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Research and Analysis of Suspension Dynamics of an Autonomous Bidirectional Road Vehicle

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Abstract

The bidirectionality of road vehicles is a novel research area, which is directly connected with autonomy, efficiency, flexibility and sustainability of transportation. The research in this area is mostly performed with stringent restrictions in corporate secrecy. This fact makes the area publicly unexplored, moreover highly future oriented.

Since so little information is publicly available about bidirectionality in road vehicles, the Master thesis project was proposed by Scania CV AB to the students of The Royal Institute of Technology (KTH). This thesis topic will focus on the suspension systems investigation as well as touch upon drivetrain systems. The main objective for this project was to find the relative importance of the different suspension angles for a bidirectional vehicle, additionally to study the settings of the suspension angles for such vehicle. The study of suitable suspension systems and drivetrains was performed with tailored requirements from the corporate side of the project.

The investigation was performed as a complementary study of the possible technical solutions for a bidirectional vehicle. This included a study of suitable suspension and drivetrain systems, analytical 2-Dimensional analysis of the selected suspensions with a further investigation of one of the systems in multi-body simulations software Adams Car from MSC software. With a later, more detailed, analysis of bidirectionality implications based on the built simulation model. Ending the project was the study of relative importance of suspension angles in bidirectional vehicle with subsequent discovery of the best performing angle setups.

This thesis project provides a comprehensive introduction to the topic of bidirectionality of road going vehicles with an outlook for further investigation of the discovered suspension setups and their optimisation in urban delivery vehicle environment.

Keywords

Bidirectional vehicle, 4WS, 4WD, suspension angles, kingpin inclination, camber angle, caster angle, toe angle, relative importance, suspension
setup, suspension system, drivetrain, efficiency.
Sammanfattning

Vägfordonsegenskaper för dubbla åkriktningar är ett nytt forskningsområde som är direkt kopplat till självkörning, effektivitet, flexibilitet och hållbarhet i transporter. Forskningen inom detta område utförs mestadels med stränga begränsningar i företagshemligheter. Detta faktum gör området offentligt outforskat likväl mycket framtidsorienterat.

Till följd av att ytterst lite information finns tillgänglig för allmänheten om dubbelriktade vägfordon föreslogs examensarbete av Scania CV AB till studenter på Kungliga Tekniska Högskolan (KTH). Detta examensarbete kommer att fokusera på undersökningen av hjulupphängningssystem samt beröra drivlinesystem. Huvudsyftet med detta projekt var att hitta den relativa betydelsen av de olika hjulvinklarna i dubbelriktade vägfordon, dessutom att studera inställningarna för hjulvinklarna för ett sådant fordon. Studien av lämpliga hjulupphängningssystem och drivlinor utfördes med skräddarsydda krav från företagets sida av projektet.

Undersökningen utfördes som en kompletterande studie av möjliga tekniska lösningar för ett dubbelriktat vägfordon. Detta inkluderade en studie av lämpliga hjulupphängnings- och drivlinesystem, analytisk 2-dimensionell analys av de valda hjulupphängningssystemen med en ytterligare undersökning av ett av systemen i stelkropps dynamikmjukvaran Adams Car av MSC Software. Där det sistnämnda är en mer detaljerad analys av dubbelriktade implikationer baserat på den byggda simuleringsmodellen. Den avslutande delen av projektet var studien av relativ betydelse av hjulupphängningsvinklar i dubbelriktade vägfordon med efterföljande upptäckt av några av de bäst presterande vinkelinställningarna.

Detta examensarbete ger en omfattande introduktion till ämnet dubbelriktat vägfordon med utsikter för vidare undersökning av de upptäckta hjulinställningarna och deras optimering för stadsleveranser.

Nyckelord

Dubbelriktat fordon, fyrhjulsstyrning, fyrhjulsdrift, hjulvinklar, Kingpin-lutningsvinkel, cambervinkel, castervinkel, toevinkel, relativ betydelse,
fjädringsinställning, hjulupphängningssystem, drivlina, effektivitet.
Preface

The work of this thesis has been equally shared, where both of us have worked throughout all processes together. Parts of the thesis were quite complex, which required extra attention as well as discussions before proceeding. Therefore, we decided to work equally on each step, together.
Acknowledgments

This Master Thesis project was performed at Scania CV AB in collaboration with The Royal Institute of Technology (KTH) in Stockholm, Sweden.

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List of acronyms and abbreviations

2CV  Citroën 2CV
4WD  4-Wheel Drive
4WS  4-Wheel Steering
A    Amplitude
CLS  Corvette Leaf Spring
COG  Centre Of Gravity
CRC  Constant Radius Cornering
DWB  Double WishBone
EM   Electric Motor
EOL  End Of Life
HITS Sustainable and Integrated Urban Transport System
IC   Instantaneous Centre
ICE  Internal Combustion Engine
KPI  KingPin Inclination
LCA  Life Cycle Analysis
LCI  Life Cycle Impact
PDEM Product Development and Evaluation Matrix
RC   Roll Centre
RK   RevoKnuckle
SA  Straight-line Acceleration
SB  Straight-line Braking
SS  Step Steer
SWA  Steering Wheel Angle
SWD  Sine With Dwell
Chapter 1

Introduction

Contemporary transport systems are being developed with focus on sustainability, adaptability and efficiency. The mentioned characteristics are taken to the extreme to satisfy the future outlook of the transport sector. The commercial vehicle industry is heavily investing in areas of autonomy, electrification and last mile delivery solutions. With regard to last mile delivery solutions, the bidirectional movement of a vehicle is often overlooked. Nevertheless, such movement can bring considerable benefits in terms of adaptability, agility, manoeuvrability and efficiency in dense urban environments. The traffic congestion, noise pollution and archaic rigidity of delivery process can be aided with bidirectional functionality. Scania and their partners have introduced the Sustainable and Integrated Urban Transport System (HITS) project with the aim to revolutionise the urban last mile delivery process. The HITS project incorporates the previously mentioned features and aims to develop and optimise last mile transport system. One key to succeed in this is to develop new wheel suspension systems.

1.1 Background

Vehicle technology is evolving at a rapid pace with a constant demand for smarter, smoother and more efficient solutions. The HITS project is a research and innovation project with the goal to develop a smart, sustainable and efficient transport system in the city. This project is led by Scania and CLOSER, where the last phase of the project is set in 2024. The HITS fleet consists of different elements, where the essential parts are vehicles and load carriers. An additional goal is to meet the needs
from both users and customers, by including versatility in the design. [1]

The delivery sector today is operating with meagre optimisation, which has a great impact on efficiency. The glaring issues of the vehicle as well as the transport system are presented in the two lists below.

Vehicle:

- Non-bidirectionality of the vehicle in urban environment.
- Low agility - delivery vehicles lack 4-Wheel Steering (4WS) technology.
- Low flexibility in terms of transport tasks (goods, waste, people).
- Delivery vehicles are not designed for autonomy and electrification.
- The driver area takes a lot of space in the vehicle layout.

Transport system:

- Low filling rates.
- Large space usage in city areas.
- Not optimised for new transport concepts (parcel hubs, goods in waste out, etc.).
- Poor utilisation of time – deliveries are made only in daytime with noise and vibrations restrictions.

A larger amount of companies share the same vision of an optimised last-mile delivery, which focuses on solving the above mentioned issues. This is why the HITS project involves considerable amount of partners.

The partnership of HITS consists of: Scania, KTH, the City of Stockholm, the municipalities of Södertörn, Fabege, Catena, Atrium, Ljungberg, Ragn-sell, HAVI, Dagab, FTL, CLOSER, RISE, IVL and LogTrade. [1]
1.1.1 Reflection on Sustainability

The vehicle industry has made drastic changes throughout the last decades in order to minimise the carbon footprint. From the inefficient vehicles in the early days of motoring to the state-of-the-art transportation today. The emission target values for vehicles are revised almost every year, with tighter and tighter regulations regarding \( \text{CO}_2 \) emissions. However, the emissions during the use phase of the vehicle is only half the issue. The material extraction, production, maintenance and End Of Life (EOL) phases together with a use phase shine more light on the actual environmental impact of a particular vehicle. Such comprehensive analysis of the vehicle with all the stages of the life cycle of the product is called Life Cycle Analysis (LCA). There is reliable evidence that the electrification of the distribution vehicles can have a tremendous impact on the sustainability of the delivery sector in the scope of vehicle as a system. [2]

As mentioned in Section 1.1, the HITS project is aimed at looking into the future of last mile delivery process, where the vehicle system part of the process is optimised for the dense urban environment, highest possible efficiency, adaptability, agility and manoeuvrability. Such vehicle system can bring not only environmental benefits of lowering the use phase \( \text{CO}_2 \) emissions but also a considerable quality of life improvement for inhabitants of urban areas and customers in last mile delivery sector.

Similar to LCA, a Life Cycle Impact (LCI) is another analysis which can be performed but on a component level. Knowing the mass of the metal parts and assuming the production process and EOL process, see Table 1.1, it is possible to calculate the total energy and total \( \text{CO}_2 \) emissions for a specific part or assembly of parts by using the Equations (1.1) and (1.2).

Table 1.1: Energy and \( \text{CO}_2 \) emission coefficients per kg for steel, [3].

<table>
<thead>
<tr>
<th>Steel</th>
<th>Process</th>
<th>( E [\text{J/kg}] )</th>
<th>( \text{CO}_2 [\text{kg/kg}] )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Production</td>
<td>( 3.24 \cdot 10^7 )</td>
<td>( 2.370 )</td>
</tr>
<tr>
<td></td>
<td>Manufacture</td>
<td>( 5.21 \cdot 10^6 )</td>
<td>( 0.391 )</td>
</tr>
<tr>
<td></td>
<td>Rolling</td>
<td>( 7.00 \cdot 10^5 )</td>
<td>( 0.049 )</td>
</tr>
<tr>
<td></td>
<td>EOL Recycle</td>
<td>(-2.38 \cdot 10^7)</td>
<td>(-1.700)</td>
</tr>
<tr>
<td></td>
<td>EOL Potential</td>
<td>(-2.38 \cdot 10^7)</td>
<td>(-1.700)</td>
</tr>
</tbody>
</table>
Total $\text{CO}_2 = m_{\text{parts}} \cdot \left( C_{P/\text{kg}} + C_{M/\text{kg}} + C_{U/\text{kg}} + C_{E/\text{kg}} \right) \text{[kg]}$, \hspace{1cm} (1.1)

where $C_{P/\text{kg}}$, $C_{M/\text{kg}}$, $C_{U/\text{kg}}$, $C_{E/\text{kg}}$ are emission coefficients of production, manufacture, use and EOL phases, respectively.

Total Energy $= m_{\text{parts}} \cdot \left( E_{P/\text{kg}} + E_{M/\text{kg}} + E_{U/\text{kg}} + E_{E/\text{kg}} \right) \text{[J]}$, \hspace{1cm} (1.2)

where $E_{P/\text{kg}}$, $E_{M/\text{kg}}$, $E_{U/\text{kg}}$, $E_{E/\text{kg}}$ are energy coefficients of production, manufacture, use and EOL phases, respectively.

Performing an LCA for a complete vehicle or an LCI for a vehicle subsystem can be quite challenging, especially if those need to be done with high accuracy. It is already challenging enough to do an LCA on a complete existing model, not to mention a completely novel concept vehicle with no physical prototype. That is why such analyses will not be performed on the concept HITS vehicle, in this thesis.

### 1.2 Problems

As previously mentioned, HITS is an extensive project, which is currently in the concept phase. The challenges to address for this thesis are defined as four research questions and are presented in the list below:

1. What are the implications of bidirectionality in vehicle design?

2. What are the effects of suspension angles (camber, toe, KingPin Inclination (KPI) and caster)?

3. What are the pros and cons of the possible suspension technical solutions for a bidirectional vehicle?

4. What suspension and drivetrain configurations are suitable and what are their respective impacts for an autonomous bidirectional vehicle in last mile city distribution in 2030, in the aspect of a transport solution provider?

The first research question targets the issues that occur with bidirectionality in a steering and suspension system. Bidirectionality is
connected to the suspension angles, which therefore lays the foundation of the next question.

The suspension angles are independent and interconnected, which leads to either positive or negative effects on the system, depending on their initial calibration. For this specific reason, the second question focuses on the influence of suspension angles and their relative importance in a suspension system during various manoeuvres. The third research question distinguishes the benefits and disadvantages, in order to understand the impact of a suspension system on a bidirectional vehicle. The last question specifies the importance of investigation on the different mentioned systems, in order to find suitable steering, suspension and drivetrain configurations for the HITS vehicle.

1.3 Research Methodology

In this report, the research methods were chosen with a focus on enriching the understanding of new or scarcely researched topics. This was done to provide a wider range of knowledge for Scania in developing transport areas. The guidelines provided by the HITS project aided the selection of methods and parametrisation of the latter.

A Product Development and Evaluation Matrix (PDEM) was used together with cross verified simulation models. Those methods were selected by reasons of wide scope and validity for the HITS project. The general approach in the research was to define the most important parameters for the HITS vehicle and implement them in the PDEM. The output of the PDEM was then used in the further creation of the simulation models, which consisted of an analytical bicycle model and a multi-body simulation model, that were cross checked for verification. The parameters in the PDEM were graded with some degree of speculation, since it was impossible to find clear values in the open sources, e.g. cost and weight of the components were impossible to find. Further on, the appropriate conclusions were made based on the newly gained knowledge from the simulations.
1.4 Structure of the Thesis

The following chapter presents the theoretical modules of the thesis project, which will include and explain the following; bidirectionality, efficiency of active suspension types, drivetrain configurations, suspension systems as well as the bicycle model of the HITS vehicle. The third chapter shows the research process as well as the method that was used throughout the thesis. This chapter also describes the model verification that was performed between two softwares, i.e. a comparison was made between the results from Mathworks Matlab and MSC Software Adams Car. Other studies presented in this chapter are the kinematic analysis performed in Autodesk Fusion 360, the single parameter analysis as well as the Adams test matrix simulations. The fourth chapter presents all results together with associated discussions, while the fifth chapter shows the conclusions and future work together with related reflections. The references are presented at the end of the report.
Chapter 2

Theory

The theoretical background of this thesis is presented in this chapter, where each section represents different parts of the project. The HITS vehicle has requirements that need to be fulfilled along challenging design parameters. The HITS project aims to deliver a vehicle, which is not only safe and stable, but also highly efficient, autonomous and bidirectional.

The following section describes bidirectionality and its importance in the vehicle industry. Assumptions and relevant terms will be presented and explained, based on theoretical research. Section 2.2 will highlight relevant PhD research topics connected to this report, whereas Section 2.3 will show the different drivetrains used in this thesis and describe them in more theoretical detail. The theory of the selected suspension systems is found in Section 2.4 together with their descriptions as well as advantages and disadvantages. The suspension angles are explained in Section 2.5, while the bicycle model for a 4WS vehicle is derived and presented in Section 2.6.

