CHALMERS

Difference in chassis setup due to different body variants.

Master's Thesis in the Masters Programme Automotive Engineering

Lucas Börjesson, Peter Wiborg

Department of Applied Mechanics
Division of Vehicle Engineering and Autonomous Systems
Chalmers University of Technology
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Abstract

The automotive industry is one of the world's most important economic sectors and it is important to reduce cost in development. One way of doing this is to reduce the test time and the number of prototype vehicles built. This thesis is investigating if it is possible to test the lateral dynamic behaviour of a station wagon with only simulations for this vehicle and physical test made with a modified sedan based on the same platform. The simulations were made in VI-Car Realtime which is a table based simulation software. The thesis includes the building procedure of a simulation model in this software, how to adapt the simulation model to the behaviour of a physical vehicle and modifications to this vehicle in order to meet requirements. It was found out that this probably is possible although this study was not extensive enough. The resulting characteristics of the modified sedan is close to the characteristics of the station wagon, but the standard sedan is also very close. This makes it hard to know if the results are due to modifications or from the fact that the sedan and the station wagon are much alike from the beginning. In order to determine this further work would need to be done.

Acknowledgements

In this master thesis the difference in chassis setup due to different body variants has been studied in order to see if it is possible to predict and tune the chassis behaviour for a wagon, with physical testing with a modified sedan and simulations with a vehicle dynamic simulation software. The thesis is a partial requirement for the Master of Science degree at the Masters programme Automotive Engineering at Chalmers University of Technology, Gothenburg, Sweden. The thesis has been performed at the Vehicle Dynamics & Calibration division at Volvo Car Corporation, Gothenburg, Sweden from January to June 2013. We would like to thank our supervisor at the Vehicle Dynamics & Calibration division Egbert Bakker and our professor at Chalmers Mathias Lidberg for their help and guidance through the project. The head of the Vehicle Dynamics & Calibration division Erik Axelsson for the help with resources and all the administrative work within Volvo. Per Hesslund for the help regarding the steering robot. Joakim Rydholm and Lars-Åke Skoglund for the help at Hällered Proving Ground. Our colleagues Anton, Christoffer and Erik for their role as speaking-partners and and last but not least all the co-workers at Vehicle Dynamics CAE and Vehicle Dynamics & Calibration for their helpfulness.

Lucas Börjesson & Peter Wiborg, Gothenburg, July 1, 2013

List of Notations

K&C - Kinematic and Compliance
CoG - Center of Gravity
Sxx - Sedan Car
Vxx - Station Wagon
Mxx - Modified Sedan
SWA - Steering Wheel Angle
ARB - Anti-roll bar
DOF - Degrees of Freedom

Contents

1

Introduction

THE AUTOMOTIVE INDUSTRY is one of the world's most important economic sectors and with about 34 passenger car manufacturers[1] in Volvo Cars segment worldwide it is also one of the toughest markets to compete in. In the year 2012 63 million passenger cars were built worldwide, 30 million of these where built in Europe, USA and Japan [2]. To maintain a strong role on the market it is important to offer the customers up to date products at an attractive price. In the automotive industry today it is therefore getting more important to shorten the time when developing a new car model as well as making the process more cost effective.

1.1 Background

In order to shorten the development time and at the same time lowering the costs, Volvo Cars wants to investigate if it, within a couple of years, would be possible to predict and tune the chassis behaviour for a wagon, with physical testing with a modified sedan and simulations with a vehicle dynamic simulation software. The fact that the sedan, Sxx and the station wagon Vxx share the same platform[1] and have the same suspension layout, track width and wheelbase makes them especially suitable for the study. The study will be focused on lateral vehicle dynamics and can be broken down into three main areas:

- Map out what differentiates a wagon from a sedan structurally (aerodynamic properties, chassis/body stiffness and centre of gravity position etc). How these differences affect the vehicle in terms of stability, lateral performance, roll etc.

- Construct a methodology for how to predict and tune the chassis behaviour for a wagon, with physical testing with a modified sedan and simulations with vehicle dynamic simulation software.

[1]See Appendix A.2.

The ambition is that it will only be necessary to manufacture a prototype of one of the body styles and still know how both the sedan and station wagon will behave in terms of vehicle dynamics.

1.2 Scope

The thesis work aims to investigate if it is possible to predict and tune the chassis behaviour of a station wagon under the conditions described in the background.

To put it succinctly:

- A simulation model of the sedan (Sxx) will be built from scratch and validated against the sedan (Sxx) in production.
- The simulation model of the sedan (Sxx) will be modified to the wagon (Vxx) specifications in terms of CoG and inertia. The modifications will be limited to those that can be implemented to the physical sedan (Sxx).
- The modified simulation model of the sedan (Sxx) will be improved to behave as similar as possible to the standard sedan simulation model (Sxx).
- A physical sedan (Sxx) will be modified according to the improved simulation model of the sedan (Sxx), from here on named Mxx, modified xx.
- The physical modified sedan (Mxx) will be compared against a physical sedan (Sxx) and wagon (Vxx).

1.3 Problem Definition

- How well do the simulation model of the Sxx represent the Sxx in production?
- How do the modified sedan with changed specifications for inertia, CoG, springs, dampers and anti-roll bars behave in terms of vehicle handling compared to a standard station wagon in production? How similar are the detailed specifications for components such as springs and anti-roll bars given by this study and those used in production?
- What are the main differences from a vehicle dynamics and design point of view between a sedan and a station wagon based on the same platform? Vehicle dynamics point of view includes areas such as handling stability and sensitivity to steering (over-/understeer), roll, and lateral performance. With construction means for example spring stiffness, anti-roll bar stiffness, inertia and center of gravity.
- Is it possible to predict the vehicle dynamic behaviour of a station wagon by performing physical testing with a modified sedan within the described limitations?

1.4 Delimitations

The study will only be done for the sedan model, Volvo Sxx, and the station wagon model, Volvo Vxx. The only changes that will be made to the suspension of the sedan in production are; springs, dampers and anti-roll bars. There will be no changes in suspension geometry. The centre of gravity will be changed in order to match the properties of the station wagon in production although no changes will be done to the stiffness or body shape. The comfort of the modified vehicle cannot be affected. The test procedures are limited to Volvo Cars standard tests for vehicle handling evaluation. The modifications to the Sxx physical model that should behave like a Vxx are limited to changed mass and centre of gravity (CoG) to represent the Vxx model.

2

Theory

It is important to know some basic theory and what is influencing the results in order to understand the work that was done in this thesis. The tyres are of great importance since they have contact with the road, if they are not correctly simulated the results will be useless. Inertia and center of gravity are two other parameters that greatly change the behaviour of the vehicles. A vehicle can be simulated based on only these two parameters and still give a hint of how a complete simulation model will behave. Last there is the aerodynamic influence. Since this thesis investigates the difference between a sedan and a station wagon, which has very different body shapes, the aerodynamic properties might differ and change the simulation results.

2.1 Tyres

Being the link between the road and the vehicle the tyre play a significant role when it comes to the vehicles overall performance. The tyres main tasks are:

- Carry the static load
- Generate longitudinal forces (brake and traction forces)
- Generate lateral forces (cornering forces)
- Isolate vertical disturbances
- Rolling with low resistance
- Rolling with low noise emissions

A road tyre is a compromise between these tasks. Stiffer sidewalls will generate larger lateral forces and therefore better cornering performance but on the other hand stiffer sidewalls will lead to less vertical compliance in the tyre and therefore worse comfort. A higher friction coefficient will give better handling performance but also a higher fuel consumption. In simulations computer models of tyres are used and it is therefore important to understand how complex and hard it is to model a tyre.

Tyre Deflection

To start with the fundamental mechanic relationship that a wheel that rolls without sliding have translational speed that is equal to the radius times the angular velocity of the wheel does nost hold for a modern car tyre when it is deflected. The deflection of a rolling tyre does not look as the idealized case displayed in Figure 2.1 either but rather like the one in Figure 2.2 where the deformation arises at the leading edge.

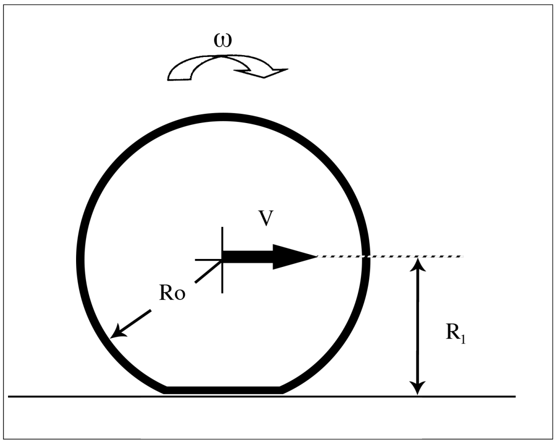

Figure 2.1: Idealized Tyre. [4]

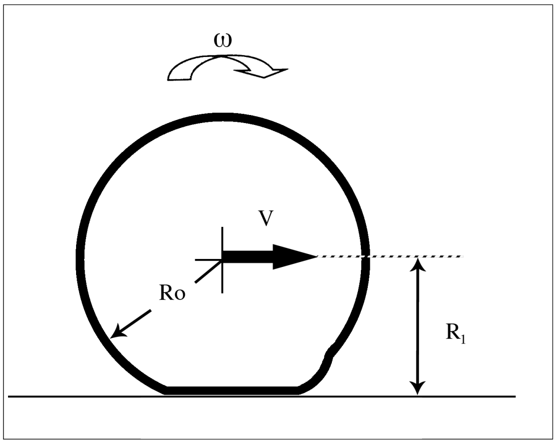

Figure 2.2: Real Tyre. [4]

The rolling radius is therefore not measurable. Instead of a measured radius a calculated effective radius, R_e is used. R_e is a proportional constant between the tires translational and angular velocity, $R_o > R_e > R_l$. The lack of a kinematical relationship between the translational speed and the rotational speed of the tyre has led to the definition slip, which is defined as the relative motion between the tire and the road surface it is moving on and a reference speed. The reference speed can be the translational speed of the tire or the circumferential speed of the tire. For a driven wheel the longitudinal slip is defined:

$$S_x = \frac{R_e\omega - V}{R_e\omega} \tag{2.1}$$

and for a braking wheel:

$$S_x = \frac{R_e\omega - V}{V} \tag{2.2}$$

2.2 Inertia and CoG

In a Kinematic & Compliance, K&C, rig it is possible to measure moments of inertia and centre of gravity (CoG). These parameters are essential for the behaviour of the vehicle and with them incorrect, the simulation model will be inaccurate. The CoG will mainly determine three outcomes. First of all the static load on the front and rear axle which depends on the longitudinal position of the CoG. The understeer gradient will change when moving the CoG along the x-axis of the vehicle. The second outcome is the load transfer, both in longitudinal and lateral direction. This is determined by the height of the CoG, the higher the CoG is placed the more load transfer will occur. Load transfer is almost never wanted, except for certain race forms, such as dragrace when all available grip should be transferred to the propelling rear axle. The final outcome is when the CoG is offset in the lateral direction of the vehicle. If this is the case, the car might steer itself when driving in a straight line.

