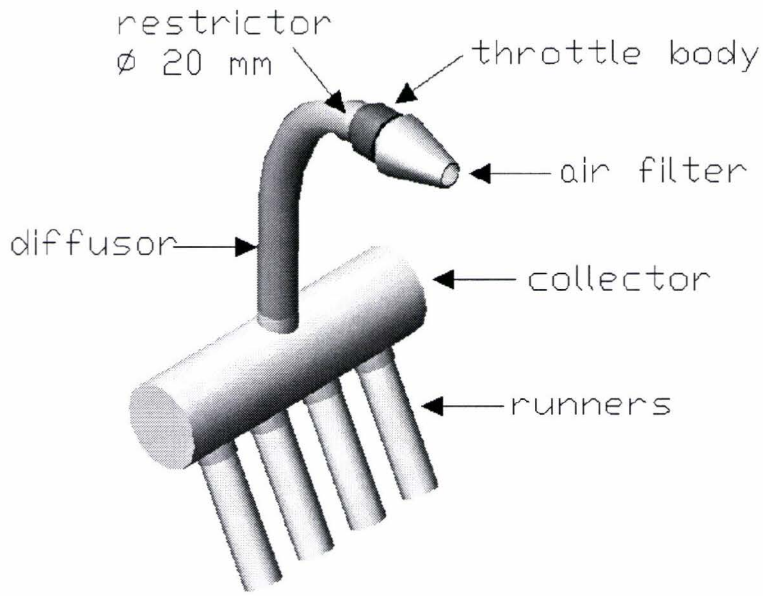


Figure 6.2: Redesign air intake system

In the curved tube the restrictor has been integrated and the inside of the tube has to contain gradual transitions to improve the air flow in the air intake system. The position of the air filter has been chosen right above the helmet of the driver. Using a funnel-shaped cylinder the necessary amount of air will be sucked in. The area of that cylinder will be determined using the assumption that the car drives at a speed of 20 m/s and the air will run through the restrictor at the sound velocity (340 m/s). Using equation 6.1 the area can be calculated.

$$\Phi_{cylinder} = \Phi_{restrictor} \qquad \text{(Equation 6.1)}$$

$$A_{cylinder} = \frac{V_{restrictor}}{V_{cylinder}} \cdot A_{restrictor}$$

$$A_{cylinder} = \frac{340}{20} \cdot \left(\frac{\pi}{4} \cdot 20^2 \right) = 5340[mm^2]$$

This funnel-shaped cylinder will have an elliptic area. The shape and the dimensions of this area have been drawn in figure 6.3.

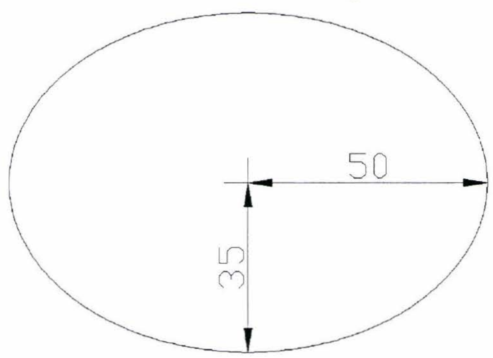


Figure 6.3: Area for air filter

For the throttle a butterfly throttle valve is selected for its simplicity, robustness, reliability and minimal flow disturbance in full throttle situation. The throttle diameter (38 mm) is set to be small enough to be compatible with the restrictor, but without acting as a restrictor itself.

The diffuser is in effect a Laval tube that maximizes the effective flow area of the restrictor. After the restrictor a diffuser is mounted at a 7 degrees diverging angle to prevent the air foil releasing the side wall. This will slow the air down and increase static air pressure, which lowers frictional rates and air turbulence. Onto the collector (Ø100 mm with a length of 350 mm) four intake runners will be connected, which run to each cylinder. Inside the collector the intake runners have a bell mouthed shape to lower the discharge coefficient and maximize the effective flow area. The fuel injectors will be mounted at the end of the runners in the direction of the intake valves. The spray ends up on the relative hot valve, which will enhance the vaporization of the fuel. In all connections of pipes to plenums the couplings are made as smoothly as possible to prevent turbulence.
After considering the redesigned air intake system a modified exhaust has to be designed, because the available space in the Formula Student car is different than for a normal motorcycle.

6.1.2: Exhaust

Just like the air intake system the exhaust has to be modified for the Formula Student car. On a motorcycle the exhaust has to be build along other parts towards the rear of the bike, which will lead to a complex system. For the Formula Student car the exhaust has to be adjusted to the shape of the chassis, which will result in a modification to a standard exhaust. The exhaust for a normal motorcycle is specially tuned for that engine type. Therefore it is desirable to use a standard exhaust and to keep it intact as much as possible. The four tubes of the exhaust run underneath the engine along the oil sump, two on either side. These two tubes will be connected to one another from two to one using a reducing piece. This will happen on both sides of the engines. The remaining two exhaust pipes will then be connected using a reducing piece so that one large exhaust remains. Onto this tube a muffler will be mounted, which is required to keep the noise of the car below 110 dB. The design for the new exhaust is shown in figure 6.4.

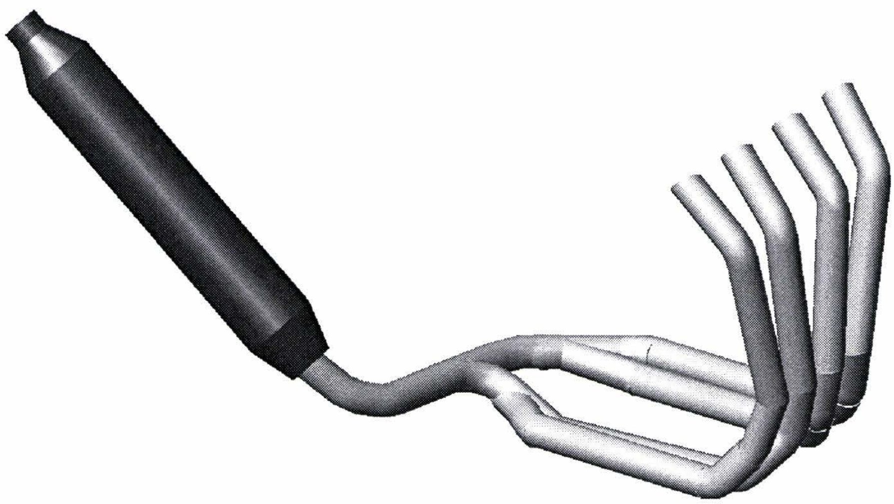


Figure 6.4: Exhaust

The layout of the runners of the exhaust has been determined using the shape of the chassis and the availability of space. The length of the runners has been determined by calculating the volume of the required air flow for the stroke of the engine. According this length the runners come together behind the sump of the engine and from that point the collecting tube goes

towards the right side of the car through the side sandwich panel. This has been done because the muffler cannot be placed through the rear frame due to lack of space. The differential and the drive shaft with the brake disc don't allow the exhaust to run towards the rear of the car. The muffler has been placed at the right sight of the car onto the sandwich panel through the rear suspension. Therefore the required wheel travel of 50 mm has been checked, so that contact between exhaust and suspension will be avoided. Furthermore the whole exhaust will be shielded, using insulating tape, so that the heat won't cause any injuries for the driver or for someone who is working near the exhaust.
Connecting the redesigned air intake system and exhaust onto the engine, figure 6.5 can be made, which shows these new parts attached to the engine.

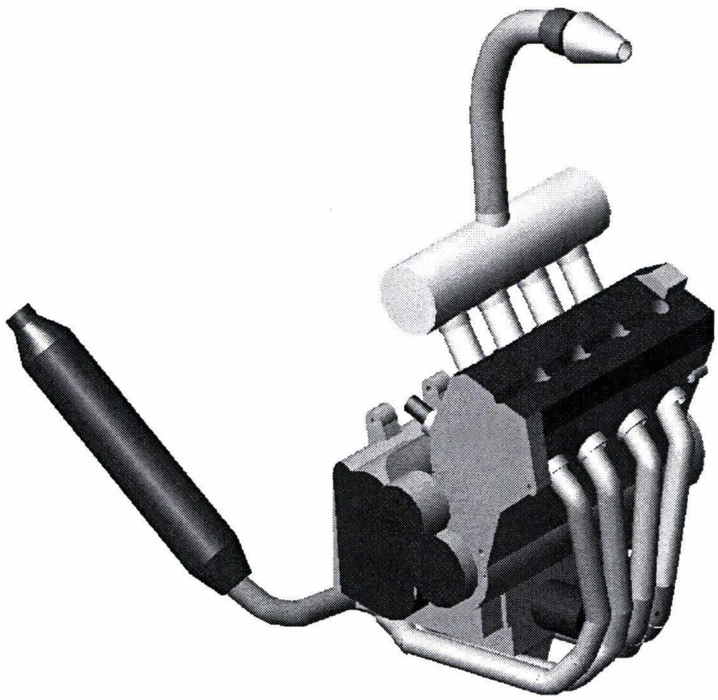


Figure 6.5: Overall engine layout

For the overall centre of gravity it is important to place the engine as low as possible in the chassis. Therefore the choice has been made to apply a ground clearance of 50 mm to the chassis, which means that the car with driver will have a ground clearance of 50 mm in static condition, the parked situation. This clearance will also count for the engine. The plug for the oil sump is the lowest point of the engine, which has been placed 50 mm above the ground.

For the cooling of the engine a radiator will be used, which will be placed upon the braces of the main roll hoop. The attachment of the radiator onto the braces has to be disconnectable, because the roll hoop and the braces have to be removed when the engine has to be lifted out of the chassis for maintenance. Underneath the radiator an aluminium plate with a thickness of 1 mm will be attached onto the braces to guide the necessary air flow through the radiator. In paragraph 6.1.3 the position and the requirements for the fuel tank will be explained.

6.1.3: Fuel tank

The fuel for the engine will be stored in the fuel, which has to be placed in the chassis. Therefore some considerations have to be made. According the rules the fuel tank is not allowed to have a bigger capacity than 2 gallon (7,57 litres). The tank used for the Formula Student car will have a capacity of 7,5 litres. For the placement of the fuel tank in the chassis one thing is very important to consider. The mass of the fuel tank is variable, because of the fuel consumption of the car. For short events this influence will be negligible, but for the endurance race the influence of the weight reduction due to fuel consumption will be significant. When the fuel tank is placed far away from the centre of gravity of the car, the drive conditions will be affected. For example the lateral and longitudinal load transfer will change when the tank becomes emptier and emptier. Also the yaw frequency will be affected, because the mass moment of inertia of the car will vary. Therefore the fuel tank has to be placed in the centre of gravity of the car to minimize the influence on the driving conditions. Technically this won't be possible, because the centre of gravity of the car lies just in front of the seat frame. Therefore the choice has been made to place the fuel tank just underneath the sloped part of the seat frame, as close as possible and just below the centre of gravity. Another advantage of that position is that the centre of mass of the fuel tank will be low in the car. The influence of the low centre of gravity is positive for the overall height of the centre of gravity. This height should be as low as possible.

This position for the fuel tank has also been chosen according the rules. According those rules the fuel tank is not allowed to be located outside the major structure of the car, to prevent hazards in case of a roll over or a collision. All parts of the fuel storage and supply system must lie in the surface defined by the top of the roll bar and the outside edge of the four tires. The final position of the fuel tank is shown in figure 6.6.

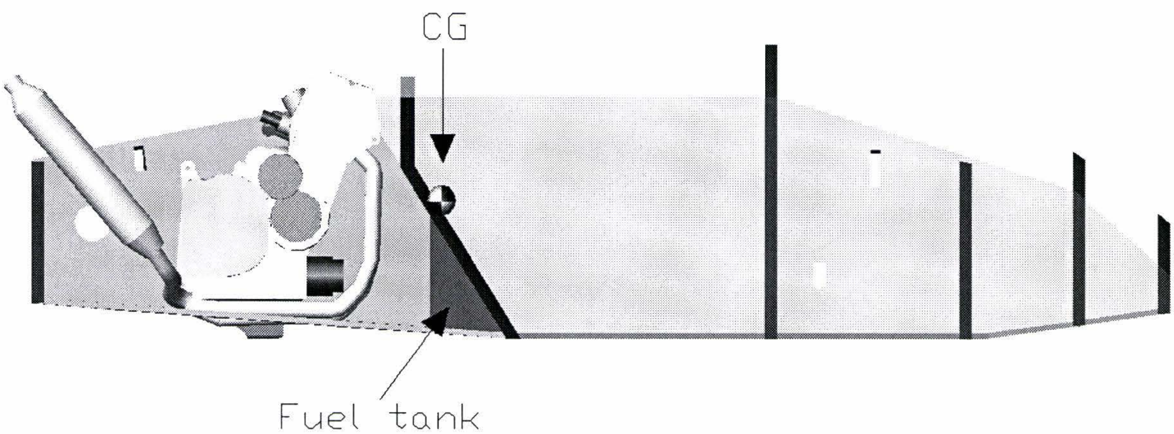


Figure 6.6: Fuel tank position

The fuel tank will be made of fibre reinforced plastic to keep the weight as low as possible and to improve the safety. The shape of the fuel tank will be adjusted to the shape of the seat frame considering the angle of the bottom panel of the car.

The shape and the position of the engine and the shape of the seat frame create a big space between them. In this part of the car the fuel tank and the batteries will be placed to fill up this free space. Furthermore the exhaust of the engine, which will generate a lot of heat, will fill this space. Furthermore it's important that the fuel tank will be screened from the exhaust to improve the fire safety for the driver. Therefore a thin aluminium plate can be used to make a shield from the straight part of the seat frame towards the bottom panel.
Now the engine layout is ready the transmission from the engine towards the rear wheels can be determined. First of all the use and the function of the differential will be explained in paragraph 6.2.

6.2: Differential

When driving through a bend, due to lateral load transfer, an inner wheel can and will be lifted off the ground. When a rear will is lifted off the ground it will spin up, because the lost friction with the road. This effect will cause a reduction of the power at the outside wheel, because most of the power from the engine will go to the lifted wheel. That is the way with the least resistance. To prevent this effect a differential will be used that divides the engine power. In a corner this means that the biggest part of the total power of the engine has to be guided to the wheel on the outside.

In short terms the functions for a differential can be divided in three parts

- To transmit the engine power to the wheels
- To act as the final gear reduction in the vehicle, amplifying the torque of the transmission before to the wheels
- To transmit the power to the wheels, while allowing them to rotate at different speeds.

Car wheels rotate at different speeds, when turning. In figure 6.7 is shown that each wheel travels a different distance through a bend, and that the inside wheels travel a shorter distance than the outside wheels. Since speed is equal to the distance travelled, divided by the time it takes to go that distance, the wheels that travel a shorter distance travel at a lower speed. Also the front wheels travel a different distance than the rear wheels.

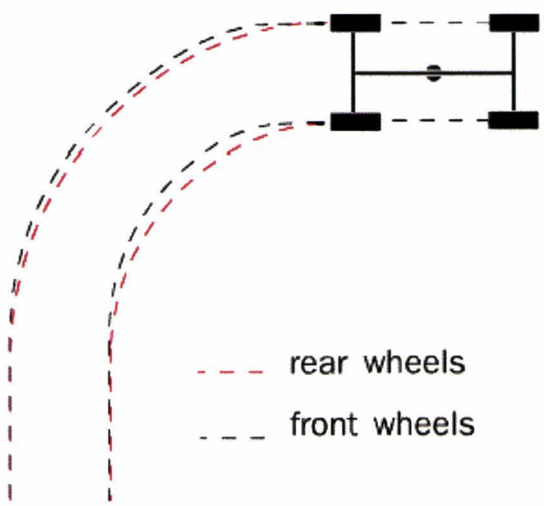


Figure 6.7: Wheel paths

For the non-driven wheels, the front wheels on a rear-wheel driven car, the difference in speed is not an issue. There is no connection between them, so they rotate independently. The driven wheels however are linked together so that a single engine and transmission can turn both wheels. If the car did not have a differential, the wheels would have to be locked together, forced to rotate at the same speed. This would make turning difficult and hard on the car: for the car to be able to turn, one tire would have to slip. With modern tires and concrete roads a great deal of force is required to make a tire slip. That force would have to be transmitted through the axle from one wheel to another, putting a heavy strain on the axle components. To avoid those problems a differential will be used. This will be a Torsen differential.

6.2.1: Torsen differential

The Torsen (**Tor**que **Sen**sing) differential is a purely mechanical device; it has no electronics, clutches or viscous fluids. The working of a Torsen differential is shown in figure 6.8a en 6.8b. In figure 6.8a the drive shafts to the rear wheels are shown. Figure 6.8b shows the different parts that have to be used in the Torsen differential.

a) b)

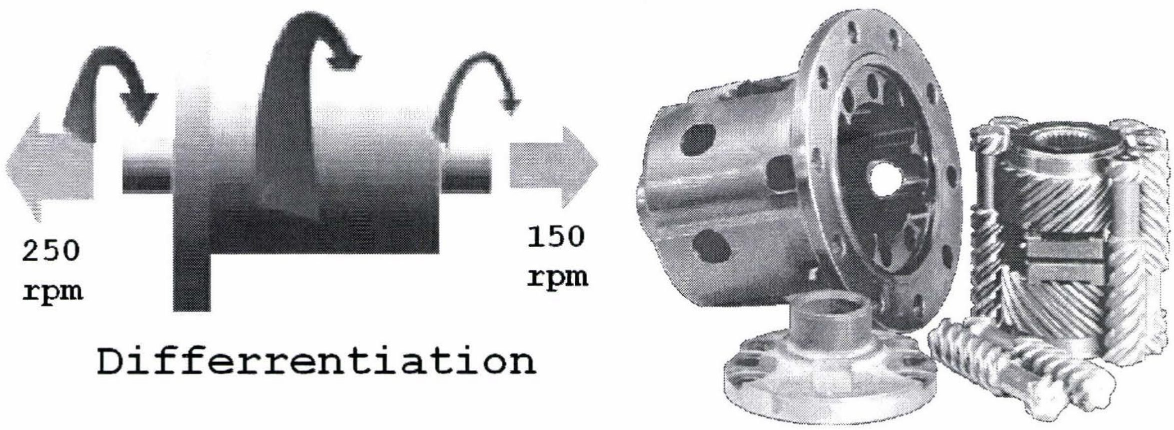


Figure 6.8: Torsen differential a) working b) differential parts

The Torsen differential works as an open differential when the amount of torque going to each wheel is equal. As soon as one wheel starts to lose traction, the difference in torque causes the gears in the Torsen differential to bind together. The design of the gears in the differential determines the torque bias ratio. The Torsen differential, which will be used for the Formula Student car is designed with a 3:1 bias ratio, it is capable of applying up to three times more torque to the wheel that has the highest traction, because the normal force on that wheel is much higher than for the wheel with the lowest tarction. For example when the engine generates 1000 rpm for the differential, one wheel will be able to rotate at a speed of 1500 rpm, while the other is rotating at 500 rpm.

Now the choice has been made to use a Torsen differential the next thing to do is to determine the transmission from the engine towards the differential.

6.2.2: Transmission

To create the transmission from the engine towards the differential a single-row chain will be used. Therefore a sprocket on the final drive shaft of the engine, a chain wheel and a chain will be necessary to create this transmission. The number of teeth on the chain wheel and the sprocket depends on the ratio needed for the Formula Student car. The Suzuki motorcycle, where has a top speed around 250 km/h, which is far above the desired top speed for the Formula Student car. For this car the top speed will be aimed at 120 km/h because the maximal reachable speed will be just over 100 km/h due to the short straights. Therefore a different ratio has to be used. Furthermore due to the obliged restriction of Ø20 mm for the intake manifold the maximum number of revolutions won't be 12700 rpm, but only 9000 rpm, which means that the top speed of the Formula Student car has to be reached at 9000 rpm. For the ratio used at a motorcycle the speed that will be achievable with the restriction can be determined with equation 6.2.

(Equation 6.2)

$$V_{GSX} = \frac{n_{res}}{n_{\max}} \cdot V_{\max}$$

$$V_{GSX} = \frac{9000}{12700} \cdot 250 \approx 178[km/h]$$

The sprocket and the chain wheel that will be used for a standard Suzuki motorcycle have respectively 16 and 45 teeth. With this gear ratio the engine can reach a speed of 178 km/h at an engine speed of 9000 rpm. The speed of 178 km/h has to be reduced to 120 km/h for the Formula Student car. Using the same standard sprocket with 16 teeth will mean that the chain wheel has to be bigger. The required number of teeth can be calculated with equation 6.3.

(Equation 6.3)

$$N_{FS} = \frac{V_{GSX,9000}}{V_{FS,9000}} \cdot N_{GSX}$$

$$N_{FS} = \frac{178}{120} \cdot 45 \approx 68$$

The diameter of a chain wheel with 68 teeth is 360 mm, assuming the use of a chain with a pitch of 5/8" (15,875 mm). The available space for the chain wheel on the rear axle is not that big, because with that diameter it will interfere with the bottom panel and the rear frame. For the stiffness of the chassis it is important that the upper and lower member of the rear frame will be rigid, because the rear wishbones of the rear suspension are connected to that frame. Therefore only a slot in vertical direction in the rear frame will be permitted, but the chain wheel with a diameter of 360 mm will be too big. Therefore the sprocket has to be smaller, so that the chain wheel also can be chosen smaller to retain the same gear ratio. For the Suzuki motorcycle sprockets with 14, 15, 16 and 17 teeth are common. For the Formula Student car the sprocket with 14 teeth will be used to reduce the size of the chain wheel. To create the required gear ratio with a chain wheel that is common used for this application a chain wheel with 57 teeth will be used. This will result in a chain wheel diameter of 288 mm, which will just fit between the rear frame and the bottom panel. The chain wheel will be attached to the differential, which is shown in figure 6.9.

