Analysis of truck steering behaviour using a multi-body model

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Master thesis

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Preface

This Master’s thesis is the result of a study that has been done for Eindhoven University of Technology in cooperation with DAF Trucks N.V. in Eindhoven. Most of the research has been done at the Product Development department of DAF Trucks in the Technical Analysis group.

This report is written under the assumption that the reader has some basic knowledge about tyre and vehicle dynamics. If necessary references [1] and [2] can provide the reader with additional information on both topics.

I would like to thank Prof. Dr. Henk Nijmeijer and Dr. Ir. Igo Besselink from Eindhoven University of Technology for their support and coaching during this graduation project. Special thanks to Ir. René Liebregts who supported me during the entire project and especially during my stay at DAF Trucks. Finally, I would also like to thank the colleagues at the department of Technical Analysis at DAF Trucks for the pleasant support during my stay at DAF.
Abstract

In the car industry, the growing influence of mechatronics has led to the development of so-called active safety systems like ABS, ESP, active steering, and so on. In order to function properly, the development of these systems requires excellent knowledge of vehicle dynamics.

To improve this knowledge and to gain insight in the steering behaviour of a truck and semi-trailer, a study is started in cooperation with DAF Trucks Eindhoven. This study analyses the steering behaviour of a truck and trailer combination with the use of a multi-body model. Where possible, an attempt is made to analyse results from a driver’s viewpoint in order to compare objective model results with the subjective quantification of a driver.

To achieve the objective, a multi-body model of a tractor semi-trailer is available, including the tyre model. However, a few modifications have to be made to ensure results are correct. Especially the tyre model needs a review, which is very important when analysing vehicle dynamics. Second, a set of virtual manoeuvres have to be formulated that characterise the vehicle’s dynamic response to a steer input. Finally, the steering behaviour of the initial model can be analysed by simulating these tests. To improve insight in the dynamic behaviour a parameter study is done. For this study, several of the vehicle’s parameters are varied over a certain range, and simulation results are analysed.

For the tyre model, the existing set of model parameters produce results that are unreliable for analysing the steering behaviour. A closer study of the results shows that additional measurements are needed to solve the problem. Therefore, the choice is made to use a simplified tyre model. This way the model represents a truck tyre with realistic characteristics.

To analyse steering behaviour, three tests are selected that characterise the vehicle response for a general steer input. A circle test is selected to create a steady state steering characteristic. The analysis shows that a truck is understeered in almost any condition. A step response is selected to study the dynamic response of the vehicle. Results of this test are especially useful for studying the dynamic response of chassis and cabin. Finally, a sine sweep is selected to generate transfer functions. These transfer functions contain a lot of information necessary to analyse the steering behaviour.

To study the influence of parameter variations, nine parameters that are supposed to affect the steering behaviour, are selected. These parameters are varied over a wide range and the influence on the steering behaviour is studied by the same three tests. Results of this study can roughly be divided in parameters that actually change the steering characteristic and parameters that only influence the feedback signals. A different steering characteristic is mainly caused by mass distribution or tyre characteristics, while parameters that purely influence feedback signals will most often change the eigenfrequency of a component.

For future research, it is important to validate results with real tests. It is essential to validate and expand the knowledge of truck tyres by doing new measurements. For the truck model it is important to validate the model with respect to steering behaviour and to provide the necessary link between objective and subjective quantification.
Samenvatting

Door de toenemende invloed van mechatronische systemen in de automotive industrie, heeft veiligheid zich kunnen uitbreiden tot diverse actieve veiligheidssystemen zoals, ABS, ESP, active steering en adaptive cruise control. Om deze systemen goed te laten functioneren is bij de ontwikkeling een zeer uitgebreide kennis van voertuigdynamica nodig.

Om de kennis en inzicht in deze voertuigdynamica te verbeteren, is in samenwerking met DAF Trucks Eindhoven een onderzoek gestart naar het stuurgedrag van een trekker-oplegger combinatie. Aan de hand van simulatie resultaten van een multi-body model wordt het stuurgedrag geanalyseerd. Daar waar mogelijk is de analyse uitgevoerd vanuit het oogpunt van een chauffeur om een vergelijk te maken tussen de objectieve resultaten van het model en de subjectieve beoordeling van de chauffeur.

Voor het onderzoek wordt een bestaand vrachtwagenmodel gebruikt dat is uitgerust met een bandmodel. Echter, om een goed onderzoek naar het stuurgedrag mogelijk te maken, zijn enkele aanpassingen aan zowel het vrachtwagen- als het bandmodel noodzakelijk. Daarna worden er een aantal virtuele manoeuvres samengesteld die het dynamisch gedrag van een vrachtwagen karakteriseren. Als laatste kan aan de hand van deze testen het stuurgedrag van de vrachtauto worden geanalyseerd. Om meer inzicht te krijgen is er aan het onderzoek een parameterstudie toegevoegd. In dit onderzoek worden diverse voertuigparameters gevarieerd over een bepaald gebied, waarbij het stuurgedrag opnieuw wordt bestudeerd.

Bij de studie naar het gebruikte bandmodel blijkt dat de resultaten onvoldoende zijn voor een goed onderzoek naar het stuurgedrag. Nader onderzoek wijst uit dat er aanvullende metingen nodig zijn om de problemen op te kunnen lossen. Daarom is er voor dit onderzoek gekozen om verder te gaan met een vereenvoudigde versie van het bandmodel.

Voor de beoordeling van het stuurgedrag, zijn er drie virtuele testen opgesteld die het gedrag van het voertuig karakteriseren. Met een cirkeltest kan een steady-state stuurkarakteristiek worden gemaakt, welke laat zien dat de vrachtwagen onder vrijwel alle omstandigheden onderstuur heeft. Met een stapstuur kan de dynamische voertuigrespons worden bestudeerd. Hieruit blijkt dat resultaten te splitsen zijn in specifieke bandkrachten en resultaten die afhankelijk zijn van het dynamische gedrag van het chassis en de cabine. Als laatste worden er door middel van sinusvormige stuurfrequenties diverse overdrachtsfuncties gegenereerd. Deze overdrachtsfuncties geven veel nuttige informatie over het dynamische gedrag van het voertuig.

Bij het onderzoek naar de invloed van verschillende parameters kunnen de resultaten worden opgesplitst in twee groepen. De eerste groep zijn parameters die de stuureigenschappen van het voertuig veranderen. De tweede groep zijn parameters die enkel invloed hebben op het stuurgevoel van de chauffeur. Van de eerste groep blijken voornamelijk de massaverdeling van het voertuig en de bandkarakteristieken een hoofdrol te spelen. Bij de tweede groep zijn voornamelijk de eigenfrequenties van diverse onderdelen van het voertuig van invloed op het stuurgedrag.

Voor toekomstig onderzoek is het erg belangrijk om het gebruikte voertuigmodel en de resultaten te valideren met metingen aan een echte vrachtauto. Deze metingen leveren ook meer inzicht in de relatie tussen objectieve resultaten en een subjectieve beoordeling. Verder is het ook van belang om de kennis over vrachtwagenbanden te verbeteren door extra metingen uit te voeren.
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1 Introduction

1.1 Motivation and background

In car development, vehicle dynamics have always played an important role in the development process to achieve good ride comfort and steering behaviour. With the introduction of seat belts and airbags, vehicle safety became a powerful marketing tool and as such an important factor in car development. The growing influence of mechantronics has led to the development of so called, active safety systems like ABS, ESP, active steering, adaptive cruise control and even active suspension systems.

Most of these systems have already been introduced in several cars and some of them have become as common as an airbag. However, in truck development, most of these systems have not yet been introduced. The development of trucks, to some extend, follows the development of cars in terms of technical innovation. This is due to the fact that truck development is mainly driven by economics and new technology usually first has to prove itself before it will be implemented in a truck.

In general, the development of a DAF truck is no exception to this process. This can be illustrated by the fact that the most recent documents of research about the steering behaviour, date from around 1990. It may seem a bit odd that a truck manufacturer leaves this topic untouched for over 15 years. However, the explanation is quite simple. First of all, the steering behaviour of a DAF truck was and still is considered to be very good. This reduced the need for further research. Secondly, the last twenty years, nothing really changed in the geometric blueprint of a truck. Most of the geometric details are optimised to fit European regulations as economically as possible. The changes that were made and did influence the steering behaviour, as was noticed during real tests, could easily be corrected by suspension adjustment. Thirdly, the absence of technical innovation in truck steering, also minimized the need for further research.

Today, not only the introduction of active safety systems requires an excellent knowledge of the dynamic behaviour, also the development of active and independent suspension systems require a better understanding. Even though most of these systems will be developed by other companies, the integration of these systems is still an internal job. For example, most of these systems have some kind of control system that has an impact on the dynamic behaviour of the vehicle. So for optimal performance it is essential to understand this behaviour. Furthermore, the implementation of systems that can change the response of the vehicle may also influence the steering behaviour. Therefore it is important to optimise these systems for both functionality and steering behaviour.

These developments explain why a study on the steering behaviour of a truck is important at this moment.
1.2 Aim and scope

For this research, a truck and semi-trailer multi-body model is used to study the steering behaviour. This model is based on a generic DAF FT XF 95 truck. The model will be used to gain insights in the dynamic behaviour of a tractor semi-trailer combination. This means that this study will take a virtual approach.

The aim of this project is:

- To improve and gain further insight in the steering behaviour of a tractor semi-trailer combination, from a driver's viewpoint. This insight should form the base for an objective quantification of vehicle handling and in future, can be used to meet development goals.

To achieve this goal, the following tasks are defined:

- Update the vehicle model. Although a vehicle model is present, there are two items that need to be reconsidered. First, the tyre model produces doubtful results. Second, the steering geometry needs to be updated.
- Find out what kind of virtual tests are suitable to investigate the steering behaviour and which signals need to be measured to represent this behaviour.
- Analyse the dynamic behaviour of the model that represents a real truck.
- Investigate the effect of parameter variations on the steering behaviour.

Note that a complete investigation of the steering behaviour includes extensive testing with different real vehicle configurations and manoeuvres and comparing these tests with driver comments. This is much more than can be done in this study. However, this report is the start of what hopefully becomes a complete and in depth analysis of the steering behaviour of a truck. The result of the research should include an effective method for the quantification of this behaviour. This report gives a first insight in truck dynamics during a steering event.

1.3 Content

This report describes the study that has been done to achieve the aim described above. Chapter 2 starts with a brief summary of what can be found in literature about how to quantify the steering behaviour or handling of a vehicle. It will give the reader an open view about the aspects on how to quantify vehicle handling. In Chapter 3 the revision of the tyre model is explained. First, the problems with the current set of parameters and measurement data are given. Then a comparison is made between the different types of tyres and finally the new set of parameters is presented. Chapter 4 is the main chapter where the model, test methods and the analysis of the initial model are explained. In Chapter 5 the effect of several model parameter variations is explained. Finally, Chapter 6 gives the conclusions and recommendations.
2 Quantification of vehicle handling

In this chapter an explanation is given on how to quantify vehicle handling, which can be done in several ways. One of these is the perception of the driver, which is the starting point of the quantification in this study [12]. The only difference between quantification by an actual driver and the quantification in this study is that here it is attempted to quantify handling as an objective measure.

The first step in quantifying handling from a driver’s perspective is to investigate on what information the driver bases his opinion. The input of the driver on the steering wheel will lead to a vehicle response. This response can trigger the driver to change his input again. The vehicles path may also be influenced by external disturbances, which will again trigger the driver to make a correction. This interaction between driver and vehicle can also be explained as a closed loop control system, which is shown in Figure 2.1.

![Figure 2.1: The ‘closed loop system’ between driver and vehicle.](image)

In this system, the ‘vehicle’ block represents the dynamic system, in other words the vehicle that is controlled by the driver. The driver can influence the vehicle’s motion by changing the steering wheel angle. The dynamic response is then evaluated again by the driver to see if further corrections are needed. Evaluation starts with a desired response that is compared with the actual vehicle response containing all signals that inform the driver about the vehicle’s motion. For example, there are three motion variables in which the vehicle will move when the driver turns the steering wheel. The lateral acceleration, yaw velocity and roll angle characterise the motions during a steering event and will therefore provide the basic feedback. Besides motion, there are some other signals, like steering wheel torque that provide useful information to the driver. Finally, external disturbances such as road irregularities or crosswinds can have a strong influence on the response of the vehicle. So when quantifying handling, these effects can not be ignored.

With this schematic representation in mind, vehicle handling can be quantified into three different aspects [3],[6],[8].

- **Steering precision and steering moment.** When the driver wants to change the steering angle, he needs to overcome a certain steering torque. The amount of torque needed, depends on the desired steering angle; most of the time a larger angle will require a larger torque. The relation between the steering angle, steering torque and the motion of the truck, informs the driver about the magnitude and precision of his correction and the amount of friction between tyre and road surface.
Vehicle response. This aspect describes the relation between a steer input and the motions of the vehicle. As soon as a driver changes the steering angle, the vehicle will change course. The motion can be characterised by the change of specific variables. The relation between these variables and the input signal, gives an indication of vehicle handling.

Vehicle directional stability. External disturbances, like road irregularities or wind influence, can cause the vehicle to leave the desired trajectory. For the driver, this means that he has to make corrections to keep the vehicle on course. The number of corrections that need to be made and the magnitude of these corrections will provide a measure for vehicle stability.

In the following sections these definitions are explained in more detail.

Steering precision and steering moment [11]

When the driver turns the steering wheel he has to overcome a certain steering torque. This torque does not remain constant but will usually increase for larger steering angles. The total steering moment consist of three independent moments, all acting on the steering system [6].

\[ M_{\text{steer}} = M_{s, \text{friction}} + M_{s, \text{static}} + M_{s, \text{dynamic}} \]  \hspace{1cm} (2.1)

\( M_{s, \text{friction}} \), is the friction moment for the entire steering system up to the stub axle.

\( M_{s, \text{static}} \), is the moment that results from the king pin’s caster and inclination angle. These angles will cause a very small vertical displacement, also called jacking effect, which results in a self aligning torque.

\( M_{s, \text{dynamic}} \), is the moment that originates from the self aligning torque of the tyres and is mainly influenced by the cornering stiffness and the velocity of the vehicle.

The question is; how to evaluate the quality of the steering moment? To answer this question, two ratios can be calculated that give a good measure of how well the steering moment informs the driver about the corrections made.

The first ratio is called the relative dynamic self aligning torque and can be calculated by dividing the dynamic steering moment by the total steering moment.

\[ R_{\text{dyn}} = \frac{M_{s, \text{dynamic}}}{M_{\text{steer}}} \]  \hspace{1cm} (2.2)

This number indicates the share of the dynamic moment to the total steering moment and informs the driver about the dynamic activity of the vehicle. For example, cornering at high velocities will require more steering torque than cornering at low velocities. In addition, sharp turns will also require higher steering torques than wide turns. In both cases, a large value for \( R_{\text{dyn}} \) means that the dynamic steering moment has a large share in the total steering moment. So during a steering event, the steering moment gives the driver a good indication of the condition of the vehicle and tyre road contact in particular.
The second ratio is called the relative total self aligning torque and can be calculated by dividing the sum of the static and dynamic steering moment, by the total steering moment.

$$R_{tot} = \frac{M_{s,static} + M_{s,dynamic}}{M_{steer}} \quad (2.3)$$

In this ratio the static moment is added to the dynamic moment, to give an indication how well the steering moment informs the driver about the steering angle. When a driver turns the steering wheel, he expects the steering torque to increase for larger steering angles. This way the driver gets informed about the magnitude of the steering angle. At low speeds the ratio will be dominated by the static moment, while for higher speeds the dynamic moment becomes more important.

To illustrate the meaning of these ratios, Figure 2.2 shows three theoretical figures. The first figure shows the theoretical course of $R_{tot}$ for four different vehicles. The second figure shows the theoretical $R_{dyn}$ for the same four vehicles and the third figure shows an example of how the moments are related to each other.

![Figure 2.2: Theoretical steering ratio and steering moment for different vehicles [8].](image)

In this case the driver of car B would have the best feedback from his steering system since $R_{tot}$ and $R_{dyn}$ both reach a high value. This indicates that a large fraction of the steering moment is considered to contain useful information.

**Vehicle response**

Vehicle response is the most common aspect to quantify handling. As the word describes, the key is to evaluate the response of the vehicle to a steer input and comparing this to the opinion of an average driver. Most people are capable to give some kind of judgement about handling from their own experience of driving a car. But when an objective measure is needed, it is important to realise there are a lot of aspects that can influence the response of a vehicle.
As explained, handling starts with a steer input. From there the input has to travel through a mechanical steering system before it reaches the front wheels. Due to the steering angle, the tyres generate a lateral force necessary to move the vehicle. This movement is initiated at the front wheels and is passed through chassis and suspension systems, to the cabin and driver. It might be obvious, that besides the loop as described above, influences like elasticity, tolerances, suspension setup, dynamic behaviour and many more, can disrupt the desired response.

During cornering, the following variables define the basic feedback for the driver and inform him about the vehicle response. Therefore, these signals are suitable for representation and analysis of the response [3].

\[ \dot{y} : \text{ representing the lateral acceleration.} \]
\[ \psi : \text{ representing the yaw velocity.} \]
\[ \phi : \text{ representing the roll angle.} \]

Subsequently standard techniques like bode plots, power spectral densities, transfer functions or just a plot against time can be used to analyse these signals.

**Vehicle directional stability**

The previous two aspects describe handling and its quantification for a closed loop system. However, external influences can also affect the vehicle behaviour. In this case, the vehicle is driving in steady state conditions. If an external disturbance causes the vehicle to leave the path, the driver reacts by correcting the vehicle until the desired trajectory is reached again.

One of the most common disturbances a vehicle will experience, are road irregularities. In truck dynamics, bump and roll steer effects are well known problems. Due to suspension and steering geometry a vertical displacement of the wheels can change the steering angle. Another irregularity that is known to cause problems, are ruts in the road surface. Since a truck will drive in a straight line for 80 percent of its time, straight line stability is an important aspect of handling. So the effect that ruts have on straight line stability is therefore also a measure of handling. Except road irregularities there are some other effects, like side wind, that might influence the vehicle stability.