2.3 Aerodynamic Influence

The difference in behaviour caused by aerodynamic effects between two vehicles can be significant. There is the drag force acting as a retarding force on the vehicle, a lift force that will either increase or reduce the normal force of the tyres and finally there might be vortices and other aerodynamic phenomena making the vehicle unstable. The lift force is mostly positive on standard road cars, meaning the resulting force is reducing the normal force of the tyres. On some sports cars the lift force is negative which gives more normal forces on the tyres. This will allow higher lateral accelerations but the rolling resistance will also increase. In order to calculate the aerodynamic drag and lift, Equation 2.3 and 2.4 is used.

$$F_D = \frac{\rho V^2 A C_D}{2} \tag{2.3}$$

$$F_L = \frac{\rho V^2 A C_L}{2} \tag{2.4}$$

where F_D is the drag force, F_L is the lift force; ρ is the density of the surrounding media, in this case air, V is the velocity of the vehicle, A is the projected area in the Y-Z plane, or as it is usually called, frontal area, C_D is the drag coefficient and C_L is the lift coefficient. In order to calculate the difference in drag and lift forces on a Sxx and a Vxx Equation 2.5 and 2.6 can be used since the velocity and the surrounding media are the same on both vehicles, and the frontal areas are almost identical.

$$\Delta F_D = \frac{\rho V^2 A}{2}(C_{D,Vxx} - C_{D,Sxx}) \tag{2.5}$$

$$\Delta F_L = \frac{\rho V^2 A}{2}(C_{L,Vxx} - C_{L,Sxx}) \tag{2.6}$$

3

VI-Car

VI-Car Realtime is a realtime vehicle simulation software. From a vehicle dynamics perspective it can be used in order to optimize and control the vehicle performance as well as visualize how different parameters influence the overall vehicle dynamics. A VI-Car model can be built from an Adams Car model, which is a geometry based model of a full vehicle, or based on results from K&C test rigs. The simplified vehicle model includes 14 degrees of freedom, DOF, distributed as follow:

- The vehicle includes five rigid parts: Vehicle chassis (sprung mass) and four wheel parts (unsprung masses).
- The vehicle chassis has 6 DOFs while wheel parts have 2 DOFs each (vertical motion with respect to the vehicle body and wheel spin).
- The suspension models do not have linkages or bushing and the steering system does not have parts for the steering wheel or rack.
- Suspension and steering system properties (kinematic, compliance and component data) are described by tables.
- Vehicle subsystems such as brakes and powertrain are described by differential and algebraic equations.
- Body chassis torsional compliance can be added.

The simplified vehicle runs much faster than real time which makes it an efficient tool for vehicle dynamics optimization. VI-Car is compatible with MATLAB and tyre files given from tyre manufactures in order to simulate tyres [5].

3.1 Suspension Model

An important and time consuming part is the construction of an accurate model of the Sxx in VI-Car. The software need input parameters in order to build a model. These values can either be a linear function with constant gradient or a more accurate curve consisting of a number of data points. The later of these two choices were chosen to most of the parameters in order to get the most accurate model possible. The majority of suspension and steering system properties were given by the result files from K&C measurements of a Sxx (fwd, 2.0l petrol engine, auto). The following tests were performed:

- Aligning Torque
- Lateral Force Compliance
- Longitudinal Force Compliance
- Vehicle Roll
- Steering
- Vertical Force versus Vertical Movement

All tests where done several times with different configurations, for example engine, anti-roll bar, brakes in on and off mode. The fact that the same tests have been done with different configurations makes it possible to calculate wanted parameters such as anti-roll bar stiffness. The data was stored in big ASCII coded documents with a lot of data points from each measurement. In order to reduce the number of data points to get a more manageable array of numbers, the curve was fitted with a fifth grade polynomial. With the new calculated function it was possible to create a new smaller array that was exported to VI-Car. The reason of this was that the raw data had several thousands data points, to use them all would have resulted in very big tables and slow simulations. There were a number of parameters that was not measured in the K&C data; camber, toe and caster compliance due to forces in F_z and moments in M_z. Since there were no data available it was decided to set the gradient of M_z to zero, which means there are no compliances in M_z in this model. The compliance in F_z where available as a standard value in Volvo documents that was used in the model.

VI-Car requires more input parameters than the available data from the K&C measurements. Examples of such data are: spring rate, damper data, bump stop data, unsprung mass weight and brake- and powertrain data. This data were provided from internal Volvo Cars documents. In order to validate the model the same load cases used in the K&C rig were simulated on the VI-Car model. Two of the inputs in the steering system needed inputs in three dimensions; camber/caster, steering input and jounce. There were no available tests for these specific movements and in order to get the needed values, a relationship between jounce and camber/caster were used from a test including these without steering input. This gave the correct data when the steering input was zero and for the remaining steering inputs it was assumed that the same distribution could be used. This assumption was made because in other models it was seen that the variance between different steering inputs were about the same as the one around zero. The result were later imported to MATLAB and compared with the original data in order to confirm an accurate model of the suspension geometry.

One of the most important inputs to the VI-Car model is the tyre. These are files provided by the tyre manufactures. How well these tyre files represent a real tyre is critical for how well the simulation model will correspond to a real vehicle. In order to determine a proper tyre for the evaluation it is good to determine what kind of tests are to be done (see Chapter 5). Since this study mainly focused on the lateral dynamics, most of the tests were configured this way. The tyre Continental Sport Contact 3 235/45 R17 was chosen, mainly because this is a tyre commonly used by Volvo. There were tyre files available for this tyre in mode 13, which means that they work in lateral cases and includes relaxation effects.

4

Simulations

THE STUDY mainly consist of simulations but verifications of the models will also be done with physical tests. The simulations were done in VI-Car and all surrounding work were made with MATLAB. The tests were done on Volvos test track in Hällered. Three different vehicles were studied; a Volvo Sxx, one Volvo Vxx and a modified Volvo Sxx.

4.1 Adapting the Sxx Simulation Model

To verify the Sxx simulation model, tests were performed at Hällered proving ground. To get the model as accurate as possible the test vehicle had a full tank of fuel during all the tests. The test vehicle was also weighed in three different configurations:

- Unloaded car except a full tank of fuel
- With the driver and a full tank of fuel
- With the driver, the robot and a full tank of fuel

The optimal scenario would be that the test where performed with the same vehicle that was used for the measurements in the K&C rig. In this thesis this was not possible and the test vehicle had a different equipment level and engine/gearbox configuration than the measured car. The test vehicle weighed 45.5 kg more than the vehicle from the K&C rig. With the function "Body Setup Data" in VI-Car it was possible to add the weight at the right position by comparing the corner weights between the simulation model and the tested vehicle. In the vertical direction the weight was positioned at the same height as the CoG. It was assumed that the vertical CoG was the same as for the measured vehicle. The software would not only add the weight to the new total weight of the vehicle but also calculate the new total moment of inertia. This made it possible to use the measured moments of inertia from the K&C rig as a basis for the vehicle model. The two other measured configurations made it possible to position weights representing the driver and the robot which weighs 77 and 81.5 kg respectively. The vertical position for the driver weight was measured from the ground according to studies stating that a drivers CoG approximately is in the same height as the driver's navel when driving. The vertical position for the robot was estimated. To get an even more realistic behaviour, the stiffness of the chassis was added to the model.

With the weight and weight distribution for the simulation model according to the test vehicle the evaluation and adaption process could be started. For each of the performed manoeuvres described in Chapter 5 data such as steering wheel angle (SWA), lateral acceleration (a_y), roll, yaw, roll rate, pitch, longitudinal velocity and pitch rate where compared between the simulation and the physical model.

4.2 Building the Mxx Model

As for the Sxx a wagon, Vxx had been measured in the K&C rig, for the Vxx only the CoG and inertia data was available. Since there is no difference in suspension or steering geometry between the Sxx and the Vxx it was possible to add weights to the Sxx model in order to match VXX in terms of CoG and inertia.

4.2.1 Basic VXX Simulation Model

A basic Vxx simulation model was built by modifying the Sxx simulation model with the CoG, inertia data and mass for the K&C Vxx. Like in the Sxx case the vehicle measured in the K&C rig did not match the Vxx that would be used for the physical test in terms of equipment level and engine/gearbox configuration. Therefore two weights with a sum of 62 kg were added to the Vxx simulation model in order to match the corner weight of the physical Vxx. In the vertical direction the weights were positioned at the same height as the vehicles CoG. The springs preload was changed so that the simulation Vxx would have the same ride height and pitch angel as the Sxx.

