Preface

This master thesis was carried out at the division of Vehicle Dynamics at Kungliga Tekniska Högskolan (Royal Institute of Technology) in Stockholm Sweden in collaboration with the company Vehiconomics which provided me with the prototype vehicle and all associated data. This thesis contains testing, measurement and simulation, which are important moments in vehicle development.

During this demanding and at the same time extremely fun period, I have acquired an understanding for the complexity of vehicle development, and the importance of being accurate and patient when undertaking projects of this magnitude.

First of all I would like to thank my supervisor and examiner Lars Drugge. He has been a huge help with his knowledge and expertise in this particular area. Lars has been my speaking-partner and even at those dark days, trough discussion we have managed to find the light. I would like to thank Håkan Lutz, the founder of Vehiconomic, for his involvement in this project and for helping me carry out certain measurements. Huge thanks goes also to Jesper Slättengren for his priceless help with ADAMS/Car and guidance to develop my own ideas. He has worked with this software for many years and is very skilled in this area.

A person who deserves a special mention is my colleague Johan Hag, for his friendship and support during the time at Kungliga Tekniska Högskolan.

Last but not least, I would like to thank my beloved family for their unlimited support and patience during this whole time.

Stockholm, 2011

Igor Kovacevic
Abstract
This thesis treats the handling analysis of the Vehiconomic's three wheeled prototype vehicle “Smite”. This is a light weight low emission vehicle intended for urban traffic but with sufficient power supply to manage highway driving as well. The purpose of this thesis is to evaluate the vehicle's stability and handling characteristics using the multibody simulation software ADAMS/Car, and also to try to find a more optimum configuration of the vehicle. The software in an excellent tool that allows you to examine different driving scenarios and easily make vehicle modifications, both quicker and safer than in real life.

In order to obtain reliable results the simulations have to be as close to the reality as possible. The ADAMS/Car model was validated with the help of logged data, derived while performing physical test runs on an air strip. Maneuvers preformed were; Severe lane change, Slalom maneuver and a J-turn maneuver. Due to simplifications made in the virtual model, the validation did not generate satisfying results for all driving scenarios, but the model is considered good enough to help draw general conclusions about the vehicle's behavior. The largest simplification made was regarding the propelling of the vehicle, which was made front wheel driven instead of rear wheel driven.

Due to the simple construction of the vehicle, very little room for modifications is available. The possibilities to affect the position of the center of gravity are very limited. The only room for modification available is in the front suspension system. Due to multiple mounting points for the spring/damper component, 12 different combinations of the front suspension system were generated. Additionally two more setups were made by changing the track width with the implementation of spacers. A suspension configuration was found that reduced the roll angle with approximately 10%.
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
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<tbody>
<tr>
<td>( \delta_s )</td>
<td>[mm]</td>
<td>Displacement of the spring</td>
</tr>
<tr>
<td>( \delta_w )</td>
<td>[mm]</td>
<td>Displacement of the wheel</td>
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<tr>
<td>( \varphi )</td>
<td>[N/rad]</td>
<td>Cornering stiffness</td>
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<tr>
<td>( C_\alpha )</td>
<td>[N/rad]</td>
<td>Slip coefficient</td>
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<tr>
<td>( c_{\text{damper}} )</td>
<td>[Ns/mm]</td>
<td>Viscous damping coefficient</td>
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<tr>
<td>( c_v )</td>
<td>[Ns/m]</td>
<td>Vertical damping coefficient</td>
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<tr>
<td>( C_f )</td>
<td>[Nm/rad]</td>
<td>Roll stiffness</td>
</tr>
<tr>
<td>( C_{rr} )</td>
<td>[-]</td>
<td>Rolling resistance coefficient</td>
</tr>
<tr>
<td>( C_{\text{stip}} )</td>
<td>[N/rad]</td>
<td>Longitudinal force coefficient</td>
</tr>
<tr>
<td>( F_{\text{damper}} )</td>
<td>[N]</td>
<td>Damping force</td>
</tr>
<tr>
<td>( F_{\text{spring}} )</td>
<td>[N]</td>
<td>Spring force</td>
</tr>
<tr>
<td>( F_{\text{torque}} )</td>
<td>[N]</td>
<td>Total force on the body</td>
</tr>
<tr>
<td>( F_w )</td>
<td>[N]</td>
<td>Vertical wheel force</td>
</tr>
<tr>
<td>( C_z )</td>
<td>[N/m]</td>
<td>Vertical tire stiffness coefficient</td>
</tr>
<tr>
<td>( L_s )</td>
<td>[mm]</td>
<td>Distance to the attachment of the spring</td>
</tr>
<tr>
<td>( L )</td>
<td>[mm]</td>
<td>Length of the lower A-arm</td>
</tr>
<tr>
<td>( t_w )</td>
<td>[mm]</td>
<td>Track width</td>
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1 Introduction

Vehiconomic’s prototype Smite is a low emission three wheeled vehicle that carries two people. The passenger sits behind the driver, thus making the vehicle narrower and more maneuverable than a regular car. The Smite is meant to be used for urban transportation, therefore making the demands on ride stability and handling very high.