The variables as described in the vehicle response section should be sufficient to detect, analyse and quantify vehicle stability. However, a different approach is needed to investigate these effects. For example, to analyse vehicle stability an external input is needed to change the vehicle’s path, while for the vehicle response this input is created by the driver. When a disturbance is simulated that changes the vehicle’s path, there are three options to quantify stability. The first one is not to correct the vehicle and simply measure the deviation. The second option is to correct the vehicle at once and measure the correction needed. The third and final option is to continuously correct the vehicle when the external disturbance also continuously vary, like the size and path of ruts. In this case, a driver model is needed to provide the necessary corrections. The amount and magnitude of the corrections can be a measure of vehicle stability.
Chapter 3: Truck Tyre Model

In vehicle dynamics, tyres are probably the most important part of the car. They are the vehicles’ connection with the road and all forces necessary to direct the vehicle, originate from the tyres. Therefore it is important, when studying the steering behaviour of a virtual truck model, that the force and moment characteristics of the tyres are represented correctly. In this study the Magic Formula (MF) tyre model is used to describe the tyre behaviour. This chapter gives a description of the tyre model parameters in order to represent truck tyre behaviour. It is assumed that the reader has basic knowledge of the Magic Formula, if not, please refer to [1].

3.1 Background information

In the year 2001 a consortium of seven different companies, including DAF trucks, under the supervision of TNO automotive, executed a project for testing truck tyres. The objective of this project was to obtain a full description of the tyre-road interaction forces under steady state and transient rolling conditions. The description of the interaction forces is based on the Magic Formula (MF) tyre model in which characteristics for pure lateral, pure longitudinal and combined slip conditions are obtained [5]. In total eight different tyres were tested under various conditions and with various types of testing equipment. In Table 3.1 an overview of the tyres measured is given.

<table>
<thead>
<tr>
<th>Tyre ID</th>
<th>Tyre size</th>
<th>Rim size</th>
<th>Tyre type</th>
<th>Tyre brand</th>
<th>Tread pattern</th>
<th>Inf. pressure [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>tyre1</td>
<td>315/80 R 22.5</td>
<td>9.00x22.5</td>
<td>driven axle</td>
<td>Bridgestone</td>
<td>M729</td>
<td>8.25</td>
</tr>
<tr>
<td>tyre2a</td>
<td>315/80 R 22.5</td>
<td>9.00x22.5</td>
<td>steering axle</td>
<td>Bridgestone</td>
<td>R227</td>
<td>8.25</td>
</tr>
<tr>
<td>tyre2b2</td>
<td>315/80 R 22.5</td>
<td>9.00x22.5</td>
<td>steering axle</td>
<td>Goodyear</td>
<td>G391E</td>
<td>8.25</td>
</tr>
<tr>
<td>tyre2c</td>
<td>315/80 R 22.5</td>
<td>9.00x22.5</td>
<td>steering axle</td>
<td>Pirelli</td>
<td>FH55</td>
<td>8.25</td>
</tr>
<tr>
<td>tyre3</td>
<td>385/65 R 22.5</td>
<td>11.75x22.5</td>
<td>trailer axle</td>
<td>Goodyear</td>
<td>G465</td>
<td>9.00</td>
</tr>
<tr>
<td>tyre4</td>
<td>265/70R 19.5</td>
<td>7.50x19.5</td>
<td>trailer axle</td>
<td>Pirelli</td>
<td>ST95</td>
<td>8.50</td>
</tr>
<tr>
<td>tyre5</td>
<td>315/70 R 22.5</td>
<td>9.00x22.5</td>
<td>steering axle</td>
<td>Pirelli</td>
<td>FH55</td>
<td>9.00</td>
</tr>
<tr>
<td>tyre6</td>
<td>295/60 R22.5</td>
<td>9.00x22.5</td>
<td>driven axle</td>
<td>Bridgestone</td>
<td>M729</td>
<td>9.00</td>
</tr>
</tbody>
</table>

Table 3.1 The tyres used in the measurement programme [4].

As can be seen in Table 3.1, truck tyres can be divided into three categories. Difference is made between tyres mounted on the driven axle, the steering axle and the trailer axle. From now on these tyres will be called, drive tyre, steer tyre and trailer tyre. For each tyre, one set of MF parameters is determined and stored in a tyre property file. During simulation, the tyre property files will be used by the tyre model to simulate the behaviour of a specific tyre. The property files are created by an optimization routine that optimises the MF parameters to minimise the differences with the measured data.

At the start of this study, tyre property files are available for each tyre. However, these files are the result of the project described above and where created several years ago for an older version of the MF. Simulation results with these files are considered to be doubtful and it is questioned whether these files are useful to study handling. Therefore an analysis is made to examine the quality of the original tyre property files. If necessary, references will be made to the final reports of the project, [4] and [5]. For detailed information of the measuring facilities, test conditions, tyres and project results, please also read [4] and [5].
3.2 Tyre properties and Magic Formula model

As explained before the vehicle model uses one tyre property file for each type of tyre. Table 3.2 shows the relation between a specific tyre, the tyre property file and the name of the dataset it was created from.

<table>
<thead>
<tr>
<th>Filename</th>
<th>Tyre</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>truck1.tpf</td>
<td>tyre1</td>
<td>driven tyre</td>
</tr>
<tr>
<td>truck2b2.tpf</td>
<td>tyre2b2</td>
<td>steering tyre</td>
</tr>
<tr>
<td>truck3.tpf</td>
<td>tyre3</td>
<td>trailer tyre</td>
</tr>
</tbody>
</table>

Table 3.2 Relation between tyre, tyre property file and tyre type.

The first and logical step to check the quality of the property files is to see how well the model fits the original measurement data. To achieve this, a program is written that calculates the tyre characteristics for a specific tyre, and plots the results together with the original measurement data. The complete results are presented and discussed in Appendix A. The most important or characteristic results will be presented and discussed in this paragraph.

Steer tyre

Figure 3.1: Characteristic results of model and measurement for the steer tyre.
The top two figures of Figure 3.1 show the longitudinal and lateral tyre forces for various vertical loads. The dotted lines represent the measurement data and the solid lines represent the tyre model. Clearly these tyre characteristics fit the measurement data very well and only a few remarks can be made. In case of pure braking, the measurement data for a vertical load of 32 kN show some strange results at $\kappa < -0.15$. However, it is more likely that this results from measurement errors, rather than it represents the actual tyre behaviour.

Furthermore, the cornering characteristics in the top left figure show a peak value around $\pm \alpha \approx 12 \text{deg}$. After this peak, the value for $F_y$ decreases, while it was expected to remain fairly constant. A possible explanation for this result is the shortage of measurement data at large slip angles. As a result, the optimisation routine will not be bounded at higher slip angles and unexpected results may occur.

Unfortunately, the results as shown in the two bottom graphs of Figure 3.1 are not as well as the first. The bottom left graph shows several friction coefficients and it becomes clear that these results can not be trusted. Note that model results are now extrapolated for vertical loads above those of the measurement data. Therefore, a brief discussion will be given, for both graphs.

**Lock friction:**
The lock friction coefficient indicates the brake force of a locked wheel at a specific vertical load in pure longitudinal slip conditions. In other words, this is the point of full longitudinal slip where $\kappa = -1$, as can be seen in the figure. The lock friction coefficient is calculated by:

$$\mu_{l,\text{lock}} = \frac{F_x}{F_z} \quad \text{for} \quad \kappa = -1$$ (3.1)

$F_x$ represents the longitudinal braking force, $F_z$ the vertical load and $\kappa$ the longitudinal slip coefficient. Looking at the curve for the lock friction, two comments can be made. First of all, the lock friction coefficient reaches the same value as the brake and drive friction coefficient, at a vertical load of 65 kN. If this is true, it means that the tyre can generate the same amount of brake force in full slip conditions, as it can with almost no slip. This is a doubtful result since it does not correspond to friction laws and experience with other tyres. The second remark is, the lock friction coefficient suddenly changes from positive to negative at a vertical load of 70 kN. In other words, if somewhere along a brake trajectory the vertical load exceeds 70 kN, the vehicle would stop braking and start accelerating again. No explanation is needed to understand that this result is completely wrong.

**Lateral friction:**
The lateral friction coefficient, determines the maximum amount of side force ($F_y$) a tyre can generate at a certain vertical load and can be calculated using the following formula.

$$\mu_{y,\text{neg}} = \frac{\max(F_y)}{F_z} \quad \text{and} \quad \mu_{y,\text{pos}} = \frac{\max(-F_y)}{F_z}$$ (3.2)

The curve for the lateral friction looks alright and no comments can be made.
Brake and drive friction:
This friction coefficient determines the maximum longitudinal force ($F_x$) that can be generated with braking or accelerating the tyre. The figure shows that the curve reaches a maximum between $0 < \kappa < 0.25$ and a minimum between $-0.25 < \kappa < 0$. The brake and drive friction coefficient can be calculated by dividing the longitudinal force $F_x$ by the vertical load $F_z$.

\[
\text{for } \kappa = \text{neg. } \mu_{\text{brake}} = \min(F_x)/F_z
\]
\[
\text{for } \kappa = \text{pos. } \mu_{\text{drive}} = \max(F_x)/F_z
\]

The only comment that can be made is the dip that appears at the exact same point where the curve for the lock friction changes from positive to negative. Only this time the curve does not flip to negative, but restores back to normal. Again, a model error can be expected because a friction coefficient does not usually change this rapid.

Longitudinal slip stiffness (bottom right of Figure 3.1):
The longitudinal slip stiffness represents the slope of the longitudinal slip curve at $\kappa = 0$, and can be calculated by:

\[
C_{F_x} = \frac{\partial F_x}{\partial \kappa} \bigg|_{\kappa=0}
\]

The longitudinal slip stiffness shows an abnormal course. At a vertical load of 40 kN, the curve starts to continuously descend and at a vertical load of 75 kN, the slip stiffness becomes negative. Obviously, this is another physically impossible result, since the slip stiffness should continuously increase instead of turning negative.

Cornering stiffness:
The cornering stiffness represents the slope of the side slip curve at $\alpha = 0$, and can be calculated by:

\[
C_{F_y} = \frac{\partial F_y}{\partial \alpha} \bigg|_{\alpha=0}
\]

This curve shows no abnormalities.

Camber stiffness:
The camber stiffness represents the vertical shift for the side slip curve and can be calculated by:

\[
C_{F_y} = \frac{\partial F_y}{\partial \gamma} \bigg|_{\gamma=0, \alpha=0}
\]

The curve for the camber stiffness looks alright and seems to fit the measurement data quite well. However, the only measurement data available is for camber angles of $\gamma = 0$ and $\gamma = -2$. In other words, experimental validation is rather limited which makes it hard to ensure results are correct.
Driven tyre

Figure 3.2 shows some characteristic fit results for the drive tyre. Because, most of the comments that can be made on the characteristics have already been discussed with the steer tyre, comments will be limited to questionable results only.

The top two graphs of Figure 3.2 again look alright. However, the bottom two graphs are again doubtful and therefore need some extra discussion.

In the bottom left figure, the lock friction coefficient shows the same abnormal behaviour as it does for the steer tyre. It reaches the same value as the brake friction coefficient at a vertical load of 65 kN. The other friction coefficients do not significantly show abnormal behaviour, although they do descend faster than expected. However, measurement data also show a steep descend which supports the model results. It is assumed that the friction coefficients for truck tyres do not change linearly with Fz. But will reach a settling point for higher vertical loads. Unfortunately, no extra measurement data is available, so this theory cannot be proven.
Aligning torque stiffness (bottom right of Figure 3.2):
The calculation of the aligning torque stiffness does not differ much from the calculation of the cornering stiffness. It represents the slope of the self aligning moment at $\alpha = 0$.

\[
C_{Ma} = \frac{\partial M_z}{\partial \alpha} \bigg|_{\alpha=0}
\]  
(3.7)

The first half of the curve looks fine, but at a vertical load of 70 kN the curve starts to descend and returns to zero. This may indicate a model error because the self aligning torque stiffness should continuously increase as a function of $F_z$.

Camber stiffness for aligning torque:
The calculation of the camber influence on the aligning torque is similar to the calculation of the camber stiffness.

\[
C_{Mr} = \frac{\partial M_z}{\partial \gamma} \bigg|_{\gamma=0,\alpha=0}
\]  
(3.8)

As for the aligning torque stiffness, the first half of the graph looks alright. However, at a vertical load of 60 kN the curve becomes negative. Again, this may indicate a model error just as for the aligning torque stiffness.

Trailer tyre

Since the results for the trailer tyre are quite similar to those of the steer and drive tyre, comments are limited to a few statements only. The corresponding figures can be found in appendix A.3, just as for the steer and drive tyre, and it is left to the reader to note the same comments.

- Friction coefficients reach zero or even turn negative.
- The longitudinal slip stiffness turns negative.
- The camber torque stiffness returns to zero.
- The longitudinal relaxation length returns to zero.

3.3 Extrapolation

From the results discussed in the previous section, the conclusion can be drawn that the model fits the measurement data quite well. However, problems occur when the model is extrapolated outside the range of the measurement data.

To gain better insight in what goes wrong and to understand the consequences of these errors, further investigation is necessary. Generally, the characteristics as presented in appendix A, fit the measurement data quite well. However, these results are only calculated for the same vertical loads as the measurement data, while errors seem to occur outside this range. Therefore, the basic characteristics are calculated again, but now for vertical loads up to 94 kN. The results are presented in appendix B.1 and will be discussed briefly.
Pure braking
- For the steer and trailer tyre, the longitudinal force $F_x$ suddenly changes from positive to negative at a certain vertical load. This results from the longitudinal stiffness becoming negative, as explained in the previous section.
- At higher vertical loads, the maximum and minimum of the curve moves from $\kappa \approx \pm 0.25$ to $\kappa = \pm 1$. This is a result of the lock friction coefficient reaching the same value as the brake and drive friction coefficient.

Pure cornering (Fy)
- In figure B.2.2 the maximum value for the lateral force at a vertical load of 93 kN is lower than the maximum value with a vertical load of 81 kN. This may indicate an error for the lateral friction coefficient.

Pure cornering (Mz)
- Extrapolation of the drive tyre, results in a decrease of the self aligning moment as can be seen in figure B.2.3. This is caused by the descending curve for the cornering stiffness, which decreases the slope of the self aligning moment at $\alpha = 0$.

Combined slip
- All friction circles have odd shapes and should not be trusted.

Camber influence on Fy and Mz
- As a result of the camber stiffnesses, the camber influence on Fy and Mz also shows some strange behaviour. Especially the self aligning moment cannot be trusted, since the camber stiffness becomes negative at a certain vertical load.

The absence of measurement data for vertical loads above 50 kN, forces the model to extrapolate non-linear truck tyre behaviour. Results so far, have proven that the fit routine used to create the tyre property files, failed to create a model that represents reliable tyre behaviour beyond the measurement range. Probably the best solution for this is to do some extra measurements that provide the data needed to create a proper model. Unfortunately, measuring truck tyres at such high loads is a complex job and no such measurements will be taken in the near future. Therefore another solution is needed to solve this problem.

So the question remaining is how to find a set of Magic Formula parameters that still fit the measurement data, and also has realistic extrapolation properties for high vertical loads.

### 3.4 Refitting the Magic Formula using MF-Tool 3.1

As explained before, the tyre property files have been created in 2001 by an optimisation routine. This routine is embedded in software called, MF-Tool that uses the Magic Formula model called MF-Tyre 5.0. Nowadays, two later versions of the magic formula are available namely MF-Tyre 5.2 and MF-Swift 6.0. One of the main modifications to the new program is a new fit routine that calculates the model parameters. For instance, the old routine does not have the possibility to add constraints while the new routine does. So recalculating the parameters, using the latest versions of MF-Tool with MF-Tyre 5.2 and MF-Swift 6.0, may solve the problems with extrapolation. Figure 3.3 shows a few results.
Figure 3.3 a,b,c,d: Fit results for MF-tyre 5.2 and MF-swift 6.0.

Figure 3.3a, shows the brake characteristic for a steer tyre, extrapolated up to 81 kN for both MF-tyre 5.2 and MF-swift 6.0. Although the characteristics no longer change sign, the model still produces wrong results with extrapolation. Three curves end with a maximum around $\kappa = 1$, while they should look like the curve produced with MF-tyre 5.2 for 51 kN. Figure 3.3b, shows the camber stiffness for a drive tyre. As seen before, the camber stiffness reaches a maximum and then drops down again. For the fit created by MF-swift 6.0 the curve goes back to zero and the fit created with MF-tyre 5.2 even becomes less than zero. As explained in the previous section, stiffnesses that return to zero can not be trusted.

Figure 3.3c shows the longitudinal friction for a trailer tyre. The figure clearly shows the friction coefficient becoming zero for both fit routines and again the result can not be trusted. Figure 3.3d shows the self-aligning stiffness for a steer tyre. As for the camber stiffness, the self-aligning stiffness reaches a maximum and then turns negative.

Of course, these figures only show a small fraction of all the fit results. However, it is shown that calculating the tyre parameters by more recent versions of MF-Tool does not solve the problem at hand. Therefore it is not useful to further discuss any of these results.
3.5 Solution proposal

Although refitting the Magic Formula did not give the desired result, there is an important lesson that can be learned. In general the Magic Formula fits the measurement data quite well. As explained, most of the problems occur when the model is extrapolated outside the range of measurement data. The problem with being short on measurement data is that the model has to predict tyre behaviour in a strong non-linear range. In other words, using a mathematical model with almost 140 variables to predict non-linear tyre behaviour, while most of the measured behaviour is roughly linear, is almost impossible. This also explains why the refit, as discussed in the previous section, did not work.

So until additional measurements are taken, it will be very unlikely that the Magic Formula in its current form can represent truck tyre behaviour outside the range of measurement data. At least, not if it is desired that the model represents tyre behaviour for a complete range of vertical loads. Therefore another solution has to be used to provide an acceptable tyre model for this study.

