Vehicle dynamics
analysis and design for a
narrow electric vehicle

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DCT 2009.110

Master’s Thesis

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Eindhoven, November 2009
Preface

Ever since my youth, my interest in cars has been great; nevertheless I had not expected this interest could also apply to the simulation of a virtual vehicle. During both my traineeship and the research leading to this work, this interest has grown. Moreover, it became clear to me that vehicle development is impossible without the application of some form of vehicle simulation. Nowadays, computational analysis and design probably are the most important tools in automobile development. It is likely that in the future the importance of this discipline will only develop further. From my perspective, knowledge about vehicle dynamics simulation is a must for my generation of automotive engineers interested in this field of technology.

The vehicle of subject in this research is an interesting new approach to personal transport, combining high maneuverability in cities and sporty driving performance with guided automated highway travel. The fully electric drive of the CITO is state-of-the-art and fits well in current developments in legislation and public opinion on road transport. However, from my perspective, electric vehicles will not be common property in the next 20 years as not only the technology inside the vehicle is not yet up to that challenge, but also the infrastructure has not reached the required level. Moreover, for me the question whether electric propulsion is the ultimate solution for a vehicle with the usage as we know it, is not yet answered convincingly. And even if it were, the storage of electricity inside the vehicle is only an energy carrier and a sound solution for the generation of electric energy has yet to be found. From a vehicle dynamics point of view however, the vehicle offers great opportunities.

During the 14 months of research, at times the work did not progress as planned in advance. Interestingly, the research done in this project was entirely new to the company Small Advanced Mobility. One particular thing I found out is the fact that the number of people able to help out grows inversely proportionally with the increase in complexity of the matter at hand. In the end, comprehensive knowledge of vehicle dynamics and especially vehicle handling is a rare good. Therefore I would like to particularly thank Igo Besselink for his competent reflection on my progress and for the knowledge transferred. Furthermore I would like to thank Alex Serrarens for his constructive criticism, trust and clear view of the subject, Henk Nijmeijer for keeping me on track during the entire research and Carlo Luijten as committee member for going into the work I present here, the result of my research in the past 14 months. Additionally, I would like to thank Dirk van Sambeek, founder of Small Advanced Mobility and committed director of his company for the opportunity to carry out this research on the interesting CITO vehicle. I would like to thank my colleagues at Small Advanced Mobility and its parent company Modesi for the healthy atmosphere at work and especially the laid-back atmosphere during our trip to the 2009 Frankfurt Motor Show.

Before the start of this research project, expectations were high. Although it is always possible to keep making improvements and every research leads to new opportunities, in the end a feeling of satisfaction remains.

*Bas van Putten, November 2009*
Summary

This research presents vehicle dynamics modeling, a modular suspension design and four wheel steering and all wheel drive algorithm development for a narrow electric vehicle. The vehicle is called CIty auTO (CITO) and is the first vehicle to be developed by Small Advanced Mobility B.V. (SAM). From a vehicle dynamics perspective, the reduced vehicle width compared to a conventional vehicle poses challenges in several fields.

This research is focused at gaining insight in and mapping of these fields in order to design a suspension concept that provides maximum performance and ensures optimized roll over resistance. The base-line vehicle geometric parameters are assessed, leading to a proposed increase in vehicle track width from 0.92 [m] to 1.20 [m]. In order to design a suspension geometry for this vehicle, a vehicle dynamics simulation tool based on the Matlab/Simulink SimMechanics environment is developed. This tool consists of a vehicle model and a driver model. The novel position feedback driver model is based on a virtual reference point in front of the vehicle, which is used to calculate tracking steering wheel angles. The vehicle model is built in a generic form and is completely parameterizable. A reference vehicle model with MacPherson strut front suspension and torsion bar rear axle is developed and verified using vehicle measurement data. An initial double wishbone suspension design is presented, using a suspension analysis tool. Simulation results of the CITO vehicle equipped with the double wishbone suspension are compared to results of the reference vehicle, showing similar dynamic performance.

A literature review is done, assessing and combining results in literature, based on which a choice is made for technologies likely to have a substantial positive effect on vehicle dynamics. Based on proposals made in literature, both novel four wheel steering and four wheel drive algorithms are developed. The velocity dependent four wheel steering design is implemented and extended in terms of functionality. A novel all wheel drive strategy is developed aiming at improving dynamical behavior of a rear wheel drive based vehicle by using vehicle yaw velocity as controlled variable. Simulation results show increased vehicle dynamic performance in several challenging maneuvers, particularly in case of the all wheel steering algorithm. Vehicle roll over resistance is ensured by the geometric suspension lay out, supported by simulation results for several maneuvers showing virtually no increase in height of the centre of gravity. Finally, an optimized double wishbone suspension design is presented, taking into account constructional considerations and altered operating conditions. A qualitative elasto-kinematic analysis is performed aimed at determining suspension bushing stiffness.

Conclusively it is stated that the end result is a vehicle which in simulation provides equal or better vehicle handling and roll over safety compared to a common mid-sized vehicle. This result is obtained in spite of the unfavorable ratio of track width over height of centre of gravity. Finally, recommendations are made for further research, derived from the conclusive statements.
Samenvatting

Dit onderzoek presenteert zowel modelvorming op het gebied van voertuigdynamica, als het ontwerp van een modulair wielophaningsconcept, als de ontwikkeling van vierwielsturings- en vierwieladrijvingsalgoritmen voor een smal, elektrisch voertuig. Het voertuig, genaamd CIty auTO (CITO), is het eerste voertuig dat wordt ontwikkeld door Small Advanced Mobility B.V. (SAM). Vanuit het oogpunt van voertuigdynamica vormt voornamelijk de geringe spoorbreedte ten opzichte van een conventioneel voertuig een probleem op diverse gebieden. Dit onderzoek is erop gericht inzicht te vergaren in en het in kaart brengen van deze gebieden, teneinde een wielophaningsconcept te ontwerpen dat maximale prestaties op het gebied van handling combineert met uitmuntende roll over veiligheid. Daartoe zijn ten eerste de initiele geometrische parameters geanalyseerd, wat leidt tot een toename van de spoorbreedte van het voertuig van 0.92 [m] tot 1.20 [m]. Teneinde een correcte wielophanging voor het voertuig te kunnen ontwerpen, is een voertuig dynamica simulatie tool ontwikkeld, baserend op de Matlab/Simulink SimMechanics omgeving. Deze tool bestaat uit zowel een bestuurdersmodel als een voertuigmodel. Het nieuwe ontwikkelde positie feedback bestuurdersmodel is gebaseerd op een virtueel tracking punt dat zich voor het voertuig uit beweegt en aanleiding geeft tot correcte, met de praktijk vergelijkbare, stuurwielhoeken. Het voertuigmodel is op generieke wijze opgebouwd en is compleet parametriseerbaar. Een referentievoertuig is met MacPherson voorwielophaning en torsie-achteras is ontwikkeld en geverifieerd gebruikmakend van meetdata. Een initieel ontwerp voor een wielophanging met dubbele draagarmen is gepresenteerd, gebruikmakend van een wielophaning evaluatie tool. Simulatieresultaten van het CITO voertuig uitgevoerd met het voorgestelde dubbele draagarm ontwerp zijn vergeleken met het referentievoertuig en tonen vergelijkbaar dynamisch gedrag. Een literatuurstudie is uitgevoerd, waarbij resultaten uit literatuur zijn geverifieerd en met elkaar gecombineerd, waarna de technologien gekozen zijn met de grootste potentie tot verbetering van het voertuigdynamisch gedrag. Gebaseerd op de voorstellen uit literatuur zijn zowel een nieuw vierwielsturingsalgoritme als een nieuw vierwieladrijvingsalgoritme ontwikkeld. Het vierwielsturingsysteem is afhankelijk van de voertuigsnelheid en is uitgebreid in termen van functionaliteit. Het vierwieladrijvingsysteem is erop gericht het dynamisch gedrag van het achterwielaangedreven voertuig te verbeteren middels giersnelheidsterugkoppeling. Simulatieresultaten tonen aan dat het dynamische gedrag van het voertuig zeer is verbeterd in een aantal uitdagende manoeuvres. Voornamelijk het vierwielsturingsysteem laat hier zeer goede resultaten zien. Roll over veiligheid wordt gegarandeerd door de specifieke orientatie van de draagarmen. Uiteindelijk is een geoptimaliseerde wielophanging gepresenteerd, waar voortschrijdend inzicht en constructieve overwegingen geïmplementeerd zijn.

Concluderend wordt gesteld dat het eindresultaat een voertuig laat zien dat, in simulatie, even goede of zelfs betere handling combineert met een hogere mate van roll over veiligheid, vergeleken met een voertuig uit de middenklasse. Dit resultaat wordt bereikt ondanks de ongunstige verhouding tussen spoorbreedte en hoogte van het voertuig. Afsluitend worden aanbevelingen gedaan voor nader onderzoek.
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1 Introduction

“That the automobile has practically reached the limit of its development is suggested by the fact that during the past year no improvements of a radical nature have been introduced.”

-Scientific American, June 2, 1909

1.1 Background

Predicting the future turns out to be rather difficult. Looking back at 100 years vehicle development since then it is fair to state that the automobile underwent a radical change. Not only the way it looks, the way it is propelled and the velocities it is able to reach have changed, but the entire perspective on public and personal transport has gradually developed into the form it nowadays has. Of course not only the automobile is responsible for this, yet it has played an exceptionally important role. In fact, these are turbulent times still, if only because of the fact that vehicle manufacturers are desperately looking for new ways to sweep a largely saturated market. For over 30 years the idea has existed to reduce the use of and dependency on fossil fuels, for a diversity of reasons. In the past 5 years, emphasis is on the transfer to electric propulsion; the electrification of personal transport. The foundation for this transition is laid by early research in electric vehicles starting from the times before the internal combustion engine was invented, and evidently by the development of hybrid vehicles combining an internal combustion engine with electric energy storage and an electric drive unit. Meanwhile, hybrid vehicles are available in almost all vehicle classes and in every imaginable form, ranging from the small Smart Fortwo MHD with start/stop system up to the limousine BMW 7-series ActiveHybrid. From an efficiency perspective, an evolution of the hybrid vehicle is the Full Electric Vehicle (FEV). Although at the moment this type of personal transport is in the centre of interest, not one major vehicle manufacturer has introduced such a vehicle into the market yet. Mainly this is due to the fact that electric storage energy density is not competitive with fossil fuels, meaning that in order to achieve an equal range, the electric vehicle is much heavier. Moreover, this added mass itself contributes negatively, because more energy is required for acceleration, leading to higher energy consumption. Therefore, the choice usually is made to keep a somewhat equal mass and consequently a smaller range. This small range subsequently demands a high density, high power distribution network, such that the vehicle can be charged for example both at home and at a given destination. Such an energy distribution network however requires enormous investment, which is not profitable as long as there are only few electric vehicles. Time will tell whether within the foreseeable future Full Electric Vehicles will start conquering the market.
1.2 Research objective and thesis outline

The CITO vehicle is the first vehicle to be developed by Small Advanced Mobility (SAM). One of the key features of the vehicle is its extreme narrowness compared to any other vehicle available on the market. The aim for a vehicle as narrow as possible originates from the desire to greatly increase transportation efficiency. From a vehicle dynamics perspective, this reduced vehicle width poses challenges in several fields. This research is focused at gaining insight in and mapping of these fields in order to design a suspension concept that provides maximum performance. The main research objective therefore is formulated as follows.

*Development of a modular suspension concept for the CITO vehicle, i.e. a narrow electric vehicle, accommodating essential technology for improving vehicle dynamic performance*

In order to achieve this goal a second, preparatory objective is formulated.

*Development of a comprehensive and generic vehicle dynamics simulation tool for the CITO vehicle in all relevant configurations*

These two objectives are treated in reversed order in this report. In Chapter 2 the baseline vehicle is discussed. Based on the parameters describing this vehicle, an analysis is done aimed at assessing the starting point vehicle geometry in order to create a solid basis to further develop the vehicle and a suspension design. The vehicle dynamics simulation tool, essential in the proceedings of this research, is discussed in Chapter 3. Not only the actual vehicle model is presented, but also a driver model calculating steering wheel angles based on a prescribed trajectory and a suspension kinematics analysis tool are described in detail. Chapter 4 discusses matching a reference vehicle model to real vehicle measurement data. This reference vehicle model is subsequently used to make a comparison with the CITO vehicle model in Chapter 5. At this point a suspension geometry is designed and analyzed. Chapter 6 opens with a literature review of vehicle dynamics enhancement technologies, leading to a choice of technologies to be used in the CITO vehicle. Subsequently, simulations are performed evaluating the functionality of each of these technologies in the CITO vehicle. Chapter 7 combines results of and experience from previous chapters to optimize the suspension design proposed in Chapter 5. Additionally, a qualitative elasto-kinematic analysis is presented. Finally, a conclusion is drawn on the presented research and recommendations are made on both the vehicle dynamics simulation tool and the suspension design and corresponding vehicle behavior.
2 General vehicle description

The Small Advanced Mobility CITO Dual Mode, SAM CDM in short, is a new mobility concept, combining the idea of increasing overall road capacity and the introduction of high tech electrical propulsion concepts. The Dual Mode system provides the driver the opportunity to drive fully automatically guided on highways as well as fully manually controlled within cities and on rural roads. As traffic congestion is currently one of the largest discomforts in daily life, costing businesses and individuals enormous amounts of money, radically new ideas for mobility are necessary. Out of the box thinking is strongly encouraged by national governments, leading to numerous grants for new development in this field of R&D. This chapter describes the general outline of the project, in section 2.1 the vehicle is presented that is subject of the research presented in this report and in section 2.2 an evaluation of the baseline vehicle determining key vehicle parameters is presented.

2.1 Small Advanced Mobility CITO Dual Mode

The CITO, which stands for CIty auTO, is a narrow vehicle in a 1+1 configuration. The idea is initially developed in 1994, starting as the three wheeled bicycle called FITO, which evolved into a completely new form of transportation. Currently, the vehicle is no longer propelled by human power, but driven by a high tech electrical drive train allowing the vehicle to drive up to 200 km as well as reaching a maximum speed of over 140 km/h. The vehicle is designed as the ideal means of transportation for commuters, traveling alone for relatively short distances at a time. For this matter, the vehicle has no need to be as wide as a regular vehicle providing space for an entire family and luggage. Although these restrictions seem to bound applications of the vehicle, development is aimed at driving comfort and agility similar to a current mid-sized vehicle. The vehicle is shown in Figure 2.1.

As its name implies, the SAM CDM can be operated using a unique Dual Mode system, allowing the driver to sit back and relax or work during automated highway travel, as well as providing manual driving in the city and on rural roads. During automated driving, the vehicle is connected via wireless communication to a batch of approximately 6 vehicles driving in a special U-shaped concrete infrastructure, preferably constructed alongside existing highways. Figure 2.2 shows the vehicle in its infrastructure, which can be prefabricated and set up without using extensive foundation. This both reduces the cost and
increases the speed at which new roads can be built, even temporarily to reduce traffic density, for example during road works or special events.

Figure 2.2 CITO and infrastructure alongside a congested highway

During automated highway travel the vehicle is not driven on or near the boundaries of vehicle dynamics, bends and highway exits are smooth and generally there is no need to swerve around road hazards. This means that from a vehicle dynamical point of view, the automated mode is hardly of interest. In manual mode however, the driver is free to use the accelerator, brakes and steering. As the vehicle is no longer in its own infrastructure, it is exposed to the same hazards as a common current vehicle and has to provide equal active and passive safety. Combining this claim with the very narrow track width, necessary to fit in the CDM infrastructure, poses a challenge in the development of this dual mode vehicle.

2.2 Evaluation of the CITO
For the SAM CDM, development is split in two phases;

- Prototype development
- Series production vehicle development

During the research, as there is no real vehicle at hand, focus is on prototype development. Naturally, series production is kept at the back of one’s mind. At present, following components are developed and ready;

- Interior and interior shell
- Exterior design including a ‘scissor door’
- Ideas about drive train, chassis and suspension

Research currently focuses on following fields;

- Chassis design
- Electric drive train, batteries and electric motors
- Connected drive, automated mode
- Vehicle dynamic simulation
- Suspension design
As part of the general idea of the vehicle, a number of basic parameters is determined ruling the dimensions of the vehicle. Table 2.1 shows these parameters for the SAM CDM and a common mid-sized vehicle [1].

Table 2.1 Initial choice of vehicle parameters CITO and those of a mid-sized vehicle

<table>
<thead>
<tr>
<th>Parameter</th>
<th>SAM CDM</th>
<th>Mid-sized vehicle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Approximate vehicle length</td>
<td>3.25 m</td>
<td>4.20 m</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>2.48 m</td>
<td>2.58 m</td>
</tr>
<tr>
<td>Approximate vehicle width</td>
<td>1.05 m</td>
<td>1.78 m</td>
</tr>
<tr>
<td>Approximate vehicle height</td>
<td>1.40 m</td>
<td>1.48 m</td>
</tr>
<tr>
<td>Approximate height centre of gravity</td>
<td>0.49 m</td>
<td>0.60 m</td>
</tr>
<tr>
<td>Track width</td>
<td>0.92 m</td>
<td>1.54 m (front and rear)</td>
</tr>
<tr>
<td>Ratio track over height CoG</td>
<td>1.88</td>
<td>2.57</td>
</tr>
<tr>
<td>Approximate vehicle mass</td>
<td>900 kg</td>
<td>1117 kg</td>
</tr>
</tbody>
</table>

From Table 2.1 it becomes clear that compared to a common mid-sized vehicle, the CITO has an exceptionally low ratio of track width over height of centre of gravity. Although the vehicle height is approximately equal, Figure 2.3 shows the difference in width compared to a common vehicle. The narrow track width increases the risk for roll over as well as reduces dynamical capabilities of the vehicle.