In order to make the Smite safe in all kinds of traffic situations an evaluation of the vehicles handling is needed. By conducting physical tests, the obtained data is used to validate a virtual model of the Smite. The virtual model is built in the multi-body software MSC.ADAMS/Car and is configured so that the geometries and the inertial characteristics of the different subsystems resemble the real Smite.

This way different parameters of the vehicle can easily be virtually modified in order to evaluate the dynamic behavior. This procedure is a quicker and easier alternative than making the changes in real life.

The aim of this thesis is to examine the dynamic behavior of the three wheeled vehicle Smite, and by modifications of the virtual car, examine if a more optimal configuration is achievable in terms of driving comfort and roll angle reduction. This possibility is given through the vehicle’s adjustable front suspension, where the spring/damper is allowed to be mounted in different positions.
2 Problem description

2.1 Background
In order to perform an evaluation of the vehicle’s handling, it is important to make some general assumptions and estimates, hence the known vehicle data obtained in this project is seemingly limited. The available general vehicle data consisted of engine power, turning radius, weight and geometric dimensions. In order to build a good estimated model of the proposed vehicle, crucial data such as suspension data, tire characteristics and the position of the center of gravity, are needed. These will be measured through various experiments which will be conducted partly at the MWL\(^1\), at the Barkarby airstrip and at Vehiconomics\(^2\) headquarters.

2.2 Purpose
The purpose with this project is to conceive an adequate virtual vehicle model which will be used in the development of a more optimal configuration for the proposed vehicle. The virtual model will be a quick and efficient tool to simulate different driving conditions, in different environments, with different vehicle setups. In this way the vehicle’s dynamic behavior can be predicted and the procedure is an easier and safer alternative than making changes in real life.

2.3 Aim
The aim is managing to build a virtual vehicle model in such a way that the dynamic characteristics are a good estimation of the real vehicle’s dynamic behavior. The model should be able to give an acceptable correct recreation of the real driving scenario in order to be relevant for this study. The validation of the virtual model will be conducted by using the data collected during various experiments, where one is a physical test run at an airstrip. After the validation of the virtual model the goal is to find a more optimum configuration of the vehicle’s front suspension parameters, thus improve the vehicle properties in terms of driving comfort. Through different mounting positions of the spring/damper it will be examined how much the roll angle can be reduced.

2.4 Problem
The problem consists of recreating the physical vehicle to a working virtual model, using estimates along with the known vehicle data. What needs to be obtained in order to validate the virtual vehicle model is;

- Position of the center of gravity
- Tire characteristics

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\(^1\) Marcus Wallenberg Laboratory at Kungliga Tekniska Högskolan.

\(^2\) Vehiconomics is the company which developed the three wheeled vehicle called Smite.
• Suspension data

• Accelerations, speed and path from the physical test drives

The possibility of measuring all of the above mentioned paragraphs will be available except for the tire characteristics. In this case estimates will be necessary in order to obtain a working model.

2.5 Delimitation

This project is limited to a time period of 20 weeks and the results will be limited to what is possible to achieve in that period of time. The difficulty lies in creating a model with good enough accuracy to describe real life scenarios. The tire characteristics and the friction between the road and the tire are tuned in to resemble the dynamic behavior of the physical vehicle.
3 Literature study

The literature review did not give a sufficient insight to the problem of evaluating the stability of a three wheeled vehicle with a 2F1R\(^3\) [1] configuration. Although there are numerous of companies today building three wheelers, there was not a lot of published work dealing with the stability- and handling characteristics of a 2F1R\(^3\) [1] configured three-wheeler. The majority of the papers presented dealt mostly with tilting three wheelers with a 1F2R configuration.

Koona Ramji, V. K. Goel, Kusum Deep and Manoj Thakur [2] conducted a parametric study and identified the parameters which influence the ride behavior most: Stiffness and damping of front and rear suspension systems, wheelbase of three wheeled vehicle and track width of the vehicle. This information is most relevant to this thesis and will be a starting point for the dynamical analysis.

The American Academy of Pediatrics [3] presented in their paper about vehicle injury prevention, the constraints and limitations of three wheeled all-terrain vehicles. In this paper it is concluded that even though the three wheel design gives a stable appearance, it has stability difficulties especially on hard surfaces. Furthermore, parameters such as high center of gravity, a poor or absent suspension system, and in case of two rear wheels, no rear-wheel differential, have strong impact on the stability.

C-C Yu and T Liu [1] described the dynamics of three wheeled vehicles with either two wheels on the front axle or two wheels on the rear axle. Their study presented a full control strategy, which means that the vehicles motion can be fully controlled by input controls on every wheel, such that a given driving scenario can be achieved. When controlling the six components of motion: longitudinal, lateral, bounce, roll, pitch and yaw, the dynamics of the vehicle can be improved and controlled more precisely. Thus decrease tire dynamic weight transfer and eliminate the lateral velocity.

