The damper levels influence on vehicle roll, pitch, bounce and cornering behaviour of passenger vehicles

A study in cooperation with Volvo Car Corporation

Master's Thesis in Automotive Engineering

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CHALMERS UNIVERSITY OF TECHNOLOGY
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Volvo S60R with Continuously Controlled Damping (CCD)

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ABSTRACT

The competition within the automotive industry is increasing. It is therefore important to reduce the resources and time spent on developing new vehicles. A significant part of the development is the chassis tuning which consumes a lot of time and resources. Dampers have a key role in chassis tuning and a better understanding of how the level of damping affects the vehicle behaviour could reduce both the number of prototypes built and the time spent on chassis tuning. This thesis investigates the possibility to develop a tool which captures the vehicle behaviour accurately enough to be used for choosing damper levels in the concept phase and as an aid in the tuning of dampers. A Simulink model was developed together with a graphical user interface allowing the user to easily change the level of damping and run simulations. In order to validate the model and see how changes to the level of damping affect the vehicle behaviour, both subjective and objective tests were performed for different damper configurations. It was found that the changes in vehicle behaviour for the various damper levels had a good correlation between the simulation model and the results from the physical tests. However in absolute numbers differences were found between simulations and objective tests. In order to verify the simulation model a limited number of objective tests are required. The tested vehicle should also be measured properly in order to receive better input data to the simulation model.

The study found that the pitch balance in the vehicle is mostly affected by the difference in compression and rebound damping between the front and rear dampers. Lower damping at low vertical wheel velocities has a negative impact on roll control. This can to a limited extent be compensated with higher roll stiffness. A good balance between the front and rear damping is also required in order to have good connection to the road and make the vehicle feel more comfortable.

The subjective and objective evaluation showed that it might be the time lag between lateral acceleration and steering wheel angle is a good indication of the vehicles turn-in capabilities and that the roll angle time lag determines the response. However more objective and subjective tests are required in order to confirm this connection. A better understanding of how to define good vehicle behaviour objectively is also required if the tool should be used in the concept phase.

Keywords: Vehicle, Dampers, Roll, Pitch, Bounce, Damper tuning, Influence

Stötdämparnivåernas påverkan på bilens krängning, hävning, pitchbeteende och kurvtagningsförmåga
Ett projekt i samarbete med Volvo Cars
Examensarbete inom Automotive Engineering
ANTON ALBINSSON
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Institutionen för tillämpad mekanik
Avdelningen för Fordonsteknik och Autonoma System
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SAMMANFATTNING

Med den ökade konkurrensen inom fordonsindustrin är det viktigt att minska kostnader och ledtider för utveckling av nya modeller. En stor del av utvecklingen av nya modeller är att hitta rätt dämpar- och chassi-inställningar. En bättre förståelse av hur dämparnivåerna påverkar bilen beteende kan reducera både antalet prototypbilar som behövs och tiden som spenderas på att hitta rätt dämparnivåer. Det här examensarbetet har undersökt möjligheten att utveckla ett verktyg som fångar bilens beteende tillräckligt bra för att användas för val av dämparnivåer i konceptfasen och som ett hjälpmedel vid dämparjusteringar i bil. Därför utvecklades en Simulink modell samt ett grafiskt användargränssnitt som gör det möjligt för användaren att på ett enkelt sätt ändra dämparnivåerna i bilen. För att validera modellen och för att få en bättre uppfattning om hur dämparnivåerna påverkar bilens beteende har både subjektiva och objektiva prov på provbana utförts för olika dämparinställningar. Valideringen visade att modellen kunde fånga bilen generella beteende och hur ändringar i dämparnivåer påverkade bilens beteende men för absoluta värden hittades skillnader mellan simuleringar och de fysiska testerna. Mer objektiv provning krävs därför för att fastställa modellens tillförlitlighet. Fordonet som används för de objektiva testerna bör också mätas upp ordentlig för att ge bättre data till simuleringsmodellen.

Studien fann att pitchbalansen påverkas mest av skillnaden i kvoten mellan kompressions- och expansionsdämpning fram och bak. Lägre dämpning vid låga vertikala hjulhastigheter ger sämre krängningskontroll. Detta kan i viss mån kompenseras med högre krängningsstyvhet. En bra balans mellan dämparnivåerna fram och bak behövs även för att föraren ska uppfatta bilen som bekväm och för att föraren ska få en bra känsla av vad som händer på vägen.

De subjektiva och objektiva testerna visade att tidsförskjutningen för lateralaccelerationen och krängningsvinkeln kan vara avgörande för bilens instyrningsförmåga respektive respons. Mer objektiva och subjektiva tester behövs däremot för att bekräfta denna koppling. En bättre förståelse i hur man definierar det önskade fordonsbeteendet objektivt krävs för att verktyget ska kunna användas fullt ut i konceptfasen.

Nyckelord: Fordon, Dämpare, Krängning, Pitch, Hävning

Contents

Preface

This thesis work has been conducted as a partial requirement for the Master of Science degree at the Masters programme Automotive Engineering at Chalmers University of Technology, Gothenburg, Sweden, in cooperation with Volvo Car Corporation. The thesis has been performed at the Vehicle Dynamics & Calibration division, Volvo Cars Research & Development, Gothenburg, Sweden as well as Hällered Proving Ground, Borås, Sweden, during January-June 2013.

We would like to acknowledge and thank our supervisors at Volvo Cars, Kenneth Ekström and Egbert Bakker, for their support and guidance and our examiner at Chalmers Mathias Lidberg for sharing his knowledge and expertise. The head of the Vehicle Dynamics & Calibration division, Volvo Cars, Erik Axelsson for his support with resources and all administrative work. Special thanks to Mathias Davidsson, Tobias Ersson, Carl Sandberg, Johan Thunmarker and Per Carlsson as well as all Co-workers at the Vehicle Dynamics division for your inputs, co-operation and unconditional help throughout this project, it has been greatly appreciated.

We would also like to thank our colleagues from Chalmers, Erik Wendeberg, Lucas Börjesson and Peter Wiborg for discussions along the way.

Gothenburg, June 2013

Anton Albinsson and Christoffer Routledge

Abbreviations

CCD	Continuously controlled damping
CoG	Centre of gravity
K&C	Kinematics and Compliances

Roman upper case letters

B	Stiffness factor
C	Shape factor
D	Peak value
E	Curvature factor
F_D	Damper force
F_Z	Vertical wheel force
F_y	Lateral wheel force
F_x	Longitudinal wheel force
F_{spring}	Spring force
F_{stop}	Bump/Rebound stop force
F_{damper}	Damper force
I_{xx}	Moment of inertia of sprung mass around x-axis
I_{yy}	Moment of inertia of sprung mass around y-axis
I_{zz}	Moment of inertia of vehicle around z-axis
Krf	Anti-roll bar stiffness front
Krr	Anti-roll bar stiffness front
L	Wheelbase
M_{roll}	Roll moment
M_{pitch}	Pitching moment
M_Z	Yaw moment
S_H	Horizontal shift
S_V	Vertical shift

Roman lower case letters

a	Longitudinal distance from CoG to axles
a_x	Longitudinal acceleration at CoG
a_y	Lateral acceleration at CoG
a_z	Vertical acceleration at CoG
b	Lateral distance from CoG to wheel centre
c	Damping coefficient
g	Gravitational constant
h	CoG height
h_0	Height of CoG above roll axis
h_{rf}	Front roll centre height
h_{rr}	Rear roll centre height
k	Spring stiffness
k_f	Effective spring stiffness at wheel centre front
k_r	Effective spring stiffness at wheel centre rear
k_{RF}	Front Anti-roll bar stiffness
k_{RR}	Rear Anti-roll bar stiffness

k_{tf}	Wheel stiffness front
k_{tr}	Wheel stiffness rear
l_f	Distance from CoG to front axle
l_r	Distance from CoG to rear axle
m	Vehicle mass
m_1	Unsprung mass for one side front
m_2	Unsprung mass for one side rear
m_s	Vehicle sprung mass
m_u	Vehicle unsprung mass
r	Yaw rate
s	Track width
v_x	Longitudinal velocity
v_y	Lateral velocity
ω	Rotational velocity
ω_D	Damped natural frequency
ω_N	Natural frequency
ω_r	Ratio between frequency and natural frequency
z	Vertical displacement
z_0	Amplitude

Greek lower case letters

α	Slip angles
δ	Steer angle at wheels
φ	Body roll angle
θ	Pitch angle
ψ	Yaw angle
ϕ	Phase angle
ξ	Damping ratio

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1 Introduction

This master thesis project is carried out in cooperation with Volvo Cars Corporation. The report describes the development of a simulation tool for use in the evaluation of dampers influence on the vehicle behaviour. The tool is also intended for future damper tuning session and to become an aid to determine initial damper characteristics before any prototype is built.

1.1 Background

With the increasing competition between car manufactures, the established manufactures need to improve their customer value of the products in order to compete with the low budget brands. Two possible aspects of quality perception which the costumer could expect from a premium brand are good ride comfort and handling performance. Damper settings have a great influence on the vehicles behaviour and can improve or ruin the vehicles handling and ride. This makes it worth spending time tuning dampers to obtain the desired vehicle performance.

Tools based on the level of critical damping are today often used as an aid during damper tuning sessions. Critical damping curves show the percentage of critical damping as a function of vertical wheel velocity and is determined through vehicle models. The target shapes of these curves are based on experience rather than theory and on bounce motions of the vehicle rather than the combination of roll, pitch and bounce motions which occur in reality. Benefits of these tools, during the tuning work, are at the moment limited and the tools are therefore not used that often.

The evaluation of damper settings is done subjectively without any objective targets for the vehicle behaviour taken into account during the tuning sessions. The subjective tuning work is very time consuming and with the current trend of reducing development time for new cars, the time spent on vehicle tuning need to be minimized. By reducing the time spent on tuning work, it is possible to launch new models more often while reducing development cost. The biggest challenge for reaching this goal is the lack of knowledge of how the level of damping affects the vehicle behaviour objectively. Tuning engineers have through years of experience developed a sense for what changes that should be made to the dampers in order to get the desired vehicle behaviour. There is however a lack of knowledge about what makes a vehicle good or bad in terms of ride and handling and how the dampers influence that behaviour. A better physical understanding of how the level of damping affect the vehicle behaviour is therefore one part of further optimizing the tuning process and the capabilities of initial damper characterization in Computer Aided Engineering (CAE).

1.2 Purpose

This thesis work aims to develop, test and implement a computer aid, based on the level of critical damping, to be used for tuning of dampers during tuning sessions and for selecting damper parameters in the concept phase. The computer aid is based on a simplified model which aims to capture the vehicles roll, pitch and bounce behaviour. The tool should be able to describe the change in vehicle behaviour due to a change of dampers or damper settings.

1.2.1 Research questions

The thesis should answer the following questions.

- Can the vehicle behaviour be captured in a simplified model?
- How does the level of critical damping affect the vehicle behaviour during roll pitch and bounce?
- Is it possible to see any trends when comparing subjective measurements, results from the simulation model and objective test results?
- Can the tool be used in future tuning sessions and does it simplify the tuning processes?

1.2.2 Limitations

No theory about subjective comfort will be presented, since the focus of the project is handling and vehicle motion. However, comfort is still an important factor that needs to be considered during the whole development.

The study will focus on trends of how the vehicle behaviour changes with damper settings rather than absolute values. This is due to lack of data found connecting objective data to subjective measurements. Even if the model is capable of capturing the real vehicle behaviour fully, there is currently not enough knowledge of how a good car should behave objectively to rely fully on computer aids.

No detailed modelling of the dynamic behaviour inside the dampers is made. All dampers are assumed to have the same dynamic behaviour but with different levels of damping. In reality two dampers with the same force vs. velocity displacement relation can behave differently in dynamic events. This is considered a damper design issue and is not taken into account.

No model of the steering system is implemented in the complete vehicle model since steering characteristics are mainly influenced by the parameters connected to the steering system and not the damper settings. Steering will therefore be carried out directly on the wheels. The vehicle body is considered to be rigid without any flexibility at all.

2 Damped mechanical systems

The vehicles suspension system consists of, among other components, springs and dampers. These together with various masses make the vehicle a complex vibrating system which is stimulated by road inputs and accelerations. The spring and damper characteristics affects the ride, handling and comfort of the vehicle. Their characteristics are tuned precisely to find the best compromise between the requirements. Generally, spring stiffness is chosen based on the vehicle mass which means that the dampers play a very important role for vehicle behaviour.

Suspensions are generally described and analysed as a 2-degree of freedom (2-dof) system. However, in order to understand the vibration modes found in the 2-dof system it is sometimes easier to analyse the 2-dof as 1-dof system (Dixon J. C., 2007). This part will therefore review free vibration of undamped and damped systems in 1-dof and damped systems in 2-dof.

2.1 Free undamped vibration 1-dof

An easy way of describing and analysing system vibrations is by considering the one degree of freedom system where only one coordinate is necessary to describe the systems motion (Dixon J. C., 2007). Figure 1 shows a picture of a 1-dof system at its equilibrium position. A mass (m) only allowed to move in z direction (vertical) is connected to the ground through a spring with linear stiffness$k$. As the mass is moved a distance z from its equilibrium position a spring force arises, given by.

$$F_z = -kz \tag{2.1.1}$$

The negative sign indicates that the force acts opposite to the direction of motion. The force gives the body an acceleration which is expressed by Newton's second law as

$$m\ddot{z} = F_z \tag{2.1.2}$$

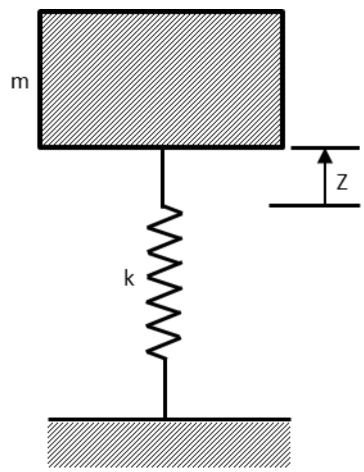

Figure 1, One degree of freedom system, undamped

By substituting for F_z, the equation of motion becomes,

$$m\ddot{z} + kz = 0 \tag{2.1.3}$$

and by dividing with the mass *m* it is expressed as.

$$\ddot{z} + \frac{k}{m} z = 0 \tag{2.1.4}$$

If the spring stiffness is assumed to be constant this equation becomes linear and is determined by

$$\ddot{z} + \omega_N^2 z = 0 \tag{2.1.5}$$

where ω_N is the natural frequency, thus

$$\omega_N^2 = \frac{k}{m} \rightarrow \omega_N = \sqrt{\frac{k}{m}} \tag{2.1.6}$$

The natural frequency for a system without any damping is the frequency at which the system will oscillate in free vibrations. It is only dependent on the mass and the spring stiffness of the system as seen in Equation (2.1.6). If damping is added to the system the natural frequency will change (Dixon J. C., 2007).

If oscillatory motions are considered, the equation of motion for the mass becomes

$$Z = Z_0 \sin(\omega_N t + \phi) \tag{2.1.7}$$

where Z_0 is the amplitude, t is the time and ϕ the phase angle. When the mass is displaced and released, it oscillates freely at its natural frequency, sinusoidal and at the constant amplitude Z_0, see Figure 2.

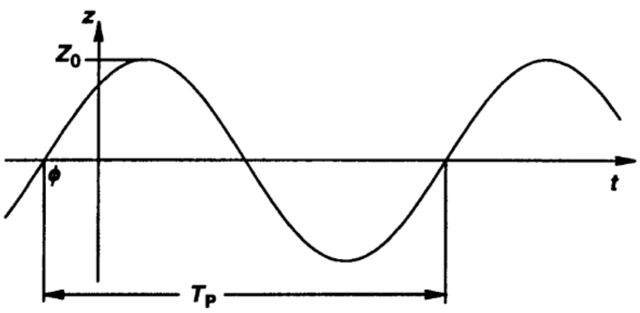

Figure 2, Free vibration motion (Pacejka, 2006)

2.2 Free damped vibration 1-dof

Figure 3 illustrates a damped system where a linear damper has been included adding a damping force determined by

$$F_D = -c\dot{z} \tag{2.2.1}$$

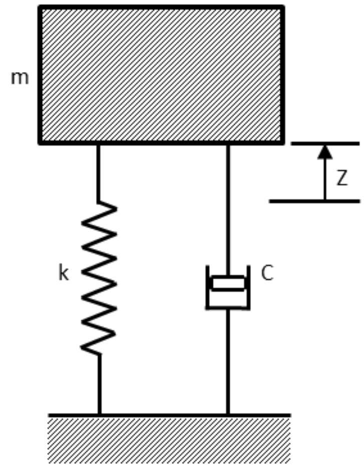

Figure 3, One degree of freedom system, damped

The damping force acting on the mass is also negative since it is directed against the direction of motion. The damping coefficient is c often expressed in the unit Ns/m. If the mass is moved a distance z with velocity $\dot{z}$ the total force acting on the body is

$$F_z = -c\dot{z} - kz \tag{2.2.2}$$

which produces an acceleration expressed, as earlier mentioned, by Newton's second law.

$$m\ddot{z} = F_z = -c\dot{z} - kz \tag{2.2.3}$$

The equation of motion, in standard form, therefore becomes.

$$\ddot{z} + \frac{c}{m}\dot{z} + \frac{k}{m}z = 0 \tag{2.2.4}$$

The system can be described by the standard equation for a damped system with 1-dof based on two parameters, the natural frequency and damping ratio.

$$\ddot{z} + 2\xi\omega_N\dot{z} + \omega_N^2 z = 0 \tag{2.2.5}$$

By comparing variables from equation (2.2.5) with equation (2.2.4) it is found that

$$\omega_N = \sqrt{\frac{k}{m}} \tag{2.2.6}$$

$$\xi = -\frac{\alpha}{\omega_N} = \frac{c}{2m\omega_N} = \frac{c}{2\sqrt{mk}} \tag{2.2.7}$$

where $\alpha = -\xi\omega_N = \frac{c}{2m}$

The damped natural frequency ω_D can be determined by (2.2.8) and decreases with increased damping.

$$\omega_D = \sqrt{\omega_N^2 - \alpha^2} = \omega_N\sqrt{1 - \xi^2} \tag{2.2.8}$$

The actual displacement at time t may therefore, as oscillatory motions are considered, be expressed according to Dixon as (Dixon J. C., 2007),

$$Z = Z_0 e^{\alpha t} \sin{(\omega_D t + \phi)} \tag{2.2.9}$$

2.3 Free damped vibration 2-dof

Most commonly vibration dampers are described with help of a 2-dof system where two masses, m_1 and m_2 are suspended and coupled by springs and dampers, see Figure 4. Since the masses are guided they only move in z direction. Because the masses can move independently of each other there are two degrees of freedom.(Den Hartog, 1985)