4.2.2 Mxx Simulation Model

The Mxx simulation model was built by modifying the Sxx simulation model with the CoG and inertia data for the K&C Vxx. The physical Sxx weighs 63.5 kg less than the physical Vxx (both with a full fuel tank). A 63.5 kg weights was therefore added to the Mxx simulation model, the longitudinal and lateral position where given by the corner weight measurement of the Vxx. The springs preload was changed so that the simulation Mxx would have the same ride height and pitch angle as the Sxx. The vertical position of the weight was given by comparisons between the Mxx and the Vxx simulation models in VI-Car, the weight was moved until the Mxx and Vxx behaved identical in simulations.

4.2.3 Optimization of the Mxx Simulation Model

With the correct inertia and weight for the Mxx the optimization work could start. The Sxx and the Vxx have the same target values in terms of performance and road handling. The goal with the optimization process was therefore to get the Mxx to behave as close to the Sxx as possible, the higher weight and CoG limiting it from behave identical. A limitation within this thesis work is that there was neither budget nor time to manufacture springs and anti-roll bars according to the desired specification. To evaluate if it possible to tune the chassis for a wagon with only simulations and physical testing with a sedan it is crucial to test the optimized springs and anti-roll bars on the physical Mxx. Therefore the optimization process of the Mxx in VI-Car has to be a compromise by making the Mxx as good as possible with the available anti-roll bars and springs. Before changing ARB:s and springs on the Mxx the weight of 63.5 kg was mounted to the vehicle. The coordinates for the weight where given by the simulation model in VI-Car. The weight was mounted in the trunk of the vehicle as can be seen in Figure 4.1.. The lateral and longitudinal position of the weight where adjusted by measuring the corner weight of the vehicle and matching the result to the Vxx results. The design of the trunk compartment being a limiting factor, the vertical position was set as close to the simulation value as possible. With the weight in place the springs and ARB:s could be changed and the spring preload adjusted so that the Mxx:s ride height complies to the Sxx:s.

Figure 4.1: Weights in the trunk of the Mxx

4.3 Damper and Spring Data

It is hard to get a good estimation of the springs from the K&C, and for the dampers it is not possible to get reliable data from these tests since the test are static and not dynamic. In this thesis it was assumed that such data do exist for one of the body styles, in this particularly case the Sxx. The choice of the springs is made according to a table based on the axle weight of the vehicle. The dampers are more dependent on the type of vehicle, if it is sport or comfort focused. All this data were available when building the simulation model of the Sxx.

4.4 Auxiliary Anti-Roll Force

When testing the vehicle roll, vertical forces are applied to all four wheels and the wheel travel for each wheel is measured. Since the roll test had been done both with the front- and rear anti-roll bar mounted to the vehicle and without them the auxiliary anti-roll forces for front and rear could be calculated as follow:

$$z_f = z_{lfa} - z_{rfa} \tag{4.1}$$

where z is the wheel travel in the vertical direction; f = front, r = rear, a = anti-roll bar, na = no anti-roll bar.

$$F_{zf} = \frac{(F_{zlfna} - F_{zlfa}) - (F_{zrfna} - F_{zrfa})}{2} \tag{4.2}$$

where F_{zf} is the force applied by the ARB to each wheel in the vertical direction. $F = force$. The ARB effect can be calculated by taking the difference in the vertical force for each wheel with and without the ARB mounted. In theory the difference should be the same whether you look at the left or right wheel. In practice this is not the case and therefore an average value between the two wheels is used. The auxiliary anti-roll force input to VI-Car should be in the format force/length [N/mm]. By plotting F_{zf} against z_f, the gradient i.e. auxiliary anti-roll force can be obtained from Equation 4.3.

$$\frac{\Delta F_{zf}}{\Delta z_f} = \frac{F_{zf,max} - F_{zf,min}}{z_{f,max} - z_{f,min}} \tag{4.3}$$

The auxiliary anti-roll force where calculated in the same way for the rear.

4.5 Aerodynamic Influence

Since all the tests in this study are made with a steering robot, it is not a problem if there is more aerodynamic resistance on one of the vehicles. This is because the robot is maintaining speed, if there is more resistance the robot will increase the throttle demand in order to overcome the resistance. The lift force has the potential to be a problem. Using Equation 2.6 with the maximum velocity used during the tests, $V = 80$ km/h, the difference in lift force between the two vehicles will be 40 N which is negligible compared to the difference in normal force [6]. When it comes to instability due to vortices and similar aerodynamic phenomena, the maximum testing velocity is too low for any major differences due to aerodynamic effects of this kind.

5

Tests

In order to test and verify the performance of a prototype vehicle it is beneficial to do a number of tests. These tests can also be done to verify a simulation model which is going to be done in this case. The following tests were done both physically with a real car as well as with simulations in VI-Car.

5.1 Constant Radius

The vehicle is driven in a constant radius turn in order to determine the steady state turning performance of the vehicle. The vehicles speed is slowly increased until the vehicle no longer can maintain the desired path. This displays the vehicles steady state performance and roll dynamics for the complete driving spectrum, from normal turning to maximum corner performance. The manoeuvre will be done for both initial left and right hand turns.

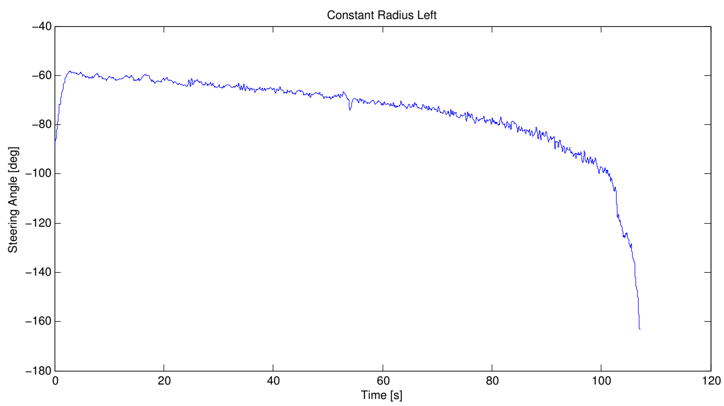

Figure 5.1: Constant Radius

5.2 Sine with Dwell Steer

The vehicle is undergoing a manoeuvre to simulate collision avoidance. The vehicle experience high lateral g forces during the test which gives good results when the vehicle is on the limit. The purpose with the tests is to objectively determine the vehicles transient response behaviour (yaw stability and response).

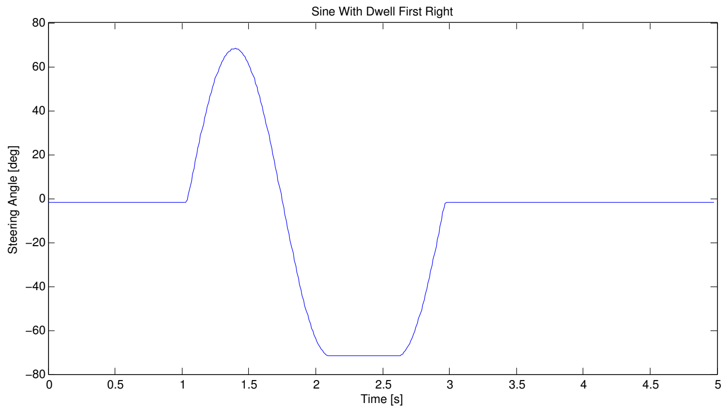

Figure 5.2: Sine With Dwell

5.3 Dynamic Catch Up

The dynamic catch up test is similar to the sine with dwell test and was mainly used to see consistency between two tests. The test show how sensitive the vehicle is for a steering catch during driving. It starts with a sharp turn in one direction in order to trigger instability in the vehicle which follows with a turn in the opposite direction and then back again in order to restore the stability in the vehicle.

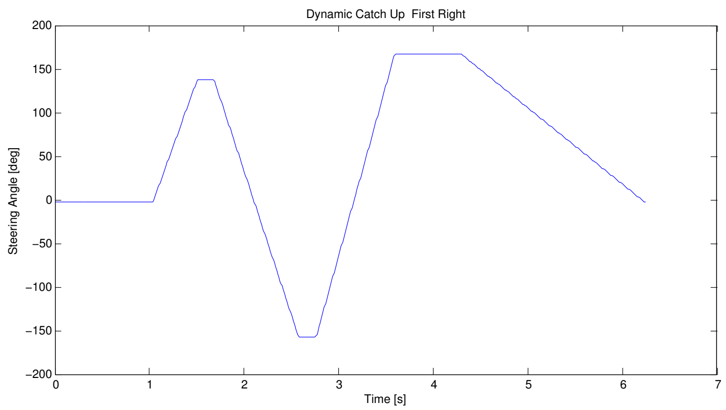

Figure 5.3: Dynamic Catch Up

5.4 High G Swept Steering

The vehicle will make a turn with speed and turning sufficient enough to record the vehicles maximum lateral acceleration. The test determine the vehicles steady state turning performance and can characterize steady state directional and roll dynamics for a complete driving spectrum, from normal turning to limit road holding capability. The vehicle will be tested for both left and right hand turns.

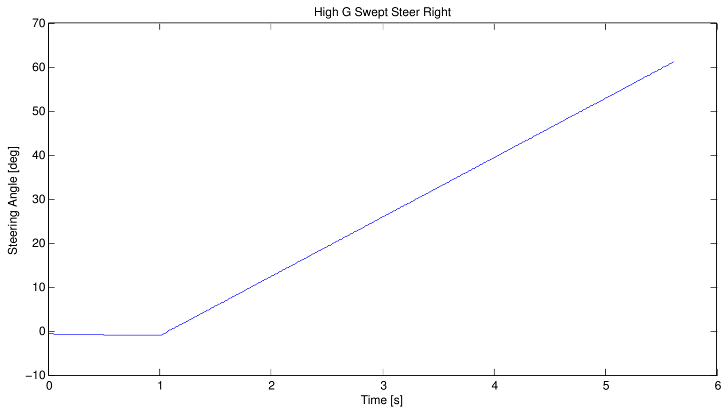

Figure 5.4: High G Swept Steer

5.5 Frequency Response

The vehicles transient and steady state turning performance in the "linear" (normal driving) domain where the maximum lateral acceleration is fairly low (0.1-0.35g) are determined by sine wave steering wheel input. The input signal starts with a high frequency and is gradually reduced to a lower frequency. It is possible to start with the low frequency but due to limitations at the test track this is the way the test was done.