\(^3\) There are two configurations for the three-wheeled vehicle. One is two wheels on the front axle and one wheel on the rear axle (2F1R). The other is one wheel on the front axle and two wheels on the rear axle (1F2R).
4 Vehicle description

4.1 The three wheeler

The Smite is a prototype vehicle conceived by the company Vehiconomics (Figure 1). The vehicle carries two people where the passenger sits behind the driver, thus making the vehicle narrower and more maneuverable than a regular car. It is propelled by a petrol engine and a CVT\textsuperscript{4} gearbox enabling it to reach a maximum speed of 90 km/h.

![Figure 1: The vehicle Smite [4].](image)

The vehicle consists of eight subsystems, namely; tires, brakes, engine and transmission, chassis, body, steering, front suspension and rear suspension. The vehicle has two steerable front wheels and one single driven wheel in the back (Figure 2-3). All the wheels are braked by disc brakes. The front suspension system consists of a double wishbone suspension, from where two tierods connect to a steering plate and a “bicycle-like” steering rod, which makes up the steering. The rear suspension consists of a single swingarm on which the engine and transmission are mounted.

\textsuperscript{4} Continuous Variable Transmission.
The chassis is a sandwich composite construction and the body is made of a thermoplastic material. The body is segmented into nine parts held together by Velcro, thus simplifying the construction by eliminating the use of screws.

4.2 The virtual model

To create the complete virtual vehicle prototype the software ADAMS/Car was used. This is a multibody software designed to build and simulate different types of vehicles. The modeling is template-based where every template is changeable by simply entering desired data. This is done in the “template builder” mode, which is an interface in ADAMS/Car. By first defining hardpoints, parts and geometry are easily created thereafter. All the templates for this vehicle, except for the body, engine and brakes, were made from scratch. The templates for the body and engine already existed as readymade templates. In the readymade templates only their data was changed so that the properties and the geometry would be consistent with the physical vehicle. The virtual models was built by means of subsystems; tires, brakes, engine and transmission, chassis, body, steering (Figure 7), front suspension (Figure 5), and rear suspension (Figure 6), that are assembled to obtain the desired geometrical layout (Figure 4). The advantage with this software is that parameters can easily be changed in order to analyze different layouts and explore “what if” studies. This way it is possible to

\[5\] Coordinates in the xyz-space.
quickly animate the vehicles behavior and plot the dynamic result, making the method effective in both time and financial cost.

![Figure 4: The assembled virtual model of the Smite.](image)

Even though the physical vehicle is rear wheel driven, the virtual model was made with a front wheel drive. This was done due to lack of time to from scratch build a complete new powertrain template. The only measure that was taken is that the powertrain map was changed to more realistic values suiting this project. With this in mind it is understandable that the limit handling of the virtual vehicle might slightly differ from the real one. Because of the fact that no severe acceleration tests will be performed in this study, the difference in handling is to be considered small enough for the results to count as relevant. The body of the virtual vehicle was imported into ADAMS/Car as a STEP-file, obtained from a preexisting CAD\textsuperscript{6} - model of the vehicle made with the software Rhinoceros 3D [5]. The imported body serves for aesthetic purposes and to better visualize the behavior of the car.

The ADAMS/Car model was built to behave as close to the real vehicle as possible. To achieve this, the following parameters needed to be estimated due to the absence of real data:

- Spring stiffness (rear)
- Damper stiffness (rear)
- Steering gear ratio
- Tire properties

The tire properties were tuned iteratively in order to obtain satisfying results. The tire property file used was the Fiala [6] tire model. This is a simplified model with very few input parameters but which still provides reasonable results. This tire model comes very handy for cases like these when no tire data (except tire dimensions) is known.

\textsuperscript{6} Computer Aided Design
Figure 5: Front suspension subsystem of the virtual Smite.

Figure 6: Rear suspension subsystem of the virtual Smite.

Figure 7: Steering subsystem of the virtual Smite.
4.3 The instrumented vehicle

The vehicle experimentally tested was the physical prototype Smite. No particular modifications of the vehicle were acquired in order to mount the measuring instruments (Figure 8-9) which consisted of a data logger and an inertial measurement unit. These were placed as close to the location of the center of gravity as possible in order to obtain the best results. By means of the devices it was possible to measure the position, the speed and accelerations of the vehicle. These data were used to validate the virtual vehicle model, i.e. to make it behave as close to the real vehicle as possible.

Figure 8: The DL2 Data Logger.

Figure 9: The IMU06 inertial measurement unit.
5 Method

In order to validate the virtual model, gathered data was used from the conducted tests at the airfield in Barkarby with the prototype vehicle. The same maneuvers were executed by the virtual model and the instrumented vehicle. The tests consisted of several types of driving maneuvers, a severe lane-change maneuver - Part 2: Obstacle avoidance (according to ISO 3888-2), a slalom course to generate a sinus-like movement and a constant radius J-turn. All of the maneuvers were performed with varying speed.