One solution that will provide a proper tyre model is, to eliminate as much Magic Formula parameters as possible to fit the model to the measurement data, using only a limited set of parameters. This way the tyre model will represent ideal tyre behaviour. However, since there are no measurement results available that provide a clear insight in realistic truck tyre behaviour, it is best to use an ideal tyre for vehicle simulations, instead of using a tyre model which can produce significant errors.

3.6 Comparing the drive, steer and trailer tyre

The original tyre property files contain approximately 140 parameters that were used to fit the magic formula to the measurement data. Simplifying the model means that the number of parameters will be reduced to a maximum of approximately 20 parameters. The benefit of this simplification is not only that it solves problems with extrapolation, but also that the model will be tuneable by hand. This way the user is able to study the effect of different tyre characteristics on the steering behaviour of the vehicle. For example, increasing the cornering stiffness will certainly influence the steering behaviour. On the other hand, there is a need to find out if there are any characteristic differences between the steer, drive and trailer tyre. If the tyre model is going to be simplified, it means that specific tyre characteristics may disappear. Until now a distinction has been made between a steer, drive and trailer tyre. If there are any characteristic differences between those types of tyres, it might be important to understand them before the tyre model is simplified. If any characteristic differences are lost by the simplification, it is still possible to create them by hand afterwards. On the other hand, if few to no differences are found it can be practical to use only one or two types of tyres.
Figure 3.4 shows the cornering stiffness for every tyre, calculated from the original measurement data.

![Figure 3.4: Comparing the cornering stiffnesses.](image)

As can be seen in Figure 3.4, the variation in measured cornering stiffness is fairly small. The average stiffness of the steer tyres seems to be slightly higher than it is for the drive and trailer tyre. Unfortunately, it is hard to draw any conclusions since there are measurement results for four different steer tyres and only two for drive and trailer tyres. Another noteworthy result is the difference in cornering stiffness for the trailer tyres. The cornering stiffness for tyre 4 is quite low compared to the other stiffnesses, while the cornering stiffness for tyre 3 is quite high. However, an explanation can be found by the fact that tyre 3 and tyre 4 have very different dimensions, see Table 3.1. Tyre 3 has a width of 265 mm, which is also the narrowest of all measured tyres. Tyre 4 has a width of 385 mm, which is also the widest of all measured tyres. It is not hard to understand that this dimensional difference can also be found in the measurement results.
Figure 3.5 shows some other important tyre characteristics that are compared. Some brief comments are given below.

**Longitudinal slip stiffness (Figure 3.5 top left):**
For the longitudinal slip stiffness the differences are also quite small, although it appears to be that the stiffness of the steering tyres is a bit higher than the stiffness of the driven tyres.

**Aligning torque stiffness (Figure 3.5 top right):**
The aligning torque stiffness gives almost the same result as for the cornering stiffness. The values lay close together but the steering tyres seem to have a slightly steeper curve than the drive tyres.

**Longitudinal lock friction (Figure 3.5 bottom left):**
At first sight the data points for the lock friction look like they are randomly spread. In general, most of the data points for low vertical loads lay between $0.5 < \mu < 0.7$ and as the vertical load increases the friction coefficient slowly descends. A few data points lie outside this range but this is probably caused by measurement errors. Most important, there is no notable difference between the types of tyres.
Lateral friction (Figure 3.5 bottom right):
For the lateral friction a few remarks can be made. The data points of all steer tyres lie very close together. As explained before, the data points of the trailer tyres mark the upper and lower bandwidth of all tyres, which is probably caused by the difference in dimension. The drive tyres have some variation compared to the other tyres. Unfortunately, the average result of the drive tyres lays very close to the average result of the steer and trailer tyre. Therefore, it remains hard to draw any conclusions.

Appendix B shows the comparison of the other tyre characteristics. Page B-4 shows the data points that are directly calculated from the measurement data, like Figure 3.4. On page B-5 the measured tyre characteristics are compared, but not by plotting the measurement data in the same figure. As explained before, the tyres are measured under various load conditions. Unfortunately, these conditions are not the same for every tyre, so the measurement data cannot be compared directly. Therefore, the characteristics are calculated for the vertical loads of 20, 35 and 50 kN, using the original tyre property files. Even though these property files contain errors, it is proven that the fit results at low vertical loads are alright and therefore the comparison should be trustworthy.

In general, it can be concluded that the characteristic differences between all tyres are fairly minimal. Most of the variations found, can be explained by dimensional differences, variation in measurement conditions and model or measurement errors. In most cases where variations do represent specific tyre behaviour, it is hard to attribute this behaviour to the tyre type. These variations do not uniformly point into one direction and therefore, can only be seen as specific characteristics of the measured tyre.

With these conclusions in mind it would be obvious to use only one tyre property file for all truck tyres, instead of three different property files. However, the choice is made to create two reduced tyre property files. The first one will represent the steer tyre and the second one will represent the drive and trailer tyre. The main reason for this choice is that for further research, it can be necessary to change the characteristics of the steer tyres separately from the drive and trailer tyres.

3.7 Simplified Magic Formula

As explained before, the original tyre property files contain about 140 parameters so that the Magic Formula can represent the tyre behaviour. The function of these parameters varies from simple dimensional information, to parameters that characterise the dynamic tyre behaviour under various conditions. To simplify the Magic Formula and reduce the chance of errors, the choice is made to use a selection of approximately 25 of the original 140 parameters. A complete explanation of all parameters of both the original and the reduced tyre property files, would lead to an in-depth discussion of the Magic Formula. This is beyond the scope of this study and therefore, the explanation is limited to the presentation of the reduced tyre property files in Appendix C. For further explanation of the parameters, please refer to [1].

For the continuation of this report, it is of more importance to see and discuss the results produced by the new tyre property files. The most important results are discussed in this section. Other results for the steer and drive tyre are presented in appendix D.
Figure 3.6 shows some simulation results for the simplified steer tyre. In comparison to Figure 3.1 the model produces much better results, especially for the friction coefficients and the longitudinal slip stiffness. Although the model still has difficulty to fit the data points for the friction coefficients, it can be concluded that the errors as seen previously are now eliminated. One might still conclude that the values for the friction coefficients at higher vertical loads are still quite low. However, as explained before it is assumed, that the problem lays in the fact that the friction curves are linearly descending, while they should probably be non-linear. In this case, the Magic Formula should be extended with friction coefficients that decrease non-linearly with an increasing vertical load.
Figure 3.7 shows some simulation results for the simplified drive tyre. Comments that can be made to the results of the simplified drive tyre are quite similar to those of the simplified steer tyre. The model performs quite well and no errors can be detected. Again the model has difficulty to correctly fit the friction coefficients.

So far, simplifying the Magic Formula seems to solve most of the problems, as seen with the original tyre property files. On the other hand, the original files did also perform well for load conditions within the range of measurement data. Most of the problems occurred when the model was extrapolated for values outside the range of measurement data. Therefore, it is important to verify the robustness of the simplified model under higher load conditions. Appendix E shows extrapolated model results for both the steer and drive tyre. To illustrate the effect of the simplified tyre model, Figure 3.8 shows the longitudinal slip characteristic of the steer tyre for both the full and simplified model. As can be seen in the figure, the longitudinal slip characteristic of the full model flips sign for a certain vertical load, while the simplified model shows the characteristic as it should be.
The results, as shown in this section, prove that simplifying the Magic Formula by reducing the number of parameters solved most problems. It can be concluded that when a tyre model has the possibility to represent tyre behaviour on a high level of detail, an equivalent of detailed information, like measurement data, is necessary to set up the model parameters. When this information is not present, all parameters that add more detail to the model become useless. In fact, they can even cause unexpected behaviour as they are still present in the formula. This is what went wrong in the original model.

The simplified model represents normal tyre behaviour even for vertical loads above 90 kN. With these results, there is enough confidence to use the new tyre property files to analyse the steering behaviour of the truck. So from now on, the steering tyres of the truck will be represented by the tyre property file called, “DAF_steer_tyre.tpf” and the drive and trailer tyres will be represented by the property file called, “DAF_drive_tyre.tpf”.

Figure 3.8: Left, Longitudinal slip characteristic for the full model
Right, Longitudinal slip characteristic for the simplified model
4 Initial Model

A complete multi-body model of a truck and trailer combination is used to analyse the steering behaviour. Before the actual analysis of the steering behaviour can be done, it is important to explain how the model is build up and to know its possibilities and limitations. Once this is understood, the possible manoeuvres and measurement signals that are necessary to do the analysis can be defined. Then, these manoeuvres can be simulated with the initial model, together with the basic vehicle parameters, to learn and understand the vehicle response. In the first section of this chapter the model is explained. The second section describes the manoeuvres and measurement signals. In the last section the response of the initial model will be discussed.

4.1 The model

In November 2005 a project was started by Eindhoven University of Technology and TNO to develop and validate a modular simulation model for commercial vehicles. The main research topics for commercial vehicles concentrate on durability, safety, ride dynamics and comfort. This model should be useful for testing active systems such as roll-over prevention, driver warning systems, drive by wire, active suspensions, etc. In other words the model needs to represent the dynamic vehicle behaviour so that it can give correct feedback for controller systems. Another requirement for the model is the modular approach, which makes it possible to easily evaluate different vehicle configurations. Therefore the choice is made to model both the tractor and the semi-trailer in SimMechanics which is a multi-body extension to Matlab/Simulink. One of the benefits of SimMechanics is the ability to implement Simulink control systems into the model, which is perfect for testing active systems. Another benefit is the ability to model the truck in components and place these components in a library. So if someone wants to try a simulation with a different configuration, for example another cabin, they only need to switch the library block in the model.

In the following subsections the configuration of the model, specifications and measurement setup are explained in more detail.

4.1.1 Configuration of the model

The tractor model is based on a DAF FT XF95 Super Space Cab 4x2 tractor and the semi-trailer is a generic three axle flat bed configuration. The total length of the vehicle combination is 15.50 m and in general conditions the trailer is loaded with 27000 kg of payload and a total vehicle mass of 37000 kg. In Figure 4.1 a schematic drawing of the entire model is shown.
As can be seen in Figure 4.1, the model exists of several blocks representing the cabin, chassis, axle, engine, etc. Each of these blocks have their own mass and inertia and are connected through joints and force elements. The joints define the degrees of freedom each block has in relation to the other and the force elements define the stiffness in a certain degree of freedom. First the tractor is considered. Figure 4.2 shows all the force elements between the various component blocks of the tractor.

The rigid cabin is elastically suspended in all 6 degrees of freedom with respect to the chassis. In this case all the springs and dampers are modelled similar to actual cabin suspension with only one exception. The anti-roll bar is modelled as a torsional spring in the x-direction. Both the suspension of the front and rear axle are modelled with 2 degrees of freedom with respect to the chassis, hop and roll. In this case the suspension is not modelled exactly the same as for a real truck, since the actual truck uses leaf-springs for translational stiffness in the z-direction. The limitation of this difference is that possible secondary dynamic effects, like bump-steer, cannot be investigated.
As for the cabin, the anti-roll bar is modelled as a torsional spring in the roll degree of freedom. Finally, the tractor chassis is modelled as two rigid bodies, connected by a revolute joint and a torsional spring to account for chassis flexibility. Note that the engine is not depicted in Figure 4.2 for practical reasons. However, the engine is elastically suspended in all 6 degrees of freedom with respect to the chassis, a picture can be found in appendix F.

Figure 4.3 reflects the semi-trailer showing all the components and connections.

The trailer chassis is also modelled as three rigid bodies of equal length and interconnected by a torsional spring to account for flexibility. This means that the trailer load, which is rigidly connected to the chassis, also needs to be divided into three separate parts. The total weight is distributed evenly over the three parts. The trailer suspension is modelled similar to the front and rear suspension of the tractor. The first trailer axle is connected to the second chassis body, and the second and third axles are connected to the third chassis body. Finally, note that at the end of every tractor and trailer axle the wheels are represented by the Magic Formula model in combination with a tyre property file, as explained in the previous chapter.

In appendix F.1 an overview is given of general model parameters and names.

Steering system [9]

In the original model the steering system did not include any geometrical properties like a king pin inclination angle (kpi), a king pin caster angle and an Ackerman steering geometry. These angles directly influence the rotation angles of the steering wheel and will therefore also influence the forces generated between the tyre and the road surface [10]. Since this study concentrates on the steering behaviour, the choice is made to add these properties to the steering system.
Figure 4.4 shows the original steering system.

![Figure 4.4: The original steering system, all geometric angles are 90 degrees and the steering angles are equal.](image)

The following remark applies to Figure 4.4. In the model, the king pin is only a virtual rotation joint whose coordinate system can be oriented in space. Also the tie rod is not physically present but is only simulated by entering the same steer angles for both the left and right wheel. However, Figure 4.4 illustrates that when the steering wheels are linked by a tie rod that is connected at right angles to the king pin, the left and right steering angle are the same. Figure 4.5 shows the changes made to the steering system.

![Figure 4.5: King pin, inclination and caster angles and Ackerman steer.](image)

In the new situation, inclination and caster are still virtual angles by changing the orientation of the joint coordinate system. The Ackerman steer however, is now implemented as a physical link between the left and right wheel.

### 4.1.2 Model specifications

To simulate certain manoeuvres with the SimMechanics model, an m-file specifies the simulation conditions, varied model parameters and the manoeuvre itself. This m-file then calls the model to calculate the manoeuvre and finally saves the measured signals in one or more data structures. Figure 4.6 illustrates the entire process with a block diagram.
The input signals can be divided into three groups:

- **General input.**
- **Driver input.**
- **Variation input.**

The general input is used to define all basic simulation conditions like simulation time, step size and initial vehicle position. The driver input is used to define the manoeuvre of the vehicle by defining steer, throttle and brake inputs exactly like a driver would apply them. In other words, the steer input is the actual steering wheel angle changing over time. The throttle input is the throttle position that will result in a certain speed. And the brake input is the percentage of brake application. So basically, it is possible to create any kind of manoeuvre as long as the input signal can be specified as a function of time.

A few remarks:

- If, at a certain point in time, a brake force is applied the drive axle torque will immediately be set to zero to simulate the clutch.
- If the throttle is set to a constant value, it means that a constant drive moment is applied to the wheels. But this does not mean that the vehicle will maintain a constant speed. For example, if the truck is driving uphill with an increasing slope, a larger drive moment is needed to maintain the same speed.
- If a constant speed is desired then it is also possible to define this speed as a function of time. This will activate the cruise control that can vary the throttle to maintain the desired speed.

The last group of input signals are the variable model parameters. Most of the vehicle parameters are set at a constant value, only a few of them can be changed to evaluate simulations with different configurations easily.

A short overview of the most important possibilities and limitations will be given.
Possibilities

The model gives a full dynamic description of a tractor and trailer combination, complete with non-linear tyre model and dampers. This gives the model the following possibilities.

- The simulation of almost any manoeuvre, including events like road obstacles.
- A simple implementation of all sorts of control systems.
- The possibility to test non-realistic situations and configurations to improve insight.
- The modular library enables a fast and easy change in configuration.
- An, almost, endless variation of measurement options and an easy way to compare real-time test results with simulation results.

Limitations

- No element elasticity. Apart from chassis stiffness, also the stiffness of the steering and suspension system, for instance leaf springs, can influence the steering behaviour of the vehicle. As for every multi-body method, the body elements are rigid and do not contain any elasticity. The only way to add elasticity to a body is to divide it into smaller pieces and then reconnect these pieces with spring elements. The truck and trailer chassis have already been divided this way, although it is still only a rough estimation of the actual stiffness. Unfortunately, to create a realistic deformation, quite a number of cut sections are needed and the element stiffness should be carefully measured to translate it to the model.
- No driver model present. Although the vehicle model has the ability to simulate almost every desired manoeuvre, in its current form it is not able to react to situations and correct the vehicle to follow a desired path. For example, straight line stability can only be tested by measuring its deflection from the original path when driving over an obstacle. But when judging the steering behaviour from a driver’s viewpoint, it will be interesting to know how fast and how easy the vehicle can be corrected to continue to follow the original path.
- No friction and play between components. Even though it is not necessary to model friction and play for the general vehicle dynamics, it will be important when the research is going to take a closer look on individual systems. For example, judging the quality of the steering moment and investigating shimmy.
- Limited geometric details. The vehicle is modelled on a low level of geometric details. This is all right for the moment, since a high level of geometric detail would increase the dynamic complexity. But for future research this may be insufficient. Friction and driver model can be implemented in the future relatively easily. Adding representative frame flexibility and more geometric detail will prove to be more difficult, and leads to an increase in simulation time.
4.1.3 Measurement positions

The final part of the model description gives a brief overview of all the measured signals and their location in the model. Of course, measurement signals can be added or deleted depending on the purpose of the simulation. But for the clarity of the results discussed later on in the report, it is best to describe the measurement setup used in this research.

The most important measurement signals can basically be divided into two groups. The first group contains the signals for the general analysis, such as accelerations, velocities and roll angles. The second group contains the signals that are necessary to create a so-called running mode model, which is a method to visualise response functions and will be explained later on in this chapter. Figure 4.7 shows the measurement positions for both groups. Note that the trailer loads are no longer depicted since they are rigidly connected to the trailer chassis and therefore give no extra information about the dynamics of the system.

Figure 4.7: Points of measurement for general analysis.

Figure 4.7 shows the measurement points for both dynamic and motion analysis. The points of measurement for dynamic analysis are numbered from one to ten. Each body has one central measurement point where position, velocity, angular velocity, acceleration and angular acceleration are measured in all three directions. Except the cabin, this has an extra measurement point at the driver’s seat. Each body has two additional measurement points that measure the local position to calculate the absolute roll. The measurement points used for motion analysis are marked with a triangular sign. At these points the accelerations of the bodies are measured in all three directions. Because this information is used to visualise the motions of the truck, these points lay around the entire geometry at every corner of a block.