2.1 Bidirectionality

Bidirectionality provides the vehicle with the ability to be driven in either direction without the need to do the turn manoeuvre. This ability is commonly overlooked in modern road vehicles, however, it can bring a multitude of benefits in crowded urban areas. The bidirectionality gives increased flexibility in parking and elevated efficiency for urban delivery vehicles. Most frequently, the bidirectionality is employed in the rail transport sector, where it saves considerable amount of funding
on developing infrastructure for unidirectional vehicles. The road vehicle sector does not provide enough recognition to bidirectionality, with the exception of few select companies, which implement this feature in their vehicles.

Prior to the research on this topic, it was agreed to establish a few assumptions, which then would be confirmed or rejected with the literature survey.

The assumptions that were made are:

- The vehicles with bidirectionality are supposed to be symmetrical in two planes - longitudinal and transversal.
- The static caster angle is set to be zero (0°) on both axles of the vehicle.

As already mentioned, the topic of bidirectionality in the road vehicle sector is considerably new, i.e. the literature and materials on this topic are vague and scarce. The interest towards bidirectionality has increased together with the growing trend of autonomy and remotely driven vehicles.

It was found that the company Zoox is preparing to launch a bidirectional urban taxi vehicle on the market in the near future, [4]. This company holds some amount of patents for such vehicle. The Independent Control of Vehicle Wheels patent 11,136,021 [5] describes a control system, which actuates the wheel and steering angles depending on the state of the vehicle and what direction it is going. This patent mentions utilisation of a positive caster angle on “front” wheels when going in one direction, same can be said about a toe in angle and a negative camber angle. The latter two angles are more related to straight-line driving and cornering, while the caster angle is more related to the direction change of the vehicle. The change of above angles is described to be happening based on control system estimation of vehicle states. The above statements are related to the examples provided in the patent, whereas the bidirectionality implications from a suspension system standpoint are not explicitly mentioned, [5]. The static suspension angles are not explicitly mentioned either.
From the information and video materials presented on the official website of Zoox, it is possible to speculate that the vehicle that is built does make use of a MacPherson suspension type with active or semi-active dampers. The vehicle also employs 4-Wheel Drive (4WD) and 4WS technologies. [4]

Komatsu holds a Dump Truck patent US20170015167A1 from 2017 [6] which is connected to a Bidirectional Autonomous Truck patent US 6,578,925 B1 from 2003 [7] and a Vector Neutral Truck patent US 6,783,187 B2 from 2004 [8]. The Dump Truck patent does not clearly specify the bidirectional functionality. Nonetheless, from the materials found on the Komatsu bidirectional autonomous mining truck, the dump truck in the patent from 2017 is assumed to be bidirectional. [9]

The Dump Truck patent explicitly describes the systems of the vehicle, i.e. it includes detailed figures and thorough description of the suspension system. From the figures of the suspension system and its description in the Dump Truck patent, it is reasonable to assume that a Double WishBone (DWB) type with 0° caster angle was later used in the production vehicle. The descriptions of vehicle systems in this patent specifically targeted the truck layout and the suspension system component placement, with regards to structural strength. The truck described in this patent employs 4WD and 4WS technologies. [6]

The Bidirectional Autonomous Truck patent from 2003 describes a bidirectional truck for mining operations with a thorough outline of the drivetrain system. From the figures presented in the patent, it was reasonable to speculate that the truck in the patent used a trailing arm suspension type. The drivetrain of that truck made use of 4WD technology. [7]

The Vector Neutral Truck patent from 2004 describes a bidirectional truck concept with an untraditional, for mining trucks, layout. The patent describes a more conceptual approach to such vehicle, with the description of manoeuvrability advantages and detailed layout of the components. It is not possible to make out useful assumptions in terms of suspension system from the figures presented in the patent. The vehicle utilises 4WD and 4WS technologies. [8]
All the above mentioned patents employ symmetry in the vehicles in two planes.

With the completed survey on bidirectionality, the prior assumptions can be summarised and reworked accordingly:

1. Symmetry of the vehicle is one of the components of bidirectionality.
2. Incorporating 4WD and 4WS technologies is beneficial for bidirectionality.
3. A $0^\circ$ static caster angle or some sort of active caster control for adding positive caster can be beneficial for bidirectionality but should be investigated thoroughly for the HITS vehicle.

### 2.2 Efficiency of Active Suspension Types

The topic of over-actuation of the suspension systems in the context of improving energy efficiency of a battery electric vehicle is not widely explored in the research publications. Peikun Sun explored a direct yaw moment control and wheel camber angle control in the context of improving energy efficiency of the vehicle during cornering manoeuvres. It was shown that it is possible to reduce the power loss on the tire by approximately 11.6% during steady state cornering, with the implementation of active camber control on the vehicle. [10]

It was also shown that the power loss reduction is decreasing with increasing velocity. It was stated that it is beneficial to use upper limits of camber angle adjustments to take full advantage of the active camber phenomenon, while at the same time not using the extreme angles to negate the efficiency gain with increase in rolling resistance loss. The direct yaw control showed marginal improvements in efficiency while cornering (approximately 1.5%) without considering electric motor efficiency maps. The combination of direct yaw control and active camber control can yield the best overall efficiency improvement. State-of-the-art optimisation of the direct yaw moment control and active camber control with addition of modelling the driveline components can yield an impressive improvement of 20% in energy efficiency in steady state cornering. The control system for the direct yaw moment control


and active camber control was developed. It should be noted that all the comparisons of efficiency were made to the simplistic equal torque distribution on all wheels. [10]

Mohammad Mehdi Davari did a thorough investigation on the influence of varying toe and camber angles on the rolling resistance coefficient and rolling loss, with a proposal of a control strategy to increase the overall vehicle efficiency while cornering. It was noted that the potential of energy saving from the active camber control of the suspension decreases with increasing velocity. Nevertheless, it was mentioned that a proposed control strategy can contribute to rolling loss reduction and promote vehicle's energy efficiency. It was stated that it is possible to achieve the desired lateral acceleration $a_y$, by combining different side slip angles $\alpha$ with appropriate camber angle. It is possible to minimise the tire slip losses using this phenomenon. [11]

The purpose of the vehicle dictates the implementation of active suspension, where considerable benefits can be achieved. However, the active suspension adds considerable complexity to the vehicle design. The active suspension was not implemented in this thesis, since the amount of knowledge for such system, in terms of bidirectionality, is not sufficient. The investigation of static suspension angles should provide baseline knowledge for further development of active suspension systems for bidirectionality.

## 2.3 Drivetrains

Vehicles are comprised of different systems where the propulsion is handled by the drivetrain. It is of high importance to select a prominent drivetrain configuration in the early stages of vehicle design, since it dictates the packaging for other systems, the functionality as well as the overall efficiency of the vehicle. Additionally, the drivetrain influences the design process and limitations for systems dependent on it.

In simplified terms, a drivetrain consists of:

- Energy storage device: battery, fuel tank or pressure tank.
- Energy conversion device, which transforms stored energy from
one type to the other: Internal Combustion Engines (ICEs) or the Electric Motors (EMs).

- Mechanical reduction device, which gives an efficiency benefit for the whole system: gearbox or fixed gear ratio.
- Driveshafts or constant velocity joints, which deliver mechanical energy to the wheels.

Numerous drivetrain versions can omit several components mentioned above and still satisfy all the designer requirements. The two most dominant types of drivetrains in today’s market are the conventional ICE and the EM. Various combinations of the two mentioned drivetrains are commonly used and resemble a hybrid-electric drivetrain category. A more obscure hydrogen electric drivetrain exists on the market but is rarely chosen, due to the technology being in its early days. The electric drivetrains comparatively provide higher efficiency levels per unit of energy and an increased packaging/integration possibilities than ICE counterparts. However, the electric drivetrains suffer from range anxiety, slow charging times and poor storage device energy capacity. The electric drivetrains are heavily dependent on the driving range design requirements, e.g. the target driving range dictates the size of the battery. A battery size and motor optimisation can be performed in order to extract the longest driving range with current battery technology.

The choice of drivetrains for this project was based on a literature study on previous research, where comparisons were made of different drivetrain configurations for electric vehicles in terms of efficiency, [12], [13], [14]. Ten promising configurations were taken for further investigation and evaluated based on the requirements of the HITS project. Three configurations were more relevant than others: chassis mounted EM with two staged gearbox (config. 9), a planetary gear set with two EMs and a differential (config. 7) and an in-wheel configuration (config. 3), [13], [14]. All three configurations can apply 4WD as well as 4WS and are schematically illustrated in Figure 2.1 below.
Figure 2.1: Schematics of studied drivetrain configurations.

The first mentioned configuration, config. 9, has a chassis mounted EM with a two staged gearbox per wheel, which can be straight cut spur gears with a synchroniser or a planetary gear set with a brake. It provides not only a higher efficiency, due to the two staged gearbox, but also individual wheel control as well as lower unsprung mass compared to config. 3. This configuration is developed by the authors of this report and is a combination of other drivetrain types.

The second choice for investigation was config. 7, which involves two EMs and a planetary gear set per axle. A previous study (done by Jinglai Wu et al.) where a comparison of different drivetrain configurations for electric vehicles was made. Their study shows that config. 7 provides an efficiency increase of 6-8% (overall) compared to the other drivetrains presented in the paper. This configuration can use smaller EMs and can be packaged in a compact way inside the chassis. It gives smooth gear transitions with no comfort compromise. Config. 7 has a lower unsprung mass compared to config. 3 and also provides a possibility to operate as continuously variable transmission (infinite ratio selection from gear range). It is not sensitive to wheel damage and will still operate the vehicle even when one of the motors fail. [14]

The third drivetrain configuration is config. 3, where the EM placement is close to the wheel and the fixed gear reduction is inside the wheel (see Figure 2.1). It provides space saving, individual wheel control as well as great flexibility in auxiliary components placement. However,
the disadvantages of this configuration are increased unsprung mass, requirement for stiffer springs for comfort as well as wheel damage sensitivity. Nevertheless, the choice to still include config. 3 in this thesis is because of its status of interest in the vehicle industry. Another reason is; low velocity urban delivery vehicles are tailored for higher efficiency and better vehicle packaging, giving lower priority to comfort.

### 2.4 Suspension Systems

A suspension is a vehicle system, which attaches a vehicle to its wheels and permits relative motion between the two. The suspension system functions are to define the wheel position relative to the vehicle body, support and distribute forces between wheel and body, satisfy the comfort requirements of the vehicle occupants and enable safe operation of the vehicle. Additionally, minimise the load variations between tire and road as well as maintain the tire in an upright position in most conditions for improved road holding.

A classical suspension system consists of:

- Springing medium to support the loads.
- Suspension arms and linkages (wishbones, swing arms, etc.) to locate the wheel relative to the vehicle body.
- Damping medium to attenuate the road imperfections and prevent unnecessary wheel movement when going over an obstacle.
- Knuckle or upright which permits rotational movement for steering system and mounting of the the drivetrain components and wheels.
- Ball joints and bushings to add compliance in some directions and prevent movement in the others.
- Anti-roll bar, which helps in load distribution during cornering and mitigates the roll movement.

The three classical suspension designs are the live axles, individual wheel suspensions and semi individual suspensions. The live axle is a common axle beam which is rigidly attached to the wheels and is suspended on the vehicle frame/body. The individual suspension has
each wheel independently suspended to the body/subframe. The third design, semi individual suspension, combines the behaviour from live axles and independently supported suspensions.

The suspension system development does not receive nearly as much attention as drivetrain development programs. However, the suspension system takes on an increasingly important role to interconnect the other systems and make the vehicle behave in a safe and stable manner on the road. The topics of suspension efficiency and over-actuation gain more and more attention with the new focus on efficient, ecological and autonomous vehicles.

As presented in the patents [5], [6], [7], [8] in Section 2.1, the suspension systems that were illustrated in the drawings were either a classic DWB or a MacPherson suspension system. Although the patents indicate that these systems work with bidirectionality, the focus of this project was shifted towards other independent suspension systems; a Corvette Leaf Spring (CLS), RevoKnuckle (RK) and a trailing arm type from Citroën 2CV (2CV).

The choice of suspension systems was founded on a study and comparison of advantages/disadvantages of each system, based on the importance for the HITS vehicle. The amount of suspension types that were evaluated in this investigation reached up to 21. The evaluation resulted in the three above mentioned suspension systems being selected for further investigation.

2.4.1 CLS

The CLS is a transverse leaf spring suspension which is a combination of DWB and a mono leaf spring. The mono leaf spring is attached to the subframe of the vehicle as well as the lower A-arms of the DWB in a lateral position, i.e. it is mounted in a transverse position between the two wheels of the same axle (see Figure 2.2), [15], [16]. The mono leaf spring replaces the regular coil springs that are otherwise used in DWB and is made of composite material, which reduces the mass of the suspension system and increases the reliability of the spring. This results in the composite mono leaf providing a lower unsprung mass with preserved spring rates as the replaced steel springs as well as a
five times increase in durability. The suspension envelope is reduced by the transverse mounted CLS, which emancipates space in the vehicle. The Centre Of Gravity (COG) of the vehicle can be lowered closer to the ground, which results in handling improvement. Additionally, the composite leaf can act as an anti-roll bar. [15] [16]

Figure 2.2: The Corvette Leaf Spring suspension, [17].

2.4.2 The RK

The next suspension system is called RK which is a modified MacPherson suspension system by Ford, (see Figure 2.3). This concept from Ford is promising for high performance front-wheel driven vehicles. The RK has improved vehicle dynamics performance (torque steer, wheelfight, steering nibble, brake judder) and reduced torque steer sensitivity to changes in tire size (low aspect ratio) and tire conicity, [18]. The RK needs very stiff top mounts in the chassis but inverted coilovers provide a good amount of suspension stiffness, which results in descent load bearing and stiffness capabilities. Compared to the DWB, the RK has a smaller KPI offset, is lighter in weight and is also less expensive. [18]
2.4.3 The 2CV

The Suspension system that Citroën used in their car model 2CV was pitch interconnected, see Figure 2.4. The 2CV is a quite simple and not expensive construction with a caster angle that adjusts with loading. The 2CV is a comfortable and soft suspension at the same time as it has a reduced pitch motion. It was inexpensive to build and required almost no maintenance. First versions of this suspension used friction shock absorbers and inertia dampers. The pitch interconnection gives considerable benefits in ride quality over rough terrain. [20], [21], [22].

As seen in Figures 2.4 and 2.5, the wheelbase has a horizontally mounted cylinder on each side, which contains a pair of coil springs. The coil springs are operated by bellcranks and rods and together with a pitch interconnection, the cylinders move horizontally. [20], [21], [22].
Figure 2.4: The Citroën 2CV chassis, [23].