5.1 Measurements - position of the center of gravity

The center of gravity of a vehicle is the point at which the object balances if placed on a pivot point. There are three axes on a vehicle: the x-axis (driving direction), the y-axis (lateral direction) and the z-axis (vertical direction). The position of the center of gravity on the x- and y-axis can be determined directly from the weight of each wheel. The vertical position requires that one end of the car be lifted. In this case the rear end of the vehicle was raised 32.4 cm. In order to get the weight contribution of each wheel scales (Figure 10) connected to a controller box (Figure 11) were placed under each wheel (Figure 12).

Figure 10: The scales for the front axis.

Figure 11: The controller box.
To conduct these measurements, the control box needed the axle width and wheel base. The axle width was measured from the outside of the left tire to the outside of the right tire on the front axle. The wheel base was measured from the middle of the front tire to the middle of the rear tire. After these parameters were entered, the scale displayed the location of the center of gravity from the left front to right front tire, and from the left front to the rear tire. For calculating the vertical center of gravity (which is a point that the car would balance on if it were turned on its side) an additional parameter needed to be entered in the control box and that is the amount the back wheel was raised from the ground while keeping the front wheels placed on the scales. By lifting the back of the vehicle the center of gravity moved forward along the x-axis. The amount of movement is related to how high up the center of gravity is in the vehicle. The control box calculated and displayed the vertical position of the center of gravity. Regarding the accuracy of the measurements, account must be taken for deviating values due to the unevenness of the ground where the scales were placed, as to the measured height to which the rear wheel was raised.
A virtual illustration was made showing the weight distribution and the center of gravity position (Figure 13).

Figure 13: The measured weight distribution on each wheel and the position of the center of gravity, recreated in the software 3d max. The red digits represent the percentage of the total weight acting on each wheel. The green digits represent the location of the center of gravity with regard to the front left wheel. The digits represent percentage of the total length in x-, y-, and z-direction.

5.2 Measurements - front suspension

One of the crucial parts needed in order to validate the virtual model is the suspension characteristics of the physical vehicle. Luckily a spare spring/damper component was obtained for evaluation using a Instron [7] testing machine designed to evaluate the mechanical properties of materials and components. The tests were conducted at the Marcus Wallenberg laboratory at the Royal Institute of Technology. Since the spring is integrated with the damper the entire component was mounted into the Instron testing machine, see Figure 14. The test was performed with a series of frequencies ranging from 2-40 Hz, with corresponding amplitudes in the span 1-15 mm (Table 1).
Table 1: Frequencies and amplitudes used to test the spring/damper component.

<table>
<thead>
<tr>
<th>Amplitude [mm]</th>
<th>Frequency [Hz]</th>
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<tr>
<td>1</td>
<td>10</td>
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<tr>
<td>1</td>
<td>15</td>
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<td>1</td>
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<td>15</td>
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The results from the testing machine were obtained as tables consisting of different load cases with corresponding amplitudes. Hence the suspension component consists of both a spring and a damper, account must be taken for the static spring force which is integrated into the damper force.

\[ F_{\text{suspension, tot}} = F_{\text{spring}} + F_{\text{damper}} \]  \hspace{1cm} (1)

In order to execute this operation \( F_{\text{spring}} \) must be calculated, thus \( k_{\text{spring}} \) is needed.

\[ F_{\text{spring}} = -k_{\text{spring}} \cdot x \]  \hspace{1cm} (2)

\[ k_{\text{spring}} = \frac{\Delta y}{\Delta x} \]  \hspace{1cm} (3)

Thanks to a separate test which was conducted for measuring the static spring force, the graph obtained (Figure 15), was used to calculate \( k_{\text{spring}}=50\text{N/mm} \). The \( \Delta y \) and the \( \Delta x \) interval covered the relevant amplitude field used in the tests.

Figure 14: Instron testing machine. 1- Load cell 100 kN, 2- Load cell 5 kN, 3- Spring/damper, 4- Position sensor, 5- Hydraulic piston, 6- bearings.
With $F_{spring}$ known, the damper characteristics can be calculated and visualized as a force-amplitude diagram (Figure 16). The force-amplitude diagrams for all amplitudes are visualized in APPENDIX A.
Since the damping is greater in the expansion direction than in the compression direction, the damper has a non-linear characteristic. With the $F_{\text{damper}}$ known, a force-velocity diagram can be created (Figure 17), using equations (4) and (5).

\[ F_{\text{damper}} = c \cdot \dot{x} \]  \hspace{1cm} (4)

\[ \dot{x} = A \cdot 2 \cdot \pi \cdot f \]  \hspace{1cm} (5)

![Figure 17: Damping characteristics.](image)

5.2.1 Vertical wheel suspension spring constant

Due to the geometry of the front wheel suspension (Figure 18), the equivalent vertical wheel suspension spring constant $k_w$ differs from the coil spring constant $k_s$. 
Figure 18: Individual front wheel suspension with double parallel A-arms.

\[ k_w = \frac{F_w}{\delta_w}, \quad \delta_w = L_w \]  

(6)

\[ k_s = \frac{F_s}{\delta_s}, \quad \delta_s = L_s \cdot \cos(\beta + \theta) \]  

(7)

\[ F_w \cdot L_w - F_s \cdot L_s \cdot \cos(\beta + \theta) = 0 \]  