Figure 2.3 CITO parked next to a common vehicle

As equal safety compared to a mid-sized car is intended, roll over properties of the CITO are investigated and compared with the mid-sized vehicle. To this end, first literature assessing roll over modeling is reviewed. Then, a roll over model is chosen and simulations are performed to determine the required track width of the CITO.

2.2.1 Roll over modeling literature review

Roll over has gained more and more interest with the introduction and massive success of the Sports Utility Vehicle (SUV), especially in the United States. Specific, standardized maneuvers such as the fishhook test, described in section 3.4.4, have been introduced to assess a vehicle's roll over properties. This focus on roll over has lead to different approaches in both modeling of roll over and detecting and preventing roll over in simulation and reality.

Firstly, in literature a distinction is made between directly caused, intrinsic roll over and so called tripped roll over, caused by an obstacle on the road. Eger et al. (2001) argue that a simplified two dimensional vehicle model with two or three degrees of freedom already
provides insight in properties of vehicle roll over [3]. This simplified model can then be used to determine an analytic stability boundary, helping to detect imminent roll over. In general, the vehicle is viewed from behind in the y-z plane and is simplified by a two-mass-spring-damper system with two rotational degrees of freedom around the x-axis. Roll over is said to occur when the vehicle hits a curb sideways with a certain impact velocity and roll velocity of the vehicle body around the x-axis does not return to zero. Odenthal et al. (1999) present a slightly different approach, focused at intrinsic roll over, starting from the same basis of a simplified model [4]. In this case, a two dimensional three degree of freedom model is proposed, where reduction of tyre vertical load to zero of one of the tyres, under influence of lateral acceleration of the centre of gravity, is said to represent a roll over situation. Although this is a simplification, it provides a relation between the moment of roll over, lateral acceleration and vehicle basic geometry. For a definition of geometrical parameters, see Figure 2.4. Please note that the roll axis shown in this figure, is neglected.

\[ R = \frac{2(h_R + h) \alpha_y}{T g} \]  

(2.1)

A roll over coefficient is defined which is 0 when no lateral acceleration is applied, meaning the two wheels on one axle have the same vertical load. When the roll over coefficient becomes 1, the vertical load of one of the wheels becomes 0 and subsequently it is lifted off the ground. Of course, this does not conclusively mean the vehicle tips over, but as the model represents a conservative definition, it is useful.

2.2.2 CITO roll over evaluation

The key equation in this comparison is the roll over coefficient described in literature. [4]

The roll over coefficient in (2.1) is based on the standard 3 DOF vehicle model shown in Figure 2.4 with rigid axles and a roll axis. Of course, this type of suspension does not resemble to reality for either the CITO or the mid-sized vehicle, but because there is no choice yet on the type of suspension to be applied, it provides a defendable approximation. As is shown in (2.1), the roll over coefficient depends on the height of the centre of gravity, vehicle track width and lateral acceleration. When the roll over coefficient reaches the value of 1, one of the wheels loses contact with the road, which is assumed to be equal to a roll over situation.
The roll over coefficient is set to 1 to be able to calculate maximum attainable lateral acceleration at the moment roll over occurs. Figure 2.5 shows the line of intersection between the roll over surface and the level plane of maximum attainable lateral acceleration of the VW Golf, i.e. a mid-sized vehicle currently available on the market. This line represents a combination of track width and height of centre of gravity that corresponds to a similar roll over coefficient as the VW Golf. It also shows the position of the CITO in respect to this line. In order to achieve the same basic behavior, the track width or centre of gravity or both should be moved towards line representing the equivalent of the middle sized vehicle, as indicated by the arrow.

![Figure 2.5 Height of centre of gravity versus track width for the CITO and mid-sized vehicle](image)

Based on the roll over model analysis using the current estimated height of centre of gravity, the track width should be increased to approximately 1.25 [m]. This corresponds to the intersection of a line in pure horizontal direction starting from the CITO point, with the line shown in Figure 2.5. Maintaining the current track width, the height of the centre of gravity has to be reduced to approximately 0.35 [m], which is almost at wheel centre height.

At present, the complete packaging of the vehicle is not yet known which means that the height of the centre of gravity is uncertain. As the narrowness of the vehicle is one of the key features and roll over properties can be influenced by suspension design, an increase of track width of the vehicle to 1.20 [m] is proposed. This does not entirely correspond to the analysis of the roll over model, but takes into account that clever suspension design can influence roll over behavior of a vehicle. For suspension design this implies that roll angles of the body have to be kept small and the height of the centre of gravity has to remain constant.

In order to achieve optimal cornering behavior, the location of the centre of gravity in longitudinal direction is aimed at 50/50 [%] between front and rear. Table 2.2 shows the final vehicle parameters used in the continuation of this research.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>CITO</th>
</tr>
</thead>
<tbody>
<tr>
<td>Approximate vehicle length</td>
<td>3.25 m</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>2.48 m</td>
</tr>
<tr>
<td>Approximate vehicle width</td>
<td>1.40 m</td>
</tr>
<tr>
<td>Approximate vehicle height</td>
<td>1.40 m</td>
</tr>
<tr>
<td>Approximate height centre of gravity</td>
<td>0.49 m</td>
</tr>
<tr>
<td>Track width</td>
<td>1.20 m</td>
</tr>
<tr>
<td>Ratio track over height CoG</td>
<td>2.45 -</td>
</tr>
<tr>
<td>Approximate vehicle mass</td>
<td>900 kg</td>
</tr>
<tr>
<td>Weight distribution (front/rear)</td>
<td>50/50 [%]</td>
</tr>
</tbody>
</table>
3 Simulation model

The simulation model is designed in a modular, entirely parameterizable way to be able to easily tune certain vehicle configurations and provide a solid basis to compare results. The simulation model provides a choice of standard maneuvers and the possibility to define new ones, for example a circuit. The entire vehicle model is parameterized, such that different vehicle configurations can be evaluated using the same vehicle model. Nonlinear spring and damper characteristics can be implemented, as well as active suspension algorithms. In this chapter, the entire simulation sequence is presented, from maneuver description in 3.1, via the open loop and closed loop driver model in 3.2 and the vehicle model itself in 3.3 to post processing of simulation results in 3.4. Additionally, the suspension kinematics evaluation model is discussed in 3.5. A manual for the vehicle dynamics simulation tool is presented in Appendix C: Simulation manual.

3.1 Maneuver preprocessing

In order to evaluate vehicle dynamics of any vehicle, a choice of standardized maneuvers is provided. Each of these maneuvers assesses a different aspect of vehicle dynamics, such that in the end clear insight is gained in vehicle behavior in both steady state and dynamic situations. The maneuvers provided with the simulation package are summed up in Table 3.1 below, along with the additional information about this maneuver.

<table>
<thead>
<tr>
<th>Maneuver</th>
<th>Standard</th>
<th>Additional information</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steady state circular</td>
<td>ISO 4138:1982</td>
<td>Closed loop, increasing velocity</td>
</tr>
<tr>
<td>Double lane change</td>
<td>ISO 3888-1</td>
<td>Closed loop, free rolling</td>
</tr>
<tr>
<td>Moose test</td>
<td></td>
<td>Closed loop, free rolling</td>
</tr>
<tr>
<td>Fishhook test</td>
<td>NHTSA Fishhook 1a</td>
<td>Open loop, free rolling</td>
</tr>
<tr>
<td>Step steer input</td>
<td>ISO 7401:2003</td>
<td>Open loop, free rolling</td>
</tr>
</tbody>
</table>

A more detailed description of each of the maneuvers can be found in section 3.4.

The maneuver preprocessing sequence derives reference path input data for the driver model, depending on maneuver prescribed in position coordinates or direct steering wheel angles. In the former case, the simulation is executed in closed loop mode, where a driver model interprets the prescribed trajectory and calculates required steering wheel angles. In the latter case, steering wheel angles are provided directly and thus there is no need for interpretation by the driver model.

In closed loop trajectory prescription, maneuvers are defined in longitudinal $x$- and lateral $y$-position. In order for the driver model to work properly, additionally the traveled distance $s$
is required. In essence, the driver model controls a defined position in the vehicle by evaluating its $x$- and $y$-position corresponding to the current $s$-position. A type identifier is introduced to be able to distinguish closed loop from open loop simulation, which in this case is set to 1. An arbitrary trajectory is presented in Figure 3.1, showing both the desired track in the top figure and corresponding traveled distance $s$ in the lower figure.

![Figure 3.1: Trajectory in x-y-coordinates; Bottom: Corresponding traveled distance $s$](image)

Table 3.2 shows the $s$-$x$-$y$-array corresponding to the trajectory presented in Figure 3.1.

**Table 3.2 Example of $s$-$x$-$y$-table defining a trajectory for closed loop simulation**

<table>
<thead>
<tr>
<th>$s$</th>
<th>$X$</th>
<th>$Y$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>2.41</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>3.41</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>5.41</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>10.41</td>
<td>6</td>
<td>7</td>
</tr>
</tbody>
</table>

In open loop trajectory prescription on the other hand, the driver model directly outputs prescribed steering wheel angles at the defined moment in time. This means the maneuver is defined in steering wheel angle $\delta_{stw}$ and time $t$. The closed loop identifier is set to 0. Table 3.3 shows an overview of outputs generated by maneuver preprocessing for both open loop and closed loop trajectory prescription.
Table 3.3 Simulation preprocessing output

<table>
<thead>
<tr>
<th>Open loop reference prescription</th>
<th>Closed loop trajectory prescription</th>
</tr>
</thead>
<tbody>
<tr>
<td>Array of steering wheel angles</td>
<td>Array of x-values</td>
</tr>
<tr>
<td>Array of time instances</td>
<td>Array of y-values</td>
</tr>
<tr>
<td>Type identifier 0 (open loop)</td>
<td>Type identifier 1 (closed loop)</td>
</tr>
<tr>
<td>Vehicle initial velocity</td>
<td>Vehicle initial velocity</td>
</tr>
<tr>
<td>Array of vehicle velocity targets</td>
<td>Array of vehicle velocity targets</td>
</tr>
<tr>
<td>Simulation stop time</td>
<td>Simulation stop time</td>
</tr>
<tr>
<td>Maneuver name</td>
<td>Maneuver name</td>
</tr>
</tbody>
</table>

3.2 Driver model

The driver model incorporates two separated control loops for velocity and steering control. First the steering controller is described, after which the velocity controller is discussed. Both control loops interact with one another, however the choice is made to separately control each loop.

3.2.1 Steering controller

The steering controller is split into two sections, one for open loop steering control and one for closed loop position control. A type identifier, defined in maneuver prescription, enables the steering controller to distinguish closed loop from open loop steering control. In case of open loop steering control, the algorithm first maps the current simulation time $t_s$ onto the specified time array $t_c$ by searching the minimum value of $\Delta t = \min (t_s - t_c)$. The index $i$ of this entry in the time array is subsequently used to extract the corresponding steering wheel angle $\delta_{stw} = \delta_t(i)$. This approach is equal to a Look-up-Table in Matlab/Simulink, however the incorporation of this open loop steering controller in the driver model s-function makes this an integrated approach.

In order to prevent unrealistic steering wheel angles, $\delta_{stw}$ is then limited to $[\frac{-3\pi}{2}, \frac{3\pi}{2}]$. There is no filtering afterwards and the steering wheel angle is transferred directly into the steer rack, where the steer ratio is applied to calculate wheel steer angles.

If the type identifier indicates closed loop control, the second section of the steering controller is activated. As the vehicle has only one actuator, namely the steering wheel angle, it already becomes clear that it is not trivial to solve the closed loop control problem at all time. Therefore, a number of assumptions is made.

- The vehicle tracking error is small (no large deviations from the intended path)
- As a result, the vehicle orientation angle needs not be controlled

A schematic view of the driver controller is presented in Figure 3.2. The rear wheel steer angles $\delta_{rear}$ are left uncontrolled at the moment, meaning these steer angles remain zero.
Starting from these assumptions, a controller is designed which operates with a virtual preview point in front of the vehicle, depending on the vehicle speed and in line with the vehicle’s longitudinal axis. A schematic representation of the vehicle in top view is shown in Figure 3.3. This figure presents an arbitrary desired trajectory, an actual trajectory which by some unknown reason is perturbed from its reference and preview point P and corresponding preview distance $\Delta s_p$ ahead of the vehicle. This preview distance is linearly dependent on vehicle forward velocity and becomes larger with higher vehicle velocity, though never under 5 [m] and never over 50 [m].

Figure 3.3 Schematic representation of vehicle on trajectory showing preview point and preview distance

Figure 3.4 shows the same situation, though mapped on an x-y-grid. The location of the centre of gravity of the vehicle is denoted as C, with corresponding current location $(X_c, Y_c)$. According to its current position, the vehicle has a traveled distance $s_c$ calculated by integration of vehicle forward velocity with initial value 0 [m/s]. The location of preview point P is only determined in terms of s by

$$s_P = s_c + \Delta s_p$$

As every closed loop maneuver is defined in a traveled distance s-array with corresponding x-y-arrays, this traveled distance $s_c$ can be mapped onto the array to extract corresponding x-y data of the reference trajectory. However if this point is chosen as target point, the approach is not robust for every situation and does not deliver optimal performance.
Therefore not only the exact corresponding x-y data is extracted from the array, but a region is cut out of the array around the exact s-position. The size of this region is determined by the preview distance and the region mainly lies ahead of the s-position of the preview point. This region that is cut out of the desired trajectory is shown in Figure 3.4, along with the region $[X_{pos}, Y_{pos}]$ of possible target points. Also the absolute vehicle orientation angle $\psi$ is shown, which subsequently is used to transform the array of possible target points to local vehicle coordinates by

$$
\begin{align*}
    x_l &= x_{pos} - x_c \\
    y_l &= y_{pos} - y_c
\end{align*}
$$

and

$$
\begin{bmatrix}
    x_{loc} \\
    y_{loc}
\end{bmatrix} = \begin{bmatrix} \cos \psi & \sin \psi \\ -\sin \psi & \cos \psi \end{bmatrix} \begin{bmatrix} x_l \\
    y_l \end{bmatrix}
$$

Both $x_{loc}$ and $y_{loc}$ consist of points in local vehicle coordinates which are possible target points. In Figure 3.5 these target points are represented as points $t_n$, where $n$ denotes the size of the cut out region.
For every point, distance $\Delta P t_n$ to point $P$ is calculated and the point with the shortest distance to $P$ is chosen as target point. In the schematic view in Figure 3.5, this is point $t_3$. This target point is subsequently used to calculate the error angle

$$e_{\psi} = \tan^{-1} \left( \frac{y_{\text{loc}}(t_d)}{x_{\text{loc}}(t_d)} \right)$$

where $x_{\text{loc}}(t_d)$ the local $x$-coordinate, respectively $y_{\text{loc}}(t_d)$ the local $y$-coordinate represents of the target point. The actual steering wheel angle corresponding to the error angle is calculated with a gain, which makes the steering driver model a closed loop P-controller, as follows

$$\delta_{\text{stw}} = K \cdot e_{\psi}$$

Figure 3.6 shows another view of the situation presented in Figure 3.3, where target point and corresponding wheel steer angles $\delta_1$ and $\delta_2$, derived using the steer ratio incorporated in the steer rack model as will be described in section 3.3.3, are displayed.
The approach with a velocity dependent preview point ahead of the vehicle corresponds to drivers in reality, tending to look further in front of their vehicle when driving faster. Furthermore, the target point on the reference trajectory corresponds to a real driver aiming at a certain point on the road.

3.2.2 Velocity controller
The driver model velocity controller is based on a table containing reference velocity data, generated by the maneuver preprocessing files presented in section 3.1. This table, in Matlab/Simulink implemented as a Look-up-Table, is used to generate a velocity set point, which subsequently is used by a P-controller to control vehicle forward velocity. Other controllers, such as PD- and PID-controllers have been tested, but not found more effective. The output of the P-controller is fed into a torque distribution algorithm, which for the basic rear wheel driven car splits the torque 50/50% between the two rear wheels.

3.2.3 Driver model simulation
To investigate the intrinsic behavior of the driver model, several simulations are performed. These simulations are subdivided in open loop simulations and closed loop simulations.

The open loop simulation is relatively straightforward and provides information about whether current simulation time is correctly mapped onto the defined reference array and whether the steering wheel angle is fed through directly. To this end, the open loop fishhook maneuver is prescribed. Figure 3.7 shows the result of this simulation, showing no difference between the reference and actual signal.