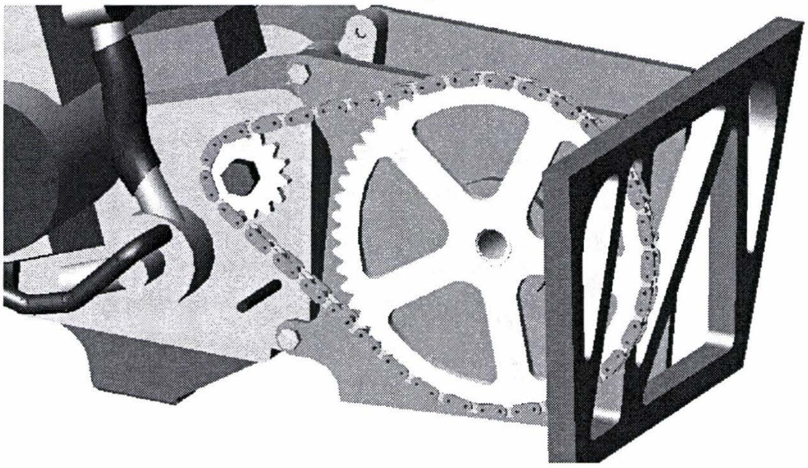


Figure 6.9: Chain wheel and sprocket

Figure 6.9 shows the slot in the rear frame. The chain wheel runs right through that frame and therefore a chain guard will be attached to the backside of the rear frame to prevent contact with the rotating chain wheel, which will do harm, when rotating at a speed of 1300 rpm.

To tension the chain a pulley will be pushed against the chain (not drawn in figure 6.9). This pulley will be placed under the chain and its position will be adjustable so that the tension on the chain will always be the same, when the chain stretches due to wear. For the use of a tensioner pulley the chain has to be a little bit longer than drawn in figure 6.9. For the circumscribed angle of the chain around the sprocket this will be very useful, because then more teeth of the sprocket will be used for the transmission. This will reduce the forces on the individual teeth of the sprocket, because the chain force will be divided over more teeth.

On to the chain wheel the differential will be mounted. The suspension for the differential will be discussed in paragraph 6.2.3. Therefore the two aluminium plates, which fix the rear side of the engine, will be used.

6.2.3: Mounting differential

For the suspension of the rear part of the engine two aluminium plates will be used to support the engine using the rear frame and the bottom panel. The Torsen differential will be mounted right between those two plates at the height of the centre of the rear wheels. The diameter of the rear tires is 567 mm, which causes a height of 283,5 mm for the drive shafts and the Torsen differential.

For the bearing of the differential in the two aluminium plates deep groove ball bearings will be used. They are suitable for very high speeds and they are robust in operation, requiring little maintenance. Two bearings will be mounted in the two aluminium plates to prevent moments on the bearings, because they will be placed very close to the path of the chain forces onto the chain wheel and the differential. These forces will be guided into the chassis, using the bearings and the two aluminium plates. The differential has to be extended, because the standard length is too short to place it through the two aluminium plates. Therefore on the right side of the car, where the aluminium plate with a thickness of 12 mm is placed, a sleeve with a length of 45 mm will be welded on the existing differential to guide the differential through the two plates.

To prevent the differential from running dry, continuous lubrication of the differential will be necessary. Therefore an aluminium box will be mounted, in two parts, around the differential to prevent the leakage of the lubricant. At the side of the chain wheel the box will be bearing mounted and at the other side the differential itself will be supported. The position of the bearings and the section of the two parts of the aluminium box are shown in figure 6.10.

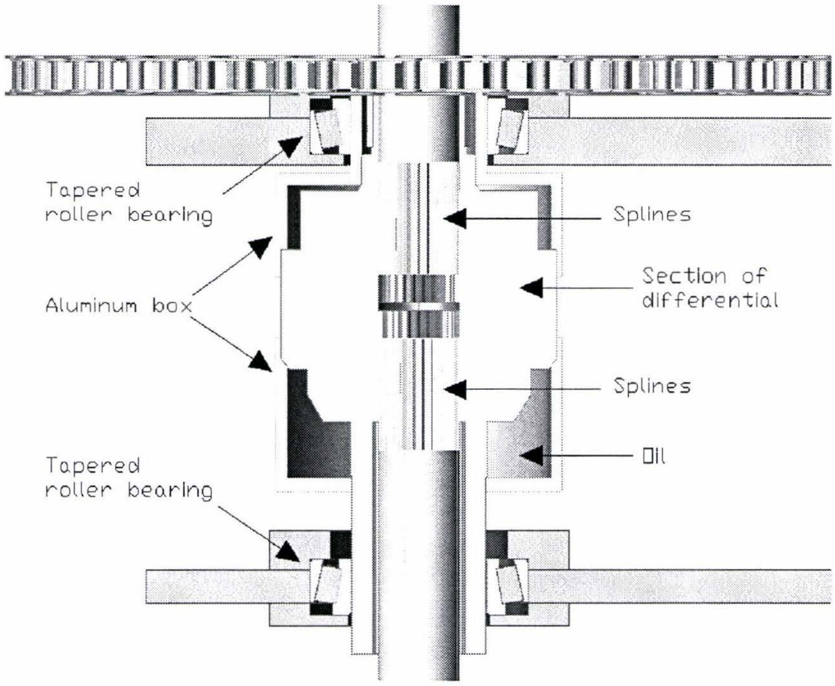


Figure 6.10: Section differential

In figure 6.10 also is shown that the outer rings of the bearing will be fixed using one aluminium flange at the side of the chain wheel and two aluminium flanges at the other side. These flanges will be connected onto the two aluminium plates using four Allen bolts M5.

For a rear wheel driven car drive shafts will be necessary to transmit the power from the engine via the chain and the differential towards the rear wheels. The shape and the layout of these drive shafts will be determined in paragraph 6.3.

6.3: Drive shaft

The forces from the differential towards the rear wheels will be transmitted using drive shafts. For these drive shafts three parts will be important: the connection with the differential, the connection with the wheels and the part between the differential and the wheels. Therefore the drive shaft will be divided in three pieces. First of all the connection with the differential will be described.

6.3.1: Splines

The rear drive shafts will be connected into the differential using splines. With these splines a solid connection can be made, because a large contact area between the drive shaft and the differential will be created. A spline is formed by multiple grooves. Each groove of the spline has its own contact patch with the differential. Adding all these small areas a large contact area between spline and differential will be created. The spline connection is shown in figure 6.11a and the drive shafts, provided with splines, will be connected in the differential, which is shown in figure 6.11b.

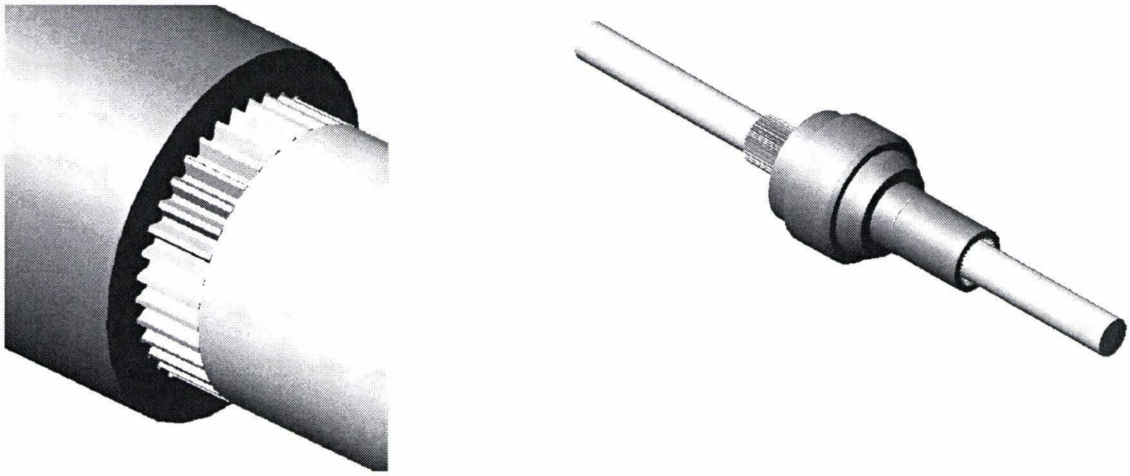


Figure 6.11: Spline connection a) shape spline b) drive shafts in differential

Figure 6.11a shows that on one side the grooves will be cut on the outside of the shaft and at the other side the grooves will be internally cut. The spline connection in the differential will be made in the widest part, so that the drive shafts will go through the whole differential. The number and the depth of the grooves determine the contact area between shaft and differential.
For the suspension movement, the 51 mm wheel travel for the rear wheels, the drive shafts have to be flexible to accept this movement. Therefore the shafts will be cut in three parts. These three parts will be connected to each other using flex plates.

6.3.2: Flex plate

Two drive shafts will connect the differential with the rear wheels. Because the Formula Student car has a required wheel travel of 51 mm the suspension and thus the drive shafts has to be able to follow that wheel travel. Therefore a flexible coupling has to be used in the drive shafts. A flex plate will be useful for that application, because the angles that have to be made are relatively small. The wheel travel of 51 mm over a length of 400 mm will cause the angles shown in figure 6.12.

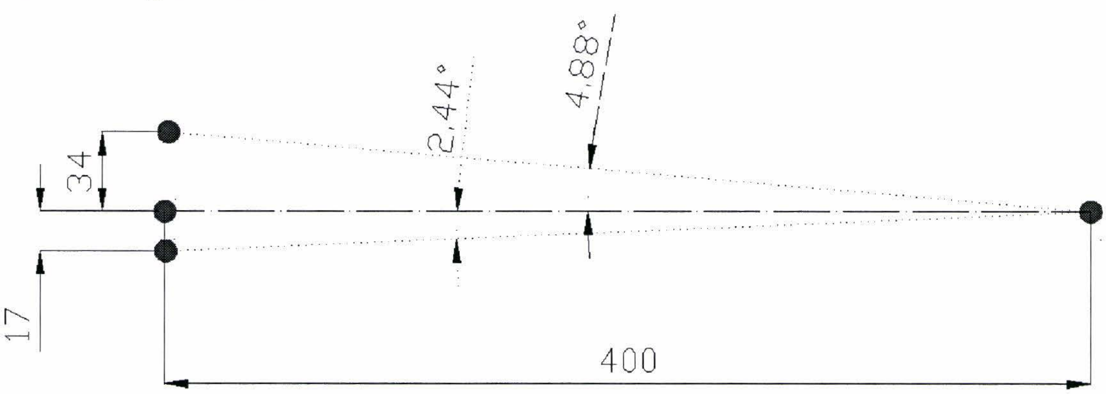


Figure 6.12: Flex plate angles

In figure 6.12 the centre line represents the rear left drive shaft. During bump the maximal vertical displacement of the chassis will be 34 mm, which causes an angle of 4,88 degrees. For droop the vertical displacement will be maximal 17 mm, which will result in an angle of

2,44 degrees. This means that the maximal angle for the flex plate will be approximately 5 degrees compared to the neutral situation.

The dimensions of the flex plates will be determined using the maximum forces on the flex plates. The two pieces of the drive shaft are connected using a thin membrane for the required angle flexibility. The drive shaft will be made of a thin wall aluminium tube with a large diameter. This will result in a low mass and a high stiffness, because the mass used is placed on the outer fibre. The mass inertia will be high using a large diameter, which results in a relative high stiffness for the thin-walled tube. These properties can be translated into two versions of a flex plate, a double and a triple version. The layout of both versions is shown in figure 6.13a en 6.13b.

a) b)

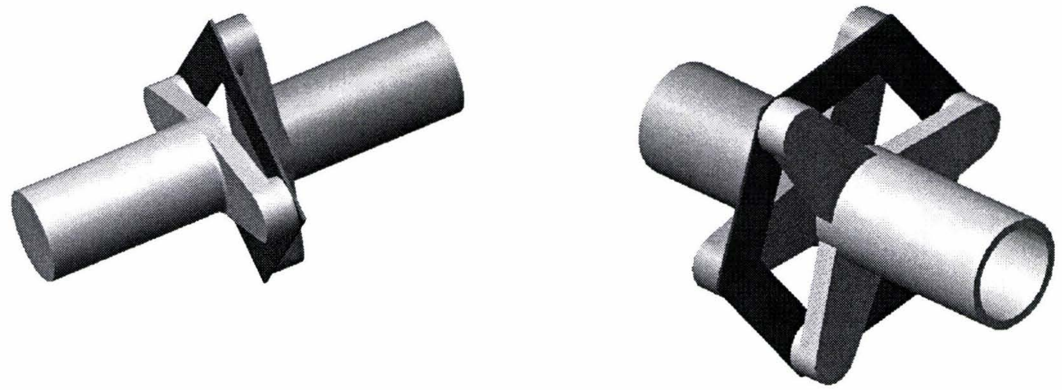


Figure 6.13: Flex plates a) double version b) triple version

Regarding the required functionality of the flex plate the choice has been made to use the double version. The triple version has the disadvantage that the angular motion will be less fluent, because the same angles have to be achievable. The bending lines of the triple version don't go through the centre point of the axle, in contrast to the double version, which is shown in figure 6.14a en 6.14b.

a) b)

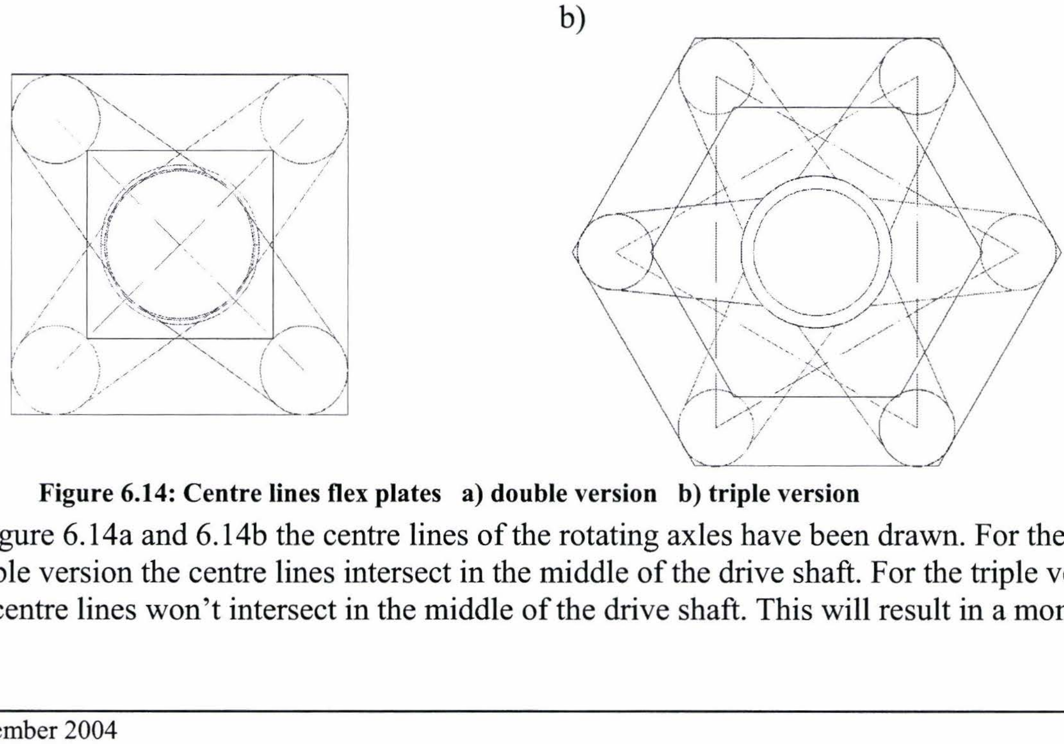


Figure 6.14: Centre lines flex plates a) double version b) triple version

In figure 6.14a and 6.14b the centre lines of the rotating axles have been drawn. For the double version the centre lines intersect in the middle of the drive shaft. For the triple version the centre lines won't intersect in the middle of the drive shaft. This will result in a moment around the centre line of the drive shaft. When using the double flex plate the forces will go right through the centre of the drive shaft and won't cause any internal moment. This will result in a lighter construction of the flex plate. The use of the doubled version flex plate will result in the design for the drive shaft shown in figure 6.15.

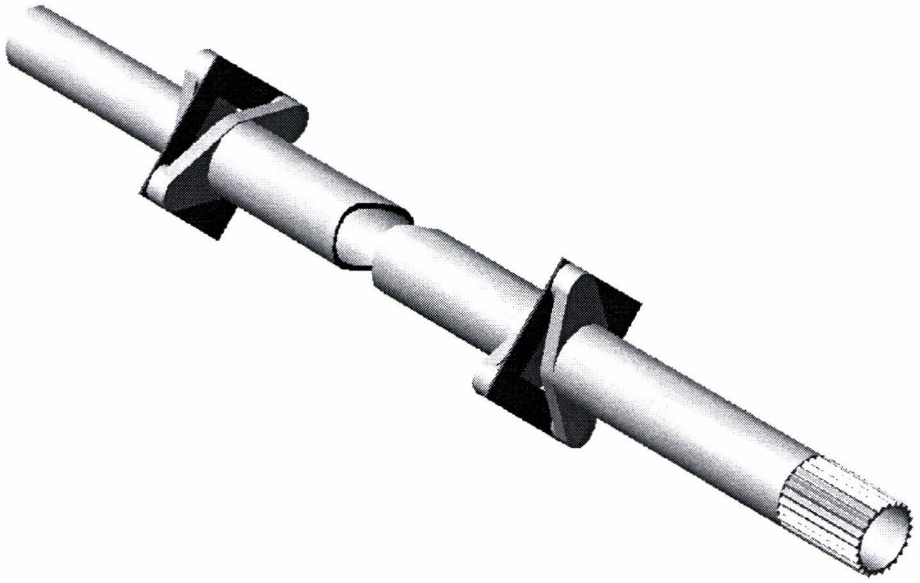


Figure 6.15: Left drive shaft

In figure 6.15 only the cut through left drive shaft with two flex plates has been shown. From the differential splines will be used and towards the wheels bar steel will be used for the bearing of the upright and the hub in the rear wheel. The relative positioning of the flex plates is very important to acquire the desired angular motion.
The connection from the drive shafts to the wheel will be explained in paragraph 6.3.3.

6.3.3: Rear axle

For the rear axle bar steel will be used with a diameter of 30 mm, which will be rolled to create a stepped form for the axle. The shape of the axle, which has to rotate in contrast to the upright, contains two raised edges. These edges will be used to place two angular contact ball bearings in the upright. Therefore two circular chambers will be milled in the upright at the front and the backside. The hub used to transmit the rotation of the axle to the rotation of the wheel will be connected onto the axle using splines. Therefore a key way has to be made in the axle. The section of the left rear axle is shown in figure 6.16.

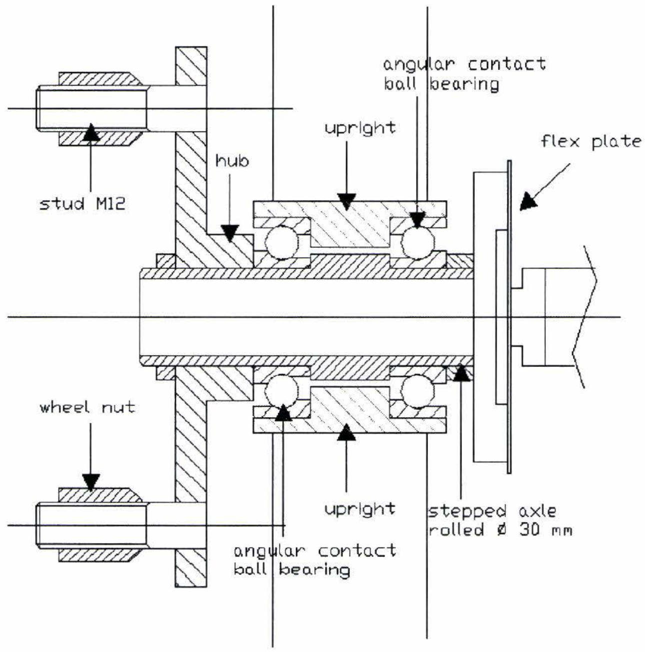


Figure 6.16: Section left rear axle

For the connection between the hub and the wheel four studs M12 and four wheel nuts will be used. These studs will be integrated in the hubs. A hexagonal nut will be used to keep all the parts fixated on to the axle. The 3D shape of the hub and the other parts for the rear left axle are shown in figure 6.17. In this figure the upright has been removed to keep the figure as clear as possible.

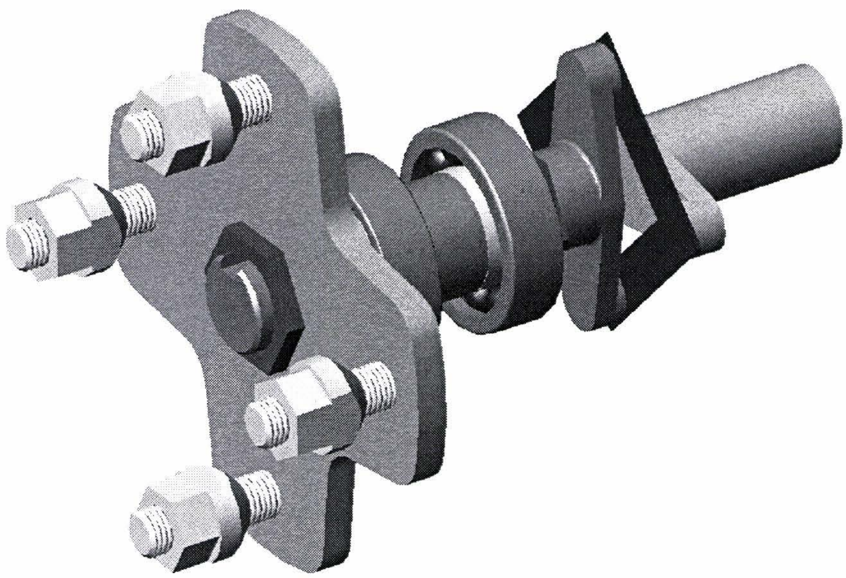


Figure 6.17: Layout rear axle

Now the design of the drive shaft and the rear axle is visible, the shape and the layout of the drive train has been determined. For the front wheels, the layout of the bearings and the axle has to be made too. The design for the front axle will be explained in paragraph 6.4.

6.4: Front axle

Because the car is rear wheel driven there will be no driving shafts needed for the front wheels. The necessary axle has to connect the wheel with the brake disc and the upright, because the brake system will be placed in the front wheels. The brake disc and wheel rotate with respect to the upright with the integrated brake calliper, bearings fitted on the axle will be needed to separate the parts that have to rotate and the non-rotating parts.