(8)

\[ \frac{F_w}{L_w} = \frac{F_s \cdot L_s \cdot \cos(\beta + \theta)}{L \cdot \cos \alpha} \]  

(9)

\[ k_w = \frac{(F_s \cdot L_s \cdot \cos(\beta + \theta))/\left(L \cdot \cos \alpha\right)}{\delta_s/(L_s \cdot \cos(\beta + \theta))} \rightarrow k_s \cdot \frac{(L_s \cdot \cos(\beta + \theta))^2}{(L \cdot \cos \alpha)^2} \]  

(10)

With inserted numerical values in expression (10) the equivalent vertical wheel suspension spring constant is calculated to \( k_w = 7.83 \) N/mm.
5.2.2 Roll axis and roll angle

Due to inadequate data to make a validation of the virtual model, as a complementary to the measured lateral acceleration, a theoretical estimation of the roll angle is needed. By determining the instant center of rotation (RC) for the vehicle (Figure 19), the roll angle can be calculated as a function of lateral acceleration.

Figure 19: Position of the instant center of rotation for the front suspension.

\[
C_F = \frac{M}{\phi} \rightarrow \frac{2 \cdot F_z \cdot \frac{t_w}{2}}{\phi} \quad (11)
\]

\[
F_z = k_w \cdot \delta_w \rightarrow k_w \cdot \frac{t_w}{2} \cdot \phi \quad (12)
\]

From equation (11) and (12) we obtain the final expression for roll stiffness.

\[
C_F = \frac{2 \cdot k_w \cdot \frac{t_w}{2} \cdot \phi \cdot \frac{t_w}{2}}{\phi} \rightarrow 2 \cdot k_w \cdot \left(\frac{t_w}{2}\right)^2 \quad (13)
\]
From the force equilibrium for the sprung mass around the roll axis (Figure 20) the roll angle is obtained.

\[ 0 = C_F \cdot \phi - m \cdot a_y \cdot h_e - m \cdot g \cdot h_e \cdot \phi \rightarrow \phi = \frac{m \cdot h_e}{(C_F - m \cdot g \cdot h_e) \cdot a_y} \tag{14} \]

where

\[ h_e = h - \frac{b \cdot e_f}{L} \tag{15} \]

### 5.3 Maneuvers description

The data needed for the virtual vehicle model was acquired during a test day at a small landing strip in Barkarby outside of Stockholm. The measurements were made using the prototype vehicle Smite. Both the physical and virtual vehicle were subjected to the same maneuvers with the same speed profile.

#### 5.3.1 Severe lane change - Part 2: Obstacle avoidance

The principle of the obstacle avoidance maneuvers is to drive a vehicle from its initial lane to the adjacent lane parallel to the first, and returning to the initial lane, without exceeding lane boundaries (Figure 21). Over the test course the throttle position was held as steady as possible in order to try to maintain a constant speed. This test only summarizes one small part of a vehicle's complete handling characteristics. The track setup was done according to the International Organization for Standardization [8]. Parameters measured are speed, position coordinates and lateral acceleration.
5.3.2 Slalom track

The second maneuver that was executed was the slalom track with a sine-like pattern (Figure 22). This handling test aim to determine the transient handling response due to steering and suspension. The test started with a minimum speed of 30 km/h to get a feel of the vehicle behavior through the track, and was then gradually increased in steps until maximum controllable speed was achieved. Parameters measured are speed, position coordinates and lateral acceleration.
5.3.3 J-turn

The J-turn maneuver is an avoidance maneuver in which a vehicle is steered away from an obstacle using a single steering input, where the idea is to examine how the vehicle reacts to an immediate steering angle applied (Figure 23). In the beginning of the maneuver, the vehicle is driven in a straight line at a constant speed and the driver then triggers a desired handwheel input.

Figure 23: J-turn track setup.
6 Validation

Before any conclusions can be drawn, the virtual model in ADAMS/Car must be validated in order to ensure that the model has a similar behavior to the real vehicle. Validation is done to conform with all the three driving scenarios. Regarding the accuracy of the model, account must be taken for the simplifications that were done. For instance the ADAMS/Car model is front wheel driven while the real vehicle is rear wheel driven. The rear suspension characteristics, the moment of inertia and the tire characteristics of the vehicle were iteratively tuned in to obtain best possible conformation between the virtual and the real vehicle. It is important to emphasize the simplifications that the software does, which can not be changed by the user and which produce some differences. ADAMS/Car uses a vehicle controller called Machine Control to simulate the control actions of a driver. Machine control uses simple mathematical models of vehicle dynamics to estimate necessary control actions, such as the steering angle and throttle position. In the upcoming paragraphs, comparisons between the virtual and the experimental data for some of the driving maneuvers are shown.

