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Development of methods for objectively quantifying performance of active suspension systems

Master's thesis in Automotive Engineering

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Abstract

Ride is one of the attributes of vital importance in modern passenger vehicles. This is closely connected to the suspension system of the vehicle, which often consists of passive springs and dampers. For these passive systems there are methods and metrics to evaluate their performance but for semi- and fully active suspension systems there are not well-defined methods and metrics. The objective of this thesis is therefore to develop a method to objectively quantify the performance of semi- and fully active suspension systems. The goal is to answer the question: *"What is a good semi-active or fully active suspension system?"*.
The evaluation of the suspension systems is done in three ways; using simulations in IPG CarMaker, objective testing of physical cars and through subjective assessments of the same cars. The results are then processed using MATLAB to find any desired metrics. Here a correlation study is also carried out to see what objective measurements correlate best to the subjective scores. From the results, it is evident that the road used for small amplitude primary ride and flat surface secondary ride in simulations does not produce results comparable to the physical tests. For the large amplitude road some correlations are found, for instance between the objective heave movement and subjective comfort rating. For secondary ride on rough surface the objective pitch RMS correlates to subjective choppiness rating. These results are not certain as the standard deviation in them is significant and data for more cars and drivers would be required to increase certainty. However, this shows that it is possible to find objective metrics to describe the performance of a suspension system, including active ones, and thereby estimate a subjective rating from a value of the objective metric.
To conclude; the method proved functional in the sense that it helps translate subjective assessments of a vehicle's ride quality with semi- and fully active suspension into objective, quantifiable metrics. While having testing procedures stricter and defined in greater detail could have produced stronger correlations, as well as having larger data sets in the form of number of test cars, test drivers and procedures, it can still be considered a proof of concept. Moreover, it forms a good basis and a useful tool for further work in creating a fully effective ride evaluation method.

Keywords: Active suspension system, ride, ride evaluation, vehicle dynamics.

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Nomenclature

Abbreviations

BS	British Standards
CAE	Computer-Aided Engineering
COG	Center Of Gravity
BS	British Standards
COG	Center Of Gravity
DOF	Degrees Of Freedom
FLC	Fuzzy Logic Control
GUI	Graphical User Interface
ISO	International Organization for Standardization
LQ	Linear Quadratic
LQG	Linear Quadratic Gaussian
LQR	Linear Quadratic Regulator
MBS	Multibody System
MF	Magic Formula
MPC	Model Predictive Control
PSD	Power Spectral Density
RMQ	Root Mean Quad
RMS	Root Mean Square
VDV	Vibration Dose Value
VVE	Virtual Vehicle Environment

Variables

ϕ	Roll angle
θ	Pitch angle
a_{tm}	Top-mount acceleration
c	Damping coefficient
F	Force
k	Spring stiffness
m	Mass
t	Time
T	Track width
v_{tm}	Top-mount velocity
W	Wheelbase
$x_{travelled}$	Distance travelled
z	z-position at wheel location
z_{def}	Deflection in z of wheel centre
z_{driver}	z-measure at drivers position
z_{heave}	z-measure at centre of vehicle
z_{tm}	Top-mount displacement

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1
Introduction

1.1 Background

Ride comfort in a modern car is an attribute of vital importance. This attribute is a measure of how well a vehicle absorbs road undulations, bumps, impacts and vibrations and ultimately how comfortable the occupants of the cars are while driving. There needs to be a balance between the comfort of the ride and the handling capabilities of the car, since these two attributes inevitably compete with each other. The handling characteristics aim to maintain as much control of the vehicle as possible. This by ensuring that the wheels maintain as much contact with the road as possible, producing the maximum amount of grip and thus following the road accurately and with confidence. On the contrary, ride quality is often associated with a feeling of isolation from the inputs and disturbances of the road to minimize fatigue of the occupants.

This balance has long been attempted to attain with passive springs and dampers. For these there are good metrics defined to be able to determine the performance of the suspension system in a car. There are also semi-active systems which use the dampers to facilitate a plush secondary ride while maintaining firm body control and capable handling. For these systems the "skyhook" algorithm, which aims to minimize the body heave, is most commonly used. Next are the fully active systems which, instead of traditional dampers, use actuators that can be independently controlled to exert a force on the suspension. This allows for the wheels to more accurately follow the road and by knowing what comes ahead through the addition of a "preview" system they can act on what is to come. This comes at a great complexity but with many great possibilities. For these newer systems, however, there are not well defined metrics for their performance and need to be developed to allow for objective evaluation of the performance of these systems. Since passive suspensions have been used and developed for a very long time there are multiple methods and metrics available for ride evaluation. When applying these to semi- or fully active suspension systems the results may not be as good or representative as for the passive ones. For instance one of the most well used metrics for passive suspensions is the root mean square (RMS) value of the vertical acceleration. But since semi- and fully active suspensions can cause significant jerk, which is the derivative of the acceleration, the RMS can give a wrongful indication. This because a high jerk does not have to have a significant impact on the RMS but will have a considerable impact on the perceived ride comfort. From this example it can be understood that suspension systems which uses active components needs to be evaluated differently compared to the method used for conventional passive suspension systems.

1.2 Objectives

This thesis aims to develop a method to allow for objectively quantifying the performance of semi-active and fully active suspension systems. Therefore it should answer the question: *"What is a good semi-active or fully active suspension system?"* This means that the performance of a skyhook damper suspension system can be used as a benchmark. From this the additional value and benefits of using preview information or a fully active suspension can be explored. Therefore a study on correlation between subjective and objective ride evaluation is required to understand present and desired characteristics of the vehicle's ride behaviour. While there are already existing studies on subjective and objective evaluation of steering and handling characteristics, there was less to be found on ride comfort. Additionally, as there are numerous standardized methods for objectively quantifying ride quality for vehicles with passive suspension systems, few exist for active systems. The results from this should later be used for aiding the decision making for development of future suspension systems.

1.3 Delimitations

There is no focus on creating or developing models and controllers for skyhook dampers or model predictive control, but rather utilizing existing ones. Physical suspension components will neither be designed nor manufactured for this project, instead the properties and attributes of existing components will be explored. The main focus will be on semi-active suspension systems with the possibility of extending to fully active systems for further evaluation.

2

Theory

2.1 Suspension systems

2.1.1 Basic theory

A conventional passive suspension system of today typically consists of a spring and damper, connecting the unsprung mass (wheel- and tyre assembly, wheel hub, wheel bearing, brake disc, brake caliper, etc.) to the sprung mass of the vehicle (body, chassis, powertrain, interior, etc.). This connection can be a dependent system and contain a solid axle connecting two wheels or an independent suspension consisting of multiple links or control arms. According to Gillespie the suspension system's main purpose is to:

- Provide vertical compliance so the wheels can follow the uneven road, isolating the chassis from roughness in the road.
- Maintain the wheels in the proper steer and camber attitudes to the road surface.
- React to the control forces produced by the tires longitudinal (acceleration and braking) forces, lateral (cornering) forces, and braking and driving torques.
- Resist roll of the chassis.
- Keep the tires in contact with the road with minimal load variations [1, p. 237].

The springs of a passive system can store energy momentarily while the dampers dissipate energy, meaning that no external energy is delivered to the system. However, passive systems come with the drawback of having fixed spring- and damping rates, resulting in a persistent compromise between ride and handling of the vehicle. They are, nonetheless, in most cases less complex, less expensive and more reliable than their semi- and fully active counterparts [2].

The behaviour of a passive damper can be linear, as shown in figure 2.1, or non-linear and with breakpoints but for any given velocity they can only produce one specific force. This due to the fixed damping coefficients.