In appendix F.2 an overview is given of the points of measurement and their names and coordinates in the model.
4.2 Measurement signals and simulation manoeuvres

In chapter 2 it is explained that the quantification of handling can be divided into three categories:

1. Steering precision and steering moment
2. Vehicle stability
3. Vehicle response

The first category can not be investigated with the current model, because of the absence of a properly modelled steering system. The second category, vehicle stability, can be investigated but only by looking at the vehicle’s deviation from the original path, as explained in chapter 2. So, to do a complete research on vehicle stability the implementation of a driver model is preferred. When investigating vehicle stability, the main aim is to investigate the effect of ruts on the straight line stability. Modelling these ruts is a complex job and the question rises if, at this moment, the model’s suspension contains enough dynamic information, like friction and play, to really get the desired effects.

This research will concentrate on the third category, vehicle response. Before explaining the approach of the research it is helpful to define the viewpoint from which all tests have been set up.

With this definition in mind, three degrees of freedom remain that characterise the steering event, according to the coordinate axes as depicted in figure 4.2:

- y-axis translation (lateral displacement)
- z-axis rotation (yaw)
- x-axis rotation (roll)

For the y-axis translation the second derivative, or acceleration \( a_y \), is chosen to reflect this degree of freedom. The reason for this is when the vehicle is driving in a straight line the lateral acceleration is zero and in steady state cornering the lateral acceleration remains at a constant value. This way the variations in lateral acceleration, during a steering event, are well visualised. For the same reason the z-axis rotation will be reflected by the first derivative, or yaw velocity \( \psi \) and the x-axis rotation will be reflected by the rotation angle, or roll angle \( \phi \).

Finally, a set of manoeuvres suitable for testing the vehicle response is required. As mentioned before, the tests need a clearly defined steer input to produce a good and repeatable result for the output signals. There are basically three possibilities to define a steer input:

1. A single prescribed steer input.
2. A continuously adaptive steer input.
3. A prescribed continuously varying steer input.

With this knowledge it is possible to define one type of manoeuvre for each type of steer input.
A step steer will create a prescribed single steer input. This is a common method for testing the response of a system and will give information about the response time, settling time and the percentage overshoot.

With a circle test it is necessary to continuously adapt the steer input to drive the desired circle. The results from this test can be used to create several steering characteristics and to measure the amount of oversteer or understeer.

A sine sweep is chosen to create a prescribed continuously varying steer input. The results from this test can be used to create the transfer functions between the input and several output signals.

In the next section each of these manoeuvres will be explained in more detail together with an extensive discussion of the results for the basic model. In practice some other test manoeuvres, like a lane change or slalom, are used to investigate the steering behaviour. However, since these tests are either an extension or a combination of the tests mentioned above, they will give no extra information and therefore will not be evaluated.

### 4.3 Simulation results for the initial model

Now the model build up, basic parameters and test methods are explained, the steering behaviour of the initial model is investigated. In this section the simulation conditions and results will be discussed for every manoeuvre.

#### 4.3.1 Circle test

A circle test is performed by driving a circle with a constant radius and gradually increasing the forward velocity. With an increase in forward velocity and thus an increase in lateral acceleration, the lateral tyre forces also have to increase in order for the vehicle to remain on a constant circle radius. Before the simulation results of the initial vehicle will be presented, a theoretical explanation of the circle test and possible results will be given. A single track vehicle model, or bicycle model, is used to support this theoretical explanation.

**Circle test in theory**

As explained later on, the truck and semi-trailer is a complex vehicle which is hard to analyse even by the equations of motion of the bicycle model. Therefore, the choice is made to start explaining the circle test by the standard bicycle model and then extending the theory to show the complexity of a truck and semi-trailer. Figure 4.8 shows a drawing of the bicycle model in a cornering situation. The model is considered to be in the x and y plane so no roll is present. The tyre and axle characteristics can be lumped into a single, equivalent tyre with centre point steering. There are no aerodynamic forces and no slopes or levelled road surfaces. For more information about the bicycle model or the mathematical description below, please refer to [2].
Figure 4.8: The bicycle model, in case of steady state cornering.

Writing down the equations of motion will result in the following coupled set of equations:

\[ m(\dot{v} + ur) = F_{y1} + F_{y2} \]  
\[ lr = aF_{y1} - bF_{y2} \]  

Where,  
- \( m \) = vehicle mass  
- \( v \) = lateral velocity  
- \( r \) = yaw velocity  
- \( u \) = forward velocity (\( \approx V \) for small slip angles)  
- \( F_{x1,2} \) = tyre side slip force  
- \( I \) = yaw moment of inertia  
- \( a, b \) = distances to C.G.

And the tyre side slip angles \( \alpha_{1,2} \):

\[ \alpha_1 = \delta - \frac{1}{u}(v + ar), \quad \alpha_2 = \frac{1}{u}(v - br) \]  

Where \( \delta \) represents the steering angle of the front wheel. The tyre side slip angles generate the tyre side force according to:

\[ F_{y1} = C_1 \alpha_1, \quad F_{y2} = C_2 \alpha_2 \]  

Where \( C_{1,2} \) are the cornering stiffness of the front and rear tyre and depend on the vertical load \( F_z \) as explained in chapter 3.

Substituting these equations and eliminating \( v \), results in the following differential equation:

\[ mlu\ddot{r} + \{I(C_1 + C_2) + m(a^2C_1 + b^2C_2)\}\dot{r} + \]  
\[ \frac{1}{u}\{C_1C_2l^2 - mu^2(aC_1 - bC_2)\}\dot{u} = muaC_1\dot{\delta} + C_1C_2l\ddot{\delta} \]  

(4.5)
In case of steady state cornering $\dot{r}, r, \delta$ are all zero. Furthermore, under the assumption that the vehicle side slip angle $\beta$ is very small the circle radius can be written as:

$$R = \frac{V}{r} \approx \frac{u}{r}$$  \hspace{1cm} (4.6)

Now the required steering angle for steady state driving a circle with radius $R$ can be written as:

$$\delta = \frac{l}{R} - \frac{mV^2}{RL} \left( \frac{a}{C_2} - \frac{b}{C_1} \right)$$  \hspace{1cm} (4.7)

Rewriting formula (4.7) and substituting $a_y$ for $V^2/R$ will give the following equation:

$$\delta = \frac{l}{R} + \frac{a_y}{g} \eta$$  \hspace{1cm} (4.8)

With $\eta$ the understeer coefficient defined according to:

$$\eta = \frac{mg}{l} \left( \frac{b}{C_1} - \frac{a}{C_2} \right)$$  \hspace{1cm} (4.9)

Furthermore, the equations for the side slip angles (4.3) and equation (4.6) can be written as follows:

$$\frac{l}{R} = \delta - \alpha_1 - \alpha_2$$  \hspace{1cm} (4.10)

Substituting equation (4.10) in (4.8) will give:

$$\alpha_1 - \alpha_2 = \frac{a_y}{g} \eta$$  \hspace{1cm} (4.11)

With formula (4.8) and (4.11), three possible situations can be defined in case of cornering.

- $\eta = 0$ than $\alpha_1 = \alpha_2$ also called “neutral steer”.
- $\eta > 0$ than $\alpha_1 > \alpha_2$ also called “understeer”.
- $\eta < 0$ than $\alpha_1 < \alpha_2$ also called “oversteer”.

With formula (4.8) and (4.11) in mind, these definitions will be discussed in more detail. In case of a neutral steered vehicle, $\eta = 0$, the forward velocity will have no influence on the required steering angle of the vehicle. So if the vehicle is doing a circle test, the required steering angle to continue the same circle will remain constant. For an understeered situation, $\eta > 0$, the steering angle has to increase in order to remain at the same circle radius while the forward velocity is increasing.
In case of an oversteered vehicle, $\eta < 0$, the situation is reversed, so the steering angle has to decrease to remain on the same circle radius.

Although the theoretical explanation for the bicycle model is quite clear, the situation for a truck and semi-trailer is far more complex. This is shown by deriving the equations of motion for a truck and semi-trailer with one trailer axle, see Figure 4.9. The situation with a semi-trailer with three axles, as used in this study, will be explained later on.

![Bicycle model of a truck and single axle semi-trailer](image)

Figure 4.9: Bicycle model of a truck and single axle semi-trailer [1].

Deriving the equations of motion will result in the following coupled set of equations, [1]:

\[
(m + m_c)(\ddot{v} + V\dot{r}) - m_c \{ (h + f)\dot{\psi} - f \dot{\psi} \} = F_{y1} + F_{y2} + F_{y3} \tag{4.12}
\]

\[
\{ I + m_c h (h + f) \} \ddot{\psi} - m_c h (\dot{v} + V \dot{r} + f \dot{\phi}) = aF_{y1} - bF_{y2} - hF_{y3} \tag{4.13}
\]

\[
(1 + m_c f^2) (\ddot{\phi} - \ddot{r}) + m_c f (\dot{v} + V \dot{r} - h \ddot{r}) = gF_{y3} \tag{4.14}
\]

The forces $F_{y1}, F_{y2}, F_{y3}$ can also be expressed in terms of motion variables, which are:

\[
F_{y1} = C_1 \alpha_1 = C_1 \delta - C_1 \frac{1}{u} (v + ar) \tag{4.15}
\]

\[
F_{y2} = C_2 \alpha_2 = -C_2 \frac{1}{u} (v - br) \tag{4.16}
\]

\[
F_{y3} = C_3 \alpha_3 = -C_3 \left( \frac{1}{u} (v - hr - g(r - \phi)) + \phi \right) \tag{4.17}
\]
Substituting (4.15), (4.16), (4.17) in (4.12), (4.13), (4.14) gives the following coupled set of differential equations:

\[ m_f \dot{\phi} + \frac{1}{u} C_g g \dot{\phi} + (m + m_g) \ddot{v} - m_e (h + f) \ddot{\phi} + C_f \phi + \frac{1}{u} (C_i + C_z + C_u) v \ldots \]
\[ + \left\{ (m + m_e) u + \frac{1}{u} (a C_i - b C_2 - h C_3 - g C_3) \right\} = C_i \delta \]  

(4.18)

\[ -m_h f \ddot{\phi} - h g C_3 \frac{1}{u} \dot{\phi} - m_e h \ddot{v} + \left\{ I + m_f (h + f) \right\} \ddot{r} - h C_i \phi + \frac{1}{u} (a C_i - b C_2 - h C_3) v \ldots \]
\[ + \left\{ \frac{1}{u} \left( a^2 C_i + b^2 C_2 + h^2 C_3 + h g C_3 - m_e h u \right) \right\} r = a C_i \delta \]  

(4.19)

\[ (I_e + m_f f^2) \ddot{\phi} - g C_i \frac{1}{u} \phi + m_e f \ddot{v} - \left\{ I_e + m_f \left( f + h \right) \right\} \ddot{r} - g C_i \phi + g C_i \frac{1}{u} v \ldots \]
\[ + \left\{ m_e f u - \frac{1}{u} \left( g h C_3 + g^2 C_3 \right) \right\} r = 0 \]  

(4.20)

As explained before, in steady state cornering conditions \( \phi, \dot{\phi}, \ddot{v}, \dot{r} \) are all zero so (4.18), (4.19), (4.20) simplify to:

\[ C_s \phi + \frac{1}{u} (C_i + C_z + C_u) v + \left\{ (m + m_e) u + \frac{1}{u} (a C_i - b C_2 - h C_3 - g C_3) \right\} r = C_i \delta \]  

(4.21)

\[ -h C_i \phi + \frac{1}{u} (a C_i - b C_2 - h C_3) v + \left\{ \frac{1}{u} \left( a^2 C_i + b^2 C_2 + h^2 C_3 + h g C_3 - m_e h u \right) \right\} r = a C_i \delta \]  

(4.22)

\[ -g C_i \phi + g C_i \frac{1}{u} v + \left\{ m_e f u - \frac{1}{u} \left( g h C_3 + g^2 C_3 \right) \right\} r = 0 \]  

(4.23)

After elimination of \( \phi \) and \( v \) these equations reduce to:

\[ \frac{1}{u} \left\{ \frac{1}{u} [l^2 C_i C_2 + 2 C_i C_3 \left\{ h (h + g + 2a) + a (a + g) \right\} - C_2 C_3 \left\{ h (2b - 2h - 2g + l) - 2 b^2 + g l \right\} \ldots \right. \]
\[ - m u^2 (a C_i - b C_2 - 2 h C_3) - m u^2 \left\{ C_i (h + a + \frac{f}{g} + a \frac{L}{g}) + C_z (h - b + \frac{h}{g} - b - \frac{L}{g}) \right\} \right\} r \]  

(4.24)

\[ = \{ I C_i C_2 + (2a + 2h) C_i C_3 \} \delta \]

In comparison with (4.5) in steady state conditions, it is easy to see that a tractor semi-trailer is far more complex than the bicycle model of a normal vehicle. In fact, the situation will even become more complex since (4.24) is derived for a model with only one trailer axle. The effect of multiple non steering axles on the steady state handling performance of a vehicle was studied by Christopher B. Winkler [14], [15]. He concluded that the behaviour of a vehicle with multiple non steering rear axles can be compared to the same vehicle with only one non steering rear axle and an equivalent wheelbase. In [15] he stated that the same theory can be applied to a multi axle semi-trailer.
To give an idea of the increased complexity of a tractor semi-trailer with three trailer axles, the theory of equivalent wheelbase, as studied by Winkler, will be explained in two steps. First, the interpretation of an equivalent wheelbase will be proven by a simple bicycle model with two non steering rear axles, see Figure 4.10. Second, a possible implementation of this theory for a tractor semi-trailer is given.

![Bicycle model of a vehicle with two non steering rear axles.](image)

For a vehicle with multiple rear axles (4.10) can be written as:

$$\delta - \frac{l}{R} = \frac{T}{lR} \left( 1 + \frac{C_{ar}}{C_{af}} \right) + a_y \eta$$  \hspace{1cm} (4.25)

Where,

- $T = \sum_{i=1}^{N} \frac{\Delta_i^2}{N}$ is the tandem factor to account for the number of rear axles.
- $N$ is the number of non steering rear axles.
- $\Delta_i$ is the longitudinal distance from the aft end of $l$ to the $i^{th}$ rear axle.
- $C_{ar}$ is the sum of the cornering stiffnesses of all rear tyres.
- $C_{af}$ is the sum of the cornering stiffnesses of all front tyres.
In [14] it is shown that the vehicle with multiple non steering rear axles remains stable at higher velocities than it would if it had the same front and rear compliances, but only one rear axle. In other words, multiple non steering rear axles have a stabilizing effect on the vehicle. It behaves as if it had a longer “equivalent wheelbase” than its actual wheelbase. To explain the idea of equivalent wheelbase rearrange (4.25) as follows:

\[
\delta - \frac{l}{R} = a, \eta \tag{4.26}
\]

Where \( l_e \) represents the equivalent wheelbase of a vehicle with the same dynamic behaviour as the multi-axle vehicle with wheelbase \( l \) and is given by:

\[
l_e = l \left\{ 1 + \frac{T}{l^2} \left( 1 + \frac{C_{af}}{C_{ar}} \right) \right\} \tag{4.27}
\]

To prove this interpretation, consider the vehicle shown in Figure 4.11, which is in steady state turn at very low speed such that \( a_y \approx 0 \). According to the requirements of static equilibrium of lateral force and yaw moment, the tire forces must develop with the polarities as indicated in the figure. In this condition the side force of the centre axle acts in opposite direction to the side forces at the front and rear axle, so the magnitude of \( F_{y2} \) equals the sum of \( F_{y1} \) and \( F_{y3} \).

The two dimensions \( a \) and \( b \) are defined as follows.

- \( a \) is the distance ahead of the front tyre at which an imaginary tyre, steered to the same angle as the real front tyre, would have to be located to experience zero side slip.
- \( b \) is the distance aft of the centre of the rear suspension where an imaginary, non steering tyre would have to be to experience zero slip.

The two imaginary tyres are the tyres of an equivalent vehicle which would experience the same low-speed turn as the vehicle with multiple rear axles, given the same steer angle. The equivalent wheelbase is:

\[
l_e = a + l + b \tag{4.28}
\]

From the requirements of static equilibrium:

\[
F_{y1} + F_{y2} + F_{y3} = 0 \tag{4.29}
\]

\[
F_{y1}(l - \Delta) = F_{y3}(2\Delta) \tag{4.30}
\]

And for the geometry, assuming all angles are small:

\[
-\alpha_1 = a / R \tag{4.31}
\]

\[
\alpha_2 = (\Delta + b) / R \tag{4.32}
\]

\[
-\alpha_3 = (\Delta - b) / R \tag{4.33}
\]

\[
\delta + \alpha_1 = (a + l) / R \tag{4.34}
\]

\[
\delta = (a + l + b) / R \tag{4.35}
\]
The equivalent rear wheel, located where the turn geometry results in zero slip.

The equivalent front wheel, located where the steer angle, $\delta$, and turn geometry result in zero slip.