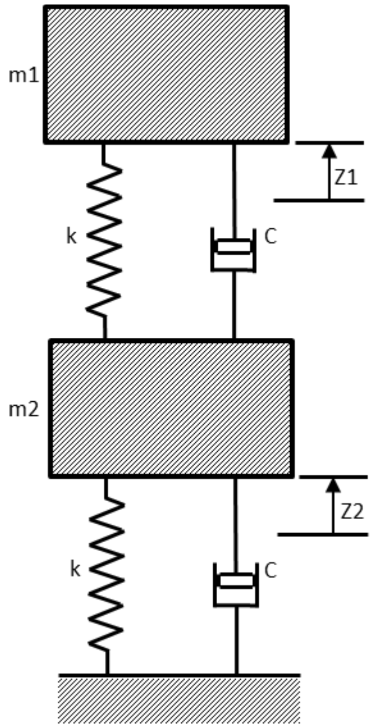

Figure 4, Two degrees of freedom system, damped

The force, in a 2-dof system, acting on the body is determined by

$$F_z = -c(\dot{z}_1 - \dot{z}_2) - k(z_1 - z_2) \tag{2.3.1}$$

which produces an acceleration of

$$m\ddot{z} = F_z = -c(\dot{z}_1 - \dot{z}_2) - k(z_1 - z_2) \tag{2.3.2}$$

Written in standard form the expression becomes

$$\ddot{z} + \frac{c}{m}(\dot{z}_1 - \dot{z}_2) + \frac{k}{m}(z_1 - z_2) = 0 \tag{2.3.3}$$

2.4 Critical damping

Critical damping is defined for free vibrating systems. The critical damping is the level of damping that will allow the system to return to its equilibrium position fastest without any overshoot (see Figure 5). If the system is over damped i.e. have a damping ratio over 1 ($\xi > 1$) it will return to its equilibrium position slower. An under damped system ($\xi < 1$) will have some oscillations before returning to its equilibrium position. If there is no damping at all ($\xi = 0$) the system will simply oscillate around the equilibrium position. This is not achievable in practice since there will always be some energy losses in the system. The damping ratio is often studied in order to understand how a dynamic system responds to disturbances. In automotive applications it is a useful measurement when studying dampers. (Dixon J. C., 2007)

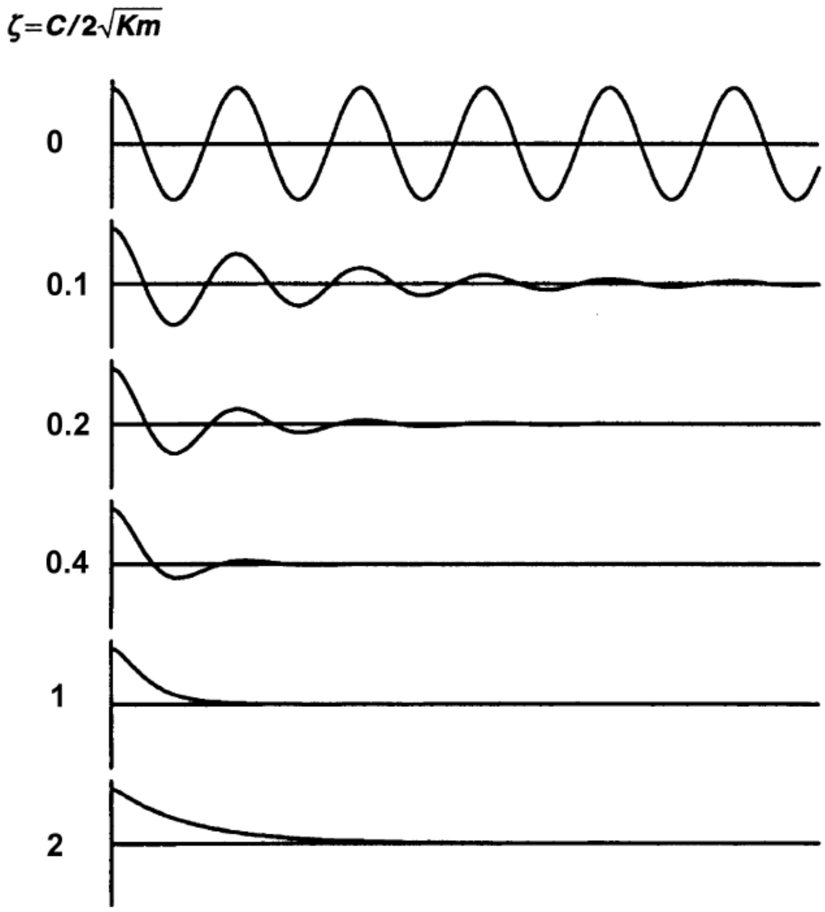

Figure 5, Time response for one degree of freedom free vibrating system with different damping ratios (Dixon J. C., 2007)

In order to see how a damped system responds to a forced vibration it is useful to look at the frequency response of the system. The response for a 1-dof system can be seen in Figure 6. The frequency ratio is the actual frequency normalized to the undamped natural frequency.

$$\omega_r = \frac{\omega}{\omega_N} \tag{2.4.1}$$

The y-axis shows the response gain from the input to the mass for different damping ratios. This gain is often called transmittability.

As seen in the Figure 6 the peak value of the transmittability decreases for higher damping ratios and is moved to lower frequencies. At higher frequencies the transmittability is smaller for lower damping ratios but below a damping ratio of 0.2 the effect is small. Even though the gain decreases for lower damping ratios at higher frequencies the system will oscillate longer in free vibration.

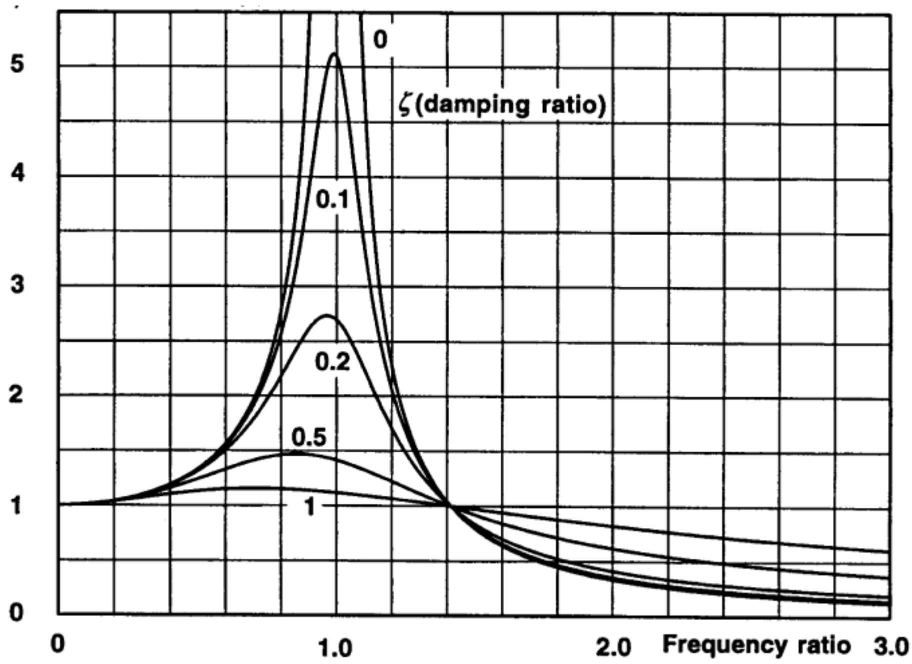

Figure 6, Frequency response for one degree of freedom system with forced vibration input for different damping ratios (Dixon J. C., 2007)

3 Vehicle models

Suspension dampers main tasks is according to Causemann to damp post vibration of the vehicle body caused by uneven roads or driving conditions as well as settle the road-induced wheel and axle vibrations while keeping constant tyre to road contact which ensures good tracking and braking performance (Causemann, 2001).

As a vehicle drives over a bump the vehicle springs and dampers are compressed causing energy to be stored in the springs. The energy stored in the springs is then released as they strive to reach their equilibrium. If the vehicle would not be fitted with dampers, the motion caused by a bump would make the vehicle oscillate until the energy in the dampers is lost through friction. The dampers prevent these oscillations by turning the kinetic energy of suspension movement into heat, which is dissipated through a hydraulic fluid. If too much damping is added to the system the comfort and safety is compromised. At high damping ratios certain excitations makes the vehicle so stiff that the vehicle bounces on the tires. This increases the natural body frequency to a level which is perceived as uncomfortable for humans (Causemann, 2001). For constant damping ratio a passenger vehicle generally have best ride at damping ratio around 0.2, and the best handling at 0.8 (Dixon J. C., 2007). Even if these targets are not exact and constant damping ratio has been assumed they highlight the need of a compromise between ride and handling.

Dampers do not only influence ride comfort of the vehicle. Since they affect the roll and pitch behaviour as well as the transient load transfer, dampers have a big influence of the planar motion of the vehicle, which makes it important to study how the dampers influence the handling and stability behaviour.

3.1 Two track model

The two-track model describes the planar motion of the vehicle. In its basic form, seen in Figure 7, the two track model does not include any roll or pitch dynamics. The model is however useful when describing the planar accelerations, velocities and displacements for the vehicle. It can also be combined with tyre-, load transfer-and/or other models to give a more accurate description of the vehicle behaviour. The equations of motion for the two track model can be derived from Newton's second law (Lidberg, 2012).

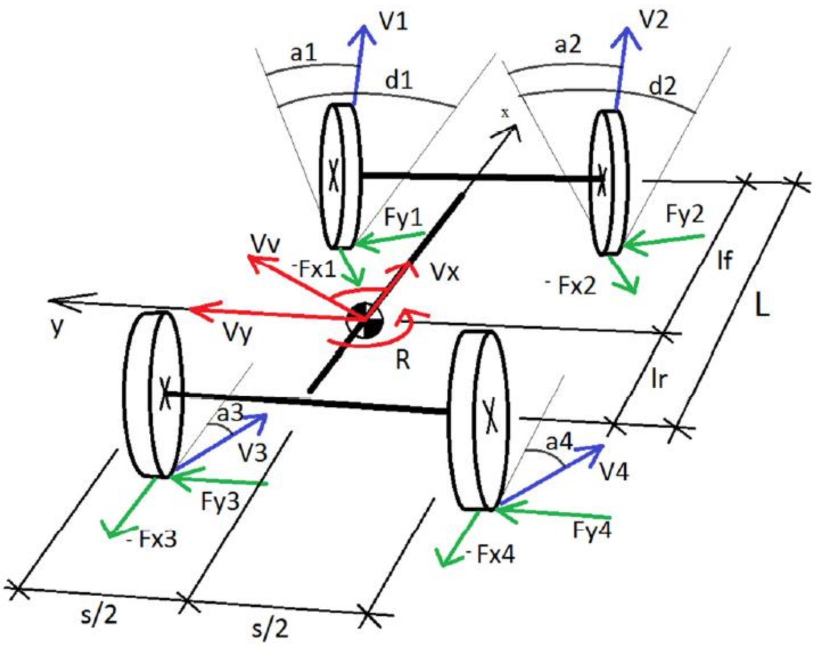

Figure 7, Two-track vehicle model (Albinsson, o.a., 2012)

$$ma_y = F_{y1}\cos(\delta_1) + F_{y2}\cos(\delta_2) + F_{y3} + F_{y4} + F_{x1}\sin(\delta_1) + F_{x2}\sin(\delta_2) \quad (3.1.1)$$

where $a_y = \dot{v_y} + v_x r$ (3.1.2)

$$I_{zz}\dot{r} = \frac{(F_{x4} - F_{x3})s_2}{2} + \frac{(F_{x2}cos(\delta_2) - F_{y2}\sin(\delta_2) - F_{x1}\cos(\delta_1) + F_{y1}\sin(\delta_1))s_1}{2} + \left(F_{x1}\sin(\delta_1) + F_{x2}\sin(\delta_2) + F_{y1}\cos(\delta_1) + F_{y2}\cos(\delta_2)\right) * l_f - \left(F_{y4} + F_{y3}\right) * l_r \quad (3.1.3)$$

$$ma_x = F_{x2}cos(\delta_2) - F_{y2}\sin(\delta_2) + F_{x1}\cos(\delta_1) - F_{y1}\sin(\delta_1) + F_{x3} + F_{x4} \quad (3.1.4)$$

3.2 Load transfer

A vehicle subjected to tyre forces will experience a change in vertical tyre forces. Due to the load transfer the vertical tyre forces will not remain constant during a transient manoeuvre where the acceleration is not constant. During steady-state events with constant lateral accelerations, the tyre vertical load will differ from its static values. The amount of load transfer is a function of the vertical, longitudinal and lateral accelerations and the vehicle setup.

The dampers do not influence the steady-state level of load transfer but they do have an impact on the load transfer during transient events and thereby also on instantaneous tyre loads. Fluctuations or changes in vertical tyre force will change the tyres lateral and longitudinal stiffness which in turn affects their ability to generate forces.

The load transfer distribution between the front and the rear axle affects the over and under steer behaviour of the vehicle. A bigger ratio of load transfer on the front axle tends to give the vehicle a more under steered behaviour at greater lateral accelerations and vice versa. The level of damping controls the initial load transfer of the vehicle and therefore also the initial over/under steer behaviour.

3.3 Full car vibrating model

The ride of a vehicle is according to Dixon the heaving, pitching and rolling motion in forced vibration caused by road roughness (Dixon J. C., 2007). If ride quality is studied the purpose of the suspension is to minimize the passenger discomfort. A good ride quality is mainly achieved by an appropriate choice of springs and dampers. The appropriate choice will depend on the road surface the vehicle will mainly operate on as well as individual preferences (Dixon J. C., 2007). The full car vibrating model described below is a tool that can be used for analysing the ride of a vehicle.

The full car vibration model has springs and dampers in all four corners of the vehicle. It describes the pitch, roll and vertical motion as a function of the spring and damper forces. Input to the system is the road displacement on all four wheels, y in Figure 8, and the sprung mass is considered to be a rigid body with infinite stiffness in all directions. Centre of gravity is assumed to be placed along the vehicles x-axis (x-axis according to the ISO-system). The basic model with notations can be seen in Figure 8. Tyres are modelled as springs connected to the road and tyre damping is neglected since it is normally small compared to the damping from the dampers (Jazar, 2008). Since most cars have anti-roll bars, which affects the roll behaviour and load transfer of the vehicle, the anti-roll bar stiffness is also included in the model. The model does not include any forces arising from acceleration and these have to be added separately to the equations of motion. The equations, with a change of notation between wheel 4 and 3 due to notation standards, of motion for the full car vibrating model becomes (Jazar, 2008):

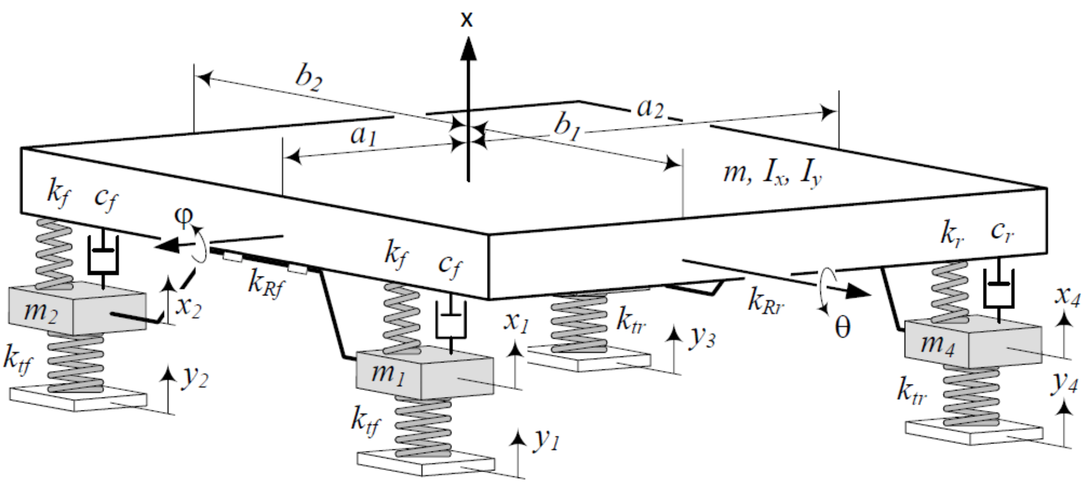

Figure 8, Full car vibration model (Jazar, 2008)

Equation which determines centre of gravity motion:

$$\begin{aligned} m\ddot{x} &+ c_f\left(\dot{x}-\dot{x_1}+b_1\dot{\varphi}-a_1\dot{\theta}\right)+c_f\left(\dot{x}-\dot{x_2}-b_2\dot{\varphi}-a_1\dot{\theta}\right) \\ &+ c_r\left(\dot{x}-\dot{x_3}+b_1\dot{\varphi}+a_2\dot{\theta}\right)+c_r\left(\dot{x}-\dot{x_4}-b_2\dot{\varphi}+a_2\dot{\theta}\right) \\ &+ k_f(x-x_1+b_1\varphi-a_1\theta)+k_f(x-x_2-b_2\varphi-a_1\theta) \\ &+ k_r(x-x_3+b_1\varphi+a_2\theta)+k_r(x-x_4-b_2\varphi+a_2\theta)=0 \end{aligned} \tag{3.3.1}$$

Equation for the roll motion:

$$\begin{aligned} &I_{xx}\ddot{\varphi}+b_1c_f\left(\dot{x}-\dot{x_1}+b_1\dot{\varphi}-a_1\dot{\theta}\right)-b_2c_f\left(\dot{x}-\dot{x_2}-b_2\dot{\varphi}-a_1\dot{\theta}\right) \\ &-b_1c_r\left(\dot{x}-\dot{x_3}+b_1\dot{\varphi}+a_2\dot{\theta}\right)+b_2c_r\left(\dot{x}-\dot{x_4}-b_2\dot{\varphi}+a_2\dot{\theta}\right) \\ &+b_1k_f(x-x_1+b_1\varphi-a_1\theta)-b_2k_f(x-x_2-b_2\varphi-a_1\theta) \\ &-b_1k_r(x-x_3+b_1\varphi+a_2\theta)+b_2k_r(x-x_4-b_2\varphi+a_2\theta) \\ &=0 \end{aligned} \tag{3.3.2}$$

Equation for the pitch motion:

$$\begin{aligned} &I_{yy}\ddot{\theta}-a_1c_f\left(\dot{x}-\dot{x_1}+b_1\dot{\varphi}-a_1\dot{\theta}\right)-a_1c_f\left(\dot{x}-\dot{x_2}-b_2\dot{\varphi}-a_1\dot{\theta}\right) \\ &+a_2c_r\left(\dot{x}-\dot{x_3}+b_1\dot{\varphi}+a_2\dot{\theta}\right)+a_2c_r\left(\dot{x}-\dot{x_4}-b_2\dot{\varphi}+a_2\dot{\theta}\right) \\ &-a_1k_f(x-x_1+b_1\varphi-a_1\theta)-a_1k_f(x-x_2-b_2\varphi-a_1\theta) \\ &+a_2k_r(x-x_3+b_1\varphi+a_2\theta)+a_2k_r(x-x_4-b_2\varphi+a_2\theta) \\ &=0 \end{aligned} \tag{3.3.3}$$

Equation for the wheel motions (1=Front left, 2=Front right, 3=Rear left, 4=Rear right):

$$\begin{aligned} m_f\ddot{x}_1 &- c_f\left(\dot{x}-\dot{x_1}+b_1\dot{\varphi}-a_1\dot{\theta}\right)-k_f(x-x_1+b_1\varphi-a_1\theta) \\ &- k_{RF}\frac{1}{b_1+b_2}\left(\varphi-\frac{x_1-x_2}{b1+b_2}\right)+k_{tf}(x_1-y_1) \end{aligned} \tag{3.3.4}$$

$$\begin{aligned} m_f\ddot{x}_2 &- c_f\left(\dot{x}-\dot{x_2}-b_2\dot{\varphi}-a_1\dot{\theta}\right)-k_f(x-x_2-b_2\varphi-a_1\theta) \\ &+ k_{RF}\frac{1}{b_1+b_2}\left(\varphi-\frac{x_1-x_2}{b1+b_2}\right)+k_{tf}(x_2-y_2) \end{aligned} \tag{3.3.5}$$

$$\begin{aligned} m_f\ddot{x}_3 &- c_f\left(\dot{x}-\dot{x_3}+b_1\dot{\varphi}+a_2\dot{\theta}\right)-k_f(x-x_3+b_1\varphi+a_2\theta) \\ &- k_{RR}\frac{1}{b_1+b_2}\left(\varphi-\frac{x_3-x_4}{b1+b_2}\right)+k_{tr}(x_3-y_3) \end{aligned} \tag{3.3.6}$$

$$\begin{aligned} m_f\ddot{x}_4 &- c_f\left(\dot{x}-\dot{x_4}-b_2\dot{\varphi}+a_2\dot{\theta}\right)-k_f(x-x_2-b_2\varphi+a_2\theta) \\ &+ k_{RR}\frac{1}{b_1+b_2}\left(\varphi-\frac{x_3-x_4}{b1+b_2}\right)+k_{tf}(x_4-y_4) \end{aligned} \tag{3.3.7}$$

3.4 Magic tyre formula

One of the most commonly used tyre models is the semi-empirical Magic tyre formula which is a mathematical curve approximation based on trigonometric functions. Developed in 1987 as a cooperative effort by Egbert Bakker and Lars Nyborg representing Volvo and Hans B Pacejka from Delft University of Technology this model computes the steady-state tyre force and moment characteristics, based on its physical characteristics and construction, in order to fit tyre test data (Pacejka, 2006).