Figure 2.5: Layout of the wheelbase and suspension system of the Citroën 2CV with the horizontally mounted cylinder, [24].
2.5 Suspension Angles

As presented in the previous Section 2.4, the purpose of the suspension system is to provide stability, safety and comfort to the vehicle according to the customers' desires. The suspension system depends on different parameters: suspension setup, suspension geometry and kinematic behaviour. The suspension setup includes the different suspension angles, which will be explained in this section. Figure 2.6 illustrates the KPI, camber, caster and toe in/out angles as well as the scrub radius. It is of high priority to set the different angles in appropriate degrees, not only to achieve high performance but also to prevent undesired or unexpected behaviour.

(a) KPI angle and scrub radius in front/rear view of the vehicle, [25].

(b) Camber angle in front/rear view of the vehicle, [26].

(c) Caster angle in side view of the vehicle, [26].

(d) Toe in/out in top view of the vehicle, [26].

Figure 2.6: Suspension angles: (a) Kingpin angle and scrub radius, (b) Camber angle, (c) Caster angle, (d) Toe in/out.

The KPI is the angle between the kingpin axle (dashed line) and the vertical axis (dotted line in centre of the tire) and is presented in Figure 2.6a. The KPI is mounted with an offset, which in turn decides the size of the scrub radius. The kingpin axis, together with the scrub radius,
create a moment arm which facilitates the wheels to return back to initial position (straight-line orientation). Positive KPI is defined when the kingpin axis is tilted towards the centre of the vehicle, as seen in Figure 2.6a. Negative and neutral KPI are defined when the kingpin axis is tilted outwards from the centre and positioned parallel to the vertical axis perpendicular to the ground, respectively.

The camber angle, which is illustrated with a front view in Figure 2.6b, can be either tilted in negative, neutral or positive directions. A negative camber angle is the result from when the top parts of the wheels are tilted inwards to the centre of the vehicle, whereas a positive camber angle has the wheels upper sections tilted outwards. A neutral camber has zero angle on the wheels and they are parallel to the vertical axis, perpendicular to the ground.

The caster angle can also be setup in negative, neutral and positive angles and is illustrated in Figure 2.6c from a side view of the vehicle. A negative caster is set when the top of the strut/upright is tilted forwards from the centre of the vehicle, while positive caster is when the strut top is tilted rearwards towards the centre of the vehicle. Neutral caster is when the strut is parallel to the vertical axis, perpendicular to the ground.

The toe angle is presented from a top view of the vehicle, in Figure 2.6d. Negative toe (toe in) is defined when the front part of the wheels are turning inwards towards the centre of the vehicle, whereas positive toe (toe out) turns in the opposite direction, i.e. outwards. Neutral toe is defined when the wheels have no angle and are positioned straight, parallel to the longitudinal axis.

In terms of bidirectionality, the angle setup for caster and toe will change depending on the driving direction, e.g. a caster angle with positive setup will become a negative setup when the vehicle changes its driving direction. Camber and KPI are unaffected by the driving direction change.

A suspension system has several parameters that need to be set correctly in order for the system to work to its full potential. Although, one setup might be optimal for one vehicle does not necessarily mean that it will be optimal for all others. The suspension setup is made on a
case by case basis for each vehicle. As formerly mentioned, the most important parameters to be investigated in this thesis project are the four angles: camber, toe, KPI and caster. The angles can take a negative, neutral or positive setup, which in turn results in different benefits and disadvantages. Bidirectionality affects the setup of the suspension angles, especially caster and toe which will be mirrored in the front and rear, whereas KPI and camber are not. The suspension angles, including their three different setups, will be described below.

2.5.1 Camber

A negative camber angle is the most commonly used setup. It enhances the handling of the vehicle when cornering by giving less deformations to the contact patch between tire and road. It can also reduce wheel vibrations and allow cornering while driving with higher velocities. However, while improving the performance for cornering, the straight-line acceleration and braking is reduced and the braking distance is increased. Straight-line stability is also compromised and premature tire wear will occur. The contact patch between tire and road is smaller compared to the contact patch with neutral camber, which also results in poor traction during wet conditions. [27]

Neutral camber is when the tires are set in zero angle position. This equals to the tire having a full contact patch against the road while driving in straight-line, which results in great handling of the vehicle. Overall, neutral camber is beneficial while driving in straight-line, however, it has disadvantages while cornering. This means that the contact patch will reduce in turning manoeuvres and restricts hard cornering. [27]

Positive camber is more likely to be used in heavy vehicles with dry weight. This is due to camber gain where an added load to the vehicle will affect the position of the wheels and push them towards neutral setup. Positive camber also assist to keep the vehicle straight on uneven roads due to camber thrust. However, similarly to negative camber, the disadvantages of this setup is increased braking distance as well as premature tire wear and tear will occur. [27]
2.5.2 Toe

Negative toe angle (toe out) gives faster turns and higher stability in cornering compared to straight-line driving, where this setup pulls the car into yaw motion. This creates a noticeable wobbly motion that might be unpleasant at higher velocities. Negative toe in the rear provides the benefit of improved grip for acceleration in the rear due to the induced slip angle $\alpha$ on the tire. However, this also causes the vehicle to oversteer in cornering, especially in rear-wheel drive vehicles. [28]

Neutral toe angle reduces both tire wear and power loss down to the minimum, while used in the front and/or rear suspension system. Compared to positive/negative toe angle, this setup increases the stability of the vehicle on straight-line driving, while the stability is compromised while cornering. [28]

Positive toe angle (toe in) improves the stability of the vehicle in straight-line, as well as increasing straight-line grip, which enables higher acceleration. Positive toe also increases the tire temperature faster compared to a neutral toe setup and can assist to reduce oversteer problems. However, the fast increasing tire temperature can also be considered a disadvantage, due to reduced tire life and energy efficiency caused by the heated up tires. Uneven tire wear issues due to bad wheel adjustment and slow steering response are also disadvantages of positive toe angle. [28]

2.5.3 KPI

The KPI and the wheel offset, together with the scrub radius, affect the steering effort of the vehicle. By producing a lifting force, a self-centering effect is created, which increases the steering effort. This also increases straight-line stability when driving in low velocities. The negative and neutral KPI values are not encountered in presented suspension types, therefore, only positive KPI will be studied in this thesis. [29]

The scrub radius is defined as the dimension between the kingpin axis and a vertical line coincidentally constrained to the contact patch point of the wheel and the ground (see Figure 2.7). [29]
2.5.4 Caster

Negative caster is rarely used due to the scarce amount of benefits. The only advantage of using negative caster is to facilitate the turn of the steering wheel, whereas the disadvantages are reduced stability as well as poor handling. [30]

Neutral caster in both front and rear creates a bidirectional setup by avoiding the use of negative caster. With no real tuning, it is clear that the benefits from positive caster will not be applied on this setup. Similar to negative caster, neutral caster also facilitates the steering effort. However, it is not in the same level as with negative caster and it is also without jeopardising the stability of the vehicle. [30]

Positive caster is the most commonly used setup in the vehicle industry due to the benefits of better handling when cornering, increased stability at higher velocities as well as enhanced straight-line tracking. This is due to the self-aligning torque, which is created by the positive caster angle. Positive caster is affected by dynamic camber while cornering, which results in the good outcome of increased tire lean and contact patch. However, if dynamic camber would be extreme, it would result in the vehicle understeering mid-corner. Another disadvantage is that the
steering effort increases proportionally to an increasing positive caster and will be noticeable in the lack of power steering. [30]

2.6 Bicycle Model

The bicycle model was introduced in the 1950’s by Leonard Segel and is the simplest vehicle dynamics model, which gives essential information on vehicle movement. The bicycle model is a single track model, which means that the virtual car only has two wheels. In turn, this implies that no Ackerman steering geometry is present. The model is assumed to be in 2-Dimensions with a body fixed coordinate system (COG, x, y, z) attached to the COG of the vehicle, see Figure 2.8. Additionally, the height over ground of the COG is assumed to be zero and an inertial coordinate system (O, X, Y, Z) is defined. [31]

![Figure 2.8: Bicycle model, [32].](image)

The specific restrictions of the model are presented below:

- Neglected lateral load transfer.
- Neglected longitudinal load transfer.
• Neglected roll, pitch and jump motion.
• No aerodynamic effect.
• No chassis or suspension compliance.
• Left and right side symmetric vehicle.
• COG in XY plane.

With all the shortcomings mentioned above, the bicycle model is, nevertheless, able to account for numerous properties of the stability and dynamic behaviour of vehicle under various conditions. The model can also be expanded to account for load transfer effects and additional motion modes.

Some of the available properties from the bicycle model:

• Ability to find lateral acceleration \( a_y \) and yaw rate \( \dot{\psi} \) of the vehicle.
• Illustrate trajectory of the vehicle travel.
• Find the vehicle response.
• Finding understeer gradient \( K_{us} \).
• Finding slip angles \( \alpha \) on the tires.
• Finding the critical and characteristic velocities \( v_{crit} \) and \( v_{char} \).

The bicycle model was further developed to include the rear wheel steering in accordance with HITS vehicle requirements. In chapter 2.3 in Vehicle Dynamics and control by Rajesh Rajamani, such model is described and derived, [33]. It was decided to derive the model in a slightly different way for easier implementation in the Matlab code and to crosscheck the results.

As previously mentioned, the bicycle model is a 2-Dimensional model for investigations of lateral vehicle motion. The need for such model appears from the fact that the velocity at each wheel of the vehicle is not following in the direction of the wheel at high longitudinal velocities. The two degrees of freedom of the model come from the definition of position of the vehicle with lateral position and the yaw angle \( \psi \). [33]
The HITS vehicle includes features such as 4WD and 4WS, which results in increased agility and stability. The bicycle model for the HITS vehicle was derived, in order to verify that 4WS is necessary to make the vehicle more stable and manoeuvrable as well as to verify the behaviour of the Adams Car model. The half car model presented in Figure 2.8 was modified to a half car model with 4WS, see Figure 2.9 below.

\[ \begin{align*}
\uparrow x: & \quad ma_x = -F_{12} \sin \delta_F - F_{34} \sin \delta_R, \\
\downarrow y: & \quad ma_y = F_{12} \cos \delta_F + F_{34} \cos \delta_R, \\
\widehat{\psi}: & \quad J\ddot{\psi} = F_{12} \cos \delta_F f - F_{34} \cos \delta_R b,
\end{align*} \]

where \( m \ [\text{kg}] = \) total vehicle mass, \( a_x \ [\text{m/s}^2] = \) longitudinal acceleration, \( a_y \ [\text{m/s}^2] = \) lateral acceleration, \( F_{12} \ [\text{N}] = \) lateral force on the front wheel, \( F_{34} \ [\text{N}] = \) lateral force on the rear wheel, \( \delta_F \ [\text{rad}] = \) angle at the front wheel, \( \delta_R \ [\text{rad}] = \) angle at the rear wheel, \( J \ [\text{kg} \cdot \text{m}^2] = \) second
moment of inertia, $\ddot{\psi} \ [rad/s^2]$ = yaw acceleration, $f \ [m]$ = distance from the front axle to COG and $b \ [m]$ = distance from the rear axle to COG.

The final derived results are presented in Equations (2.4) - (2.7) and are verified by the previous derivations made by Davari and Rajamani, where other vehicles with 4WS were studied, [11], [33].

\[
m(\ddot{v}_y + \dot{\psi}v_x) = - C_{12} \alpha_{12} \cos \delta_F - C_{34} \alpha_{34} \cos \delta_R,
\]
\[
f\ddot{\psi} = - C_{12} \alpha_{12} \cos \delta_F f + C_{34} \alpha_{34} \cos \delta_R b,
\]

where $\ddot{v}_y \ [m/s^2]$ = lateral acceleration, $v_x \ [m/s]$ = longitudinal velocity, $\dot{\psi} \ [rad/s]$ = yaw rate, $C_{12} \ [N/rad]$ = cornering stiffness at the front wheel, $\alpha_{12} \ [rad]$ = slip angle at the front wheel, $C_{34} \ [N/rad]$ = cornering stiffness at the rear wheel and $\alpha_{34} \ [rad]$ = slip angle at the rear wheel.

The slip angles $\alpha$ are derived as:

\[
\alpha_{12} = \arctan \left( \frac{v_y + f\dot{\psi}}{v_x} \right) - \delta_F,
\]
\[
\alpha_{34} = \arctan \left( \frac{v_y - b\dot{\psi}}{v_x} \right) - \delta_R,
\]

where $v_y \ [m/s]$ = lateral velocity.

From the equations presented above and the chapter 2.3 in Rajamani [33], it was concluded that the basic equations of motion for the 4WS bicycle model were correct.

Further on, the functionality for vehicle path tracking was added to the Matlab bicycle model code. It was based on coordinate frame transformations from chapter 2.4 in Rajamani [33]. The calculation of steering sensitivity $\frac{\partial \dot{\phi}}{\partial \delta}$ was added to see the vehicle behaviour in a Constant Radius Cornering (CRC) test. The understeer gradient $K_{us}$ as well as the critical and characteristic velocities $v_{crit}$ and $v_{char}$ calculations were added into the code functionality.
Chapter 3

Methods

This chapter presents the different methods used in this thesis project and describe them more in detail in the following sections. The different parts of the project required different approaches in order for the results to be as realistic and true as possible.

The first section of this chapter will explain the research process model that was used together with the approach of selecting the drivetrain configurations and suspension systems in this thesis. Section 3.2 presents the PDEM, Section 3.3 describes the model verification between the two softwares Matlab and Adams Car and Section 3.4 presents the kinematic analysis of the suspension systems performed in Fusion 360. A single parameter study was performed and is described in Section 3.5, while the complete study of all suspension angles is presented in Section 3.6.

An important note, the methods including Adams simulations are only performed on the CLS. The RK and 2CV were analysed through the kinematic analysis together with all the CLS test results. The choice of testing only one suspension type was made because of restrictions on time and software related issues.

Another note, in order to investigate drivetrain configuration 3, it was decided to test the CLS Adams model with additional unsprung mass and without (where the additional unsprung mass portrays configuration 3 in Adams). This was done due to configuration 3 utilising hub EMs, which increase the unsprung mass.
3.1 Research Process

The research process in this work was chosen in direction of a general research model. The research plan was to build analytical and multi-body models, run an evaluation on the basis of selected tests and then crosscheck the results, in order to verify the models. The diagram with the related research steps is outlined in Figure 3.1. Steps 1 - 3 were done according to the research process diagram. Step 4 (Choosing the study design) was selected to be an experiment. In step 5, the PDEM matrix was selected as sample design. On the basis of this matrix process, the analysed combinations were selected in accordance with the most important characteristics for the HITS project. Additionally, a benchmarking process was done to aid the final selection of samples. Steps 6 - 8 were done in accordance to the general research model.

![Research Process Diagram](image_url)

Figure 3.1: The research process model, [34].
3.2 PDEM

In order to evaluate the drivetrain configurations and suspension systems, a reference for evaluation must be defined in the initial phase of the project. A proper scale and suitable parameters must be specified for understanding if the drivetrain configurations and suspension systems are considered 'good' or 'bad'. The PDEM is a $nxm$ matrix, where $n$ represent the number of rows and $m$ the number of columns. In this report, $n$ includes the number of parameters, while $m$ lists the different combinations of drivetrain configurations and suspension systems that were evaluated in this thesis, see Table 3.1.