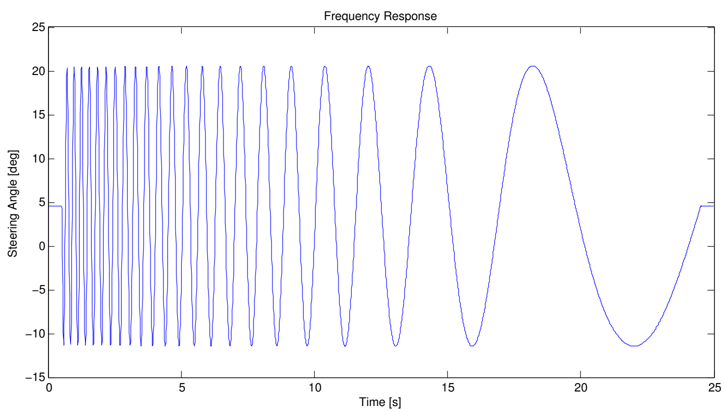

Figure 5.5: Frequency Response

5.6 On Center Steering

The purpose with the on center steering test is to measure the steering performance of the vehicle at low steering frequency and low to moderate accelerations. Conditions typical for highway driving and roads requiring mild to moderate turning.

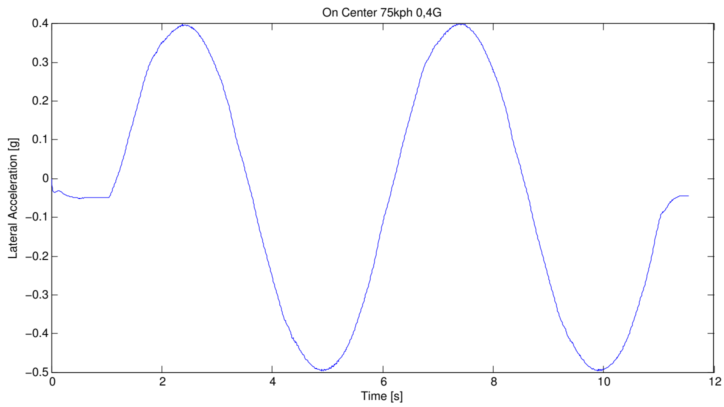

Figure 5.6: On Center Steering

5.7 Turning Diameter

The vehicle is turned 180 degrees and the turning diameter is measured to ensure that the steering ratio is the same for the S/V/M-xx as for the simulation models.

6

Results

In the results section the simulation model for the sedan, Sxx, is evaluated and compared with the actual test data for both dynamic tests and K&C measurements. The dynamic test (simulations and physical) results for the modified sedan, Mxx, are analyzed and compared with the verified sedan, Sxx. Finally the dynamic test results (simulations and physical) are compared between the three vehicles (Sxx, Vxx and Mxx).

6.1 Sxx Simulation Results

The Sxx simulation model was verified for both static (K&C) and dynamic tests. The static tests were used to verify the suspension characteristics while the dynamics tests were the basis for fine tuning the dynamic behaviour of the vehicle.

6.1.1 Suspension Modelling

The suspension geometry was modelled by using results from K&C measurements as described in Section 3.1. It was not possible to match the curves in all cases. When looking at Figure 6.1 it is seen that the two curves do not match entirely. This is because it was very hard to get the wheel angles to behave perfect in both bounce and roll.

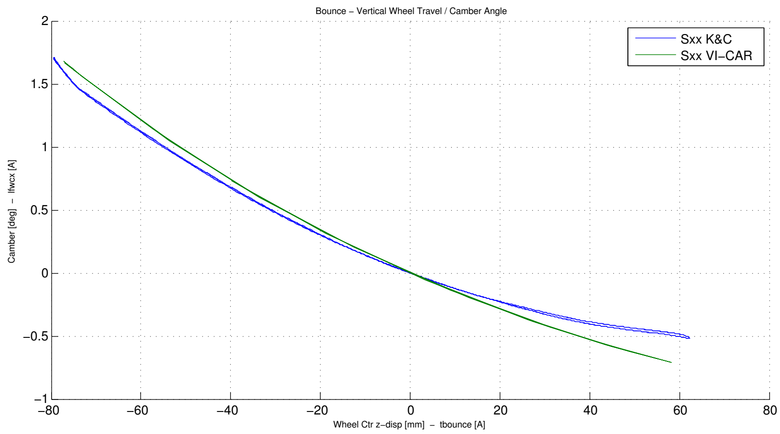

Figure 6.1: Simulation vs K&C, Bounce

This is because the test in the K&C rig is not performed in the same way as in the simulation. In the physical test the wheels are mounted to the ground and the body is rolling, in the simulation the body is fixed and the wheels are moving. Since this thesis is focused on the lateral dynamics, it was decided to optimise for roll instead of bounce. Roll is more important in lateral dynamics because of the large roll angles the vehicle experience during the lateral manoeuvres. Figure 6.2 shows how well the curve for camber change matches the measured data in the roll test.

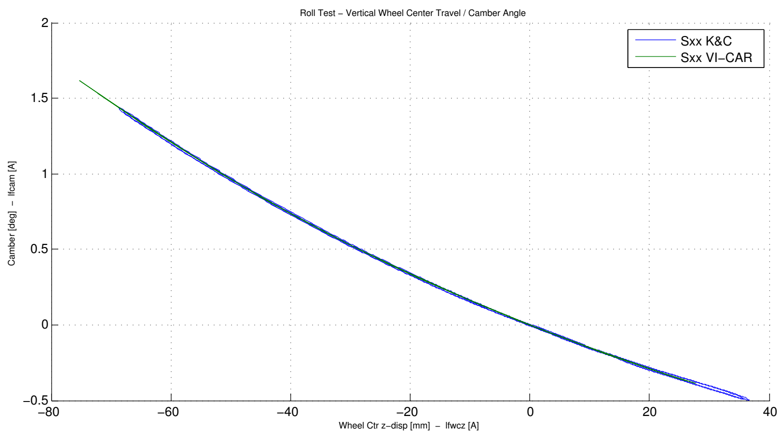

Figure 6.2: Simulation vs K&C, Roll

In Figure 6.3 it is possible to see the influence of the bump and rebound stops. The parts were the curve gradients are different from the main part of the curve are determined of this. The very beginning depends on the rebound stop properties and the end is determined by the bump stop properties. These had to be changed quite a lot in order to get a proper model. The bump stop is used to change the behaviour of the suspension during bump and roll and the rebound stop is mainly used as a safety measure to save the suspension from damage.

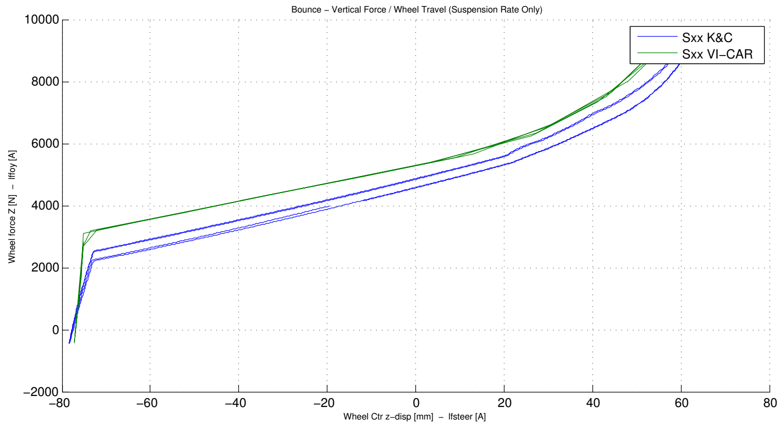

Figure 6.3: Simulation vs K&C, Bounce

Figure 6.4 shows the toe change when applying a force in the longitudinal direction. The measured data consists of two lines because the measurements are quasistatic which means that the tests are not completely static, there are movements occurring between the measurements. Since the testing procedure is cycling the system are experiencing hysteresis, which means that the measured value depends on the previous states. This phenomena was seen in all test, but the most significant effect was occurred in the compliance tests. It was not possible to get the simulation model to experience hysteresis so it was chosen to do an average of the two and apply to the model.

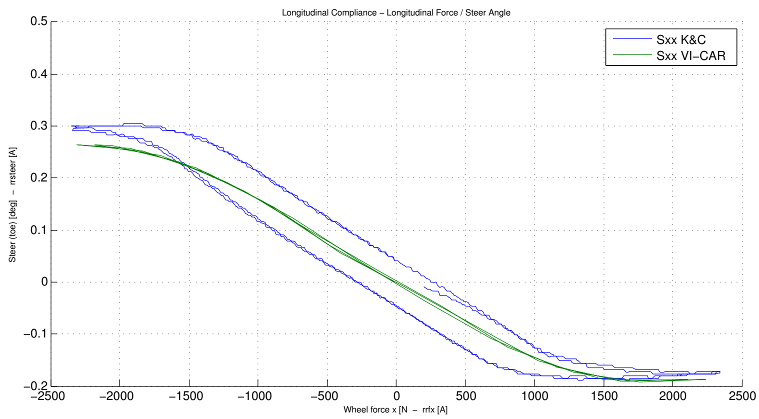

Figure 6.4: Simulation vs K&C, Longitudinal Compliance

Figure 6.5 displays the ackermann curves. As it can be seen, the difference between the left and right wheel are minimal between -10 and 10 degrees, which corresponds the steering wheel angles mainly used while driving normally in higher velocities. With greater steering wheel inputs there will be more ackermann which is beneficial in lower velocities, where greater steering angles are often used. As it can be seen the simulated values are almost identical to the measured values.