6.1 Speed control

To generate the speed profile, logged data was used from the physical tests with 0.01 second intervals. In order to maintain and control the speed a powertrain template was used to drive the front wheels. The template is a modified preexisting powertrain template which was adjusted to suit the three wheeler. As mentioned earlier, this configuration is a simplification and does not conform with the real vehicle, since the real vehicle is rear wheel driven and propelled with only one wheel. As seen in Figures 24, 25 and 26 the two speed profiles follow the same trend, even though the ADAMS/Car model show a more oscillating behavior. As seen in all the figures, the speed profile deviates the most in cases where the input signal contains most noise, such as in figure 24.
Figure 24: Comparison of speed profile between the real vehicle and the ADAMS/Car model during the execution of the Severe lane change track.

Figure 25: Comparison of speed profile between the real vehicle and the ADAMS/Car model during the execution of the Slalom track.
Figure 26: Comparison of speed profile between the real vehicle and the ADAMS/Car model during the execution of the J-turn maneuver.

6.2 Chassis displacement

The logged data from real test executions was used to generate a path file consisting of x- and y-coordinates over time with 0.01 second intervals. The idea is to try and make the ADAMS/Car model follow the exact same path as the real vehicle. As seen in Figures 27-29, the curves are slightly displaced in the x-direction (vehicle driving direction). This small difference might be caused by the ADAMS/Car driver model's inability to exactly follow the specified path due to noise in the measured data. Figure 29 illustrates how the driver model in ADAMS/Car deviates from the specified path and follows a smoother arc, thus simplifying the real vehicle motion. The path referenced in the event file is the target file used by the steering controller with no smoothing or other manipulation to alter the path shape. Therefore, any irregularities included in the path description, such as the noise coming from the logged data, are seen by the steering controller and can produce a changing steering input.
Figure 27: Comparison of path profiles between the real vehicle and the ADAMS/Car model during the execution of the Severe Lane Change track.

Figure 28: Comparison of path profiles between the real vehicle and the ADAMS/Car model during the execution of the Slalom track.
Figure 29: Comparison of path profiles between the real vehicle and the ADAMS/Car model during the execution of the J-turn maneuver.

6.3 Lateral acceleration

With the speed profile and position coordinates as input parameters the generated lateral acceleration was compared with data recovered from the real test executions, see Figures 30-32.

Figure 30: Comparison of lateral acceleration between the real vehicle and the ADAMS/Car model during the execution the Severe Lane Change track. What is
noticeable is how the lateral acceleration deviates from the measured, despite that the difference in chassis.

Figure 31: Comparison of lateral acceleration between the real vehicle and the ADAMS/Car model during the execution of the Slalom track.

Figure 32: Comparison of lateral acceleration between the real vehicle and the ADAMS/Car model during the execution the J-turn track.
Considering that the Severe lane change track and the J-turn track exhibit the largest deviations in speed and position, consistently they possess the greatest error. It's worth mentioning that the ADAMS/Car model doesn't take into account environmental influences, such as weather conditions and road surface unevenness. The real tests were conducted on an uneven ground, while in the simulated case the ground was flat. The differences in lateral acceleration might be caused due to the models difficulties to follow the specified path coordinates. The validated results for the slalom track are satisfactory where the lateral acceleration from the simulation is considered to agree reasonably good with the measured values. Despite trouble shooting for errors in the Adams model, no defects were able to be found.

6.4 Roll angle

Due to the lack of logged roll angle data, a theoretical roll angle was calculated using equations (12)-(16) and compared with the generated roll angle from the simulations in ADAMS/Car. Consequently also here the Severe Lane Change track (Figure 33) exhibits the greatest error whereas the Slalom track (Figure 34) presents the best agreement between the theoretical and simulated values. For the J-turn scenario (Figure 35) the curves follow the same trend and both curves almost reach the same peak value. Considering the deviations in path, the results are considered satisfactory.

![Figure 33: Comparison of the theoretical and the simulated roll angle during execution the Severe Lane Change track.](image-url)
Figure 34: Comparison of the theoretical and the simulated roll angle during execution of the Slalom track.

Figure 35: Comparison of the theoretical and the simulated roll angle during execution of the J-turn maneuver.
6.5 Validation discussion

With deviation in speed and position, the model is not considered valid for all the driving scenarios. Greatest inaccuracy was exhibited during the simulations of the Severe Lane Change track and the J-turn maneuver, whereas the Slalom track gave more conforming results. The difference in driving behavior might origin from several reasons. One reason might be that the front wheel driven virtual model has difficulties to follow the same trajectory as the real rear wheel driven one. The fact that the vehicle is configured to be driven on the front wheels instead of the rear might limit the model's driving ability through certain maneuvers. Although the calculated theoretical roll angle is considered arbitrary, the absence of real roll angle data contributes to more limitations in validating the model. The fact that the logged speed data, from the test runs, contained noise might also contribute to the error.

Additional sources of errors are due to the simplifications made in the model, for instance the assumptions used in the rear suspension. Because there was no possibility to measure the spring and damper characteristics of the rear suspension, the suspension data used was designed to satisfy all driving scenarios best way possible. Additional simplifications of the reality are the flat ground assumption and the tires. It is reasonable to state that the ground is rarely flat and even in reality, whereas in ADAMS the road is totally flat and with constant friction.