Figure 4.11: Simplified model of a three-axle vehicle in a low-speed turn. [14]

And for $C_{af}$ and $C_{ar}$:

$$F_{y1} = C_{af} \alpha_1$$ \hspace{1cm} (4.36)

$$F_{y2} = \frac{1}{2} C_{ar} \alpha_2$$ \hspace{1cm} (4.37)

$$F_{y3} = \frac{1}{2} C_{ar} \alpha_3$$ \hspace{1cm} (4.38)

Substituting (4.31), (4.32), (4.33) into (4.36), (4.37), (4.38) and substituting the result into (4.29) gives:

$$a C_{af} = b C_{ar}$$ \hspace{1cm} (4.39)

Equivalently (4.30) can be rearranged to:

$$a(l - \Delta) C_{af} = -\Delta(b - \Delta) C_{ar}$$ \hspace{1cm} (4.40)

Substituting (4.39) into (4.40) yields:

$$b = \frac{\Delta^2}{l^2}$$ \hspace{1cm} (4.41)
Substituting (4.41) into (4.39) yields:

\[ a = \left( \frac{C_{ar}}{C_{af}} \right) \left( \frac{\Delta^2}{T^2} \right) \]  

(4.42)

Then, with (4.28), (4.41) and (4.42) the equivalent wheelbase is:

\[ l_e = l \left[ 1 + \frac{T}{l^2} \left( 1 + \frac{C_{ar}}{C_{af}} \right) \right] \]  

(4.43)

and with the tandem factor \( T \) where \( N = 2 \) and \( \Delta_1 = \Delta_2 \):

\[ l_e = l \left[ 1 + \frac{\Delta^2}{l^2} \left( 1 + \frac{C_{ar}}{C_{af}} \right) \right] \]  

(4.44)

Finally, with the equivalent wheelbase explained the same theory can be used to calculate the equivalent wheelbase of a multi-axle semi-trailer. Figure 4.12 shows model of a tractor semi-trailer with three trailer axles in a slow speed turn.

\[ g_e = g \left[ 1 + \frac{T}{g^2} \left( 1 + \frac{C_{ar}}{C_{af}} \right) \right], \text{ and } N = 3 \]  

(4.45)
Simulation results

The choice is made to drive a circle with a radius of 40 m because it is possible to drive this circle at the DAF test track at St. Oedenrode (NL). If in a later stage, reference measurements will be made, these can be directly compared to the model results. Besides, one steering characteristic has already been measured on a vehicle that is comparable to the model also driving a circle with a radius of 40 m. Since the vehicle model has not been equipped with a driver model that can automatically steer to follow a certain curve, the circle test has to be completed with an iterative approach.

The basic idea behind this approach is first to find out what steering angle is required to drive a circle with a radius of 40 m at a certain speed. This circle is driven in stationary conditions to find the corresponding yaw velocity and lateral acceleration. When this is done with sufficient variation in speed, it will be possible to create a steering characteristic of the vehicle. To determine the required steering angle, several simulations will be made where the speed is kept constant between 10 and 80 km/h and the steering angle continuously increases from 0 to 10 deg. Figure 4.13 shows some basic results for one of these tests at a speed of 40 km/h.

Since the vehicle is driving along a spiral curve this test is called the spiral test. Next, the theoretical yaw velocity is calculated, according to (4.46). This is the yaw velocity a vehicle will have when driving the 40 metre circle at a certain speed, see Figure 4.14.
From Figure 4.14, one can choose a certain speed and the corresponding yaw velocity. With this information and the results of the spiral test it is possible to determine the required steering angle for the circle test. Figure 4.15 shows both the yaw velocity and the lateral acceleration against the steering angle as a result of the spiral test.

The right graph of Figure 4.15 shows that the vehicle reaches a maximum lateral acceleration around 4.6 m/s². Beyond this point the vehicle either starts sliding, or will simply roll over. For example, to determine what steering angle is required to drive a circle with a radius of 40 m at a speed of 40 km/h. With (4.46) it can be calculated that the theoretical yaw velocity is 15.9 deg/sec. In Figure 4.15 the corresponding steer angle of 6.07 deg and a lateral acceleration of 3.08 m/s² are determined. Although this information is sufficient to determine the steering characteristic, a steady state circular test is performed to check the results and to see if the vehicle is indeed driving a circle with a radius of 40 m. The results for the steady state circle test for V=40 km/h are shown in Figure 4.16.
The figures confirm that the vehicle is driving the desired circle and all signals reach a steady state condition. These circle tests are carried out for 10, 15, 20, 25, 30, 35, 40 and 45 km/h to create a steering characteristic for the standard vehicle. Figure 4.17 shows the results for these simulations.
The top left plot of Figure 4.17 shows the lateral acceleration versus the change in steering angle $\delta$, for both the model and measurement. The right plot shows the g-force versus the total steering angle for the model. From these results two conclusions can be drawn, the first is that the simulation results are quite comparable to the measurement results [7]. This makes the results for the model more trustworthy even though there are some differences to be noticed. Taking in account the model simplicity and the fact that the measurement conditions are not known exactly, this difference is acceptable. The second conclusion that can be drawn is that the vehicle is slightly understeered. This can be concluded from the fact that the steering angle has to increase in order to keep the vehicle on the desired circle for increasing acceleration levels.

The three figures at the bottom of Figure 4.17 show the side force $F_y$, the vertical tyre load and the slip angle for all truck tyres. Since the vehicle is understeered one would expect the slip angles, left plot, of the front wheels, to be larger than those of the rear wheels. It looks as if the vehicle is oversteered at low speeds and then turns into an understeered vehicle when the speed is increasing. According to (4.4) it can be concluded that the cornering stiffness is changing during the manoeuvre. As already explained, the cornering stiffness depends on the vertical load $F_z$. For the circle test these loads are depicted in the middle plot at the bottom of Figure 4.17. If the loads for each tyre are added to see the total axle load, as is done in Figure 4.18, it can be seen that with increasing speed the load of the front axle decreases. At the same time, the load of the rear axle increases with the same amount. This effect is caused by the roll moment of the trailer and the jack angle between truck and trailer. In a cornering event the longitudinal centre line of the truck and trailer are under an angle, in this situation the trailer roll moment causes some lift of the front axle.

So if the total vertical load of the front axle is decreasing, then also the cornering stiffness of the front tyres and with it the side force will decrease. This can only be compensated with a larger slip angle and therefore also a larger steering angle.
Besides the fact that the balance between the vertical load of the front and rear axle is changing, there is a second effect that causes a decrease in the total side force of an axle. This can be explained by looking at the curves for the cornering stiffness as presented in chapter 3. This curve is non-progressively increasing which means that with an increase of the vertical load, the increase of the cornering stiffness will not be as large as when a decrease of the vertical load will also decrease the cornering stiffness. To illustrate the effect, a theoretical example of the steer axle with a vertical load of 30 kN for each tyre is assumed. As a result of roll, the balance in vertical load will change. Table 4.1 shows the effect of the change in vertical load on the cornering stiffness of the tyres.

<table>
<thead>
<tr>
<th>Roll (%)</th>
<th>0</th>
<th>16.5</th>
<th>33</th>
<th>49.5</th>
<th>66</th>
</tr>
</thead>
<tbody>
<tr>
<td>Left Tyre</td>
<td>Fz [N]</td>
<td>30000</td>
<td>25000</td>
<td>20000</td>
<td>15000</td>
</tr>
<tr>
<td></td>
<td>Cfa [N/deg]</td>
<td>3278</td>
<td>2799</td>
<td>2284</td>
<td>1741</td>
</tr>
<tr>
<td>Right Tyre</td>
<td>Fz [N]</td>
<td>30000</td>
<td>35000</td>
<td>40000</td>
<td>45000</td>
</tr>
<tr>
<td></td>
<td>Cfa [N/deg]</td>
<td>3278</td>
<td>3715</td>
<td>4115</td>
<td>4476</td>
</tr>
<tr>
<td>Total Axle</td>
<td>Fz [N]</td>
<td>60000</td>
<td>60000</td>
<td>60000</td>
<td>60000</td>
</tr>
<tr>
<td></td>
<td>Cfa [N/deg]</td>
<td>6556</td>
<td>6514</td>
<td>6399</td>
<td>6217</td>
</tr>
</tbody>
</table>

Table 4.1: Theoretical change of the vertical load and the cornering stiffness.

Table 4.1 shows that if the balance of the vertical load on an axle changes, the cornering stiffness and the side force will also change. In cases of both the theoretical example of Table 4.1 and the simulation results of truck model, the cornering stiffness and the side force will decrease as an effect of vehicle roll. This again, needs to be compensated with a larger slip angle.

In Figure 4.17 it is noticed that the vehicle is understeered. However, for future reference it may be useful to know how much the vehicle’s steering characteristic differs from a neutral steered vehicle. Therefore the steady state yaw velocity gain, $K_s$, is introduced as:

$$K_s = \frac{r}{\delta} \quad (4.47)$$

For a neutral steered vehicle this curve will increase linearly, since the steering angle will remain constant. For an oversteered vehicle the curve will bend upwards from the neutral curve and for an understeered vehicle the curve will bend downwards from the neutral curve. Figure 4.19 shows the steady state yaw velocity gain for the truck model.
Figure 4.19 shows the yaw velocity gain for the initial truck model and the theoretical yaw velocity gain if the model would have been neutral steered. As expected the vehicle is only slightly understeered, which is indicated by the fact that the deflection of the yaw velocity gain from the neutral curve is only small. Only at the end, the curve reaches a maximum value at a speed of 40 km/h. This however, can be explained by the fact that some of the left wheels reach a maximum side force as a result of a very low vertical load.

### 4.3.2 Step Steer

A well known method for analysing the response of a dynamic system is a step response, or in this case the response of the vehicle to a step steer input. Since the SimMechanics truck model is a continuous time model with numerical time integration, it is not possible to perform a real step. As an alternative the choice is made to create a fast steer input with a steer angle of 1 degree in 0.1 second. The step could even be faster since the iteration step is 0.01 second but this may cause calculation errors due to large changes in tyre forces and model response, so a step of 0.1 second is chosen.

As described in chapter 2 the three signals that describe the dynamic behaviour of the system are the lateral acceleration \(a_y\), the yaw velocity \(r\) or \(\dot{\psi}\) and the roll angle \(\phi\). These signals will initially start at zero and will end at a constant value when the vehicle is driving a stationary circle. In between these stationary conditions they will describe the exact motion of the vehicle. Figure 4.20 shows the results for the step response at various driving speeds.
Figure 4.20: Step steer response for a steer wheel angle of 30 degrees (1 degree steer angle) at 8 different speeds.

The two figures at the top of Figure 4.20 show the lateral acceleration of the front axle and the cabin. The first result that immediately draws the attention is the peak at the start of the curve that is completely analogous to the step steer input. This can easily be explained by the tyre side slip forces as depicted in Figure 4.21.

Figure 4.21: Tyre side forces for a step at 80 km/h.
The front tyres show the same peak as for the lateral acceleration. The explanation for this is that when a step steer is applied to the front wheels, the vehicle will still move in a straight line. So at this point the steer angle will equal the tyre side slip angle explaining the peak in tyre side force. After the step is applied the vehicle will start moving sideward and the tyre side forces will reach their final value.

For the lateral acceleration of the cabin the result is slightly different because the cabin will also have a dynamic roll effect during the step response. The roll behaviour of the cabin can be seen on the bottom right of Figure 4.20. Generally spoken the roll behaviour can be explained the same way as for the lateral acceleration. Only this time the excitation starts a fraction later since roll is the effect of a lateral acceleration. Although the fluctuations in the response signal after the initial step may seem trivial, it can be of interest to see if the trailer has any influence on the behaviour. Therefore the step steer of 80 km/h has been performed two additional times. One time with a solid weight mounted on the truck to simulate the trailer weight but not the trailer dynamics and one time without the trailer at all. The results of these tests for the roll angle are shown in Figure 4.22.

![Figure 4.22: Step response at 80 km/h for the original vehicle, the truck without trailer and with only the trailer load.](image)

Apparently the fluctuations are larger when the truck is manoeuvring without the trailer, although the total roll angle is larger with the trailer present. The explanation for this result can be found in the fact that the trailer is able to transfer its roll moment to the truck. Especially since the weight of the trailer is dominating the weight of the total vehicle combination, this moment will overrule some of the truck dynamics.

### 4.3.3 Sine Sweep

In chapter 2 it was explained that the combination of a driver and vehicle can be seen as a closed loop control system, with the driver as the controller and the vehicle as the controllable dynamic system. In this system, the vehicle reacts to a steer input of the driver and the response signals of the vehicle will inform the driver about the quality and quantity of the action. With this feedback system in mind it seems quite obvious that when the research is focussing on the steering behaviour of the vehicle, the transfer functions from input and output signals should contain all the necessary information.
To determine these transfer functions a sine sweep is performed at a constant speed. This sine sweep is created by a sinusoidal steer input of which the frequency will increase over time:

\[ \delta = \frac{\delta_{sw\text{ max}} \sin(0.05\pi t^2)}{\text{steer ratio}} \]  \hspace{1cm} (4.48)

Where \( \delta \) represents the steer input, \( \delta_{sw\text{ max}} \) represents the maximum steer wheel input (30 degrees), \( t \) represents the time and the steer ratio is the ratio between the steer wheel angle and the steer input (in this case 30). With a simulation time of 250 seconds and a step size of 0.01 second, the frequency of the sinusoidal steer input will gradually increase, creating a well defined transfer function between 0.01 and 10 Hz. With the steer signal being the only variable input of the vehicle, the following transfer functions are of interest to examine the steering behaviour.

- Steer input to lateral acceleration \( (a_y) \)
- Steer input to yaw velocity \( (r \ or \ \dot{\psi}) \)
- Steer input to roll angle \( (\phi) \)

In order to get a complete picture of the dynamic behaviour of the vehicle, these transfer functions are determined on several locations of the vehicle. Figure 4.23 shows the results for the transfer function of the lateral acceleration, measured at the front axle, truck chassis, driver seat in the cabin and on the trailer chassis.

Figure 4.23: Transfer function, steer angle to lateral acceleration.
Although the transfer functions probably contain most information about the dynamic behaviour of the vehicle, they are also very hard to analyse. Therefore, the analysis will be limited to describe the motions of the vehicle to try and understand what is really happening. The transfer function of the lateral acceleration as depicted in Figure 4.23 can roughly be divided into four sections.

- **< 0.1 Hz**: For steering activity below the frequency of 0.1 Hz the entire vehicle responds with the same magnitude and phase. Or, in other words, with very slow steering the vehicle response is steady state since there is enough time for all components to react.

- **0.1 – 0.5 Hz**: In this range the magnitude and phase of all the truck’s components are slightly descending, which indicates that the steering activity becomes faster and the truck response will have a small delay on the input. However, for the truck the delay is still quite small and all components are still in phase. The delay of the trailer starts to become significantly larger than the delay of the truck.

- **0.5 – 3 Hz**: This is the most interesting frequency range for two reasons. First, it is the range with a lot of dynamic activity. Second, most of the steering activity will take place in this range. As can be seen in Figure 4.23 the motion of the trailer is rather small within this frequency range. Apparently this frequency is too fast for the trailer mass to follow. Then there is a lot of activity going on for the truck axle, chassis and cabin. First look at the magnitude of these signals. Although, at first sight, they may look very different, there is a resemblance for each of these curves. They all reach a maximum and minimum after each other and after the minimum they will rise again to a certain maximum value. Even though the magnitudes are very different, which troubles the resemblance of the curves, this behaviour seems to be very consistent. Looking at the phase of the same signals between 1 and 2 Hz, the phase of the cabin is opposite to the phase of the front axle and chassis. Both effects can be explained by the fact that the virtual roll centre of the vehicle is moving up. As explained by the theoretical drawing in Figure 4.24, a virtual roll centre is the theoretical centre of rotation. This point is theoretical because in practice it is not truly a fixed point in space but will move sideward along with the vehicle. As a result of the tyre forces the front axle is moving from left to right at a certain frequency. At approximately 1.1 Hz, (situation 1) the virtual roll centre will lie close to the front axle creating a larger swing for the chassis and cabin. When the frequency is increased to 1.2 Hz, (situation 2) the virtual roll centre will lie close to the chassis and therefore the movement of the front axle and cabin will be larger. Finally, at around 2 Hz, (situation 3) the virtual roll centre will lie somewhere in the bottom half of the cabin which results in larger movements of the front axle and chassis. Because the virtual roll centre lies between the origin of the motion (front axle) and the driver seat, the situation will occur that the motion or phase of the driver seat is opposite to the motion of the front axle and chassis.

- **> 3 Hz**: For two main reasons the steering activity above 3 Hz is no longer interesting. First of all, steering behaviour in this frequency range will be unrealistic and second, the dynamic behaviour of the vehicle will decrease as a result of mass inertia. As can be seen there is some extra dynamic behaviour around 8 Hz, but this is purely caused by the vibrating modes of the engine block.
Figure 4.24: Theoretical movement of the virtual roll centre.

The transfer function for the yaw velocity is presented in Figure 4.25.

Figure 4.25: Transfer function, steer angle to yaw velocity.

Again the response function will be divided into four different sections.

- **< 0.1 Hz**: As explained with the lateral acceleration, steering activity in this frequency range is too slow to see any dynamic activity.
- **0.1 – 0.9 Hz**: In this range the first delay in the response signal is visible. Although it is still quite small for the tractor, the yaw velocity of the trailer almost completely disappears.
• **0.9 – 2 Hz**: In this specific area two different phenomena can be distinguished. First of all, there is quite a large dip in the yaw velocity of the tractor. This can be explained by the fact that the movement of the trailer is almost negligible, which also limits the rear end movement of the tractor. At the front end, the motion of the cabin is opposite to the motion of the chassis, which reduces the movement of the front end. These two effects together result in a reduced yaw velocity around 1.7 Hz. The second phenomenon that can be seen is that the magnitude and phase between the cabin and the chassis are no longer the same. In other words, the cabin suspension has some extra yaw activity above 1.2 Hz.

• **> 2 Hz**: The magnitude of the yaw velocity will continuously decrease for frequencies above 2 Hz. But as explained with the lateral acceleration this range is of less interest since steering activity at these frequencies is no longer realistic.

The transfer function for the roll angle is presented in Figure 4.26.

In general, the roll angle transfer function can also be divided in the same four sections as explained before. Since some of the effects are already discussed with the lateral acceleration and the yaw velocity the explanation for the roll angle will only be brief.

- **< 0.1 Hz**: The initial difference in magnitude between the chassis and the cabin is a direct result of the cabin suspension roll stiffness.
- **0.1 – 0.5 Hz**: A slight descend in both the magnitude and phase for all signals, just as can be seen for the lateral acceleration and the yaw velocity.
- **0.5 – 2 Hz**: Between 1 and 2 Hz the magnitude of the roll angle reaches a peak, which is caused by the roll eigen frequency and the fact that the movement of the trailer damps out. The latter effect results in a higher relative movement of the front axle and therefore an increase in roll angle.
- **> 2 Hz**: Above 2 Hz the roll amplitude decreases and does no longer play an important role in the steering behaviour.
Now the basic transfer functions and with it, the dynamic behaviour of the truck are known, it is interesting to see if anything changes when one of the input signals is varied. To do this the sine sweep has been performed again for the speeds of 40 and 60 km/h and for a steer input of 60 degrees. The results for the lateral acceleration are shown in Figure 4.27.