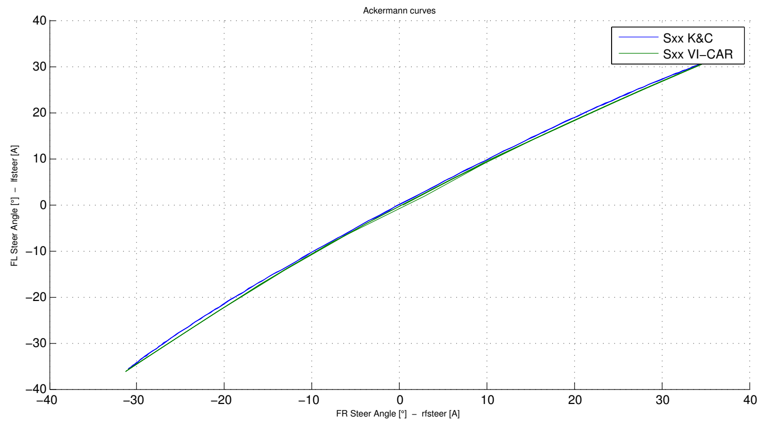

Figure 6.5: Simulation vs K&C, Ackermann

6.1.2 Dynamic Modelling

The data from the tests was analysed and compared with the simulation data. When the simulated vehicle was unmodified, with spring and anti-roll bar stiffness according to the physical vehicle, they did not match entirely as it can be seen in Figure 6.6.

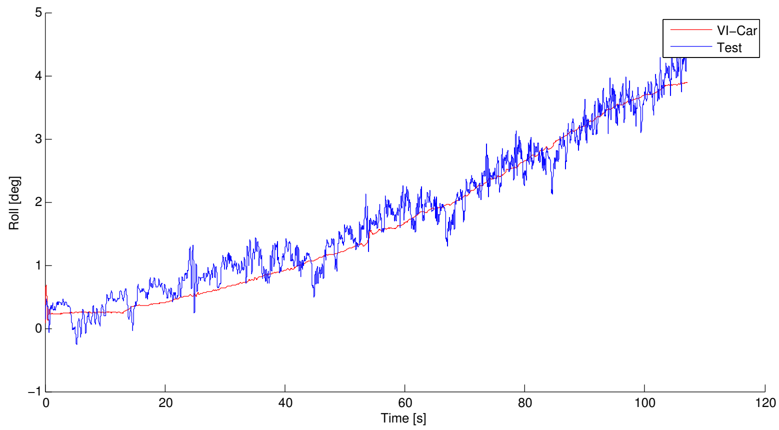

Figure 6.6: Constant Radius, Sxx No Modifications, Test vs Simulation

When looking at the roll plot it is clear that the simulation model is too stiff, it does not roll as much as the real vehicle. Since Figure 6.6 shows a constant radius test, a test were dampers do not have much influence due to very small roll rates, the best way to address this issue was to use softer springs and more compliance in the suspension components, see Figure 6.7.

The most common choice to increase roll would be to make the anti-roll bars softer, which was done. This lead to that the simulation model was too oversteered in the sine with dwell manoeuvres, see Figure 6.8. To increase the cars overall stability and shift the cars balance toward understeered behaviour the front anti-roll bar stiffness was increased

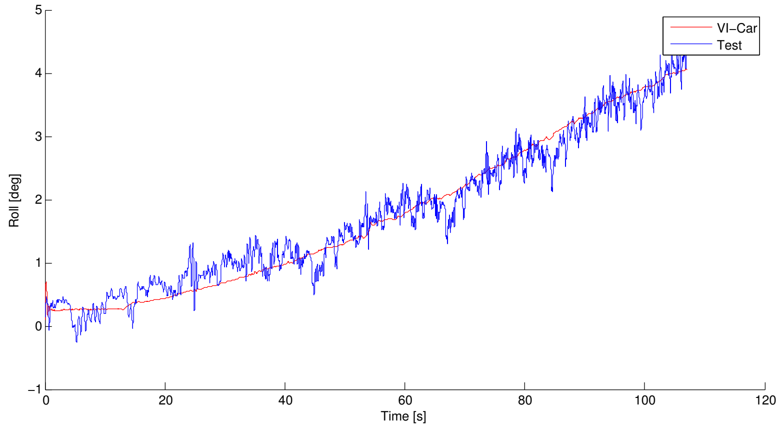

Figure 6.7: Constant Radius, Sxx, Test vs Simulation

and the rear anti-roll bar stiffness was decreased. It was possible to make the simulation equally understeered as the real vehicle, but that changed the roll behaviour too much. Since this stability problem only was a problem in a sine with dwell manoeuvre with a maximum lateral acceleration of 1 G, it was decided that a compromise between roll and oversteer could be done, see Figure 6.9. The changes made to the springs and anti-roll bars can be seen in table 6.1. Since the sine with dwell is a very dynamic manoeuvre the dampers had a big impact on the result as well. The dampers needed to be modified from the standard values for the simulation model to fit the test data.

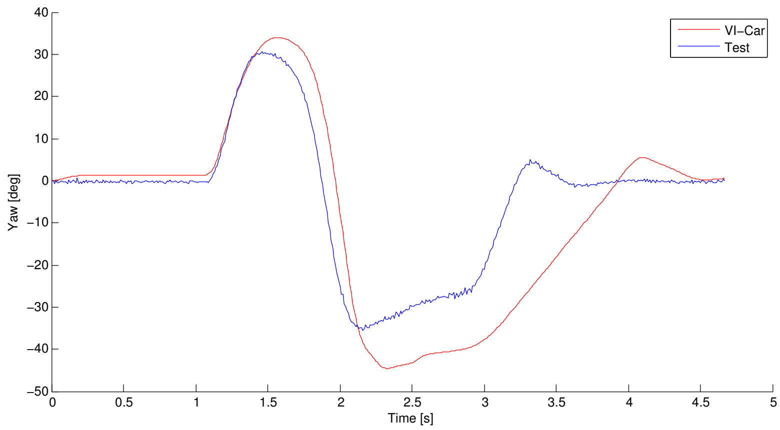

Figure 6.8: Sine With Dwell, Sxx No Modifications, Test vs Simulation

When comparing Figure 6.8 and 6.9 it can be seen that the difference after the last peak has decreased. This shows that the vehicle has become less oversteered. The hardest part was to match the pitch, see Figure 6.10. This was something that was not solved. It was not as prioritised as other parameters since the peak values are relatively small compared to roll or yaw for example.

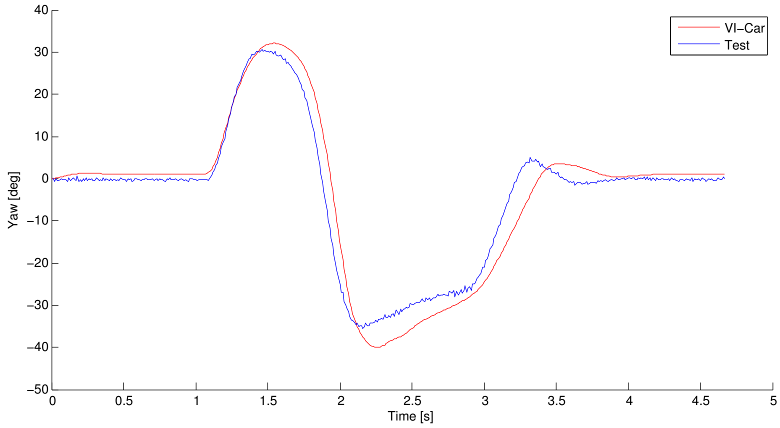

Figure 6.9: Sine With Dwell, Sxx, Test vs Simulation

Since the compromise with the anti-roll bars was done and the simulation model was a bit oversteered compared to the test vehicle, the trajectory of the two vehicle was not identical. As it can be seen in Figure 6.11 the simulated vehicle did take a wider turn than the real vehicle.

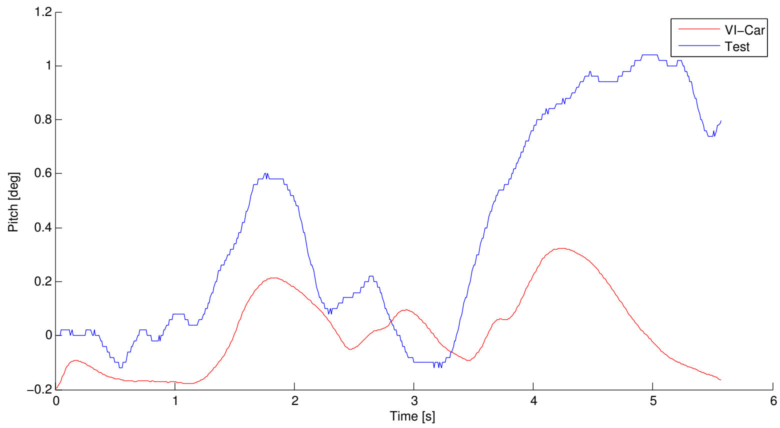

Figure 6.10: Dynamic Catch Up, Sxx, Test vs Simulation

This is because the yaw angle in the simulation will be greater than in the physical tests due to the oversteered behaviour. It is not ideal to have it like this, but compromises have to be done and this was found to be the best one. Since the Mxx will be build from these results it is assumed that if the Mxx simulation matches the Sxx simulation it will be as far of from the real values as the Sxx is and thereby be correct in the reality.