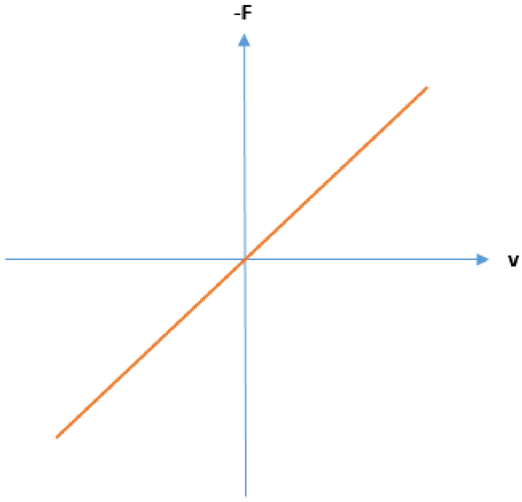

Figure 2.1: Characteristics of linear passive damper.

2.1.2 Semi-active

Semi-Active suspension systems combine elements of both passive and fully active systems. They consist of springs and dampers, as the passive systems, but have the ability to control the attributes and the properties of the damper. Generally, semi-active suspension systems can be categorized by various functions and properties: The first one, **Slow-active** suspension (often called adaptive suspension), is reactive and responds to the rolling, pitching and heaving motions of the vehicle body as well as different road surfaces and obstacles by altering the damping rate. **Low-bandwidth** is the second type where the damping is continuously adjusted to control the low-frequency motions of the sprung mass (1-3 Hz). Finally, **High-bandwidth** systems have the ability to control the low-frequency motions as the low-bandwidth system with the added benefit of controlling the high-frequency motions of the axle (10-15 Hz) [1].

As explained by Guglielmino *et al.* [3] the benefit of a semi-active suspension over a conventional passive suspension is the ability to adjust the damping in real time. This is, however, without adding much energy to the system as in the case of a fully active system. This means that the dampers no longer have a fixed damping coefficient as for the passive ones but instead a variable one. With a variable damping coefficient a given velocity no longer corresponds to a specific force. Instead, a range of forces can be achieved. This results in the characteristics of the damper being described as a an area between two curves, as shown in figure 2.2, and not as a single curve. These adjustments and modulations depend on the control method of the system and how it controls the damping forces required at a given time. The effect of this can be reduced accelerations of the chassis and dynamic tyre forces, resulting in greater ride comfort and improved handling respectively.

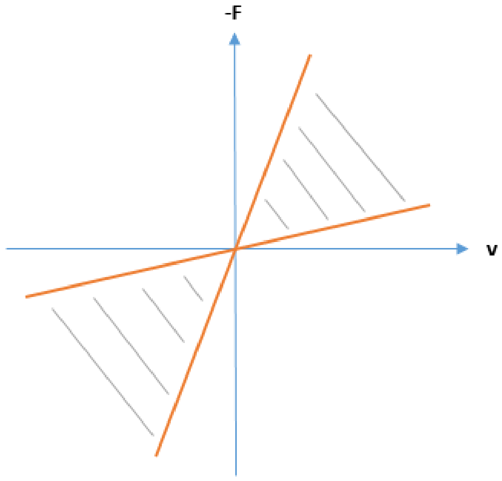

Figure 2.2: Characteristics of semi-active damper.

2.1.3 Fully active

Instead of using conventional springs and dampers a fully active suspension system uses actuators in order to fulfill the demands set on the suspension [1]. This adds a significant amount of complexity and cost to the system as well as additional weight. The active systems also require an external power source to enable the suspension to add energy into the system when desired. This increases the overall energy consumption of the vehicle resulting in an increased fuel consumption. The added energy, however, allows the system to generate the desired forces at any moment and thereby offers a great amount of control for the vehicle ride and behavior. This means that there is no traditional damper characteristics curve as the system can generate any force for any given velocity, illustrated in figure 2.3. This also means that criteria set on handling and ride can be achieved independently which, in turn, allows for less compromises [2]. Another important factor with these systems is the need for some form of fail-safe or redundancy in case of a malfunction.

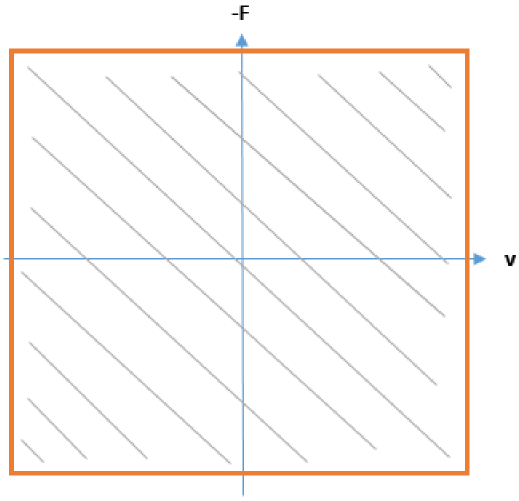

Figure 2.3: Characteristics of fully active damper.

2.2 Vehicle models

There are many different types of vehicle suspension models used in vehicle dynamics theory. Which one to use depends on the important properties and level of detail needed for the specific task.

2.2.1 Quarter car

A quarter car model is a simplified one or two degree of freedom (DOF) model, effective in explaining basic vertical dynamics of a vehicle. As shown in figure 2.4, it is a mass-spring-damper system representing one of the four corners of a vehicle. The model is comprised of a sprung mass m_s and a unsprung mass m_u with a set of a spring k_s and a damper c_s connecting the two, and another set, k_u & c_u, connecting the unsprung mass and the ground. This gives the model two DOFs, x_s and x_u corresponding to the vertical movement of the two masses.

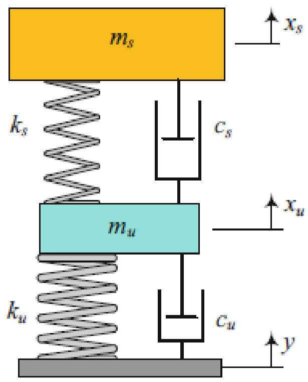

Figure 2.4: Quarter car model [4].

2.2.2 Half car

By extending the quarter car model to a half car model additional DOFs are introduced. Now with two unsprung masses, m_1 & m_2 and a sprung mass, m, connecting them, shown in figure 2.5. The half car model can either translate pitching motions of the vehicle, if the unsprung masses represent the front and rear of the vehicle, or roll motions, if they instead represent the left and right sides of the vehicle through the angle θ. Both variations can still translate the heave motions of the vehicle. This gives the model a total of 4 DOFs, the three first being x, x_1 and x_2, corresponding to the vertical movement for each mass. This is accompanied with the angle θ, which as mention can represent either the roll or pitch angle of the car.

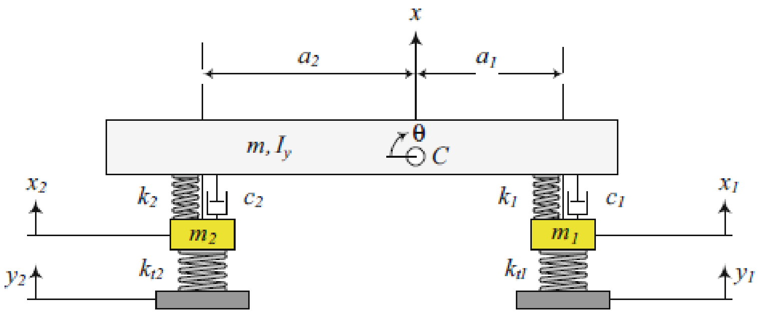

Figure 2.5: Half car model [4].