The Magic tyre formula is named after its unique formulation that makes it possible to fit any test data regardless of the tyres physical properties through a set of mathematical parameters (Mohammadi, 2012).

The general form of the formula can be written as

$$y = D\sin\left[C\,arctan\left\{Bx - E\left(Bx - arctan(Bx)\right)\right\}\right] \tag{3.4.1}$$

and

$$Y(X) = y(x) + S_V \tag{3.4.2}$$

$$x = X + S_H \tag{3.4.3}$$

Where Y(x) is the output variable F_x, F_y or M_z defined as the longitudinal force, side force or aligning moment respectively. The remaining coefficients of the Magic Tyre Formula are described as (Pacejka, 2006),

B Stiffness factor

C Shape factor

D Peak Value

E Curvature factor

S_H Horizontal shift

S_V Vertical shift

The relationship between these coefficients and the tyre slip relation is shown in Figure 9.

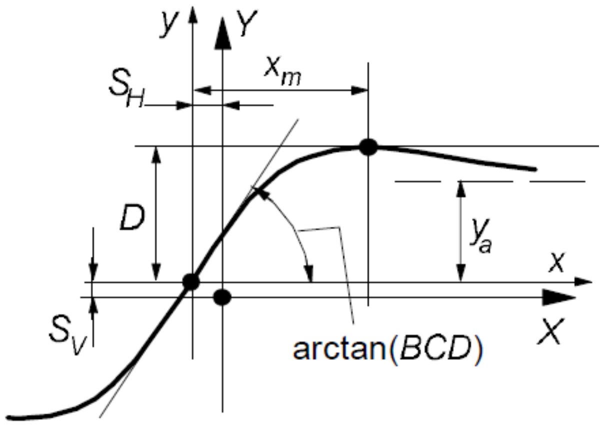

Figure 9, Magic tyre formula, tyre parameters (Pacejka, 2006)

For the complete explanation and formulation of the model refer to (Pacejka, 2006).

4 Simulation model

In order to study the vehicle behaviour during roll pitch and bounce motion a simulation tool was created in Simulink, which computes the behaviour depending on inputs given. The Simulink model allows simulations of almost any driving event wanted and gives the advantage of repeatability. This makes it possible to compare vehicle behaviour between various vehicles and damper characteristics in a simple and fast manner.

4.1 Black box

A black box containing required input and wanted output visualizes the basic structure of the simulation model, see Figure 10. Required inputs to the black box were vehicle- and tyre data as well as wanted test manoeuvres. The desired outputs were the vehicle motions, damper forces along with the vehicle requirements, which is described more thoroughly later.

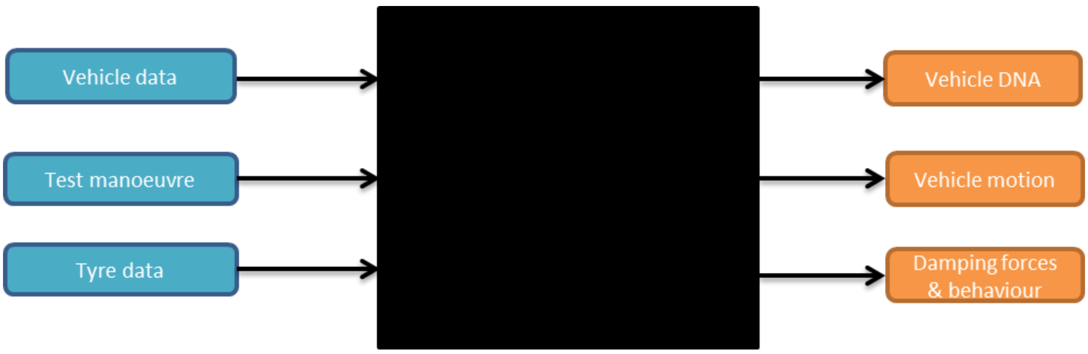

Figure 10, Black box describing the requirements for the simulation model

4.2 Simulation model

Planar motions of the vehicle can easily be described by the two track model but since roll, pitch and bounce behaviour was investigated as well, this model alone was not enough. The two track model was therefore combined with the full car vibrating model in order to better describe the vehicle behaviour. Simulink makes it easy to build these vehicle models separately and connect them.

Since essentially two different mathematical vehicle models were used, the two track model and the full car vibrating model, a link between these was needed. The two track model does not include any vertical, roll or pitch motions and the full car vibrating model does not include any lateral and longitudinal accelerations nor yaw motions. To overcome this problem the lateral and longitudinal acceleration from the two-track vehicle model was used as an input for the roll and pitch moment in the full car vibrating model as shown in equations (4.2.1) and (4.2.2).(Dixon J. C., 1991)

$$M_{roll} = \frac{m_s h_0}{I_{xx}} a_y + \frac{m_s g}{I_{xx}} \varphi \qquad (4.2.1)$$

$$M_{pitch} = \frac{m_s h}{I_{yy}} a_x \tag{4.2.2}$$

Due to lack of data for the investigated vehicle, no anti-dive or anti-squat has been included in the model. However, for the lateral acceleration the kinematic roll axis was considered. The roll centre height for both front and rear suspension as a function of roll angle was found from vehicle data and added to the model. No consideration of lateral roll centre movements was taken into account.

When evaluating the ride of the vehicle the full car vibrating model needs the road input on each wheel. In order to keep track on how far the vehicle have travelled the centre of gravities longitudinal displacement was added as an input to the full car vibrating model.

The lateral force produced by the tyre varies with the vertical load. The vertical tyre load is a function of the spring and damper forces which can be calculated from the full car vibrating model including anti-roll bars, see equations (4.2.3)-(4.2.6). The input to the two-track model, the lateral tyre forces, is therefore also dependent on the full car vibrating model. When these two models are combined they describe the vehicle body motion with 6 degrees of freedom. Together with vertical wheel motions and steer angles the model has 12 degrees of freedom.

$$dF_{z1} = F_{spring1} + F_{stops1} + F_{damper1} + \frac{1}{4\left(\frac{s}{2}\right)}(-K_{rf}\varphi - h_{rf}m_f a_y) \tag{4.2.3}$$

$$dF_{z2} = F_{spring2} + F_{stops2} + F_{damper2} + \frac{1}{4\left(\frac{s}{2}\right)}(K_{rf}\varphi + h_{rf}m_f a_y) \tag{4.2.4}$$

$$dF_{z3} = F_{spring3} + F_{stops3} + F_{damper3} + \frac{1}{4\left(\frac{s}{2}\right)}(-K_{rr}\varphi - h_{rr}m_r a_y) \tag{4.2.5}$$

$$dF_{z4} = F_{spring4} + F_{stops4} + F_{damper4} + \frac{1}{4\left(\frac{s}{2}\right)}(K_{rr}\varphi + h_{rr}m_r a_y) \tag{4.2.6}$$

$F_{springi}$ is the spring force currently acting on the wheel, $F_{stopsi}$is the force from the bump and rebound stops (if engaged)and $F_{damperi}$ is the damper force. The force from anti-roll bars and forces through the suspension linkage were also added to the equations.

Damper forces were calculated through a lookup table where the damper speed, derived from the full car vibrating model was used as an input. In order to get a more accurate vehicle model behaviour the bump and rebound stop had to be added as an input to the model. Tables describing the force versus displacement characteristics for the stops were added to the model together with the clearance before the stops are engaged.

Tyre forces were computed with MF-Tyre which is a Simulink compatible software developed by TNO Delft that uses Magic Tyre Formula to describe the tyre behaviour based on tyre files supplied by manufactures. With help of the wheel centre-, lateral- and rotational- velocity of the wheel together with the vehicles vertical load it determines the lateral forces produced in steady state for each tyre separately. Good tyre data is therefore required in order to get an accurate result from the model.

Figure 11 visualizes how the different subsystems were connected in Simulink and gives a better overview of how a change to subsystems affects the models behaviour. The green boxes shows the required inputs to the model, the blue boxes main subsystems in the model and the arrows represents the information flow in the model.

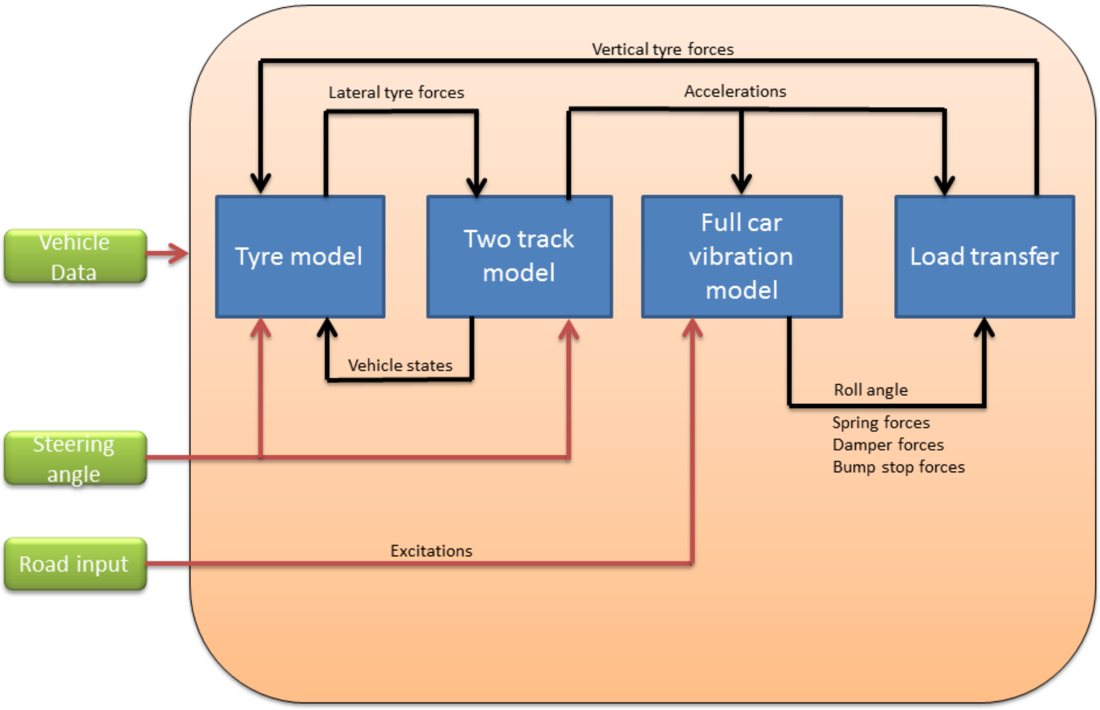

Figure 11, Flow chart of the Simulink model

In order to run the simulation tool the user has to know a number of set parameters for the specific vehicle. The required input parameters can be seen in Table 2 found in

Appendix A. Since the springs and dampers are not located directly above the wheels, the spring stiffness and damper data had to be modified accordingly. The effective spring stiffness, the spring stiffness at the wheels, is given by the spring stiffness corrected by the motion ratio between the spring and the wheel centre. The damper forces at the wheels are given by the same correlation. For instructions of how to run the simulation model, refer to Appendix B, where a description of the graphical user interface can be found.

5 Test procedures

Tests have been performed with various damper settings. Volvo Cars receives damper force versus velocity data for different damper settings from suppliers. Figure 12 shows an example of how the force vs. velocity graph for an unspecified vehicle can look like. These documents have a number of fixed test velocities which are used in each test.

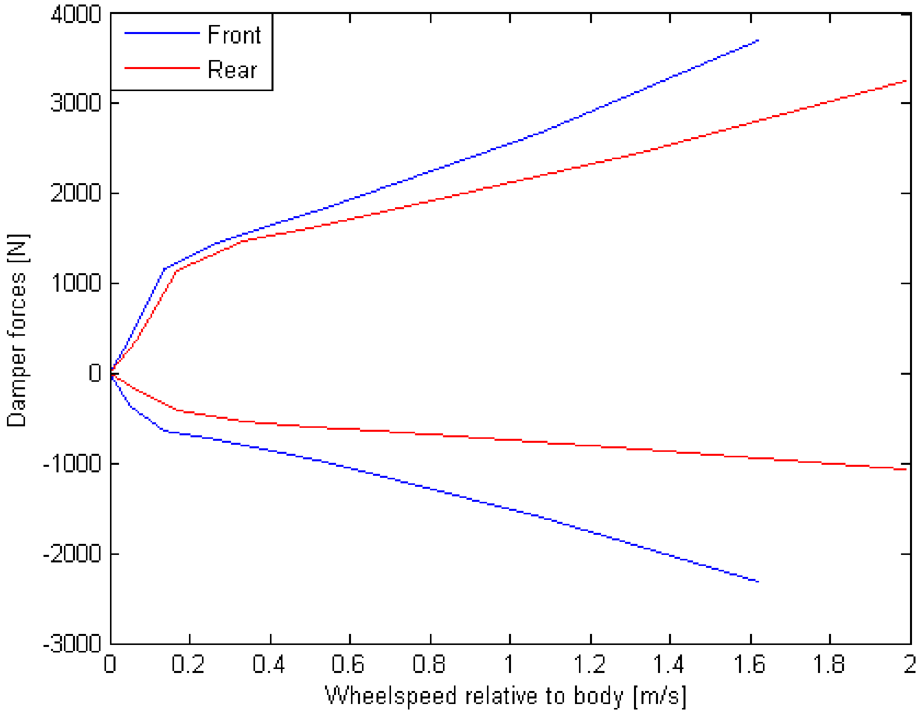

Figure 12, Force vs. velocity plot for an unspecified vehicle

When damper settings were changed, both in simulations and real vehicle tests, damper forces were varied at these points. These damper velocities are in the report referred to as the first, second, third, and so on, damper speeds or points according to Table 1. The same velocities are used for rebound and compression damping.

Table 1, Damper speeds at which the damper force is given by the supplier

Point/Speed	1	2	3	4	5	6	7
Damper velocity [m/s]	0.052	0.131	0.262	0.393	0.524	1.047	1.571

5.1 Test events

A number of test procedures have been chosen to evaluate the dynamic behaviour of the vehicle. These tests were used to make objective measurements for evaluation of the vehicle steering and handling performance.

5.1.1 Single sinusoidal

The most studied test was the single sinusoidal input also referred to as a single lane change. The vehicle is driving in a straight line at 80 km/h when a single sinus steering input at 0.5 Hz is given. The amplitude of the steering input is calibrated to achieve a lateral acceleration of 4 and 7.5 m/s^2 in two different tests. The tests output is mainly the time lags from the two peaks of the steering wheel angle to yaw rate, lateral acceleration and the roll angle, respectively, see Figure 13. These measurements should give an indication of the vehicles responsiveness and agility. The time lag between yaw rate and lateral acceleration was also investigated as well as pitch and roll behaviour.

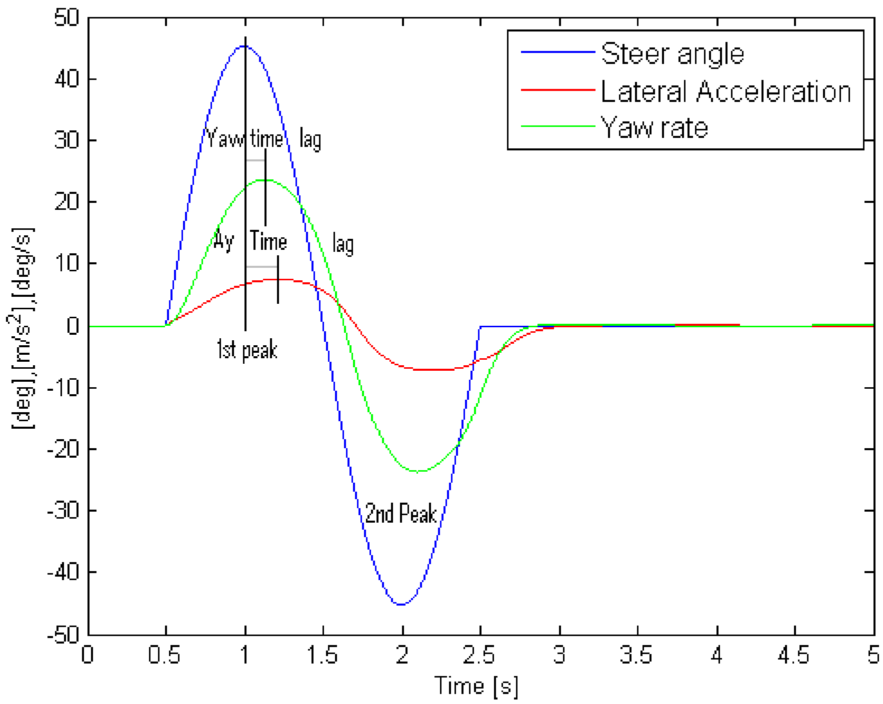

Figure 13, Single Sinusoidal time lags

5.1.2 Frequency response

The frequency response test is performed at 75 and 120 km/h. A sinus wave with increasing frequency from 0.1 to 3 Hz at constant amplitude is given as a steer input. The amplitude is calibrated to achieve lateral acceleration of 0.32 g for a sinusoidal steering input of 0.2 Hz according to the standard test procedure used. Most important outputs from this test are gains and phase lags for yaw and roll motions at frequencies of 0.1 to 3 Hz. These phase lags indicates how fast the vehicle responds to a steering wheel input for different frequencies and the gains shows the magnitude of the response compared to the input. A larger phase lag means a slower response and vice versa. It is a useful test for comparing the roll response of vehicles with different damper settings. Another measurement which evaluates the roll damping of the vehicle is the roll rate to lateral acceleration gain at 1 Hz.