Table 3.1: PDEM with parameters.

<table>
<thead>
<tr>
<th>Main Parameters</th>
<th>Sub-parameters/Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Safety/Stability</td>
<td>Safety/Stability</td>
</tr>
<tr>
<td>Bidirectionality</td>
<td>Bidirectionality</td>
</tr>
<tr>
<td>Cost</td>
<td>Product cost (direct cost)</td>
</tr>
<tr>
<td></td>
<td>Wearable parts (indirect cost)</td>
</tr>
<tr>
<td>Energy efficiency</td>
<td>Drivetrain efficiency</td>
</tr>
<tr>
<td>High up-time</td>
<td>High up-time</td>
</tr>
<tr>
<td>Weight</td>
<td>Weight</td>
</tr>
<tr>
<td>Quietness (night drive)</td>
<td>Noise level</td>
</tr>
<tr>
<td>Kinematic behaviour</td>
<td>Kinematic behaviour</td>
</tr>
<tr>
<td>Manoeuvrability</td>
<td>Steering angle</td>
</tr>
<tr>
<td>Serviceability</td>
<td>Easy access</td>
</tr>
<tr>
<td></td>
<td>Low complexity of service tasks</td>
</tr>
<tr>
<td></td>
<td>Long service intervals</td>
</tr>
<tr>
<td>Compactness</td>
<td>Dimensions (Z)</td>
</tr>
<tr>
<td></td>
<td>Suspension envelope</td>
</tr>
<tr>
<td>Robustness</td>
<td>Mechanical strength</td>
</tr>
<tr>
<td></td>
<td>Failure</td>
</tr>
<tr>
<td>Comfort</td>
<td>Noise and vibration levels</td>
</tr>
<tr>
<td>Sustainability</td>
<td>Sustainability</td>
</tr>
</tbody>
</table>
3.3 Model Verification – Matlab vs Adams

The experiments were separated in different stages in order to ensure validity across the research process. The first stage was to implement the bicycle model in Matlab. This was done to verify that the 4WS is needed in the vehicle. The next stage was to create and simulate a full vehicle model in Adams which was verified against the bicycle model to ensure validity.

The outputs from the Adams simulations were used as inputs for the bicycle model in Matlab, such as the Steering Wheel Angle (SWA) and the longitudinal velocity $v_x$. The vehicle path was used as an indicator of proper operation between the two softwares. The 4WS strategy in both softwares is an opposite phase steering in the "rear" axle with identical values of SWA, see Figure 3.2. The parameters that were compared in this study were the lateral acceleration $a_y$ as well as yaw rate $\psi$.

![Figure 3.2: The 4WS 'opposite phase' strategy.](image)

3.3.1 Matlab

The SWA and the longitudinal velocity $v_x$ are output values taken from Adams, which then were used as input in the bicycle model. The vehicle
parameters were defined in Matlab, where most of them were given requirements from HITS. However, the cornering stiffness $C$ and the friction coefficient $\mu$ had to be defined through an iterative process. The friction coefficient $\mu$ was later found in the tire property file in Adams and the value could then also be verified to be accurate enough. The value of the friction coefficient $\mu$ is presented below.

$$\mu = 0.52$$

As previously mentioned, the lateral acceleration $a_y$ and yaw rate $\dot{\psi}$ are the final outputs to be studied and compared between Adams and Matlab. The bicycle model also provides information such as the steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$, understeer gradient $K_{us}$ as well as the critical and characteristic velocities $v_{crit}$ and $v_{char}$, see Equations (3.1) and (3.2). Equation (3.3) presents the steering sensitivity for a neutral steered vehicle ($K_{us} = 0$).

$$v_{crit} = \sqrt{\frac{L}{+K_{us}}} \, \text{[km/h]}, \quad (3.1)$$
$$v_{char} = \sqrt{\frac{L}{-K_{us}}} \, \text{[km/h]}, \quad (3.2)$$
$$\frac{\dot{\psi}}{\delta} = \frac{v_x}{L}, \quad \text{[1/s]} \quad (3.3)$$

Where,

$$L = 4.5 \, m,$$
$$K_{us} = 0, \quad \text{(value for neutral steered vehicle)}.$$

Due to the understeer gradient $K_{us}$ being equal to zero, the velocities $v_{crit}$ and $v_{char}$ are not valid. This results in the steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ graph being a straight line.

### 3.3.2 Adams

In order to verify the Adams model of the CLS, three tests were run and compared against the bicycle model. The tests that were chosen are presented in the list below, together with the focus of each test,
respectively:

- CRC – to find the balance of the vehicle.
- Step Steer (SS) - to find proper tire limits of the tire model and correct value of the tire friction coefficient $\mu$.
- Slow slalom - to find proper cornering stiffnesses $C_{12}$ and $C_{34}$.

The test standards ‘Road vehicles – Lateral transient response test methods – Open-loop test methods ISO 7401’, ‘Passenger cars – Steady-state circular driving behaviour – Open-loop test methods ISO 4138’ and ‘Road vehicles – Vehicle dynamics test methods – Part 2: General conditions for heavy vehicles and buses ISO 15037-2’ [35], were used for this study and guided the setup for the Adams simulations (including all other Adams simulations in the rest of the thesis). The CRC test was run with a radius of $120m$ and lateral acceleration $a_y$ ramp up from $0.1 m/s^2$ to $6m/s^2$, test runtime was $90s$. Such setup clearly illustrated the balance of the vehicle at the limit of cornering grip. The SS test was run with $90^\circ$ of SWA, a ramp up of $0.15s$ for the steering input and initial velocity of $60km/h$, the test was run for $30s$. This setup of the SS illustrated the limits of the tires in an adequate way. The swept slalom test was run with maximum SWA value of $75^\circ$, initial frequency of $0.01Hz$, maximum frequency of $0.5Hz$, frequency rate of $0.5$, velocity of $45km/h$ and a runtime of $20s$. Such slalom test gave an accurate representation of the lateral stiffnesses of the vehicle.

### 3.4 Kinematic Analysis in CAD 2D

The method used for a kinematic analysis in this study required all three models to be projected to simplified 2D models in a CAD environment - Fusion 360. The analysis, which included parallel wheel travel and roll motion, investigated the change of the suspension angles as well as the change of Roll Centre (RC) height. The models were drawn in both front and side view, where the front presents the information of camber, KPI, RC and scrub radius, whereas the side view presents the caster and mechanical trail (see Figure 3.3). The wheel travel is moved in $\pm80mm$ in vertical direction and the roll motion was set to a $5^\circ$ roll to the left (see Figure 3.4 and 3.5 where the two motions are represented on the RK model, respectively).
Figure 3.3: Models in front and side view (front view of 2CV was excluded): (a) CLS and RK side view, (b) RK front view, (c) 2CV side view, (d) CLS front view.
The 2CV front view is omitted from the CAD analysis due to the fact that the kinematic behaviour of the model is theoretically known. The toe angle was not included in the CAD analysis, however, it was studied in Adams car. This section presents the assumptions made for the simplified 2D models as well as the method used for the kinematic analysis. The results and discussion are presented in the next chapter.
3.4.1 Models

In order to perform the kinematic analysis on the three suspension models in 2D, a couple of assumptions were made to the simplified models. The assumptions are presented below:

1. Due to the models being simplified in 2D, the side view is equally alike for the CLS and the RK. As can be seen in Figure 3.6, the side view model of the CLS and RK lacks a pivot point, compared to the 2CV. The wheel is supposed to move along an arc when moving in the upwards motion, where the arc is found by the pivot point. Due to the lack of the pivot point, there is no possibility to perform the upwards wheel motion in this 2D drawing in order to study the change in caster angle and mechanical trail change in parallel wheel travel. However, because of the upper and lower control arms of the CLS (lower control arm and strut for RK) moving parallel, there is mostly no change in caster angle nor mechanical trail.

2. The 2CV is assumed to have the chassis walls and the vertical side of the wheel parallel to each other. This enables a kinematic study of this model in roll motion.

3. The KPI in RK is set to 0° in initial state in this analysis and is therefore parallel to Camber, i.e. the upright axis of rotation in this model is parallel to the side of the wheel (see Figure 3.7). This results in KPI being equal to camber in initial state and have equal change in dynamic motion.

4. The A-arms in the CLS are parallel to each other and completely horizontal as well as having the RC positioned on the ground in initial state (see Figure 3.3d). In a roll motion of 5° to the left (see Figure 3.8), the lines only intersect on one side of the vehicle and this is the only possible assumption to make for finding the movement of the RC.

5. This next assumption is made for the modelling of the contact patch movement. Other softwares can be used in order to produce a more accurate contact patch model, however, this is unnecessary in a simple 2D analysis. The scrub radius in this analysis is now defined as a dimension between the contact patch point on the wheel and a point of the intersection of KPI angle with the
horizontal line, which is coincident with the contact patch point (see Figure 3.9). The horizontal line will move together with the contact patch point in every kind of motion the wheel is forced into, including movements where the wheel lifts off the ground. This approach provides a different result compared to the vertical line being attached to a contact patch point on the ground (see the difference of the values in Figure 3.9).

Figure 3.6: Assumption 1 - no pivot point in the CLS and RK.

Figure 3.7: Assumption 3 - KPI parallel to camber (RK).
Figure 3.8: Assumption 4 - lines only intersect on one side of the vehicle due to horizontal A-arms (CLS).

Figure 3.9: Assumption 5 - redefined scrub radius.
3.4.2 Performed Analysis

Two projections of the suspension systems, front and side view, were analysed through different motions; parallel wheel travel and roll motion.

Studied from front view, the parallel wheel travel in CAD was performed by moving the wheel upwards +80mm, as well as downwards −80mm. The camber and KPI change, together with the scrub radius, were studied through this analysis. The assumptions made in Section 3.4.1 enable the scrub radius to follow the up and down motion of the wheel.

Studied from front view, the roll motion was performed by fixing the lower mid-centre point of the chassis to the vertical axis and then rolling the vehicle 5° to the left. The wheels have the contact patch point tangentially constrained to the horizontal axis and roll together with the whole vehicle. The assumption of the scrub radius here is the same as the one for parallel wheel travel.

Studied from side view, the analysis focuses on the change of caster angle and mechanical trail. For the 2CV, the wheel is moved upwards +80mm around its pivot point (see Figure 3.3c) and the changes of caster angle and mechanical trail are studied. As stated in assumption 1 in Section 3.4.1, regarding the lack of pivot point in the CLS and RK, the results in the next chapter will be more theoretical. Roll motion is not performed in side view due to the rotation being around a third axis in a perpendicular plane.

The roll stiffness $K_\phi$ was calculated for all three models where the 2CV was treated as a modified swingarm, i.e. the distance between the spring and the pivot point was considered to be the same as the distance between the bellcrank and the pivot point (see Figure 3.10).

![Figure 3.10](image-url)

Figure 3.10: Side view of the 2CV with spring position approximation.
Even though the RK is a modified MacPherson, the calculation for the roll stiffness $K_\phi$ is still the same. Equations (3.4), (3.5) and (3.6) are used for the roll stiffness calculations of the CLS, RK and 2CV, respectively.

$$K_{\phi, \text{CLS}} = \left( \frac{s}{2} \right)^2 \left( \frac{f_F}{f_L} \right)^2 \cdot k \quad [\text{Nm/\text{rad}],} \quad (3.4)$$

$$K_{\phi, \text{RK}} = \left( \frac{b \cdot d}{a} \right)^2 \cdot k \quad [\text{Nm/\text{rad}],} \quad (3.5)$$

$$K_{\phi, \text{2CV}} = \left( \frac{s}{2} \right)^2 \left( \frac{f_F}{f_L} \right)^2 \cdot k \quad [\text{Nm/\text{rad],} \quad (3.6)$$

The geometrical definitions can be seen in Figures 3.11, 3.12 and 3.13.

Figure 3.11: CLS geometry, [32].
The ride frequency for loaded and unloaded cases was calculated for all three suspension types by using Equation (3.7).

\[
f = \sqrt{\frac{\text{Equivalent wheel stiffness}}{\text{sprung mass}}} \cdot \frac{1}{2 \cdot \pi} \quad [Hz]
\]  

(3.7)
3.5 Single Parameter Study

The single parameter study was done in order to investigate the bidirectional behaviour and to gain additional knowledge about the suspension angle performance before the test matrix investigation. The study includes a neutral setup, positive bidirectional setup as well as a non-bidirectional setup. The tests used in this study were also revised in order to see which tests to perform in the final Adams Test Matrix Simulations. The series of tests were run in Adams and included the five tests: SS, CRC, Straight-line Acceleration (SA), Straight-line Braking (SB) and Sine With Dwell (SWD). The test setups were correlated to the input values in Section 3.3.2, with minor changes in some of the tests. The tests which required velocity input had it chosen as the most frequently occurring velocity value: 50 km/h.

Each test was run once for each single parameter change. The lateral acceleration $a_y$, yaw rate $\dot{\psi}$, steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$, SWA, forces in the steering system as well as the longitudinal velocity $v_x$ when the instability of the steering system starts were studied in the process.

The steering input in the tests SS and SWD was adjusted according to the change of angles in the suspension. This was done in order to replicate identical angles at the wheels regardless of the changes in KPI and caster angles. Since, in this particular study, the adjustment of the caster angle results in a slight change in the steering leverage length. This, in turn, changes the overall steering ratio, which then calls for SWA input change. See Table 3.2 below for the steering input for both tests:
Table 3.2: Steering input in SS and SWD.

<table>
<thead>
<tr>
<th>Gear and velocity</th>
<th>Test</th>
<th>Caster setup</th>
<th>Input SWA</th>
<th>Angle at the inner wheel</th>
<th>Angle at the outer wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>2nd gear 50 km/h</td>
<td>SS</td>
<td>Caster +2°</td>
<td>90</td>
<td>2.9174</td>
<td>2.8267</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster 0°</td>
<td>86.9</td>
<td>2.9193</td>
<td>2.8231</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster −2°</td>
<td>83.9</td>
<td>2.924</td>
<td>2.8214</td>
</tr>
<tr>
<td>5th gear 90 km/h</td>
<td>SS</td>
<td>Caster +2°</td>
<td>38.5</td>
<td>1.4891</td>
<td>1.4639</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster 0°</td>
<td>37.3</td>
<td>1.4885</td>
<td>1.4618</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster −2°</td>
<td>36.1</td>
<td>1.4875</td>
<td>1.4591</td>
</tr>
<tr>
<td>3rd gear 50 km/h</td>
<td>SWD</td>
<td>Caster +2°</td>
<td>69</td>
<td>2.3828</td>
<td>2.3215</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster 0°</td>
<td>66.7</td>
<td>2.3864</td>
<td>2.3211</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster −2°</td>
<td>64.4</td>
<td>2.3894</td>
<td>2.3198</td>
</tr>
<tr>
<td>5th gear 90 km/h</td>
<td>SWD</td>
<td>Caster +2°</td>
<td>25.9</td>
<td>1.0452</td>
<td>1.0319</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster 0°</td>
<td>25.1</td>
<td>1.0428</td>
<td>1.0287</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Caster −2°</td>
<td>24.4</td>
<td>1.0441</td>
<td>1.029</td>
</tr>
</tbody>
</table>

### 3.5.1 SS

The SS test was run with the same steering ramp up and steering angle inputs as in Section 3.3.2. The velocity was reduced from 60 km/h to 50 km/h to satisfy the vehicle velocity operating range. This test gave an understanding of the vehicle response and influence of suspension angles on it.