2.2.3 Full car

By further extending the half car model to include all four corners of the vehicle a full car model is created, as shown in figure 2.6. This has a sprung mass m along with four unsprung masses m_1, m_2, m_3 & m_4 which represents each corner of the vehicle. The model also includes the roll angle ϕ and the pitch angle θ. This expands the number of DOFs to seven, which consists of the heave movement x, roll angle ϕ and pitch angle θ of the sprung mass. These are accompanied by x_1, x_2, x_3 and x_4 which are the vertical movements of the corner masses.

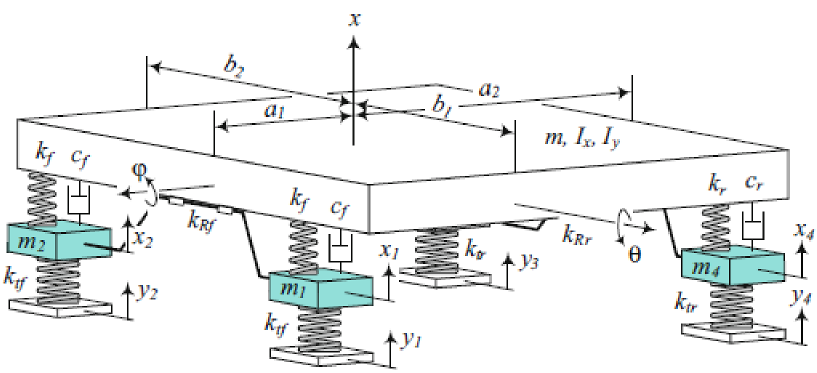

Figure 2.6: Full car model [4].

2.2.4 Multibody system

In order to capture more DOFs than the full car model and create more advance models which often are used for simulations, the multibody system (MBS) approach can be used. In a MBS the vehicle is divided first into subsystems (Steering system, suspension, etc.) and then further into component level where each component is accompanied by its own DOFs. For example can the chassis be represented by one single rigid body and a suspension system can consist of multiple rigid bodies, links and force elements [5]. These models allow for a high flexibility but the more that is modeled and included the more complex and computationally heavy it becomes.

2.2.5 IPG CarMaker

CarMaker uses a Virtual Vehicle Environment (VVE) where a representation of a vehicle is computer modeled to simulate the attributes and characteristics of an actual vehicle. This vehicle model is built up of mathematical models and formulas containing the equations of motion and the kinematics of the system. In regard to axis systems, kinematics, forces and moments, CarMaker adheres to conventions of the modified ISO 8855 2011 standard. CarMaker has the ability to simulate all different parts of a vehicle such as powertrains, complex tyre models, braking systems, control and assistance systems and chassis and suspension systems.

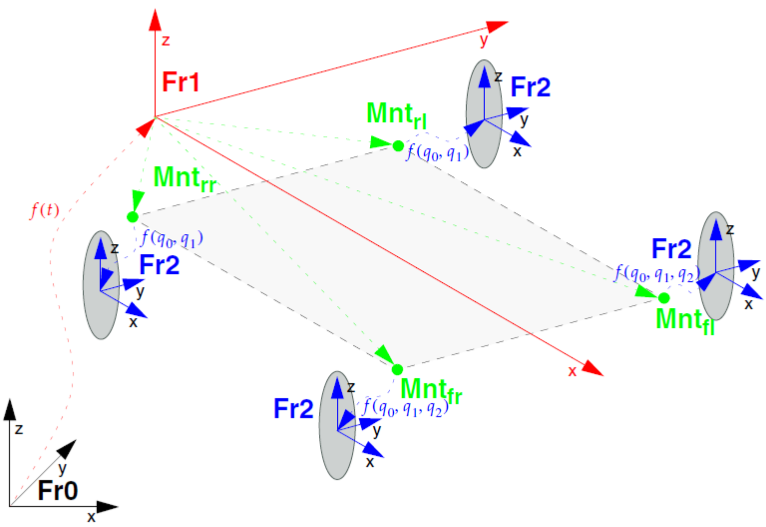

Figure 2.7: CarMaker coordinate systems [6]

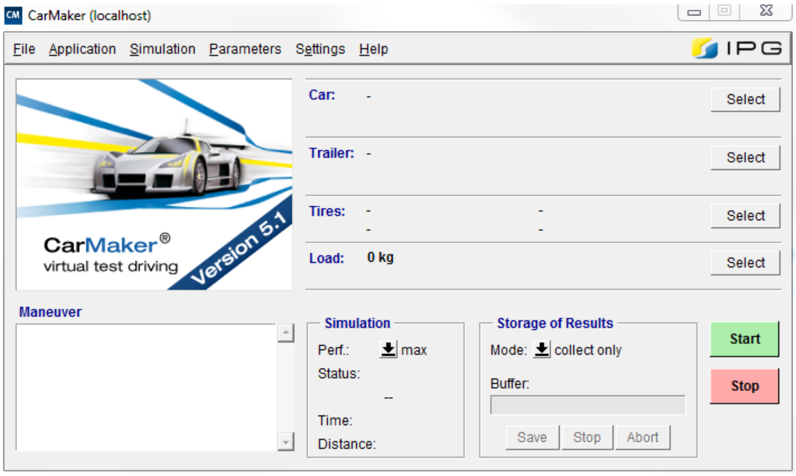

Figure 2.8: Screenshot of CarMaker GUI.

The vehicle simulated in CarMaker is represented by a multibody system where some of the main bodies are the actual body of the vehicle as well as its wheels. These bodies are placed in the virtual environments inertial axis frame, called frame zero and denoted Fr0. The vehicle travelling in Fr0 has its own axis system which moves with the vehicle, called Fr1 and additional to this each wheel carrier has a axis system attached to it, called Fr2, as shown in figure 2.7. This system of bodies is then described using differential and algebraic equations. The differential equations are the equations of motion for the generalized coordinates of the system and are displayed in equation 2.1:

$$\boldsymbol{A}\ddot{\boldsymbol{q}} = \boldsymbol{B}(\dot{\boldsymbol{q}}, \boldsymbol{q}) \tag{2.1}$$

where $\boldsymbol{A}$ is the mass matrix, $\ddot{\boldsymbol{q}}$ the second order derivatives of the generalized coordinates and $\boldsymbol{B}$ the projected forces acting on the free modes of motion. The calculation of $\boldsymbol{B}(\dot{\boldsymbol{q}}, \boldsymbol{q})$ is done through the use of the algebraic equations which for instance contains gyroscopic moments and spring/damper forces [6].

2.3 Ride

2.3.1 Theory

The ride of a vehicle is related to how the vehicle moves and reacts to the road and the road roughness. This then translates into vibrations in the vehicle and further as vibrations experienced by the occupants of the vehicle. Therefore the human response to vibrations is very important when evaluating the comfort of a vehicle. Ride comfort is hard to define but the overall comfort and well-being of the occupants of the vehicle is known as the ride comfort [7]. This is known to have a considerable impact on the quality impression of the vehicle [1]. How the motions are affecting the vehicle occupants is difficult to determine since it depends on a variety of factors, such as the characteristics of the motion, the person and other aspects of the environment [8]. This makes it difficult to objectively measure the level of comfort experienced by the occupants. Ride comfort can be evaluated for instance through measuring different accelerations, but since it is directly related to the occupants it is ultimately a subjective measure. Despite comfort being subjective the human body has different natural frequencies, with some variation between different people. As mentioned by Heissing and Ersoy [7] some of the more important frequencies are the frequencies around 0.5-0.75 Hz which can cause motion sickness, frequencies between 3 and 11 Hz which affects the torso and the head and neck is sensitive to frequencies of approximately 25 Hz. For passenger cars vertical vibration dominates but there are also horizontal components, which are more pronounced for taller vehicles [7].