5.1.3 On centre steering

The on centre steering manoeuvre is performed at 50, 75 and 120 km/h at 0.2 and 0.4 g maximum lateral acceleration. The steering input is a sinus wave with a frequency of 0.2 Hz. This test evaluates the vehicles response to small steering wheel inputs. Some useful measurements from this test is the yaw rate gain and lateral acceleration gain. The test is also used to measure the steering wheel torque response. This is mainly influenced by the steering system, which is not included in the simulation model used, and the tyres.

5.1.4 Primary ride evaluation

The primary ride evaluation is performed on three different roads, two roads with large amplitude disturbances and one road with small amplitude disturbances. The vehicle is driven on these three roads separately and the vehicle motion is measured. Main outputs from this test are vehicle roll-, pitch- and bounce- behaviour. It is also used to find any sudden changes to the roll and vertical acceleration of the body and its jerkiness. These sudden changes, measured by an abruptness index, should be avoided in order to have a comfortable vehicle.

5.1.5 High and low g swept steer

The high and low g swept steer tests are Steady-state cornering tests. The steering wheel angle is increased slowly to achieve an increase in lateral acceleration of 0.06 to 0.1 g/s. The test is performed at 75 and 120 km/h. Outputs from the tests are the steering dead band window as well as the understeer- and roll- gradient. The steering dead band window shows how much the steering wheel have to be turned to get any noticeable response from the vehicle. The tests are also used to measure the steering wheel torque response. However this is mainly influenced by the steering system and the tyres which are not modelled in detail in the simulation model used.

5.2 Vehicle evaluation

Volvo Cars use vehicle requirements to objectively measure the performance of the vehicle. The main features describing the dynamics of the vehicle are the steering-, handling- and ride requirements. The evaluation contains a number of metrics which should be within certain boundaries. These boundaries are different for most car models and target values are set based upon which type of car that should be developed. Metrics considered to be of importance and that would be affected by changes in damper characteristics were calculated from the simulation results and evaluated. Since the purpose of this thesis was to study the dampers influence on vehicle behaviour there is no point in studying steady-state events, due to the lack of damping forces. What the vehicle evaluation sheet measures can be found in Appendix C.

5.3 Subjective and objective test methods

Subjective and objective tests have been carried out at Hällered Proving Ground (HPG) to verify the simulation model. HPG is Volvo Car Group's test facility where most of their vehicle tests and verification work are done. This is also were most of the damper tuning work is carried out.

A test vehicle fitted with Continuously Controlled Damping (CCD) made it possible to adjust the level of damping quick and easy. The CCD system also allowed testing several damping configurations without changing any dampers. Volvos CCD system uses magnetic valves in the dampers, controlled by an applied current to change the characteristics of each damper continuously. Normally the system is active when the vehicle is moving and changes the damper settings according to a control algorithm. However, for the tests, a static damper characteristic was desired and the active continuous control was therefore disabled.

A number of damper settings were defined based on results from the simulation model. These settings were then validated through subjective and objective measurements in order to see if there was any correlation between these and the simulation results.

5.3.1 Objective measurements

A steering robot was installed in the vehicle for objective validation. The aim of this assessment was to see how well the simulation model performed compared to real vehicle behaviour and if changes made to vehicle had the same effect in both cases.

The test procedure chosen for validation was the single sinusoidal since the purpose of the test was also to study the vehicle response and turn-in for different damper settings. Furthermore, the test made it possible to study the vehicles roll and pitch behaviour. Collected data was then compared to simulated results as well as subjective assessments.

5.3.2 Subjective assessments

The subjective validation was carried out by experienced engineers used to the tuning process. The aim of the tests was to see if the drivers could feel difference in vehicle response, ride as well as roll and pitch behaviour between the standard dynamic chassis and any of the predefined damper settings.

The drivers were asked to evaluate the car during different driving events which they were left free to choose themselves. The dynamic damper setting was first driven as a reference. This was followed by a period of any pre-defined chassis settings unknown for the driver. The drivers were then asked to fill in an evaluation document by rating the vehicle behaviour from 1-10. The subjective evaluation sheet is found in Appendix D. Additional notes and comments were also recorded during the assessment in order to better understand what difference the driver felt.

6 Results

The results are separated into simulation results, objective results and subjective results. Objective results carried out by robot tests are compared to simulated results in order to examine the validity of the simulation tool. Subjective results are also presented indicating what difference the driver experiences during the different damper settings.

A critical damping curve is shown in Figure 14. It displays the percentage of critical damping for front and rear dampers in compression and rebound at various wheel velocities. The front compression damping starts at a high value, almost double the amount of rear compression for really low velocities, and is reduced as the velocity is increased. Rebound damping levels are higher at low velocities and increases even more until $0.2\ \mathrm{m/s^2}$. The damping is then reduced until $1.2\ \mathrm{m/s^2}$ where it levels out. Noticeable is also that the rear damper levels in compression and rebound has the largest gap between them. Enclosing the front damper levels this gives a higher damping ratio, see Figure 15.

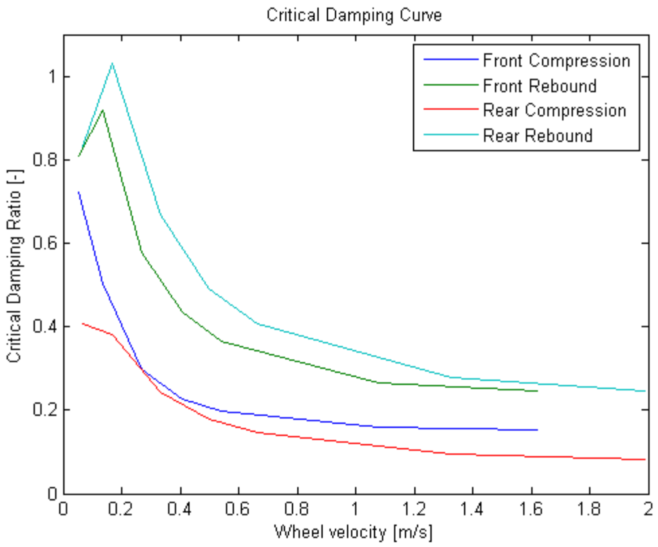

Figure 14, Damping ratio for a standard dynamic chassis

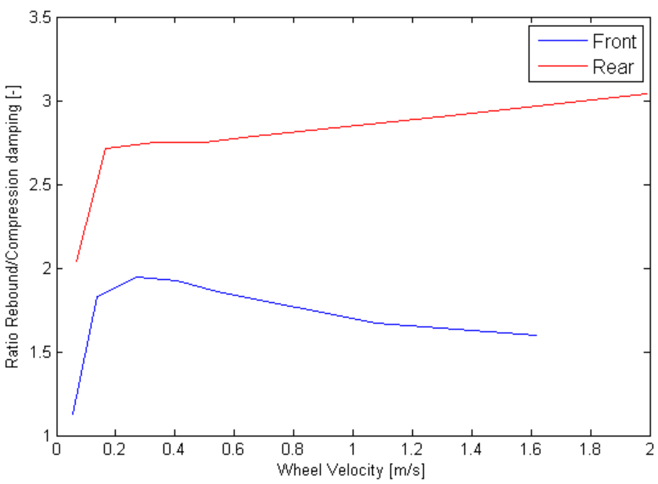

Figure 15, Ratio between rebound and compression damping, standard dynamic chassis

6.1 Simulation results

Results from simulations have been divided into roll, pitch and the vehicles handling behaviour. They are based on the standard tests, explained earlier, and carried out for investigation of vehicle behaviour during transient manoeuvres.

6.1.1 Roll

Figure 16 and Figure 17 shows the vehicles roll behaviour, with increased damping in compression and rebound both front and rear, during a single sinus manoeuvre at the lateral acceleration of 7.5 m/s^2. As seen in Figure 16, an increase of damping does not seem to affect the vehicles roll behaviour at all for this test except for the roll acceleration. However, if the graph is zoomed, see Figure 17, it is seen that the roll angle is reduced if front and rear rebound damping is increased. The roll rate is reduced for all changes except rear compression where it increases.

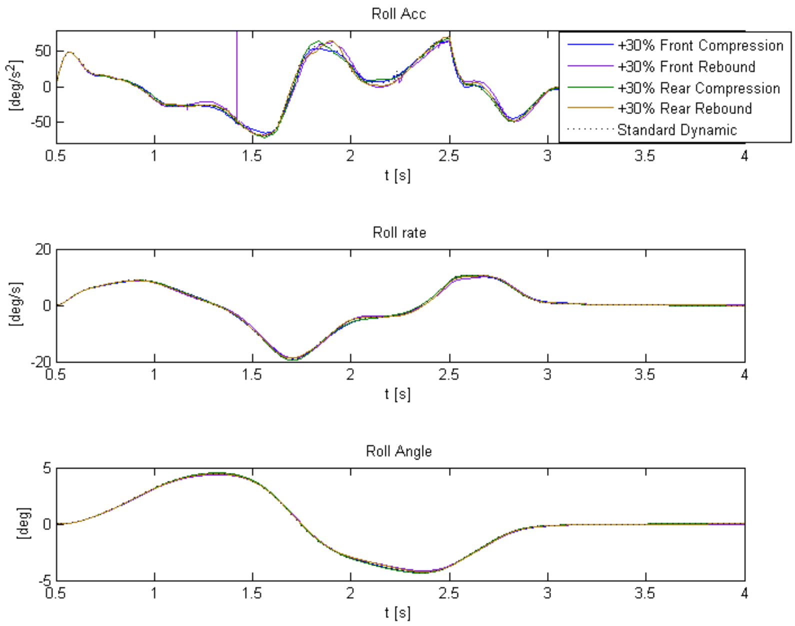

Figure 16, Roll behaviour during single sinus event at 7.5 m/s^2, 30 % added damping

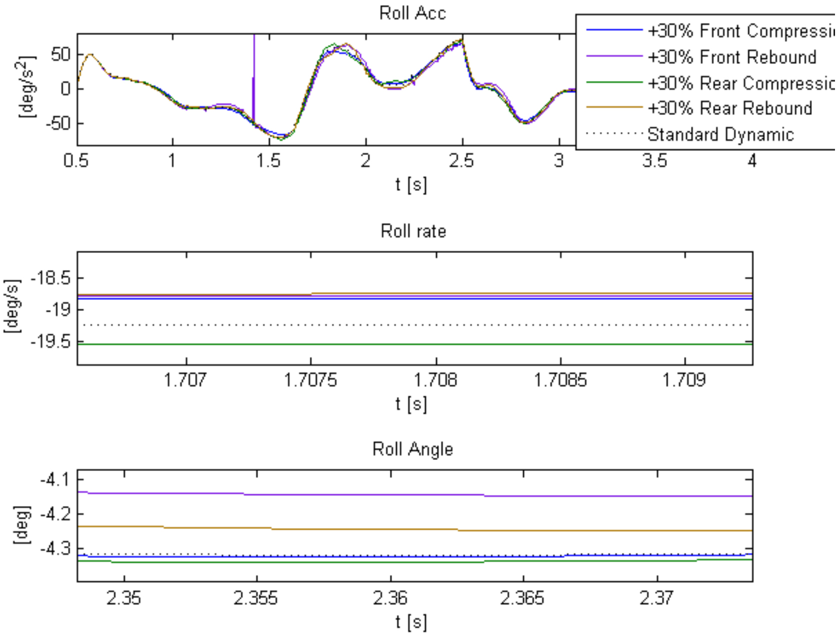

Figure 17, Roll behaviour during single sinus event at 7.5 m/s^2, 30 % added damping, Zoomed

The same event also showed that the vehicle hits the bump stop regardless of damping level. This was also noticed during the high g test. Figure 18 shows the suspension deflection during a high g event when turning left. Positive values on the horizontal axis correspond to rebound motion and negative to compression motion. As the vehicle is steered left it can be seen that the vehicle hits the bump stop in the right front corner after 6 seconds and is not compressed noticeably after that. This is also realised in Figure 19 where the vertical tyre force on the right front corner increases until the vehicle hits its bump stop. As the bump stop is reached more force is applied to the rear right corner which reaches the same vertical force as the front.

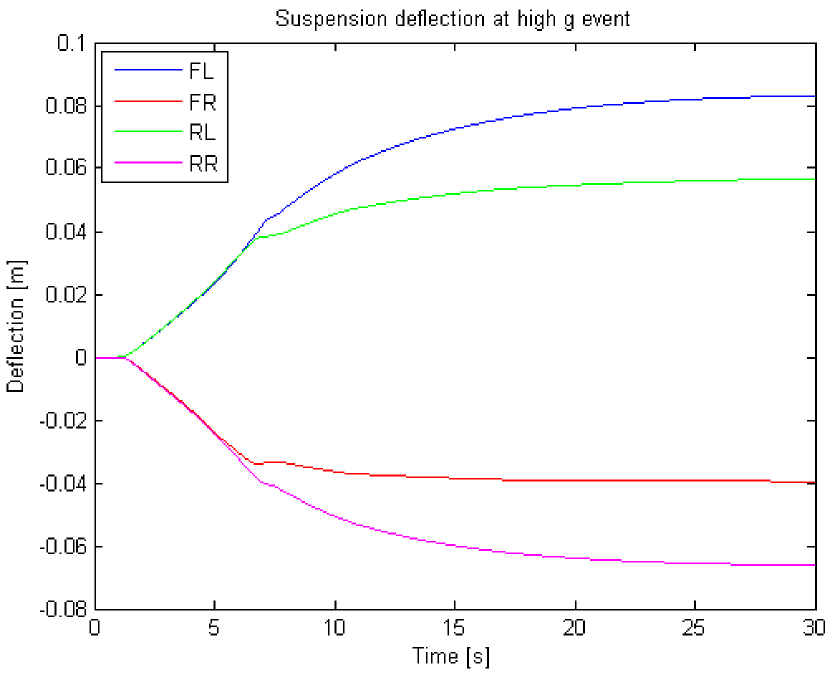

Figure 18, Suspension deflection at high g event

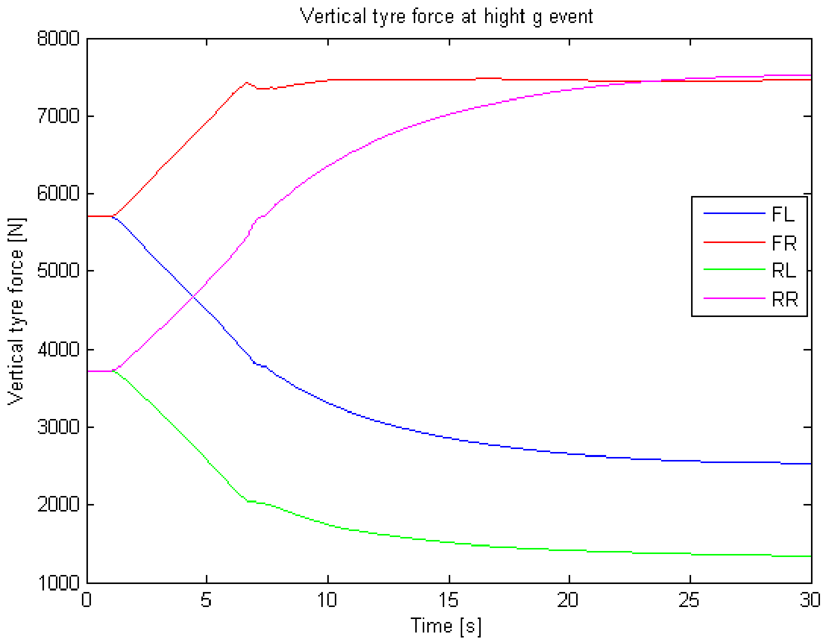

Figure 19, Vertical tyre forces at high g event

Figure 20 illustrates the difference in roll behaviour for the standard dampers and a smoothened damper curve where the low speed damping has been reduced, see Figure 21. Biggest difference is seen in the roll rates peak value. When the low speed damping was reduced the peak value increased with 6.5 % for a step steer manoeuvre. The anti-roll bar stiffness was then raised with 50 % and the same level of critical damping was kept during the same manoeuvre. With a stiffer anti-roll bar the roll rate for the smoothened damper levels increased with 5 % compared to the basic damper levels. When the spring stiffness was increased with 50 % the difference between the damper settings was smaller. The smother damping curve gave only a 4 % increase in roll rate. The level of roll damping for the smoothened and the standard dynamic damper lever, for normal anti-roll bar and spring stiffness, are shown in Figure 22.