### 3.5.2 CRC

Adams has a default solver setting of error tolerance level $10^{-4}$. However, the solver error tolerance level had to be changed to $10^{-1}$, in order for the CRC test to actually run. Furthermore, there was minimal change of the behaviour of the vehicle in this test. The test setup for the CRC is identical to the one in Section 3.3.2 with only one difference: reduced radius. The radius of the circle was altered to 150 m, since 120 m was deemed insufficient for the suspension angle study. The vehicle did not complete the full circle and was skidding off with the highest lateral acceleration $a_y$. This gave a good representation of the nature of the vehicle with different suspension angles.

### 3.5.3 SA

The SA test allows the vehicle to run in 10 km/h and start to accelerate after 2s of runtime. The throttle was applied to 100% after 2s with a step of 0.1s and the test runtime was 20s. This test was run to find out
how stable the vehicle is in the straight-line under acceleration and what influence the suspension angles have on such test. The test input was set to have a free steering wheel, which provides a full study of the vehicle, without the drivers interference while accelerating.

### 3.5.4 SB

This test is fundamentally similar to the acceleration test. The SB test was set up to decelerate from *50km/h* with the brakes applied at 80% with a time step of 0.1s. In the end, it was decided that the SB test is to be excluded from the Adams Test Matrix Simulations, due to the high similarity to the SA test.

### 3.5.5 SWD

The SWD test was performed with the test standard ECE R13H [36] as a reference. The SWA for the lateral acceleration $a_y$ of $3m/s^2$ was determined from a swept steer test, which gave a baseline Amplitude (A) (see Table 3.2). The test was done with the steering amplitudes of 1.5A to 6.5A with a step of 0.5A (which equates to 10 test runs). For the Adams test matrix simulations, only the first four runs were taken, since the remaining runs did not provide any useful information (see Table 3.3). The dwell time was set to 0.5s and the steering frequency was set to 0.7Hz, which is consistent with the test standard. The outputs from this test are the maximum yaw rate $\dot{\psi}$ after the first zero crossing of SWA as well as the instant yaw rate $\dot{\psi}$ at 1s after the completion of steering. The criteria for passing the test can be found in the test standard. [36]

<table>
<thead>
<tr>
<th>Yaw rate $\psi$</th>
<th>Amplitude A</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.5</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>3</td>
<td>2.5</td>
</tr>
<tr>
<td>4</td>
<td>3</td>
</tr>
</tbody>
</table>


3.6 Adams Test Matrix Simulations

From the single parameter study in the previous section, it was decided to have the velocity excluded from the list of factors as well as excluding the SB test, in this analysis. The CLS model in Adams was run in the four tests SS, CRC, SA and SWD against an L9 test matrix, which consist of nine different suspension setups, see Table 3.4. [37]

<table>
<thead>
<tr>
<th>Suspension angles</th>
<th>Caster</th>
<th>KPI</th>
<th>Toe</th>
<th>Camber</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test/Main factor</td>
<td>A</td>
<td>B</td>
<td>C</td>
<td>D</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>1</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>5</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>6</td>
<td>2</td>
<td>3</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>7</td>
<td>3</td>
<td>1</td>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>8</td>
<td>3</td>
<td>2</td>
<td>1</td>
<td>3</td>
</tr>
<tr>
<td>9</td>
<td>3</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
</tbody>
</table>

Such approach is common in experimental design studies. The approach takes into account testing of multiple parameters in an efficient way by varying multitude of factors simultaneously and then analysing the results by distinguishing the effect and relative importance of each factor. Such philosophy shortens the test stage time without sacrificing the amount of knowledge from the result. It is possible to uncover the interactions between the factors, i.e. how suspension angles affect each other. This would require a higher resolution of the test matrix, which results in an enormous amount of tests. Therefore, an analysis of the interaction between factors will be excluded in this thesis.

The L9 test matrix consists of four main factors (camber, toe, KPI and caster) as well as three levels low (1), medium (2) and high (3). The velocity was ruled out from the test matrix simulations since it was deemed as an inappropriate main factor from a study standpoint.

The values of all four main factors in each level are presented in Table 3.5 below.
Table 3.5: The four main factors and the three levels.

<table>
<thead>
<tr>
<th></th>
<th>Low 1 Negative</th>
<th>Medium 2 Neutral</th>
<th>High 3 Positive</th>
</tr>
</thead>
<tbody>
<tr>
<td>A Caster</td>
<td>-2</td>
<td>0</td>
<td>+2</td>
</tr>
<tr>
<td>B KPI</td>
<td>+6.7</td>
<td>+9.7</td>
<td>+12.7</td>
</tr>
<tr>
<td>C Toe</td>
<td>-0.24</td>
<td>0</td>
<td>+0.24</td>
</tr>
<tr>
<td>D Camber</td>
<td>-1</td>
<td>0</td>
<td>+1</td>
</tr>
</tbody>
</table>

The front and rear setups of the four suspension angles of the bidirectional vehicle that were studied in this thesis, can be seen in the figures below. Figures 3.14, 3.15, 3.16 and 3.17 show the negative, positive and neutral setup of camber, toe, KPI and caster, respectively.

Figure 3.14: Different setups of camber from front/rear view, [38].
Figure 3.15: Different setups of toe from top view, [38].

Figure 3.16: Positive setup (negative and neutral setup excluded) of KPI from front/rear view, [25].
A high velocity investigation was done in the SS and SWD, since for those tests, the setup was velocity dependant. A higher velocity of $90 \text{ km/h}$ was selected as the new velocity test input for the SS and SWD. The two tests were run once more with full matrix investigation, for all nine configurations, now with $90 \text{ km/h}$. An investigation of increased unsprung mass was also performed, with an additional mass of $50 \text{ kg}$ on each wheel in the vehicle. The complete set of tests was run once again with full matrix investigation, for all nine configurations, in $50 \text{ km/h}$ (for SS and SWD).

It should be noted that in such investigation, it is important to select
a proper test output to be used in factor and relative importance calculations. Since the output of the test and the values dictate the importance and influence of the suspension angles on this particular output. In some sense, the output of the test dictates what importance will be shown. The selected outputs for the SS, CRC, SA and SWD tests were: maximum lateral acceleration $a_{y,\text{max}}$, steering sensitivity $\frac{\partial \psi}{\partial \delta}$, longitudinal velocity $v_x$ at which the oscillations of the steering wheel start and maximum yaw rate $\dot{\psi}_{\text{max}}$ after the first SWA zero crossing, respectively.

In order to calculate the effect of the parameters as well as the relative importance on the output of the test, it is essential to find the overall mean value of the output (see Equation (3.8)), where $\eta$ is the output variable of the test, [37]. Then, the average of the specific level is taken as illustrated in Equation (3.9). The $m_A$ is the effect of factor A on level 3. In order to find the relative importance, it is necessary to find the sum of squares per factor and the total sum of squares, see Equations (3.10) and (3.11). The relative importance of the factor was calculated as a ratio between sum of squares per factor and total sum of squares, see Equation (3.12). It is possible to predict the best output of the optimal solution $\eta_{\text{optimal}}$ of the most important factors. This can be done by implementing the formula in Equation (3.13), where the $m_A$ and $m_{B1}$ are the most important factor effects in the particular test matrix run. The output $\eta_{\text{optimal}}$ gives a good prediction on the best performing angle setup. However, $\eta_{\text{optimal}}$ will not be presented in this report. [37]

\[
m = \frac{1}{9} \sum_{n=1}^{9} \eta_i = \frac{1}{9} (\eta_1 + \eta_2 + \ldots + \eta_9) \tag{3.8}
\]

\[
m_{A3} = \frac{1}{3} (\eta_7 + \eta_8 + \eta_9) \tag{3.9}
\]

\[
\text{Total sum of squares} = \sum_{n=1}^{9} (\eta_i - m)^2 \tag{3.10}
\]
Sum of squares due to factor A

\[
3(m_{A1} - m)^2 + 3(m_{A2} - m)^2 + 3(m_{A3} - m)^2
\]  \hspace{1cm} (3.11)

Relative importance

\[
\frac{\text{sum of squares due to factor}}{\text{total sum of squares}} \cdot 100 \% \hspace{1cm} (3.12)
\]

\[
\eta_{optimal} = m + (m_{A1} - m) + (m_{B1} - m)
\]  \hspace{1cm} (3.13)
Chapter 4

Results and Discussions

The results and discussions of each study performed in this thesis are presented in this chapter.

The first section presents the results from the model verification between Matlab and Adams. The results from the kinematic analysis in 2D, the single parameter analysis and the Adams test matrix simulations are presented in Sections 4.2, 4.3 and 4.4, respectively. In the Adams test matrix simulations in Section 4.4, the simulated results of drivetrain configuration 9 and 7 had almost no difference between them and will therefore be represented by the same results. Another set of tests were run on the CLS model but with additional unsprung mass of 50 kg, which represents drivetrain configuration 3. The results of the models RK and 2CV are presented and discussed in Section 4.5.

4.1 Model Verification – Matlab vs Adams

As previously mentioned in Section 3.3, the comparison of Matlab and Adams was done on the basis of lateral acceleration \( a_y \) and yaw rate \( \dot{\psi} \) in the three tests: CRC, SS and Slalom. The iterative procedure of finding realistic values for lateral stiffness \( C \) as well as the tire friction coefficient \( \mu \) gave the results presented in Table 4.1 below:

<table>
<thead>
<tr>
<th>Coefficient</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C_{12}, C_{34} )</td>
<td>440 000 [N/rad]</td>
</tr>
<tr>
<td>( \mu )</td>
<td>0.52 [-]</td>
</tr>
</tbody>
</table>
From the figures presented in this section, it is apparent that there is a difference in results between the two softwares. It was observed that the Matlab model has a more conservative estimation of lateral acceleration $a_y$ and yaw rate $\dot{\psi}$. As seen in Figures 4.1 and 4.2, these values are lower the closer the vehicle is to losing control, in the CRC test.

![Figure 4.1: Lateral acceleration $a_y$ of the CLS in the CRC test.](image1)

![Figure 4.2: Yaw rate $\dot{\psi}$ of the CLS in the CRC test.](image2)

In the SS test, the lateral acceleration $a_y$ and yaw rate $\dot{\psi}$ from the bicycle
model do follow the overall trend of the Adams model, although they are not reaching the exact peak values (see Figures 4.3 and 4.4).

Figure 4.3: Lateral acceleration $a_y$ of the CLS in the SS test.

Figure 4.4: Yaw rate $\dot{\psi}$ of the CLS in SS test.

In the Slalom test, the bicycle model does not reach the peak values of the Adams model in lateral acceleration $a_y$ and yaw rate $\dot{\psi}$. Additionally, there is a slight phase delay, see Figures 4.5 and 4.6.
The above statements can be explained through the complexity level difference between the models. The Adams model is 3D double track, while the bicycle model is 2D single track. The present bicycle model does not account for the lateral load transfer during cornering, compliance in the suspension provided by the bushings and tires nor the gravitational component of lateral acceleration $a_y$ at high roll angles. The percentage differences in the results are at the level of $10-15\%$, which in statistical
Results and Discussions

The points mentioned above make it possible to assume that the Adams model is valid.

The understeer gradient $K_{us}$ for the vehicle was found to be zero, which is consistent with the figure presenting steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ (see Figure 4.7). As seen in the figure, the initial part of the CRC graph is following the linear shape as the analytical model does. From the classical analysis of critical velocity $v_{crit}$, the theoretical critical velocity can be found, which is represented by Equation (3.1). In the case of classical theoretical analysis, the critical velocity $v_{crit}$ goes to infinity, since the $K_{us}$ is equal to zero for a bidirectional vehicle. However, the vehicle exhibits understeering behaviour at critical lateral acceleration $a_y$ levels. The vehicle starts to understeer at $68 \text{ km/h}$, which is a value of dynamic critical velocity $v_{crit, dyn}$. The difference in values for a dynamic velocity $v_{crit, dyn}$ and purely theoretical velocity $v_{crit}$ stems from the fact that the 3D model incorporates considerable amount of complex, nonlinear and compliant joints, which fundamentally make the assessment of such critical parameter quite demanding. The lateral stiffness of the full dynamic system is not identical to the theoretical bicycle model values, which gives the difference between simulation results and theoretical calculations. The theory is an ideal case, while the simulation represents the non-ideal case.

From the Table 4.1, it can be concluded that the low tire friction coefficient $\mu$ is one of the reasons for the degraded performance of the vehicle, since truck tires have lower $\mu$ values compared to regular passenger vehicles.

$$v_{crit} = \infty,$$

$$v_{crit, dyn} = 68 \text{ km/h}.$$
4.2 Kinematic Analysis in CAD 2D

As stated in Section 3.4, the kinematic analysis was performed on the three aforementioned suspension types. The geometry for RK and 2CV was assumed on the basis of width between wheels and suspension packaging envelopes. The geometry for CLS was selected to be similar to the axle supplier specification.

As can be seen in previous Section 2.5, the different suspension parameters provide different advantages and disadvantages, depending on how they are tuned, i.e. if they are negative, neutral or positive. This kinematic analysis provides information regarding the behaviour of the different parameters for the different suspension models.

The camber angle provides better control of the contact patch shape, during cornering, as well as additional lateral force. However, high camber angle values increase uneven tire wear. Adding a toe angle to the vehicle provides great stability when driving on a straight road, however, it becomes unstable when turning in corners. The KPI together with the scrub radius provides better returnability for the steering and decreases the tire wear, when tuned in positive values. The caster and mechanical trail affect the steering in the same way, however, from a different plane.
(i.e. the side view of the vehicle). It requires more effort for steering, the higher the values of these parameters are. It is preferred to have the RC as close as possible to the COG of the vehicle, since it results in less force pushing the vehicle to roll. However, when moving the RC closer to the COG, in order to decrease the roll force, a jacking force on the vehicle increases. The roll axis was also evaluated in this analysis. It can be seen as parallel to the ground (seen from side view) for a bidirectional vehicle, compared to asymmetrical vehicles where the roll axis is tilted/angled.