The main source of vibrations is the road but there are also other sources such as the engine and aerodynamics [3]. A common way of expressing these vibration environments, such as a road, is to use the power spectral density (PSD) function [8]. These vibrations vary significantly in both amplitude and frequency and therefore ride can be divided based on the frequency of the disturbance. Frequencies between 0 and 3 or 4 Hz are normally considered to be primary ride and are related to the rigid body motion and general suspension motion of the vehicle. Frequencies in the 3-50 Hz span are considered to be secondary ride and are instead more related to the disturbances caused by various surface imperfections. It is agreed that secondary ride can be further divided to allow for a better description of the motions, but the exact frequency spans vary between literature. For instance Neal *et al.* [9] says there is a choppy motion occurring between 3 and 5 Hz, shake around 12 to 14 Hz and harshness in the interval between 25 and 50 Hz. These frequencies below 50 Hz are the ones most important for ride comfort for vehicle speeds up to 200 km/h [7].

These attributes can be measured and evaluated on different road types at different speeds. Primary ride is generally measured at higher speeds over roads exciting the lower primary ride frequencies for both small and large amplitudes. Secondary ride is, on the other hand, evaluated over different road surfaces and textures, measuring lower amplitude imperfections at higher frequencies. These measurements are usually performed at medium- to higher speeds. Finally, impact harshness is usually measured by driving over single, uniform impacts of various dimensions at lower speeds.

2.3.2 Evaluation

As mentioned the ride quality can be evaluated from, for instance, accelerations but there is no generally accepted method to objectively assess the complete comfort in a vehicle. One of the commonly preferred methods is to use the RMS value of the acceleration, but the reasoning to use this is not that it predicts the human response better than other measures but rather the convenience of measurement and analysis [8]. Ride performance can also be evaluated through measuring the vibration isolation through the displacement transmissibility from the road to the sprung mass [10]. Another way to measure it is through the vibration dose value

(VDV) which uses a frequency weighting and incorporates shock events better than the RMS acceleration [8]. There are many more measures that can be used and incorporated in the ride evaluation and some of them are the peak values of velocities or accelerations, peak-to-peak values, the crest factor and the root-mean-quad [8].

There are standards such as the ISO 2631 [11] and BS 6841 [12] that propose ways to evaluate the human response to vibration. The ISO 2631 evaluates this through frequency weighted RMS values of accelerations and the BS 6841 determines an overall ride value from ride component values. The use of frequency weighting is to account for the fact that the human body is more sensitive to vibrations of specific frequencies. The ISO standard indicates that the human body has the greatest sensitivity between 4 and 8 Hz for vertical vibrations. One common issue with the objective methods is that they do not directly connect to subjective ratings which are often used since, as mentioned, ride comfort is a subjective measure.

Various attributes of the ride need to be considered to minimize these vibrations which the vehicle's occupants are exposed to. As explained by Neal *et al.* [9], attributes such as body motion smoothness and -control, end of travel, coarse road isolation, impact harshness and shake are all greatly affected by the damper tuning.
Body motion smoothness is desired w.r.t. heave, pitch and roll motions and is controlled mostly by the ride frequencies in both the front and rear. These frequencies normally range from 1.1 - 1.6 Hz. Low damping forces produce two natural frequencies known as body heave ($\sim$ 1.3 Hz) and suspension hop ($\sim$ 12 Hz). The suspension hop and frequencies around are the shake in the vehicle. Excessively high damping forces creates a scenario where the sprung- and unsprung masses move in-phase at frequencies around 3 - 5 Hz, which corresponds to choppiness, as described above.
Body motion control is best evaluated at higher speeds on undulating road surfaces. This attribute is related to the body motion smoothness, mentioned above, and aims to a achieve motions which could be described as "fluid" and "hydraulic". This is a balance obtained by a system which is damped but not over-damped and requires different amount of damping forces for different speeds and surfaces.
End of travel performance is tuned for severe scenarios in order to prevent abrupt dynamic changes in jounce under full compression of the suspension. Excessive dependency on end of travel tuning can lead to increased harshness.
Course road isolation and impact harshness is caused by continuous high frequency, small amplitude disturbances ($\sim$ 25 - 50 Hz) and can be controlled by thorough rebound to compression balancing.

2.4 Preview

Preview technology in suspension systems uses cameras and sensors to scan the road surface ahead of the vehicle to get a preview of the road profile. For a given road the average slope within the range of the sensors is calculated and forms a so-called reference plane. With that reference plane the controller activates the actuators in the suspension by setting the heave position of the vehicle, along with roll and pitch angles to zero. Then the information from that can be used to, in the best way possible, prepare the active components in the suspension for upcoming obstacles and thus improve the vehicle's response to these obstacles [13].

2.5 Control theory

When moving away form a passive suspension towards ones with active components there is a need to control these active parts. There are many different approaches for how this can be done where some of these are linear quadratic regulator (LQR), linear quadratic Gaussian (LQG), fuzzy and neuro-fuzzy control and skyhook and groundhook approaches. A linear quadratic (LQ) control aims to minimize a performance index or cost function with weighting factors. The optimal control method of LQ approaches has been used by Hrovat [14] when investigating possible gain for ride and comfort. Further improvements to ride quality using LQ were sought by Prokop and Sharp [15] through the use of preview information. According to Kashem *et al.* [2] however, there are downsides such as that the LQG controller is complex to use and it is difficult to determine the weighting coefficients involved.

Since suspension systems on vehicles are nonlinear, using fuzzy logic control (FLC) can be appropriate. This because they can work with complex systems without a precise mathematical model and can be used to control the disturbance rejection in semi- and fully active suspensions [2]. FLC with proper rule bases and membership functions are able to be insensitive to model and parameter inaccuracies which is a clear advantage. When used in semi-active systems it has shown similar results to a skyhook control but with an increased variation of the dynamic tyre contact force [2].

2.5.1 Ground- and Skyhook logic

The groundhook and skyhook logic have a very similar approach with the idea that the system is hooked to a fixed point, in the ground or in the sky respectively. The goal of the groundhook logic is to reduce the dynamic tyre force, which in turn improves handling and reduces road damage [16]. The most common approach for active control, however, is the skyhook control method. This was originally developed by Karnopp *et al.* [17] and aims to minimize the body heave, which makes it an effective vibration control algorithm [2]. Because of this it also gives a positive effect on the primary ride component of the ride comfort. The skyhook damping method produces a damping force in the opposite direction and proportional to the absolute velocity of the sprung mass, as shown in equation 2.2.

$$F_d = F_{skyhook} = \begin{cases} c_{sky}\dot{x_1} & \dot{x}\dot{x_1} > 0 \\ 0 & \dot{x}\dot{x_1} \leq 0 \end{cases} \tag{2.2}$$

This equation shows the approximation to an ideal skyhook [3], where $\dot{x}$ is the relative velocity between the chassis and the ground and $\dot{x_1}$ the absolute velocity of the chassis.