Angular contact ball bearings will be used. When using two bearings in an O-position they can resist both the axial forces in two directions. A double row angular contact ball bearing can be used instead of two single row bearings. This will decrease the build in length with 9,4 mm, which is desirable, in view of the small amount of axial space in the wheels. The difference between double row and two single row bearings is shown in figure 6.18a and 6.18b.

a) b)

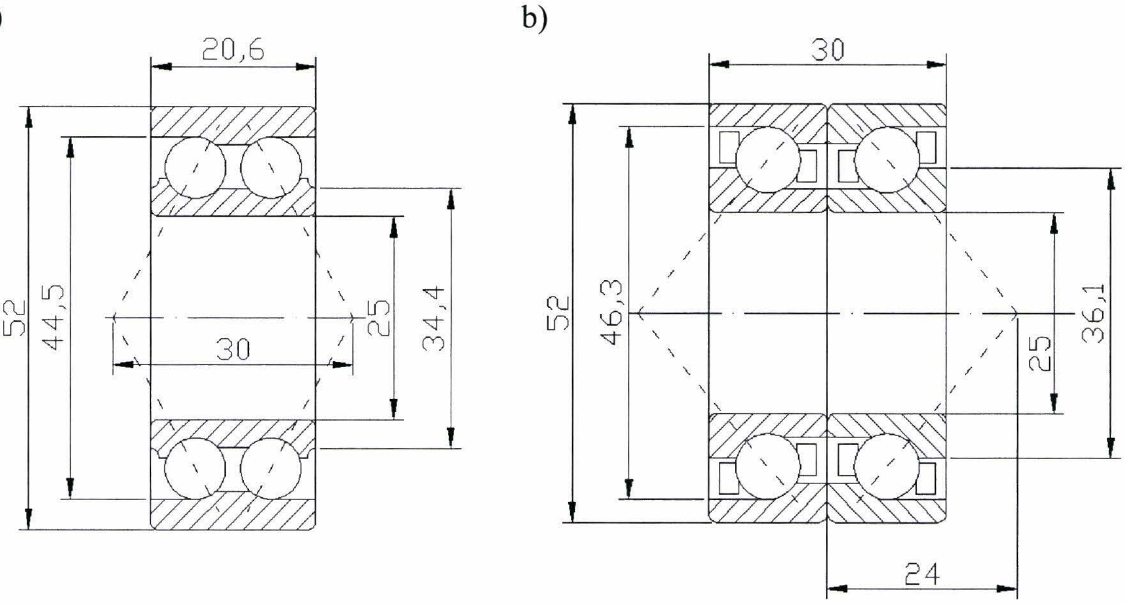


Figure 6.18: Bearing comparison a) double row bearing b) two single row bearings

The suspension geometry determines the placement of the upright, because the length of the A-arms and the required movement of the tires have been determined in chapter 4. The amount of offset of the rims will be decided by the space needed for the hub and the bearings. Using the double row bearing this space will be minimized and the offset of the rim won't have to be that much. The offset for the front and the rear wheel is shown in appendix V. The configuration of the front axle is shown in figure 6.19a where the section of the front left axle is shown. In figure 6.19b the 3D layout has been drawn.

a) b)

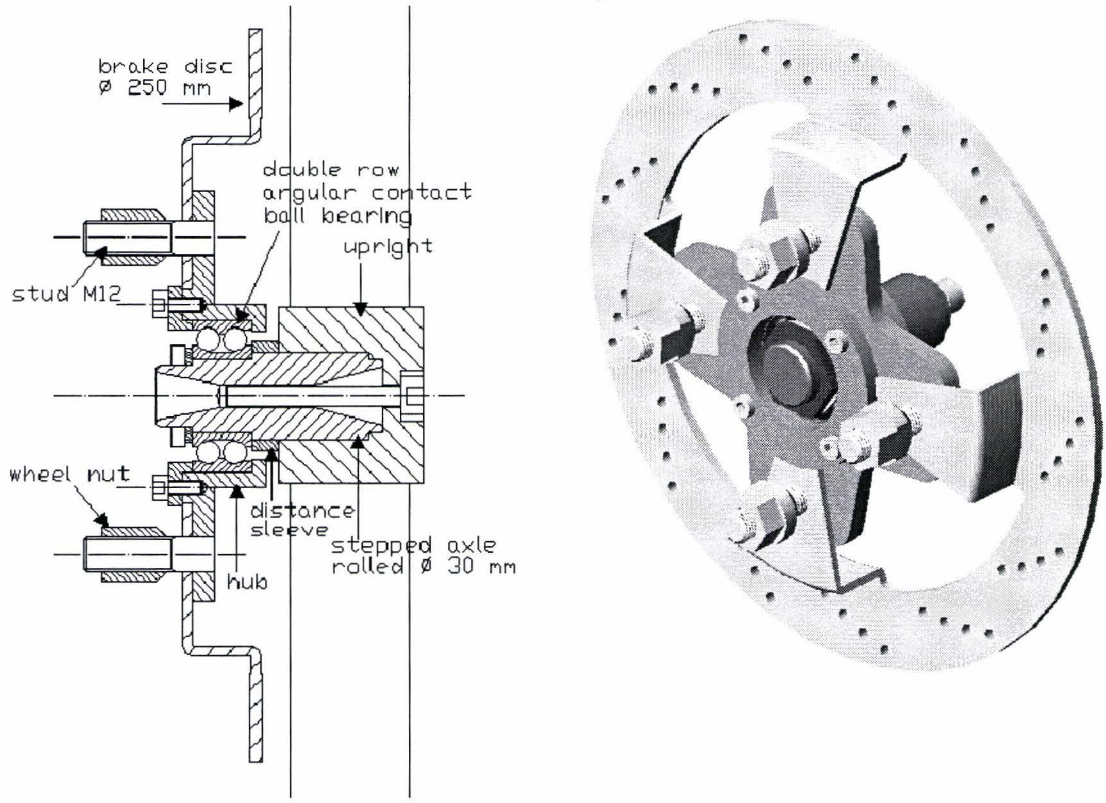


Figure 6.19: Left front axle a) section front axle b) layout front axle

The base material for the axle shown in figure 6.19 is a bar steel with a diameter of 30 mm. The advantage of a stepped axle is that the bearings won't have to be that large and in the upright extra millimetres (step from 25 to 30 mm) can be used for the stiffness of the axle, because the forces from the tires will be guided through the axle and the upright. The double row angular ball bearings will be fitted into the hub, which means that the outer ring of the bearing will be an interference fit. The inner ring of the bearing will be fitted with a high quality loose fit and locked axially. A negative offset of 45 mm for the front wheels is required to create the desired suspension layout and the necessary space for the front axle, the upright and the brake system.

The brake disc will be mounted between the rim and the hub and will have a flanged form. The holes in the brake disc have been made to give better performance in wet conditions. The dimensions and the material properties of the brake discs will be explained and calculated in chapter 7.

Chapter 7: Braking and steering

The overall layout for the Formula Student race car has been made. The chassis, the suspension and the drive train have already been described in the previous chapters, but the layout for the brake system and the steering mechanism have do be determined yet. Therefore in this chapter first of all the brake system and in paragraph 7.2 the steering and shifting mechanism will be described.

7.1: Brake system

For a race car the brakes are very important, because when a driver can brake some tenths of a second later than his opponent he can overtake him before a bend. Therefore the Formula Student car needs a good brake system. Furthermore good brakes are important, because of the drive ability of the car. When the brakes react very well on the input of the driver, the driver will be able to use the car optimally. Therefore the input from the driver on the brake pedal has to result in an immediate response at the brakes. This means that the volume of the oil, that will be used to guide the forces from the brake pedal towards the brake calliper, has to be minimized, because the stiffness of the oil is much lower compared to the stiffness of steel. The difference between these two materials is a factor 400.

The maximal braking force and hence the maximum negative acceleration that can be generated, depends on the tires. In chapter 2 the tire characteristics have been analyzed, which resulted in a 1,3g lateral acceleration in a bend. For the braking forces the assumption has been made that the maximum negative longitudinal acceleration will be the same as the maximal lateral acceleration of the front tires, which will be 1,3g.

During braking the disc brakes have to dissipate the kinetic energy of the car. At the front wheels a brake disc will be placed into the wheel, between the upright and the rim. At the rear side two brake discs will be placed on the two drive shafts. This position results in a mass reduction in the rear wheels to keep the unsprung mass as low as possible, because the rear rims and tires are wider and heavier. For the front brakes the wheels are the only place to mount the brake discs and callipers into. The calculation for the dimensions of the disc brakes will be done in paragraph 7.1.1.

7.1.1: Brake disc dimensions

The calculation for the thickness of the brake discs will be done using the brake balance. The brake balance for this kind of cars is usually set at 75/25, which means that the front brakes have to generate 75 percent of the brake force and the rear brakes only 25 percent. The necessary braking energy will be calculated from the mass and the maximal negative longitudinal acceleration of 1,3g. Using equation 7.1, when driving at a speed of 25 m/s and braking till a speed of 5 m/s using the required energy can be calculated.

$$\Delta E_{kin} = \frac{1}{2} \cdot m \cdot \left| (V_{end}^{\ 2} - V_{begin}^{\ 2}) \right| \qquad \text{(Equation 7.1)}$$

$$\Delta E_{kin} = \frac{1}{2} \cdot 300 \cdot \left| (5^2 - 25^2) \right| = 90 \cdot 10^3 [J]$$

For the calculation of the brake discs the amount of heat that has to be dissipated by the discs is important. When using a brake balance of 75/25 one front disc has to dissipate 37,5% of the total kinetic energy, which means 0,375 x $90 \cdot 10^3 = 33{,}75 \cdot 10^3$ J. Considering the fact that stainless steel will be used as disc material the heat dissipated by the disc can be calculated using equation 7.2. The assumption has been made that the brake disc will be heated up 200 degrees, during one brake action.

$$E_{heat} = \Delta E_{kin} \qquad \text{(Equation 7.2)}$$

$$E_{heat} = m_{disc} \cdot c_p \cdot \Delta T$$

$$m_{disc} = \frac{33{,}75 \cdot 10^3}{c_p \cdot (T_{end} - T_{begin})} = \frac{33{,}75 \cdot 10^3}{502 \cdot (493 - 293)} = 0{,}34 [kg]$$

Using the fact that a safety factor of 2 will be desired the mass of the disc has to be 0,7 kg. The space in the rim is limited, because the inner diameter of the rim is 305 mm. This will result in a maximum diameter of 250 mm for the brake disc. A space of 25 mm around the disc is necessary to place the brake calliper over the brake disc. Before designing the brake disc the last assumption that has been done is that the disc has an inner diameter of 190 mm, which means that the height of the disc is 30 mm (250 – 190 / 2). The thickness of the disc will be calculated using equation 7.3.

$$t_{disc} = \frac{m_{disc}}{A_{disc} \cdot \rho_{st,steel}} \qquad \text{(Equation 7.3)}$$

$$t_{disc} = \frac{m_{disc}}{\frac{\pi}{4} \cdot (D_{out}^{\ 2} - D_{in}^{\ 2}) \cdot \rho_{st,steel}} = \frac{0{,}7}{\frac{\pi}{4} \cdot (0{,}25^2 - 0{,}19^2)}$$

From equation 7.3 can be derived that a thickness of 4 mm will be used, when considering that a safety factor of 1,9 will be sufficient. This will result in a brake disc of 0,65 kg, which will be mounted as described in paragraph 6.4.

For the rear discs much lighter and smaller discs can be used, because the brake balance is 75/25. For one rear disc 12,5 percent of the total kinetic energy has to dissipated, which means 0,125 x $90 \cdot 10^3 = 11{,}25 \cdot 10^3$ J. This is one third of the amount of energy dissipated by one front disc, which means that the mass only has to be one third too. The disc has to have a mass of 0,22 kg. The rear disc will have an outer diameter of 175 mm and an inner diameter of 125 mm. This will result in a thickness of 2,5 mm, using equation 7.3. These dimensions will result in a mass for the rear discs of 0,24 kg each.

7.1.2: Brake calliper

For the layout of the calliper the choice has been made to use a floating calliper with only on one side two pistons. Callipers with pistons on both sides of the brake disc are also available, but because of the lack of axial space in the rim, small build in dimensions will be preferable.

Figure 7.1 shows the section of the calliper and its position in the wheel regarding the position of the brake disc. Figure 7.1b shows that the brake disc will only have a piston on one side, because between the brake disc and the wheel, only a small axial space is available.

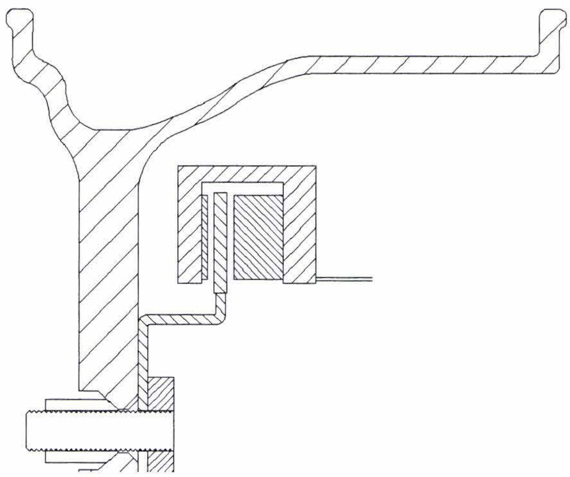


Figure 7.1: Floating brake calliper

In figure 7.1 the brake calliper has been drawn in the upper part of the wheel. This will not be the final position of the calliper. Figure 7.1 only determines the position of the calliper regarding the brake disc.
The position of the calliper within the rim is very important. The placement of the calliper can be done in two different ways: the calliper can be placed before or behind the front axle. Considering the mass of the calliper it is preferable to place it behind the front axle. This will ease the steering when driving through a bend. This effect will be explained in figure 7.2, which shows the front wheels in top view.

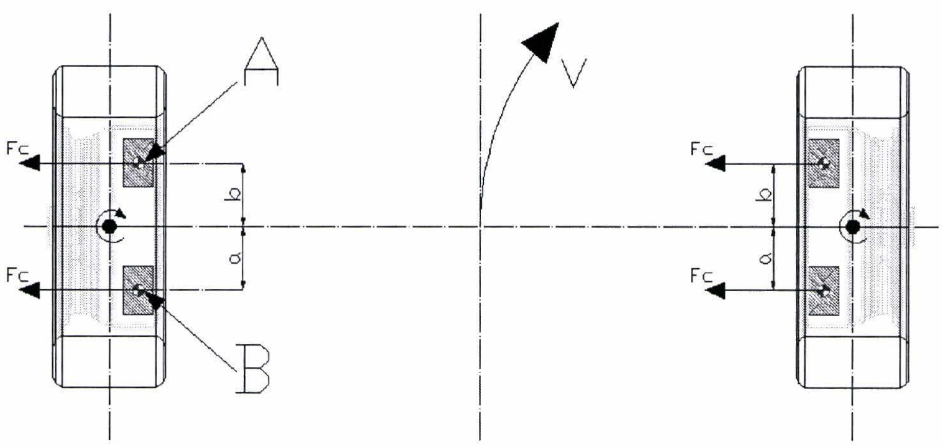


Figure 7.2: Brake calliper position

When the calliper will be positioned before the front axle (position A in figure 7.2), the lateral force (Fc) on the calliper in a bend, which will point towards the outside of the bend, will have a negative effect on the steering of the car. This moment on the steering input will be calculated by multiplying the lateral force (Fc) with the arm for that force (b). This moment will cause extra under steer. When the calliper is placed behind the front axle (position B in figure 7.2) it will generate a positive moment when multiplying the lateral force (Fc) with the arm (a). This moment can be added to the steering input, which will have lighter steering as result. The height of the position of the calliper in the rim depends on the availability of

space, because the upright has been designed as tall as possible to diminish the forces on the suspension. Therefore the available space in the rim is small. The final place of the calliper will be behind and below the front axle, because there is the availability of space and a low centre of gravity is desirable.
The position for the rear brake discs and callipers will be different. This will be described in paragraph 7.1.3.

7.1.3: Rear brake system

For the rear brake system an other configuration will be used. The unsprung mass has to be as low as possible for the handling of the car, which means that the weight of the four wheels has to be as low as possible. Therefore the rear brake disc will be placed onto the drive shafts next to the differential, so that the mass of the brake discs and the brake callipers won't be placed in the rear wheels. This will result in a lower unsprung mass for the rear wheels. The position of the rear brake discs is shown in figure 7.3.

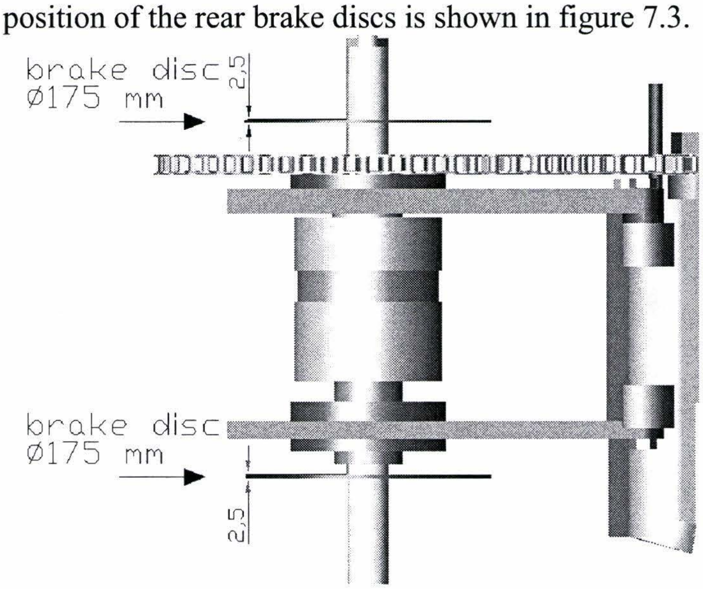


Figure 7.3: Rear brake discs position

The brake disc will be mounted on the drive shafts. The rear brake callipers will be mounted on the bottom sandwich panel and will be supported by the two aluminium plates that bear the differential. The availability of space is underneath the brake discs and that position is preferable regarding the height of the overall centre of gravity.

The friction coefficient of 1,3 between the tires and the road surface limits the maximum deceleration to 1,3g. This requires the front and rear brakes to provide a braking force of 489 Nm and 310 Nm respectively. These demands, next to low mass and costs, led to choosing the 2 piston Grimeca callipers. The calliper pistons are 28 mm in diameter. Ferodo brake pads and two individual Grimeca master cylinders (6/8 inch) are used for the front and rear brakes to increase safety. The master cylinders are connected by a balance bar to allow the brake bias to be adjusted.

Now the brake system has been designed the required steering mechanism will be determined.

7.2: Steering and shifting mechanism

For the handling of the Formula Student car it's important that the steering wheel is placed in such a way that it can rotate completely, between -180 and 180 degrees. This movement of the steering wheel should not be hampered when rotating the steering wheel with two hands. These hands have to act freely and should not make contact with the driver's legs or with the chassis during a steering input. Therefore the steering wheel will be placed as high as necessary so that it will be able to rotate freely.

For the position of the steering wheel some different choices have to be made. According the rules some aspects have to be taken into account for the positioning of the steering wheel. First of all the shape of the steering wheel has to have a nearly circular perimeter, which means that "H", "Figure-8" or cut-out wheels are not allowed. The second rule for the position of the steering wheel is that the front roll hoop, which is integrated in the main frame, is not allowed to be lower than the top of the steering wheel. This will mean that the hands of the driver will be protected during a rollover. The last rule for the steering wheel is that it has to be attached to the steering axle with a quick release. The driver has to be able to disconnect the steering wheel in driving position with his racing gloves on, using this quick release.

For the safety of the driver hands on wheel shifting will be done using two levers behind the steering wheel. For gearing up the right lever will be used and for gearing down the left lever will be used. When using this mechanism the driver won't have to remove his hands of the steering wheel means that he can shift faster and it will be safer, because he always can use both hands. Then the driver will be able to react faster on unpredictable situations. These shifters will be connected to the shifter shaft at the engine using steel cables. The choice has been made to shift mechanically so that the driver knows whether the correct gear has been reached. When shifting electronically, uncertainty will always exist whether the shifting action has been completed or not as there is no force feedback.

For the position of the steering wheel the side view of the car with the position of the driver has been used to determine the optimal point for the steering wheel. Figure 7.4 shows the front part of the chassis, where the position for the steering wheel has to be determined.

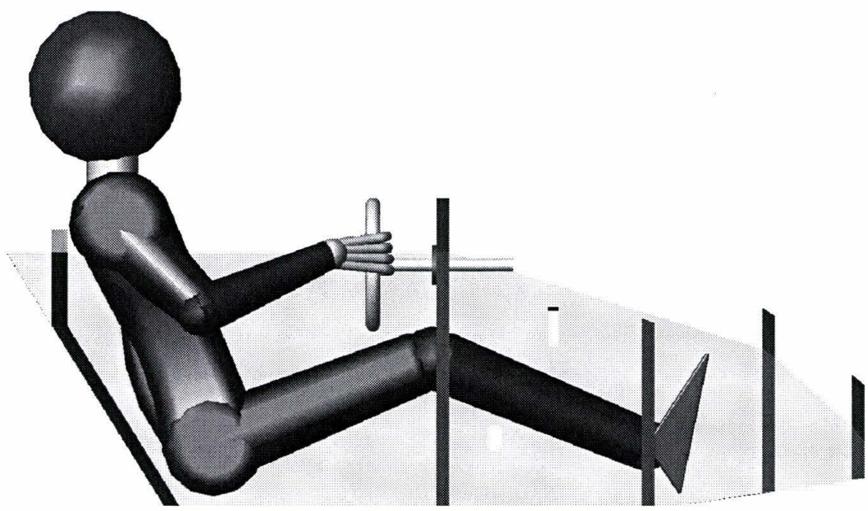


Figure 7.4: Steering wheel position

In figure 7.4 the driver has both hands on the steering wheel and the steering axle has been determined to be horizontal. When trying to steer with a steering wheel under an angle, which means that the steering axle will be sloped downwards like the angle of the legs, the two extreme positions, -180 and 180 degrees, cannot be reached without touching the legs or the chassis. This was not desired. Therefore the steering axle has been placed horizontally. In figure 7.4 the point where the tie rod intersects with the chassis is shown: the lowest slot in the side panel of the chassis. This will result in a vertical steering mechanism, because the steering axle is right above that slot.

To join the tow tie rods of the front suspension and the steering axle a mechanism has to be designed to translate the input on the steering wheel into the tie rods. The extreme positions of the steering wheel have to result in the largest steering angles of the wheels with neutral Ackermann, which means that the inner wheel will make an angle of 29 degrees and the outer wheel an angle of 21 degrees. For the tie rods these angles mean that the horizontal translation of the connection points with the upright has to be 21 mm for the largest angle and 19 mm for the angle of 21 degrees.