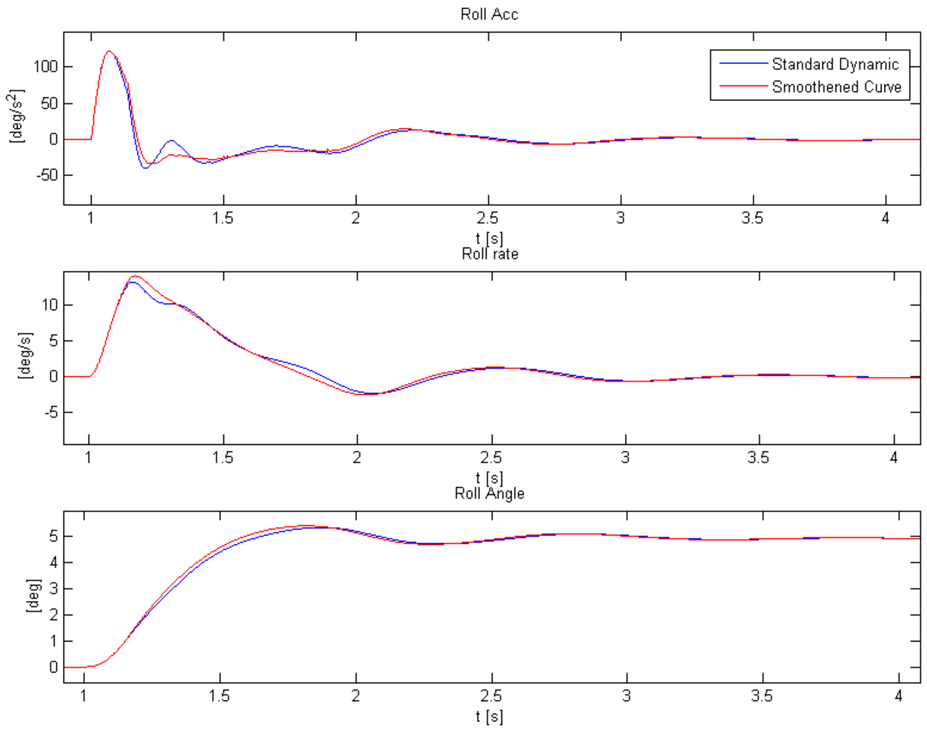

Figure 20, Roll behaviour for a step steer input comparing the standard damper setting with the smoothened damper curve

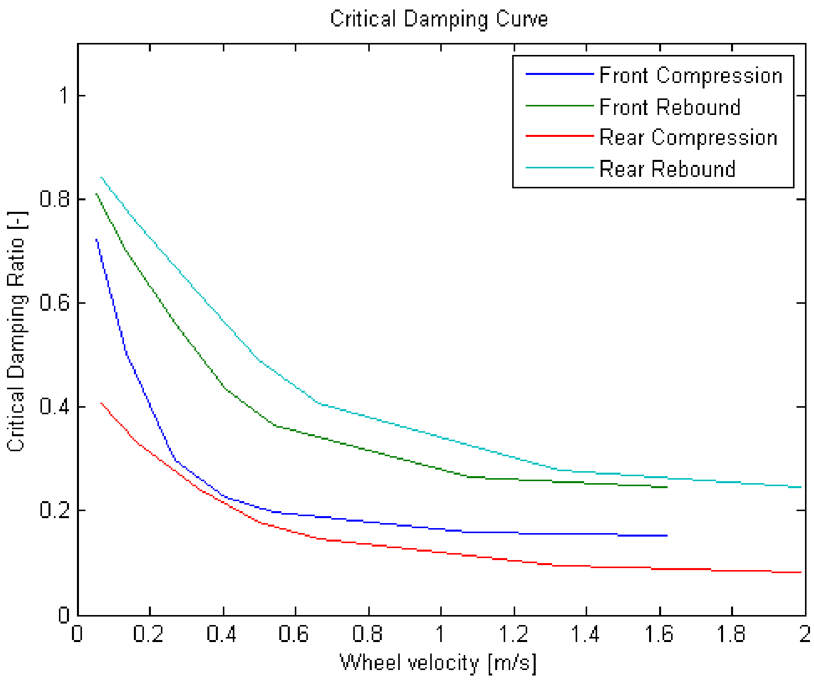

Figure 21, Smoother damping curve

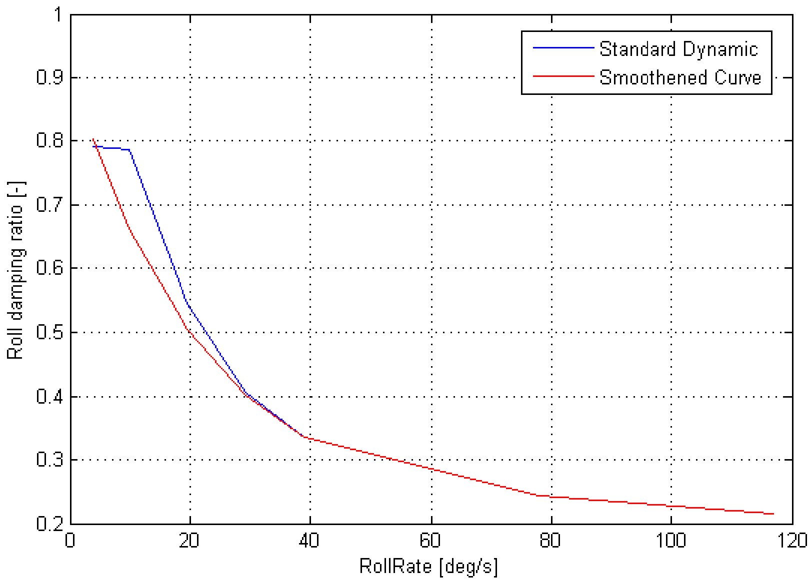

Figure 22, Roll damping ratio for the standard dynamic damper setting and the smoothened damper curve with normal spring and anti-roll bar stiffness

6.1.2 Pitch

The pitch angle, pitch rate and pitch acceleration during a single sinusoidal test at a lateral acceleration of 4 m/s^2 for increased damping can be seen in Figure 23. If the front compression or rear rebound damping is increased the vehicle tends to pitch more rearwards compared to the standard dynamic damper level. The front rebound and rear compression have the opposite effect on the pitch angle if increased. By adding more front rebound damping or rear compression damping, the pitch rate is kept closer to zero during the whole manoeuvre compared to the other configurations. Figure 24 shows the same manoeuvre at 7.5 m/s^2 with equal damper settings. The pitch angle is affected in a similar way as for 4 m/s^2 but the pitch rate is higher for the configurations where front rebound and rear compression is increased.

Figure 33 and Figure 34 found in Appendix E shows the same single sinusoidal manoeuvre at 4 and 7.5 m/s^2 for reduced damping. These figures show the same relation as previous tests with added damping. A lower front compression or rear rebound damping tends to promote a forwards pitching motion and a lower rear compression or front rebound damping promotes a rearward pitching motion. This is also true on disturbance roads as seen in Figure 25. The lower front compression and rebound damping keeps the pitch rate at lower levels during the test at 4 m/s^2 but gives a higher pitch rate at 7.5 m/s^2.

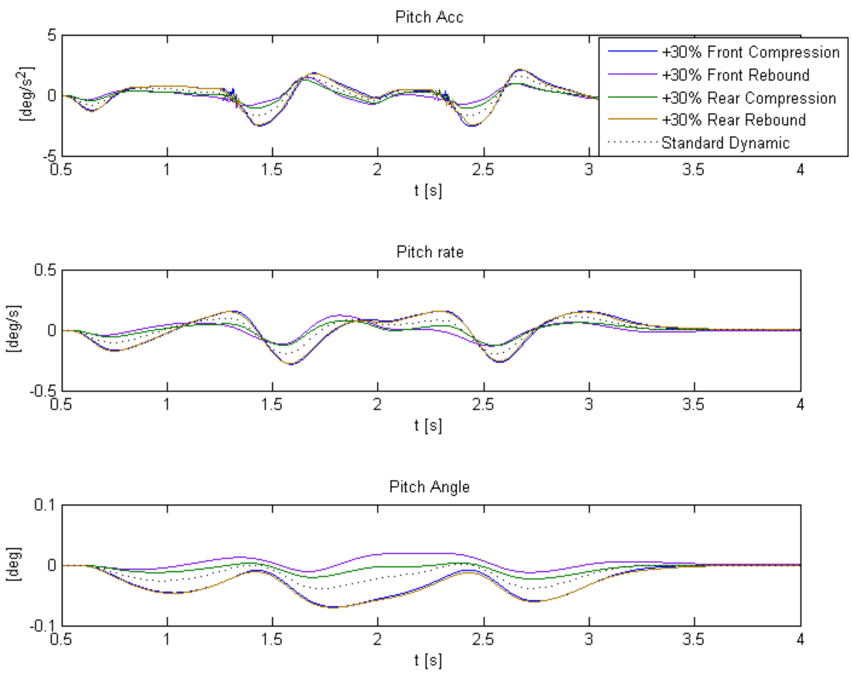

Figure 23, Pitch behaviour during single sinus event at 4 m/s^2, 30 % added damping

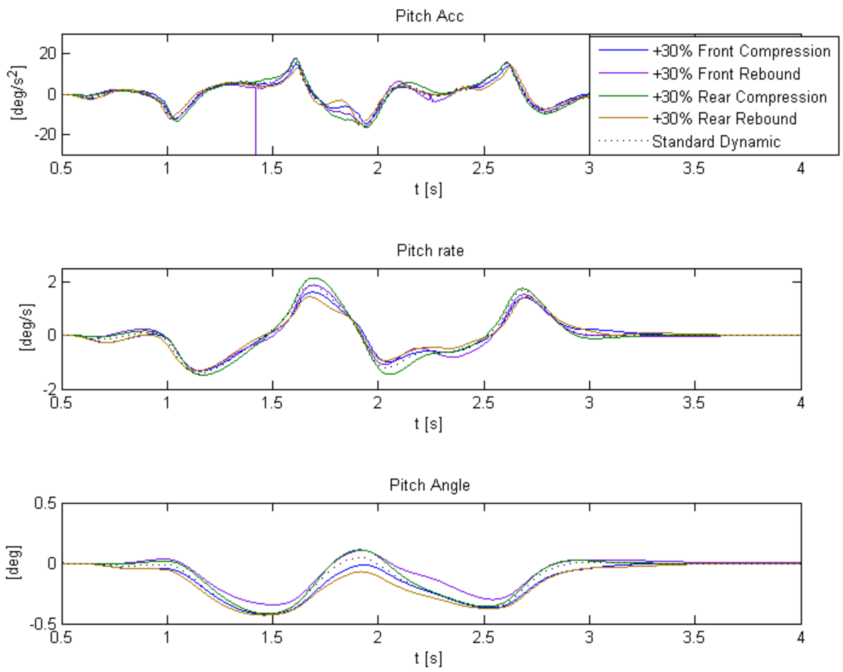

Figure 24, Pitch behaviour during single sinus event at 7.5 m/s^2, 30 % added damping

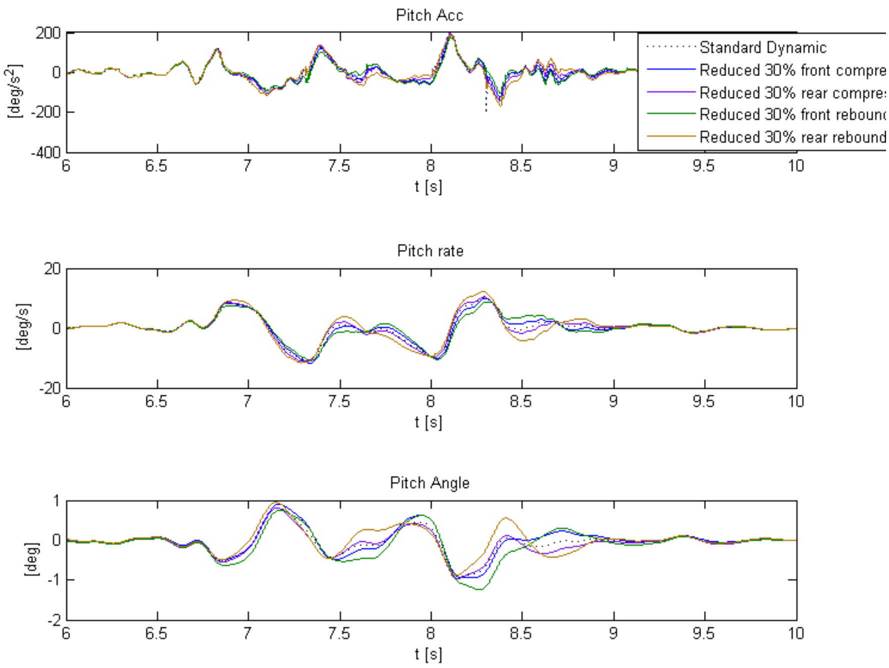

Figure 25, Pitch motion long dips, 30 % reduced damping

6.1.3 Bounce

The centre of gravity displacement for single sinusoidal manoeuvre is shown in Figure 26. A negative displacement was found for all damper settings with added damping. If the rebound damping is increased, either front or rear, the centre of gravity has a bigger negative displacement compared to the standard dynamic. Increasing the compression damping has the opposite effect on the vehicle.

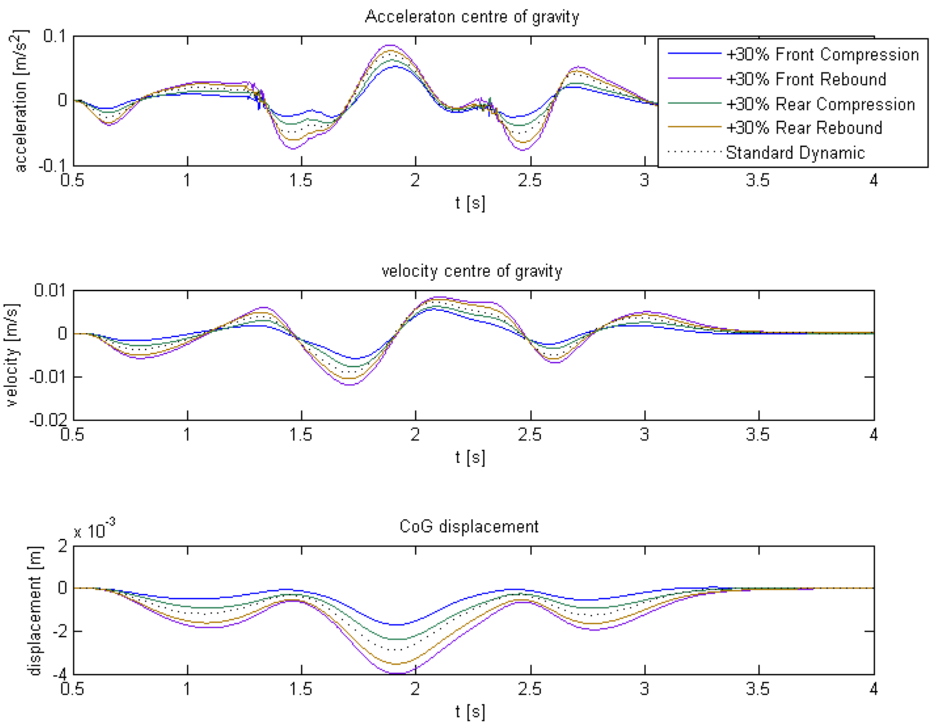

Figure 26, Centre of gravity displacement during single sinus event at $4\ \mathrm{m/s^2}$, 30 % added damping

The centre of gravity displacement compared to the ground on a road with large amplitude disturbances can be seen in Figure 27. When rebound damping is reduced the centre of gravity motion reaches higher positive values compared to standard dynamic setting. Some differences can also be observed in the negative travel for reduced rebound damping but these differences are smaller than in bounce. If the compression damping is reduced the negative motion increases more than the positive displacement.

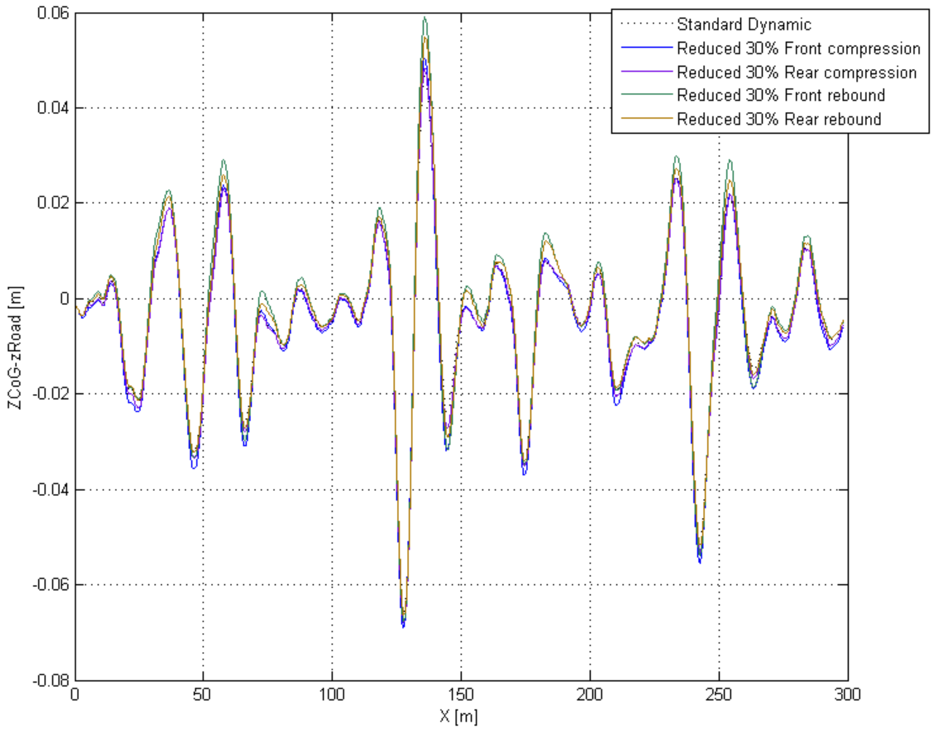

Figure 27, Centre of gravity displacement during FEC road, 30 % reduced damping

6.1.4 Single sinus time lag

Results from the time lag evaluation for the single sinus event at 7.5 $\mathrm{m/s^2}$ can be found in Appendix F as Table 4. Damper forces where only changed for the three first defined damper velocities since higher damper velocities were not reached during this event. This is also seen in Table 4, as damper forces were changed at the third velocity no changes were found for the first peak where damper speeds are low.

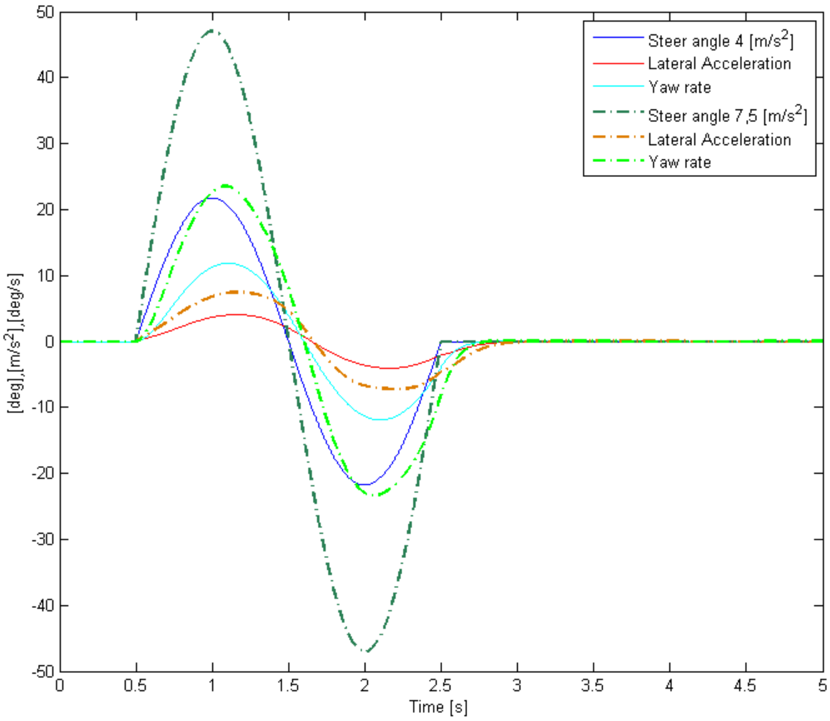

Figure 28, Comparison between 4 $\mathrm{m/s^2}$ and 7.5 $\mathrm{m/s^2}$ during the single sinus event

Simulations show that the roll angle time lag decreases if damping is reduced at the first damper velocity. A lower compression damping, both front and rear, at the first and second damper velocity reduces the time lag for yaw rate and lateral acceleration at the first peak. An increased compression damping has the opposite effect. When the rear rebound damping was increased at the second damper velocity, lower lateral acceleration and yaw rate time lags at the first peak were found. At the first damper velocity however, an increase in rear rebound damping increases the yaw rate time lag. The time lag varies depending on the lateral acceleration which is seen in Figure 28.