There are many versions of the skyhook control method where some are continuous skyhook control, modified skyhook control and optimal skyhook control [2]. One reason for these different versions existing is that there are issues with the basic skyhook control. One big issue is that the semi-active systems have a limited working region, which is caused by the fact that they do not add any energy to the system. When the system enters and exits this region there can be an abrupt change in the applied control force, which causes a rapid change of acceleration [18]. A fast change of acceleration can also occur because of the fact that the skyhook control attempts to minimize the heave. These fast changes in acceleration lead to high derivatives and thereby excessive jerk [18], which is the derivative of the acceleration and has a significant impact on the ride comfort. This high jerk would not affect the RMS value of the acceleration significantly and thus the RMS value would still indicate a good ride. Therefore the use of the RMS metric might give a wrongful indication of the performance of the semi-active system while it gives a more correct indication for a passive one. From this it can be seen that systems with active components need to be evaluated differently from passive systems since they have different factors affecting ride.

2.6 Power Spectral Density (PSD)

Road profiles can be thought of as a collection of random signals transmitted through the vehicle as it drives and can be represented by the Power Spectral Density (PSD) function. It is used to illustrate the distribution of power of a certain signal over a certain frequency. This gives a representation of what frequencies are present and the most pronounced in, for instance, a stretch of road. A signal, which can be the elevation profile over a given stretch of road, can be described in either time or frequency domain. The signal in time domain can be transformed to the frequency domain using the Fourier Transform. The Fourier Transform brakes down the signal into a series of sine waves of different phase and amplitudes. The PSD, subsequently, is the plot of the amplitude and spatial frequency, i.e. the inverse of the wavelength of a given sine wave, also known as wavenumber. This can be expressed by the following equation:

$$G_Z(v) = G_0[1 + (v_0/v)^2]/(2\pi v)^2 \tag{2.3}$$

where $G_Z(v)$ corresponds to the PSD amplitude in length^2/cycles/length, v the spatial frequency or wavenumber in cycles/length, G_0 the roughness level which is dimensionless $[-]$ and finally v_0 which is the cutoff spatial frequency or cutoff wavenumber in cycles/length [1].

2.7 Correlation analysis

There are many different ways available to model correlation, where one of the most commonly used way is through linear analysis. The main advantage with using a linear analysis method is that it is simple and fast to calculate. On the downside, however, it can only represent linear relations which means that the relations are only be described as straight lines. In order to find more complex correlations non linear methods needs to be used. These can for instance be models using quadratic correlation or models able to find if preferred ranges exists.

Linear regression analysis can be carried out using either one single explanatory variable, called simple linear regression, or with multiple explanatory variables, then called multiple linear regression. For simple linear regression the correlation is made between one output value, y_i and this one explanatory variable, x_i, as seen in equation 2.4. This equation also contains other variables where f_i is the regression value, μ_i a random component/error/residual, β_0 a constant and β_1 the regression coefficient. These are summarized in table 2.1. A straight line is through the use of this method fitted to the data using the least squares minimization method, i.e. taking the difference between the data and the regression line and minimizing the sum of its squares.

$$y_i = f_i + \mu_i = \beta_0 + \beta_1 x_i + \mu_i \tag{2.4}$$

Table 2.1: Parameters in simple linear regression model.

Parameter	**Description**
y_i	True value
f_i	Regression value
μ_i	Random component/error/residual
β_0	Constant
β_1	Regression coefficient
x_i	Explanatory variable

For multiple linear regression the process to find the best fit is very similar to the one for simple linear regression. There is still one output value, Y_i, but it is instead correlated to the changes in all of the explanatory variables, x_{i1} to x_{ip}, as shown in equation 2.5. Here μ_i is still a random component/error/residual and β_0 a constant but there are multiple regression coefficients, β_1 to β_p. This where p is the total number of explanatory variables used. All variables can be found in table 2.2.

$$Y_i = \begin{bmatrix}\beta_0 + x_{i1} \cdots x_{ip}\end{bmatrix} \begin{bmatrix}\beta_1 \\ \vdots \\ \beta_p\end{bmatrix} + \mu_i \tag{2.5}$$

Table 2.2: Parameters in multiple linear regression model.

Parameter	**Description**
Y_i	True value
μ_i	Random component/error/residual
β_0	Constant
$\beta_1 \cdots \beta_p$	Regression coefficients
$x_{i1} \cdots x_{ip}$	Regressors
$p = 1, 2, \cdots, n$	Indicator of n different explanatory variables

For linear analysis a measure of the strength of the correlation is needed. One commonly used such measure is the Pearson correlation coefficient, commonly denoted r for a sample and ρ for a population. This is, for two random variables X and Y, found as the covariance between them divided by the product of the standard deviations for the random variables, as shown in equation 2.6.

$$\rho_{X,Y} = \frac{\mathrm{cov}(X, Y)}{\sigma_X \sigma_Y} \tag{2.6}$$

In the case of finding correlation between objective metrics and subjective assessments a linear approach can prove successful if a higher objective rating results in a more positive subjective assessment and vice versa. This would indicate that that a driver prefers a higher objective rating. However, this is not always the case when it comes to ride evaluation as it varies between when a high or a low objective score is preferred, as well as expectations for the type of vehicle being evaluated whether it be a large luxury car or a light sports car [19].

3
Test Procedures

Objectively quantifying the performance of a suspension system can be done at two stages of the development process. Either at an early stage in a virtual environment using a model of the designed system or at a late stage using a physical vehicle where it can be driven and measurements can be taken. As has been mentioned before, undulations and disturbances of certain frequency ranges represent specific ride measures, whether it be broad categorization (primary ride, secondary ride) or narrower, more specific attributes (bounce, choppiness, etc.). These attributes are tested on different tracks using a variety of different cars, where the cars available for physical tests are mentioned in table 3.1. These cars were chosen to provide a variety in weight, center of gravity height and in ride and handling characteristics. Therefore all the cars are not the same model which in turn means that there will be some deviations in results which are not solely due to the suspension type and specification but also the differences between the models. Cars 1 to 3 are equipped with passive suspensions while cars 4 and 5 have semi-active systems. Car 4 is utilizing a skyhook algorithm while car 5 has a preview system in combination with model predictive control (MPC).

Table 3.1: Cars used for physical testing.

Car	**Suspension type**	**Specification**
Car 1	Passive	-
Car 2	Passive	-
Car 3	Passive	-
Car 4	Semi-active	Skyhook
Car 5	Semi-active	Preview MPC

3.1 Simulation

3.1.1 Cars

CarMaker has a broad variety of pre-existing car models available to use for simulation. However, since the results of the simulations are to be compared with results from physical tests where Volvo cars are used, these are not adequate. Therefore, models for cars 1, 2, 4 and 5 in table 3.1 were acquired from an in-house CAE department. These models have a relatively accurate representation of the actual vehicle's dimensions, weight, weight distribution, drivetrain, suspension geometry and -characteristics.

These models use a Pacejka magic formula (MF) tyre model, which is the one available for use. The MF model is a handling tyre model and does therefore not give the optimal results when used for ride evaluations. It will give limited results at higher frequency input from the road, since it is a single contact point model, which means that wheel enveloping effects are not considered.

3.1.2 Tracks

The virtual roads that the vehicle model drives on are created in two ways. Either by manually defining individual segments of road, i.e. straights and curves, and their respective length, width, slope, curvature and friction coefficient and joining them together to create the test environment. The other method to create a road is to utilize a real-life existing road that has been scanned and digitized. This way test environments near identical to real-world environments can be created and simulated. This, combined with the simulated car models mentioned above, is an essential part to allow for comparison between simulated and actual results for a given vehicle and test track.