The ratio between the input on the steering wheel and the output on the tie rod results in the layout of the steering mechanism. This mechanism contains a rack-and-pinion drive, which means that onto the steering axle a pinion will be mounted. This will rotate over a gear rack. The gear rack is mounted on a part of a circle (22,5 degrees), which has been fixed on a box profile. This box profile will rotate around a needle bearing, which will be mounted in the bottom panel. This needle bearing is the centre for the circle part of 22,5 degrees. The steering axle has a diameter of 25 mm, but will be stepped at the end where the pinion will be mounted. The reduced diameter will be 20 mm, so that the size of the pinion will be chosen smaller. This will result in a shorter gear rack, because the length of the gear rack will be determined by the circumference of the pinion. A shorter gear rack will result in a smaller rotation angle for the box profile, so that the driver won't be hampered by the steering mechanism between the legs. The steering mechanism is shown in figure 7.5.

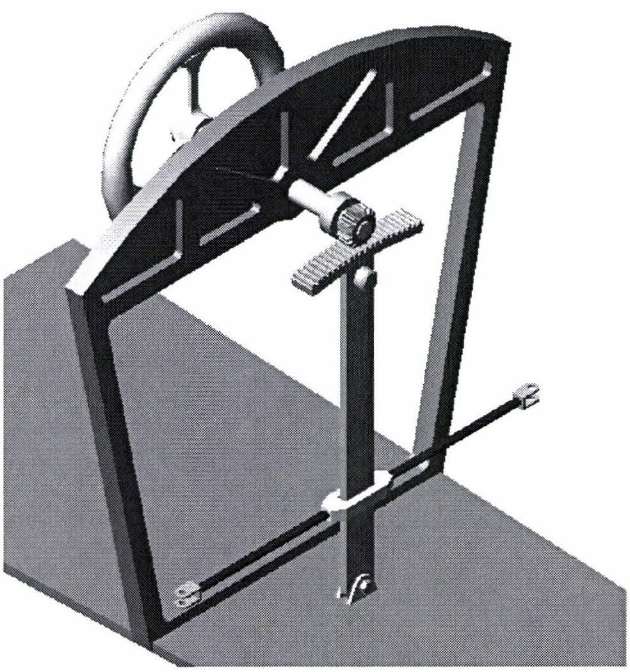


Figure 7.5: Steering mechanism

Using a box profile 40x10x1 mm the input of the steering wheel will be translated to the tie rods, which will be mounted into the little aluminium blocks at the end of the tubes. The height of the tubes in figure 7.5 has been determined by the ratio between the total length of the box and the distance between the tubes and the pivot point in the bottom panel. The tubes will be connected rigidly with each other using an aluminium block that will be mounted around the box profile.
The steering axle will be supported with two bearings, one in the dashboard of the main frame and the other just behind the pinion (see figure 7.5). Before the pinion isn't an option, because then the steering axle will be too long and will interfere with the anti-roll mechanism. Over the tie rods onto the bottom panel an aluminium plate will be made in the chassis to protect the steering mechanism. When a driver is getting out of the car the legs and the feet won't damage the mechanism when a cover plate will be used.

Chapter 8: Conclusions and recommendations

In this chapter first of all the results obtained will be described in the conclusions, after which some recommendations will be made to improve the car and its performance. The conclusions have been arranged according the sequence of the chapters.

8.1: Conclusions

Analysis car properties (chapter 2)
The most important dimensions for the Formula Student car will be the wheelbase (1600 mm), the front (1250 mm) and the rear track (1200 mm) and the centre of gravity height (330 mm). The mass of the car will be 300 kg using a weight distribution of 45/55. Aluminium wheels will be used for the front tires (6 inch) and the rear tires (8 inch). The forces during driving will be calculated with the friction coefficients for the front (1,3) and the rear tires (1,5). Cornering and braking can be done with 1,3g and accelerating with 1,5g. Longitudinal and lateral load transfer will result in a difference between the static and the dynamic forces. The yaw frequency of the car will be 8,8 Hz using the estimated moment of inertia (117,8 kgm^2) and the stiffness ($3{,}62 \cdot 10^5$ N/m).

Chassis (chapter 3)
For the chassis laminated aluminium sandwich panels will be used, which will be connected using aluminium frames (25 mm high). The sandwich panels will be made of two aluminium skins (1 mm) and a polyurethane core (10 mm). Reinforcements will raise the stiffness of the chassis and they will be used as protection for the driver. For a roll over (head clearance 51 mm) a steel roll hoop (Ø25,4 mm, 2,1 mm wall thickness) has been integrated in the chassis. This hoop is supported with two braces towards the rear frame and removable using a bolted connection. Inserts in the hoop and the sandwich panels provide extra stiffness for the connection from roll hoop to sandwich panel. For the engine suspension aluminium parts will be integrated in the chassis. The ground clearance for the chassis is 50 mm.

Suspension (chapter 4 and 5)
The overall suspension geometry is a double wishbone configuration with unequal length A-arms, which will provide 51 mm wheel travel. The centre lines for the wishbones have been placed to guide the forces nicely into the chassis and to prevent bump steer. Rod ends will be used to provide the mutual rotation between the chassis and the suspension. The front and rear roll moment is equal, which results in a higher rear roll centre (137,4 mm above the ground) than the front roll centre (94,6 mm). The roll centres and the dive centre, below and in front of the centre of gravity, determine the behaviour of the chassis during braking, cornering and accelerating. The static alignments for the wheels will be 2 mm toe in for the front wheels and 2 mm toe out for the rear wheels. Driving will be done with neutral Ackermann, using the KPI (6,3°) and the caster angle (7,6°), with a static negative camber (0,5°). This will cause an inner wheel angle of 29° and an outer wheel angle of 21°. A monoshock is used to provide the wheel travel for the front (44 N/mm) and the rear suspension (53 N/mm). A rotation/ translation mechanism will provide the anti-roll stiffness. The bump and the roll stiffness can be adjusted separately.

Drive train (chapter 6)
A Suzuki GSX-R600 engine will be used to power the car. The air intake system has been adjusted to the engine to provide the necessary air restriction (Ø20 mm). A standard exhaust will be used, which will be adjusted to the shape of the car, regarding the available space. A Torsen differential will be used to match the available engine power with the changing traction conditions of the tires. On the differential a chain wheel (57 teeth) will be mounted, which will be connected to the sprocket at the engine (14 teeth) using a chain (5/8"). This ratio will provide a top speed of 120 km/h at 9000 rpm. The differential will be connected with the rear wheels using two drive shafts. A spline connection will be used in the differential and two flex plates for each drive shaft will provide the required flexibility. One flex plate will connect the drive shaft with the rotating rear axle to create the rotation of the wheels. For the front axle a fixed axle will be used, because these wheels won't be driven.

Braking and steering (chapter 7)
For the braking system 4 brake discs and 4 brake callipers will be used. The front discs (Ø250 mm, 4 mm thick) will be placed in the front wheels with the brake callipers integrated in the front upright. The rear discs (Ø175 mm, 2,5 mm thick) will be rigidly connected onto the drive shafts. The front discs will be bigger, because the brake balance between front and rear is 75/25. A rack-and-pinion mechanism will be used for steering. This mechanism will be placed between the legs of the driver and will be connected to the tie rods. Behind the steering wheel (Ø260 mm), which will contain a quick release, two levers will be placed for shifting up and down. The clutch will be integrated in the shifting mechanism.

8.2: Recommendations

An enumeration will be made for the recommendations.

- The oil sump for the engine should be redesigned and removed from the underside of the engine to create a flat underside of the engine. This will result in a possible lowering of the engine of 60 mm. This adjustment however has to result in a complete different exhaust system, because the runners won't be able to be placed underneath the engine any more. Also the air intake system has to be reshaped, regarding the position of the driver.
- The position of the pivot points in the upright has been fixated, only the length of the wishbones will be adjustable. For the tuning of the suspension it will desirable to create a couple of possibilities for the position of those pivot points, using multiple holes or a slot in stead of one hole. Shims could be used too between the upright and the aluminium block for the upper A-arm, to keep the position for the aluminium block variable. Then the amount of camber will be adjustable.
- For the engine cooling it is desirable to remove the rear side panels, but the torsion stiffness of the chassis will than be reduced. Holes in those side panels, between the seat frame and the front wishbones for the rear suspension, will improve the air stream around the engine and the exhaust for extra cooling. In these holes a lamination should be made to guide the air stream.
- The reduction of the total mass will improve the driving conditions, considering a smaller load transfer and a lower moment of inertia for the car. A light carbon monocoque will have this mass reduction as result. However this will be more difficult and more expensive than the sandwich construction with the frames.

- The unsprung mass, the mass of the wheels has to be as low as possible. The use of magnesium wheels in stead of aluminium wheels will lower the mass and a reduction of wheel size will be useful too. The reduction of the wheel size will result in a redesign for the front and rear suspension and the reduction of the brake disc size in the front wheels.
- When the comfort for the driver will be less important, the chassis can be made tighter to crate a lighter and stiffer chassis compared with the current chassis.
- Using more detailed finite element analysis for the chassis the mass distribution in the chassis can be optimized. The thickness of the sandwich panels in some parts of the chassis can be reduced, regarding the required stiffness. The use of different core thicknesses will be useful too for an optimal mass distribution.
- In appendix W is shown that some parts have been analyzed for the internal stresses. The optimization for the mass and the mass distribution in all the car parts will lower the total weight and will result in parts with a higher stiffness/mass ratio. This has to be done using finite element analysis.
- For the exact forces on the car not only the steady state forces has to be calculated, but also the forces in the transitional phases: from braking into cornering and from cornering into accelerating.
- The rod ends used in the upright for the upper A-arm horizontally have to be placed vertically to use the rod end in plane. This will mean that the upright has to be adjusted.
- When the engine will be equipped with the necessary reservoirs and its wiring and piping, the position for the rear monoshock and the anti-roll mechanism will be questionable. Therefore the use of two shock absorbers for the rear suspension has to be considered, but the extra weight will be a disadvantage.
- For extra safety aspects, regarding the position of the chain, a chain guard above the sprocket and the chain wheel has to be used to screen the chain to prevent injuries.
- To optimize the engine performance the length of the runners for the air intake system and the exhaust has to be determined. This has to be done using a finite volume method.
- The dimensions for the drive shafts have to be determined regarding the forces from the engine onto the differential and the transmission from those forces from the differential to the rear wheels.
- The aluminium blocks that connect the suspension with the chassis have been designed, considering the angles in the suspension layout. For the production costs this won't be desirable. Therefore a redesign of those connection points is recommended here.
- The Formula Student car has to be suitable to be made in a series of 1000 pieces. The shape of some parts has to be reconsidered, regarding the fabrication and the production costs. Some milled parts will be too complex, when 1000 cars have to be made.
- When all the parts have been completely designed their mass and position has to be determined to calculate the exact position of the overall centre of gravity. For example the yaw frequency has now been calculated using a simplified model, but will be different when all the properties of the separate parts will be determined.
- The building and testing of the car will clarify whether the design for the suspension results in the desired movement of the wheels. The suspension stiffness can be tested and adjusted for the required movement of the car.
- For the brake calliper in the front wheels an interface has still to be designed from the upright to the calliper
- Construction drawings, which haven't been made yet, will be necessary to build the car.

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Appendices

Appendix A: Track and velocity curve

The track for the endurance contains a lot of bends. Figure A.1 represents the track that has to be driven during the endurance race and figure A.2 shows the velocity curve of the car during one round over the endurance track.

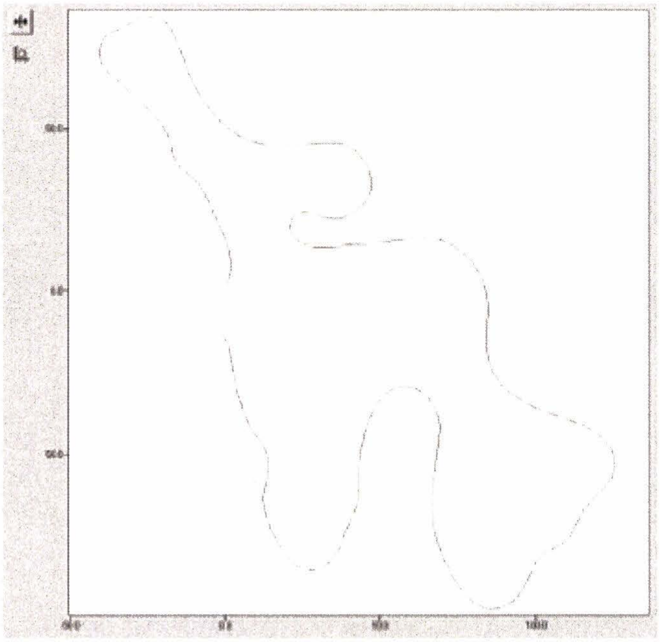


Figure A.1: Endurance track

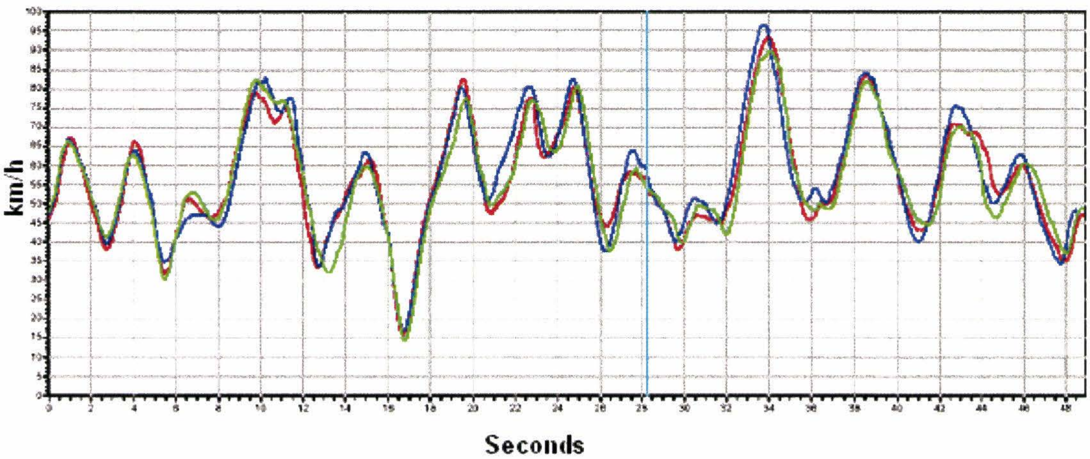


Figure A.2: Velocity on endurance track

Appendix B: Rear tire characteristics

To determine the friction coefficient for the rear tires the tire characteristics will be used. For the rear tires 8.2/22.0-13 3 ply Pro Series tires will be used. The tire characteristics for these tires are shown in figure B.1.

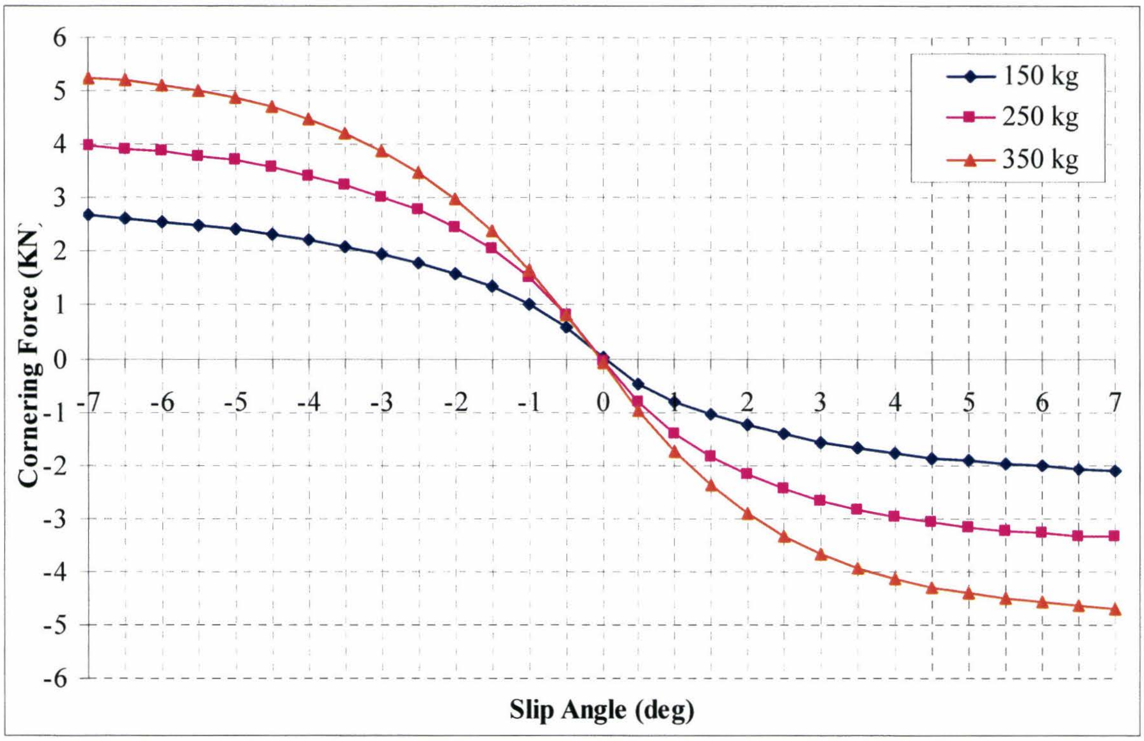


Figure B.1: Rear tire characteristics

Appendix C: Load distribution

Longitudinal and lateral load transfer cause a difference between the static and the dynamic normal forces on the four wheels. The following sequences of handlings will be analyzed:

1. Driving with constant velocity
2. Braking with 1,3g
3. Driving with constant velocity
4. Cornering with 1,3 g
5. Accelerating with 1,5g
6. Driving with constant velocity

The left tires will be used as the outer tires during cornering. The loads on each wheel have been shown in figure C.1 till C.4.

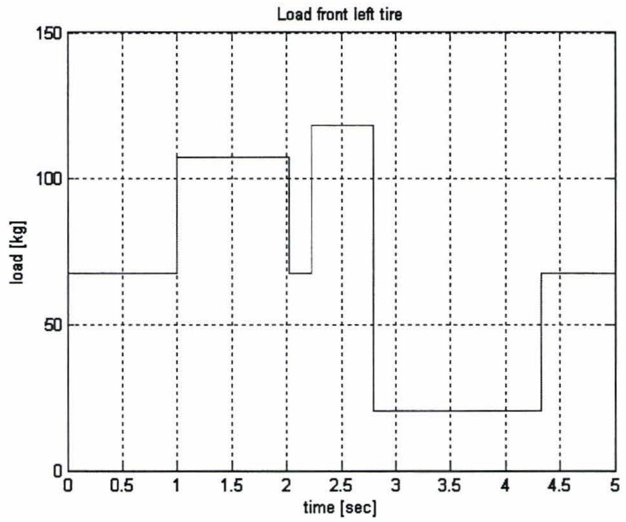


Figure C.1: Load on front left tire

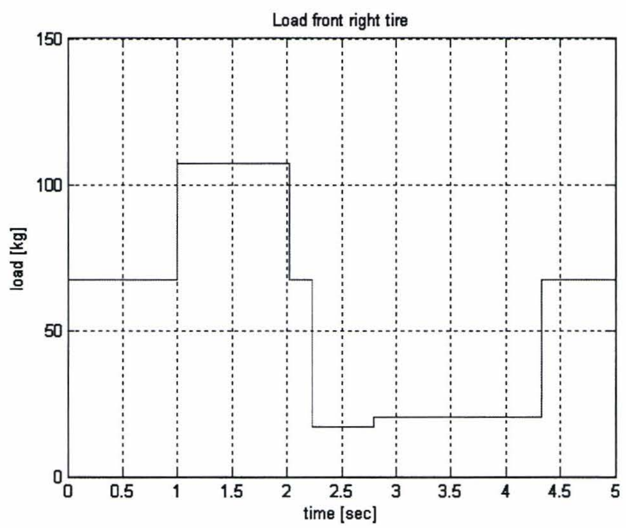


Figure C.2: Load on front right tire

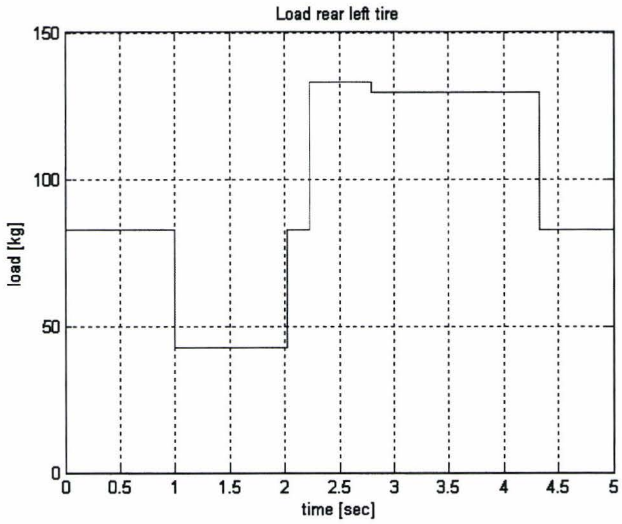


Figure C.3: Load on rear left tire

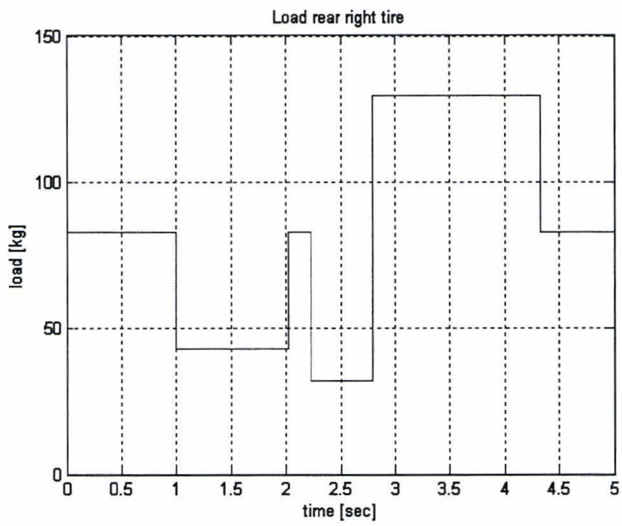


Figure C.4: Load on rear right tire

Appendix D: Other than normal forces on wheels

The forces that work on the tires during driving can be analyzed, using the following sequence:

1. Driving with constant velocity
2. Braking with 1,3g
3. Driving with constant velocity
4. Cornering with 1,3 g
5. Accelerating with 1,5g
6. Driving with constant velocity

The left tires will be used as the outer tires during cornering. The forces on each wheel have been shown in figure D.1 till D.4.