In suspension design, the RC heights, Instantaneous Centre (IC) points and roll axis position, influence the roll stiffness $K_\phi$ and jacking forces in the suspension. The procedure for general roll analysis is to find the distance between the COG position and RC, then make conclusions on the length of the moment arm as well as movement of the RC, compared to initial position. The rule of thumb in this analysis is: the shorter the moment arm between COG position and RC, the lower the rolling moment. In turn, this implies that the vehicle is harder to roll (meaning higher roll stiffness $K_\phi$ in the suspension). The IC of rotation and its height from the ground determine the jacking forces coming in the suspension arms, during cornering. The jacking forces are not mitigated by the spring but are tackled in suspension arms and mounts. [40]

It should be highlighted, that in the roll analysis, high values of scrub radius come from the way how the contact patch is modelled and the assumptions made for its motion. The scrub radius affects the steering effort, which in turn, directly affects the sizing of the steering system. The real behaviour of the contact patch and changes in scrub radius are considerably more complicated in the dynamic motion of the vehicle, since the tire deformation and suspension compliance are hard to replicate in the simulation environment. Nevertheless, such simplified 2D analysis provides a comparative estimation of scrub radius changes in different systems.

The static values as well as the results of the kinematic analysis of all three suspension models are presented in Tables 4.2, 4.3 and 4.4 below:
### Table 4.2: Static values of the CLS, RK and 2CV.

<table>
<thead>
<tr>
<th></th>
<th>Camber angle [°]</th>
<th>Toe angle [°]</th>
<th>KPI [°]</th>
<th>Caster angle [°]</th>
<th>Scrub radius [mm]</th>
<th>Mechanical trail [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>CLS</strong></td>
<td>0</td>
<td>0</td>
<td>9.7</td>
<td>2</td>
<td>100.75</td>
<td>17.81</td>
</tr>
<tr>
<td><strong>RK</strong></td>
<td>0.3</td>
<td>0</td>
<td>0.3</td>
<td>0</td>
<td>95.19</td>
<td>0</td>
</tr>
<tr>
<td><strong>2CV</strong></td>
<td>0</td>
<td>0</td>
<td>0.1</td>
<td>0</td>
<td>30</td>
<td>16.63</td>
</tr>
</tbody>
</table>

### Table 4.3: Parallel wheel travel of ±80mm of the CLS, RK and 2CV.

<table>
<thead>
<tr>
<th></th>
<th>Camber angle [°]</th>
<th>KPI [°]</th>
<th>Caster angle [°]</th>
<th>Scrub radius [mm]</th>
<th>Mechanical trail [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>CLS</strong></td>
<td>-0.363</td>
<td>+0.7 - -0.54</td>
<td>+10.07</td>
<td>+10.00</td>
<td>-194.79</td>
</tr>
<tr>
<td><strong>RK</strong></td>
<td>+1.3</td>
<td>+1.21 - +1.56</td>
<td>+13.01</td>
<td>-108.75</td>
<td>+275.7</td>
</tr>
<tr>
<td><strong>2CV</strong></td>
<td>0</td>
<td>?</td>
<td>0</td>
<td>30</td>
<td>+62.346</td>
</tr>
</tbody>
</table>

Values from -80 mm to +80 mm.

### Table 4.4: Roll investigation of the CLS and RK.

<table>
<thead>
<tr>
<th></th>
<th>Camber angle [°]</th>
<th>KPI [°]</th>
<th>Caster angle [°]</th>
<th>Scrub radius [mm]</th>
<th>Mechanical trail [mm]</th>
<th>Roll centre height [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>CLS</strong></td>
<td>4.39</td>
<td>3.1</td>
<td>4.1</td>
<td>135.6</td>
<td>70.2</td>
<td>-28.57</td>
</tr>
<tr>
<td><strong>RK</strong></td>
<td>4.1</td>
<td>3.1</td>
<td>?</td>
<td>194.79</td>
<td>20.2</td>
<td>-275.7</td>
</tr>
</tbody>
</table>

4.2.1 **CLS**

The CLS suspension type is a variation of a common DWB suspension. Such suspension type takes pride in improved wheel control, high configurability as well as relatively simple design. Compared to the other two suspension types in this thesis, the CLS is more expensive.

During compression and rebound motion, the transverse leaf spring DWB suspension does not exhibit the same behaviour as the RK. The CLS suspension type controls the wheel movement to follow an arc shape without the negative to positive camber angle sign change, see Table 4.3. This fact signifies that in this particular suspension setup, the inner wheel during cornering will have a negative camber gain, which will negatively affect the cornering performance (camber force directed to the outside of the corner).
The RC in the CLS suspension is situated at the ground level. This is due to the upper and lower control arms being set up parallel to each other in static condition. This indicates that the roll moment arm length is similar to the 2CV suspension type, which results in high roll susceptibility. It should be noted that theoretically, it is possible to account for roll stiffness $K_\phi$ with smart suspension design. The addition of anti-roll bar greatly increases the roll stiffness $K_\phi$. During the roll motion with 5° on the chassis, the RC point moves 28.5 mm below ground, see Table 4.4. This indicates that such suspension setup loses some of its roll stiffness $K_\phi$ during motion. Nonetheless, this suspension type will not introduce unwanted excitations to the vehicle during high dynamic manoeuvres, since the RC point does not have as great change as in the RK suspension. Similarly to the 2CV, the roll moment will be quite high because of moment arm length. The high change of scrub radius during the roll motion partly comes from the assumptions made in Section 3.4 and partly from the kinematic setup of the CLS suspension with the movement of the wheels in the perfect arc. The KPI and camber angles change in accordance with wheel movement, see Table 4.4.

The CLS roll stiffness $K_{\phi, CLS}$ is presented below:

$$K_{\phi, CLS} = \left(\frac{s}{2}\right)^2 \cdot \left(\frac{f_F}{f_L}\right)^2 = 647530 \text{ Nm/rad},$$

Where,

$$f_F = 581 \text{ mm},$$
$$f_L = 892 \text{ mm},$$
$$k = 511.88 \text{ N/mm},$$
$$s = 2442 \text{ mm}.$$
\[ f_{\text{loaded, CLS}} = 0.677 \text{ Hz}, \]
\[ f_{\text{unloaded, CLS}} = 1.105 \text{ Hz}. \]

The ride frequencies indicate that the presented CLS suspension will be similar in ride quality to the RK in both loaded and unloaded states.

The IC position of the control arms changes with the movement in the suspension and body. In the initial static position, the CLS lacks the IC point, because of the above stated relation between the control arms. Nevertheless, the IC point becomes apparent when the suspension undergoes motion. In the up/down wheel test, the IC point location moves below ground with a position of 278.1\text{mm} in up motion and 380.8\text{mm} below ground in down motion. In the rolling test, the IC point coordinate is 119.2\text{mm} above ground. Such results indicate that there will be apparent jacking forces in the mounting points and both arms of the suspension.

The caster gain will not be seen in this suspension type, due to the lack of pivot point in side view (no intersection point for the parallel mounted suspension arms, see Assumption 1 in Section 3.4.1).

The scrub radius for the CLS has a higher value compared to RK. This is an inherent difference between the two suspension types, see Table 4.2.

Since the presented CLS geometry does not change the camber angle sign during suspension motion and the wheel is moving in a perfect arc, it is valid to connect these phenomena to a low toe angle variation. It should be noted that it was attempted to decrease the bump steer in this suspension type by having similar lengths between the tie rod and lower control arm, hence, the low toe variation, see Table 4.3.

It should also be noted that the DWB is one of the most configurable suspension types out of the three selected suspensions. It is possible to minimise the roll movement and replicate the favourable camber change values, as in the RK. This can be accomplished with the change of control arm lengths, mounting points on the chassis and height of the upright (angling of the control arms).
Anti-feature mitigation in heavy vehicles with this suspension type can be carried out similarly to the RK.

4.2.2 RK

The RK suspension type is intended to be an improvement of the MacPherson type. The RK is supposed to have a better vehicle dynamics performance in torque steer, wheelflight, steering nibble and brake judder compared to the MacPherson suspension, yet still retaining the same package [18]. Lower scrub radius in this suspension type, compared to MacPherson, is achieved with addition of two more joints in the upright area.

The compression and rebound characteristics of the RK suspension are similar to the MacPherson type. The corollary being that during motion, some suspension angles will change their sign because of the wheel “tucking in” towards the chassis at the compression stroke (negative camber) and “flaring out” at the rebound stroke (positive camber). A good example of this phenomenon is the dynamic camber angle sign change in up/down motion, see Table 4.3. Such sign change is beneficial in improving cornering behaviour, since the inside wheel in the corner will be in rebound with positive camber (force towards the inside of the curve) and outside wheel will be in compression stroke with negative camber (force towards the inside of the curve).

The RC for the RK suspension at static position is situated 66.32mm above ground level. This means that this suspension type will have a shorter roll moment arm, which in turn means higher roll stiffness $K_\phi$ compared to ground level RC. Moreover, during the roll motion of the body to $5^\circ$, the RC increases its height to 275.7mm. In turn, this means that the more the vehicle rolls with this suspension type, the higher the roll stiffness $K_\phi$ (which is beneficial in cornering behaviour). Nonetheless, considerable movement in the RC during high dynamic manoeuvres can upset the balance of the vehicle and cause unpredictable behaviour, see Table 4.4. As stated before, the high scrub radius change in roll motion partly comes from the assumptions made in Section 3.4 and partly from the RK wheel movement peculiarities. Therefore, the wheel will change the camber angle sign during motion. The change of the KPI and camber
angles is not as severe as in the CLS roll motion, since the KPI axis is not tilting as much, see Table 4.4.

The RK requires similar considerations to finding the correct tie rod positions to mitigate bump steer. The lower control arms are shorter and finding the correct steering geometry to avoid bump steer is less complicated of a task than in the 2CV. The bump steer issue is not as severe nor does it consume nearly as much useful space, as in the 2CV.

The RK roll stiffness $K_{\phi, \text{RK}}$ is presented below:

$$K_{\phi, \text{RK}} = \left( \frac{b \cdot d}{a} \right)^2 \cdot k = 942920 \text{ Nm/rad},$$

where,

$$b = 3434.28 \text{ mm},$$
$$d = 1222.864 \text{ mm},$$
$$k = \frac{356.63}{2} \text{ N/mm},$$
$$a = 3652.6 \text{ mm}.$$

The loaded and unloaded ride frequencies can be seen below:

$$f_{\text{loaded, RK}} = 0.746 \text{ Hz},$$
$$f_{\text{unloaded, RK}} = 1.218 \text{ Hz}.$$

The ride frequencies indicate that the presented RK suspension is managing the drastic load change very well. At full load, the ride will be quite smooth and plush, whereas with no load, the ride will still be acceptable and similar to a regular passenger car.

As stated in the previous section, the IC moves with the movement of the suspension. During the wheel upward movement, the IC height changes from 198.64 mm to 330.62 mm. During the roll motion, the IC height from the ground equates to 508.18 mm. Such drastic changes in IC height,
together with a high initial value, will put additional jacking forces into the suspension arms and struts. This calls for stronger mounting points and thicker control arms.

The implementation of anti-features in suspension design of heavy vehicles has limited use, since the loads in such vehicles are high and anti-features put additional strain on the suspension arms. However, these features are still implemented in a more advanced way, with semi-active and active suspension designs. In the case of RK, the implementation of active damper design will be sufficient to provide the anti-feature benefits, without sacrificing the control arm strength optimisation.

As previously mentioned, the geometry for this suspension type was assumed on the basis of width between the wheels and the packaging envelope.

The design flexibility of this suspension type is marginally better than the 2CV, since the restrictions on the strut mounting points are a considerable design consideration. The adjustability of camber and toe gain gives some design freedom for vehicle dynamics. The benefits of the RK over MacPherson cannot be overlooked.

4.2.3 2CV

From Tables 4.2 and 4.3, it is apparent that the 2CV suspension type has excessive caster gain during the up and down wheel motion. In turn, this elongates the wheelbase of the vehicle during load cases, braking, acceleration and cornering. The increase in mechanical trail should give the benefit of steering returnability as well as straight-line stability with the increased vehicle loading.

This suspension system is a variation of a swing arm suspension, which in turn makes certain assumptions valid:

- The wheel travel in compression and rebound is completely vertical, in the frontal projection.
- The wheel travels in an arc, from a side view.
- The KPI and scrub radius are engineered into the suspension and are constant, i.e. no change in up/down motion.
• The change in camber angle only occurs during roll motion and is equal to the roll angle of the chassis. (Wheels tilt with consistent deviation of the angle dependency to the chassis).

With the above assumptions, it is valid to conclude that in dynamic manoeuvres (which do not include rolling motion), the 2CV suspension does not have any change in camber, KPI nor scrub radius. The tie rod connection in the 2CV suspension is of prime importance to the toe angle change during compression and rebound (bump steer). The 2CV is the most sensitive suspension system to the position of the tie rod in the steering system. Usually, in order to avoid bump steer, it is beneficial to have the tie rods with similar length to the lower control arm and as close as possible to its pivot point. This ensures that the tie rod and the lower control arm follow the same curvature during travel. Since the 2CV has only one swing arm, it results in an abnormally long tie rod to mitigate bumpsteer. In turn, it limits the flexibility of suspension mounting points and takes considerable space for the steering system (the tie rods are connected in the centre of the axle).

The RC for a 2CV suspension is situated at the ground level. This implies that there is a considerable distance between the COG position and the RC of the suspension. This fact indicates that roll moment arm in the suspension will be quite long.

The 2CV roll stiffness $K_{\phi, 2CV}$ is presented below:

$$K_{\phi, 2CV} = \left( \frac{s}{2} \right)^2 \cdot \left( \frac{f_F}{f_L} \right)^2 = 3767300 \ Nm/rad,$$

where,

$$f_F = 70 \ mm,$$
$$f_L = 650 \ mm,$$
$$k = 108940 \ N/mm,$$
$$s = 2442 \ mm.$$
It should be noted that some approximations were made in the calculations for the 2CV suspension. In order to achieve the same roll stiffness as the CLS, the 2CV requires considerably more stiff springs. This is dictated by the dimension of the bellcrank arm, since this dimension cannot be extended indefinitely (according to the packaging considerations).

The loaded and unloaded ride frequencies can be seen below:

\[
\begin{align*}
  f_{\text{loaded, } 2CV} &= 1.633 \text{ Hz}, \\
  f_{\text{unloaded, } 2CV} &= 2.66 \text{ Hz}.
\end{align*}
\]

The ride frequencies indicate that the presented 2CV suspension is quite harshly sprung, in order to cope with considerable amount of load. This indicates that in the unloaded state the ride will be harsh. On the contrary, the ride at full load will be closer to a regular passenger car. This drastic change in ride harshness can be solved by implementing progressive non-linear springs, in order to manage the high change in load and to provide decent ride quality in unloaded state.

The pitch motion during braking and acceleration manoeuvres can be taken care of with front/rear suspension coupling, in the same way as in the original 2CV suspension system. If done correctly, this eliminates the need to engineer anti-features in this suspension, which keeps the mounting points for the swing arm in reasonable places.