3.2 Physical testing

The simulations carried out need to be compared and correlated towards something which, in this case, is the data from physical testing. This data can be in two formats, either from objective or subjective tests. Data from the simulations can be directly compared and correlated towards the data from the objective physical tests, to see if they yield similar results. The subjective data can be used to find correlations between both the physical and simulated objective data in order to find which measures and metrics carries the most weight in a subjective assessment. Physical tests were performed at Volvo Cars test facilities at Hällered Proving Grounds, shown in figure 3.1.

Figure 3.1: Aerial view of Hällered Proving Grounds.

3.2.1 Subjective evaluation

All the available cars in table 3.1 were included in a subjective evaluation. Here, test drivers drove as many cars as their schedule allowed meaning that some cars got driven more than others and thus gathered more data. The team consisted of expert drivers, all of whom are professional vehicle dynamics engineers, as well as two amateur drivers, albeit with substantial driving experience and academic knowledge of vehicle dynamics. The tests were conducted on a predefined set of tracks to give all the drivers similar road excitation to give a common basis for the assessment to be made. The focus of the tests was on primary and secondary ride. Small and large amplitude undulations for the primary ride and smooth and rough surfaces for secondary ride to allow for a thorough ride evaluation. For simplification purposes and to not take up too much time impacts were excluded and no focus was put on other factors which does not affect normal ride. For instance the effect of reaching end of travel causing topping and bottoming of the suspension.

While performing the evaluation each driver had a copy of an evaluation form in the car so he/she could evaluate their findings quickly and while their memory was still fresh. The intention of this form is to translate subjective evaluations experienced by the driver to objective metrics and also to rate the vehicle on a higher more overall level. Therefore its purpose is to provide a subjective judgment on objective metrics on a scale from 1-5 as well as to give a rating of how good or bad certain categories are experienced, on a scale of 1-10. A common terminology to describe primary ride is to divide it into three categories: control, balance and comfort. In the evaluation form the two major categories of control and comfort had focus. Control relates to how controlled the vehicle is experienced in comparison to the road. Therefore it includes the bounce, bounce delay, roll and pitch of the vehicle in relation to the road, whether it is copying or feels disconnected or is heavily or lightly controlled. Included is also the damping on the front and the rear axle to also incorporate the balance part and a overall control rating. For the comfort part instead incorporates the experienced overall abruptness and the abruptness and displacement related to headtoss. Abruptness can be described as jerky transitions interrupting otherwise smooth and fluent body movements.

Figure 3.2 shows a road suitable for evaluating large amplitude primary ride.

Figure 3.2: Large amplitude road for primary ride evaluation.

To capture the different parts of secondary ride the three main areas of choppiness, shake and harshness are all included for both a smooth and a rough road. For the choppiness and the harshness there is an evaluation of the level, if the car is experienced as smooth or choppy/harsh, accompanied by an overall rating for each attribute. The shake is instead divided into three smaller areas as shake can occur differently for different parts of a vehicle. The three sources to evaluate for amount of shake is the floor, steering wheel and seat, this again accompanied by an overall rating.

Each test lasted for approximately one hour per car. Some drivers only managed to drive one car which meant they had no comparison to the rest, other than their previous experience of driving nearly identical cars in the past for various purposes connected to their jobs.

Figure 3.3 shows an example of two types of different road surfaces which can be used for secondary ride evaluation: smooth on the left and right of the car and coarse in the middle where the car is driving.

Figure 3.3: Example of different road surfaces for secondary ride evaluation.

3.2.2 Objective evaluation

For the objective testing the five cars (or with the same specification) in table 3.1 were tested according to predefined procedures. In this case accurate measuring equipment was used to measure for instance different displacements and accelerations at various locations of the vehicle, which can then be used in post-processing to extract the desired metrics. The instrumentation consists of speed sensors, optical triggers, string potentiometers measuring wheel deflection and accelerometers on seat rail, steering wheel and suspension top mounts. As for the subjective tests the primary ride was tested for small and large amplitudes while secondary ride was tested for smooth and rough roads. Specific tracks at Hällered Proving Ground were used for specific measurements and each test was run three times to ensure uniform and reliable results. To make sure all measurements are valid general test conditions need to be within acceptable limits. That means an ambient temperatures between 5°C and 30°C, wind speeds between 0 and 8 m/s and a dry or lightly humid track surface. Objective testing is performed by professional drivers, following strict procedures with minimal deviations in throttle and steering inputs in order to produce repeatable, representative results.

4

Post-Processing

4.1 Objective data

In order to find the desired metrics the raw data from each test needs to be loaded into MATLAB and corrected if needed. From the raw data the base parameters for the road and vehicle can be found and these work as a basis to find any desired metrics.

4.1.1 Data conversion

The data gathered from all the different types of tests can be read in MATLAB, where the post-processing takes place. The MATLAB program is set up to allow the user to choose whether the objective input data originates from simulations in CarMaker or from physical tests. Independent of where the data originates from, all the same parameters are extracted. This allows for all the same metrics to be calculated and as fair a comparison as possible to be made. The goal of the post-processing is to find all the metrics which are considered to be of interest and to perform a correlation analysis between both the objective data sets and the subjective data. Of the parameters available in the data from physical tests the ones of interest are the vertical accelerations at the four top-mounts, $a_{tm,i,j}$, which are measured using accelerometers and the relative vertical movement of each wheel, $z_{def,i,j}$, measured from string gauges. These same measurements can be extracted from the CarMaker simulation and with these known for each of the four wheels most other measures of interest can be found.

Before the data from the accelerometers and string gauges used for the physical tests can be used to calculate any metrics it needs to be corrected for drift and noise in the signal. The drift is compensated for by subtracting the mean value from the data. The noise is instead compensated for by passing the data through an ideal low pass filter, this as the noise is of higher frequency than the changes relevant for ride evaluation. The filter used is a moving average filter using the MATLAB function *filter*. The filtered output, $y(n)$, of a moving average filter applied on a vector x with a window size s is defined by equation 4.1. Since the simulation data isn't measured using real equipment, but instead taken from an ideal simulation environment, there is no drift nor noise and therefore it can be used as is.

$$y(n) = \frac{1}{s}(x(n) + x(n-1) + ... + x(n-(s-1))) \tag{4.1}$$

With the data corrected and adjusted the accelerometer data can be integrated to first find the velocity, $v_{tm,i,j}$ and then again to find the displacement, $z_{tm,i,j}$, of the top-mounts. To get a better and easier representation of the road profile and vehicle relative movement, the displacements for both types of data are passed through a similar low pass filter as before. This to find the very low frequency content which consists of major elevation changes such as hills, which can then be subtracted. The data available also contains t which is current time in measurement and $x_{travelled}$ which is the distance travelled. With these derived parameters added to the measured data a base set of data is available for further use, these parameters are summarized in table 4.1.

Table 4.1: Parameters available from physical and simulated data.

Parameter	**Unit**	**Comment**
t	s	Time
$x_{travelled}$	m	Distance travelled
$z_{def,i,j}, i = (l,r), j = (f,r)$	m	Deflection in z of wheel centre
$a_{tm,i,j}, i = (l,r), j = (f,r)$	m/s^2	Top-mount acceleration
$v_{tm,i,j}, i = (l,r), j = (f,r)$	m/s	Top-mount velocity
$z_{tm,i,j}, i = (l,r), j = (f,r)$	m	Top-mount displacement

Outside of these parameters from the test there is also some general information available. This includes parameters such as sampling frequency, track information and vehicle information. These parameters can now be used as part of finding the base parameters for the vehicle and the road. The base parameters for both of these are presented in table 4.2. They include the z-position at the positions the four wheels, $z_{i,j}$, roll angle ϕ, pitch angle θ and z-position at the centre of the vehicle, z_{heave}, as well as the z-position at the driver's position, z_{driver}. These are calculated as shown in equation 4.2-4.8. Since there is no live measurement of the road profile it is estimated as the difference between the top-mount z-position and the z-position of the wheel, as shown in equation 4.3.