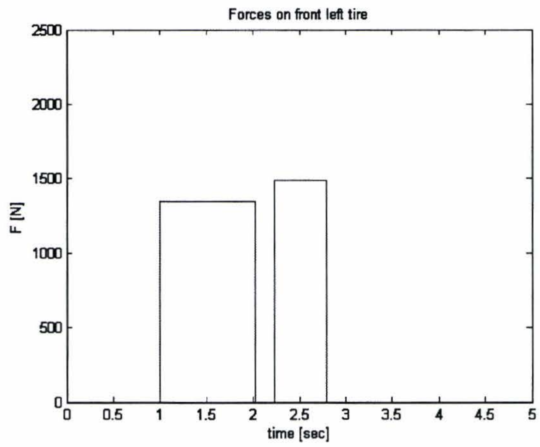


Figure D.1: Forces front left tire

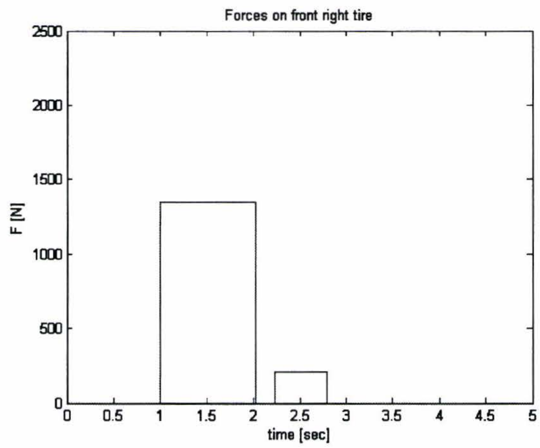


Figure D.2: Forces front right tire

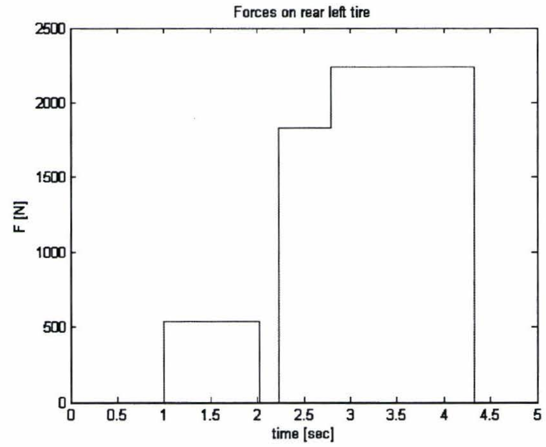


Figure D.3: Forces rear left tire

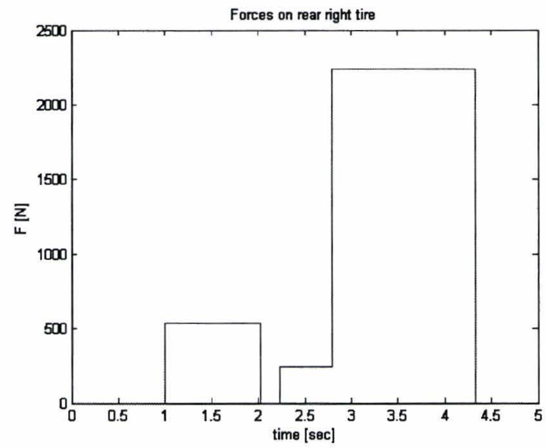


Figure D.4: Forces rear right tire

Appendix E: Forces on suspension

All the forces on the different parts of the suspension have been calculated considering the longitudinal and lateral load transfer during driving and shown in table E.1. These forces depend on the forces at the tires and the dimensions and position of the upright in the wheels. Braking and cornering with 1,3g and accelerating with 1,5g.

Corner force front wheel (1,3g)	1720	N
Corner force front wheel outside	1505	N
Corner force front wheel inside	215	N
Corner force rear wheel (1,3g)	2105	N
Corner force rear wheel outside	1860	N
Corner force rear wheel inside	235	N
Dimensions upright		
distance ground-lower A-arm	152,5	mm
distance ground-upper A-arm	377,5	mm
height upright	225,0	mm
Forces on suspension for accelerating		
lower A-arm rear wheel	3710	N
upper A-arm rear wheel	1500	N
Forces on suspension for braking		
lower A-arm front wheel	2290	N
upper A-arm front wheel	925	N
lower A-arm rear wheel	915	N
upper A-arm rear wheel	370	N
Forces on suspension for cornering		
lower A-arm front wheel outside	2525	N
upper A-arm front wheel outside	1020	N
lower A-arm front wheel inside	360	N
upper A-arm front wheel inside	145	N
lower A-arm rear wheel outside	3120	N
upper A-arm rear wheel outside	1260	N
lower A-arm rear wheel inside	395	N
upper A-arm rear wheel inside	160	N

Table E.1: Forces on A-arms

Appendix F: FEA frames

The frames used for the connection with the suspension are the main frame, the front suspension frame and the rear frame. The forces from the suspension have been applied using point forces. The results are shown in figure F.1, F.2 and F.3.

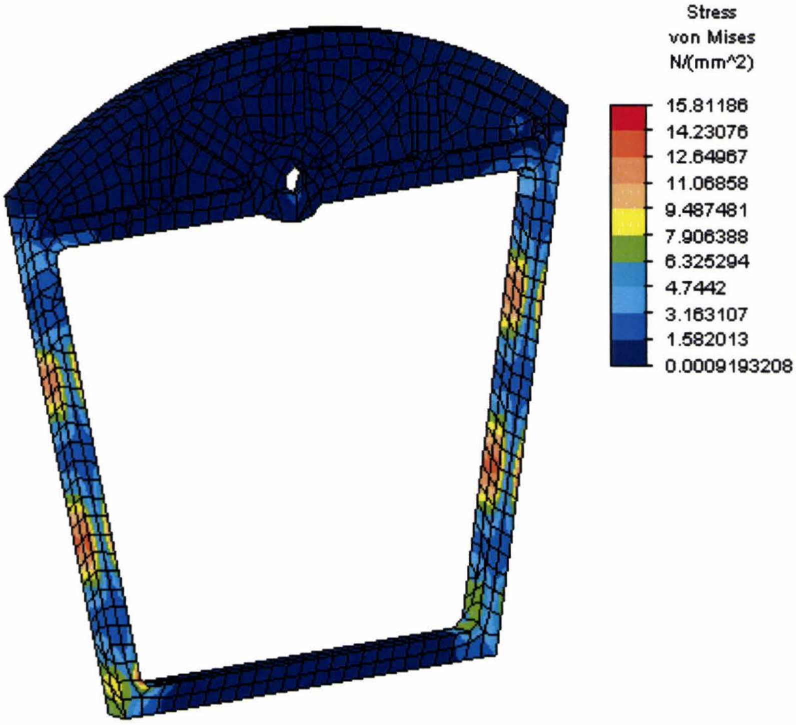


Figure F.1: Stress distribution in main frame

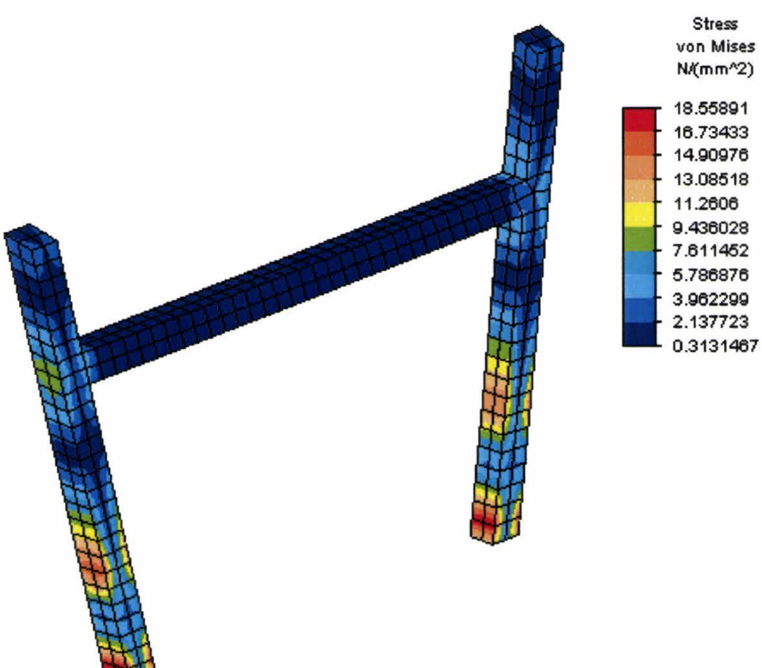


Figure F.2: Stress distribution in front suspension frame

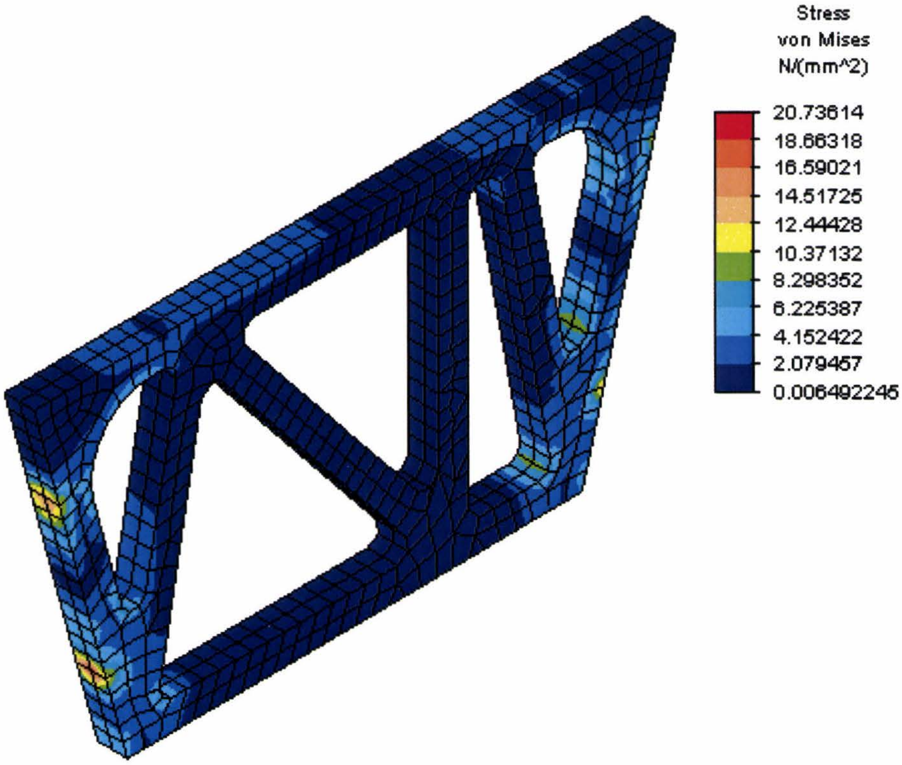


Figure F.3: Stress distribution in rear frame

Appendix G: Side impact analysis

For the analysis of the side impact six point forces have been applied to the side panel of the chassis. The results are shown in figure G.1 and G.2.

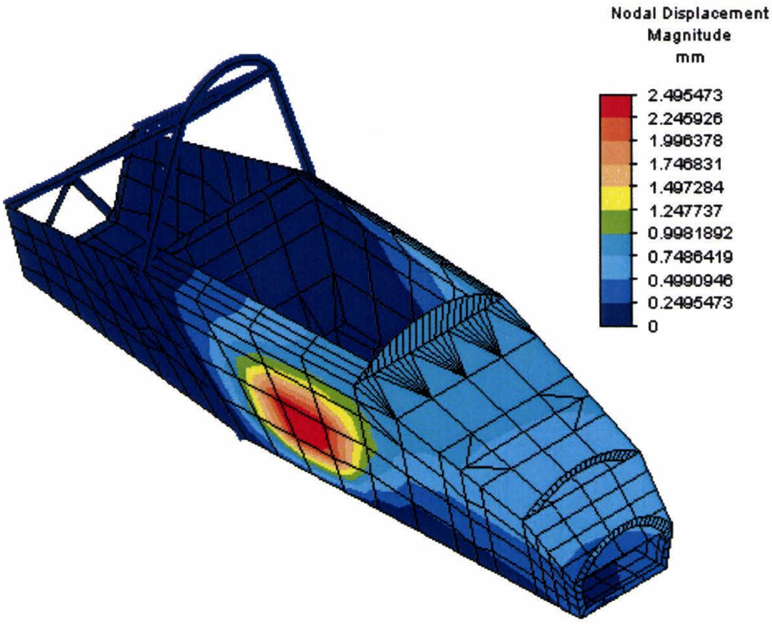


Figure G.1: Displacement side panel for side impact

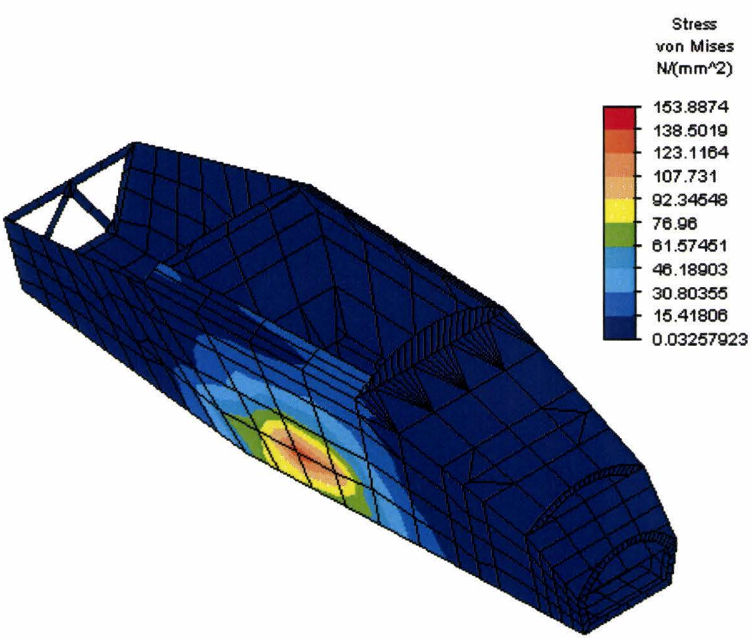


Figure G.2: Stress in side panel for side impact

Appendix H: Stress in chassis

To determine the maximum stresses in the chassis a finite element analysis has been made. For accelerating, braking and cornering the forces have been applied to the suspension, which results in stresses in the chassis. The highest stresses have been measured to be in the rear side panels. The stresses during driving are shown in figure H.1, H.2 and H.3.

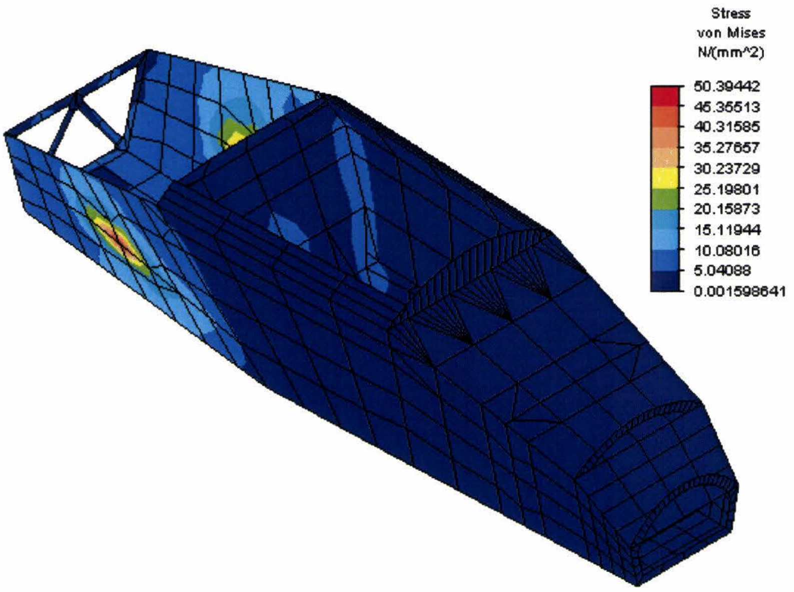


Figure H.1: Stress distribution in chassis during acceleration

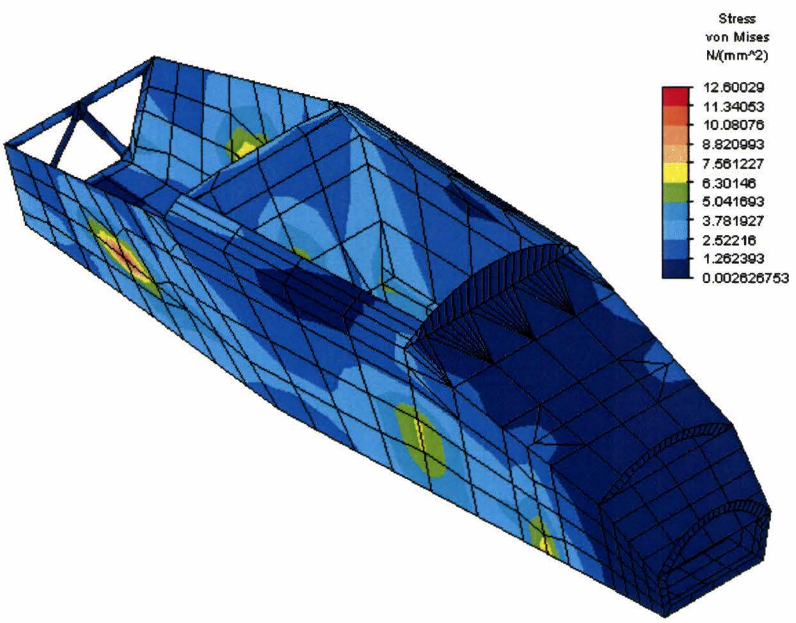


Figure H.2: Stress distribution in chassis during braking

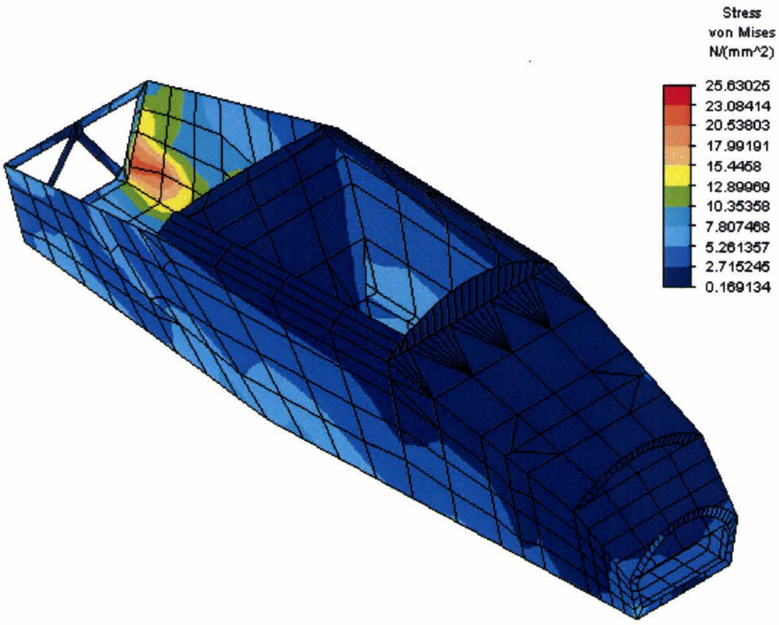


Figure H.3: Stress distribution in chassis during cornering

Appendix J: Head clearance

To determine the head clearance for the driver a line has been drawn between the highest point of the main hoop and the main frame. The head clearance of 51 mm has been set using an offset of 51 mm for the line between the main hoop and the main frame. This is shown in figure J.1.

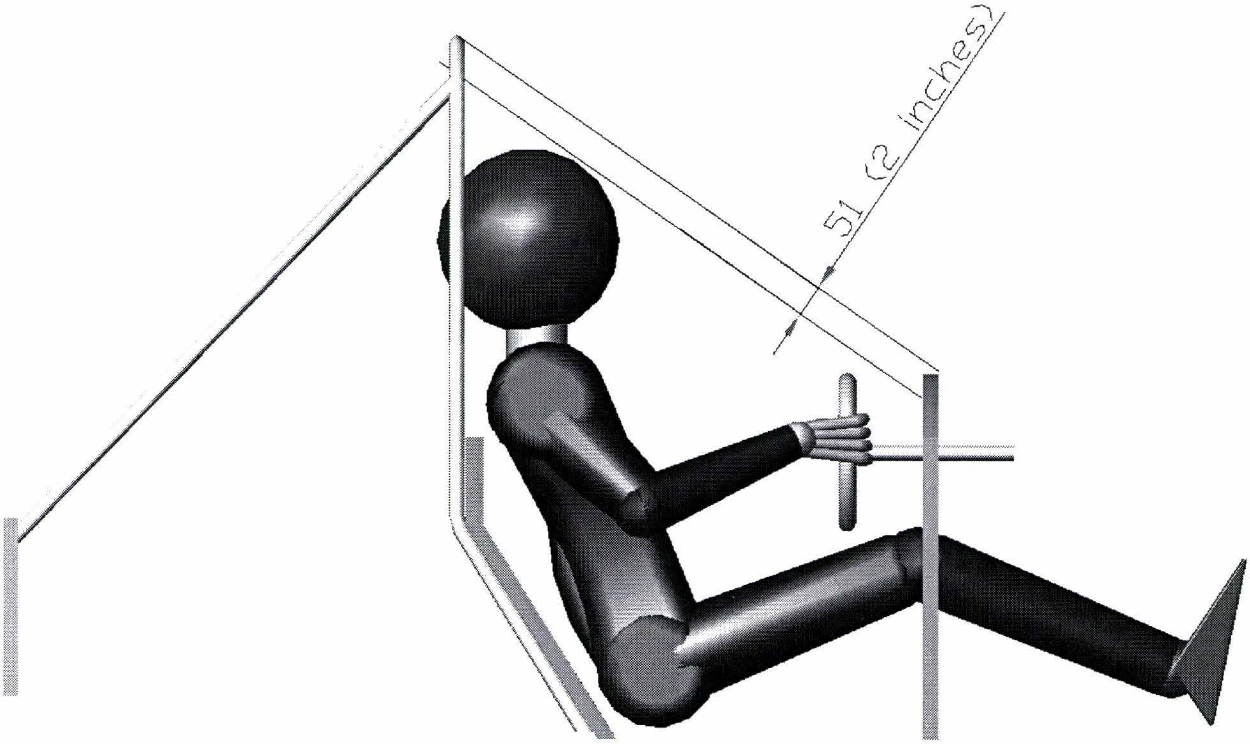


Figure J.1: Head clearance construction

Appendix K: Ackermann angles

The angles that have to be made for riding with neutral Ackermann have been determined in figure K.1. These angles have been determined in table K.1 using the dimensions of the car.