From the results it can also be noticed that if the lateral acceleration to yaw time lag was small at the first peak it was often greater at the second peak. There are however, some exceptions that do not follow this pattern. The biggest reduction in time lags overall was found for the configuration where 30 % front and rear compression damping at the second damper velocity was removed.

6.1.5 Vehicle evaluation

Due to confidentiality the evaluation document as whole cannot be displayed. However, it was discovered that the vehicle evaluation document for steering and handling does not capture the damper level changes very well and no big changes could be observed. The ride evaluation on the other hand managed to catch significant changes, especially in abruptness and bounce. How the abruptness and bounce varies with damper level is displayed in Table 5 and Table 6 found in Appendix G.

6.2 Objective test Results

Results from the comparison between simulation results and objective measurements are presented in Figure 29. In the objective test a lateral acceleration of 7.5 $\mathrm{m/s^2}$ was reached, the same level as in the simulations. A higher steering wheel angle was required in the real car compared to the simulations as illustrated by the figure and a higher yaw rate was reached in the simulation compared to the real test. With a roll centre height from Adams, a multibody dynamic software, the roll angle was greater in the simulations, 4.5 degrees compared to 3 degrees during the test. When a roll centre height based on K&C measurements was used in the simulation model, a maximum roll angle of 3.3 degrees could be found. The maximum simulated longitudinal acceleration reached in the manoeuvre was -0.5 $\mathrm{m/s^2}$ compared to around -0.55 $\mathrm{m/s^2}$ carried out in the physical test.

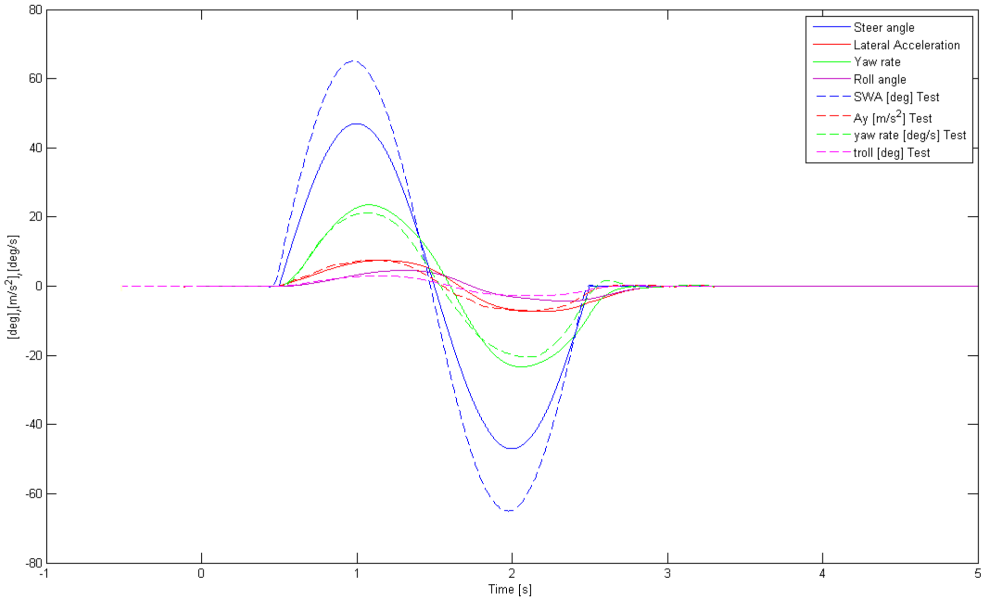

Figure 29, Simulation results and objective test data from single sinusoidal at 7.5 $\mathrm{m/s^2}$ with roll centre height from Adams model

Pitch rate measurements from the steering robot data is presented in Figure 30. With roll centre height from measurement the pitch rate was lower compared to simulation results based on data from Adams. The manoeuvre starts at 1 second in the physical test plot and at 0.5 seconds in the simulation plot.

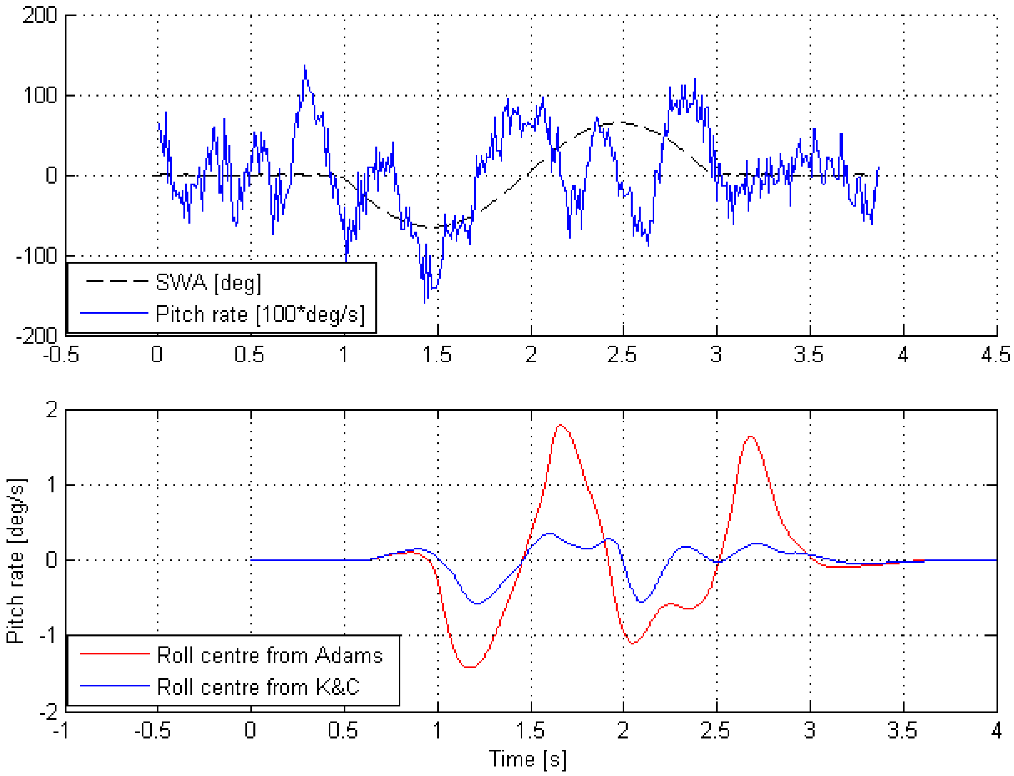

Figure 30, Pitch rate, measurements versus simulations for the two different roll centre heights

Time lags from the real vehicle single sinus test at 7.5 m/s^2 can be seen in Table 7 found in Appendix H. Results show the mean values from six tests performed for each configuration. The difference in time lag compared to the standard configuration, which used as a starting point, is also included in the table. Table 8, shows the simulation results from the same test for the different configurations and Figure 35, also found in the appendix, displays the trend lines plotted for all configurations. A few observations can be made by looking at the data. The time lag for yaw rate is decreasing from the first to the second peak in the simulation but the opposite is true for the real vehicle as seen in the data. The lateral accelerations time lags have generally higher values in the simulations compared to the test data. Lateral acceleration to yaw time lags was also smaller for the real vehicle compared to the test. This is especially true for the second peak where the simulation model indicates higher time lags compared to the first peak and the test data suggests a shorter time lag. The time lags for the roll angle is also smaller in the test data compared to the results from the simulation but both sources indicates the same pattern with increased time lag for the second peak for the roll angle.

6.3 Subjective test results

The subjective grading of the different damper configurations from the test drivers can be seen in Appendix I. The results are only based on if they are better or worse than the benchmarked dynamic setting. Comments from the test drivers regarding the different damper settings are presented below. The second point is defined in Table 1.

6.3.1 Configuration A, -30% compression front & rear second point

The drivers thought that the car became much softer dynamically, especially on long dips and other roads with small disturbances. Even though it was softer the vehicle still felt balanced and controlled since it settled quite fast after excitation. The roll behaviour was still parallel and good but the roll control became worse since the vehicle bounced on the bump stops during cornering. The response, how fast the vehicle responds to steer input, was lower. The turn-in, how much the vehicle responds to steer input, increased a bit, most likely because the vehicle felt less under steered according to the driver.

6.3.2 Configuration B, -30% compression front second point

First impression was that the vehicle had a tendency to pitch more forward. Wheel control became worse, even though the road inputs wasn't felt that much. A feeling of increased movements in the front gave the impression of the vehicle not following the road. The roll behaviour became unpredictable and uncontrollable. It felt like the vehicle body moved more towards the outer front wheel during cornering and the rear end drifted out. No big difference in response was felt but the turn-in was increased.

6.3.3 Configuration C, -30% compression rear second point

This configuration made the vehicle feel jerkier, thus small road disturbances were felt more by the driver, especially in the front. The feeling of the dampers not utilizing their full movement was also noticeable. One driver on the other hand thought that this configuration gave a better impression in comfort since the inputs felt smoother. The drivers agreed that the vehicle became more controlled for low frequency motions.

This configuration also resulted in more vehicle roll and the vehicle hit the bump stops which made the vehicle under steered at first and over steered when the rear bump stops were activated. The response was reduced and the steering became less sharp and the feeling of connection between the steering wheel and wheels became worse, especially around zero steering angle.

6.3.4 Configuration D, +30% compression rear second point

The rear end of the vehicle became jerkier and the pitch balance was worse. A feeling of less compression damping in the front and a more controlled vehicle during roll was experienced. The vehicle felt more parallel with the road in the long dips. The driver experienced that the vehicle raised the front during roll, which was thought to be a consequence from increased roll angle. Initial response was reduced and the vehicle became less understeered.

6.3.5 Configuration E, +20% rebound front second point

The final configuration felt jerkier. Whether it came from the front or rear was not perceived but it made the vehicle more nervous on roads with small disturbances. The drivers were very confident that the vehicle became more balanced in long dips, even though they experienced that the vehicle was “sucked down” in the front. The roll control became better and the roll angle was reduced. The vehicles response increased just like its turn-in capabilities.

7 Discussion

Is the simulation model valid or not, how does the level of damping affect the vehicle behaviour and are the physical tests carried out enough to be able to draw any conclusion between subjective evaluation and objective measurements? This chapter combines and discuss the results based on these topics.

7.1 Validation of model

The five damper configurations that have been used in order to validate the simulation model were chosen based results in Table 4 found in Appendix F. The single sinus event allowed investigation of roll and pitch behaviour, it also made it possible to investigate the vehicles different response times. These were investigated in order to see if any correlation could be found between a perceived responsive vehicle and objective measurements. Thus, its yaw rate-, lateral acceleration- or roll rate lag time. According to a study by C. G. Fernandes et al.,(da Hora Passos, Fernandes, & de Mello, 2007), there is a significant correlation between a quick subjective steering response and phase lag between steering wheel angle and roll rate. The time lags were also investigated in order to see if there was any correlation between results from the simulation model and the robot test as the damper configurations were changed.

Since the simulation model is a simplified vehicle model less attention has been paid to absolute numbers. Focus has instead been on trends and vehicle reactions to changes. Figure 35 which shows the trend lines of yaw-, lateral acceleration- and roll angle- time lags for all settings allows an easy way of comparing the vehicle behaviour between simulations and robot test result throughout the configuration range. It is noticed that the simulation results are delayed in most cases compared to real test data. The reasons for this behaviour can be explained by various reasons. First of all, the model is a basic model based on a rigid chassis which does not consider compliance effects such as roll steer and no effort has been put into modelling i.e. the camber and toe change as the vehicle rolls. The tyre model used is simplified and determines the tyre lateral force in steady state only and the tyre data might not be as valid as thought.

During the validation phase it was also found that the roll centre behaviour received from physical testing did not match the roll centre from CAE. Kinematic simulations from CAE states that the roll centre height decreases as the vehicle rolls while a kinematic and compliance (K&C) measured data claims the opposite trend. The CAE behaviour was used in the comparison between the simulated and objective results. However, when comparing the measured roll centre behaviour with simulated and physical tested results, see Figure 29, it is realised that the measured roll centre behaviour gives a better correlation with the physically tested vehicle. This was also the case when pitch was considered as seen in Figure 30. Furthermore, according to measurement data it is determined that the physical vehicle hits the bump stop much earlier than the simulated. This may also explain why the simulated vehicle allows a higher roll angle. Another difference between the simulation model and the results from the test is the steering wheel angle required to reach the lateral acceleration specified. The fact that no steering compliances were included in the model could also be an explanation of the difference.

However, if the delays are neglected and a comparison between the different trend lines is carried out it is possible to determine some correlations. In most cases the trend lines between the robot test and simulated results have similar levels, especially if compared to the dynamic setting. The fact that the vehicle data in the simulation model is based on old K&C measurements and not the tested vehicle gives some doubts. Difference in i.e. vehicle mass, roll and pitch inertia directly affects the vehicle behaviour and would also affect the result significantly.

7.2 Level of damping

In several textbooks about vehicle dynamics the damping ratio is often assumed to be constant, one value for rebound and one for compression. As seen in Figure 14 this is not true for real dampers. The damping ratio is actually higher at lower velocities and lower at higher damper velocities. If a constant damping ratio would be used a compromise between the low speed and high speed damper velocities have to be made. This would most likely, depending on the priorities, increase the high speed damping and introduce larger damper forces to the body on disturbances that produce higher damper velocity. It would also lead to lower low speed damping levels which would give less body and wheel control. It is therefore beneficial to have a damping ratio which varies with the vertical wheel velocity.

The tuning engineers stated that the damper curves should be smooth. If the curve is not smooth, for instance if there is a flat part on the curve, the damping force would change unpredictably and noticeable for the driver. The force versus displacement curve, for a damper setting with a constant damping ratio between the second and third damping velocities in front compression, can be seen in Figure 31. This curve is often referred to as the egg plot. As seen in the figure, at $0.524\mathrm{m/s}$ the force will increase quite rapidly for the front damper in compression until the flat part of the curve is passed. The damper force will then decrease until the velocity decreases which occurs at 0 displacement. This has been experienced by the tuning engineers and can be felt as if the damping is lost for a moment. Compare this to the rear damper where the force is smooth during the whole displacement of the damper. The egg plot is a useful tool to find any irregularities in the damping force and has therefore been added to the simulation tool.

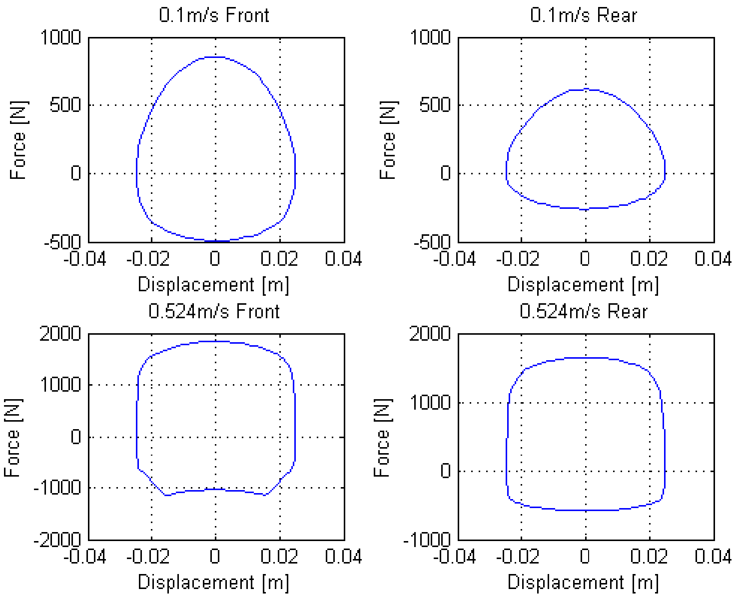

Figure 31, Force versus displacement curve for a damper setting with a flat part of the front compression curve

This study has focused both on the influence of specific low speed damping ratios on the vehicles roll, pitch and bounce behaviour but also on the influence of changing the level of damping over the whole range of vertical wheel velocities. During handling manoeuvres on a flat surface it is hard to reach high damper velocities which mean that the damping ratio for the fourth velocity and above does not influence the vehicle behaviour during such events. When these events are examined it is therefore not useful to study the influence of any higher damper velocities. In everyday driving on the other hand the roads are not completely flat and disturbances will force the wheels to move vertically at higher velocities.

As seen in both the objective and subjective measurements, changing the damping ratio the second damper velocity on either front or rear rebound or compression have a great impact on the steering and handling of the vehicle. When low speed compression damping was removed the turn-in of the vehicle became better for most of the configurations. As seen in the subjective data this also leads to worse roll control for both small steering inputs in a straight line and during cornering. Since the vehicle is close to the bump stop, even without any suspension deflection, allowing more roll motions makes the tendency for the vehicle to hit the bump stop worse. A problem with engaging the bump stop is the unpredictable handling behavior it gives the vehicle. This was also notices by one of the test drivers, especially with configuration C. The driver experienced that the vehicle became understeered initially when the front bump stop was engaged and oversteered when the rear bump stop was engaged. Another drawback with decreasing the low speed damping was the negative impact on connection to the road. Or in other words, the amount of information the driver receives from the road through the vehicle.

Simulations show that if the rebound damping is reduced at low damper speeds there is no or minor improvement in vehicle response and turn-in. One effect this change probably has on the car is worse roll control due to the reduced roll damping. An interesting observation made from the simulation results was that a lower damping at first damper velocity decreased the time lag for roll angle during the single sinusoidal test. This is probably because the vehicle allows a higher roll rate at the start of the maneuver while the dampers catch the roll motion at higher roll rates. However these results could not be verified in the physical tests since the damper force at the first damper velocity could not be changed, due to limitations in the CCD system, for the real vehicle.

When low speed damping was increased the subjective measurements stated that the vehicle became jerkier, which means that small inputs were felt more by the driver and the comfort was reduced. If the rear compression damping was decreased, corresponding to configuration D, the driver thought that the jerkiness of the vehicle increased in the front. This shows that just decreasing the damping without considering the balance between the front and rear damping does not always improve ride comfort.