Table 4.2: Parameters calculated for the vehicle and road.

Parameter	**Unit**	**Comment**
$z_{i,j}, i = (l,r), j = (f,r)$	m	z-position at wheel location
z_{heave}	m	z-measure at centre of vehicle
z_{driver}	m	z-measure at drivers position
ϕ	rad	Roll angle
θ	rad	Pitch angle

$$z_{vehicle,l/r,f/r} = z_{tm,l/r,f/r} \tag{4.2}$$

$$z_{road,l/r,f/r} = z_{tm,l/r,f/r} - z_{def,l/r,f/r} \tag{4.3}$$

$$z_{heave} = \Big(\frac{z_{l,f} + z_{r,f}}{2} + \frac{z_{l,r} + z_{r,r}}{2}\Big)/2 \tag{4.4}$$

$$z_{driver,f/r} = \frac{\Big(z_{l,f/r} - z_{r,f/r}\Big)\Big(\frac{T_{f/r}}{2} - y_{driver}\Big)}{T_{f/r}} \tag{4.5}$$

$$z_{driver} = z_{driver,f} + \big(z_{driver,r} - z_{driver,f}\big)\frac{x_{driver}}{W} \tag{4.6}$$

here, x_{driver} is the distance from the front axle to the driver and y_{driver} the distance form the center-line of the vehicle to the driver. The track width is denoted T and the wheelbase W.

$$\phi = 1/2\left(\frac{z_{l,f} - z_{r,f}}{T_f} + \frac{z_{l,r} - z_{r,r}}{T_r}\right) \tag{4.7}$$

$$\theta = \frac{(z_{l,f} + z_{r,f}) - (z_{l,r} + z_{r,r})}{2W} \tag{4.8}$$

4.1.2 Metric calculation

There are many different metrics which could be used to try and describe the ride of a vehicle. These can try and capture the vehicle response in primary or secondary ride in many different ways. They can be related to how the vehicle is connected to the road, the balance of the vehicle, how it reacts to inputs of specific frequencies and so forth.

4.1.2.1 Connection to road

One important factor to capture when evaluating the ride of the vehicle is how it behaves in relation to the road. This can be done by looking at the relative heave, pitch and roll of the vehicle to the road. When doing this one can get a better understanding on how connected or disconnected the vehicle is to the road, or how well it copies the changes in the road. For the heave this is done by taking the RMS value of the difference between the drivers z-position in the vehicle and the roads z-position at the drivers location, as shown in table 4.3. Much similar the roll and pitch metric is found as the RMS of the difference between the roll/pitch angle of the vehicle and the road, also as shown in table 4.3. This can also be done in the same manner using a low pass filtered version of the road data. The purpose of doing this is to determine whether the car copies the exact road or if it more follows the main changes and absorbs the minor disturbances.

Table 4.3: Metrics based on vehicles connection to road.

Metric	Calculation	Unit
Heave metric	$\text{RMS}(z_{vehicle,driver} - z_{road,driver})$	m
Heave metric low frequency	$\text{RMS}(z_{vehicle,driver} - z_{road,driver,lowfrequency})$	m
Roll metric	$\text{RMS}(\phi_{vehicle} - \phi_{road})$	rad
Roll metric low frequency	$\text{RMS}(\phi_{vehicle} - \phi_{road,lowfrequency})$	rad
Pitch metric	$\text{RMS}(\theta_{vehicle} - \theta_{road})$	rad
Pitch metric low frequency	$\text{RMS}(\theta_{vehicle} - \theta_{road,lowfrequency})$	rad

4.1.2.2 Balance

Another aspect which is important to attempt to capture is how the front end of the vehicle reacts and moves compared to the rear end. If there are large difference between the front and the rear the vehicle can be experienced as incoherent or strange, for instance. To capture this the mean value of the z-position at the centre of the front axle is divided with the same value for the rear axle, shown in equation 4.9.

$$\begin{gathered} z_{mean,f/r} = \text{mean}\left(\frac{z_{vehicle,l,f/r} - z_{road,l,f/r} + z_{vehicle,r,f/r} - z_{road,r,f/r}}{2}\right) \\ Travel\ balance = \frac{z_{mean,f}}{z_{mean,r}} \end{gathered} \tag{4.9}$$

This can also be done for the roll, where the mean value of the roll on the front axle is divided with the roll on the rear axle, as shown in equation 4.10.

$$\begin{gathered} \phi_{mean,f/r} = \text{mean}\left(\frac{z_{vehicle,l,f/r} - z_{vehicle,r,f/r}}{T_{f/r}}\right) \\ Roll\ balance = \frac{\phi_{mean,front}}{\phi_{mean,rear}} \end{gathered} \tag{4.10}$$

Table 4.4: Balance metrics.

Metric	Calculation	Unit
Travel balance	Equation 4.9	$[-]$
Roll balance	Equation 4.10	$[-]$

4.1.2.3 Velocity metrics

As mentioned in 2.3.2 one of the measures which can be used to evaluate ride is the vertical velocity. Therefore the RMS value as well as the peak value of the vertical velocity of the vehicle are included as metrics.

Table 4.5: Metrics based on vertical velocity.

Metric	Calculation	Unit
RMS vertical velocity	$\mathrm{RMS}(\dot{z}_{vehicle})$	m/s
Peak vertical velocity	$\max(\dot{z}_{vehicle})$	m/s

4.1.2.4 Acceleration metrics

As the most well used metric for ride is the RMS value of the vertical acceleration it is also included here. This is accompanied by the crest factor as in the ISO 2631 which can later allow for a comparison to said standard. This is however complimented with additional metrics based on the acceleration, which can be used in the correlation analysis and for comparison. The peak value of the acceleration is included together with the root mean quad (RMQ) value [8], as some studies has spoken for the RMQ over the RMS. The RMQ follows the same idea as the RMS value but instead of taking the square and square root as it is to the power of four and the fourth root. All acceleration metrics are presented in table 4.6.

Table 4.6: Metrics based on vertical acceleration.

Metric	Calculation	Unit
RMS vertical acceleration	$\mathrm{RMS}(\ddot{z}_{vehicle})$	$\mathrm{m/s^2}$
Peak vertical acceleration	$\max(\ddot{z}_{vehicle})$	$\mathrm{m/s^2}$
Crest factor	$\frac{\max(\ddot{z}_{vehicle})}{\mathrm{RMS}(\ddot{z}_{vehicle})}$	[−]
RMQ vertical acceleration	$\mathrm{RMQ}(\ddot{z}_{vehicle})$	$\mathrm{m/s^2}$

4.1.2.5 Jerk metrics

Multiple potential metrics can also be based on the jerk, which as mentioned is the derivative of the acceleration and is more related to the abruptness experienced. As it is related to the abruptness the value of most interest is its peak value. Jerk can also be induced from semi-active or fully-active suspensions as they try to produce the, for the moment, most optimal force and that could change quickly causing high jerk values. Therefore the peak absolute values of the jerk are included for vertical motion at the driver as well as for roll and pitch motion of the vehicle, which is shown in table 4.7.

Table 4.7: Metrics based on jerk.