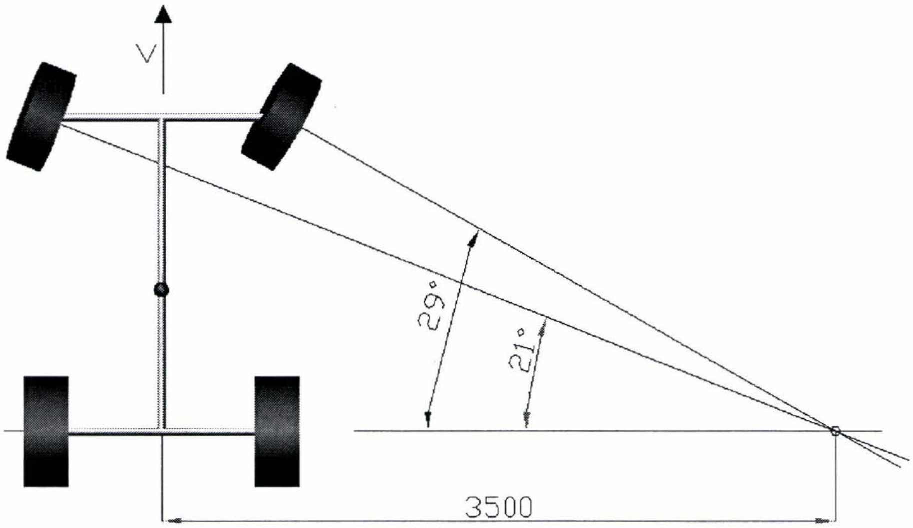


Figure K.1: Steering angles

Ackermann angles		
Frontwheel diameter	530	mm
Rearwheel diameter	567	mm
Wheelbase	1600	mm
Front track	1250	mm
Rear track	1200	mm
Turning radius	3500	mm
Distances to turning point		
Frontwheel right	3290	mm
Frontwheel left	4425	mm
Rearwheel right	2900	mm
Rearwheel left	4100	mm
Angles in top view		
Frontwheel right	29,1	°
Frontwheel left	21,2	°
Angles in front view		
Frontwheel right	4,6	°
Frontwheel left	3,4	°
Rearwheel right	5,6	°
Rearwheel left	4,0	°

Table K.1: Calculation Ackermann angles

Appendix L: Camber change in corners

Using the angles in the suspension layout steering will cause a change of camber for the front wheels. The amount of camber depends on the steering input of the driver. This is shown in figure L.1.

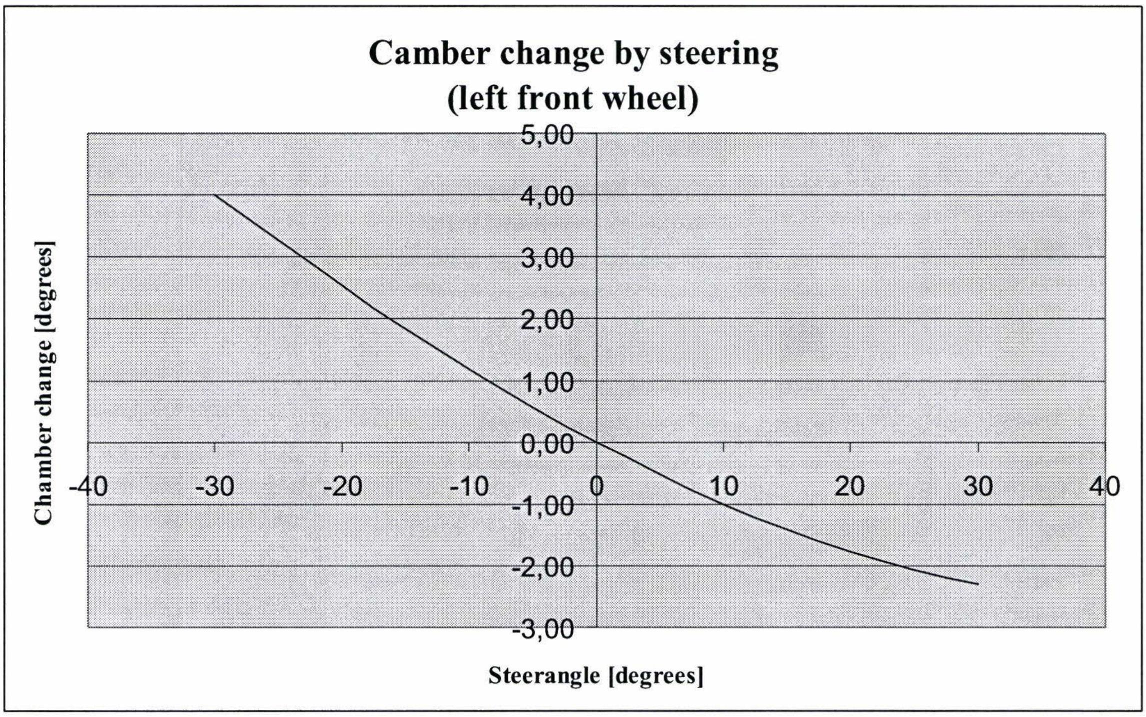


Figure L.1: Camber change by steering

Appendix M: Stress in suspension

To determine the maximum stresses in the suspension a finite element analysis has been made. For accelerating, braking and cornering the forces have been applied to the upright. The red wishbones represent a tube that contains tension and a blue tube contains pressure. The stresses during driving are shown in figure M.1, M.2 and M.3.

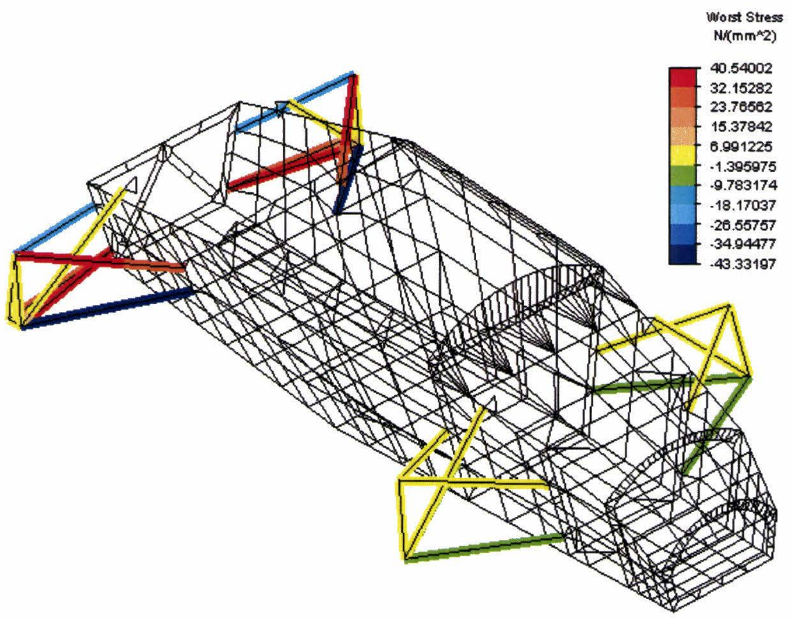


Figure M.1: Stress distribution in suspension during acceleration

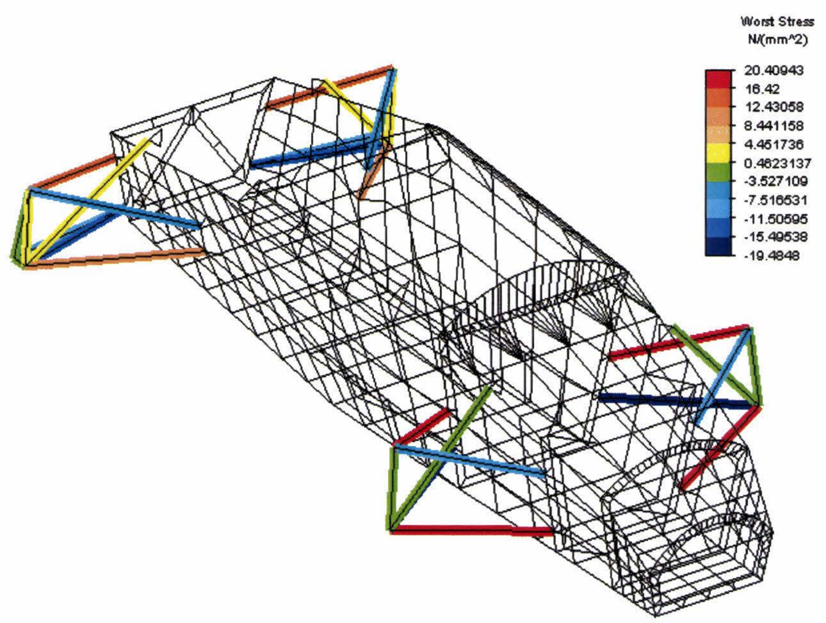


Figure M.2: Stress distribution in suspension during braking

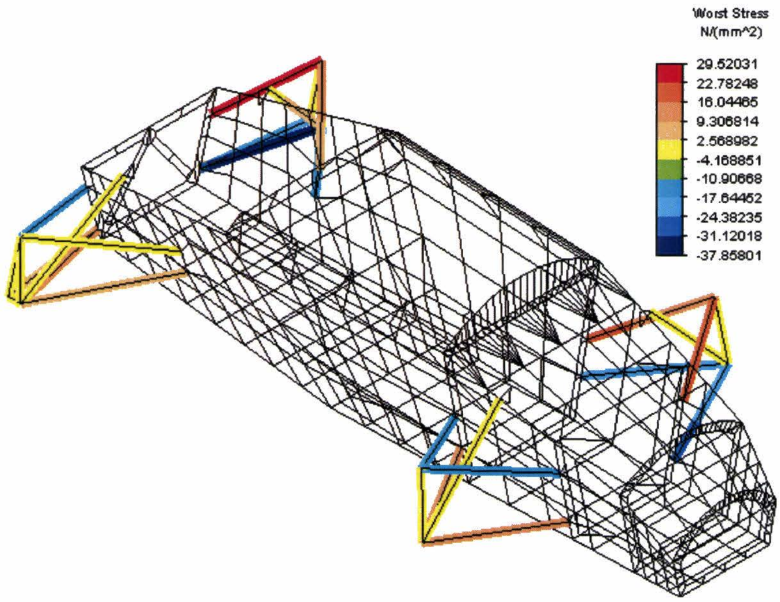


Figure M.3: Stress distribution in suspension during cornering

Appendix N: Bump steer rear suspension

The lines that have to be drawn for the determination of the tie rod position for the rear suspension will be constructed in the same sequence as the bump steer prevention for the front suspension. The position of the tie rod is different, but the construction is equal to the front suspension. This is shown in figure N.1.

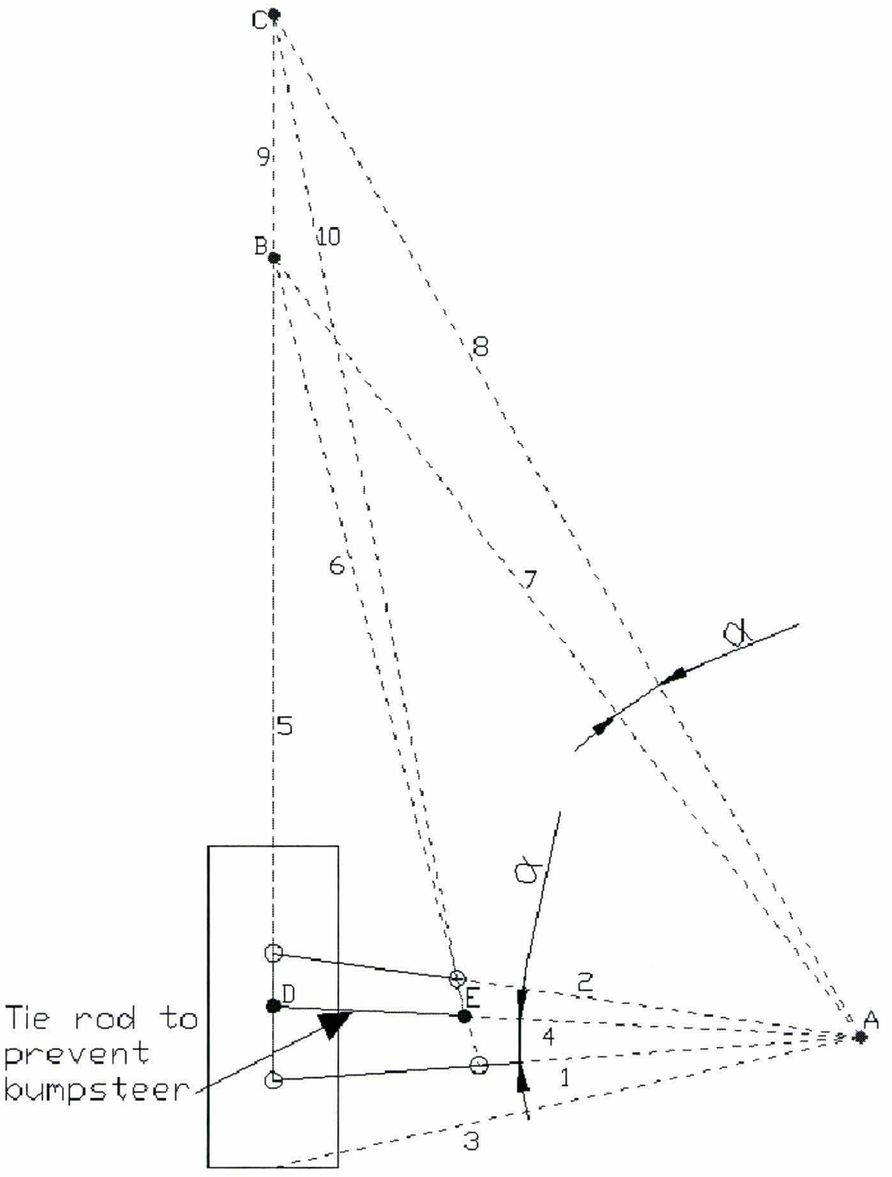


Figure N.1: Construction rear tie rod

Appendix P: Upright analysis

The closed box structure for the upright has been chosen by considering the size of the stresses in the upright for an open structure and a closed box structure. This shown in figure P.1 and P.2 with the maximum stress in brackets under the figures.

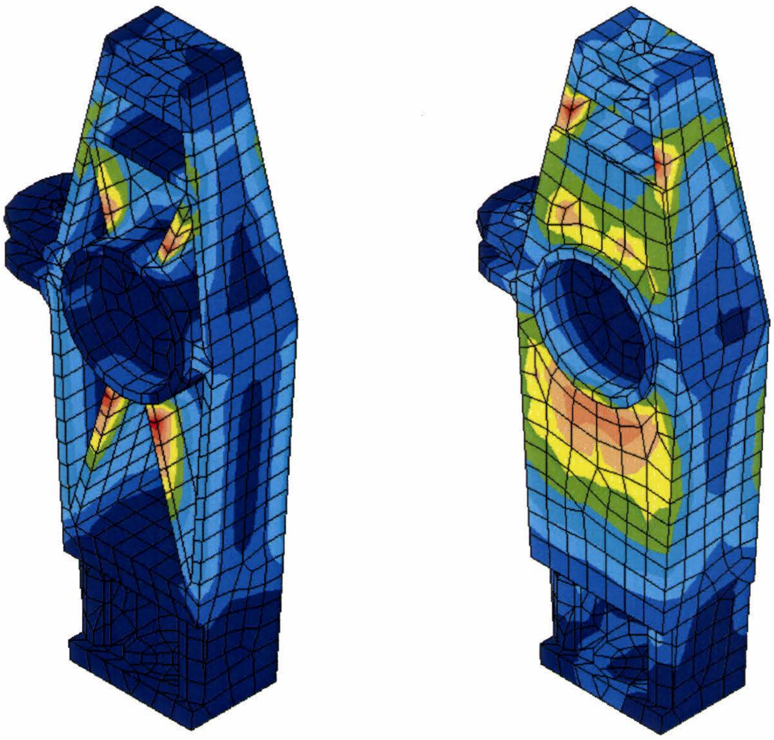


Figure P.1: Rear upright during cornering a) open (212 N/mm^2) b) closed box (76 N/mm^2)

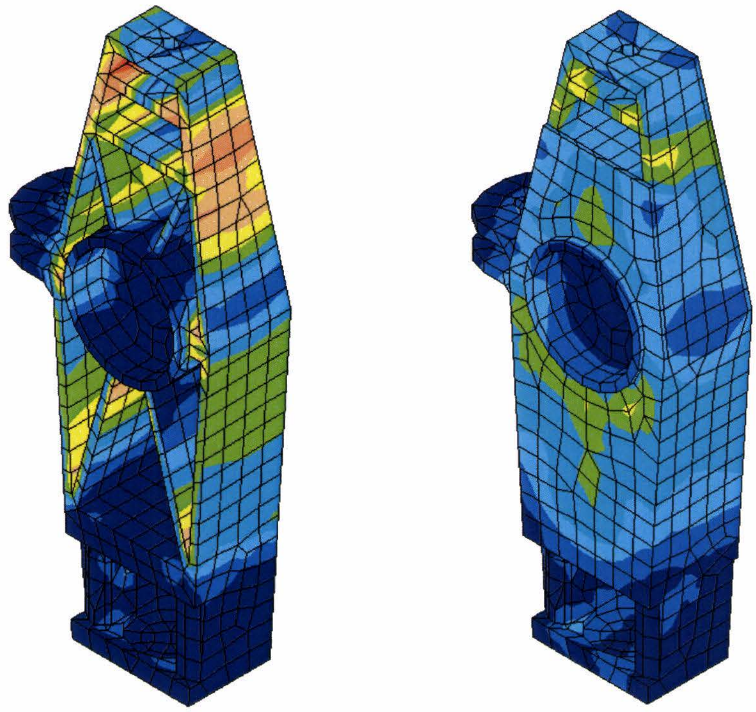


Figure P.2: Rear upright during braking a) open (134 N/mm^2) b) closed box (94 N/mm^2)

Appendix Q: Rear wheel stiffness

For the required stiffness for the spring of the rear suspension the static and dynamic forces on the rear wheels have been determined. The accelerating force has to provide a rear wheel travel of 25,5 mm. The required wheel stiffness has been determined in the same as for the front wheels, which is shown in figure Q.1.