In order to examine the closed loop behavior, two simulations are performed. Firstly, to examine steady state target determination, a circular test is prescribed. This simulation provides insight in target determination under sub-critical circumstances, with relatively low velocity and no large changes in steering wheel angle. The test conditions are described in Table 3.4.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circle radius</td>
<td>100 m</td>
</tr>
<tr>
<td>Initial vehicle velocity</td>
<td>5 m/s</td>
</tr>
<tr>
<td>Simulation time</td>
<td>7 s</td>
</tr>
</tbody>
</table>
As results both the actual tracking behavior and the target determination and search area are presented. The former provides clear insight in overall performance of the driver model, whereas the second provides more detailed information and insight in the driver model control sequence. Figure 3.8 shows the reference trajectory during simulation as well as the actual path of the vehicle centre of gravity and sampled target points determined by the driver model. Figure 3.9 shows the region of y-positions as a function of time, used by the driver model to calculate the point closest to the preview point. This search area naturally grows as the simulation progresses, because the steady state circular test prescribes an increasing velocity and hence an increase in preview distance and desired trajectory cut-out.

Figure 3.8 Semi-steady state tracking behavior

Figure 3.8 presents perfect tracking and well defined driver targets during this part of the steady state circular test. Figure 3.9 on the other hand shows the expanding y-region, which is used for target point determination. An equal figure could have been presented for the x-region. In section 3.2.1 these regions combined are called $[X_{\text{pos}}, Y_{\text{pos}}]$. 

Figure 3.9 Semi-steady state target determination and expanding search area because of increasing velocity

The steady state results show correct and stable target calculation in sub-critical driving conditions. This provides a basis on which to proceed to a critical maneuver in which steering wheel angle change rates are large and the vehicle is driven on the limits of tyre grip. A moose test, as described in section 3.4.3, is performed to investigate driver model behavior in critical circumstances. The initial vehicle speed in this maneuver is $15 \text{ m/s}$. 

24
during the maneuver the vehicle is in free rolling condition, meaning braking nor driving torque is applied to the wheels.

![Figure 3.10 Driver model tracking evaluation during moose test](image1)

Figure 3.10 shows the tracking behavior of the driver model, presenting both the driver model targets and the actual trajectory of the centre of gravity of the vehicle. It is clear that tracking during this maneuver, where lateral accelerations of up to 7 [m/s²] are reached, is within a boundary of 0.3 [m]. For the nose of the vehicle, this result is even better. Sharp edges in the prescribed trajectory are rounded off because of the preview distance.

![Figure 3.11 Target calculation with y-target boundaries](image2)

Figure 3.11 shows the search area which is used by the driver model to calculate target point. In this particular situation it seems the target point is located halfway between the minimum and maximum value for the Y-target, which to a large extent holds for situations where the grip boundaries of the vehicles tyres are not reached.

It is possible to even further optimize tracking behavior, but it has to be kept at the back of one’s mind that the goal for the driver model is to reflect real driver behavior. To verify the steering wheel angle calculated as output of the driver model, simulation results are compared to measurement data. As a reference, the steering wheel angle of a test driver measured during a severe double lane change maneuver is used, see section 3.4.2. [5] In simulation, the Peugeot 306 GTI model is used, see Chapter 4.
As is shown clearly in Figure 3.12, output steering wheel angles reflect realistic behavior of a test driver. However, there are some dissimilarities noticeable, especially in the smoothness of the steering wheel signal. There are several reasons for these differences. Firstly, differences are caused by the fact that the driver model reacts more quickly to changes in vehicle stability and the influence of this on the vehicles orientation than a real driver. Secondly, the driver model and trajectory description cause the vehicle to also track the line connecting the two lanes. A real driver would only focus on the two straight lanes, the trajectory in between is irrelevant. The fact that the 306 GTI model does not exactly correspond to the real vehicle used in measurements, adds a third reason for differences. Finally, between $t=2.5 \, [s]$ and $t=3.5 \, [s]$, the steering wheel angle in simulation does not go back to zero, showing a relatively large deviation from the measured angle. This is explained by the vehicle model showing a slightly more understeered behavior than the real vehicle.

3.3 Vehicle model

The vehicle model is developed in the generic Matlab/Simulink environment. This platform on the one hand provides the comprehensive multi body modeling toolbox SimMechanics and on the other hand comfortable integration of possible control strategies, such as all wheel steering, ESP and active suspension. The main components of the SimMechanics vehicle model are discussed in this section, starting with an overview, suspension modeling, the transfer of steering wheel angles from the driver model and finally the implementation of nonlinear spring characteristics.
3.3.1 Vehicle model overview
Providing maximum resemblance with real vehicle lay out, the vehicle model is built using the same structure as a regular vehicle. Figure 3.13 shows an overview of the vehicle model.

Figure 3.13 Four wheel steering, all wheel drive vehicle simulation model overview containing: 1. Vehicle body; 2. Suspension; 3. Tyre model; 4. Steer rack; 5. Roll stabilizer

The model overview shows that the model is prepared for four-wheel steering and all-wheel drive. Depending on the vehicle configuration, either one of those options can be chosen. Additionally, it is possible to define multiple, interchangeable, suspension topologies. This way, different suspension geometries can be evaluated using the same vehicle model.

The parameters controlling the entire simulation, starting from basic geometry data of the vehicle up to suspension stiffness and suspension hard points, are loaded from one parameter file. As the entire vehicle model is parametrized, each of these parameters, such as track width, roll stabilizer thickness etc., can be changed and differences in results can be assessed quickly. For each different suspension set up, there is a separate parameter file containing all hard points, defined entirely in vehicle local coordinates, such that the suspension automatically adapts to the general vehicle geometry, such as wheelbase and track width. An example of this parameter file can be found in Appendix A.

3.3.2 Suspension
Being one of the most important components of the vehicle model, the suspension is modeled as a separate block. This block is masked, providing the possibility to automatically adjust suspension coordinates to their current location on the vehicle, leaving the user the option to choose the location of the wheel the suspension is attached to. The suspension model itself is composed in a comprehensive way, providing insight to the user. An example of a double wishbone suspension model is shown in Figure 3.14 below.
The suspension bodies are connected to the large dark shaded dummy block on the left, number 1 in Figure 3.14, which is welded onto the vehicle body. Doing so, both the top level of the vehicle model remains relatively clean and different suspension geometries can be easily connected to the vehicle body using the same weld location. At the very right of Figure 3.14, the lightly shaded upright, number 5, and wheel connector are shown. Between the dummy body and the upright, the separate medium shaded links are shown; in this case these are triangular wishbones number 2 and 3. In this particular case, the spring and damper, number 4, rest on the lower triangular link and a nonlinear spring characteristic is implemented, explained in more detail in subsection 3.3.4. The movement of the track rod, number 6, is transferred from the connector port on the lower left, via flexibility onto the upright. More on steering is found in the next section.

3.3.3 Steering
The driver model provides the vehicle model with steering wheel angles, as described in section 3.2. As in a real vehicle, steering wheel angles are transferred to the suspension upright via a track rod. A regular rack and pinion steer rack is modeled, having a thread pitch prescribed in the general parameter file, which makes the steer ratio tunable. A real steering column has a finite stiffness, adding up to the vehicles understeer property. This stiffness causes the driver having to apply a larger steering wheel angle to achieve a wheel steer angle. In order to reflect real vehicle behavior, this stiffness is crucial, as it limits the speed at which wheel steer angles can be developed. As discussed in section 3.3.3, the finite stiffness of the steering column is modeled as part of the suspension block. Figure 3.14 shows in the lower left corner two bodies connected via a prismatic axis and a spring and damper. These two bodies with low mass can translate in one direction, in the course of which the stiffness of the spring and damper are excited.

3.3.4 Spring, damper and stabilizer
As shown in Figure 3.13, the roll stabilizer is modeled in a separate subsystem. This enables the user to define stabilizing forces depending on the difference in suspension travel between the left and right side of each axle. Additionally, this subsystem incorporates a nonlinear spring description, which could serve progressive or degressive spring characteristics, but in general is used to model a bump stop stiffness in the suspension. This bump stop stiffness
models the collision of two bodies, in this case the entirely pressed spring and the spring mount. Without this bump stop stiffness, the movement of the suspension model is not bounded within the limits of a real suspension. In order for the vehicle to rest in its initial position, an exact spring preloading distance is calculated and fed forward into the suspension look-up-table. To achieve more realistic results, also the damper is implemented in form of a look-up table, prescribing damping forces according to spring travel velocity. Figures showing both spring and damper characteristics are presented in Appendix B. Figure 3.15 below shows the implementation of the nonlinear spring and damper characteristic and roll stabilizer in the separate spring/damper/stabilizer subsystem.

![Figure 3.15 Nonlinear spring, damper and roll stabilizer; 1. Look-up-tables representing nonlinear spring and damper; 2. Roll stabilizing force based on suspension travel difference](image)

### 3.4 Post processing

Several sensors are distributed throughout the vehicle model to monitor various states and characteristics of the vehicle. Depending on the maneuver chosen, different figures are plotted. The plotted results are discussed according to each maneuver.

#### 3.4.1 Steady state circular test

As the steady state circular test, as the name implies, is more or less a steady state maneuver. As the vehicle velocity is increased with low acceleration and constant circle radius, this maneuver provides the opportunity to evaluate intrinsic vehicle behavior and investigate the course of various properties over increasing lateral acceleration. Post processing of the steady state circular test plots the figures which are summed up below.

- Steering wheel angle versus lateral acceleration
- Vehicle side slip angle versus lateral acceleration
- Vehicle roll angle versus lateral acceleration
- Lateral velocity versus lateral acceleration
- Yaw gain versus vehicle forward velocity
- Steering wheel angle versus vehicle side slip angle
• Required and maximum lateral force per tyre versus lateral acceleration
• Wheel camber angles versus simulation time

3.4.2 Double lane change
The double lane change test is a dynamical maneuver which assesses the overall lateral dynamics of a vehicle. The maneuver is carried out at high speed 100 [km/h], i.e. high speed, to simulate swerving to avoid an obstacle on the highway. The maneuver is performed in free rolling condition, meaning there is no force applied via braking or driving. During the maneuver, the vehicle has time to somewhat settle between the two lane changes. This makes the maneuver suitable to determine dynamic understeer or oversteer of the vehicle. Figure 3.16 is a graphical representation of the maneuver, showing the boundaries depending on vehicle width between which the vehicle should remain.

Figure 3.16 Double lane change maneuver ISO 3888-1

The following figures are plotted in post processing after executing the double lane change maneuver.

• Absolute y-position versus absolute x-position for both reference trajectory and vehicle trajectory
• Vehicle forward velocity versus simulation time
• Vehicle lateral acceleration versus simulation time
• Vehicle yaw velocity versus simulation time
• Vehicle roll angle versus simulation time
• Vehicle roll angle versus lateral acceleration
• Vehicle height of centre of gravity change versus simulation time
3.4.3 Moose test

In addition to the double lane change maneuver, the so called moose test is performed. This maneuver became well known after a Swedish journalist overturned Mercedes A-Class during this test. It is more critical from a vehicle dynamic point of view than the double lane change as the vehicle has no time to settle between the two direction changes and steering wheel angles and steering wheel angle change rates are larger. In addition, as was clearly shown by the 1997 Mercedes A-Class, combining the roll and pitch movement of the vehicle body on the springs and high lateral acceleration can cause roll over. For a narrow vehicle as SAM CDM, passing this maneuver flawlessly is as critical as essential for the safety of the vehicle.

![Figure 3.17 Moose test](image)

Figure 3.17 Moose test

Figure 3.17 shows a schematic representation of the moose test, clearly indicating the smaller distance between the direction changes and the larger y-displacement compared to the double lane change, as described in section 3.4.2. In post processing, the same figures are plotted as for the double lane change maneuver.

3.4.4 Fishhook test

In contrast with the previous maneuvers, the fishhook test is an open loop maneuver, where steering wheel angles are directly prescribed. The test is performed under the same free rolling condition as the lane change maneuvers, meaning there is no traction or braking applied. Because of its gentle left turn before steering hard right, the maneuver is very suitable to investigate roll over properties of the vehicle.

![Figure 3.18 Fishhook test](image)

Figure 3.18 Fishhook test NHTSA Fishhook 1a with top: schematic representation of the trajectory; bottom: steering wheel angle versus simulation time
Figure 3.18 shows a schematic representation of the fishhook maneuver. Table 3.5 presents the parameters corresponding to Figure 3.18.

### Table 3.5 Parameters NHTSA Fishhook 1a maneuver

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \theta_{\text{inst}} )</td>
<td>Steering wheel angle at ( ay=0.3g )</td>
</tr>
<tr>
<td>( \theta )</td>
<td>( 6.5 \cdot \theta_{\text{inst}} )</td>
</tr>
<tr>
<td>( \dot{\theta} )</td>
<td>( 720 , [\text{deg/s}] )</td>
</tr>
<tr>
<td>( T_1 )</td>
<td>( 0.250 , [\text{s}] )</td>
</tr>
<tr>
<td>( T_2 )</td>
<td>( 3.000 , [\text{s}] )</td>
</tr>
</tbody>
</table>

As the fishhook maneuver is intended to investigate roll over, the following figures are plotted in post processing.

- Vehicle height of centre of gravity change versus simulation time
- Tyre vertical forces versus simulation time

### 3.4.5 Step steer input

The step steer input test, according to ISO 7401:2003, prescribes a step in the steering wheel angle at a certain moment in time. This maneuver gives information about the settling times for lateral acceleration and yaw velocity and thus provides knowledge about the response of the vehicle. In post processing, the following two figures are plotted.

- Lateral acceleration versus simulation time
- Yaw velocity versus simulation time

### 3.5 Suspension kinematics evaluation model

The suspension kinematics model is a supplement to the entire vehicle simulation package, specifically focused on evaluating the kinematic properties of a suspension design. In essence, the suspension kinematics model makes use of the SimMechanics suspension model as described in section 3.3.2, however additional information is collected by numerous supplementary sensors. As the basic lay-out of the model is equal to the suspension model described in 3.3.2, for a schematic representation is referred to Figure 3.14. In essence, a quarter car model with elaborate suspension design is used, where the vehicle body is fixed at suspension neutral position.

In kinematic analysis of a suspension, a distinction is made between bump properties and steer properties. Bump properties are understood to mean phenomena occurring during suspension travel. For this matter, a dummy wheel connected to the suspension is excited causing a vertical displacement \( \Delta h \), shown in Figure 3.19 on the left.
Assessing steer properties of the suspension is done by exciting the track rod, connected to the upright, shown in Figure 3.19 on the right. This displacement $\Delta s$ causes the wheel to rotate mainly about its vertical axis by an angle $\Delta \delta$.

After a suspension travel simulation, the following figures are plotted in post processing.

- Wheel steer angle change versus suspension travel (also called: bump steer)
- Camber angle change versus suspension travel
- Caster angle change versus suspension travel
- Roll center height change versus suspension travel

The following figures are plotted in post processing after a suspension kinematical steering simulation.

- Wheel steer angle versus steering wheel angle
- Camber angle change versus steering wheel angle

In addition, the suspension evaluation model provides the initial, static value of the following design parameters.

- Camber angle
- Caster angle
- King pin inclination angle
- Scrub radius
- Anti-dive / anti-rise percentage for front and rear
- Anti-lift / anti-squat percentage for front and rear

Additionally a transfer function for vehicle ride can be derived. This is done by adding pure vertical movement, i.e. one degree of freedom, to the vehicle body in the kinematics evaluation model. This however transforms the kinematic model into a dynamic model. A specific, known input signal is subsequently to be applied to the wheel. This leads to a quarter car vehicle model, which can be used for a ride analysis in order to roughly derive spring and damper rates according to a chosen ride frequency and comfort criteria.
4 Reference vehicle model

To be able to benchmark behavior of the CITO Dual Mode vehicle, a reference vehicle model is defined. Measurement data of a Peugeot 306 GTI [5] provides the basis for this vehicle model. This vehicle matches best the requirements for the CITO Dual Mode, being a sporty vehicle with dynamic properties similar to a regular mid-size vehicle. Simulation results of the vehicle model are matched with provided measurement data, to ensure optimal correspondence between the real vehicle and the model. First, the model lay out is discussed in section 4.1, after which the results of parameter optimization to match the measurements is presented in section 4.2.

4.1 Reference vehicle lay out

The reference vehicle is based on the same structure as presented in Chapter 3, leading to a separate driver model and vehicle model. The Peugeot 306 GTI is front wheel driven and front wheel steered and uses a MacPherson front suspension, having one large spring and damper strut and a lower suspension arm, together securing the required 5 degrees of freedom, left out finite bushing stiffness.

An approximation of suspension geometry data is derived from this schematic overview. The MacPherson front suspension is modeled in SimMechanics. The SimMechanics scheme is presented in Figure 4.1.

![Figure 4.1 SimMechanics overview of MacPherson front suspension; 1: MacPherson strut; 2: lower arm; 3: upright; 4: track rod](image)

As is clear from the figure above, the basic lay out of the suspension is the same, a massless dummy block connects the suspension to the vehicle body and suspension arms, number 2,
and in this case the main strut, number 1, connects the upright, number 3, to the dummy body. The same flexibility is shown as before, modeling the finite stiffness of the steering column, number 4.

The rear suspension of the Peugeot 306 GTI is a torsion beam rear axle. In the reference vehicle model, this rear suspension is modeled as a prismatic joint, providing the rear wheel with only vertical movement. The vehicle dynamical simplification made is that in the model wheel camber does not change during suspension travel, whereas the real vehicle will show a slightly negative camber under compression. This can be implemented by placing the prismatic axis under a small angle, but as this approach already is a simplification, the choice is made not to. The SimMechanics implementation of the rear axle is shown in Figure 4.2.