The 2CV suspension has considerable restrictions in component placement, since this suspension type is relatively simple in construction. This means that the tie rod locations, scrub radius, KPI as well as mechanical trail increase should be decided in the design phase beforehand. The simplicity of this suspension type makes it difficult to have decent flexibility in component placement. For example, in terms of tie rod mounting points, there is only one option to make the tie rods as long as possible and mount them as close as possible to the centre of the vehicle. This arrangement takes a large amount of useful space that can be used by other components.

The scrub radius in the 2CV is determined by the design of the suspension
and the wheel used with the design. For this suspension type, the scrub radius and KPI do not change with the up/down motion, see Tables 4.2 and 4.3. Roll motion, however, changes these parameters in accordance to the chassis roll angle.

### 4.3 Single Parameter Study

The four tests that were run in this study had different outputs where each of them had a certain value in focus; either maximum or steady values. The maximum values in the tests were taken as the peak values of the graphs, except for in the CRC test, where the value of SWA was taken at 10s after the beginning of the test and steering sensitivity $\frac{\partial \psi}{\partial \delta}$ was taken at $65\text{km/h}$. The overall results show that there is no clear dominant setup, since the angles behave differently in all four manoeuvres.

An example of a results selection in the SWD test is presented in Figure 4.8. The output of this test was selected as maximum physical value of yaw rate $\dot{\psi}$ after the first peak (red circle). The other possible outputs of the yaw rate $\dot{\psi}$ are represented as first peak of the parameter (purple circle) and yaw rate $\dot{\psi}$ at 3s of time (blue circle). The steering amplitude A to yaw rate $\dot{\psi}$ relation can be seen in Table 3.3.

![Figure 4.8: Example of SWD output as yaw rate $\dot{\psi}$.](image)

Tables 4.5, 4.6 and 4.7 show the output results between the bidirectional vs the non-bidirectional vehicle, while Table 4.8 displays the order of the three compared vehicle setups (in the SS and CRC test), with a ranking of lowest to highest values for each suspension angle.
It is important to keep in mind that there are more suspension angle setup combinations that can be investigated, however, they have been excluded in this study. For instance, the non-bidirectional vehicle can have another opposite setup, e.g. it can have negative toe in front and rear instead of positive in both ends, which was studied here.

Table 4.5: Output results from the SS test.

<table>
<thead>
<tr>
<th>Camber</th>
<th>Toe</th>
<th>KPI</th>
<th>Caster</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bidirectional</td>
<td>17.39</td>
<td>17.97</td>
<td>17.39</td>
</tr>
<tr>
<td>Non bidirectional</td>
<td>17.45</td>
<td>17.55</td>
<td>17.51</td>
</tr>
<tr>
<td>Max difference</td>
<td>0.06</td>
<td>0.42</td>
<td>0.12</td>
</tr>
<tr>
<td>SS (maximum values)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lateral acceleration $a_y$</td>
<td>4.09</td>
<td>4.22</td>
<td>4.08</td>
</tr>
<tr>
<td>Max difference</td>
<td>4.1</td>
<td>4.15</td>
<td>4.1</td>
</tr>
</tbody>
</table>

Table 4.6: Output results from the CRC test.

<table>
<thead>
<tr>
<th>Camber</th>
<th>Toe</th>
<th>KPI</th>
<th>Caster</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bidirectional</td>
<td>21.93</td>
<td>20.16</td>
<td>21.8</td>
</tr>
<tr>
<td>Non bidirectional</td>
<td>21.6</td>
<td>21.52</td>
<td>21.72</td>
</tr>
<tr>
<td>Max difference</td>
<td>0.33</td>
<td>1.36</td>
<td>0.08</td>
</tr>
<tr>
<td>CRC (steady values)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SWA</td>
<td>10.51</td>
<td>11.62</td>
<td>10.7</td>
</tr>
<tr>
<td>Max difference</td>
<td>10.8</td>
<td>10.73</td>
<td>10.64</td>
</tr>
<tr>
<td>Steering sensitivity $\frac{\partial \psi}{\partial \delta}$</td>
<td>0.29</td>
<td>0.89</td>
<td>0.06</td>
</tr>
</tbody>
</table>

Table 4.7: Output results from the SWD test.

<table>
<thead>
<tr>
<th>Camber</th>
<th>Toe</th>
<th>KPI</th>
<th>Caster</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max difference</td>
<td>-0.06</td>
<td>-0.3</td>
<td>-0.14</td>
</tr>
<tr>
<td>SWD (maximum values)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_1$</td>
<td>23.78</td>
<td>23.79</td>
<td>23.74</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_2$</td>
<td>-23.78</td>
<td>-23.79</td>
<td>-23.8</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_3$</td>
<td>-21.96</td>
<td>-21.94</td>
<td>-21.96</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_4$</td>
<td>-20.34</td>
<td>-20.37</td>
<td>-20.34</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_5$</td>
<td>-23.8</td>
<td>-23.82</td>
<td>-23.8</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_6$</td>
<td>-23.78</td>
<td>-23.79</td>
<td>-23.74</td>
</tr>
<tr>
<td>Max difference</td>
<td>0.08</td>
<td>0.02</td>
<td>0.23</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Camber</th>
<th>Toe</th>
<th>KPI</th>
<th>Caster</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bidirectional</td>
<td>32.74</td>
<td>32.85</td>
<td>32.85</td>
</tr>
<tr>
<td>Non bidirectional</td>
<td>32.79</td>
<td>33.07</td>
<td>33.07</td>
</tr>
<tr>
<td>Max difference</td>
<td>0.05</td>
<td>0.22</td>
<td>0.22</td>
</tr>
<tr>
<td>SWD (maximum values)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_1$</td>
<td>36.41</td>
<td>36.41</td>
<td>36.41</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_2$</td>
<td>36.42</td>
<td>36.42</td>
<td>36.42</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_3$</td>
<td>36.44</td>
<td>36.44</td>
<td>36.44</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_4$</td>
<td>36.44</td>
<td>36.44</td>
<td>36.44</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_5$</td>
<td>36.42</td>
<td>36.42</td>
<td>36.42</td>
</tr>
<tr>
<td>Yaw rate $\dot{\psi}_6$</td>
<td>36.44</td>
<td>36.44</td>
<td>36.44</td>
</tr>
<tr>
<td>Max difference</td>
<td>0.01</td>
<td>0.25</td>
<td>0.25</td>
</tr>
</tbody>
</table>
As can be seen in Figure 4.9, the lateral acceleration $a_y$ is almost identical for all the settings in the single parameter study. This comes from the fact that the vehicle drives on the same circle radius every time, with the same longitudinal acceleration. However, the steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ is different for each setting, meaning that the vehicle characteristics change with regard to suspension angles, see Figure 4.10.
As a general statement, it is reasonable to imply that the camber angle has little to no impact on the test results in the single parameter study. However, camber provides a drastic increase in steering force in the rear with a non-bidirectional setup in the SA test. This means a negative camber setting in the rear is not favourable for a non-bidirectional setup. The caster angle as well as the toe angle have considerable prevalence across all three studies (single parameter, bidirectionality, test matrix). The interactions between the angles can improve the overall vehicle performance, compared to optimising one suspension angle at a time without interactions. However, the importance of the angles does not change in case of interactions, i.e. the caster angle will still be the prevalent angle in the suspension design. Then again, the variation
of caster together with KPI and camber can yield a more optimised suspension setup or compromise an already well performing caster setup.

The toe angle exhibits greater influence in manoeuvres with steering inputs. This is connected to the principle of adding the toe angle in the suspension setup. When the static toe angle is added, it directly affects the slip angle $\alpha$ on the wheels, which generates additional forces in the tires. Excessive static toe angle can adversely influence the wear of the tires. Camber also generates lateral forces, through reaction force induced by the lean angle, but to a far lesser extent than toe angle. The camber angle is the only suspension angle that increases the steering force in the rear for a non-bidirectional vehicle. On the other hand, the toe angle significantly decreases the steering force in the rear in a non-bidirectional setup. This can decrease the use of energy in the rear steering system, when driving in straight-line.

4.3.1 SS and CRC

As can be seen in the results from the SS and CRC manoeuvres (see Tables 4.5, 4.6 and 4.8), the order of the angles with highest difference does not really show a clear pattern. Overall, camber seems to be the least changing angle of almost all the outputs studied and seems to have least importance of all the suspension angles. Camber also shows that a bidirectional setup is the one giving the lowest output value when looking at the results, except for in the SWA, where a non-bidirectional setup has the lowest angle. When it comes to the CRC tests, the angle influence on the results is still dominated by caster and toe. However, caster setup has more influence in non-bidirectional setups, whereas toe has more influence in symmetrical bidirectional cases. It is apparent that the caster setup is more sensitive to symmetry than toe in the CRC tests.

KPI has also quite a low change and has more mixed results than camber when looking at the lowest influence setup. KPI shows that the setup with lowest values for yaw rate $\dot{\psi}$ and lateral acceleration $a_y$ is bidirectional, whereas SWA and steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ has lowest values in non-bidirectional and neutral setup, respectively.

As stated in the theory, a positive/negative toe angle setup is most suitable for straight-line driving, whereas a more neutral setup is
preferred when cornering. As can be seen in Tables 4.5, 4.6 and 4.8, the lowest values in the outputs are in fact produced by the neutral setup, followed up by the non-bidirectional vehicle and lastly the bidirectional vehicle. Except for the SWA, where the order is the other way around, i.e. bidirectional, non-bidirectional and neutral.

The caster angle shows, overall, the highest difference in both tests and their respective outputs. As for KPI, it is more mixed results when looking at the highest output setup. Bidirectional setup gives the lowest yaw rate $\dot{\psi}$ and steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$, whereas a neutral setup produces the lowest lateral acceleration $a_y$. The non-bidirectional setup gives the lowest SWA.

What is interesting is that the positive front and negative rear caster setup has lower yaw rate $\dot{\psi}$ and steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ than the neutral setup. This might imply that zero caster is not necessarily equal to optimal caster setup (according to these manoeuvres). However, when studying the lateral acceleration $a_y$, zero caster angle has the lowest result.

4.3.2 SA

In comparison to the other tests in this thesis, the SA test was not as successful. As seen in Figure 4.11, the vehicle starts to oscillate while accelerating on a straight road. The SWA and steering force are proportional and this is inferred from the test setup, where the steering wheel input is “free”. Meaning that the steering wheel does not have any force input from the driver and the oscillations on the steering wheel are purely from driving straight ahead.
A closer investigation of the CLS Adams model as well as the SA test setup was done and it was found that the oscillations in the system can be induced by the numerical error in the software. This means that the oscillations are appearing from the marginal numerical error between the left side wheels and the right side wheels, with the given conditions: no road imperfections, ideal wheels and symmetric suspension geometry. This leads to the oscillations appearing only from a growing error of the software numerical algorithm. It follows that the test setup should be reworked and redefined in order to provide a valid comparison of suspension angles. The result that the negative caster setup is the most stable goes against the encountered theoretical background in the literature. Additionally, in order to receive more consistent results in the SA test, the steering system should be adjusted to give identical performance for each caster setting. The suggestions on how to rework the test setup can be found in Section 5.2.

### 4.3.3 SWD

The overall behaviour of the vehicle in the SWD tests resembles considerable non-linearity. This is illustrated with the vehicle failing a set of tests with amplitude 2A and passing the tests with amplitudes 1.5A and 2.5A, see Figure 4.12. The test passed at amplitude 3A in the single parameter study for camber and KPI changes and failed for caster and toe. Similar results appear in the bidirectionality study, where caster and toe angle changes fail the test at 3A. Failure of the test run at 2A
indicates a non-linear vehicle behaviour, which can be connected to the bidirectional nature of the vehicle as well as simplistic control of the 4WS system. The tests with amplitude 3.5A and above were failed by the first condition of SWD test standard (35% or less of peak yaw rate $\dot{\psi}$ at 3s). The toe angle gives the biggest change in yaw rate $\dot{\psi}$ across most of the amplitudes, which can be connected to the principle of the toe angle setup.

![Figure 4.12: Yaw rate $\dot{\psi}$ (1.5 A to 4A) of the SWD test for KPI 12.7°.](image)

When comparing the bidirectional and non-bidirectional setup, the toe and caster have the biggest influence on the results. In a non-bidirectional vehicle, the caster angle affects the test results the most at an amplitude of 1.5A. The effect of caster gradually becomes less important with the increase of steering amplitude, while maintaining level of high influence. On the contrary, the toe angle gradually gains influence on the results in a non-bidirectional vehicle, see Table 4.7.

The single parameter study also resulted in excluding the SB test in the Adams Test Matrix Simulations as well as only including the four suspension angles as the main factors in the L9 test matrix.

## 4.4 Adams Test Matrix Simulations

As previously mentioned in Chapter 3, the Adams test matrix simulations were performed exclusively on the CLS. The complete set of Adams simulations were performed twice, where the second set of tests studied the bidirectional vehicle with additional unsprung mass of 50kg. The
results presented below consist of the main factor relative importance in a particular manoeuvre. The interaction between the angles was not investigated, which might have a considerable impact on the results, i.e. the change of an insignificant angle can affect the other main factors. The proposed best performing setups for each test are presented in the following Subsections 4.4.1, 4.4.2, 4.4.3, 4.4.4 and 4.4.5.

4.4.1 SS 50km/h

The output studied in this test is maximum lateral acceleration $a_y$. The toe angle is the most dominant in this test, followed up by the caster angle (see Figure 4.13). The importance of camber and KPI can be considered as insignificant.

Since the toe angle can add slip angle $\alpha$ directly to the tire, the significance of it in the SS test is expected at such velocity. The best performing toe setup is positive front and negative rear.

![STEP STEER 50 km/h 2nd gear graph]

Figure 4.13: SS at 50km/h.

4.4.2 SS 90km/h

By increasing the velocity, the order of importance of the suspension angles changed. The most dominant angle is now caster, followed up by toe, KPI and camber (see Figure 4.14). The vehicle response at higher velocities relies on caster, whereas at lower velocities it relies on toe. Looking back on the assumptions on bidirectionality in Section 2.1, it
can be seen that the caster truly is the most important angle to have an accurate setup of (at least in higher velocities in SS like manoeuvres). The best performing setup for toe and caster is neutral setup, whereas KPI is set as low positive in both front and rear.

![Graph of STEP STEER 90 km/h 5th gear](image)

**Figure 4.14: SS at 90km/h.**

### 4.4.3 CRC

The output studied in this test is the steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ plotted over the longitudinal velocity $v_x$ of the vehicle. The measure for steering sensitivity $\frac{\partial \dot{\psi}}{\partial \delta}$ is taken at the point right before the vehicle loses control, which is at 65 km/h. The caster angle is of highest importance, followed up by toe (see Figure 4.15). The KPI and camber angle has minor affect in this test.