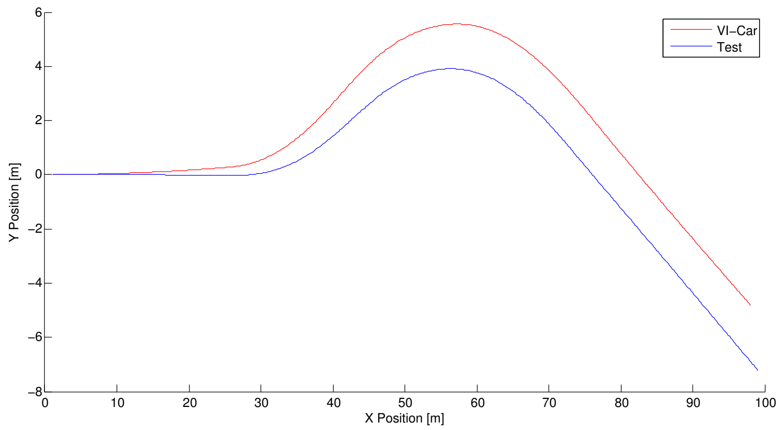

Figure 6.11: Sine With Dwell, Sxx, Test vs Simulation

Table 6.1: Changes in stiffness of springs and ARB, Sxx

	Original	Adapted	Difference
ARB Front	12.8 N/mm	21 N/mm	+ 64 %
ARB Rear	14.2 N/mm	5 N/mm	- 65 %
Springs Front	29 N/mm	26 N/mm	- 10 %
Springs Rear	35 N/mm	30 N/mm	- 14 %

6.2 Mxx Simulation Results

When the Sxx was tested and the simulation model was validated it was possible to create the Mxx model. In order to match the Vxx the CoG was raised and moved. This lead to that the behaviour of the Mxx changed, see Figure 6.12.

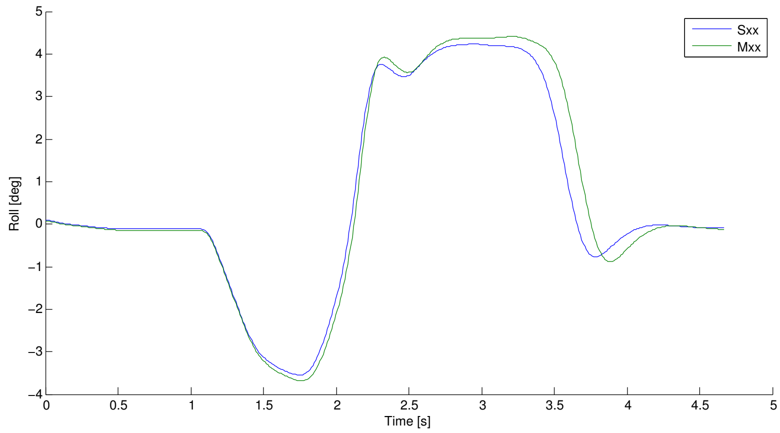

Figure 6.12: Sine With Dwell, Mxx Unmodified vs Sxx, Simulations

The higher CoG lead to more roll as it can be seen in Figure 6.12. This could be corrected with stiffer springs and anti-roll bars. This made it possible to reduce the roll to that point it behaved the same as the Sxx. When it comes to the lateral acceleration it is hard to do anything about it due to higher mass and inertia. It remained unchanged when changing springs and anti-roll bars within reasonable limits. The most significant parameter was roll, but the yaw was also slightly off, mostly in the sine with dwell test. This car also showed oversteered behaviour, see Figure 6.13.

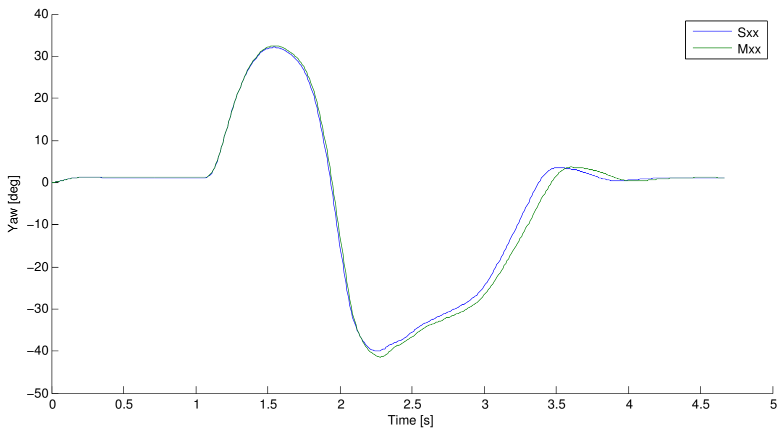

Figure 6.13: Sine With Dwell, Mxx Unmodified vs Sxx, Simulations

This could be solved with the same method as for the Sxx, stiffer anti-roll bars in the front. The chosen stiffness for the Mxx are displayed in table 6.2. Since the stiffness of springs and anti-roll bars were changed from the value of the parts mounted on the actual car for the Sxx model, it was not possible to get the spring and anti-roll bar data straight out of VI-Car. This was solved by calculate the change in percent instead and apply the result to the real springs and anti-roll bars. The dampers was not changed since it was not necessary.

Table 6.2: Stiffness of springs and ARB

	Sxx	Vxx	Mxx	Mxx Best Found Solution
ARB Front [%]	100	110	110	120
ARB Rear [%]	100	110	110	105
Springs Front [N/mm]	29	28	30	30.5
Springs Rear [N/mm]	35	37	37	36.8

The desired parts were not available in Volvo's range and since it is too expensive to manufacture special parts a compromise had to be done. Therefore are there two Mxx in the list, one with the best found configuration and one with the best compromise. The difference in spring stiffness between the two Mxx models did not make a big impact on the results. The anti-roll bars did change the behaviour to the worse. There were two alternatives for the front anti-roll bar; either 10 % stiffer or 30 % stiffer. The best result of these two was to use the one only 10 % stiffer.

Figure 6.14 shows the two different configurations of the Mxx as well as the Sxx. Here it is seen that the oversteered behaviour is compensated for. The best found solution for the Mxx is almost behaving identical to the Sxx. The chosen version is not behaving as well, but still very good.

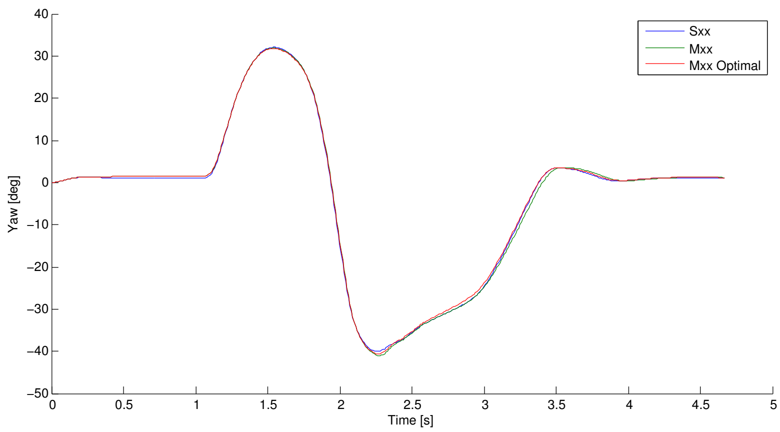

Figure 6.14: Sine With Dwell, Two Mxx vs Sxx, Simulations

The largest difference is still experienced in roll, mainly in the sine with dwell manoeuvre. When looking at Figure 6.15 it clearly shows the difference between the two Mxx versions. The compromised model is too soft which makes it roll to much, but it is less roll than it was before the changes. This will not affect the result that much, since it is still possible to see the difference between the simulation and the tests of the Mxx even though it does not behave exactly as the Sxx.

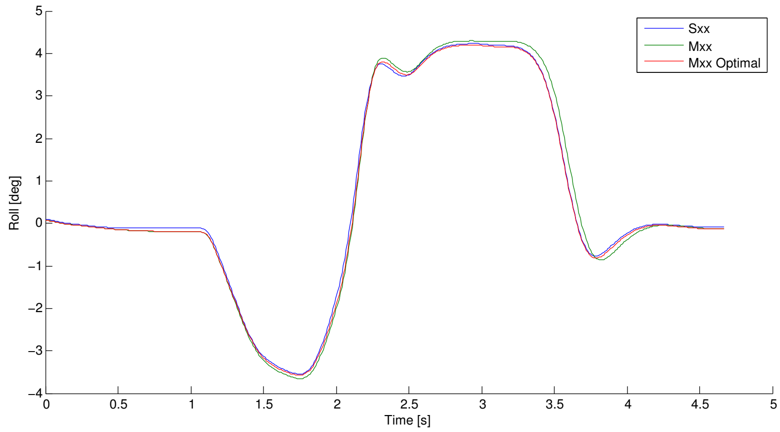

Figure 6.15: Sine With Dwell, Two Mxx vs Sxx, Simulations

6.3 Evaluation of Combined Results

The final evaluation is comparing the three vehicles, Sxx, Mxx and Vxx, with each other. It is hard to say anything about the result if the physical tests are compared with the simulation results of each of the three vehicles. The best way to analyse the results is to compare the difference of the three vehicles both in simulation and in reality.

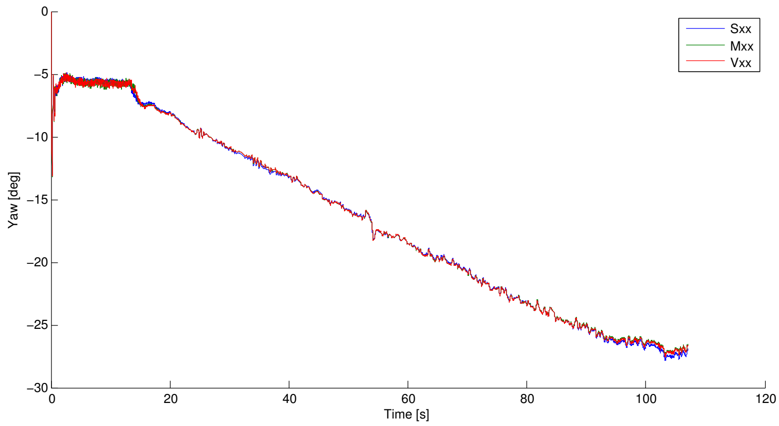

Figure 6.16: Constant Radius, Sxx, Mxx and Vxx, Simulation

Figure 6.16 compares the three vehicles yaw in a simulated constant radius test. As it can be seen the Sxx and Mxx are very much a like while the Vxx is slightly different.