Increased compression damping lead to reduced steering response while increased front rebound damping improved the response as seen in the subjective evaluation and Appendix I. Increased damping would probably also give better roll control since the roll damping is increased. Difference in connection to the road was also felt between increasing rear compression, making it worse, and front rebound damping which made it better. The fact that the connection to the road increases as the front rebound damping is increased might be due to reduced body motions in the front. The vehicle will also almost certainly have a lower dynamic ride height compared to the basic setting because of the difference in compression and rebound damping. When the rear compression is increased the vehicle will most likely have a higher dynamic ride height, this might give the driver an impression of being more disconnected to the road. As seen in Figure 14, the front compression damping is significantly higher in low speeds compared to the rear compression, especially in the first point. One of the reasons for this could be to increase the sense of connection to the road. The driver might be more interested in what inputs the front end of the vehicle is experiencing since the steering is located there. If rear inputs would be felt more in the car there is a risk that these inputs would dominate over the inputs from the front end of the vehicle.

The vehicles pitch behaviour during cornering is an important factor that affects the perception of ride. According to the tuning engineers this was referred to as a "flat roll". The pitch motion arises from the difference in net force front and rear from the dampers as well as the longitudinal acceleration. A good balance between compression and rebound, front and rear, is therefore a crucial factor when tuning dampers. Since the vehicle naturally want to pitch forwards without any damping forces, the difference between the rebound and compression force is greater in the rear than at the front of the vehicle. This results in a negative pitching moment forcing the vehicle to pitch backwards.

The optimum level of damping for keeping the vehicle as close to zero pitch as possible varies for different manoeuvres. It mainly depends on the level of lateral acceleration compared to the level of longitudinal acceleration. If for instance a very high lateral acceleration is achieved with small steering input the longitudinal acceleration remains quite small. However, since the lateral acceleration is large the vehicle will experience a high roll rate and the damper forces will create a large negative pitching moment. This would lead to a large negative pitch angle and a high negative pitch rate.

As seen in Figure 22 the increase in rebound damping from first to second damper velocity keeps the roll damping at a constant level. This have the effect of keeping the roll rate down and increases the roll control of the vehicle. The influence of the roll damping is lowered if roll stiffness is increased. For sports cars which generally have higher spring and anti-roll bar stiffness compared to conventional cars the level of critical damping can be lowered in this area without compromising roll control.

According to the simulation results from the primary ride evaluation document Appendix G adding damping to the vehicle reduces the bounce motion of the vehicle. Reducing the level of damping has generally the opposite effect as seen in Figure 27 and in ride evaluation. Some differences can be seen when comparing small and large amplitude roads. When the front rebound damping was reduced by 30% the bounce motion became smaller for the small amplitude road and bigger on the large amplitude road. This is probably since the disturbances on the small amplitude road have a higher frequency compared to the large amplitude road. Less damping allows more wheel motion resulting in less energy transfer to the vehicle body. On the large amplitude road the inputs are closer to the natural frequency of the body and more damping is needed to keep the body motion small. It was possible to see some connections regarding the level pitch motion for added and reduced damping in the evaluation document. Adding or removing damping have the opposite effect on the pitch motion of the vehicle. However, changes in front or rear damping affects the pitch motion differently, probably due to the change in pitch balance. In order to confirm the simulation results for primary ride, objective measurements should be carried out.

7.3 Test methods& material

Of all the tests used to evaluate the model and the influence of damper levels on vehicle handling behaviour the single sinusoidal test was found to be the most useful. The lag times gave a good possibility to study changes in the vehicle yaw-, lateral acceleration- and roll response to a steering input. It was also used to study the pitch balance in transient cornering at different levels of lateral acceleration.

The other handling and steering manoeuvres used did not achieve high enough damper velocities to be useful when studying dampers. This was also a problem when handling and steering requirements was studied since these are based on the same tests. When studying dampers it was found more useful to simulate specific driving events and evaluate the results from these manoeuvres instead. The frequency response test showed only minor or no changes for significant changes to damper settings. This could also be due to the low damper velocities. However when the speed was increased to achieve higher damper velocities slightly bigger but still small changes could be observed. It might help to run the simulation for a longer time in order to have more measurements points at all frequencies.

The primary ride evaluation showed a significant difference between different damper settings and was useful when studying the body control of the vehicle. It can be used to study how well the suspension handles the road input and the low frequency body motions as well as the balance between the front and rear. These tests should also be used for objective measurements and validation of the model since the primary ride is an important aspect of damper tuning. A test for evaluating the wheel hop should also be defined and evaluated in order to find a good balance between wheel hop and body motions.

One aspect which was not tested in the simulations is the roll control at straight ahead driving. This is a characteristic which is commonly used by the engineers as an evaluation of the vehicle performance. Excessive roll at small steering inputs should be avoided to give a more stable feeling. The engineers consider this an important aspect of the vehicle performance and this test could be defined as an objective test for the evaluation of the vehicle.

7.3.1 Continuously controlled damping

The vehicle fitted with Continuously Controlled Damping (CCD) was very useful for making quick changes to the damper setting without having to build new dampers. The damper configurations could be changed instantly by loading new parameters in the electronic control unit. A drawback with the used vehicle was that the possibility to change damper force at the first damper velocity was very limited. This was also the reason why no configurations that included changing the damper force at this velocity was tested. For the rear rebound and front compression damping the desired damping forces could not be achieved at the first damper velocity. This meant that the front compression damping was around 28% lower compared to the basic damper setting and the rear rebound damping was around 20% lower at the first damper speed. However, this was not a problem since the damper levels were easily adjusted in the simulated model.

No verification that the desired damper force versus velocity relations was achieved was made. The dampers should therefore be measured in a damper dyno to verify that the expected damper forces are achieved. However, since the damper forces are controlled by increasing or decreasing the current they should at least change in the right direction. That the damper settings were changed was confirmed both by the objective and subjective test since the vehicle behaviour changed for different configurations.

7.3.2 Vehicle measurements and data

Data from the steering robot was generally of good quality. However the pitch rate and pitch angle from the test did not match the expected values. It was also found that the pitch angle changed sign between some tests depending on if the vehicle turned left or right initially. One possible reason for this measurement error could be that the gyro was not installed completely straight in the vehicle or moved during the tests. The pitch angle measurements could then be influenced by the roll angle. It could also be that the surface at which the tests were performed is not completely flat. New tests should be made where the pitch angle is measured correctly in order to verify the pitch behaviour of the model and the dampers influence on the pitch behaviour. The signal for the pitch rate and pitch angle had a lot of disturbances which is hard to separate from the real signal. As seen Figure 30 the pitch rate has reached a high value even before any steer input is given.

The vehicle used for the tests should also be measured properly to ensure that the input data for the simulation is correct. The free height of suspension travel before the bump stop was engaged was found to be smaller on the tested vehicle compared to simulations. It is also not clear how stiff the springs were that was fitted to the car. This would change the level of critical damping compared to the simulation model. The damper force versus velocity relation was however the same, assuming that the CCD system managed to achieve the correct damper forces. The yaw-, pitch- and roll-inertia used in the simulation was measured for a car with different specifications. These aspects should be taken into account when comparing the objective measurements with the simulation results.

7.4 Connection between objective and subjective evaluation

For configuration A, B and E the turn-in capabilities increased compared to the basic damper settings according to the subjective tests. The only time lags that were shorter for all of these three configurations were the lateral acceleration to steering wheel angle at the first peak, lateral acceleration to yaw at the first peak and the roll angle time lag at the first peak. According to the subjective tests the initial response was worse for configuration D compared to the basic damper settings. The yaw time lag at the first peak was increased for this configuration. The lateral acceleration to steering wheel angle time lag at the first peak was very close to the benchmarked dynamic setting, but the lateral acceleration to yaw rate time lag was reduced for the first peak. The time lag for the roll angle at first peak was reduced for this configuration.

According to these results it seems like the lateral acceleration time lag at the first peak was the most important time lag to study regarding turn-in capabilities. Configuration D did actually have slightly smaller time lag compared to the dynamic setting for the lateral acceleration but the yaw response at the first peak was the worst of all the tested configurations. This indicates that the time lag for yaw rate at the first peak is also important to study. According to the results from configuration E, it seems like a bigger time lag for the yaw rate can still be perceived as better turn-in capabilities if the lateral acceleration time lag is smaller. The time lag for roll angle at the first peak does not seem to be of great importance when the turn-in capabilities are considered. According to the study performed on the topic the phase lag for the roll angle is one of the most important factors to study when considering perceived vehicle agility (da Hora Passos, Fernandes, & de Mello, 2007).A tendency of this correlation was also seen for the vehicle response when comparing subjective and objective measurements.

More tests should be performed both objectively and subjectively to be able to verify or dismiss these correlations. All configurations should be tested subjectively by a number of drivers to get a better database and the single sinus test should be performed in both directions. Objective tests could also be performed to test the roll control in cornering and in a straight line.

7.5 Damper tuning sessions

One of the main problems with using a simulation tool for damper tuning is to capture the correct damper dynamics. According to Daggupati et. al. (Daggupati, Mangaraju, Chavan, & Babu, 2010)two different dampers can have the same force versus velocity relation but have very different behaviour. In the simulation model used all dampers are assumed to have the same dynamic behaviour. This paper is also supported by the fact that during the last stage of the damper tuning sessions no or very small changes can be seen in the damper force versus velocity curves. However, if it would be possible to find the desired damper levels without actually driving the vehicle the time needed for tuning could be used to achieve a good dynamic behaviour of the damper.

Another problem faced when analysing the simulation results was the lack of objective metrics defining a good vehicle. Even if the simulations model would describe the vehicle behaviour fully it would not be possible today to use the results to get the desired characteristics. In order for a simulation tool to be really useful more studies should be conducted on how subjective and objective measurements correlate to each other. One way of achieving this and increase knowledge of how a change in the dampers affects the vehicle would be to start measuring iterations in the tuning process. Today, all dampers are measured during the tuning process and notes are taken on how the new dampers changed the behaviour of the car. If objective tests would be performed it would be possible to study both how the damper affect the vehicle behaviour and the correlation between subjective and objective measurements.

The main problem with this strategy is the limited time available for tuning work. It is therefore not possible to install a steer robot in the vehicle to get good repeatable measurements. One strategy could be to always use the same route on the country roads and keep a constant velocity, this would make it possible to at least analyse the vehicle behaviour on the disturbance roads. It could even be possible to measure the signals from the cars CAN bus to get a better understanding of what has changed since the last damper setting.

The current simulation tool could be used as a guidance to study trends of how the vehicle behaviour would change for a certain damper change. If for instance the tuning engineer has an idea of how they would like to change the damper setting to change the vehicle behaviour, they could use the simulation tool to compare the current settings with the new settings. This would give an understanding if the studied parameter (bounce control, travel balance or etc.) moves in the right direction and what other parameters would change due to the new dampers.

8 Conclusions and recommendations

The simulation model is capable of capturing the general vehicle behaviour. When compared to physical testing results the model shows similar trends as damper levels are changed. Subjective measurements also confirm these trends. Compared to the real vehicle, the simulation results showed a vehicle which was slower in yaw-, roll- and lateral acceleration for the single sinusoidal manoeuvre. The model should therefore not be used to study absolute values for vehicle behaviour but is useful when studying changes in vehicle behaviour.

Pitch balance is mostly affected by compression/rebound damping ratio front and rear and it is important to keep the vehicle as parallel as possible during cornering and on disturbance roads. The damping ratio front and rear also determines the travel balance of the vehicle in bump and roll which is important if good connection to the road is prioritised.

Changed damping at low damper velocities affects the roll behaviour, roll damping and roll control of the vehicle. Roll damping can generally be lowered if the roll stiffness is increased. Low damping can lead to the vehicle hitting its bump stops in roll and bounce which make it hard for the driver to predict the cornering behaviour. It is therefore important to consider the free suspension travel when choosing damper levels. Changed damper levels at low damper velocities also affect the vehicles steering response.

No conclusion whether it is the yaw rate-, lateral acceleration-, or roll angle- time lag that gives a feeling responsive vehicle can be drawn. However, comparison between simulated, objective and subjective results show that lateral acceleration time lag is a good indication of the vehicle turn-in capabilities and the roll angle time lag a good indication for vehicle response.

As more development and tuning work will be carried out with simulations in future, objective measurements are crucial. A suggestion is therefore to increase the amount of data logging during tuning sessions today since objective data is required in order to know how desired vehicle behaviour can be reached. More subjective measurements are also required to be able to create a better correlation between the objective and subjective measurements.

The purpose of the simulation tool is to use it as an aid in future tuning sessions. In order to increase its confidence level the model needs to be further developed and tested. The test vehicle needs to be measured properly in a K&C rig which increases the input validity in the simulation model and a thorough investigation on the vehicles roll centre behaviour needs to be carried out and added to the model. Without adding to much complexity to the model such as, bushings, wheel angles and other properties the model should be able to become a tuning aid capable of capturing the vehicles main changes in behaviour when damper settings are changed. In order to use the simulations tool without any physical vehicle to compare with a definition of how to reach the desired vehicle behaviour objectively is required.

9 Bibliography

Albinsson, A., Börjesson, L., Koitrand, S., Maisonneuve, M., Rehnberg, S., Vijayakumar, E., et al. (2012). *Design Study of a solar challenge vehicle.* Gothenburg.

Causemann, P. (2001). *Automotive shock absorbers.* Augsburg: Verlag moderne industrie.

da Hora Passos, L., Fernandes, C. G., & de Mello, K. P. (2007). *A New Approach to Improve Steering Response Development Using Bode Diagrams.* São Paulo: SAE.

Daggupati, G., Mangaraju, V., Chavan, C., & Babu, R. (2010). *A New Test Method to Characterize the Behavior of Hydraulic Damper.* SAE International.

Den Hartog, J. (1985). *Mechanical Vibrations.* New York: Dover Publications, Inc.

Dixon, J. C. (1991). *Tyres, Suspension and Handling.* Cambridge: Press Syndicate of the University of Cambridge.

Dixon, J. C. (2007). *The Shock Absorber Handbook.* Professional Engineering Publishing Ltd & John Wiley and Sons Ltd.

Jazar, R. N. (2008). *Vehicle Dynamics Theory and Application.* New York: Springer Science +Business Media, LLC.

Lidberg, M. (2012, March 15). LECTURE NOTES TME102 VEHICLE DYNAMICS, ADVANCED. *Lecture 2*. Gothenburg, Sweden.

Mohammadi, F. (2012). *Tire Characteristics Sensitivity Study.* Gothenburg, Sweden: Department of Applied Mechanics.

Pacejka, H. B. (2006). *Tyre and Vehicle Dynamics.* Oxford: Elsevier.

Appendix A

Table 2, required input parameters for the simulation model

Required input	*Unit*	*Variable name in Matlab*	*Used for*
Mass	*kg*	*m*	*All simulations*
Roll, yaw and pitch inertia	*kgm^2*	*Ixxs,Izz,Iyy*	*All simulations*
Wheelbase	*m*	*L*	*All simulations*
Distance from CoG to Front axle	*m*	*lf*	*All simulations*
Distance from CoG to Rear axle	*m*	*lr*	*All simulations*
Mass on front axle	*kg*	*mf*	*All simulations*
Mass on rear axle	*kg*	*mr*	*All simulations*
Front roll centre height [table]	*m*	*hrf*	*All simulations*
Rear roll centre height [table]	*m*	*hrr*	*All simulations*
Roll angle for roll centre height table	*deg*	*rollangle_RCH*	*All simulations*
CoG height	*m*	*h*	*All simulations*
Unsprung mass for one front wheel	*kg*	*m1*	*All simulations*
Unsprung mass for one rear wheel	*kg*	*m2*	*All simulations*
Wheel rate front	*N/m*	*kf*	*All simulations*
Wheel rate rear	*N/m*	*kr*	*All simulations*
Tire stiffness front	*N/m*	*ktf*	*All simulations*
Tire stiffness rear	*N/m*	*ktr*	*All simulations*
Anti-rollbar stiffness front	*Nm/rad*	*Krf*	*All simulations*
Anti-rollbar stiffness rear	*Nm/rad*	*Krr*	*All simulations*
Damper efficiency front	*-*	*C_eff_F*	*All simulations*
Damper efficiency rear	*-*	*C_eff_R*	*All simulations*
Damper speed front from measurements	*m/s*	*VF*	*All simulations*
Damper speed rear from measurements	*m/s*	*VR*	*All simulations*
King pin inclination angle	*rad*	*KPIL*	*Only swept steer test*
Caster angle	*rad*	*Ca*	*Only swept steer test*
Kingpin offset @ground	*m*	*ba*	*Only swept steer test*
Mechanical trail	*m*	*c_offset*	*Only swept steer test*
Kingping offset @wheel centre	*m*	*KPI_offset_centre*	*Only swept steer test*
Bumpstop data			
Front			
Bump contact	*m*	*Bumpstop.Bump_contact_F*	*All simulations*
Rebound contact	*m*	*Bumpstop.Rebound_contact_F*	*All simulations*
Deformation of bumpstop for table	*m*	*Bumpstop.Def_bump_F*	*All simulations*
Deformation of reboundstop for table	*m*	*Bumpstop.Def_rebound_F*	*All simulations*
Force from bumpstop for table	*N*	*Bumpstop.Force_bump_F*	*All simulations*
Force from rebound stop for table	*N*	*Bumpstop.Force_rebound_F*	*All simulations*
Rear			
Bump contact	*m*	*Bumpstop.Bump_contact_R*	*All simulations*
Rebound contact	*m*	*Bumpstop.Rebound_contact_R*	*All simulations*
Deformation of bumpstop for table	*m*	*Bumpstop.Def_bump_R*	*All simulations*
Deformation of reboundstop for table	*m*	*Bumpstop.Def_rebound_R*	*All simulations*
Force from bumpstop for table	*N*	*Bumpstop.Force_bump_R*	*All simulations*
Force from rebound stop for table	*N*	*Bumpstop.Force_rebound_R*	*All simulations*

Appendix B

Instruction Manual, How to run a Simulation

A Graphic User Interface (GUI) is primarily developed to make it easier to visualize changes made to the level of damping. The GUI makes it possible to use the program even if the user knowledge in Matlab is limited.

The damping levels are expressed as the damping ratio in pure heave for rebound, compression and front/rear separately as a function of vertical wheel speed. Four editable tables with seven points are used for each of these curves, see Figure 32. The GUI allows the user to choose which tests that should be simulated from a number of standard tests. This means that the user can change the damper settings and select which test that they want to run without editing any Matlab script.

The level of critical damping is calculated as:

$$\xi = \frac{C}{2\sqrt{km}}$$

$$C = \frac{D_{eff}^2\left(F + \frac{F_{friction}}{D_{eff}}\right)}{V}$$

$$k = k_{spring} + k_{parasite}$$

A save/load function has also been included in the GUI which allows the user to save the current damper settings in Excel. This Excel file contains both the damping ratios which are loaded into the GUI and force vs. damper speed relation usually given by the supplier to Volvo.