As can be seen on the left side of Figure 4.27, the variation in speed has a large impact on the transfer function below 0.6 Hz. Meanwhile, the effect of speed on the transfer function above 0.6 Hz, is of far less importance or can even be neglected. On the right side of Figure 4.27, the result for a maximum steer input of 30 and 60 degrees is given, but this does not seem to have any effect on the transfer function.

Finally a few remarks as a general result of the transfer functions can be made.

- For steering activity below 0.7 Hz, speed appears to be the parameter with the largest influence.
- For steering activity in the range between approximately 0.7 to 3 Hz, system dynamics appears to be the parameter with the largest influence.
- Steering activity above 3 Hz generally seems to be frequency dependent but at the same time less relevant.
- Up until approximately 1 Hz the yaw velocity appears to be the dominating feedback signal for the driver, while above 1 Hz roll becomes more dominating.
- The magnitude of the steer angle, used to perform a certain manoeuvre, does not influence the dynamic response, but only the stationary conditions.
4.4 Conclusions for simulation results of the initial model

In the previous section, the dynamic and steering behaviour of the initial model was studied by the simulation results of three different types of manoeuvres. For clarity, the most important comments that are made with respect to the steering behaviour are listed in this section.

- From simulation results of the circle test it was learned that the maximum allowable lateral acceleration of the vehicle lies at 4.6 m/s² while the vehicle is continuously understeered. It was shown that during a steering event the jack angle between truck and trailer and the roll moment of the trailer causes a slight lift effect on the front axle. This effect will have a progressive effect on the understeered behaviour of the vehicle.

- Simulating a step steer and studying the dynamic response of the cabin showed that there is quite a lot of dynamic activity before steady state conditions are reached. This response will most certainly influence the driver’s perception of the steering behaviour. From a closer study, it was shown that two elements are of great influence to the response of the vehicle:
  - The steer tyres generate an initial side force almost simultaneously with the steer input causing an immediate lateral acceleration of the cabin.
  - The trailer roll moment is of great influence on the dynamic response of the truck and therefore also on the steering behaviour.

- Simulation of a sine sweep makes it possible to generate transfer functions which give a lot of information about the dynamic behaviour of the vehicle. Even though the dynamic response of the vehicle is sometimes hard to translate into actual steering behaviour, it is still possible to subdivide steering activity into four categories.
  - < 0.1 Hz: Steering activity is to slow to have any effect on the steering behaviour. All components move in phase. Only the vehicle's speed might influence the response.
  - 0.1 – 0.5 Hz: Transitional phase where truck components start moving in different phase. In general, the steering behaviour will not really be influenced since steering activity is still relatively slow. Most important change within this range is the response of the trailer which will strongly decrease.
  - 0.5 – 3 Hz: For steering activity within this frequency range there is a lot of dynamic response from several of the vehicles components, and some eigenfrequencies can be detected. As a result, the steering behaviour will be influenced to a high degree. Simulation results show that there are often large phase shifts between components like the front axle, chassis and suspension. These phase shifts will almost certainly lead to a confusing feedback for the driver.
  - For two reasons steering activity above 3 Hz is no longer of interest to study steering behaviour. First of all, dynamic behaviour of the vehicle will strongly decrease. Second, steer inputs above 3Hz are less realistic and will usually not occur.

These conclusions can be made by studying simulation results of the standard vehicle. Unfortunately no real time test data is available so it remains hard to actually quantify results with regard to steering behaviour. Therefore, to improve insight, a different approach will be discussed in the next chapter.
5 Variation of model parameters

In the previous chapter, an extensive description of the vehicle model has been given. Second, a set of virtual tests, suitable to study the steering behaviour of a vehicle, was determined. These tests give a lot of results and improve the knowledge about the dynamic behaviour of a truck and semi-trailer. However, it remains quite hard to actually give any quantification to this steering behaviour. Especially, since there is no reference material available, like measurements, that can be used to quantify virtual results.

Therefore it is helpful to find a method that can improve the knowledge about truck dynamics and at the same time gives a better insight in how to quantify the results. The choice is made to study the effects parameter variations have on the steering behaviour of the vehicle. The parameters varied and simulation results are presented in this chapter.

5.1 Parameter selection

In a truck and trailer combination, there is almost an endless amount of variables that can be changed. Although the model is much simpler than a real truck, there are still a few hundred variables that can be changed. Each variable is of potential interest to study its effects on the dynamic behaviour. To prevent this study from escalating and creating an overwhelming amount of data, it is important to wisely choose a limited set of parameters that create significant results.

Through several conversations with experienced (test)-engineers at DAF Trucks, the following set of parameters came up to be known as potentially influential to the steering behaviour.

- Cabin suspension: From past results it is known that decreasing the yaw and roll stiffness of the cabin suspension, has a negative influence on the steering behaviour. Especially the judgement of the driver will be influenced.
- Truck suspension: Obviously, the stiffness of the truck suspension system will influence the steering behaviour.
- Geometric layout of the suspension system: The geometric layout of the suspension system can also influence the system dynamics. This geometric layout includes suspension travel, location of the suspension mounts and other geometric details. Unfortunately, the model contains only limited geometric information, and therefore the possibilities to study this parameter are limited.
- Suspension flexibility: It is common knowledge that flexibility of suspension components also affects the dynamic behaviour of a vehicle. For example, leaf springs can cause the front axle to move backwards when the truck drives over an obstacle. This can result in a small steer angle because the steering rod does not move at the same time. The effect is also known as bump-steer. Furthermore, a second effect called roll-steer can occur when the left and right side of the suspension are not equally pressed. This will result in a misalignment of the axle and chassis, because one side of the suspension will have a larger longitudinal displacement. However, at this moment these effects are not yet implemented in the model and will therefore not be studied.
• Chassis torsion stiffness: Due to heavy load conditions and a relatively high centre of mass, the truck experiences quite large roll moments. These moments are passed over to other sections of the truck through the chassis. So, the chassis stiffness will play an important role in how the roll moment is divided over the vehicle. Eventually, the roll moment will influence the vertical tyre load and therefore also the steering behaviour. Although the implementation of chassis flexibility in the vehicle model is simplified, it should be enough to study the effects on the steering behaviour.

• Tyre cornering stiffness: As already explained in chapter 3, tyres are one of the most important components in vehicle dynamics. However, it was also explained in chapter 3 that the tyre model has a lot of parameters that can be changed. Since this study is focussing on the steering behaviour, changing the model parameters for the cornering stiffness should have the most significant influence.

• Roll stiffness trailer suspension: The total weight of the truck and trailer combination is dominated by the weight of the trailer load. This means that in case of extreme cornering, the trailer is the decisive factor in keeping the combination upright. With this in mind it is interesting to see the influence of the trailer suspension roll stiffness on the dynamic behaviour of the entire vehicle.

• Trailer weight distribution: Weight distribution is also known to be an important factor in vehicle dynamics. Especially in truck dynamics where, as described above, the trailer weight is dominating the total vehicle weight.

From these issues, the following parameters have been chosen for this study:

1. Cabin suspension roll stiffness
2. Truck suspension roll stiffness
3. Trailer suspension roll stiffness
4. Truck chassis torsion stiffness
5. Trailer chassis torsion stiffness
6. Longitudinal weight distribution
7. Vertical weight distribution
8. Cabin yaw stiffness
9. Tyre cornering stiffness

The parameter values of the initial model, as described in chapter 4, are standard or original values of 100%. All stiffnesses will be varied with 40, 70, 130 and 160% of the original value. For the longitudinal weight distribution, the front, mid-, and rear loaded trailer are distinguished. For the vertical weight distribution, the centre of gravity of the trailer load will vary with 0.25, 0.5, 0.75, 1 and 1.25 m above the trailer floor.

In the following sections, simulation results for each parameter will be discussed. For each simulation, one figure represents the most significant result. Otherwise the discussion would end up with a huge number of figures. At the end of each paragraph the main conclusions will be given. The results for the roll stiffnesses, the chassis torsion stiffnesses and the trailer weight distribution are combined in section 5.2, 5.3 and 5.4. The cabin yaw stiffness is presented in section 5.5 and the tyre cornering stiffness in section 5.6.
5.2 Roll stiffness

In this section, variations of the cabin, truck and trailer roll stiffnesses are discussed. For each simulation, the most significant result is presented in a graph followed by a brief comment. At the end, a short summary of main conclusions will explain the general influence of roll stiffness.

5.2.1 Cabin suspension roll stiffness

Circle test

The left graph of figure 5.1, shows the steering characteristics for 40, 70, 100, 130 and 160% of the cabin roll stiffness.

As expected, changing cabin roll stiffness does not change the steering characteristics of the vehicle.

Step steer

The right graph of figure 5.1, shows the cabin roll angle for a step response at 80 km/h and 1 degree steer angle. A few remarks on the step response can be made.

- The effect of decreasing the roll stiffness is larger than increasing the roll stiffness. Apparently, the original roll stiffness is already quite high, so further increase has less effect.
- At 5.5 seconds, the response shows some fluctuation. This can be explained by the tyre forces that show the same fluctuation for a step steer as explained in paragraph 4.3.2. It can be noticed that for low cabin roll stiffnesses the fluctuation moves to higher values, but this effect is expected since low roll stiffness induces more roll.
- Between 6 and 8 seconds there is again a small fluctuation of the response. This time, the magnitude and frequency of oscillation do change so it is probably caused by the cabin roll stiffness. Although these effects can not be seen in the steering characteristics, it is quite likely that they will distort the drivers' perception of the steering behaviour.
Sine sweep
Figure 5.2 shows the transfer functions of the cabin roll angle for all roll stiffnesses.

Changing the cabin roll stiffness, results in a change of the cabin roll eigenfrequency. This is shown by the small shift of the magnitude and phase between 0.7 and 2 Hz. The change of cabin roll eigenfrequency also proves that the feedback for the driver will change. However, at this point it is hard to give any quantification.

5.2.2 Truck suspension roll stiffness

Circle test
The left graph of figure 5.3, shows the steering characteristics for 40, 70, 100, 130 and 160% of the suspension roll stiffness. The changes are applied to both the front and rear wheel suspension at the same time.

Figure 5.3: Left, Steering characteristics for the vehicle with a variation in roll stiffnesses.
Right, Step response, cabin roll angle for variable suspension stiffnesses.
Unlike the cabin roll stiffness, the suspension roll stiffness does have a significant influence on the steering behaviour. As can be seen in figure 5.3, decreasing the roll stiffness results in the vehicle becoming more understeered. This can be proven by looking at the vertical load distribution between the front and rear axle of the truck. Figure 5.4 shows the vertical load of the front and rear axle for the standard vehicle and for the truck with 40% suspension roll stiffness. As can be seen, the vertical load of the rear axle slightly increases while the load of the front axle remains the same. As a result, the slip angle of the rear axle decreases and the vehicle becomes more understeered. A closer look at the vertical load distribution of the entire vehicle learns that some of the trailer weight is transferred from the trailer axles to the rear axle of the truck. Normally, this could also result in more oversteer due to the required increase in side force for a heavier loaded rear axle, as will be explained later on. However, in this case the changes in vertical load are quite small and only result in a decreased slip angle.

![Vertical load distribution](image.png)

Figure 5.4: Distribution of vertical load for the standard vehicle and 40% suspension roll stiffness

**Step steer**

The right graph of figure 5.3, shows the cabin roll angle for a step response.

The fact that most of the vehicle’s roll originates from the suspension, is immediately highlighted by the large difference in roll angle. In the previous section, it was explained that the cabin roll stiffness does not influence the fluctuation of the response at 5.5 seconds. However, the truck suspension does influence the oscillation, as can be seen in figure 5.3. When the suspension stiffness is 40%, the oscillation has almost disappeared, while a stiffness of 160% results in a very large oscillation. This can be explained quite easy by the fact that low suspension stiffness will absorb most of the step of the tyre forces as explained in paragraph 4.3.2.
Sine sweep

Figure 5.4 shows the transfer function of the cabin roll angle.

The influence of the suspension stiffness is also shown by the change in magnitude of the transfer function. However, the most important result can be seen between 0.8 and 2 Hz. The roll eigenfrequency increases and is slightly less well damped when the roll stiffness is increased.

5.2.3 Trailer suspension roll stiffness

Circle test

The left graph of figure 5.5 shows the steering characteristic for 40, 70, 100, 130 and 160% of the trailer suspension roll stiffness.
Increasing the trailer roll stiffness has very little effect on the steering characteristic, as can be seen in figure 5.5. The result can be explained by the fact that the trailer suspension has been designed to carry the weight of a loaded trailer. In other words, the roll stiffness of the trailer is very high and only a large decrease will influence the results. This is also shown in figure 5.5 where the steering characteristic of 40% trailer roll stiffness shows a slight decrease in understeer. Lowering the trailer roll stiffness creates an extra roll moment at the rear axle of the truck. As explained before, this will decrease the side force of the rear wheels and has to be compensated by a larger slip angle.

**Step steer**

The right graph of figure 5.5 shows the cabin roll angle for a step response.

The results for the step response show that changing the trailer suspension stiffness hardly influences the dynamic behaviour of the truck. The only difference is the increasing roll angle of the truck when the trailer roll stiffness is decreased. This supports the conclusion of the steering characteristic that the roll moment at the rear axle has increased.

**Sine sweep**

Figure 5.6 shows the transfer function of the cabin roll angle.

![TF lateral acceleration $a_y$ to steer angle $\delta$ (sine sweep @ $V=80$km/h $\delta_{sw_{max}}=30$deg)](image)

As expected, the transfer functions show no difference when the trailer roll stiffness is changed. The trailer mass inertia is very large and will dominate the dynamic behaviour of the trailer. As shown in the transfer functions of chapter 4, the dynamic response of the trailer is negligible above 0.7 Hz.
5.2.4 Conclusions roll stiffnesses

So far, the effect of roll stiffness on the steering behaviour has been discussed for three individual parameters. Even though some results were quite different, a few comments can be made to summarise the general conclusions.

− The cabin roll stiffness of the original vehicle has already been set to such a high value, that an increase has very little result. Decreasing the cabin roll stiffness will change the cabin roll eigenfrequency. Although this will influence the driver’s judgement, it will not influence handling.

− Most of the vehicles’ roll originates from the suspension of the truck. Therefore, changing the suspension roll stiffness will have a lot of influence on the steering behaviour and vehicle handling.

− Of all roll stiffnesses, the trailer roll stiffness has the least influence on the steering behaviour. First of all, the trailer roll stiffness is very high due to the fact that it has to carry heavy loads and prevent the trailer from roll over. This makes it less perceptive to variations. Second, a high mass inertia causes the trailer’s response to be less sensitive for dynamic behaviour above 0.7 Hz, as was shown in chapter 4. Since the roll eigenfrequencies of the truck lie above this frequency, the influence of the trailer roll stiffness will be negligible. However the trailer roll stiffness does have some effect on vehicle handling which can not be neglected.

5.3 Chassis torsion stiffness

In this section, the chassis torsion stiffnesses are varied for both the truck and trailer chassis. The effects on the steering behaviour are studied and the most significant results are discussed. At the end, a short summary of the main conclusions is given.

5.3.1 Truck chassis torsion stiffness

Circle test

The left graph of figure 5.7 shows the steering characteristic for 40, 70, 100, 130 and 160% of the chassis torsion stiffness.

Figure 5.8: Left. Steering characteristics for a variation of chassis torsion stiffnesses. Right, Step response, cabin roll angle for variable chassis torsion stiffnesses.
At first sight, changing the truck's chassis torsion stiffness seems to have very little effect on the steering characteristic. However, there is a notable difference. Decreasing the torsion stiffness, results in an increase of understeer. This effect can be explained by looking at the effects of both the front and rear axle.

If the chassis stiffness is decreased, the front axle will experience a larger roll moment from the cabin. In other words, less roll will be transferred to the rear axle. As explained before, this will influence the side slip angle of both the front and rear axle as follows.

- The lateral force of the front axle will decrease and has to be compensated by a larger side angle.
- The lateral force of the rear axle will increase and will be compensated by a decreasing side slip angle.

So changing the truck chassis torsion stiffness, has a combined effect that influences the steering characteristic of the vehicle.

**Step steer**

The right graph of figure 5.7 shows the cabin roll angle for a step response.

The results for the circle test, showed that the front axle will experience a larger roll moment if the chassis stiffness is decreased. The results for the step response correspond with this conclusion, as can be seen in the right graph of figure 5.7.

When the stiffness is decreased, the roll angle becomes larger for the dynamic response, as well as for steady state conditions. An important conclusion for the driver, is that the information he will receive about the dynamic response of the trailer, becomes less accurate if the stiffness is decreased.

**Sine sweep**

Figure 5.8 shows the transfer function of the cabin roll angle.

Figure 5.9: Transfer function, steer angle to roll angle for different chassis torsion stiffnesses.
In addition to the conclusions of the circle test and step response, the transfer function clearly shows a change of the cabin roll eigenfrequency. Although the steering characteristics showed that the actual vehicle handling does not change a lot by the torsion stiffness of the truck chassis. The influence of the cabin roll eigenfrequency will certainly be noticed by the driver and therefore change his perception about the steering behaviour.

5.3.2 Trailer chassis torsion stiffness

The results for variation of the trailer chassis torsion stiffness show a remarkable resemblance with the results for the trailer roll stiffness of section 5.2.3. In fact, a discussion of the results for the trailer chassis torsion stiffness would lead to the same comments that were made in section 5.2.3. Therefore, only figures are presented in this section. For comments, the reader is referred to section 5.2.3.

Circle test

The left graph of figure 5.9 shows the steering characteristic for 40, 70, 100, 130 and 160% of the trailer chassis torsion stiffness.