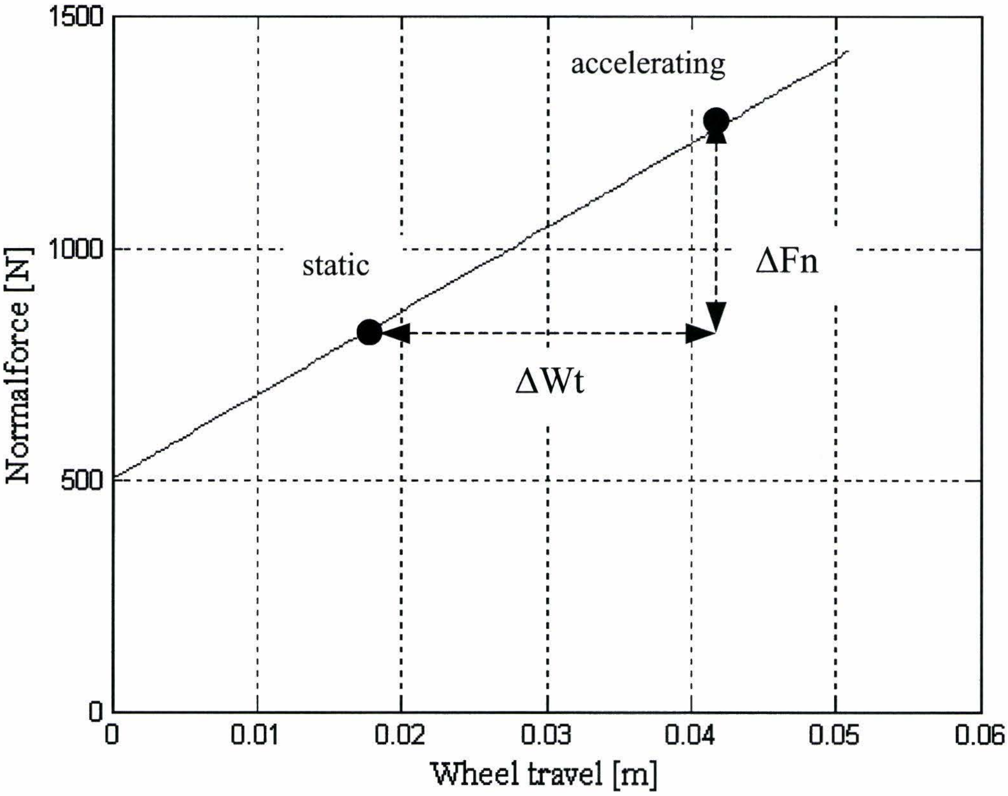


Figure Q.1: Rear wheel stiffness

The required rear wheel stiffness is the steepness of the line in figure Q.1.

$$C_{wheel,rear} = \frac{\Delta Fn}{\Delta Wt} = \frac{Fn_{acc} - Fn_{stat}}{Wt_{acc} - Wt_{stat}}$$

$$C_{wheel,rear} = \frac{1271 - 809}{0{,}0425 - 0{,}017} = 1{,}81 \cdot 10^4 [N/m]$$

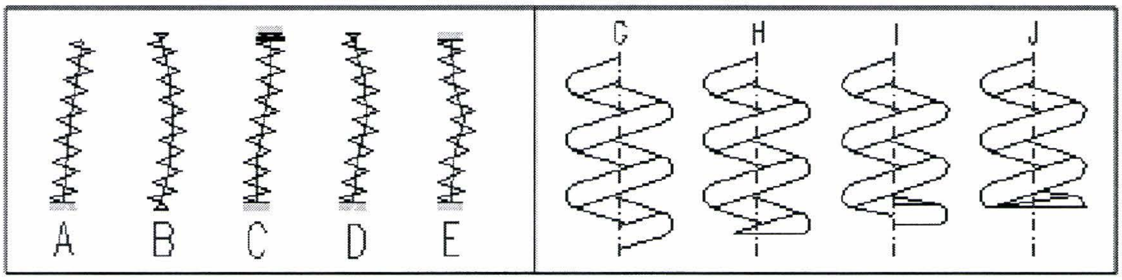
The rocker for the front suspension will be used for the rear suspension as well. This will mean that the ratio for the rocker will also be 1,7.

$$C_{spring,front} = i_{rocker}{}^2 \cdot C_{wheel,front}$$

$$C_{spring,front} = 1{,}7^2 \cdot 1{,}81 \cdot 10^4 = 52{,}3 \cdot 10^4 [N/m]$$

Appendix R: Spring front suspension

?	Input parameters section			
1.0	☑ **Selection of load conditions, spring operational and production parameters.**			
1.1	**Working cycle operational parameters**			
1.2	Method of loading		Static loading	
1.3	Working temperature	T	100,0	[° C]
1.4	Working environment		Non corrosive	
1.5	**Spring design**			
1.6	Seating of the spring		E ... Clamped - clamped ends without lateral restraint	
1.7	Design of spring ends		J ... Closed ends ground	
1.8	Surface treatment		Shot peened springs	
1.9	Direction of coil winding		Right	
1.10	Number of end / ground coils	n_c / n_c	2,0 / 1,0	

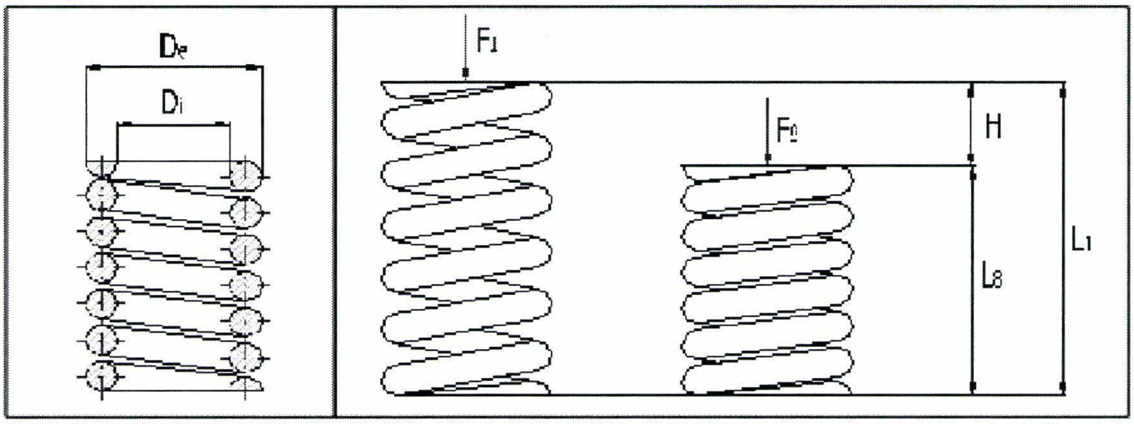


1.11	**Spring exposed to static loading**			
1.12	Operational loading mode		Heavy duty service	
1.13	Desired level of safety	s_s	1,25	
1.14	Method of stress curvature correction		Without correction	
2.0	☑ **Options of spring material.**			
2.1	Spring material :		Stainless steel wire EN 10270-3-1.4568	
2.2	**Field of use of the selected material**			
2.3	Suitability for fatigue load		Good	
2.4	Relative strength		High	
2.5	Corrosion resistance		Good	
2.6	Max. operational temperature		370	[° C]
2.7	Delivered wire diameters		0,25 - 8	[mm]
2.8	**Mechanical and physical properties of the material**			
2.9	Modulus of elasticity in shear	G_{20}	78000	[MPa]
2.10	Modulus of elasticity at operational temperature	G	76050	[MPa]
2.11	Density	ρ	7900	[kg/m³]
2.12	**Strength characteristics of the material**			
2.13	Ultimate tensile strength	S_u	1510	[MPa]
2.14	Permissible torsional stress	τ_A	755	[MPa]
2.15	Endurance limit in shear	τ_e	498	[MPa]
2.16	Endurance limit by finite life	τ_f	498	[Mpa]

☑

3.0 ☑ Spring design.

3.1	**Desired parameters of working cycle**			Deviat. [%]	
3.2	Maximum working loading	F_8	1788,0	2,0	[N]
3.3	Minimum working loading	F_1	1126,0	2,0	[N]
3.4	Fully loaded spring length	L_8	120,0	30,0	[mm]
3.5	Required spring working stroke	H	15,0	10,0	[mm]
3.6	Preloaded spring length	L_1	135	27,78	[mm]



3.7	**Filters of the designed solution**			
3.8	☐ Maximum permissible spring outer diameter	D_{emax}	80,00	[mm]
3.9	☑ Minimum permissible spring inner diameter	D_{imin}	50,00	[mm]
3.10	Permissible division of the number of active coils		1/10	
3.11	Permissible exceeding of spring limit dimensions		20,0	[%]
3.12	Perform check of buckling		Yes	
3.13	Perform check of the limit working length		Yes	
3.14	Keep to the required level of safety with the strength check		Yes	
3.15	Quality criterion		Deviation from desired dimensions	
3.16	Number of design iteration		Medium	
3.17	**Options of solutions**			
3.18	Sort design result by		Qualities of solutions	
3.19	Run design calculation			

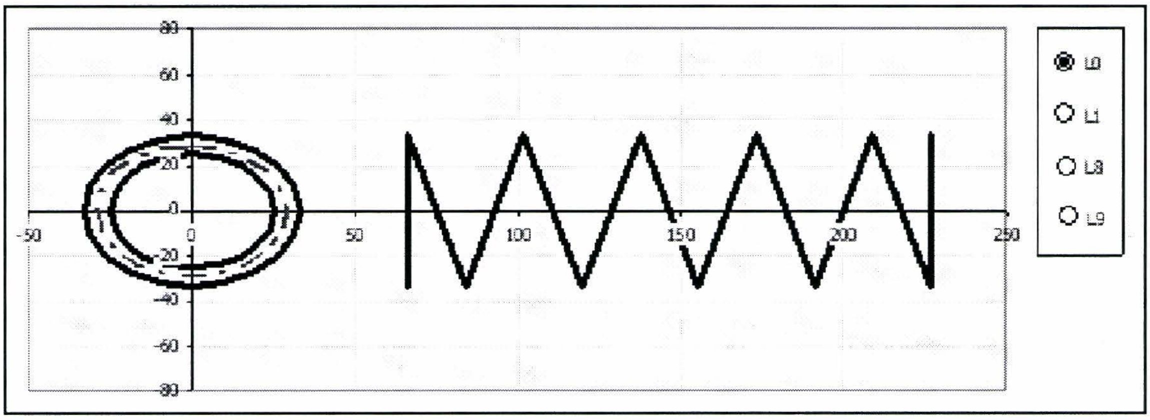
3.20

ID	D	D_e	D_i	d	n	L_0	L_1	L_8	F_1	F_8	τ_8	s_s	s_f	m	quality
1.	58.1	66.1	50.1	8.00	4.50	160.5	135.0	120.0	1126.0	1788.0	517	1.46	0.00	406.0	0.00

Results section

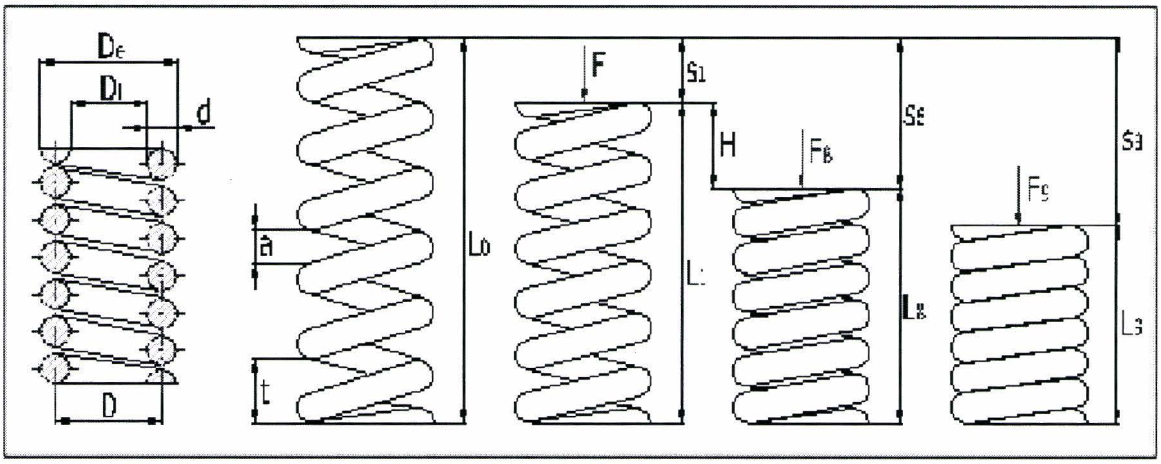
4.0 ☑ Summarized list of designed spring parameters.

4.1 Refresh results from the selected spring design



4.2	**Spring loading**				
4.3	Minimum working loading	F_1	1126,00		[N]
4.4	Maximum working loading	F_8	1788,00		[N]
4.5	**Spring dimensions**				
4.6	Mean spring diameter	D	58,09		[mm]
4.7	Recommended limits of wire diameter	d_{min} / d_{max}	2,90	8,00	[mm]
4.8	Wire diameter	d	8		[mm]
4.9	Outer / inner spring diameter	D_e / D_i	66,09	50,09	[mm]
4.10	Spring index	c	7,26		
4.11	Number of active coils	n	4,5		
4.12	Recommended limits of free spring length	L_{0min} / L_{0max}	94,42	172,84	[mm]
4.13	Free spring length	L_0	160,51		[mm]
4.14	Recommended pitch limits	t_{min} / t_{max}	17,43	34,85	[mm]
4.15	Space / pitch between coils of free spring	a / t	24,11	32,11	[mm]
4.16	**Parameters of preloaded spring**				
4.17	Spring deflection	s_1	25,51		[mm]
4.18	Spring length	L_1	135,00		[mm]
4.19	Spring stress	τ_1	325,32		[MPa]
4.20	**Parameters of fully loaded spring**				
4.21	Spring deflection	s_8	40,51		[mm]
4.22	Spring length	L_8	120,00		[mm]
4.23	Spring working stroke	H	15,00		[mm]
4.24	Spring stress	τ_8	516,58		[MPa]
4.25	**Parameters of spring limit state**				
4.26	Theoretic spring limit loading	F_9	4789,85		[N]
4.27	Theoretic spring deflection / length	s_9 / L_9	108,51	52,00	[mm]
4.28	Teoretic stress	τ_9	1383,87		[MPa]
4.29	Sum of min. permissible spaces between active coils	s_{amin}	6,447		[mm]
4.30	Minimum spring limit length	L_{minf}	58,45		[mm]

4.31	**Spring mechanical and physical properties**			
4.32	Spring constant	k	44,14	[N/mm]
4.33	Spring deformation energy	W_8	36,21	[J]
4.34	Critical spring speed	v_k	25,02	[m/s]
4.35	Natural spring frequency	f	183,96	[Hz]
4.36	Developed wire length	l	1208	[mm]
4.37	Spring weight	m	0,480	[kg]

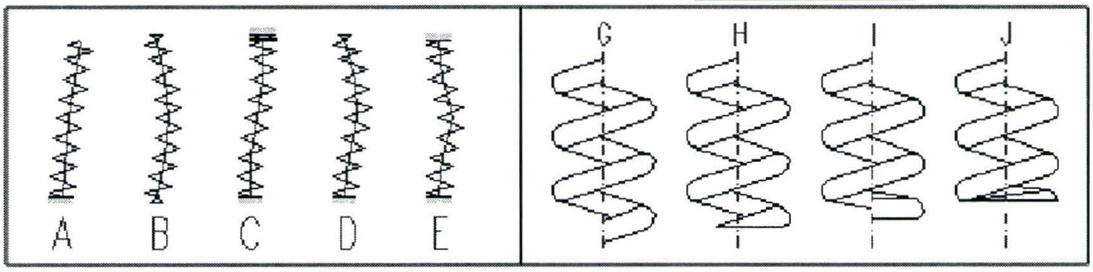


4.38	**Spring strength check**				
4.39	Curvature correction factor	K_s	1,0000		
4.40	Corrected stress of fully loaded spring	τ_{8K}	516,58		[MPa]
4.41	Permissible torsional stress	τ_A	755		[MPa]
4.42	Level of safety		1,462		
4.43	**Check of buckling**				
4.44	Permissible / actual max. working compression of spring		100	25,24	[%]

☑

Appendix S: Spring rear suspension

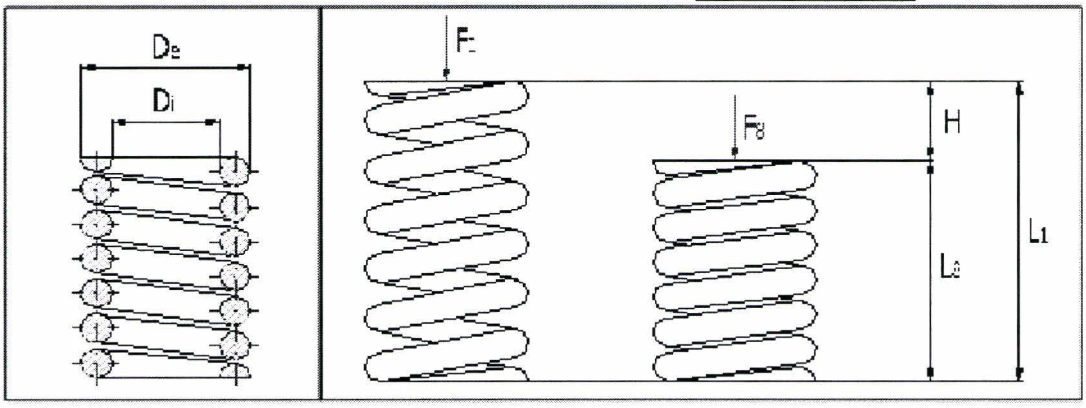
?	Input parameters section			
1.0	**Selection of load conditions, spring operational and production parameters.**			
1.1	**Working cycle operational parameters**			
1.2	Method of loading		Static loading	
1.3	Working temperature	T	100,0	[° C]
1.4	Working environment		Non corrosive	
1.5	**Spring design**			
1.6	Seating of the spring		E ... Clamped - clamped ends without lateral restraint	
1.7	Design of spring ends		J ... Closed ends ground	
1.8	Surface treatment		Shot peened springs	
1.9	Direction of coil winding		Right	
1.10	Number of end / ground coils	n_Z / n_G	2,0 / 1,0	



1.11	**Spring exposed to static loading**			
1.12	Operational loading mode		Heavy duty service	
1.13	Desired level of safety	s_s	1,25	
1.14	Method of stress curvature correction		Without correction	
2.0	**Options of spring material.**			
2.1	Spring material :		Stainless steel wire EN 10270-3-1.4568	
2.2	**Field of use of the selected material**			
2.3	Suitability for fatigue load		Good	
2.4	Relative strength		High	
2.5	Corrosion resistance		Good	
2.6	Max. operational temperature		370	[° C]
2.7	Delivered wire diameters		0,25 - 8	[mm]
2.8	**Mechanical and physical properties of the material**			
2.9	Modulus of elasticity in shear	G_{20}	78000	[MPa]
2.10	Modulus of elasticity at operational temperature	G	76050	[MPa]
2.11	Density	ρ	7900	[kg/m³]
2.12	**Strength characteristics of the material**			
2.13	Ultimate tensile strength	S_u	1510	[MPa]
2.14	Permissible torsional stress	τ_A	755	[MPa]
2.15	Endurance limit in shear	τ_e	498	[MPa]
2.16	Endurance limit by finite life	τ_f	498	[Mpa]

3.0 ☑ Spring design.

				Deviat. [%]	
3.1	**Desired parameters of working cycle**				
3.2	Maximum working loading	F_8	2160,0	2,0	[N]
3.3	Minimum working loading	F_1	1375,0	2,0	[N]
3.4	Fully loaded spring length	L_8	120,0	30,0	[mm]
3.5	Required spring working stroke	H	15,0	10,0	[mm]
3.6	Preloaded spring length	L_1	135	27,78	[mm]



3.7	**Filters of the designed solution**			
3.8	☑ Maximum permissible spring outer diameter	D_{emax}	80,00	[mm]
3.9	☑ Minimum permissible spring inner diameter	D_{imin}	45,00	[mm]
3.10	Permissible division of the number of active coils		1/10	
3.11	Permissible exceeding of spring limit dimensions		20,0	[%]
3.12	Perform check of buckling		Yes	
3.13	Perform check of the limit working length		Yes	
3.14	Keep to the required level of safety with the strength check		Yes	
3.15	Quality criterion		Deviation from desired dimensions	
3.16	Number of design iteration		Medium	
3.17	**Options of solutions**			
3.18	Sort design result by		Qualities of solutions	
3.19	Run design calculation			

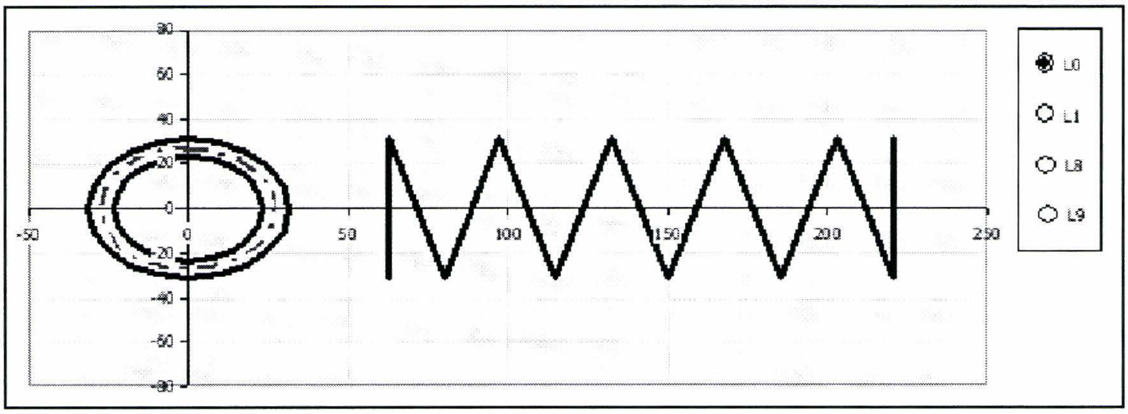
3.20

ID	D	D_e	D_i	d	n	L_0	L_1	L_8	F_1	F_8	τ_8	s_s	s_f	m	quality
4.	54.5	62.5	46.5	8.00	4.60	161.3	135.0	120.0	1375.0	2160.0	585	1.29	0.00	456.9	0.00

Results section

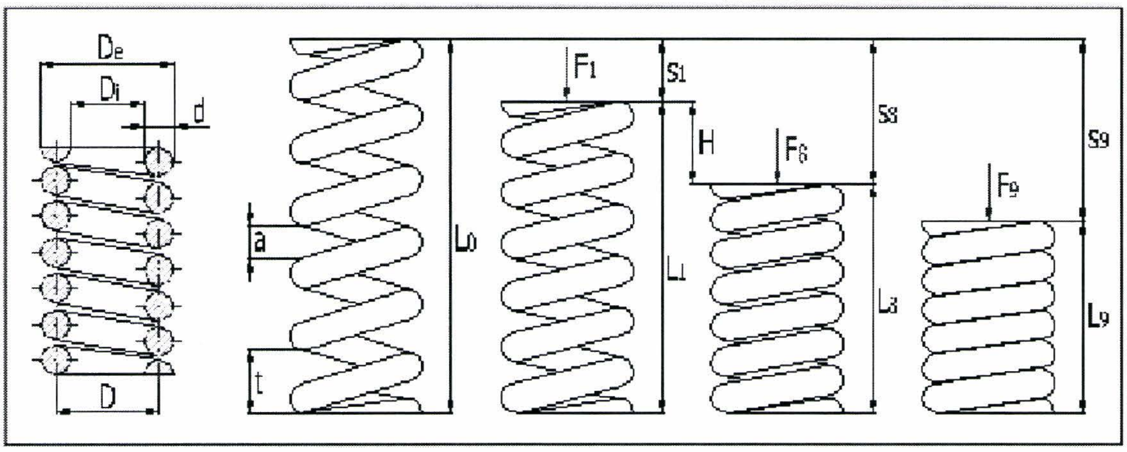
4.0 ☑ Summarized list of designed spring parameters.