![Figure 4.2 Rear axle implementation in SimMechanics; 1: wheel carrying arm; 2: spring/damper; 3: upright](image)

### 4.2 Matching of relevant parameters

To accomplish correspondence between the real vehicle measurements and the reference vehicle model, two important simulations are executed. Firstly, a steady state circular test, see section 3.4.1, is performed in closed loop with the driver model and under the same conditions as the measurements, using the measured velocity profile as input. Secondly, a severe double lane change, see section 3.4.2, is performed to ensure correspondence in dynamical situations. The parameters to be optimized during the simulation sequences are presented in Table 4.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring stiffness front and rear</td>
<td>N/m</td>
</tr>
<tr>
<td>Damper stiffness front and rear</td>
<td>Ns/m</td>
</tr>
<tr>
<td>Roll stabilizer stiffness front and rear</td>
<td>N/m</td>
</tr>
<tr>
<td>Steer ratio</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.1 Parameters to be optimized in matching simulation with measurements
4.2.1 Steady state circular test results

Vehicle dynamics of the aforementioned Peugeot 306 GTI are thoroughly assessed by Dutch Organization for Applied Scientific Research (TNO) in 1997. [5] Measurements are performed on a dry asphalt road surface at the dynamic platform of Idiada situated 70 km south-west of Barcelona (Spain). The driver who carried out the maneuvers, according to ISO standards, is a skilled professional test driver. Vehicle geometric properties such as vehicle mass, dimensions and weight distribution are available. Other particularly for vehicle dynamics analysis important characteristics are to be estimated by assessing the measurement data.

A manual optimization process is chosen in order to estimate unknown vehicle parameters such as suspension settings and steer properties. A first estimate is achieved by manually optimizing suspension settings on vehicle roll behavior during steady state cornering.

![Roll angle vs lateral acceleration in steady state cornering](image)

Figure 4.3 Correspondence vehicle roll angle simulation model to measurements

The vehicle roll angle analysis shows optimal results in tuning both the spring stiffness and roll stabilizer stiffness on the front and rear axle, see Figure 4.3. Final roll angle rate is about 4.5 [deg/g]. Only a roll angle analysis however is not sufficient to determine suspension settings for both axles.

Subsequently other characteristics are assessed in order to determine damping constants front and rear and to fine-tune suspension settings. The following figures show correspondence between the measurement data and vehicle reference model simulation. It is clear that in all cases the reference model only shows slight dissimilarities.
Initial simulations showed larger dissimilarities between the measured and the vehicle model, especially for maximum lateral acceleration. A main cause for the occurrence of these differences is the difference in tyres used. The standard 205/55R15 tyre which comes with the DELFT-TYRE simulation pack is used. To correct these differences, a small adjustment is made to the tyre property file, increasing the peak friction force in lateral direction and cornering stiffness of the tyre by factor 1.2 [1]. Afterwards, the simulation model shows slightly larger vehicle side slip angles compared to the measurement data at high lateral acceleration, which is caused by a slightly larger drift angle of the rear wheels under these circumstances. For this result, see Figure 4.4. However, this difference is small and acceptable.

The slightly larger side slip angles of the vehicle are discussed before, what is shown in Figure 4.5 indicates that the required steering wheel angles to achieve a certain vehicle side slip angle are larger in the simulation model than in measurement data, for high vehicle side slip angles. Overall correspondence is very good.
Overall differences in yaw gain between the measurements and the reference vehicle model are very small, only the peak value for yaw gain is slightly higher for the measured vehicle. This might be due to a slightly lower rear axle complementary cornering stiffness for the simulation model. Possibly the tyre property file for the rear axle should be differentiated from the front tyre property file to compensate for the difference in cornering stiffness. The rear maximum lateral force should be increased by a small amount.

After the manual optimization process over a sequence of simulations, the following values are found for the required parameters, see Table 4.2.

Table 4.2 Optimized parameters for reference vehicle model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring stiffness front</td>
<td>25000 N/m</td>
</tr>
<tr>
<td>Spring stiffness rear</td>
<td>15000 N/m</td>
</tr>
<tr>
<td>Damping constant front</td>
<td>2200 Ns/m</td>
</tr>
<tr>
<td>Damping constant rear</td>
<td>1700 Ns/m</td>
</tr>
<tr>
<td>Roll stabilizer stiffness front</td>
<td>13000 N/m</td>
</tr>
<tr>
<td>Roll stabilizer stiffness rear</td>
<td>30000 N/m</td>
</tr>
<tr>
<td>Steer ratio</td>
<td>1/20 [-]</td>
</tr>
<tr>
<td>Steering column stiffness</td>
<td>1000000 Nm/rad</td>
</tr>
</tbody>
</table>
5 Prototype suspension design

Ever since electric vehicles regained attention because of current discussions about transportation efficiency and CO₂ emissions, ideas for new, modular suspensions are developed. At the centre of these developments is the in-wheel electric motor. This motor is not mounted in the center of the car, but directly drives one of the wheels of the vehicle and can even be used as a constructive part, for example replacing a part of the rim. However, at present a state of the art electric in-wheel motor weighs about 1 [kg/kW], adding up to 15 [kg] to the unsprung mass of the vehicle. This chapter firstly presents goals, boundaries and design limitations for suspension design in section 5.1, after which suspension design for the prototype CITO and its kinematic properties are discussed in section 5.2 using the suspension evaluation tool described in section 3.5. Section 5.3 concludes this chapter with simulation of the entire vehicle containing the developed suspension, assessing ride comfort, vehicle dynamics and roll over. Additionally, a comparison with the behavior of the reference vehicle described in Chapter 4 is made, simulating various maneuvers.

5.1 Goals, boundaries and design limitations

The state of development of the CITO leaves design freedom in suspension design. There exists no structural chassis towards which suspension hard points can be engineered. This makes suspension design leading in technical development of the vehicle. However, requirements exist the suspension design has to comply with. Basic principle of the suspension design is modularity. As several examples in industry indicate, a completely integrated suspension and traction module provides numerous advantages, [6]. Examples of such design are Michelin ActiveWheel and Siemens e-Corner. Advantages of this type of suspension module are summed up below.

- Modularity; same suspension module, for example with different power output stages, can be assembled to any vehicle
- Modularity; all components, suspension, traction, braking and steering, are incorporated in one module

Contrasting with the advantages particularly in the field of cost reduction and space usage are disadvantages of vehicle dynamical nature.

- Increased unsprung mass by up to 50 [%], leading to decreased ride comfort
- Increasing dynamical wheel load fluctuations, leading to decreased grip levels on imperfect road surfaces
- Inferior wheel guidance by one degree of freedom compared to existing suspension designs
- Limited design freedom, for example to reduce roll over tendency for narrow vehicles
Based on this analysis of the existing modular suspension design concepts, goals for suspension design of the CITO are formulated. These goals comply with the main goal of section 1.2.

- Modular design by using mainly the same components on every corner of the vehicle
- High roll over resistance
- Comfortable to sporty ride, corresponding to a sporty mid-sized vehicle
- Dynamic cornering behavior corresponding to a sporty mid-sized vehicle
- Design for all wheel drive and four wheel steering

For basic parameters of the CITO, see section 2.2.

5.2 Suspension design

Combined with the geometrical properties of the vehicle, the suspension kinematics determines the intrinsic behavior of the vehicle. Especially in the field of roll over resistance, ride comfort and vehicle dynamics, suspension geometry is an important factor. Firstly, a conceptual choice is made, determining the main geometric properties of the suspension. Then, optimization of the chosen suspension geometry is done, using the suspension evaluation model of section 3.5. Based on the outcome of the evaluation model, iteratively improvements are made. In this section only the final design after making these optimization steps is presented.

5.2.1 Suspension geometrical concept

During the past century, suspension design has evolved from designs with limited complexity, such as solid beam axles, to highly complex multilink designs with flexible joint bushings and compliances which only since the introduction of the computer can be designed correctly. As this research has a broader scope than just extensive suspension design, the conceptual choice combines maximal design freedom with comprehensiveness. Recalling the basic vehicle lay-out of section 3.3, the vehicle is supposed to be equipped with both four wheel steering and four wheel drive. The exhaustive and respected works of Matschinsky (1987) [7] and Mitschke (1984) [8] provide the basis for suspension design in this research.

Based on the requirements and literature research, the choice is made for a double wishbone suspension design. Figure 5.1 shows a schematic representation of this type of suspension and its main components and the implementation of a double wishbone suspension in 3D CAD.
The suspension design provides control of all essential properties, whereas a conventional suspension such as a MacPherson strut does not provide the opportunity to reduce scrub radius. This is important for driven axles, as the scrub radius amplifies driving and braking forces into the steering column. The double wishbone design additionally provides a solid basis to do more profound research in the direction of a four link or multilink suspension, providing kinematical control of every degree of freedom. In essence, these more elaborate suspension designs are evolved from the double wishbone design.

Figure 5.2 Development of double wishbone suspension to multilink suspension

Figure 5.2 shows the development of a double wishbone suspension on the left to a full multilink suspension on the right. First, either one of the nodes connecting the upright to the wishbones is subdivided into two separate joints. In Figure 5.2 center, this is the lower node. By subdividing the lower joint, two separate control arms are created, leading to an additional degree of control over upright movement. In the right figure in Figure 5.2 also the upper node is subdivided, leading to a single control arm for every degree of freedom of the upright. This solution provides maximum control over the movement, but also is by far the most challenging to design correctly. In this research, the basic double wishbone lay-out as of Figure 5.2 left is developed in detail.

5.2.2 Kinematic analysis

As described in section 3.5, a distinction is made between bump and steer properties and two separate simulations are executed to prevent interference between these two situations. To analyze bump properties of the suspension, the wheel is displaced from -0.10 [m] to 0.10 [m], which represents the absolute realistic maximum bump displacement. Results are shown in the figures below. Special interest is on bump steer, because this effect is disadvantageous for handling and handling feel of the vehicle. Bump steer means wheel steer angle changes due to deflection of the suspension, which makes the vehicle self-steer on bad road surfaces.
As is visible from Figure 5.3, bump steer is almost reduced to zero in the main area of interest. Bump steer on rebound is less important, in Figure 5.3, rebound is negative suspension travel. Over the suspension travel from -0.06 to 0.10 [m], the wheel steer angle change is maximally 0.1 [deg].

It is important that the suspension gains negative camber while positively deflecting. This ensures optimal road holding during cornering, as the outer suspension is compressed. In this case, the outer wheels during cornering gain negative camber, which is advantageous for handling. Maximum camber gain of -7 [deg] is achieved at maximum deflection. This amount of camber gain on steering is large and might compromise driver handling feel and security.
Because the rotation axes of the two wishbones do not line up with the longitudinal axis of the vehicle, the caster angle changes under deflection of the suspension. In this case a negative caster angle means that the suspension attacks road irregularities better. During deflection, the caster angle is lightly increased.

Initially, the roll centre approximately lies at road surface. As the suspension is deflected, the roll centre height declines with a slope of about 1. This, combined with the low initial location of the roll centre, is advantageous for roll over prevention. However, it has to be noted that because of the low roll centre, an anti-roll bar with higher stiffness, whether or not combined with tuned low speed damping, has to be used, to prevent the vehicle from getting too large roll angles. As stated in the subsection dealing with ride, the springs are kept relatively soft so that the stiffer anti-roll bar does not compromise comfort too much.

The steer simulation is executed by translating the end of track rod, which would be connected to the steer rack, by a displacement of -0.07 to 0.07 [m]. As results both the
resulting wheel steer angles are shown in Figure 5.7 and camber gain due to caster angle in Figure 5.8.

![Figure 5.7 Track rod displacement versus wheel steer angle](image)

**Figure 5.7 Track rod displacement versus wheel steer angle**

Figure 5.7 shows that, due to the double wishbone lay-out, wheel steer angles are largely linearly related to track rod travel. This however also implies no Ackermann steering is implemented in the kinematic design of the suspension. Kinematic design of the steering linkage geometry, in this case a steer-by-wire electric steering system, should be aimed at obtaining Ackermann steering.

![Figure 5.8 Camber angle change versus wheel steer angle](image)

**Figure 5.8 Camber angle change versus wheel steer angle**

Figure 5.8 shows camber gain during steering, which is essential to ensure negative camber angles on the outer wheels during cornering. This effect counteracts the camber loss due to vehicle body roll.

Finally, some key values of the suspension are summed up in Table 5.1.
Table 5.1 Key static attributes double wishbone suspension design

<table>
<thead>
<tr>
<th>Attribute</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Camber angle front / rear</td>
<td>0 [deg]</td>
</tr>
<tr>
<td>Caster angle</td>
<td>6.5 [deg]</td>
</tr>
<tr>
<td>Anti-dive front percentage</td>
<td>29 [-]</td>
</tr>
<tr>
<td>Anti-rise rear percentage</td>
<td>19 [-]</td>
</tr>
<tr>
<td>King-pin inclination angle</td>
<td>6.2 [deg]</td>
</tr>
<tr>
<td>Scrub radius</td>
<td>0.01 [m]</td>
</tr>
<tr>
<td>Caster distance at wheel center</td>
<td>0.03 [m]</td>
</tr>
</tbody>
</table>

The difference in anti-dive of the front suspension and anti-rise of the rear suspension, though the suspension is (almost) mirrored, is caused by the brake force distribution between front and rear, which is assumed to be 0.6.

5.3 Full vehicle simulation

In order to verify behavior of the entire vehicle, the simulation package as described in Chapter 3 is used to simulate the CITO with the double wishbone suspension, front wheel steering and rear wheel drive. This simulation is done to identify overall ride comfort data, such as a ride comfort index as well as dynamical behavior and specific roll over properties. A comparison with the reference vehicle, described in Chapter 4, is additionally made to identify how the vehicle’s performance relates to a current mid-sized vehicle.

5.3.1 Ride

To identify ride behavior of the vehicle, two simulations are performed. Firstly, an adjusted single suspension model is used for quarter car simulation. This simulation provides information about bounce and wheel hop frequencies. Alternatively, an entire vehicle equipped with four double wishbone modules is driven over an uneven road which is assumed to accommodate all relevant road frequencies in daily driving. This simulation provides ride performance figures, such as suspension travel and ride comfort index. The frequency analysis for vehicle body acceleration is shown in Figure 5.9.

![Figure 5.9 Ride frequency analysis](image)

Figure 5.9 shows a ride frequency of somewhat more than 1 [Hz], which is typical for current road vehicles. It has to be noted that the CITO is to be equipped with stiff roll stabilizers.
and high damping, as shown in Table 5.2, leading to a comfort level which could prove to be unacceptable. The values shown in Table 5.2 are implemented in the model via a Look-up-Table.

Table 5.2 Suspension spring and damper characteristics

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring stiffness front / rear</td>
<td>30000 [N/m]</td>
</tr>
<tr>
<td>Bump stop stiffness front / rear</td>
<td>1000000 [N/m]</td>
</tr>
<tr>
<td>Spring effective stroke</td>
<td>0.07 [m]</td>
</tr>
<tr>
<td>Damping constant low speed in</td>
<td>29430 [Ns/m]</td>
</tr>
<tr>
<td>Damping constant high speed in</td>
<td>2235 [Ns/m]</td>
</tr>
<tr>
<td>Damping constant low speed rebound</td>
<td>10326 [Ns/m]</td>
</tr>
<tr>
<td>Damping constant high speed rebound</td>
<td>5464 [Ns/m]</td>
</tr>
</tbody>
</table>

In order to assess overall ride comfort, the entire vehicle is simulated, driving over an uneven road surface. This simulation provides insight in overall ride comfort, including stabilizer stiffness and roll and pitch movement of the vehicle body. As reference position, the centre of gravity of the vehicle body is chosen, as yet no exact location of driver and passenger is known. This location is an approximation of the body of the driver.

The simulation is performed at 70 [km/h] and afterwards, comfort filtering according to ISO 2631-1:1997 is done to calculate the ride comfort indices. This weighting filter differentiates between frequencies which are experienced as more uncomfortable and those that are hardly felt. The weighting function, which calculates standardized vertical acceleration \( \ddot{z}_{s,iso} \) based on measured vertical acceleration \( \ddot{z}_s \), looks as follows.

\[
\ddot{z}_{s,iso}(f) = H_{iso,x}(f)\ddot{z}_s(f)
\]  

Additionally a comfort number is calculated based on the standardized vertical acceleration. After a simulation of 50 [s], results shown in Table 5.3 are obtained.

Table 5.3 Ride comfort indices @70 [km/h] according to ISO 2631-1:1997(E)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>RMS value</th>
<th>Peak value (3 * RMS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical acceleration</td>
<td>0.84 [m/s²]</td>
<td>2.52 [m/s²]</td>
</tr>
<tr>
<td>Ride comfort index</td>
<td>0.76 [m/s²]</td>
<td>2.28 [m/s²]</td>
</tr>
<tr>
<td>Suspension travel</td>
<td>44 [mm]</td>
<td>132 [mm]</td>
</tr>
</tbody>
</table>

To accommodate driving over the simulated road surface, the vehicle requires a suspension travel of 88 [mm] in total. During the suspension design, an effective suspension travel of 200 [mm] is aimed at. For the simulated road surface, the ride comfort index is in the region “fairly uncomfortable”. In all, the suspension design and spring and damper settings are sufficient from a ride comfort point of view, as focus is on roll over stability and sporty driving performance.

5.3.2 Vehicle handling
The vehicle dynamics are assessed using several maneuvers, as described in chapter 3. Here the most important results are presented, providing detailed insight in the behavior of the total system of suspension and CITO vehicle geometry and parameters.