The Fiala tire model used in the simulation is a simplified model of a real tire. Despite very few input parameters the tire model still provides reasonable results, however it is still an approximation of the real tire. The parameters modified are; Vertical tire stiffness, Vertical damping coefficient, CSLIP (Partial derivative of longitudinal force with respect to longitudinal slip ratio at zero longitudinal slip) and CALPHA (Partial derivative of lateral force with respect to slip angle at zero slip angle). These parameters were estimated and the parameters differ between the front tires and the rear tire. Appendix C shows a screen dump of the tire data used by the virtual vehicle.

It is also worth mentioning that the ADAMS/Car model is constructed of rigid parts, whereas in reality the parts are not totally rigid. Due to simplifications made in the virtual model, the validation didn't generate satisfying results for all driving scenarios, but the model is considered good enough to help draw general conclusions about the vehicle behavior.
7 Vehicle improvement

The validation of the Smite was made without any modifications or changes in the geometry, except for the propulsion design. Considering that validation worked best for the slalom simulation, further modifications will build on these results. The idea is to investigate if there are any possible modifications that can reduce the roll angle. Because cost is an issue, improvements are to be made with the existing components without adding additional parts to the vehicle. Due to the vehicle's simple construction, the possibility to change the position of the center of gravity is very limited. To improve the handling without any advanced mechanical interventions the only options available are changing the position and angle of the front dampers, or changing the track width. These procedures are made simple due to the smart construction of the front suspension.

7.1 Modification of the front suspension system

As seen in Figure 36, the design of the front suspension system allows the spring/damper component to be mounted in different positions. Figure 37 also illustrates the different positions for the spring/damper component that will be simulated in ADAMS/Car with the aim to try and find a more optimum suspension configuration for the three wheeler. Four different lower and three top mounting positions give a total of 12 different mounting possibilities of the spring/damper component. From the equations \((7)-(11)\) the vertical spring constant was calculated to 3.70 N/mm. By adjusting the spring/damper position the goal is to examine if any of the proposed positions can generate a greater vertical spring constant, thus making the vehicle more roll resistant.

![Figure 36: The different mounting points of the front suspension system.](image-url)
Table 2 illustrates the different combinations made out of the different mounting points and resulting vertical spring constant. As can be seen in the table, the more horizontal the damper is positioned, the greater spring stiffness is generated. It is worth mentioning that these calculations are theoretical and were not validated with physical tests.

Table 2: Number of combinations with corresponding mounting points and resulting vertical spring constant.

<table>
<thead>
<tr>
<th>Nr. of combinations</th>
<th>Connected points</th>
<th>$k_w$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1 &amp; 4</td>
<td>7.40</td>
</tr>
<tr>
<td>2</td>
<td>1 &amp; 5</td>
<td>9.62</td>
</tr>
<tr>
<td>3</td>
<td>1 &amp; 6</td>
<td>12.8</td>
</tr>
<tr>
<td>4</td>
<td>1 &amp; 7</td>
<td>16.4</td>
</tr>
<tr>
<td>5</td>
<td>2 &amp; 4</td>
<td>5.20</td>
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<td>6</td>
<td>2 &amp; 5</td>
<td>7.50</td>
</tr>
<tr>
<td>7</td>
<td>2 &amp; 6</td>
<td>10.0</td>
</tr>
<tr>
<td>8</td>
<td>2 &amp; 7</td>
<td>13.0</td>
</tr>
<tr>
<td>9</td>
<td>3 &amp; 4</td>
<td>3.90</td>
</tr>
<tr>
<td>10</td>
<td>3 &amp; 5</td>
<td>5.46</td>
</tr>
<tr>
<td>11 (original setup)</td>
<td>3 &amp; 6</td>
<td>7.83</td>
</tr>
<tr>
<td>12</td>
<td>3 &amp; 7</td>
<td>9.92</td>
</tr>
</tbody>
</table>

7.2 Adjusting the track width

An easy way to improve the stability of the vehicle is to increase the track width. Due to limitation in the suspension geometry, the track width can't be widened without spacers. Spacers are cylindrical plates mounted between the rim and the hub, thus providing additional width. In this simulation theoretical spacers with a width of 30 mm will be used, thus increasing the total track width with 60 mm. Simulation with the spacers will be carried out with the original vehicle setup and also with the modified suspension setup combination number 4, hence the configuration that generated the highest vertical spring constant.
8 Comparison between the original and the improved vehicle

The original vehicle setup was compared with 13 different front suspension configurations. These 13 configurations consist of 11 different placements of the spring/damper component, and additional two more configurations where the trackwidth was extended on both the original vehicle version and on suspension setup number 4. The driving scenario used for this simulation was the Slalom track, hence the validation was most successful for this driving scenario and the logged data conformed best with the simulated values. The plot containing the roll angle for all the simulations can be viewed in APPENDIX B, but to easily overview the results a chart has been complied showing all of the suspension configurations and resulting changes in roll angle.

Table 3: Complete chart of all the configurations and resulting changes in roll angle.