As previously mentioned, the vehicle nature does not change with the angle setup. The angle setup can only enhance or compromise the vehicle performance, without changing the overall behaviour (i.e. oversteered becomes oversteered regardless). The best performing caster setup is negative caster front and positive rear, whereas the best performing toe setup is positive toe front and negative rear.
4.4.4 SWD 50km/h

The output studied in this test is the maximum yaw rate $\dot{\psi}_{max}$, after the first 0.7s of steering input. The tests with amplitudes of 3.5A and above are failed everywhere, i.e. for all configurations in all SWD tests at all velocities. This resulted in only the first four runs (1.5A - 3A) being taken as a meaningful result. However, it was noted that the second run (at 2A) failed for all configurations, while the fourth run (at 3A) failed for a couple configurations only (test setup 5 and 7), see Table 3.4 in Section 3.6. Across the investigated runs, the toe angle is the most dominant (see Figure 4.16). One can also see that the caster angle increases its importance together with the increasing steering amplitude. Camber and KPI change does not resemble any kind of pattern. However, the camber angle gains importance at higher steering amplitudes, in comparison of having no importance at lower steering amplitudes.
The best recommended caster and toe setups for the SWD test are shown in Table 4.9. Negative, neutral and positive in Table 4.9 are referring to the angle setups in the L9 test matrix in Section 3.6.

Table 4.9: Best recommended setups for caster and toe in SWD at 50 km/h.

<table>
<thead>
<tr>
<th>The amplitude $A$ of the test</th>
<th>Toe front</th>
<th>Toe rear</th>
<th>Caster front</th>
<th>Caster rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>Positive</td>
<td>Negative</td>
<td>Neutral</td>
<td>Neutral</td>
</tr>
<tr>
<td>2</td>
<td>Negative</td>
<td>Positive</td>
<td>Positive</td>
<td>Negative</td>
</tr>
<tr>
<td>2.5</td>
<td>Negative</td>
<td>Positive</td>
<td>Neutral</td>
<td>Neutral</td>
</tr>
<tr>
<td>3*</td>
<td>Positive</td>
<td>Negative</td>
<td>Positive</td>
<td>Negative</td>
</tr>
</tbody>
</table>

* Failed test.

If the maximum yaw rate $\dot{\psi}_{max}$ would be studied at a different test time, e.g. 3s, the relationship of importance of the angles will change (see Figure 4.17). This change in the output of the test will result in finding the angle importance, in order to pass the SWD test. However, this study will not be discussed in this report.
4.4.5 SWD 90km/h

Equivalent to Section 4.4.2; increasing the velocity changed the order of importance of the suspension angles. The toe angle started as the lowest importance angle in the first run, however, it increases massively and becomes the most dominant angle with the increasing steering amplitude (see Figure 4.18). On the contrary, caster has the highest importance in low steering amplitude and decreases together with the increasing steering amplitude. Camber and KPI has no resembling pattern, however, the order of importance is in opposite behaviour compared to the SWD in 50km/h. In other words, the camber angle has some initial significance at the low steering amplitude and high velocity tests, however, it loses the significance at higher steering amplitudes in high velocity. At high amplitude and high velocity, the camber angle has higher significance compared to caster and KPI.
Negative, neutral and positive in Table 4.10 are referred to the angle setups in the test matrix, see Section 3.6.

Table 4.10: Best recommended setups for all four suspension angles in SWD at 90 km/h.

<table>
<thead>
<tr>
<th>The amplitude A of the test</th>
<th>Camber front</th>
<th>Camber rear</th>
<th>Toe front</th>
<th>Toe rear</th>
<th>KPI front</th>
<th>KPI rear</th>
<th>Caster front</th>
<th>Caster rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>Neutral</td>
<td>Neutral</td>
<td>Positive</td>
<td>Negative</td>
<td>Low positive</td>
<td>Low positive</td>
<td>Positive</td>
<td>Negative</td>
</tr>
<tr>
<td>2</td>
<td>Positive</td>
<td>Positive</td>
<td>Positive</td>
<td>Negative</td>
<td>Medium positive</td>
<td>Medium positive</td>
<td>Positive</td>
<td>Negative</td>
</tr>
<tr>
<td>2.5</td>
<td>Positive</td>
<td>Positive</td>
<td>Positive</td>
<td>Negative</td>
<td>Medium positive</td>
<td>Medium positive</td>
<td>Positive</td>
<td>Negative</td>
</tr>
<tr>
<td>3</td>
<td>Positive</td>
<td>Positive</td>
<td>Positive</td>
<td>Negative</td>
<td>Medium positive</td>
<td>Medium positive</td>
<td>Positive</td>
<td>Negative</td>
</tr>
</tbody>
</table>

* KPI does only take positive setups in this study, with the levels of low positive (1), medium positive (2) and high positive (3).

4.4.6 Increased Unsprung Mass of 50 kg

The increased unsprung mass of 50 kg did not affect the importance of the angles. The relative importance charts display the same general behaviour, with minor differences in the displayed importance values. Figures 4.19 and 4.20 below present the increased unsprung mass relative importance charts:
Results and Discussions

Figure 4.19: Relative importance charts of increased unsprung mass of 50kg in SS and CRC.

Figure 4.20: Relative importance charts of increased unsprung mass of 50kg in SWD.

4.5 RK and 2CV

With the presented results in Sections 4.2, 4.3 and 4.4, it is fair to assume that with the CLS suspension system investigated in 2D analysis and in multi-body 3D analysis, the gathered results are sufficient to make analytical discussions on the other two suspensions systems; RK and 2CV. The camber gain of CLS is relatively small as well as no caster
gain occurs in up/down motion, as inferred from the 2D analysis. These facts provide sufficient confidence to assume that the static setup of angles in the CLS and results from its static setup, can be applied to other suspension types. This means that the static suspension angle importance and influence do not change between the suspension types.

As shown in previous sections, the importance of caster and toe angles in a bidirectional vehicle implies that these angles need particular attention in the suspension system. The 2CV exhibits one characteristic of changing the caster angle and wheelbase, depending on the load of the vehicle. This characteristic is a significant benefit for the bidirectional vehicle and the 2CV suspension type can exploit this phenomena. Perhaps, the 2CV can even implement an active caster adjustment mechanism to aid the vehicle in straight-line driving and cornering manoeuvres. It should be noted that the 2CV suspension has a few major weaknesses: the requirement for considerably stiffer springs (compared to the other two systems), poor expected ride quality with high load variation on the chassis, not space efficient due to sensitive positioning of tie rods as well as lower expected roll stiffness $K_\phi$ with similar spring rates. All the mentioned weaknesses can sway the choice of suspension system away from the 2CV.

From the results presented in Sections 4.2, 4.3 and 4.4, it is clear that the zero setting of the KPI angle in RK and 2CV systems is unrealistic. It is beneficial to have some positive setting in the KPI angle. The positive KPI angle together with a reasonable scrub radius and adequate caster setting, provides the benefits of returnability of the steering system to centre position, higher straight-line stability and useful feedback force to the steering system. The complicated part in the KPI setting is finding the right level of the angle, while satisfying the efficiency and force requirements for the steering system. Additionally, the KPI angle can influence other angles in the suspension in a positive or negative way, while simultaneously not having a lot of direct apparent influence on the overall result on its own.

The toe angle has a great impact on the vehicle cornering ability as well as stability in the straight-line, with no exception of vehicles with RK or 2CV suspension types. The toe angle change depends on the tie rods in both length and position, especially in the 2CV. It was found out
that by placing the tie rods inner ends towards the centre of the steering rack, it resulted in lower toe gain in the 2CV. It is possible to find and implement an adequate static toe angle for the RK and 2CV, however, an active toe is preferred for achieving finer performance.

As illustrated in the Adams test matrix simulation investigation (see Section 4.4), the camber angle has lower importance compared to toe and caster. It was previously discussed, that the camber angle can add additional lateral force to the tire, although, not to the same extent as the toe angle. It is beneficial to have a considerable camber angle setting in a unidirectional competition vehicle. However, in terms of a low velocity urban delivery vehicle, camber angle does not provide any sufficient benefits. That is why the camber gain in such a delivery vehicle should be minimised to an extent. To achieve the best efficiency, it is reasonable to keep the camber angle in neutral setting or in a marginal negative setting. It is also useful to mention that the camber angle is not direction dependent, i.e. the setting does not change its sign regardless of direction of motion, compared to caster and toe angles. In other words, if camber is positive in the front, it will also be positive in the rear. It should be noted that there is a considerable increase in the steering force when the camber angle setting is not symmetrical in a bidirectional vehicle. It should be mentioned that the camber gain in the RK should be minimised as much as possible, due to above mentioned points as well as the fact that the RK has the highest camber gain across all systems. The absence of camber gain in the 2CV and low camber gain in the CLS are considered acceptable. It is possible to utilise the benefits of camber angle during cornering events to minimise the rolling loss but that will require an active camber system with sophisticated control algorithm. The implementation of such system is debatable, since it will add enormous complexity to the suspension system and will provide the benefits only during cornering.
Chapter 5

Conclusions and Future Work

5.1 Conclusions

A general statement is that the vehicle that has been tested in this study was based on concepts. The choice for either using a bidirectional, neutral or non-bidirectional setup depends on what the prioritisation for the vehicle is. For instance, in motorsport, some vehicles perform better in turns than in straight-line driving, which means that some parts need to be sacrificed and have lower performance in order to optimise other parameters.

This results in assumption 1 in Section 2.1 being true and shows that a mirrored caster angle between front and rear is preferred for a bidirectional vehicle. The overall assumption of symmetry in bidirectional vehicle was confirmed by multiple studies. It is implied that there is a possibility to have a static angle setup, which will satisfy all the requirements on safety, stability and efficiency. However, if the vehicle performance needs to be outstanding, then implementation of active angles in the suspension system is worthwhile. As to be expected, there was no optimal solution which would perform outstandingly well in all driving scenarios. Assumption 3 in Section 2.1 can also be stated as true, however, this depends on the required performance of the vehicle.

It should be explained that the bidirectionality implications are not only limited to suspension systems. Bidirectionality in a vehicle means specific requirements on driveline and dynamic brake force distribution, which changes with vehicle direction, as well as symmetrical and uniformly
stiff chassis. It should be noted that when the vehicle needs to be fully autonomous, it is beneficial to set it up to be easy to control as for a human driver. This means it will be less challenging to control with a control system later.

It was found that the static suspension angle importance does not depend on the suspension type.

It should be clarified that the RK and 2CV were created in Adams, where the models were built to a certain maturity level. They were both created with arbitrary input, which created a chain reaction of errors. The models required an extensive amount of time in order to solve the issues. This led to the exclusion of the RK and 2CV from the testing phase, due to a strict time restriction.

From the performed studies, it is rational to conclude that the most important suspension angles for the bidirectional vehicle are caster and toe. Since those are direction dependant and give the biggest change in vehicle behaviour when altered. The camber angle has limited influence on the vehicle behaviour in bidirectionality. The KPI angle does not influence the vehicle behaviour directly but instead, it is most influential in the combination with other angle setups. The KPI can improve or compromise a seemingly well performing setting. Relevant values of the angles can be viewed in Table 3.5. As stated before, the interactions of the angles were not explicitly studied in this report. Hence, the conclusion on this topic needs further investigation.

The possible suspension technical solutions for a bidirectional vehicle are the three suspension types presented in this study. The CLS and RK suspension types are inherently better choices for a bidirectional vehicle according to the findings in the 2D analysis, Adams simulations and the fact that those suspension types are based on DWB and MacPherson types (which are already used in some prototype bidirectional vehicles). The 2CV suspension type has some considerable advantages for bidirectionality (the self-adjusting caster angle, low profile, apparent simplicity of the construction and no apparent undesirable angle gains). However, this suspension type also has subtle weaknesses (sensitive to bump steer, space inefficient tie rod placement, lower roll stiffness with similar spring rates and expected poor ride quality when high
loading capacity is needed). It should be stated that with enough optimisation, any of the suggested suspension types will be suitable to use in a bidirectional vehicle, together with the analysed drivetrain configurations.

5.2 Future Work

Bidirectionality is a relatively new trend in the road vehicle sector, which results in many unexplored research areas. The future work that will be mentioned in this section will only cover a fraction of what can be analysed hereafter.

This study required a wide base of assumptions as well as arbitrary dimensions, compared to studies performed on existing models/vehicles. Repeating this study with a model with known dimensions, will lead to more realistic results.

The RK and 2CV can be modelled and fully analysed in Adams as the CLS. A deeper analysis can be performed by using the full test matrix, which includes the interactions of the suspension angles. This will give a deeper understanding of how the angles truly interact in a bidirectional vehicle. The models can also be analysed through other output parameters or even simulated in other test manoeuvres. A kinematic study can also be performed in Adams, with created ‘requests’ in the models. This will provide a deeper kinematic analysis compared to the one in 2D CAD environment, which was performed in this thesis. The arbitrary dimensions in RK and 2CV were assumed due to lack of information on existing models for these suspension types in heavy vehicles. The assumed dimensions in these models can be revised to resemble their existing counterparts.

The suspension angles setup can be optimised together with the suspension geometry. This will naturally improve the performance as well as the efficiency, stability and safety of the vehicle.

The 4WD functionality was decided to not be fully implemented in the Adams model. The decision was made after a comparison against 2WD simulations were done and with differences being extremely small. A full
implementation of a sophisticated 4WD system in Adams would require a considerable amount of time and was also deemed unnecessary for what was investigated in this thesis.

Brake torque control as well as direct yaw control can be implemented and their effect on the overall efficiency can be analysed.

A fully active suspension can be implemented and the effects of such suspension on overall efficiency can be analysed.

Regarding the SA test, an investigation was performed where a new and improved definition of the test was made for the CLS Adams model. The new proposed setup for this test was aiming at investigating the amount of force exerted into the steering systems during a side wind force. This investigation can give an insight into the effect of caster angle settings on the straight-line stability of the vehicle during a disturbance. The test matrix approach can be applied to the new SA test setup, in order to find the best performing setting.

The setup consists of straight-line steady state driving at 100 km/h with exertion of side wind impulse force of 1000 N at 5 s. The side wind force is applied at the chassis COG point. The test simulation time is 12 s with a step of 0.01 s. The force in the steering rack is registered at the first peak and is marked with a black line in the Figure 5.1. In this particular example, the steering rack force reached 0.22 N at 6.33 s simulation time. It should be noted that the caster angle setup is equal to ‘Caster setting 3’, as described in Figure 3.17, while the other angles are set to neutral (see Table 3.5). The side wind impulse force is shown in Figure 5.2.

Figure 5.1: Force in the steering system during side wind.
The steering system adjustment, which is mentioned in Subsection 4.3.2, is highly recommended to include in the new SA test setup as well as a future development of the suspension angle testing.

On a last note, the readers are also encouraged to use their imagination to expand the possibilities of investigation on bidirectionality.
Conclusions and Future Work
References