The steady state circular test provides information about the intrinsic behavior of the vehicle. Figure 5.10 shows the required steering wheel angle to achieve a certain lateral acceleration. Initially, because of the chosen steer ratio in this simulation, the CITO requires...
larger steering wheel angles compared to the reference vehicle. This can easily be changed by changing increasing the steer ratio, leading to less required steering wheel angles at low velocity. Lateral acceleration in this maneuver increases because of increased forward velocity as the corner radius remains constant. CITO has a 50/50 [%] weight distribution between front and rear, compared to the 63/37 [%] weight distribution for the reference vehicle. Additionally, CITO is rear wheel drive, whereas the reference vehicle is front wheel drive. These two basic lay-out properties cause CITO to require less steering wheel angle at higher velocity and higher lateral acceleration. This property makes the car more neutrally steered in the handling region up to 5 [m/s²] and substantially less understeered in the high lateral acceleration region.

Figure 5.10 Required steering wheel angle to achieve lateral acceleration

In addition to these two lay-out based aspects, also the double wishbone suspension and relatively stiff anti-roll bars add to the reduction of understeer. One of the main goals of suspension design is to keep vehicle roll angles small, Figure 5.11 shows roll angles as result of lateral acceleration for both CITO and the reference vehicle. Although already the reference vehicle allows only small roll angles of about 4 [deg/g], CITO even further reduces vehicle body roll to under 3 [deg/g]. The distribution of roll stiffness between front and rear axle reduces weight transfer on the front axle slightly more than the reference vehicle, which makes the vehicle less understeered and obtains a sportier driving feel.
An important result of the steady state circular test is the fact that the maximum attainable lateral acceleration for the narrow CITO is even slightly higher than for the reference vehicle. Although the general idea is that narrow vehicles cannot reach equally high cornering speeds as wider vehicles, this simulation shows that for CITO with a track width of 1.20 [m] and 0.49 [m] height of centre of gravity, this is not the case.

The highly dynamical moose test assesses vehicle stability and dynamic response. This maneuver executed at 18 [m/s] shows no major differences between CITO and the reference vehicle. Results of the moose test are presented in Figure 5.12.
The upper plot in Figure 5.12 shows the exit speed for CITO is slightly higher compared to the reference vehicle. Also clearly visible is the lower roll angle, as expected. The maneuver also shows the vehicle has little to no time to settle between the two lane changes, from \( t=2.5 \text{ [s]} \) to \( t=3.5 \text{ [s]} \). The lateral acceleration and roll angle of the reference vehicle changes during this period, whereas CITO shows somewhat constant values. This behavior is analyzed by assessing vehicle yaw velocity. Figure 5.13 shows yaw velocity is initially generated more quickly for CITO. At about \( t=2.2 \text{ [s]} \), yaw velocity stabilizes for a short period in time at 0. This is caused by the driver model, which not only controls the vehicle to reach the two lanes, but also controls the path in between. At this point, exactly between the two lanes, the steering wheel angle is reduced, because the control error reduces to about 0 for CITO. Then the vehicle reaches the second lane and yaw velocity and lateral acceleration have to be generated again, which costs some more time compared to the reference vehicle which reacts slightly slower. Therefore it has a smooth transition from positive to negative yaw velocity and develops slightly higher yaw velocity and lateral acceleration around \( t=2.7 \text{ [s]} \). Although results at first sight do not imply this, CITO has slightly higher dynamical response in this maneuver compared to the reference vehicle and also carries more velocity through the maneuver resulting in higher exit speed.

![Figure 5.13 Yaw velocity during moose test](image)

5.3.3 Roll over

Roll over is assessed in several maneuvers, as every maneuver possibly can cause the vehicle to lose contact with the road. Especially the change in height of the centre of gravity is investigated, as well as vertical tyre reaction forces for each wheel. A comparison is made between the behavior of CITO and the reference vehicle. Firstly, the results of the maneuvers executed in section 5.3.2 are presented, after which the fishhook maneuver dedicated to roll over is performed.

Figure 5.14 shows the change in height of the centre of gravity during the steady state cornering maneuver. Initially, the height of the centre of gravity decreases, due to acceleration of the vehicle. Compared to the reference vehicle it is clear that during this steady state maneuver, the change in height is smaller.
Figure 5.14 Change of height of centre of gravity during steady state cornering

The moose test is also from the roll over perspective more challenging. In this maneuver not only the effective roll stiffness of each axle combined with kinematic aspects determines the change in height, but additionally roll damping is more important. As is shown in Figure 5.15, damping in the low speed area is suitably tuned to control body movement during dynamic maneuvers. The change in height of the centre of gravity is lower compared to the reference vehicle and more adequately damped.

Figure 5.15 Change in height of centre of gravity during moose test

Finally, the fishhook maneuver is performed for both the reference vehicle and CITO. In this maneuver not only the height of centre of gravity is monitored, but also the vertical load of each of the tyres, which combined provides all necessary information about intrinsic roll over behavior of the vehicle. As expected based on previous simulations, changes in height of centre of gravity induced by lateral acceleration and body roll are reduced for CITO compared to the reference vehicle. This is additionally founded by assessing tyre vertical loads.
Figure 5.16 Tyre vertical loads during fishhook maneuver

Figure 5.16 shows results of the fishhook maneuver for the reference vehicle and CITO, clearly showing less reduction of vertical load on the inner wheels during this maneuver, especially on the rear axle. The reference vehicle lifts the rear inner wheel off the ground, which is typical for a front wheel drive vehicle with dependent rear axle and sporty suspension tuning. For CITO this reduction of vertical load for both front and rear axle is less, assuring high roll over resistance and well-balanced driving behavior.

5.4 Summary

In this chapter, initial double wishbone suspension geometry is presented and both a kinematic and a dynamic evaluation are performed. The kinematic analysis shows that key characteristics are taken into account, leading to relatively large camber gain on steering and suspension travel. Bump steer however is almost reduced to zero. The dynamic analysis shows that the vehicle is slightly more resistant to roll over compared to the benchmark vehicle, mainly because of its suspension arm orientation and high roll stiffness. Vehicle handling results all in all suffer on both the effects of the very stiff suspension and the rear wheel drive configuration. Obtained results however are almost similar to the reference vehicle, but leave room for improvement. In general, especially roll over prevention is better for CITO with the proposed double wishbone suspension design, compared to the reference vehicle with larger track width and MacPherson suspension on the front axle and a torsion beam rear axle.
6 Vehicle dynamics improvement

In the past century not only kinematic suspension design has strongly developed, also new technologies have been introduced enabling suspension to dynamically adapt to any situation. Though a large number of these innovations is based on mechanical improvements, in the past twenty years the number of active, electronically controlled systems has expanded enormously. The well known Electronic Stability Program is one of the most striking and important of those, greatly increasing active safety. Not only are vehicles made safer by these innovations, also ride comfort and dynamic behavior are improved. Examples of technologies increasing ride comfort are air springs and dampers on luxury vehicles and improved active oil dampers where damper characteristics can be influenced. To improve vehicle dynamics, four wheel drive with active individual torque distribution is introduced, but also active damping can improve road holding and vehicle dynamical response. This chapter first presents a literature review of current and state of the art technologies that can be applied in SAM CDM in section 6.1, already making an evaluation of each of these technologies indicating the most interesting technologies. Based on this evaluation, a number of technologies is simulated using the vehicle simulation package as described in chapter 3. In section 6.2 the development and simulation of a four wheel steering algorithm for the CITO is presented, in section 6.3 all wheel drive is simulated and discussed.

6.1 Literature review

The width of the vehicle poses a challenge in several fields, mainly ride comfort, vehicle handling and the specific case of roll over. Therefore, each of these fields is assessed, comparing and combining the potential of several works in literature.

6.1.1 Vehicle handling

Within vehicle dynamics improving technologies, several different approaches can be taken. Essential for each solution however is the tyre and its behavior. Each technology is either aimed at improving tyre orientation w.r.t. the road surface or at control of tyre vertical load, both principally under specific circumstances. In addition, systems such as the Electronic Stability Program (ESP), an electronic Limited Slip Differential (eLSD) and driving torque vectoring can be classified as vehicle dynamics improving technologies, influencing vehicle yaw and lateral dynamics by applying or reducing longitudinal forces.

The most interesting method to improve vehicle handling especially because of the modular structure of the suspension geometry, is four wheel steering. Generally, two systems can be distinguished. In literature, most four wheel steering controllers are based on a closed loop system, where for example vehicle yaw velocity is fed back. Another approach is open loop feedforward four wheel steering, where rear wheel steer angles are derived from front wheel steer angles. Firstly, closed loop systems are assessed. Ackermann [9] presents a controller structure based on a linear single track model with nonlinear tyre behavior. This single track model is used to calculate reference values for lateral acceleration and vehicle yaw velocity, which subsequently are used to calculate steer angles. A front wheel steer angle is calculated...
based on a position error, which is converted into a required lateral acceleration using a
linearized vehicle model. A proportional controller controls the linearized system dynamics.
Rear wheel steer angles are calculated based on the vehicle yaw velocity target, required
lateral acceleration of the rear axle and front axle steering wheel angles. Controller tuning
however poses a problem, which leads to questionable robustness of the closed loop system
under all operating conditions. One of those is too high damping on high velocity, as well as
robustness of the system to disturbances such as road irregularities or crosswind. No
simulation or experimental results are presented, so performance of the system is left to be
determined. Akar [10] describes a sliding mode controller based on yaw velocity and side slip
angle tracking. The controller is based on a modified single track model with one central
wheel. Simulation results appear feasible and robust for parameter errors up to 10% and
crosswind. However the calculated front and rear wheel steer angles are highly unrealistic
and yaw velocity build up is not damped effectively. This four wheel steering algorithm
appears to be purely mathematical and not feasible to be used on a real vehicle.

Contrary to closed loop four wheel steering algorithms, a feedforward system has the
advantage to be very robust to operating condition changes. Besselink et al. [11] make a
comparison between a closed loop four wheel steering controller based on a yaw velocity
reference and feedforward control. The open loop four wheel steering system uses a filter to
determine rear wheel steer angles based on front wheel steer angles. Three controller tuning
parameters depending on vehicle forward velocity are used to tune vehicle yaw velocity
response. The feedforward controller proves to provide equal performance, though requiring
less control effort and providing a higher level of robustness. Another important advantage is
the fact that controller parameters are much less velocity dependent compared to a closed
loop controller.

A very interesting controller is proposed by Ono et al. [12]. This controller combines the
opportunities of four wheel steering with all wheel drive. Based on tyre grip estimation, or in
simulation directly based on the current tyre grip level, an overall resulting force is divided
over all four wheels and subsequently divided into longitudinal and lateral forces. This
controller requires a three level cascaded controller structure, with a top level handling
driver input, calculating the required resulting force and moment. The second level nonlinear
optimal controller calculates the longitudinal and lateral forces required for each wheel,
subsequently followed by an individual wheel controller and actuator. The largest
contribution to an increase in resulting force is provided by an individual wheel steer angle
distribution between front and rear, followed by individual torque distribution. The integral
controller adds 2.5 % to the end result.

Chieli et al. [13] present an all wheel drive torque vectoring algorithm for a high power
vehicle, which basically is laid out with rear wheel drive. The all wheel drive algorithm
proves to be effective in achieving better lap times at a simulation track and currently is
being tested in a real vehicle. Osbourne et al. [14] assess performance of individual front-rear
and left-right torque distribution. A combined controller is based on two PI-controllers, one
controlling vehicle yaw velocity to determine front-rear distribution and one controlling
lateral acceleration to define torque distribution from left to right. A comparison is made
between front wheel drive, rear wheel drive and several forms of all wheel drive. The front-
rear torque distribution proves to be the most effective in improving vehicle handling,
although combined control even further increases performance.

Lu et al. [15] investigate the effect of camber changes on vehicle dynamics and roll over using
a simulation package. The main conclusion is that camber changes can have a large effect on
the occurrence of roll over and that simulation packages often overestimate the roll over
hazard.
Zuurbier et al. [16] provide a comprehensive simulation overview of different technologies in achieving pro- and contra-cornering moments. A combination of active steering and braking is most effective in achieving contra-cornering, stabilizing moment. Rear wheel steering and braking proves to be most effective in situations where pro-cornering torque is required, however it is questionable how efficient braking is in order to achieve vehicle turn-in. Active suspension provides an increase in performance in both cases, but less than steering and braking effects. Schiebahn et al. [17] come to somewhat equal results, however state that active front steering is particularly important to reduce understeer and active rear steering has the potential to reduce oversteer. A braking intervention by an Electronic Stability Program (ESP) is useful especially in reducing oversteer in the maximum lateral acceleration region. Other technologies, such as torque distribution via limited slip differentials or vehicle body control via active roll stabilizers, only prove to add a small amount of desired behavior compared to steering and ESP.

6.1.2 Roll over
Choi [18] presents an assessment of effectiveness of brake interventions in order to reduce vehicle roll over. As is generally accepted, reducing vehicle velocity reduces lateral acceleration and therefore the tendency to roll over. This system however is very dependent on a reference vehicle model, for example a single track model, which is used as an observer. Parameter differences can cause ESP interventions to occur too often or cause system failure. Robustness for a vehicle speed of 100 [kmh] is proven using simulations.

6.2 Four wheel steering
Four wheel steering has been a hot topic in vehicle dynamics in the 1980’s, leading to several production vehicles being equipped with this type of technology in the early 1990’s. These systems, introduced among others by Honda, Mitsubishi and Nissan, were of the mechanical kind or included a second steer rack at the rear and improved vehicle yaw dynamics for sporty vehicles such as the Honda Prelude. As cost reduction gained more and more importance and economies were made, these complex rear axle constructions became too costly. Additionally, the Electronic Stability Program (ESP) was introduced by Bosch, pursuing the same goal, though offering even more safety improving potential. The 2008 Renault Laguna Coupe is the first European volume model to arrive at the market equipped with all wheel steering, followed by the 2009 BMW 7-series. These systems no longer are mechanically operated, but controlled electronically and therefore providing extended possibilities. Meantime, extensive research is done on all wheel steering.

6.2.1 Chosen algorithm
SAM CDM is to be used in manual mode in both city traffic and secondary roads, connecting the high way to the destination. In between, the vehicle is driven in automatic mode in its own specific infrastructure. During automatic mode, no vehicle dynamics hazards occur, as the one-way track is built of closed U-shaped concrete forms and the vehicle is connected to other vehicles. Therefore, four wheel steering for SAM CDM has two main goals;

- Reduce turning radius in city driving and especially parking
- Increase high velocity stability of the vehicle

Several other goals can be pursued when applying four wheel steering, such as controlling the vehicle position in its narrow concrete infrastructure. However, these goals are not in the scope of this research.

To achieve the goals with conflicting requirements w.r.t. the wheel steering angles of the rear wheels, no straightforward four wheel steering algorithm can be applied. For the first goal,
rear wheels have to be steered in opposite direction to the front wheels. This can be done by applying a linear reciprocal relation between front and rear steering angles. If this four wheel steering scheme were to be applied at high speed, vehicle stability is strongly negatively affected. Compare the behavior of a counter steering all wheel steered vehicle at high velocity to that of a vehicle driving in reverse at high velocity. As can be concluded from this observation, rear wheels should steer by some relation in the same direction as the front wheels to stabilize vehicle behavior at high velocity.

Reduction of the turning radius is achieved by applying maximally 50% of the steering angle of the front wheels to the rear wheels in opposite direction. In contrast with conventional rear axle constructions, CITO’s modular suspension design allows for equal maximal wheel steer angles front and rear. This extends possibilities compared to existing systems, accommodating the proposed steer angles. Figure 6.1 left shows the behavior for a left hand corner.

![Figure 6.1 Left: Low velocity turning radius reduction; right: high velocity stabilization](image)

A rear wheel steering algorithm based on velocity dependent filtering is chosen as stabilizing four wheel steering system at high velocity. This system provides approximately equal stabilizing performance whilst requiring less computational and control effort and providing optimal robustness. This algorithm functions as a filter between front and rear wheel steer angles. A schematic view of rear wheel steering in the same direction as front wheels is shown in Figure 6.1 right. The transfer function relating front and rear wheel steering angles is adopted from [11] and shown below.

\[
H_{\delta_1,\delta_2}(s) = \frac{K(t_1 - t_2)s}{(t_1s + 1)(t_2s + 1)} \tag{6.1}
\]

As stabilizing performance is required only at high velocity, though turning radius reduction is required at low velocity, the choice is made to introduce a velocity dependent filter, accommodating the transfer between the two algorithms. For the low velocity contribution, the following, arbitrary, relation between forward velocity \( V \) and filter gain is proposed.

\[
k_{low} = \frac{1}{2e^{\alpha}} \tag{6.2}
\]

\[
\alpha = \left(\frac{V}{10}\right)^4 \tag{6.3}
\]
Although the rear wheel steering filter operates from about 70 [km/h], the choice is made to introduce a filter gain, which is multiplied with the steering angle from the rear wheel steering filter. For the high velocity contribution, the following, arbitrary, relation between vehicle forward velocity $V$ and control gain is proposed.

$$k_{high} = \left(\frac{V}{15}\right)^6$$  \hspace{1cm} (6.4)

The range of this high velocity filter gain is then saturated to [0,1].

These two relations are shown graphically in Figure 6.2.