4.1 Refresh results from the selected spring design



4.2	**Spring loading**				
4.3	Minimum working loading	F_1	1375,00		[N]
4.4	Maximum working loading	F_8	2160,00		[N]
4.5	**Spring dimensions**				
4.6	Mean spring diameter	D	54,49		[mm]
4.7	Recommended limits of wire diameter	d_{min} / d_{max}	3,41	8,00	[mm]
4.8	Wire diameter	d	8		[mm]
4.9	Outer / inner spring diameter	D_e / D_i	62,49	46,49	[mm]
4.10	Spring index	c	6,81		
4.11	Number of active coils	n	4,6		
4.12	Recommended limits of free spring length	L_{0min} / L_{0max}	91,20	166,39	[mm]
4.13	Free spring length	L_0	161,27		[mm]
4.14	Recommended pitch limits	t_{min} / t_{max}	16,35	32,69	[mm]
4.15	Space / pitch between coils of free spring	a / t	23,58	31,58	[mm]
4.16	**Parameters of preloaded spring**				
4.17	Spring deflection	s_1	26,28		[mm]
4.18	Spring length	L_1	134,99		[mm]
4.19	Spring stress	τ_1	372,64		[MPa]
4.20	**Parameters of fully loaded spring**				
4.21	Spring deflection	s_8	41,29		[mm]
4.22	Spring length	L_8	119,98		[mm]
4.23	Spring working stroke	H	15,00		[mm]
4.24	Spring stress	τ_8	585,38		[MPa]
4.25	**Parameters of spring limit state**				
4.26	Theoretic spring limit loading	F_9	5675,06		[N]
4.27	Theoretic spring deflection / length	s_9 / L_9	108,47	52,80	[mm]
4.28	Teoretic stress	τ_9	1538,00		[MPa]
4.29	Sum of min. permissible spaces between active coils	s_{amin}	6,241		[mm]
4.30	Minimum spring limit length	L_{minF}	59,04		[mm]

4.31	**Spring mechanical and physical properties**			
4.32	Spring constant	k	52,32	[N/mm]
4.33	Spring deformation energy	W_8	44,59	[J]
4.34	Critical spring speed	v_k	27,48	[m/s]
4.35	Natural spring frequency	f	204,52	[Hz]
4.36	Developed wire length	l	1151	[mm]
4.37	Spring weight	m	0,457	[kg]



4.38	**Spring strength check**				
4.39	Curvature correction factor	K_s	1,0000		
4.40	Corrected stress of fully loaded spring	τ_{8C}	585,38		[MPa]
4.41	Permissible torsional stress	τ_A	755		[MPa]
4.42	Level of safety		1,290		
4.43	**Check of buckling**				
4.44	Permissible / actual max. working compression of spring		100	25,60	[%]

Appendix T: Anti roll mechanism

The tubes for the anti-roll mechanism have to be rotated 90 degrees around the axle through point 15 and 16, but for the working of the mechanism these tubes are placed in one plane with the suspension. The braking and acceleration movement are shown in figure T.1, T.2 and T.3 and the cornering movement is shown in figure T.4, T.5 and T.6.

braking and accelerating

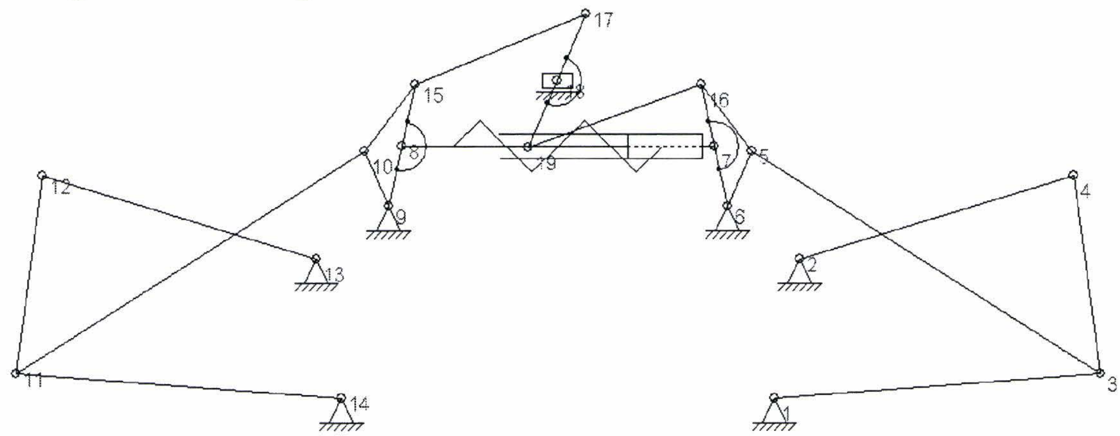


Figure T.1: Bump 34 mm

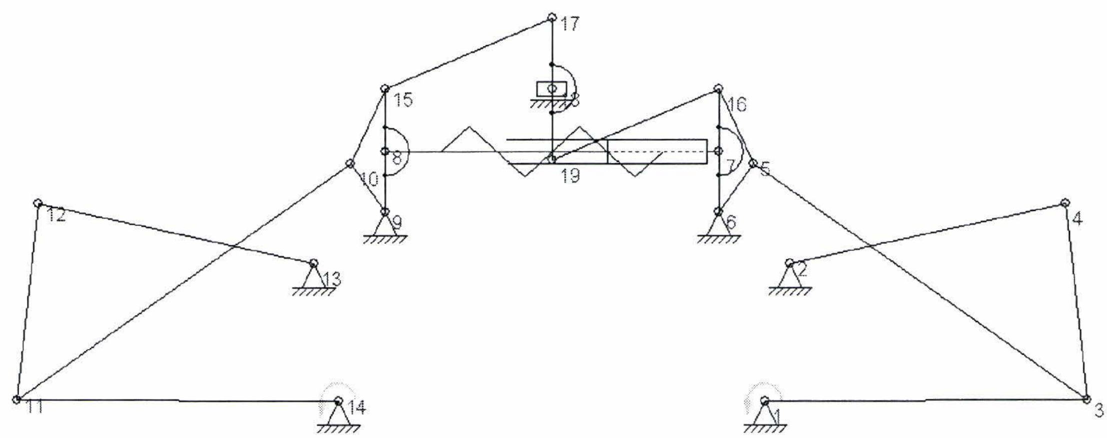


Figure T.2: Static situation

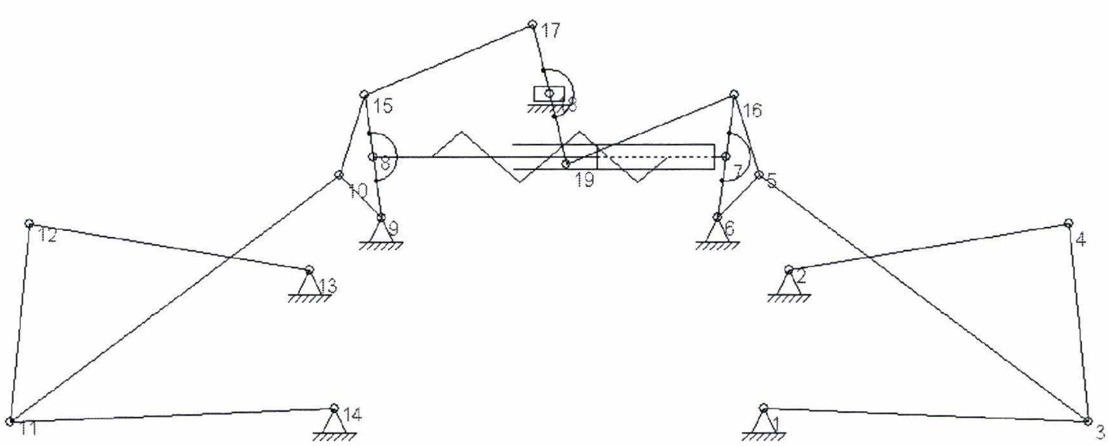


Figure T.3: Droop 17 mm

Cornering

For the movement during cornering the assumption has been made that the suspension will rotate in respect to the chassis. Therefore the road surface will be considered to be parallel to the surface patch of the tires.

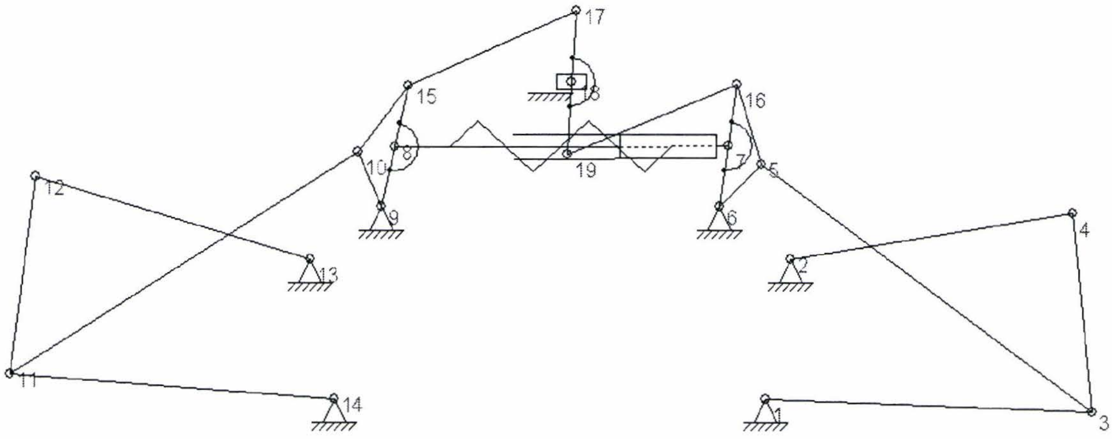


Figure T.4: Right wheel is outer wheel

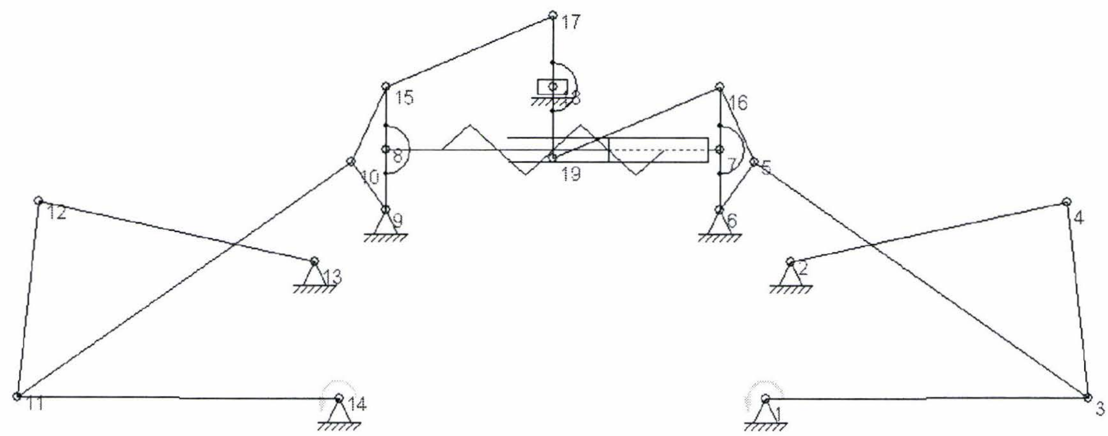


Figure T.5: Static situation

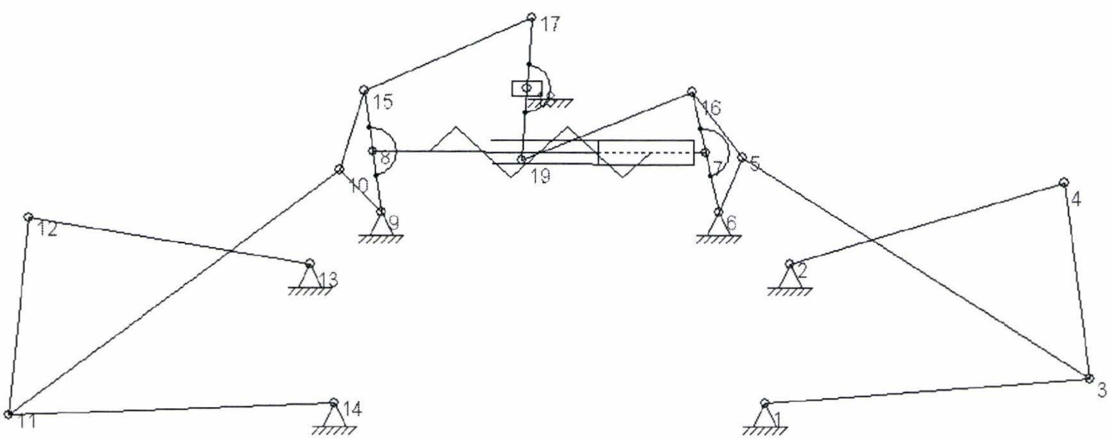


Figure T.6: Right wheel is inner wheel

Appendix V: Offset front and rear wheel

The shape of the wheels has been determined using the required space for the front and the rear suspension and the desired handling of the car. Figure V.1a shows a section of the front wheels and figure V.1b a section of the rear wheels.

a) b)

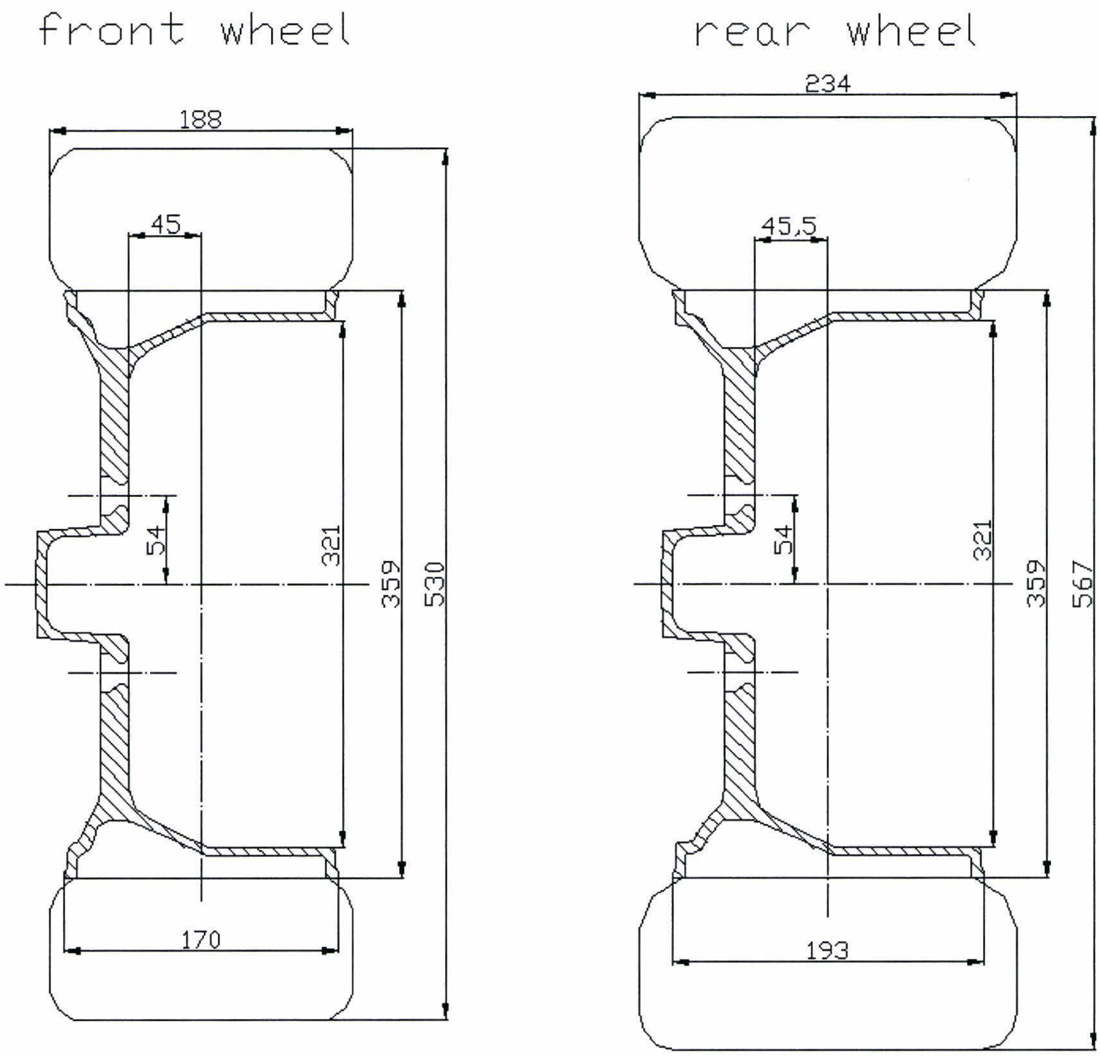


Figure V.1: Offset wheels a) section front wheel b) section rear wheel

Appendix W: FEA parts

To determine the points with the highest stresses some parts have been analyzed regarding the load cases that will be applied to those parts. Some parts are shown in figure W.1 till W.5.

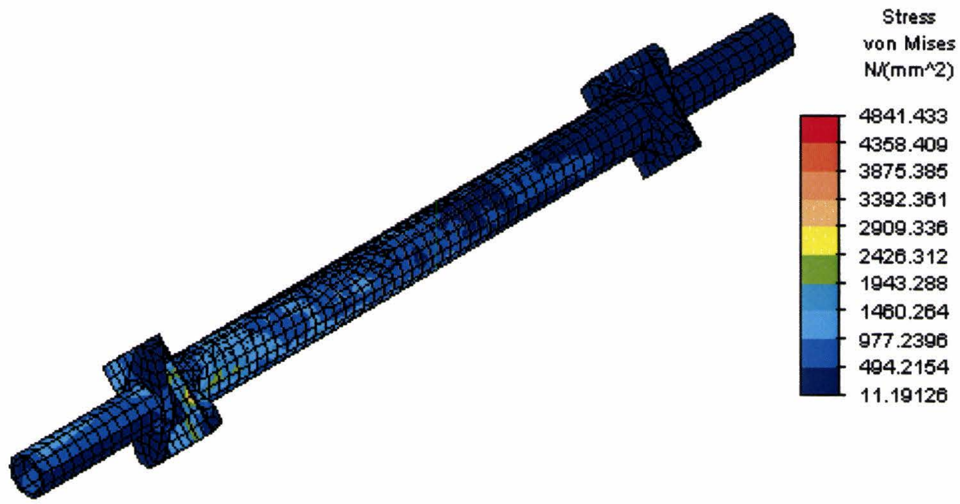


Figure W.1: Stress distribution in drive shaft

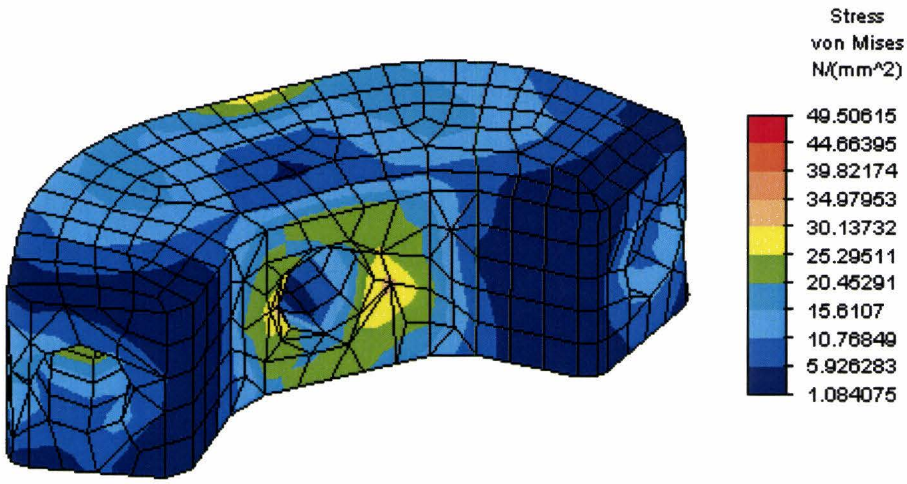


Figure W.2: Stress distribution in aluminium suspension block

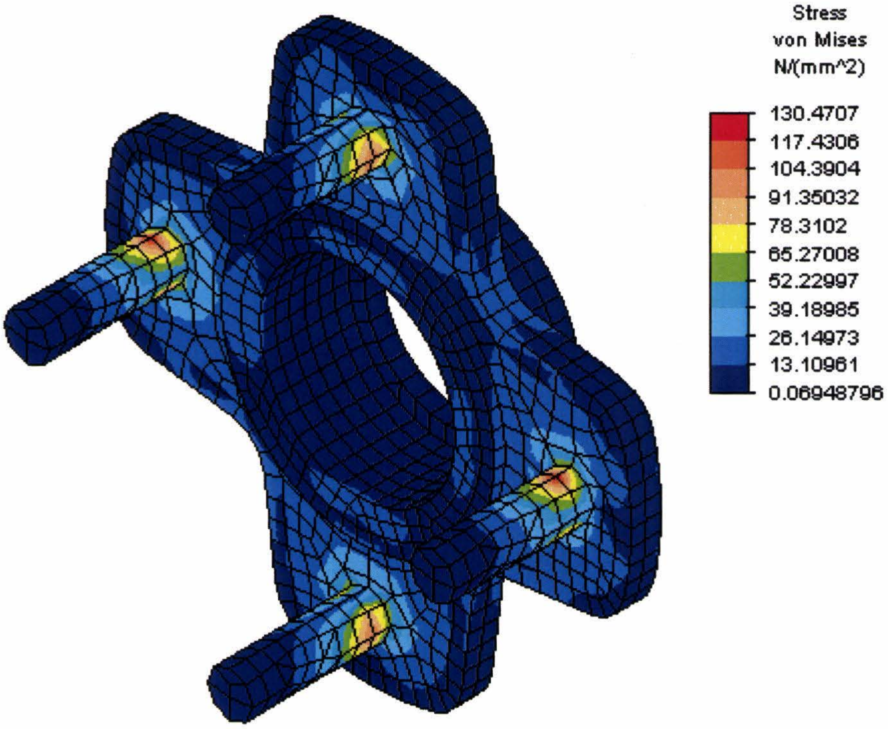


Figure W.3: Stress distribution in hub

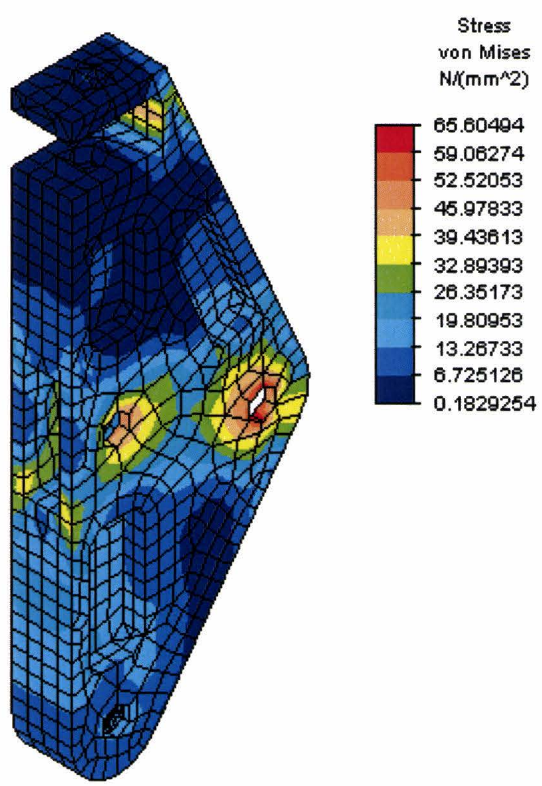


Figure W.4: Stress distribution in rocker during braking