**Figure 5.10:** Left, Steering characteristic for a variation of chassis torsion stiffnesses. Right, Step response, cabin roll angle for variable chassis torsion stiffnesses.

Step steer

The right graph of figure 5.9 shows the cabin roll angle for a step response.

Sine sweep

Figure 5.10 shows the transfer function of the cabin roll angle.
5.3.3 Conclusions torsion stiffnesses

Section 5.3 described the effects of both truck and trailer chassis torsion stiffnesses, on the steering behaviour. Although results are quite different, two main conclusions will summarise the outcome.

- The chassis torsion stiffness of the truck, determines the dynamic “communication” between the front and rear end of the truck. It will affect the balance of lateral forces between the steer and drive axle, resulting in different handling characteristics. Second, it will change the vehicles’ roll eigenfrequency, which influences the driver’s feedback.
- The trailer chassis torsion stiffness has far less influence than the truck’s chassis stiffness. Results for the trailer chassis stiffness can be compared to the results of the trailer suspension stiffness. In both cases, a lower stiffness will lead to a larger roll moment at the drive axle. This will slightly influence the steering characteristic and the cabin roll angle.
5.4 Trailer weight distribution

In this section, the effects of both longitudinal and vertical weight distribution of the trailer load will be studied. For each variation the most significant result is discussed and at the end a summary of main conclusions is given.

5.4.1 Longitudinal weight distribution

Until now, variations are studied by changing the parameter by 40, 70, 100, 130 and 160% of the original value. However for the longitudinal weight distribution a different approach is necessary. As explained in chapter 4, the trailer chassis consist of three individual sections with a solid weight attached to represent the load. To provide a realistic weight distribution the weight of each section is changed with the values from table 5.1.

<table>
<thead>
<tr>
<th>Trailer section:</th>
<th>front</th>
<th>middle</th>
<th>rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Variation 1</td>
<td>8000</td>
<td>4000</td>
<td>2000</td>
</tr>
<tr>
<td>Variation 2</td>
<td>3000</td>
<td>8000</td>
<td>3000</td>
</tr>
<tr>
<td>Variation 3</td>
<td>2000</td>
<td>4000</td>
<td>8000</td>
</tr>
</tbody>
</table>

Table 5.1: Longitudinal weight distribution.

With this distribution, the total mass of the load remains constant, while the centre of gravity moves from front, to the rear end of the trailer.

Circle test

The left graph of figure 5.11 shows the steering characteristic of a truck with a trailer load that varies according to table 5.1.

Figure 5.12: Left, Steering characteristics for various longitudinal weight distributions. Right, Step response, cabin roll angle for a front, mid and rear loaded trailer.
Changing the longitudinal weight distribution of the trailer has quite a large effect on the steering characteristic of the truck. If the centre of mass is moved towards the front end of the trailer, the amount of understeer decreases. This result can be explained quite easy. Moving the centre of mass to the front end of the trailer will result in an increased load at the rear axle of the truck. This leads to larger slip angles at the rear axle and therefore the steering characteristic of the vehicle will move to a more oversteered situation.

**Step steer**

The right graph of figure 5.11 shows the cabin roll angle for a step response.

The main effect that can be seen in figure 5.11 is that in semi-static conditions, the roll angle increases when the load is moved to the front end of the trailer. This result is comparable with the result as seen for the trailer chassis torsion stiffness. When the load is placed at the rear end of the trailer the trailer suspension will carry most of the weight and since the roll stiffness is very high, vehicle roll will be limited. If the load is moved to the front end of the trailer, more weight is carried by the truck and the roll moment of the truck suspension will be larger.

**Sine sweep**

Figure 5.12 shows the transfer function of the cabin lateral acceleration.

![TF lateral acceleration $a_y$ to steer angle $\delta$ → (sine sweep @ $V=80\text{km/h}$ $\delta_{sw_{max}}=30\text{deg}$)](image)

The differences found in the transfer function of figure 5.12 are hard to explain, because they do not uniformly move in one direction. However, it can be concluded that the longitudinal position of the trailer load, has quite a lot of influence on the dynamic response of the truck. In addition it can be concluded that also the steering behaviour will change.
5.4.2 Vertical weight distribution

To study the effects of the vertical weight distribution, the full trailer load of 27000 kg is used again. This time the height of the centre of gravity (c.o.g.) is varied with steps of 0.25, 0.5, 0.75, 1 and 1.25 m above the trailer floor.

**Circle test**

The left graph of figure 5.13 shows the steering characteristic of a truck with a load height that varies between 0.25 and 1.25 m.

![Graph showing steering characteristics for various vertical load distributions.](image)

Figure 5.14: Left, Steering characteristics for various vertical load distributions. Right, Step response, cabin roll angle for various vertical load distributions.

As can be seen in figure 5.13 the vertical weight distribution does not have a lot of influence on the steering characteristic. Increasing the height of the centre of gravity will result in a higher roll moment of the trailer. From previous explanations, one would expect this to result in less understeer. However, since the stiffnesses of both chassis and suspension remain the same, the roll moment on the axles will be divided by the same amount. In other words, changing the height of the centre of gravity does not change the balance between lateral forces of the front and rear axle.

Although the vertical location of the centre of gravity does not affect the steering characteristic, it will have a large influence on the maximum lateral acceleration before vehicle roll-over or sliding occurs. The steering characteristic with a c.o.g. of 1.25 m has a maximum lateral acceleration of 3.1 m/s², while the other limits are close to 4 m/s². So the truck will turn over more easily.

**Step steer**

The right graph of figure 5.13 shows the cabin roll angle for a step response.

The cabin roll response shows that linear increase of the height of the centre of gravity also causes the cabin roll angle to increase linear for semi-static conditions. On the other hand, the dynamic response shows very little change.
Sine sweep

Figure 5.14 shows the transfer function of the lateral acceleration of the cabin.

Figure 5.15: Transfer function, steer angle to lateral acceleration, for various vertical load distributions.

The most significant result for the transfer function of the roll angle lies in the range of 0.3 to 0.9 Hz. In this range the roll response becomes less fast and the dip in the magnitude a bit larger. Both results can be explained by the larger roll moment of the trailer, which also increases inertia. After 0.9 Hz, the variations disappear because amplitude of the trailer response will strongly decrease. Before 0.3 Hz, conditions can be considered as semi-static, which results in a larger roll angle as explained with the step response.

5.4.3 Conclusions trailer weight distribution

The results for the longitudinal and vertical weight distribution can now be summarised into a few main conclusions.

- The longitudinal weight distribution mainly influences the vertical axle load and lateral tyre forces. If the lateral tyre forces are not changed with the same relation between front and rear axle, this will also change the steering characteristic of the vehicle.
- A second result of the longitudinal weight distribution is the influence that the trailer roll stiffness will have on the roll moment of the load. For instance, if the load is placed at the front end of the trailer, flexibility of the trailer chassis will decrease the influence of the trailer suspension stiffness. Therefore, the trailer roll moment working on the truck will be relatively larger as for a rear loaded trailer.
- The vertical weight distribution mainly influences the maximum lateral acceleration that can be reached before a vehicle will turn over.
5.5 Cabin yaw stiffness

In this section, the cabin yaw stiffness is varied with 40, 70, 100, 130 and 160% of the initial value. Unlike previous sections, variation of the cabin yaw stiffness will not be discussed separately for each simulation. Instead, figures for the circle test, step response and sine sweep will be shown at once and discussed together. This is done, because changing the cabin yaw stiffness within this range has almost no effect on the steering behaviour.

The left graph of figure 5.15 shows the steering characteristic and the right graph shows the cabin yaw velocity for a step response. Figure 5.16 shows the transfer function of the yaw velocity.

Figure 5.16: Left, Steering characteristics for various cabin yaw stiffnesses. Right, Step response, cabin yaw velocity for various yaw stiffnesses.

Figure 5.17: Transfer function, steer angle to cabin yaw velocity, for various yaw stiffnesses.
The steering characteristic displayed on the left side of figure 5.15 shows absolutely no difference when the yaw stiffness is changed. This is not a surprise because the yaw stiffness has no influence on the vertical axle load. The cabin yaw response, which is depicted on the right side of figure 5.15, shows only minor differences when the yaw stiffness is changed.

The explanation for these changes can be found when looking at the transfer function of figure 5.16. The transfer function of the yaw velocity reaches a minimum between 1 and 2 Hz. The minimum becomes larger when the yaw stiffness is increased. This indicates that the cabin yaw eigenfrequency has changed.

### 5.5.1 Conclusions cabin yaw stiffness

- The cabin yaw stiffness has a negligible effect on vehicle handling. This can be explained by the fact that the cabin has very limited freedom for rotation. Even a complete removal of the yaw stiffness would not result in enough mass displacement to influence handling.
- Although handling will not be influenced by the cabin yaw stiffness, the driver’s feedback will. This is proven by the transfer function of figure 5.16, which shows a dip in yaw velocity between 1 and 2 Hz.

### 5.6 Tyre cornering stiffness

Finally, the tyre cornering stiffnesses will be changed to study the influence on the steering behaviour. The same variation of 40, 70, 100, 130 and 160% of the original value will be used to study the effects. However, a slightly different approach is necessary since the cornering stiffness depends on the vertical load. In other words, not a single value, but the entire curve for the cornering stiffness needs to be changed.

To achieve this, the Magic Formula parameters pky1 and pky2 need to be changed so that the curves for the cornering stiffness represent the desired variation. The nominal tyre load will be used as the criterion to see if a curve has reached the desired cornering stiffness. For example, a steer tyre has a nominal load of 37 kN. The cornering stiffness for a steer tyre at nominal load is 3885 N/deg. If the cornering stiffness for the steer tyre is varied by 70% then pky1 and pky2 need to be changed, such that the curve for the cornering stiffness passes through 2720 N/deg, at a nominal load of 37 kN. Table 5.2 shows the values for pky1 and pky2 for every variation.

<table>
<thead>
<tr>
<th>%</th>
<th>Cf/α</th>
<th>pky1</th>
<th>pky2</th>
<th>%</th>
<th>Cf/α</th>
<th>pky1</th>
<th>pky2</th>
</tr>
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<tbody>
<tr>
<td>40</td>
<td>1242</td>
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</tr>
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<td>70</td>
<td>2174</td>
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<td>70</td>
<td>2720</td>
<td>-6.65</td>
<td>2.8</td>
</tr>
<tr>
<td>100</td>
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<td>-14.34</td>
<td>4.45</td>
<td>100</td>
<td>3885</td>
<td>-9.513</td>
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<td>-15.1</td>
<td>3.5</td>
<td>130</td>
<td>5051</td>
<td>-13.1</td>
<td>3.0</td>
</tr>
<tr>
<td>160</td>
<td>4968</td>
<td>-19.5</td>
<td>3.7</td>
<td>160</td>
<td>6216</td>
<td>-16.05</td>
<td>3.0</td>
</tr>
</tbody>
</table>

Table 5.2: Cornering stiffness parameter values.
Figure 5.17 shows the curves for the cornering stiffness created by the parameters as presented in table 5.2.

![Figure 5.17: Cornering stiffness curves](image)

Figure 5.18: Cornering stiffness variations for the drive and the steer tyre.

Circle test

The left graph of figure 5.18 shows the steering characteristic for a truck, with the tyre cornering stiffnesses varying from 40 to 160%.

![Figure 5.18: Steering characteristics](image)

Figure 5.19: Left, Steering characteristics for different tyre cornering stiffnesses. Right, Step response, cabin roll angle for various vertical tyre cornering stiffnesses.

The steering characteristics of figure 5.18 illustrate the huge influence tyres have on the steering behaviour. Decreasing the cornering stiffness results in more understeer of the vehicle. This result can be explained quite easy knowing that due to a larger cornering stiffness, a larger slip angle is needed to reach the same lateral force Fy. Notice the fact that the vehicle remains understeered, even when the cornering stiffness is increased by 160%. This is the result of changing the cornering stiffness for all tyres at the same time. For example, if only the stiffness of the front tyres was to be increased, it could have resulted in an oversteered vehicle.
**Step steer**

The right graph in figure 5.18 shows very clearly that decreasing the cornering stiffness slows down the response of the vehicle. This can be explained by the fact that for lower cornering stiffnesses it takes longer before the lateral tyre forces reach their final value. As a second effect the final value of $F_y$ decreases with a lower cornering stiffness, resulting in less lateral acceleration. Therefore, the cabin roll angle does not reach the same final value in a step response.

**Sine sweep**

Figure 5.19 shows the transfer function of the cabin lateral acceleration.

![Transfer function](image)

Figure 5.20: Transfer function, steer angle to lateral acceleration, for tyre cornering stiffnesses.

From the transfer function, depicted in figure 5.19 two main conclusions can be drawn. The response of the vehicle becomes much slower and smaller, when the cornering stiffness decreases. Both effects were seen in the two previous simulations and will therefore not be discussed again.

### 5.6.1 Conclusions cornering stiffness

- As explained in chapter 3, tyres are the vehicles’ contact with the road and therefore one of the most important parameters in vehicle dynamics. This is clearly shown again by the results of the tests described above. The tyre cornering stiffness determines how fast and by what amount a tyre is able to transfer forces from road to the vehicle. The dynamic characteristics of the vehicle then determine the final response a driver will feel. A fast response from the tyres is usually seen as a good quality for the steering behaviour. However, this does not mean that the largest cornering stiffness, also results in the best steering behaviour. As explained in the simulation results, a very large cornering stiffness can result in extra dynamic response of the cabin. This could influence the steering behaviour in a negative way.
5.7 Conclusions for the variation of model parameters

By studying the influence of several model parameters on the steering behaviour, it becomes clear that steering behaviour can be quantified from two different viewpoints.

- The first viewpoint from which steering behaviour can be quantified is the influence of parameters that change the driver’s feedback during a steering event, but not vehicle handling itself. In other words, the perception of the driver will change due to different motion of his seat, while the vehicle will follow the exact same path as before.

These changes are mainly caused by parameters that influence the eigenfrequencies of one or more of the vehicle’s components and therefore, also influence the response of the cabin.

- The second viewpoint from which steering behaviour can be quantified is the influence of parameters that actually change the handling characteristics of the vehicle. In other words, the vehicle will no longer follow the same path as before. A different steering characteristic is one of the most typical results for these parameters.

In most cases, these changes are caused by parameters that change either the static or dynamic mass distribution of the truck. For clarity, a change of the static mass distribution means that in static conditions the vertical tyre loads will be different. The dynamic mass distribution describes the change in vertical tyre loads during a steering event. In addition, tyre characteristics also play a very important role with respect to vehicle handling. They have a lot of influence on both handling and feedback signals.
6 Conclusions and recommendations

In this chapter the main conclusions of the study are given and some recommendations for further research are formulated.

The aim of this master thesis was to gain insight in the steering behaviour of a tractor semi-trailer combination, from a driver’s viewpoint. Until now, most knowledge about steering behaviour and handling is gained by test drivers who give a subjective quantification. Although this knowledge is very important for a truck manufacturer, recent developments like active safety systems, demand a more objective quantification. To achieve this, a multi-body simulation model of a truck and semi-trailer is used to study the steering behaviour. Before the actual analysis could take place, both truck and tyre model had to be modified to reduce the risk of incorrect results. In addition, a set of virtual test had to be configured to simulate the steering behaviour.

6.1 Conclusions

The results of the study, as discussed in this report, have been divided in three sections. These are the tyre model, the initial model and the model parameter variations. Since each of these sections has its specific outcome, discussion of the conclusions will also be subdivided into the same sections.

Tyre model

- In this study, the Magic Formula (MF) tyre model is used to represent tyre behaviour. In order to represent specific truck tyre behaviour, a set of model parameters had to be derived from tyre measurement data. Unfortunately, the available measurement data appeared to be inadequate to derive a set of parameters that produce acceptable results when extrapolating to high vertical loads. In general, the measurement data have the following shortcomings.
  - Not enough measurement data is available to understand the differences between various types of truck tyres.
  - Some of the measurements had to be completely removed because the results appeared to be unreliable.
  - Measurements were taken with a too small range of vertical loads. This causes the model to create wrong results, when extrapolated for higher vertical loads.

- The MF used in this study, may not be sufficient to represent truck tyres. The friction coefficients are represented by a linear curve, while measurement data indicate a non-linear course.

- In this study, a simplified MF is used to represent the tyres. The simplified MF contains less parameters, which results in simulating an ideal tyre. However, the simplified tyre model does not contain all necessary information to completely describe a truck tyre. It does not produce physically impossible results even when extrapolated for higher vertical loads, which is most important at this point.
Initial model

The SimMechanics model used in this study is a simplified multi-body model of a truck and semi-trailer. Because of the absence of detailed modelling the following virtual tests are formulated to give useful results to study the steering behaviour.

- Circle test, to create a steering characteristic.
- Step response, to study the response of a single steer input.
- Sine sweep, to create transfer functions.

• The circle test clearly showed an understeered vehicle. It was shown that during a steering event the jack angle between truck and trailer and the roll moment of the trailer causes a slight lift effect on the front axle. This effect will have a progressive effect on the understeered behaviour of the vehicle.

• The step response has two different effects. An initial response is caused by the steer tyres that generate a side force almost instantaneously to the input. After the step input, the remaining lateral force is generated. All other variations in the response signal are caused by the dynamic properties of the chassis, cabin and trailer. In an additional test it was shown that the trailer has great influence on the dynamic response of the cabin.

• The transfer functions, created by a sine sweep, give a lot of information about the dynamic behaviour of the vehicle. Therefore, these functions contain most information necessary to analyse the steering behaviour. However, at this moment it is difficult to draw conclusions from the transfer functions since there is no verification with reality. In general, the transfer functions can be divided into four sections.