As can be understood from Figure 6.2, low velocity turning radius reduction is operative from approximately 0 [m/s] to 10 [m/s], clearly falling from 5 [m/s]. High velocity stabilizing effort is subsequently applied from about 16 [m/s] onward. Around 15 [m/s] transfer between low velocity and high velocity control takes place, somewhat making the vehicle behave as conventionally front wheel steered.

A schematic view of the four wheel steering algorithm is shown in Figure 6.3.
6.2.2 Four wheel steering simulation
Firstly, the ability to reduce turning radius is investigated. To this end, a low velocity test with maximum steering wheel angle is performed. This test executed for both the front wheel steered vehicle and the four wheel steered vehicle. Results are presented in Figure 6.4.

![Figure 6.4 Turning circle for front wheel steered and four wheel steered vehicle](image)

The above figure shows an initial turning circle of 8 [m] for the front wheel steered vehicle, whilst the four wheel steering algorithm reduces this value to approximately 5.5 [m]. Note that this value represents the turning circle of the centre of gravity, the curb-to-curb circle is 6.9 [m], which is very tight for a production vehicle. The turning radius can be further reduced, however to maintain the required level of safety and a somewhat conventional driving feel, this is not advised.

The stability increasing high velocity algorithm is assessed in the same way as in section 5.3 a comparison is made between the CITO and the reference vehicle. A number of maneuvers determining dynamical behavior are performed. Firstly, the controller parameters corresponding to (6.1) are determined based on a step steer maneuver, where yaw stabilization is pursued. The maneuver is executed at 100 [km/h] with a target steering wheel angle of 0.8 [rad]. The required values are presented in Table 6.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K$</td>
<td>0.8</td>
</tr>
<tr>
<td>$\tau_1$</td>
<td>0.7</td>
</tr>
<tr>
<td>$\tau_2$</td>
<td>0.3</td>
</tr>
</tbody>
</table>
The yaw response for this set of parameters is presented in Figure 6.5, clearly showing the stabilizing performance. Also the front and rear wheel equivalent steering wheel angles are shown in the right figure, achieved by this four wheel steering algorithm. The set of parameters found in this maneuver are used in further investigation.

![Figure 6.5 Left: yaw velocity stabilization by four wheel steering; right: steering wheel angles](image1)

**Figure 6.5 Left: yaw velocity stabilization by four wheel steering; right: steering wheel angles**

Yaw stabilization alone is not sufficient; other properties should not be affected. Lateral acceleration shows more smooth build up, see Figure 6.6, though reaching an equal maximum value. This means cornering behavior becomes a more smooth experience using four wheel steering, as well as safer because of increased yaw damping and smoothened tyre force fluctuation. Initial understeer is not created by adding steering to the rear wheels, yaw response even seems slightly better, but excessive oversteer is effectively damped.

![Figure 6.6 Lateral acceleration build up](image2)

**Figure 6.6 Lateral acceleration build up**

To verify if steady state cornering has not changed compared to the conventional front wheel steered vehicle, the steady state circular test is performed. This test, as expected, does not show differences compared to the front wheel steered vehicle. As the rear wheels mainly receive steering angles depending on the steering wheel angle velocity, rear wheel steering angles can be neglected. This means that during highway cornering, the vehicle behaves as a conventional front wheel steered vehicle.
Dynamic tests are more suited to investigate true stabilizing behavior. To this end, in addition to the step steer test, a fishhook test and moose test are performed. The fishhook test not only indicates if stability is improved, but also provides insight in whether or not roll over characteristics are affected by four wheel steering. Figure 6.7 shows the vehicle roll velocity during the fishhook maneuver at 70 [km/h]. As results of the step steer maneuver indicated, lateral acceleration is built up more smoothly for the four wheel steered vehicle. In this case, that leads to strongly reduced vehicle roll rates, though the effective roll angle is approximately equal.

![Figure 6.7 Vehicle roll rate during fishhook test](image)

Figure 6.8 shows the effect of the reduced roll rate on tyre vertical loads. Peaks that occurred for the front wheel steered vehicle are entirely damped, which effectively increases vehicle stability and controllability. Additionally, these smooth tyre load fluctuations enhance the effectiveness of a possible Electronic Stability Program because tyre forces can be estimated with greater accuracy. The change in height of the centre of gravity during this maneuver is almost unaffected, though has slightly decreased for the four wheel steered vehicle. Together with the damped tyre load fluctuations, this leads to slightly increased roll over resistance for the four wheel steered vehicle.
The moose test, executed at 68 [km/h], confirms the conclusions from previous simulation tests. Vehicle stability is increased to a large extent, though vehicle orientation angles are not affected dramatically. Figure 6.9 shows that the orientation of the vehicle during the moose test has not changed significantly. It has to be noted that this change in orientation is partly due to the fact that the tracking behavior of the vehicle during the moose test is increased when four wheel steering is applied. Figure 6.9 also clearly shows in the right figure that vehicle yaw velocity is better controlled and peaks are damped.

In Figure 6.10 both vehicle trajectory and equivalent steering wheel angles for the executed moose test are presented. For this entrance velocity, no significant difference between the trajectory of the front wheel steered and four wheel steered vehicle is noticed. This is due to
the fact that this particular velocity, the front wheel steered vehicle is exactly on the edge between stability and instability. If entrance velocity is increased only by a small amount, the front wheel steered vehicle demonstrates oversteer, whilst the four wheel steered vehicle still is controlled. In Figure 6.10 on the right, the steering wheel angles for front and rear wheels are shown. As expected the rear wheels during this maneuver behave as depicted in Figure 6.1 and rear wheel steer angles maximally reach about 30% of the front wheel steer angles. The rear wheel steer angles achieved by this type of rear wheel steering appear to be in line with vehicles currently available on the market, accommodating rear wheel steer angles of up to several degrees. Note that Figure 6.10 shows steering wheel angles, which are to be multiplied with the steer ratio of the rear axle, for example \(1/15\). Real life tests with a vehicle equipped with a four wheel steering algorithm as presented in this research will have to demonstrate whether or not such behavior is desirable.

6.3 All wheel drive

In addition to steering in general, vehicle stability and handling is also affected by the driveline topology. As tyres operate within a so called friction ellipse, the maximum available force is divided between longitudinal and lateral components. Generally speaking, this observation leads to the conclusion that driven front wheels push a vehicle towards understeer, whereas driven rear wheels increase the tendency to oversteer. For conventional vehicles equipped with only one power source and one driven axle, such as most road vehicles on the market, this means when torque is applied, the vehicle either tends to understeer or to oversteer. Four driven wheels, depending on the torque distribution between front and rear, can influence this behavior. However in conventional vehicles equipped with only one power source, an entirely individual torque distribution between the front and rear axle is hard to achieve. The lay-out of the CITO with the possibility to equip the vehicle with one, two or four electric motors, does provide this opportunity without the need for complex mechanical constructions. In addition to distribution of drive torque between front and rear, also a distribution from left to right is advantageous. By doing so, at any moment in time a specific tyre can be driven, braked or left freely rolling, increasing the level of control over tyre utilization. The all wheel driver algorithm is based on a reference value for vehicle yaw velocity, pursuing a neutrally steered vehicle.
This reference value is calculated every time step and is based on the following equation.

\[
\begin{align*}
    r_{\text{ref}} &= \frac{V}{R} \\
    R &= \frac{l}{\delta}
\end{align*}
\]

(6.5)  
(6.6)

Where \( V \) is vehicle forward velocity, \( R \) is the estimated turning radius, \( l \) is vehicle wheelbase and \( \delta \) is the front wheel steer angle. This reference value for vehicle yaw velocity is subsequently used to calculate a yaw velocity error, which serves as input for the yaw based four wheel drive algorithm. This is done by subtraction the reference value for yaw velocity from the current yaw velocity in absolute value, because this eliminates the difference between left hand and right hand corners.

\[
e_r = |r| - |r_{\text{ref}}| 
\]

(6.7)

This error is then used to calculate a control parameter ruling the fore-aft torque distribution \( D_{FR} \). Starting point for propulsion of the vehicle is rear wheel drive and depending on the error in yaw velocity, the front wheels are driven. The torques lead to the front and rear wheels is given in (6.9) and (6.10) respectively, where \( T_{in} \) is the input torque from the driver velocity controller.

\[
\begin{align*}
    D_{FR} &= \frac{2\tan^{-1}(5e_r)}{\pi} \\
    T_F &= \frac{D_{FR}}{2} T_{in} \\
    T_R &= \frac{(1 - D_{FR})}{2} T_{in}
\end{align*}
\]

(6.8)  
(6.9)  
(6.10)

\( D_{FR} \) saturates to 1 for high yaw velocity errors, transferring all torque to the front wheels. When no error in yaw velocity occurs, the vehicle is entirely rear wheel driven which is advantageous for driving comfort and handling. Also in the case where current yaw velocity is smaller than reference yaw velocity, the vehicle is entirely rear wheel driven. This condition is not covered by the error definition of (6.7) and therefore leaves the torque distribution unaffected to rear wheel drive. Figure 6.11 shows the torque distribution factors (6.9) and (6.10) as a function of the yaw velocity error of (6.7).
In addition to the fore-aft torque distribution, a wheel spin control system is implemented. This system controls the torque applied to the two wheels on each axle. While the above-mentioned yaw velocity control regulates the fore-aft torque distribution, this controls the left-right distribution. The end result is an individual torque distribution between all four wheels.

The left-right torque distribution is based on a reference difference in rotational velocity of each axle. The reference value is calculated by calculating the cornering radius of each wheel, using the vehicle center radius of (6.6), denoted below as $R_2$. This radius is subsequently used to calculate the respective wheel rotational velocity of each wheel on the axle.

\[
V_L = \frac{R_1}{R_2} V; R_1 = \frac{l}{\delta} \frac{tw}{2} \tag{6.11}
\]

\[
V_R = \frac{R_3}{R_2} V; R_3 = \frac{l}{\delta} + \frac{tw}{2} \tag{6.12}
\]

\[
\Delta \omega_{\text{ref}} = \frac{V_L}{r_L} - \frac{V_R}{r_L} \tag{6.13}
\]

\[
e_{\omega,i} = \frac{(\omega_{L,i} - \omega_{R,i}) - \Delta \omega_{\text{ref}}}{\omega_{L,i}}; i = F, R \tag{6.14}
\]

This scheme calculates a velocity difference factor. Depending on the sign of $e_\omega$ and some threshold value $e_{\text{thres}}$, torque is reduced at respectively the left or right wheel by a factor 10. See the operating logic below, where $e_{\text{thres}} = 0.05$, so roughly 5% wheel spin is allowed.

```matlab
if e_w_F < -0.05
    T_Fl = T_FL / 10;
elseif e_w_F > 0.05
    T_FR = T_FR / 10;
end

if e_w_R < -0.05
    T_RL = T_RL / 10;
elseif e_w_R > 0.05
    T_RR = T_RR / 10;
end
```
This overrules the torque calculated in (6.9) and (6.10) and effectively prevents the wheel from spinning freely, as torque transferred to this wheel is immediately reduced. Note that subsequently the torque output of the drive algorithm is filtered, see section 3.2.2, smoothing out potential torque peaks. However no braking torque is applied, because the braking system is not modeled in enough detail.

6.3.1 All wheel drive simulation
To identify intrinsic behavior of the all wheel drive algorithm, a semi steady state circular test is performed. As in previous simulations, the velocity is increased from 20 \([\text{km/h}]\) until maximum lateral acceleration is reached whilst attempting to keep a constant cornering radius of 100 \([\text{m}]\). During this maneuver, the all wheel drive algorithm is expected to transfer driving torque from entirely rear wheel drive to front wheel drive, because the vehicle is slightly oversteered due to rear wheel drive, see section 5.3.2. Figure 6.12 shows the course of both front and rear driving torques during this maneuver. Initially the vehicle is entirely rear wheel driven, while at high lateral acceleration a transfer is made over four wheel drive to front wheel drive.

![Figure 6.12 Driving torque transfer front-rear during semi steady state circular test](image)

Compared to the rear wheel drive vehicle, this reduction of torque at the rear wheels effectively reduces the oversteer tendency by reducing the yaw velocity at high lateral acceleration. Figure 6.13 shows that the reduction of torque at the rear axle effectively reduces the yaw velocity at maximum lateral acceleration and therefore reduces oversteer.

![Figure 6.13 Vehicle yaw velocity reduction by variable all wheel drive](image)
To further investigate the stabilizing effect of all wheel drive, a modified moose test is performed. The modification to the original ISO 3888-1 maneuver is in the fact that the vehicle is not in free rolling condition, but a constant velocity of 19 [m/s] is pursued during the maneuver. This modification leads to required driving torques because the generated side forces of the steered wheels have a component in the direction opposite to the vehicle velocity. The reduction in vehicle velocity during the modified moose test maneuver is shown in Figure 6.14, showing a noticeable decrease in velocity for the driven vehicle in all configurations. The all wheel drive system achieves the highest velocity through the maneuver, followed by the front wheel drive configuration. The rear wheel drive vehicle shows excessive oversteer caused by overloaded rear tyres. Partially, the success of the all wheel drive system in achieving a higher exit velocity lies in the fact that wheel spin control is incorporated. In the front wheel drive configuration, wheel rotating velocity differences are not taken into account, leading to spinning wheels on the inside of the corner. Compared to the rear wheel drive configuration the effect is clear, reduction of torque to the rear wheels effectively damps the oversteer reaction.

![Figure 6.14 Velocity during moose test for different drive configurations](image)

The vehicle orientation angle achieved by this form of all wheel drive during the modified moose test is in between the front wheel drive and rear wheel drive configuration, though closer to the front wheel drive configuration. This is as expected, because the all wheel drive system transfers torque to the front axle when yaw velocity exceeds the reference value.

Figure 6.15 shows the absolute vehicle orientation angle during the moose test for different drive configurations in the top figure. From this figure it can be extracted that the orientation between the two lane changes is controlled more effectively by the all wheel drive system compared to the rear wheel drive configuration. The lower figure in Figure 6.15 shows the drive torque transferred to each of the front wheels. This shows that already during the first lane change, between $t=1.5 \text{ [s]}$ and $t=2.5 \text{ [s]}$ torque is transferred to the front wheels. Combined with the large torque transfer between $t=3 \text{ [s]}$ and $t=4 \text{ [s]}$ this reduces the tendency to oversteer and therefore stabilizes the vehicle.
Figure 6.15 Vehicle orientation angle and front-rear torque distribution during modified moose test

Also the wheel rotating velocity control is tested during this maneuver, as already indirectly shown in Figure 6.15.

Figure 6.16 presents in the upper figure the rotational velocity of the front right wheel during the moose test for both front wheel drive and all wheel drive. The front right wheel is the front inner wheel during the second lane change and therefore vertical load is reduced, making this wheel prone to spin. The lower figure shows the torque applied to this wheel, which evidently is reduced when the wheel shows the tendency to spin.

Figure 6.16 Rotational velocity and torque applied to front right wheel
6.4 Summary

Based on a literature review assessing technology able to improve vehicle dynamic performance, the choice is made to thoroughly investigate the effect of four wheel steering and all wheel drive. Based on literature, a novel velocity dependent four wheel steering algorithm is developed and simulated on the CITO vehicle model. This four wheel steering algorithm on the one hand reduces the turning radius for high maneuverability and on the other hand stabilizes the vehicle at high velocity. Furthermore, also based on the results in literature, a novel all wheel drive algorithm is developed. Dependent on a yaw velocity error, the drive torque is distributed between the front and rear axle, whereas in neutral conditions the vehicle is rear wheel driven. This system is proven effective in stabilizing the vehicle at high lateral acceleration, reducing the tendency to oversteer. Additionally the system increases the maximum attainable average and exit velocity during the challenging modified moose test. As both systems are complementary, a combination between four wheel steering and all wheel drive is expected to provide optimal results. An integral vehicle dynamics controller however possibly is needed to get the most out of the potential of both systems.

Of the two systems simulated however, the four wheel steering algorithm is most effective in enhancing vehicle handling. This is mainly due to the reason that oversteer is very well controlled by application of small steer angles to the rear wheels. The all wheel drive system on the other hand has more potential in decreasing understeer in specific situations, because driving torque can be applied to any wheel or axle individually, though the fore-aft torque distribution accounts for the largest improvement. Wheel-spin detection and reduction is essential to prevent wheels with reduced vertical load from spinning, which would compromise all wheel drive performance. In a real all wheel drive configuration, the wheel-spin prevention can also be performed by the brake system, equal to a common ASR-system.

Simulation is performed with both systems operating together, but performance of the combination of systems is not optimal. The results of this simulation are not presented in this report.
7 Suspension design optimization

The initial suspension design presented in Chapter 5 has been developed before entire vehicle simulations with four wheel steering and all wheel drive were executed. Also constructional considerations and restrictions are not taken into account. An optimized suspension design is presented in section 7.1, assessing kinematics and presenting a schematic overview of the suspension mounted on both front and rear axle. In addition, a qualitative recommendation is done about the choice for suspension bushings in section 7.2, making a qualitative elastokinematic analysis of the suspension design.

7.1 Suspension kinematic optimization

Although the design presented in Chapter 5 underwent several iteration steps in order to achieve kinematic targets, the design is further optimized to better suit changing requirements. The optimization mainly leads to minor relocations of suspension nodes, the main geometric concept of the suspension design, a double wishbone lay-out, consequently remains similar. Also the location of a combined spring damper, which in the preceding of this research is assumed to be two individual parts, invokes changes in suspension geometry. Additional information about constructional issues, such as conflicts following from 3D CAD analysis, also requires small changes in geometry. Obviously, dynamic performance is not to be affected or rather to be improved. A selection of results is presented in this section to assess changes made in suspension geometry.