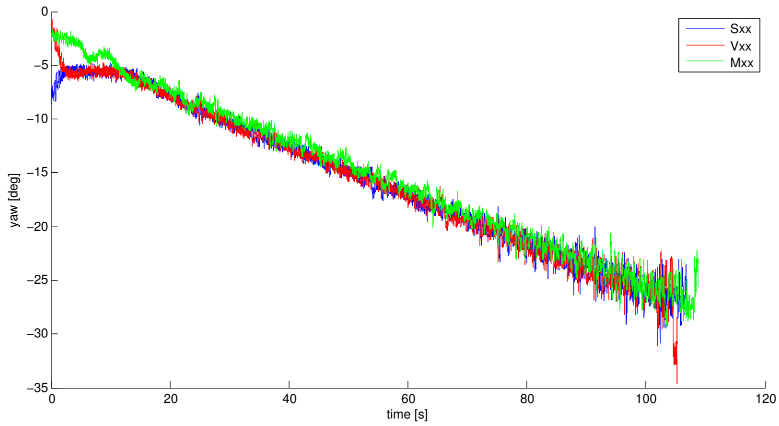

Figure 6.17: Constant Radius, Sxx, Mxx and Vxx, Test

In Figure 6.17 the same tendency can be seen, even though the difference between the vehicles is larger. The Sxx has more yaw than the other two vehicles which is reasonable because of the lower mass and CoG which reduces the inertia and load transfer. Figure 6.18 shows the simulated yaw of the three vehicles.

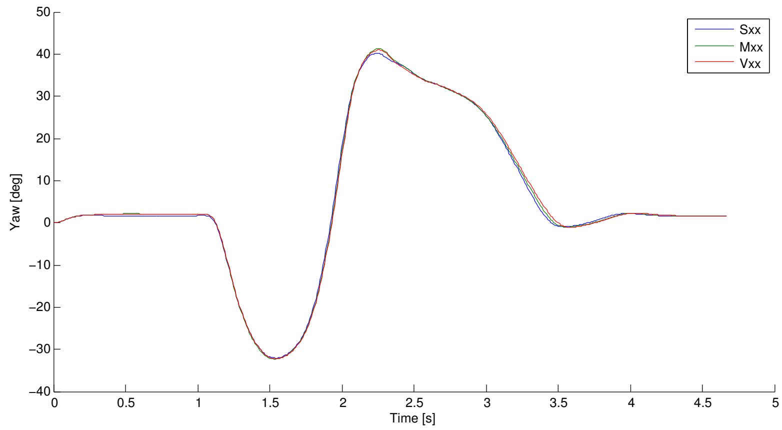

Figure 6.18: Sine With Dwell, Sxx, Mxx and Vxx, Simulation

In the simulated result in Figure 6.18 the Mxx is slightly oversteered compared to the Sxx and the Vxx shows even more oversteered behaviour. But in Figure 6.19 it can be seen that the Vxx is a bit oversteered and the Mxx is understeered compared to the Sxx when looking at the second peak. This is because the front springs of the Mxx are stiffer and on the Vxx they are softer. Stiffer springs in the front will increase the load transfer on the front axle and make the vehicle more understeered and vice versa with softer springs.

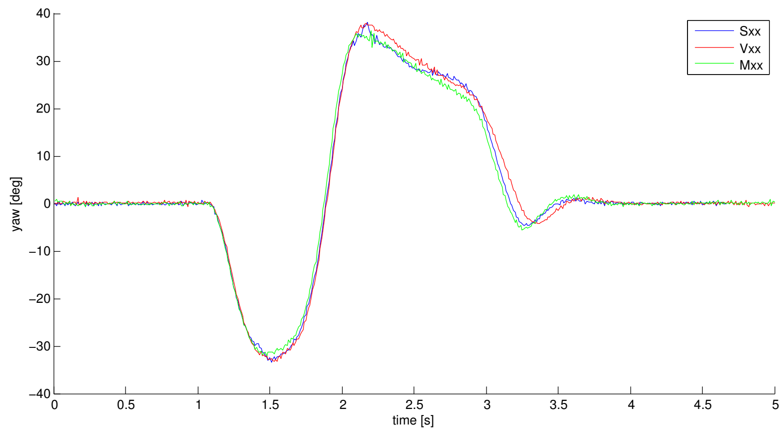

Figure 6.19: Sine With Dwell, Sxx, Mxx and Vxx, Test

The roll behaviour of a simulated sine with dwell test is shown in Figure 6.20. As it can be seen the Mxx is rolling a bit more than the Sxx and the Vxx is rolling even more. The reason why the Mxx is rolling more than the Sxx is because of the compromises that needed to be done, described in Section 6.2. The Vxx has softer springs in the front than the Mxx which makes it roll more.

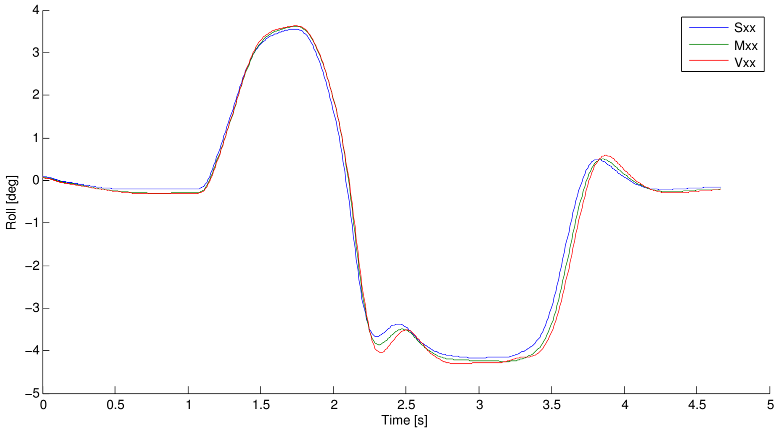

Figure 6.20: Sine With Dwell, Sxx, Mxx and Vxx, Simulation

When looking at Figure 6.21 it is clear that the roll of the Vxx is much smaller than for the other two vehicles. This seems strange since the mass and CoG of the Vxx is the same or somewhat higher than for the Mxx. This together with the softer front springs should lead to more roll on the Vxx.

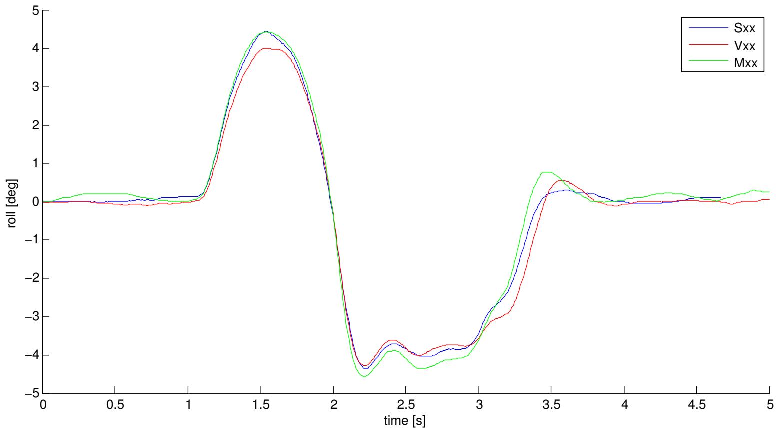

Figure 6.21: Sine With Dwell, Sxx, Mxx and Vxx, Test

It is not only in the sine with dwell manoeuvre this phenomena is present. In Figure 6.22 the Sxx still rolls a lot more than the Vxx. But this time the Mxx also rolls less than the Sxx, about as much as the Vxx.

In order to verify that the Mxx simulation model does not differ from the Vxx simulation model a simulation of the Vxx with the suspension of the Mxx was made. Figure 6.23 and 6.24 shows that the two vehicles behaves almost identical, the small differences are because of the slightly weaker body stiffness of the Vxx.