  - < 0.1 Hz
    Response in this frequency range is speed dependent. Frequencies below 0.1 Hz are too low to cause a phase shift or any other effect on the steering behaviour.
  - 0.1 – 0.5 Hz
    Transitional phase where truck components start moving in different phase. In general, the steering behaviour will not really be influenced since steering activity is still relatively slow. Most important change within this range is the response of the trailer which will strongly decrease.
  - 0.5 – 3 Hz
    For steering activity within this frequency range there is a lot of dynamic response from several of the vehicles components, and some eigenfrequencies can be detected. As a result, the steering behaviour will be influenced to a high degree. Simulation results show that there are often large phase shifts between components like the front axle, chassis and suspension. These phase shifts will almost certainly lead to a confusing feedback for the driver.
  - > 3 Hz
    For two reasons steering activity above 3 Hz is no longer of interest to study steering behaviour. First of all, dynamic behaviour of the vehicle will strongly decrease. Second, steer inputs above 3Hz are less realistic and will usually not occur. Besides, the model is too simple and inaccurate to correctly react to these motions.
Model parameter variations

- Due to model properties and conversations with DAF engineers, a selection of nine parameters has been made to study the influence on the steering behaviour. Again, it is difficult to draw conclusions because there is no verification with reality. However, most results can be divided into two categories.

  - Handling. This category describes the variations that actually change the steering characteristics of the vehicle. In most cases, these changes are caused by parameters that change either the static or dynamic mass distribution of the truck. For clarity, a change of the static mass distribution means that in static conditions the vertical tyre loads will be different. The dynamic mass distribution describes the change in vertical tyre loads during a steering event. In addition, tyre characteristics also play a very important role with respect to vehicle handling.

  - Feedback of dynamic behaviour. This category describes the variations that do not change the steering characteristic of the vehicle, but only influence the driver's feedback. These changes are mainly caused by parameters that influence the eigenfrequencies of one or more of the vehicle’s components and therefore, also influence the response of the cabin.

When quantifying the steering behaviour of a truck, it contains both properties of feedback and handling.

6.2 Recommendations

For future research a few recommendations can be made. These recommendations can also be subdivided in three sections namely, the tyre model, the truck model and quantification of steering behaviour.

Tyre model

- **New and improved measurement program.** A large number of measurements are necessary to ensure a good knowledge of truck tyre behaviour and to understand the physical differences between various tyres. Not only the number of measurements will contribute to knowledge, a wide range of measurement conditions will also help understanding the physical tyre properties. Especially the range of vertical loads should be increased to be able to study truck tyre behaviour under heavy load conditions. This information is necessary to create a truck tyre model that generates correct results even for heavy loaded conditions. In addition, it will be useful to set up special tests to measure various friction coefficients. As already noticed the Magic Formula has difficulty to fit the correct friction coefficients and it is assumed these are non-linear for truck tyres. Additional measurements should provide the necessary knowledge.

- **Magic Formula for truck tyres.** If all tyre properties are known it is useful to study if the Magic Formula, in its current form, is adequate to represent truck tyres. From the measurement data available, questions arise if the tyre characteristics for car tyres, as represented by the MF, are the same as those of truck tyres. As explained before, the friction coefficients are possibly non-linear and it is not unlikely that there are more deviations from car tyre characteristics. If these expectations appear to be correct, modifications to the MF, or even a different tyre model may be necessary improve results.
This study showed once more the huge influence tyres have in vehicle dynamics. So for obvious reasons additional research will be useful.

**Truck model**

- **Validation.** So far, validation of the truck model with respect to handling is very minor. Although the model was validated with respect to vehicle dynamics from test drive results, the same truck has not yet been used to do real handling tests like random steer. For both the results from this study and for future research it is very important to validate the truck model with practical tests. Not only is it important to check if the model is correct, real time tests can also greatly improve insight in the steering behaviour itself. One of the first tests that can take place is the circle test, since the circle test, as simulated in this study, has the same radius of 40 m as the available DAF test ground at Sint Oedenrode.

- **Detailed modelling.** Adding more detail to the model is necessary to study all aspects of steering behaviour. For example, modelling a full steering system, including friction and play, to study steering precision and steering torque. Adding leaf springs to the suspension system will add bump and roll steer effects. And increasing the number of flexible elements will improve insight in the effects of flexibility on the steering behaviour. Especially chassis flexibility, which is quite an issue in truck dynamics, is modelled with very little detail and can therefore be improved.

**Quantification of steering behaviour**

- **Validation.** The most important step in quantifying the steering behaviour and creating a reliable link between subjective and objective quantification is, to validate simulation model results in practice. Otherwise it is almost impossible to create an analytical method for quantifying the steering behaviour of a truck with the perception of a driver. For the continuation of the study the following remarks can be made with respect to the model used and simulations that can be done.

As explained before steering behaviour can be subdivided into two categories, handling and feedback of dynamic behaviour. For parameters that purely influence the feedback signals and not handling itself, it is probably best to use a FEM-model to study these influences. The level of detail necessary to study pure eigenfrequency behaviour is much more extensive than with SimMechanics. Note that practical validation is still necessary to create a link between test results and the perception of a driver.

To study handling itself the SimMechanics model can be used. Besides extending the study of handling with practical tests, it can also be very useful to start an in depth study to the steering behaviour of the bicycle model with trailer. In future, this model, together with results from the SimMechanics model and practical tests, might lead to an analytical method for quantifying truck handling.

Finally, the SimMechanics model is also useful to study the effects of events like road obstacles. Since obstacles like ruts are very common to truck drivers, it can be of great value to study the dynamic behaviour of a truck driving a rut. However, modelling ruts is a complex job and therefore will also need practical validation to trust the results.
Bibliography


Appendix A

A.1 Characteristics Steer Tyre
Fig. A.2.1 Fig. A.2.2

Fig. A.2.3 Fig. A.2.4

(*) = tyre2a    (×) = tyre2b2    (o) = tyre2c    (x) = tyre 5

cornering

camber

relaxation length x

relaxation length y

contact length

pneumatic

relaxation length x

relaxation length y

contact length

pneumatic

relaxation length x

relaxation length y

contact length

pneumatic

relaxation length x

relaxation length y

contact length

pneumatic
A.2 Characteristics Driven Tyre

- Combined Slip (tyre1) $P_z=22000N$
- Camber influence on $M_z$ (tyre1) $P_z=22000N$
- Pure Cornering (tyre1) $F_y=16500N$ $F_y=29000N$ $F_y=41000N$
- Combined Slip (tyre1) $P_z=22000N$
- Camber influence on $M_z$ (tyre1) $P_z=22000N$
- Camber influence on $F_y$ (tyre1) $F_z=22000N$
- Camber influence on $M_z$ (tyre1) $P_z=22000N$

\[
\begin{align*}
\text{Fig. A.3.1} & \quad \text{Fig. A.3.2} \\
\text{Fig. A.3.3} & \quad \text{Fig. A.3.4}
\end{align*}
\]
A.3 Characteristics Trailer Tyre

Fig. A.5.1

Fig. A.5.2

Fig. A.5.3

Fig. A.5.4

Fig. A.5.5

Fig. A.5.6
Appendix B

B.1 Extrapolated Tyre Model Characteristics
Pure Cornering (drive)

- $F_r = 41kN$
- $F_r = 53kN$
- $F_r = 65kN$
- $F_r = 81kN$
- $F_r = 93kN$

Pure Braking (drive)

- $F_z = 41kN$
- $F_z = 53kN$
- $F_z = 65kN$
- $F_z = 81kN$
- $F_z = 93kN$

Camber influence on $M_z$ (drive)

- $\gamma = \pm 10^\circ$
- $\gamma = \pm 5^\circ$
- $\gamma = -2^\circ$
- $\gamma = 0^\circ$
- $\gamma = 4^\circ$

Camber influence on $F_y$ (drive)

- $\alpha = -8^\circ$
- $\alpha = -5^\circ$
- $\alpha = \pm 2^\circ$
- $\alpha = \pm 1^\circ$

Combined Slip (drive)

- $F_z = 70000N$
- $\gamma = \pm 10^\circ$
- $\gamma = \pm 5^\circ$
- $\gamma = -2^\circ$
- $\gamma = 0^\circ$
- $\gamma = 4^\circ$

Fig. B.2.1  Fig. B.2.2  Fig. B.2.3  Fig. B.2.4  Fig. B.2.5  Fig. B.2.6
B.2 Comparing the Tyres using measurement data

- Camber Stiffness
- Camber Stiffness for Aligning Torque
- Longitudinal Peak Friction
Appendix C

C.1 Reduced Tyre Property Files

Reduced tyre property file for the steer tyre: DAF_steer_tyre.tpf

```
# MF-Tyre/MF-Swift 6.0.2 Express sample tyre property file
# MF-Tyre/MF-Swift 6.0.2 Express sample tyre property file

#--------------------------------------------------------------------------units
[UNITS]
LENGTH             = 'meter'
FORCE              = 'newton'
ANGLE              = 'radians'
MASS               = 'kg'
TIME               = 'second'

#--------------------------------------------------------------------------model
[MODEL]
FITTYP             = 60       #Magic Formula Version number
TYRESIDE           = 'SYM'
LONGVL             = 16.537   #Measurement speed
VXLOW               = 1        #Lower boundary of slip calculation
ROAD_INCREMENT     = 0.02     #Increment in road sampling
ROAD_DIRECTION     = 1        #Direction of travelled distance

# The next lines are only used by ADAMS and ignored by other MBS codes

# USE_MODE specifies the type of calculation performed:
# 0: Fz only, no Magic Formula evaluation
# 1: Fx,My only
# 2: Fy,Mx,Mz only
# 3: Fx,Fy,Mx,My,Mz uncombined force/moment calculation
# 4: Fx,Fy,Mx,My,Mz combined force/moment calculation
# +00: steady state behaviour
# +10: including relaxation behaviour < 10 Hz
# +20: including rigid ring dynamics < 100 Hz
# +300: smooth road contact (long wave lengths)
# +400: basic function road contact (circular cross section, motorcycles)
# +500: moving road contact
# +600: basic function road contact (3D road profile, short wavelengths)
# +700: basic function road contact (3D road profile)
# example: USE_MODE = 434 implies:
# - combined slip
# - rigid ring dynamics
# - basic function road contact

PROPERTY_FILE_FORMAT    = 'USER'
USER_SUB_ID             = 815
N_TIRE_STATES           = 4
USE_MODE                = 104       #Tyre use mode switch (ADAMS only)
HMAX_LOCAL              = 2.5E-4     #Local integration time step (ADAMS only)
TIME_SWITCH_INTEG       = 0.1        #Time when local integrator is activated (ADAMS only)

#-----------------------------------------------------------dimensions
[DIAMENSION]
UNLOADED_RADIUS        = 0.548     # Free tyre radius
WIDTH                  = 0.2        # Nominal section width of the tyre
RIM_RADIUS             = 0.2        # Nominal rim radius

#-------------------------------------------------------------inertia
[INERTIA]
MASS                   = 62.7       # Tyre mass
IXX                    = 6.6        # Tyre diametral moment of inertia
IYY                    = 11.9       # Tyre polar moment of inertia
GRAVITY                = -9.81      # Gravity acting on tyre in Z-direction

#--------------------------------------------------------------vertical
[VERTICAL]
VERTICAL_STIFFNESS     = 1.19E+006  # vertical stiffness
VERTICAL_DAMPING       = 500        # vertical damping
FNOMIN                 = 36700      # Nominal wheel load

#-------------------------------------------------------------structural
[YAW_STIFFNESS]        = 2.2E+04    #Yaw stiffness

#-------------------------------------------------------------longitudinal
[VERTICAL_FORCE_RANGE] = 2.1E+04    #vertical_force_range

#-------------------------------------------------------------lateral
[LATERAL_COEFFICIENTS] =
PD1L                 = 0.74563  # Longitdinal friction Mus at Fznom
PD2L                 = -0.12962  # Variation of friction Mus with load
PKL                  = 12.444    # Longitudinal slip stiffness Kfz/Fz at Fznom

#-------------------------------------------------------------rolling resistance
[ROLLING_COEFFICIENTS] =
QST1                 = 0.0085    # Rolling resistance torque coefficient
QST2                 = 0.01999    # Peak trail/X0 at Fznom
```

/department of mechanical engineering
Reduced tyre property file for the drive tyre: DAF_drive_tyre.tpf

\[ \text{UNITs} \]
\[ \text{LENGTH} = \text{meter} \]
\[ \text{FORCE} = \text{newton} \]
\[ \text{ANGLE} = \text{radians} \]
\[ \text{MASS} = \text{kg} \]
\[ \text{TIME} = \text{second} \]

\[ \text{model} \]
\[ \text{FITTY} = 60 \]
\[ \text{TYRESIDE} = \text{SYM} \]
\[ \text{LONGVL} = 16.537 \]
\[ \text{VXLOW} = 1 \]
\[ \text{ROAD_INCREMENT} = 0.02 \]
\[ \text{ROAD_DIRECTION} = 1 \]

\[ \text{dimensions} \]
\[ \text{UNLOADED_RADIUS} = 0.548 \]
\[ \text{WIDTH} = 0.318 \]
\[ \text{RIM_RADIUS} = 0.2 \]

\[ \text{inertia} \]
\[ \text{MASS} = 62.7 \]
\[ \text{IXX} = 6.6 \]
\[ \text{IYY} = 11.9 \]
\[ \text{GRAVITY} = -9.81 \]

\[ \text{vertical} \]
\[ \text{VERTICAL_STIFFNESS} = 1.1281 \times 10^6 \]
\[ \text{VERTICAL_DAMPING} = 500 \]
\[ \text{FNOMI} = 28900 \]

\[ \text{structural} \]
\[ \text{YAW_STIFFNESS} = 3.4 \times 10^4 \]

\[ \text{longitudinal} \]
\[ \text{PDY1} = 0.7547 \]
\[ \text{PDY2} = -0.1497 \]
\[ \text{PKY1} = 14.34 \]
\[ \text{PKY2} = 4.4486 \]
\[ \text{PDY6} = -0.13 \]

\[ \text{lateral} \]
\[ \text{QDY1} = -0.7575 \]
\[ \text{QDY2} = 4.445 \]
\[ \text{QDY1} = 0.0085 \]
\[ \text{QDYI} = 0.06719 \]
Appendix D

D.1 Characteristics simplified steer tyre

Fig. D.1.1 Combined Slip (tyre2b2)

Fig. D.1.2 Camber influence on Fx (tyre2b2)

Fig. D.1.3 Camber influence on My (tyre2b2)

Fig. D.1.4 Camber influence on Mz (tyre2b2)
Fig. D.2.1

Fig. D.2.2

Fig. D.2.3

Fig. D.2.4
**D.2 Characteristics simplified drive tyre**

Figs. D.3.1 to D.3.4 show the characteristics of a simplified drive tyre. The graphs illustrate the influences of camber and slip angles on the forces and moments acting on the tyre at different loads. The figures display how the tyre performance changes with varying camber and slip angles, providing insights into the tyre's behavior under different conditions.
Fig. D.4.1 Fig. D.4.2
Fig. D.4.3 Fig. D.4.4

(+)=tyre1 (*)=tyre6
camber
cornering
contact length
relaxation length x
relaxation length y

Fz [N]
effective rolling radius [m]
contact length x
relaxation length y

contact length
pneumatic trail
Appendix E

E.1 Simplified model extrapolated
Fig. E.2.1 Fig. E.2.2 Fig. E.2.3

Fig. E.2.4 Fig. E.2.5 Fig. E.2.6

Pure Cornering (drive)

\[
\begin{align*}
F_{z} &= 41 \text{kN} \\
F_{z} &= 53 \text{kN} \\
F_{z} &= 65 \text{kN} \\
F_{z} &= 81 \text{kN} \\
F_{z} &= 93 \text{kN}
\end{align*}
\]

\[
\begin{align*}
\alpha &= -8 \text{Deg} \\
\alpha &= -5 \text{Deg} \\
\alpha &= \pm 2 \text{Deg} \\
\alpha &= \pm 1 \text{Deg}
\end{align*}
\]

 Combined Slip (drive)

\[
F_{z} = 70000 \text{N}
\]

\[
\begin{align*}
\gamma &= \pm 10 \text{Deg} \\
\gamma &= \pm 5 \text{Deg} \\
\gamma &= -2 \text{Deg} \\
\gamma &= 0 \text{Deg} \\
\gamma &= 4 \text{Deg}
\end{align*}
\]

Camber influence on \(F_{y}\) (drive)

Camber influence on \(M_{z}\) (drive)
Appendix F

F.1 Tractor semi-trailer vehicle model parameters

In this appendix most of the model parameter names, locations and values are given.

General Parameters

Sign convention
- $X$  positive in the driving direction
- $Z$  positive upwards
- $Y$  positive to the left

$Z=0$  road level
$Y=0$  plane of symmetry of the vehicle
$X=0$  centre of the front wheels

Units
Within the model SI units (m, kg, N, s, rad) are always used.

General
A model is made of the DAF FT XF95 Super Space Cab 4x2 tractor.
The semi-trailer is a generic three axle configuration. Coordinates are given for the vehicle in a fully loaded condition.

Main dimensions
front axle: $X=\circ$
rear axle: $X=-3.8$ (WB=3.8)
5th wheel: $X=-3.13$, $Z=1.15$ (KA=0.67)
1st trailer axle $X=-9.63$ (distance to 5th wheel: 6.5)
2nd trailer axle $X=-11.03$ (distance to 5th wheel: 7.9)
3rd trailer axle $X=-12.43$ (distance to 5th wheel: 9.3)

Overall length of the vehicle combination: 15.50
Tractor

Engine/driveline

Engine and gearbox are modelled as a single rigid body, elastically suspended in all 6 degrees of freedom with respect to the chassis.

The drive torque is calculated based on engine power and angular velocity and scaled using the throttle input (which is between 0-1).

The driveshaft is massless and infinitely stiff. The drive torque is applied to a revolute joint between engine and driveshaft and (taking into account the final drive ratio and differential) to the wheels.
In this appendix the points of measurement and coordinates, used for frequency response analysis are given. Figure F.4 shows the points of measurement.

Figure F.4: Points of measurement for frequency response.

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Table F.1 Coordinates of points of measurement