Because vehicle roll angles are kept small, within the range of 3 [deg/g], suspension travel during cornering is limited. This observation makes camber gain on suspension travel less important. A reduced camber gain on suspension travel makes the vehicle behave more predictable on bad road surfaces, as less side force is generated by these disturbances. Camber gain reduction is achieved by increasing the length of the upper wishbone. Figure 7.1 shows the camber behavior of the changed geometry compared to the design presented in Chapter 5.
To ensure equal or better bump steer properties, the location of the track rod is revised. Compared to the previous design, it is moved slightly upward and vehicle inward. As Figure 7.2 shows, bump steer is even further reduced, to a maximum of 0.03 [deg] over the entire suspension travel region from -0.07 [m] to 0.10 [m]. The improved bump steer properties, combined with the reduced camber gain, ensures better road holding and increased steering comfort on bad road surfaces.

A constructional conflict between the node connecting the lower wishbone and the upright and the brake disc, requires a translation of the node vehicle inward by 0.04 [m]. Because only moving this node creates a scrub radius of 0.10 [m] and a negative king-pin inclination angle, other nodes are changed accordingly. The end result is shown in Figure 7.3, showing slightly increased camber gain for equal wheel steer angles. This is advantageous during cornering at high lateral acceleration.
Figure 7.3 Camber angle versus wheel steer angle

In addition to this kinematic analysis, the vehicle equipped with four equal suspension modules is simulated. Camber characteristics are of particular interest, because correct wheel camber angles are essential for handling and ultimate grip levels. Figure 7.4 shows the wheel camber angles for all four wheels during a steady state circular test with increasing forward velocity and therefore increasing lateral acceleration. The vehicle performing this maneuver is equipped with four wheel steering, as described in section 6.2.1.

Figure 7.4 Wheel camber angle under lateral acceleration for all four wheels during steady state circular test

Most noticeable is the fact that three of the four wheels continuously have negative camber angles, although no static camber angles are prescribed. This is due to the suspension geometry, high roll stiffness of the vehicle and the relatively large camber gain on steering. As this maneuver essentially is a large left-hand corner, it is expected that front and rear right camber angles are less negative than front and rear left camber angles due to body roll. This is supported by the simulation results, but interestingly positive camber angles on the
right side never exceed 1 °/deg. This is an important merit of the suspension design and is not only good for handling and maximum attainable lateral acceleration, but also for roll over prevention. Outer wheels with negative camber angles are more capable of supporting side forces that could cause roll over.

A comparison between the key suspension attributes of the old and optimized design is presented in Table 7.1. A small negative camber angle of -1 °/deg is proposed, which contributes to vehicle stability at high lateral acceleration, as the rear tyres are able to generate slightly higher maximum forces compared to the front tyres. This camber angle is necessary because of the rear-biased drive train lay-out and 50/50 weight distribution between front and rear axle. Table 7.1 also shows an increased scrub radius. This is almost inevitable when the node connecting the lower wishbone and upright, as describe before, is translated 0.04 [m] vehicle inward and king-pin inclination angle is desired to be kept equal in order to maintain a low level of disturbance forces in steering caused by the drive train.

<table>
<thead>
<tr>
<th>Attribute</th>
<th>Old design</th>
<th>Optimized design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Camber angle front/rear</td>
<td>0 °/deg / 0 °deg</td>
<td>0 °/deg / -1 °deg</td>
</tr>
<tr>
<td>Caster angle</td>
<td>5.5 °deg</td>
<td>6.2 °deg</td>
</tr>
<tr>
<td>Caster distance</td>
<td>0.03 m</td>
<td>0.01 m</td>
</tr>
<tr>
<td>King-pin inclination angle</td>
<td>6.2 °deg</td>
<td>6.2 °deg</td>
</tr>
<tr>
<td>Scrub radius</td>
<td>0.01 m</td>
<td>0.04 m</td>
</tr>
</tbody>
</table>

Table 7.1 Key suspension attributes for previous and optimized design

Figure 7.5 shows a top view of the designed suspension, mounted on a front axle. Main dimensions are displayed, as well as main components. The spatial arrangement of nodes ensures a for a double wishbone suspension design high decoupling between longitudinal and lateral forces, which is advantageous for elasto-kinematic design as described in the proceeding section.

A schematic representation of the vehicle side view is presented in Figure 7.6, showing the remaining space between the front and rear axle. Also the height of the suspension design is shown, which is usual for a driven and steered double wishbone axle. The upper wishbone can also be located within the wheel envelope, causing reaction forces in the chassis to become up to a factor 2 higher. The main reason to choose for this particular location of the
upper wishbone however, is the fact that the limited space between the wheels in Figure 7.5 is increased in height creating space for a drive train.

![Figure 7.6 Suspension design side view, showing dimensions](image)

In order to create a modular concept for the entire suspension around the vehicle, the same suspension geometry is used at the rear end as is mounted at the front. If the suspension would be mirrored, which is advantageous for braking and driving support angles, caster angle would be negative instead of positive. This is undesired, as the steered rear axle would create positive camber gains under steering. Additionally, the upper wishbone would force the upright to move forward under suspension travel. Comfort is strongly affected by this, because the suspension cannot yield upon impact. The effect of an equal suspension front and rear is that braking and driving force support angles are not well defined. This makes the rear end of the vehicle rise more during braking and dive more upon acceleration. This effect can be reduced by more low-speed damping on the rear axle, but cannot entirely be suppressed. Another solution is changing the orientation angle of the upper and lower wishbone, however constructional considerations have to be taken into account.

### 7.2 Qualitative bushing analysis

Elasto-kinematic properties of a suspension are particularly important for driving comfort. Disturbances to be eliminated by suspension bushings mainly are vibrations with frequencies that range from approximately the eigenfrequency of the unsprung mass of $10 \ [Hz]$, to the sonic region. The spring-damper is responsible for eliminating remaining disturbances with frequencies lower than the second ride frequency. In current suspension designs, elasto-kinematic properties however fulfill a second goal. Deformation of suspension bushings influences the position and orientation of the upright and thereby the orientation of the tyre w.r.t. the road. In current suspension design, this effect is mainly used to influence tyre toe angle to create desired behavior, for example under braking.

By application of elastic suspension bushings, the suspension geometry becomes deformable. Clever arrangement of rubber bushings can not only make sure the flexibility does not create undesired effects, but even positive adjustments of tyre orientation can be realized. In general this concerns small changes in tyre toe angle, creating small side forces for example under the influence of longitudinal force such as driving or braking. Figure 7.7 shows an elasto-kinematic suspension model in top view, where on the right clearly the tyre toe angle change is visible. Longitudinal force $Fx$ which compresses the flexible elements shown in Figure 7.7 on the left as springs. These elements, in this case the track rod in particular, force the wheel to translate slightly backwards and pivot around the king pin axis, causing the toe angle change.
Of course, this situation is only an example. Any force acting upon the suspension comprising flexible elements will cause a movement or change in orientation of the upright and therefore of the tyre.

What is essential in the qualitative choice of suspension bushings in this particular research is the fact that the vehicle is equipped with electronically controlled four wheel steering using steer-by-wire. This means that according to requirements set by the current situation, wheel toe angles can be controlled individually and independently. The fact that the vehicle has the ability to adjust its wheel toe angles independently roughly reduces the task of suspension bushings to filtering of undesired vibrations. Generally speaking any silentblock available on the market nowadays provides this quality in some extent. However it is still of great importance that wheel toe angles are left unaffected by elasto-kinematic movement of the upright, while the wheel has the ability to move slightly backwards on sharp bumps.

Figure 7.8 schematically shows the desired elasto-kinematic behavior of the double wishbone suspension design when a longitudinal disturbance force $F_x$ acts upon the wheel.

---

**Figure 7.7 1: Elasto-kinematic model; 2: wheel toe angle change due to longitudinal force**

---

**Figure 7.8 Desired elasto-kinematic properties**

Figure 7.8 left shows the five tunable effective stiffnesses which contribute to the required end result in the right figure. $k_1$ and $k_2$ represent the bushings belonging to the lower wishbone, $k_3$ and $k_4$ belong to the upper wishbone and $k_5$ represents flexibility in the track rod. Please note that every stiffness shown in Figure 7.8 is an equivalent stiffness, not
necessarily working in the direction of the rod it is connected to. The longitudinal disturbance force $F_x$ desirably only causes backward movement of the wheel and upright, but does not cause wheel toe angle change or wheel camber change.

The suspension is a three dimensional structure, meaning not only flexibility of all bushings in top view as shown in Figure 7.8 can provide the requested effect. A difference between lower and upper wishbone stiffness causes the upright to rotate around the ball joint connecting upright and upper wishbone. Effectively, this causes the upright to swing backwards and the wheel center to move backwards. Figure 7.9 shows this movement $\Delta s$ under influence of longitudinal disturbance force $F_x$. The upper wishbone in this case is rigidly connected, whereas lower wishbone bushings $k1$ and $k2$ have limited stiffness.

![Elasto-kinematic wheel movement with stiff upper and softer lower bushings](image)

The qualitative analysis above leads to Table 7.2, presenting the proposed bushing stiffness. Stiffness of $k5$ is set to stiff, because although this might cause the wheel toe angle to change under impact, it is necessary for steering feel that the track rod does not have low stiffness. If track rod stiffness is low, and therefore the total complementary stiffness of the steering system, the wheel toe angle is not robustly defined and is also changed easily by disturbance forces in other directions which are not important for ride comfort. This effect is unacceptable, meaning concessions have to be made.

<table>
<thead>
<tr>
<th></th>
<th>$k1$</th>
<th>$k2$</th>
<th>$k3$</th>
<th>$k4$</th>
<th>$k5$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x$</td>
<td>Soft</td>
<td>Stiff</td>
<td>Stiff</td>
<td>Stiff</td>
<td>-</td>
</tr>
<tr>
<td>$y$</td>
<td>Stiff</td>
<td>Stiff</td>
<td>Stiff</td>
<td>Stiff</td>
<td>Stiff</td>
</tr>
<tr>
<td>$z$</td>
<td>Soft</td>
<td>Soft</td>
<td>Soft</td>
<td>Soft</td>
<td>-</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>Soft</td>
<td>Soft</td>
<td>Soft</td>
<td>Soft</td>
<td>-</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Soft</td>
<td>Soft</td>
<td>Soft</td>
<td>Soft</td>
<td>-</td>
</tr>
</tbody>
</table>
The combination of the in all directions soft bushing $k1$ and in rotational direction soft bushing $k2$ allows the lower wishbone to pivot around the front bushing. Figure 7.10 shows the actual elasto-kinematic model for the design at hand and the movement of the upright and wishbones on disturbance force $F_x$. This movement mainly concerns longitudinal displacement $\Delta x$, possibly resulting in a toe angle change $\Delta \delta$. By application of goniometric relations on the suspension geometry, a large longitudinal travel $\Delta x$ of 0.04 $[m]$ leads to a track rod travel of 0.002 $[m]$, meaning the influence of disturbance force $F_x$ on wheel toe angle is negligible.

Figure 7.10 Left: kinematic suspension model; right: movement upon disturbance force $F_x$

As the entire suspension is designed in a modular way and is used on all 4 corners of the vehicle, the reaction of the rear suspension is exactly copied from the front suspension.
8 Conclusion and recommendations

In this research, a suspension geometry for a novel, narrow electric vehicle is developed and evaluated using simulation. To this end, a comprehensive multi-body vehicle dynamics simulation tool is developed in Matlab/Simulink, which is used to develop both a novel four wheel steering algorithm and a novel all wheel drive algorithm. Conclusions on this research are drawn in section 8.1, whereas recommendations for further research arising from this study are discussed in section 8.2.

8.1 Conclusion

Based on a literature survey and a conservative, simplified roll over model, the baseline vehicle track width of 0.92 [m] is proven insufficient to obtain equal roll over safety compared to a mid-sized vehicle currently on the market. Therefore, a track width of 1.20 [m] is proposed, which still ensures CITO vehicle's unique narrowness. Well suited suspension geometry and tuning is required to ensure optimal vehicle safety.

The presented novel feedback tracking driver model based on a virtual preview point ahead of the vehicle is able to produce steering wheel angles comparable to a real driver and is proven to be robust for all maneuvers and operating conditions presented in this research.

It is proven possible to match a simplified multi-body vehicle simulation model on measurement data by adjusting the standard 205/55R15 tyre property file and tuning of suspension characteristics and steering column stiffness. This is demonstrated using measurement data of a Peugeot 306 GTI.

A double wishbone suspension is designed, which provides a solid basis for the CITO vehicle with predictable behavior and specific anti roll over characteristics. Essential is a high roll stiffness and the orientation of the suspension arms, which prevent jacking up of the vehicle body. Other significant characteristics such as reduction of bump steer, camber gain on suspension travel and steering and a low scrub radius for driven axles are taken into account. The double wishbone suspension is designed such that it is possible to be used in a modular way on each corner of the vehicle. Only the upright is mirrored left to right, all other components are interchangeable. This reduces cost in design, production and assembly.

Based on a literature review, four wheel steering and all wheel drive are chosen as technologies with the largest potential for improving vehicle handling and safety. A novel four wheel steering with rear wheel steering angles based on velocity dependent filtering of front wheel steer angles is proven to be particularly effective in improving vehicle handling, especially in stabilization of the vehicle at high velocity and reduction of turning radius. The four wheel steering algorithm is implemented in the Matlab/Simulink vehicle model and simulated under several challenging operating conditions. Turning radius of the CITO vehicle is reduced with 30 % to 6.9 [m] curb-to-curb. The dynamic moose test exit speed is increased with 25%.
A novel yaw velocity feedback all wheel drive algorithm is developed and proven to be effective in reducing the tendency to oversteer on the edge of slip-slide by reducing torque on rear axle and driving front wheels. If a choice were to be made between either of these two technologies, four wheel steering is advised. The system provides by far the greatest improvement in vehicle handling and safety.

An integral vehicle dynamics controller necessary to get the most out of the all wheel drive four wheel steering infrastructure. This controller should combine the principle of a virtual preview point of the driver model with path tracking and a resultant force target for the entire vehicle. From this force target, both magnitude and direction of the resultant force for each wheel can be determined. This force can be generated by precisely controlling the steer angle and torque applied to each wheel. A combination of the four wheel steering and all wheel drive systems proposed in this research is also possible, but is proven not to be robust for particular occasions because conflicting goals are pursued.

The quantitative bushing analysis provides insight in desired elasto-kinematic characteristics. It increases however complexity of the steer-by-wire steering system as no elasto-kinematic self-steer properties are incorporated.

The end result is a vehicle which, in simulation, provides equal or better handling and roll over safety compared to a mid-sized vehicle, in spite of having an unfavorable ratio of track width over height of centre of gravity. The combination of, and integral approach to, suspension geometry design, four wheel steering and all wheel drive is essential to achieve this result.

### 8.2 Recommendations

In addition to the conclusion based on the results of this research, a number of recommendations is made. Model based suspension design and vehicle modeling in general pre-eminently are fields of science requiring comprehensive knowledge and a time schedule which is not possible to cover entirely within the scope of this research. Therefore a number of recommendations to extend functionality of the vehicle simulation model and suspension design are listed below.

- The driver model is purely mathematical, based on a virtual tracking point. Although this delivers driver-like response, it is not derived from real human behavior. A thorough study should be done regarding driver behavior and driver response in order to increase correspondence between vehicle simulation results and measurements.
- A verification of the CITO simulation model should be done by executing experiments with a real vehicle prototype. The prototype is planned to be built in April 2010, measurements on this vehicle increase reliability of simulation results and can pinpoint weaknesses leading to improvements in the simulation model.
- The level of detail of the vehicle model should be increased when the requested maneuver functionality is extended. This mainly concerns modeling of a braking system, drive train, chassis flexibility and driver aid systems.
- If the vehicle simulation package is to be used extensively in vehicle development, a Graphical User Interface should be developed, enabling easy access to the entire functionality of the package. This also enables an end-user to do simulation work without going in depth into the Matlab/Simulink environment.
• Simulation speed should be increased, on a state-of-the-art mobile computer, simulation currently runs at about 5% real time. The extensive multi-body suspension modeling adds to a large extent to this high simulation time, but also the SimMechanics environment slows down simulation. Possibly 3D kinematic maps, derived using the suspension evaluation model, can be used to describe suspension behavior instead of the multi-body model.

• The tyre model used in this research is a modified standard 205/65R15 tyre property file included in the DELFT-TYRE tyre library. This tyre does not correspond to a sporty tyre that is to be used on the CITO vehicle, possibly leading to noticeable incongruity between measurements and simulation results.

• The number of available and executed maneuvers should be increased to be able to assess vehicle behavior in detail. This concerns for example combined braking and steering, cross wind tests and a more elaborate comfort evaluation. To be able to simulate such maneuvers, the model complexity is to be increased simultaneously.

• The effects of elasto-kinematics on suspension behavior should be further investigated. This research should additionally be aimed at obtaining quantitative knowledge about bushing stiffness.