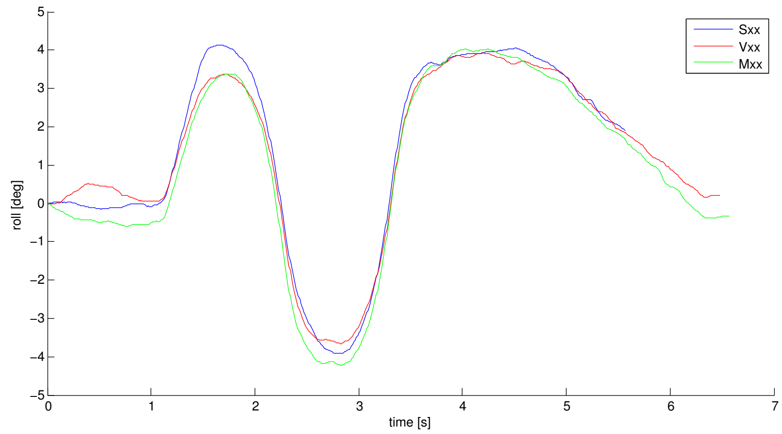

Figure 6.22: Dynamic Catch Up, Sxx, Mxx and Vxx, Test

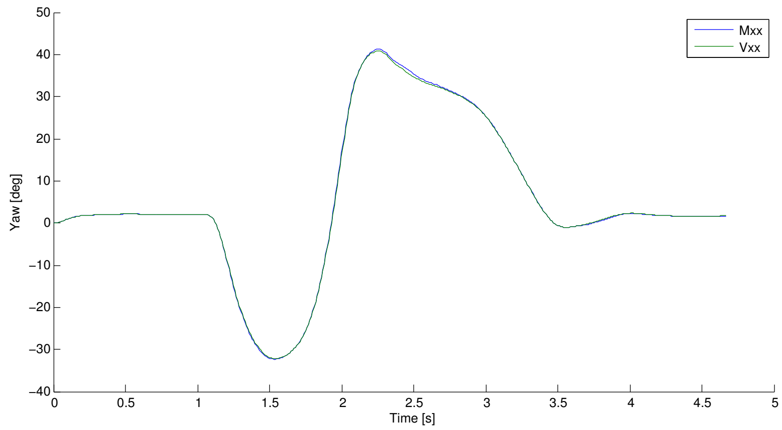

Figure 6.23: Sine With Dwell, Mxx and Vxx with Mxx Suspension, Simulation

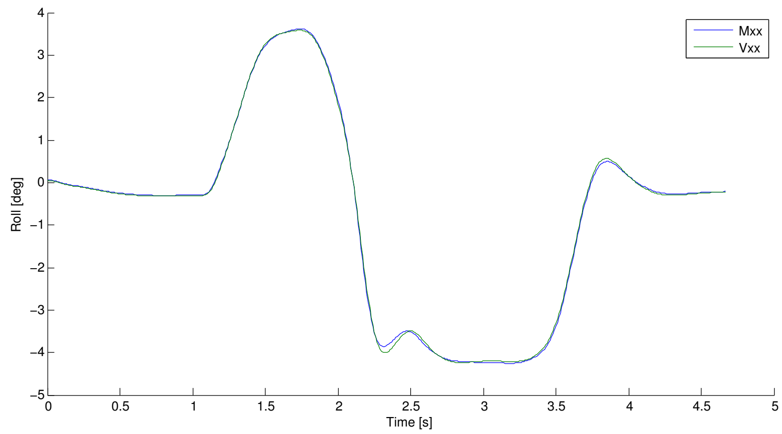

Figure 6.24: Sine With Dwell, Mxx and Vxx with Mxx Suspension, Simulation

7

Discussion and Conclusions

The results has been fairly satisfying and the methodology used in this thesis seems like an promising tool for predicting the behaviour for a wagon if a sedan based on the same platform exist. The fact that the Sxx and Vxx have the same suspension layout, track width and wheelbase makes them especially suitable for this method. Differences within these areas would make the work more complex. Although Volvo Cars have been very generous and supportive this project has been run as a thesis work which has led to some limitations. To evaluate the method and to benefit from its full potential there are room for improvements along the way. Example of such areas and improvements will be discussed below.

Suspension Geometry and Simulations

The initial idea of the project was to build upon existing VI-Car models of the Sxx and the Vxx. It turned out that Volvo cars worked with different software when the Sxx and Vxx was developed, as a result of this more of the thesis than planned had to be focused on building VI-Car models. The original plan was to convert an Adams Car model to VI-Car, unfortunately no such complete model was available. A possible improvement for the future could therefore be to use a complete Adams model and convert it to VI-Car as a base when building the VI-Car model.

The majority of the suspension and steering system properties are given by the results from K&C measurements. The optimal scenario would be that the same vehicle that is used for K&C measurements will be used for the tests at the proving ground, this would give much more accurate K&C-data. Other possible improvements regarding the K&C measurements would be to measure more for VI-Car requested data such as camber/caster depending on both steering angle and jounce. Accurate and detailed measurements of data not given from the K&C measurements such as damper and bump stop data would also lead to a better model. In general a good and accurate CAD model and weight specification for all parts as for example unsprung mass would result in a better VI-Car model.

When building the Mxx model the CoG and inertia data were given by K&C measurements of a Vxx These values should be given by the CAD-model of the Vxx since a physical Vxx should not exist at this point of the process. As a limitation in this thesis the K&C data were used since it was too time consuming and hard to get hold of the latest accurate CAD-model of the Vxx and get the CoG and inertia data from it.

Modifications and Testing

A limitation within this thesis has been that there was not either budget or time to manufacture springs and anti-roll bars according to the desired specification for the Mxx. To evaluate if it is possible to tune the chassis for a wagon with only simulations and physical testing with a sedan it was crucial to test the optimized springs and anti-roll bars on the physical Mxx. Therefore the optimization process of the Mxx in VI-Car was a compromise by making the Mxx as good as possible using available anti-roll bars and springs. The possibility to order anti-roll bars and springs according to the desired specification would most likely make the Mxx perform closer to the Vxx.

Another important factor for the outcome of this method is the tyre files. How well these tyre files represent a real tyre is critical for how well the simulation model will correspond to a real vehicle. Choosing a tyre file is a bit of a compromise since a physical tyre suitable for the tests must be paired with a thoroughly worked out tyre file. The tyre file chosen for this study works well for lateral cases and include relaxation but does not include longitudinal tyre behaviour. A complete tyre model which also works with longitudinal cases would be a better choice since it would make it possible to include tests like braking/lift of/acceleration while cornering, such manoeuvres would make the optimization and evaluation work even better.

Pitch is an important parameter, which does not match between simulation and physical tests. The pitch angle does not seem to be correct in the physical tests. One possible reason for the error may be that the gyro is not mounted perfectly straight in line with the vehicles x-axis, instead the gyro may have been slightly rotated around the z-axis. Such an error would cause the roll angle to be affected by the pitch angle and vice versa. Since the roll angle in absolute numbers change more than the pitch angel the roll angle will have a greater impact on the pitch angle than vice versa if the gyro is not perfectly straight in line with the vehicles x-axis and if the gyro does not compensate for such an error. To improve the test results in future the issue should be further investigated, preferably by varying the rotation around the z-axis of the gyro between different tests.

Although the simulations resulted in specifications for the anti-roll bars and springs that are in line with those used for the wagon in production it was difficult to draw general conclusions from the physical tests of the Sxx, Vxx and Mxx. To begin with all three vehicles are very similar which makes it difficult to discern differences. In the simulations the differences between the vehicles where consistent for the various tests which enabled conclusions to be drawn despite the vehicles being very similar. In the case of the physical tests it is not as easy. The desired result would be that Mxx consistency was closer to Vxx in behaviour than the Sxx or at least consistent in its behaviour compared to the Sxx and Vxx. Unfortunately this is not the case, in the Dynamic Catch Up tests the Mxx behaves more like the Vxx than the Sxx in terms of roll and yaw. In the Sine With Dwell tests the situation is reverse and in terms of roll and yaw the Mxx is closer to the Sxx. The Constant Radius tests indicate that the Sxx and Vxx are more alike each other than the Mxx is like them. The only general conclusion that can be drawn from the tests are the Mxx rolls more than both the Vxx and the Sxx. The result seems strange since the mass and CoG of the Vxx is the same or somewhat higher than for the Mxx and with softer springs on the Vxx compared to the Mxx the Vxx should roll more. It may be several reasons for these results, as previously mentioned the position of the gyro could affect the roll value in all tests. To be able to say more about the tests and draw better conclusions it necessary to perform more tests. It would have been very interesting to test the Mxx without ballast and test to the Vxx with the same front springs as Mxx as well as the Sxx with the ballast.

It should be mentioned that the thesis only focused on objective testing, and that the subjective testing is at least as important. To achieve the best possible results subjective testing should be included and play a big role in the development work. With more testing and implementation of the improvements mentioned in this chapter, the method described in this thesis could be to be reckoned with in future development and tuning work in vehicle dynamics.

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A

Appendix

A.1 Body Styles

A.1.1 Sedan

A sedan or saloon car is one of the most common passenger car configurations. A sedan is based on a three-box design with A, B and C-pillars where principal volumes are divided into three different boxes: engine, passengers and cargo. The passenger compartment consists of two rows of seats big enough for adult passengers. The cargo compartment is for front engine cars located in the rear and accessed through a horizontal or nearly horizontal trunk lid. [7]

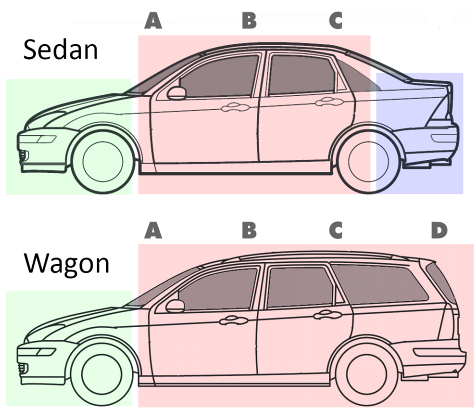

Figure A.1: Description of different body styles. (Source: `http://en.wikipedia.org/wiki/Station_wagon`)

A.1.2 Wagon

Wagon, also known as station wagon, estate car or estate is a vehicle type with a body style variant of a sedan/saloon. The wagon has its roof extended to the rear which creates a larger passenger/cargo volume that can be accessed through a fifth door at the rear of the vehicle. A wagon uses a two-box design with A, B, C and D-pillar. In order to increase the cargo volume it is often possible to fold-down the rear seats. Wagons are popular among families with children due to its big cargo volume. [7]

A.2 Automotive Platform

An automotive platform include per definition parts such as underbody and suspensions (with axles). Underbody is the collective name for front floor, underfloor, engine compartment and frame. Mechanical components that define a platform are:

- The floorpan
- Front and rear axles and the distance between them (wheelbase)
- Steering mechanism
- Type of engine, engine placement and other powertrain components

Platform sharing in the automotive industry refers to creating different models from similar mechanical key components. The main advantages with platform sharing is that fewer components needs to be designed and built, development costs are reduced and the economic resources can be focused on other areas such as development and tuning. [8]

A.3 K&C

K&C stands for Kinematics and Compliance. Kinematics is the controlled orientation of the wheels by suspension links, toe, caster and camber change etc. Compliance on the other hand is the controlled movement of the wheels by springs, bushings and part deflections, in other words how far suspension components bends when loads from the road travel through the tyres. Vehicles are often tested in K&C test rigs, in these test rigs the vehicle is undergoing static tests such as vertical bounce, roll and pitch where forces, movements and angles such as toe and camber are measured for all four wheels.