Since the plot in the GUI only describes the heave damping, two pushbuttons plotting the roll damping and the pitch damping based on the damper forces are therefore included. A pushbutton which plots the force-displacement curves for the front and rear damper, usually referred to as egg plots, is also included.

Before running any simulations the user has to start the GUI Matlab script. After choosing desired events the user presses the "Start Simulation button" which runs the simulation. The results are automatically plotted from a Matlab script.

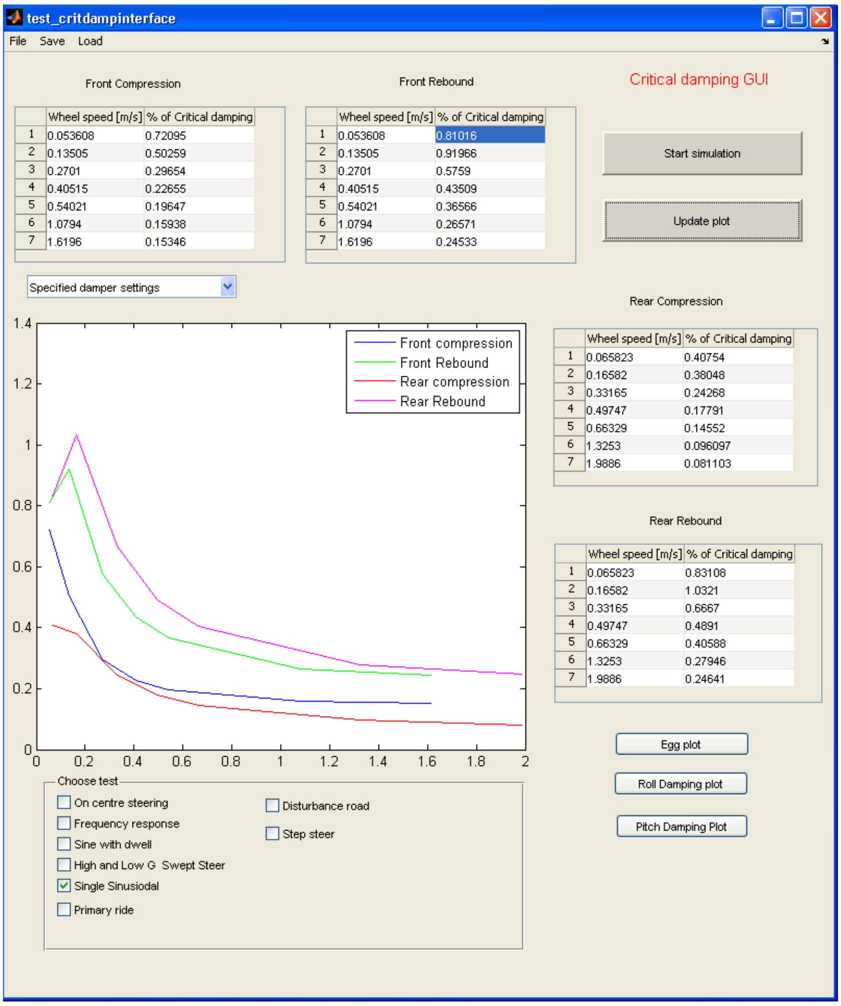

Figure 32, Graphical user interface

Appendix C

- Primary ride
    - Bounce metric
    - Roll metric
    - Pitch metric
    - Abruptness index
    - Head toss index
    - Balance index
- Handling Evaluation Sheet
    - Understeer gradient
    - Max lateral adhesion
    - Total Roll gradient
    - Roll rate at 1 Hz
    - Roll gradient at 1Hz
    - Total pitch gradient
- Steering Evaluation Sheet
    - SWA at 0.05g
    - Average on centre yaw gain
    - Overall steering sensitivity g/100degSWA at two velocities
    - Gain linearity
    - Roll rate gradient at 1 Hz
    - SWA at 1.3 Nm
    - Off centre yaw gain
    - Understeer gradient
    - Yaw linearity
    - Total roll gradient
    - SWT/ SWA Nm/100degSWA
    - SWT/AY Nm/g
    - Torque gradient at 0.3-0.5 g
    - Torque at 0.3 g
    - On centre yaw gain 50 km/h
    - Overall steering sensitivity
    - Sine time lags

Appendix D

Table 3, Subjective measurement evaluation sheet

Subjective measurements Thesis	Vehicle	Standard Dynamic	Config A	Config B	Config C	Config D	Config E
RIDE							
PRIMARY RIDE							
Primary Ride Control							
Primary Ride Comfort							
SECONDARY RIDE							
Choppiness							
Shake							
Rolling Feel							
HANDLING							
CORNERING STABILITY							
Over / understeer steer							
Connection to the Road							
Roll Control							
STEERING							
STRAIGHT AHEAD CONTROLLABILITY							
Response							
Roll Control							
CORNERING CONTROLLABILITY							
Response							
Roll Control							

Appendix E

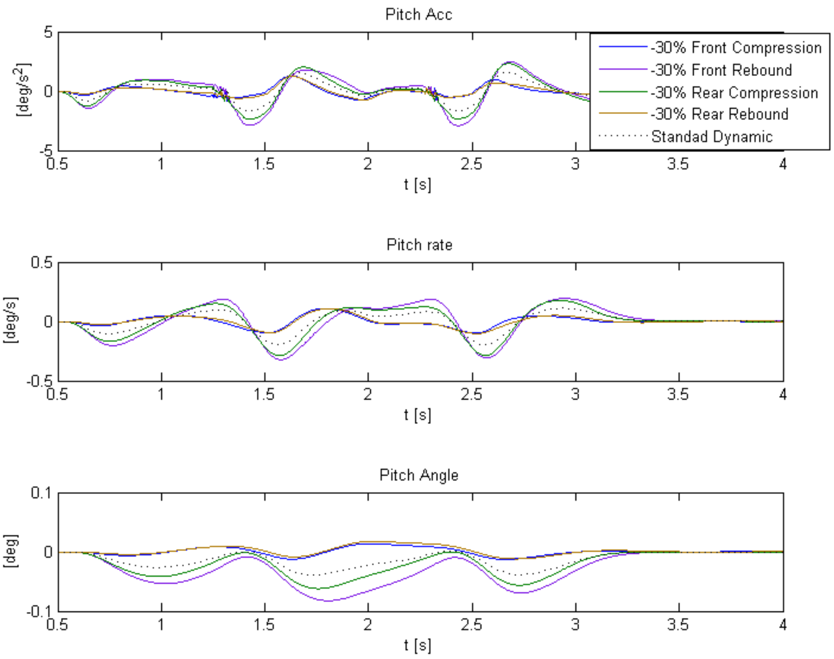

Figure 33, Pitch behaviour during single sinus event at 4 $\mathrm{m/s^2}$, 30% reduced damping

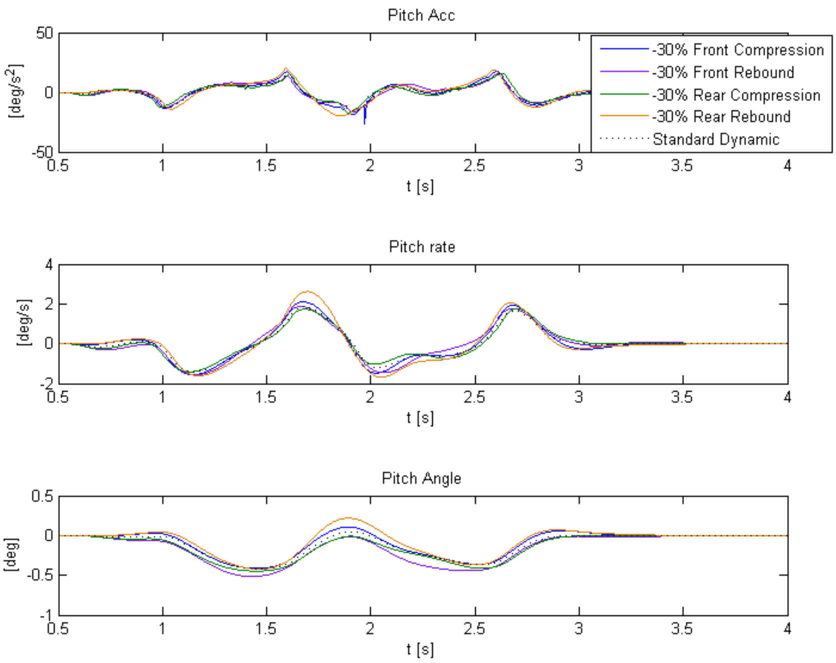

Figure 34, Pitch behaviour during single sinus event at 7.5 $\mathrm{m/s^2}$, 30% reduced damping

Appendix F

Table 4, Time lag based on simulated results

Lat Acc. Time Lag [ms]				Lat Acc. - Yaw Rate Time Lag [ms]				Roll Angle Time Lag [ms]		
1st		2nd		1st		2nd		1st		2
160	Diff.	209	Diff.	78	Diff.	152	Diff.	314	Diff.	
160	0,00%	203	-2,87%	82	5,13%	146	-3,95%	305	-2,87%	
152	-5,00%	210	0,48%	71	-8,97%	153	0,66%	308	-1,91%	
165	3,13%	206	-1,44%	84	7,69%	146	-3,95%	310	-1,27%	
159	-0,62%	206	-1,44%	79	1,28%	151	-0,66%	309	-1,59%	
160	0,00%	208	-0,48%	78	0,00%	152	0,00%	313	-0,32%	
152	-5,00%	212	1,44%	70	-10,26%	154	1,32%	309	-1,59%	
149	-6,88%	212	1,44%	64	-17,95%	158	3,95%	317	0,96%	
156	-2,50%	212	1,44%	71	-8,97%	152	0,00%	318	1,27%	
156	-2,50%	210	0,48%	73	-6,41%	152	0,00%	314	0,00%	
161	0,63%	211	0,96%	78	0,00%	155	1,97%	317	0,96%	
130	-18,75%	217	3,83%	60	-23,08%	162	6,58%	307	-2,23%	
166	3,75%	203	-2,87%	81	3,85%	146	-3,95%	308	-1,91%	
146	-8,75%	208	-0,48%	67	-14,10%	151	-0,66%	310	-1,27%	
155	-3,13%	217	3,83%	82	5,13%	162	6,58%	310	-1,27%	
154	-3,75%	206	-1,44%	72	-7,69%	146	-3,95%	306	-2,55%	
170	6,25%	207	-0,96%	84	7,69%	153	0,66%	316	0,64%	
164	2,50%	210	0,48%	79	1,28%	153	0,66%	318	1,27%	
162	1,25%	206	-1,44%	76	-2,56%	151	-0,66%	319	1,59%	
159	-0,62%	211	0,96%	76	-2,56%	158	3,95%	320	1,91%	
154	-3,75%	214	2,39%	74	-5,13%	153	0,66%	312	-0,64%	
160	0,00%	225	7,66%	78	0,00%	166	9,21%	314	0,00%	
160	0,00%	209	0,00%	78	0,00%	151	-0,66%	314	0,00%	
160	0,00%	216	3,35%	78	0,00%	161	5,92%	314	0,00%	
160	0,00%	214	2,39%	78	0,00%	154	1,32%	314	0,00%	
160	0,00%	209	0,00%	78	0,00%	151	-0,66%	314	0,00%	
160	0,00%	208	-0,48%	78	0,00%	152	0,00%	314	0,00%	
160	0,00%	200	-4,31%	78	0,00%	138	-9,21%	314	0,00%	
160	0,00%	204	-2,39%	78	0,00%	149	-1,97%	314	0,00%	
160	0,00%	209	0,00%	78	0,00%	153	0,66%	314	0,00%	
160	0,00%	210	0,48%	78	0,00%	152	0,00%	314	0,00%	

Appendix G

Table 5, Requirement evaluation sheet added damping

		Requirement Evaluation Sheet		Standard dynamic	Dynamic chassi +30% Front Compression		Dynamic chassi +30% Front Rebound		Dynamic chassi +30% Rear Compression		Dynamic chassi +30% Rear Rebound	
Small Amplitude Road	control	Bounce Motion	mm	2,662	2,452	-7,91%	2,618	-1,66%	2,614	-1,83%	2,688	0,97%
		Pitch Motion	°	0,082	0,095	15,63%	0,079	-3,91%	0,073	-10,50%	0,088	7,69%
		Roll Motion	°	0,063	0,062	-2,22%	0,063	-0,79%	0,063	-0,63%	0,063	0,32%
	Comfort	Abruptness Index	-	2,059	2,033	-1,29%	2,058	-0,05%	2,061	0,10%	2,073	0,65%
		Head Toss Index	-	0,360	0,360	0,00%	0,360	0,00%	0,360	0,00%	0,360	0,00%
Large Amplitude Road	Control	Bounce Motion	mm	19,896	18,842	-5,30%	18,575	-6,64%	19,536	-1,81%	19,478	-2,10%
		Pitch Motion	°	0,307	0,279	-9,03%	0,264	-13,99%	0,314	2,35%	0,329	7,44%
		Roll Motion	°	0,260	0,248	-4,62%	0,254	-2,16%	0,253	-2,58%	0,258	-0,54%
	Comfort	Abruptness Index	-	3,491	3,389	-2,90%	3,468	-0,66%	3,472	-0,55%	3,499	0,24%
		Head Toss Index	-	0,840	0,830	-1,19%	0,840	0,00%	0,840	0,00%	0,840	0,00%

Table 6, Requirement evaluation sheet reduced damping

		Requirement Evaluation Sheet		FNE495 standard dynamic chassi New Roll Centre Behaviour	Dynamic chassi -30% Front Compression		Dynamic chassi -30% Front Rebound		Dynamic chassi -30% Rear Compression		Dynamic chassi -30% Rear Rebound	
Small Amplitude Road	control	Bounce Motion	mm	2,662	2,918	9,62%	2,733	2,66%	2,730	2,54%	2,644	-0,69%
		Pitch Motion	°	0,082	0,072	-11,60%	0,088	7,45%	0,095	15,63%	0,079	-3,66%
		Roll Motion	°	0,063	0,065	3,17%	0,064	1,43%	0,064	0,95%	0,063	-0,48%
	Comfort	Abruptness Index	-	2,059	2,092	1,61%	2,066	0,32%	2,061	0,06%	2,051	-0,39%
		Head Toss Index	-	0,360	0,360	0,00%	0,360	0,00%	0,360	0,00%	0,360	0,00%
Large Amplitude Road	Control	Bounce Motion	mm	19,896	21,123	6,17%	21,780	9,47%	20,313	2,09%	20,562	3,35%
		Pitch Motion	°	0,307	0,357	16,54%	0,376	22,70%	0,303	-1,21%	0,286	-6,69%
		Roll Motion	°	0,260	0,278	7,04%	0,269	3,62%	0,267	2,73%	0,260	-0,08%
	Comfort	Abruptness Index	-	3,491	3,621	3,72%	3,555	1,83%	3,526	1,02%	3,523	0,92%
		Head Toss Index	-	0,840	0,850	1,19%	0,840	0,00%	0,840	0,00%	0,830	-1,19%

Appendix H

Table 7, Time lag based on objective results

	Lat Acc. Time Lag [ms]			
	1st		2nd	
	132	Diff.	98	Di
3,56%	108	-17,72%	133	
5,95%	110	-16,46%	110	
5,78%	107	-18,99%	100	
1,69%	130	-1,27%	112	
3,47%	117	-11,39%	95	

Table 8, Time lag based on simulated results

	Lat Acc. Time Lag [ms]			
	1st		2nd	
	165	Diff.	209	Diff
33%	131	-20,61%	217	
00%	143	-13,33%	207	-
33%	151	-8,48%	218	
33%	164	-0,61%	205	-
00%	162	-1,82%	209	

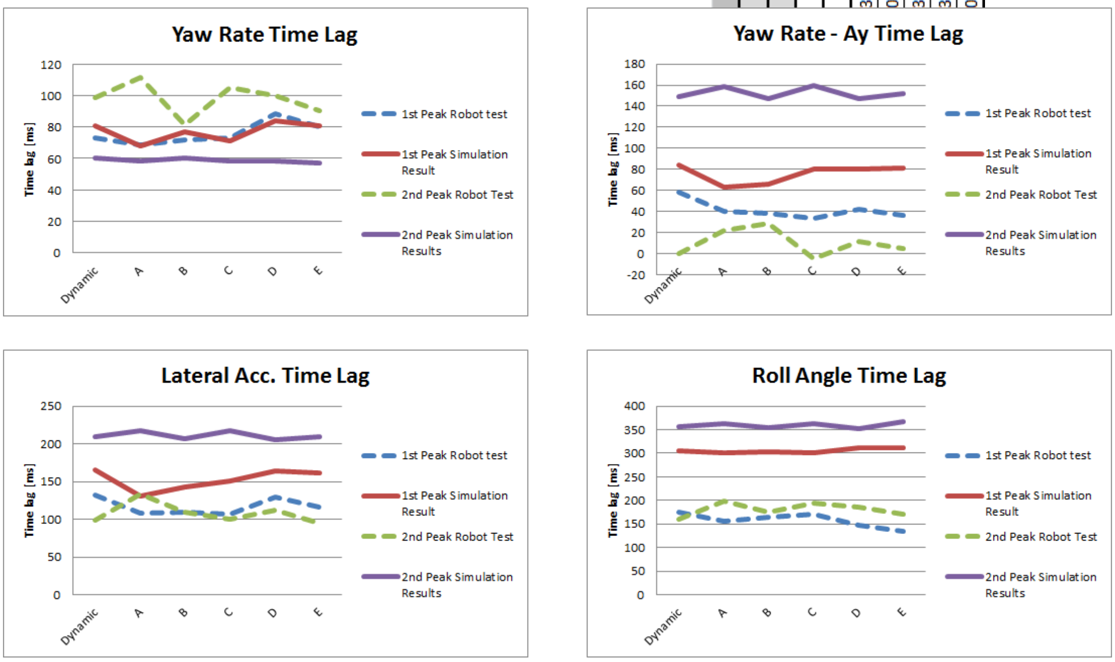

Figure 35, Trend lines based on the time lags for different configurations A to E

Appendix I

Table 9, Subjective results

Subjective measurements thesis 13w20	Vehicle	Standard Dynamic	Config A	Config B	Config C	Config D	Config E
RIDE							
PRIMARY RIDE							
Primary Ride Control							
Primary Ride Comfort							
SECONDARY RIDE							
Choppiness							
Shake							
Rolling Feel							
HANDLING							
CORNERING STABILITY							
Over / understeer steer							
Connection to the Road							
Roll Control							
STEERING							
STRAIGHT AHEAD CONTROLLABILITY							
Response							
Roll Control							
CORNERING CONTROLLABILITY							
Response							
Roll Control							