• Due to constructive issues or packaging requirements, further iteration steps in suspension design are likely to be necessary. Essential is to keep bump steer at about zero in any design, as well as meeting requirements regarding scrub radius and king pin inclination angle. Generally speaking, for every iteration step the entire suspension geometry has to be revised.

• The all wheel drive algorithm should be both optimized and extended with wheel spin control between front and rear axle. The current algorithm is based on a yaw velocity target and therefore is suitable only to improve vehicle yaw dynamics. Other qualities attributed to all wheel drive systems such as grip optimization on road surfaces with low friction coefficient, are not incorporated.

• The optimal solution to get the most out of the opportunities provided by both four wheel steering and all wheel drive, is an integral vehicle dynamics controller. This controller can be based on a principle of force vectoring for each tyre, depending on the situation.

• The steer-by-wire system, with which the CITO vehicle is to be equipped, should be developed and modeled in detail to be able to investigate its effect on vehicle behavior and to be able to tune the four wheel steering algorithm.

• Extensive testing with a prototype equipped with the proposed double wishbone suspension is necessary to assess all aspects of vehicle behavior, such as experienced level of comfort and steering feel.

• Currently the vehicle is designed as a 1+1, where the driver and his eventual passenger cannot sit next to each other. As, because of the analysis presented in this research, vehicle width is increased to 1.40 [m], one could consider a revised interior lay-out. The room that is created by the increase in vehicle width however can also be used to create a spacious interior concept and a sporty exterior design.
9 References


[2] NHTSA database of cars, vans, SUV, pickup trucks; SAE paper 1999-01-1336


[5] Results of handling behavior test program, Peugeot 306 GTI 2.0-16V 120kW, 1997, TNO Road-Vehicles Research Institute


[10] Mehmet Akar, 2006, ‘Yaw Rate and Sideslip Tracking For 4-Wheel Steering Cars Using Sliding Mode Control, International Conference on Control Applications (Munich, Germany)


Appendix A: Sim parameter file

clear varinf1 varinf2 varinf3 varinf4 VR_data

%% Vehicle Parameters
r_ul = 0.205*.5 + 0.075*2.54;
r_ul1 = 0.195*.45 + 0.08*2.54;
m_s = 900;     %Sprung mass
wb = 2.48;      %Wheelbase
twf = 1.2; %Trackwidth front
twr = 1.2; %Trackwidth rear
LfCoG = .5*wb; %Distance front axle to CoG
LrCoG = .5*wb; %Distance rear axle to CoG
hCoG = 0.49;    %Height CoG
Ixx = 305;      %Moment of inertia x-dir
Iyy = 600;      %Moment of inertia y-dir
Izz = Iyy;      %Moment of inertia z-dir
c_steer_front = 0.5e5;
damp_steer_front = 2500;
c_steer_rear = 0.5e5;
damp_steer_rear = 5000;

% Driver model parameters
load optimal_parameters.mat
k1 = 0;
k2 = 40;
k3 = 25*k3_best(length(k3_best), length(k3_best)); % All others
% if strcmp(driver.maneuver, 'S-S circular test ')
%     k3 = 10*k3_best(length(k3_best), length(k3_best)); % Sim Circle
% end

driver.a = LfCoG;
driver.b = LrCoG;
driver.twf = twf;
driver.twr = twr;
driver.r_ul = r_ul;
driver.k1 = k1;
driver.k2 = k2;
driver.k3 = k3;
driver.fourws = 1; %0 = front wheel steering; 1 = all wheel
steering
driver.fourwd = 2; %0 = rear wheel drive; 1 = front wheel drive; 2
= all wheel drive
driver.Tmax = 1200;
driver.statusbar = 0; %0 = no statusbar; 1 = simulation completion
monitor
save driver driver

% SAM

suspension_double_wishbone

% Nonlinear spring stiffness
% Calculate spring pre loading factor based on geometry

\[
d_1_x = p13(1) - p2(1);
d_1_y = p2(2) - p13(2);
\]

\[
\text{angle1} = \text{atan}(d_1_x / d_1_y);
\]

\[
d_2_x = p1(1) - p2(1);
d_2_y = p2(2) - p1(2);
\]

\[
\text{angle2} = \text{atan}(d_2_x / d_2_y);
\]

\[
d_3 = \sqrt{(d_1_x^2 + d_1_y^2)};
\]

\[
d_{\text{wheel}} = \sin(\pi - \text{angle1} - \text{angle2}) \cdot d_3;
\]

\[
d_4_x = p7(1) - p2(1);
d_4_y = p2(2) - p7(2);
\]

\[
\text{angle3} = \text{atan}(d_4_y / d_4_x);
\]

\[
d_5 = \sqrt{(d_4_x^2 + d_4_y^2)};
\]

\[
d_{\text{spring}} = \sin(\pi/2 - \text{angle2} + \text{angle3}) \cdot d_5;
\]

\[
\text{pre_fact} = d_{\text{wheel}} / d_{\text{spring}};
\]

% Compose springs

\[
c_z11 = 30000; \quad \% \text{Spring stiffness front linear}
\]

\[
x_0_z1 = \text{pre_fact} \cdot (LrCoG \cdot m_s / (2 \cdot \text{wb})) \cdot 9.81 / c_z11; \quad \% \text{Spring natural length front}
\]

\[
c_z12 = 1e6; \quad \% \text{Bump stop stiffness front}
\]

\[
c_z1_l = 0.07; \quad \% \text{Spring effective stroke front}
\]

\[
c_z21 = 30000; \quad \% \text{Spring stiffness rear linear}
\]

\[
x_0_z2 = \text{pre_fact} \cdot (LfCoG \cdot m_s / (2 \cdot \text{wb})) \cdot 9.81 / c_z21; \quad \% \text{Spring natural length rear}
\]

\[
c_z22 = 1e6; \quad \% \text{Bump stop stiffness rear}
\]

\[
c_z2_l = 0.07; \quad \% \text{Spring effective stroke rear}
\]

% Spring table front

\[
\text{stroke1} = 0:0.001:c_z1_l;
\]

\[
\text{force11} = (0:0.001:(c_z1_l + x0_z1)) \cdot c_z11;
\]

\[
\text{bump1_l} = 0.001:0.001:((\max(\text{stroke1})-(c_z1_l + x0_z1)) + 0.001);
\]

\[
\text{force12} = \max(\text{force11}) + \text{bump1_l} \cdot c_z12;
\]

\[
\text{force1} = [\text{force11}, \text{force12}];
\]

% Spring table rear

\[
\text{stroke2} = 0:0.001:c_z2_l;
\]

\[
\text{force21} = (0:0.001:(c_z2_l + x0_z2)) \cdot c_z21;
\]

\[
\text{bump2_l} = 0.001:0.001:((\max(\text{stroke2})-(c_z2_l + x0_z2)) + 0.001);
\]

\[
\text{force22} = \max(\text{force21}) + \text{bump2_l} \cdot c_z22;
\]

\[
\text{force2} = [\text{force21}, \text{force22}];
\]

% Nonlinear damping stiffness

\[
k_z11 = 57\cdot9.81 / 19; \quad \% \text{Damper stiffness in low speed (+10)}
\]

\[
k_z12 = 18\cdot9.81 / 79; \quad \% \text{Damper stiffness in high speed (+10)}
\]

\[
k_z1_l = 19;
\]

\[
k_z1_t = 98;
\]

\[
k_z21 = 20\cdot9.81 / 19; \quad \% \text{Damper stiffness out low speed (+30)}
\]

\[
k_z22 = 44\cdot9.81 / 79; \quad \% \text{Damper stiffness out high speed}
\]

% Damper table
% In
k_speed1 = 0:0.1:k_z1_1;
k_speed2 = 0.1:0.1:(k_z1_t-k_z1_1);
k_force1 = k_speed1 * k_z11;
k_force2 = max(k_force1) + k_speed2 * k_z12;
% Out
k_speed3 = -k_z1_1:0.1:-0.1;
k_speed4 = -(k_z1_t - k_z1_1):0.1:-0.2;
k_force3 = k_speed3 * k_z21;
k_force4 = min(k_force3) + k_speed4 * k_z22;
% Total
k_speed = [k_speed4+min(k_speed3), k_speed3, k_speed1, max(k_speed1)+k_speed2];
k_force = [k_force4, k_force3, k_force1, k_force2];
roll_front = 1.7e5;
roll_rear = 1e5;
i_s = 0/3.5;
i_s = -15/3;
i_s_r = 15/3;

%% MacPherson front suspension
%Coordinates left
p_1 = [LfCoG, -twf/2 + 0.65, .30];
p_2 = [LfCoG, -twf/2 + 0.1, .20];
p_3 = [LfCoG - 0.025, -twf/2, .30];
p_4 = [LfCoG - 0.1, -twf/2 + 0.233333, .60];
p_5 = [LfCoG - 0.1375, -twf/2 + 0.283333, .75];
p_6 = [LfCoG - 0.1, -twf/2 + 0.3, .80];
p_7 = [LfCoG + 0.1, -twf/2 + 0.1, 0.30];
p_8 = [LfCoG + 0.1, -twf/2 + 0.4, 0.3];

%Coordinates right
p_9 = [LfCoG, twf/2 - 0.65, .30];
p_10 = [LfCoG, twf/2, .30];
p_11 = [LfCoG, twf/2 - 0.15, .40];
p_12 = [LfCoG, twf/2 - 0.15, .60];
p_13 = [LfCoG, twf/2 - 0.15, .75];
p_14 = [LfCoG, twf/2 - 0.15, .80];
p_15 = [LfCoG, twf/2 - 0.15, .80];

%Torsion bar rear suspension
p_15 = [-LrCoG, -twr/2, 0.3];
p_16 = [-LrCoG, twr/2, 0.3];
p_17 = [-LrCoG, -twr/2, 0.4];
p_18 = [-LrCoG, twr/2, 0.4];

%Cruise control parameters
p = 300;
i = 50;
d = 100;
Appendix B: Spring/damper characteristics

Figure B.1 Spring characteristic showing linear spring range, bump stop stiffness and static spring deflection

Figure B.2 Damper characteristic showing nonlinear behavior and different low velocity and high velocity characteristic
Appendix C: Simulation manual

This manual provides knowledge about how to use the vehicle simulation tool presented in Chapter 3. As the entire tool is built in the generic Matlab/Simulink SimMechanics environment, the model is completely adjustable without specific programming knowledge.

Run one of the maneuver preprocessing m-files. Table C.1 presents the available maneuvers and their respective file names.

<table>
<thead>
<tr>
<th>Maneuver</th>
<th>File name</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steady state circular test, see 3.4.1</td>
<td>sim_circle</td>
</tr>
<tr>
<td>Double lane change, see 3.4.2</td>
<td>sim_DLC</td>
</tr>
<tr>
<td>Moose test, see 0</td>
<td>sim_moose</td>
</tr>
<tr>
<td>Fishhook test, see 3.4.4</td>
<td>sim_fishhook</td>
</tr>
<tr>
<td>Step steer test, see 3.4.5</td>
<td>sim_stepsteer</td>
</tr>
</tbody>
</table>

This preprocessing file defines the trajectory prescription for closed loop maneuvers and a steering wheel angle prescription for open loop maneuvers. All maneuver files are built similarly and write their respective variables into the structure called “driver”. In preprocessing m-files variables such as initial vehicle velocity and simulation time can be defined. Table C.2 shows the adjustable variables and a variable description.

<table>
<thead>
<tr>
<th>Variable name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>v_targ</td>
<td>Vehicle target velocity</td>
</tr>
<tr>
<td>t_free</td>
<td>Time where no driving torque is provided, free rolling condition</td>
</tr>
<tr>
<td>t_stop</td>
<td>Simulation end time</td>
</tr>
</tbody>
</table>

Except for sim_circle, all maneuvers are defined in a separate Simulink model, which is called in the first lines of the m-file. A “Signal Builder” block is used to define a signal which subsequently is used to derive the maneuver profile. This “Signal Builder” block provides a graphical interface to design any desired maneuver profile. The fishhook preprocessing file first runs a short steady state circular test in order to determine the steering wheel angle at 0.3g. This steering wheel angle is stored and used subsequently during the simulation process.

After running any of the maneuver m-files in Table C.1 the maneuver is fully defined. Additionally, for visualization a number of 24 cones is defined, which are located along the track. Now that the maneuver is prescribed, simulation can be performed. The m-file governing vehicle assets and vehicle behavior during simulation is the overall parameter file.
in this case parameters_SAM.m. An example of this parameter file is presented in Appendix A: Sim parameter file. In this parameter file, variables such as vehicle size and weight and spring and damper characteristics are declared. Also parameters required for the driver model, described in section 3.2 are defined and written into the aforementioned structure “driver”. Table C.3 Error! Reference source not found. presents the available tunable parameters as summed up in the parameter m-file. For information about the steering and drive algorithms, see section 6.2, respectively section 6.3.

Table C.3 Tunable parameters in parameter file and respective description

<table>
<thead>
<tr>
<th>System</th>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle geometry</td>
<td>r ul</td>
<td>Wheel radius unloaded [m]</td>
</tr>
<tr>
<td></td>
<td>m_s</td>
<td>Sprung vehicle mass [kg]</td>
</tr>
<tr>
<td></td>
<td>wb</td>
<td>Vehicle wheelbase [m]</td>
</tr>
<tr>
<td></td>
<td>twr</td>
<td>Track front [m]</td>
</tr>
<tr>
<td></td>
<td>twl</td>
<td>Track rear [m]</td>
</tr>
<tr>
<td></td>
<td>lICoG</td>
<td>Distance front axle to centre of gravity [m]</td>
</tr>
<tr>
<td></td>
<td>lRCoG</td>
<td>Distance rear axle to centre of gravity [m]</td>
</tr>
<tr>
<td></td>
<td>hCoG</td>
<td>Height centre of gravity [m]</td>
</tr>
<tr>
<td></td>
<td>lxx</td>
<td>Moment of inertia xx-axis [kgm/s²]</td>
</tr>
<tr>
<td></td>
<td>lyy</td>
<td>Moment of inertia yy-axis [kgm/s²]</td>
</tr>
<tr>
<td></td>
<td>lzz</td>
<td>Moment of inertia zz-axis [kgm/s²]</td>
</tr>
<tr>
<td>Vehicle config</td>
<td>driver.fourws</td>
<td>0 = front wheel steering; 1 = all wheel steering</td>
</tr>
<tr>
<td></td>
<td>driver.fourwd</td>
<td>0 = RWD; 1 = FWD; 2 = AWD</td>
</tr>
<tr>
<td>Spring/damper</td>
<td>c_z11</td>
<td>Spring stiffness front linear [N/m]</td>
</tr>
<tr>
<td></td>
<td>c_z21</td>
<td>Spring stiffness rear linear [N/m]</td>
</tr>
<tr>
<td></td>
<td>c_z12</td>
<td>Bump stop stiffness front [N/m]</td>
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<td></td>
<td>c_z22</td>
<td>Bump stop stiffness rear [N/m]</td>
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<tr>
<td></td>
<td>c_z1_l</td>
<td>Spring linear stroke front [m]</td>
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<tr>
<td></td>
<td>c_z2_l</td>
<td>Spring linear stroke rear [m]</td>
</tr>
<tr>
<td></td>
<td>k_z11</td>
<td>Damper stiffness in low speed [Ns/m]</td>
</tr>
<tr>
<td></td>
<td>k_z12</td>
<td>Damper stiffness in high speed [Ns/m]</td>
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<tr>
<td></td>
<td>k_z21</td>
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<td>Damper stiffness out high speed [Ns/m]</td>
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<tr>
<td></td>
<td>roll_front</td>
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</tr>
<tr>
<td></td>
<td>roll_rear</td>
<td>Roll stiffness rear [N/m]</td>
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<tr>
<td>Steering system</td>
<td>i_s</td>
<td>Steer ratio front [-]</td>
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<tr>
<td></td>
<td>i_s_r</td>
<td>Steer ratio rear [-]</td>
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<td></td>
<td>c_steer_front</td>
<td>Eq. steer column stiffness front [N/m]</td>
</tr>
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<td></td>
<td>damp_steer_front</td>
<td>Eq. steer column damping front [Ns/m]</td>
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<tr>
<td></td>
<td>c_steer_rear</td>
<td>Eq. steer column stiffness rear [N/m]</td>
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<tr>
<td></td>
<td>damp_steer_rear</td>
<td>Eq. steer column damping rear [Ns/m]</td>
</tr>
</tbody>
</table>

Now that both the maneuver is defined and vehicle parameters are determined, simulation can be performed. This simulation can either be started in Simulink or in the Matlab command window. During simulation, the screen shown in Figure C.1 is displayed.
The simulation information screen presents information about the chosen drive configuration and steering configuration. Additionally, the chosen maneuver is displayed, as is the simulation time and initial vehicle velocity.

After simulation, the line "Simulation finished successfully!" is displayed. Also information about simulation speed is shown, in form of a real time percentage.

In post processing, several figures can be plotted, depending on the simulation target. All essential variables are stored in the Matlab workspace. A visualization providing insight in vehicle tracking behavior and vehicle movement is accessed by executing visual.m. A screenshot is shown in Figure C.2. The visualization program has the ability to show both current simulation results and results from a previous simulation, automatically stored after the previous time visual.m is executed. The visualization program shows vehicle dimensions based on the vehicle geometry defined in the parameter file, see Table C.3. Tyre longitudinal and lateral forces are shown additionally, as are wheel steer angles.
Figure C.2 Simulation visualization screen shot