<table>
<thead>
<tr>
<th>Nr. of combinations</th>
<th>$k_w$</th>
<th>Max. Roll angle</th>
<th>Min. Roll angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7.40</td>
<td>1.00</td>
<td>-1.14</td>
</tr>
<tr>
<td>2</td>
<td>9.62</td>
<td>0.99</td>
<td>-1.09</td>
</tr>
<tr>
<td>3</td>
<td>12.8</td>
<td>0.92</td>
<td>-1.05</td>
</tr>
<tr>
<td>4</td>
<td>16.4</td>
<td>0.84</td>
<td>-0.92</td>
</tr>
<tr>
<td>5</td>
<td>5.20</td>
<td>1.06</td>
<td>-1.22</td>
</tr>
<tr>
<td>6</td>
<td>7.50</td>
<td>0.96</td>
<td>-1.06</td>
</tr>
<tr>
<td>7</td>
<td>10.0</td>
<td>0.89</td>
<td>-1.00</td>
</tr>
<tr>
<td>8</td>
<td>13.0</td>
<td>0.90</td>
<td>-1.02</td>
</tr>
<tr>
<td>9</td>
<td>3.90</td>
<td>1.03</td>
<td>-1.15</td>
</tr>
<tr>
<td>10</td>
<td>5.46</td>
<td>0.93</td>
<td>-1.06</td>
</tr>
<tr>
<td>11 (original setup)</td>
<td>7.83</td>
<td>0.90</td>
<td>-1.04</td>
</tr>
<tr>
<td>12</td>
<td>9.92</td>
<td>0.85</td>
<td>-0.95</td>
</tr>
<tr>
<td>11 + spacers</td>
<td>7.83</td>
<td>0.86</td>
<td>-0.93</td>
</tr>
<tr>
<td>4 + spacers</td>
<td>16.4</td>
<td>0.81</td>
<td>-0.89</td>
</tr>
</tbody>
</table>
8.1 Discussion

The purpose with the simulations of the modified vehicle was to find a more roll resistant configuration of the front suspension system. Comparison between the different combinations shows that in cases where the vertical spring constant exhibited the greater value also resulted in lower roll angle. But this conclusion is not valid for every case. For cases where the vertical spring stiffness, $k_v$ varied between 10.0 N/mm-13.0 N/mm the roll angle didn't follow a consistent trend. Combination nr. 4 generated the best results in terms of roll angle reduction, and even better results were achieved when mounting spacers on that same configuration. The greatest improvement obtained, if excluding the implementation of spacers, was 0.18 degrees, and if that would be perceived by the driver is unclear. Regarding changes in suspension configuration it's worth mentioning that every combinatorial placement of the spring/damper component changed the complete geometry of the suspension, thus leading to changes in ride height and the track width due to the varying positioning of the control arms.
9 Future work

To get an overall better understanding of the three wheeler, it is important to further enhance the resemblance between the real vehicle and the virtual model. A lot of approximations and estimations were used in this thesis due to uncertain and absent data, and as a result of this the model didn't exhibit high validation accuracy through all of the presented driving scenarios. If there is interest to improve the work in this thesis, it can be done by acquiring more detailed information about the vehicle, for instance more information about the rear suspension, the tires and moment of inertia. The rear spring/damper component needs to be disassembled and tested in the same way as the front component, as for the tires and the moment of inertia, this data has to be measured in some way.

A more accurate placement of the test equipment should also contribute to a more conforming simulations. Due to limited mounting possibilities the test equipment could not be placed in a desired position, which is the location of the center of gravity. A compromise was made and the equipment was placed on the same longitudinal coordinate as the center of gravity leaving out the height, which is approximately 50% lower than desired. The outcome of this measure may have affected the model's ability to perform more satisfactory. Checking the test equipment's functionality before undertaking any measurements is crucial. Due to unexplained malfunction, the test equipment failed to log yaw rate, which would have been useful to have, in order to obtain a more accurate validation.

Regarding the ADAMS/Car model and room for improvements, greatest emphasis should be on the powertrain modification. In order to pursue more realistic properties of the virtual vehicle the powertrain template ought to be modified to propel just the rear wheel. The existing ADAMS/Car model is not flawless but it serves its purpose to provide general feedback about the vehicle behavior.
References


APPENDIX A: Damper characteristics - The force-amplitude diagram for all amplitudes

Figure 38: Force-amplitude diagram for the amplitude 2 mm.

Figure 39: Force-amplitude diagram for the amplitude 5 mm.
Figure 40: Force-amplitude diagram for the amplitude 10 mm.

Figure 41: Force-amplitude diagram for the amplitude 15 mm.
APPENDIX B: Roll angle for all combinatorial placements of the spring/damper component

Figure 42: The roll angle for the original, the best and the worst suspension setup.

Figure 43: The roll angle for all the 12 different combinations (not including configurations with spacers).
APPENDIX C : Fiala tire data for the virtual vehicle model

Figure 44: The input parameters for the front tires.

Figure 45: The input parameters for the rear tire.
APPENDIX D : Confidential data

- Technical description
- Outer dimensions
- Chassis dimension
- Body specifications.