Metric	Calculation	Unit
Peak vertical jerk	$\max(\dddot{z}_{driver})$	$\mathrm{m/s^3}$
Peak roll jerk	$\max(\dddot{\phi}_{vehicle})$	$\mathrm{rad/s^3}$
Peak pitch jerk	$\max(\dddot{\theta}_{vehicle})$	$\mathrm{rad/s^3}$

4.1.2.6 Head movement

One approach is to assume that the movement of the head can be connected to the perceived ride. This requires a conversion from roll and pitch angles to displacements in y- and x-direction respectively. In order to accomplish this the vertical distance from the road to the head is needed. This can be found by using the hip point which is a design point in combination with the 95th percentile male sitting shoulder height. This 95th percentile height is found to be $0.848m$ [20]. The displacement due to roll is then calculated as the length of a circle sector according to equation 4.11 and due to pitch according to equation 4.12. These can then be combined using Pythagoras theorem to find a total displacement, as shown in equation 4.13.

$$Roll\ displacement = \sin\phi_{vehicle}(z_{95th} + z_{hip\ point}) \tag{4.11}$$

$$Pitch\ displacement = \sin\theta_{vehicle}(z_{95th} + z_{hip\ point}) \tag{4.12}$$

$$Head\ displacement = \sqrt{(Roll\ displacement)^2 + (Pitch\ displacement)^2} \tag{4.13}$$

From the total head displacement, d_{head}, the RMS and peak value of this displacement can be found and used. This total displacement can also be differentiated three times in order to find the jerk experienced by the head. Then both the RMS and the peak value of this jerk can also be found and used as metrics. The metrics based on head movement are displayed in table 4.8.

Table 4.8: Metrics based on head movement.

Metric	**Calculation**	**Unit**
RMS head displacement	RMS(d_{head})	m
Peak head displacement	max(d_{head})	m
RMS of head jerk	RMS($\dddot{d}_{head}$)	m/s^3
Peak of head jerk	max($\dddot{d}_{head}$)	m/s^3

4.1.2.7 Absolute comparison metrics

For a passive suspension the relation it keeps to the road in terms of pitch and roll angles is of more interest than the absolute values of these. When considering the ride in a car with a fully active suspension however this might change as the capabilities are very different. The perfect ride can be considered to be when the vehicle is kept perfectly flat. This means a complete disconnection from the road and that the angles relative the road does not matter but instead the absolute values as a measure on how well the suspension manages to keep the vehicle flat. Keeping the vehicle flat at all times is also what a perfect skyhook attempts to do, which means these metrics can be seen as deviations from skyhook. Therefore the RMS values of the absolute roll and pitch angles in table 4.9 are added as metrics. For the heave the RMS and peak value of the difference with the mean is used.

Table 4.9: Metrics based on absolute movements.

Metric	Calculation	Unit
RMS roll	$\mathrm{RMS}(\phi_{vehicle})$	rad
RMS pitch	$\mathrm{RMS}(\theta_{vehicle})$	rad
RMS heave	$\mathrm{RMS}(z_{vehicle} - \mathrm{mean}(z_{vehicle}))$	mm
Peak heave	$\max(z_{vehicle} - \mathrm{mean}(z_{vehicle}))$	mm

4.1.2.8 Frequency related metrics

As described in section 2.3.1 ride can be divided in primary and secondary ride. For secondary ride it is interesting to look at the PSD generated from the vertical acceleration. The secondary ride is as mentioned divided into choppiness, shake and harshness but to cover the frequency range better the suggested ranges are extended. The ranges used here are extended to be choppiness from 3 to 7 Hz, shake from 7 to 20 Hz and harshness from 20 to 50 Hz. The PSD estimate is found using the Welch method because of its noise reducing abilities. This PSD estimate can then be combined with the weights found in BS6841 [12] and shown in table 4.11, to account for the human sensitivity to vibrations. However both the weighted and non-weighted measures are kept for the correlation analysis. In order to use the weights for the specific frequencies interpolation is carried out.

Table 4.10: Metrics based on the frequency of the content.

Metric	Calculation	Unit
RMS choppiness	$\mathrm{RMS}(\mathrm{PSD}(3 < f < 7))$	m/s^2
RMS weighted choppiness	$\mathrm{RMS}(\mathrm{PSD}(3 < f < 7) weights(3 < f < 7))$	m/s^2
RMS shake	$\mathrm{RMS}(\mathrm{PSD}(7 < f < 20))$	m/s^2
RMS weighted shake	$\mathrm{RMS}(\mathrm{PSD}(7 < f < 20) weights(7 < f < 20))$	m/s^2
RMS harshness	$\mathrm{RMS}(\mathrm{PSD}(20 < f < 50))$	m/s^2
RMS weighted harshness	$\mathrm{RMS}(\mathrm{PSD}(20 < f < 50) weights(20 < f < 50))$	m/s^2

Table 4.11: BS 6841 weighting coefficients.

Frequency	Weight	Frequency	Weight	Frequency	Weight
0.5	0.334	3.15	0.662	20	0.708
0.63	0.367	4.0	0.889	25	0.600
0.8	0.381	5.0	1.025	31.5	0.492
1.0	0.385	6.3	1.055	40	0.393
1.25	0.386	8.0	1.025	50	0.313
1.6	0.392	10	0.974	63	0.239
2.0	0.417	12.5	0.907	80	0.171
2.5	0.494	16	0.810		

4.2 Subjective data

The data from the subjective testing forms is summarized in an excel sheet which allows for easy import into MATLAB. Once imported the mean, median and standard deviation values for each car can be found and used for further analysis. The subjective data can then be visualized based on what a specific driver scored each car he or she drove or based on how each car was scored by all drivers who drove it. From these visualizations it can be seen to which extent the drivers agree on the cars attributes, this based on the spread of the scores. Specific cars can also be compared by plotting their mean or median with the standard deviation marked around.

4.3 Correlation study

To find correlations between the different types of data a linear regression model is used. This due to its simplicity but with the knowledge that it excludes any non-linear correlations in the data. The correlations study is carried out in MATLAB using the function called *corrcoef*. This function preforms linear analysis through the calculation of Pearson's correlation coefficients between each of the different inputs and returns this as a matrix. What is considered to be a strong correlation varies from case to case but the lowest acceptable absolute value for Pearson's coefficient is set to 0.5. Using this the correlations between the objective metrics and the subjective assessments can be found. When significant correlation is found a linear approximation can be made and its 95% confidence intervals found using the MATLAB functions *polyfit* and *polyval*. The linear approximation is made using the least square method in *polyfit*. The output from *polyfit* can then be used as input into *polyval* which outputs a standard error estimate which is used for the 95% confidence intervals. The 95% confidence intervals are used to provide a quick and rough overview of how accurate the linear approximation is.

5

Results

As mentioned throughout, the work is divided in three main stages, simulation, physical testing and a correlation study. The physical testing is then split into an objective part and a subjective part. This results in objective data from two sources; CarMaker simulations and physical tests, which are compared to one another and later correlated to the subjective data.

5.1 Objective

5.1.1 Simulation

As mentioned the simulations are carried out in CarMaker where the available cars were car 1, 2, 4 and 5. In figure 5.1 and 5.2 the simulation results over the large amplitude primary ride track are shown for car 2 and 5. The results for car 1 and 4 are shown in Appendix A. From these it can be seen that the behaviour of the passive cars are similar to one another and that they behave as one would expect. They have a large primary heave motion after an undulation and significant overshoots around the "neutral" state relative to the road until it stabilizes. For the roll angle of both cars there are not many large deviations except for around the undulations, especially for car 2. The pitch angle of both passive cars also behaves in a very similar way to one another with a pronounced peak around the middle. There is however a significant difference of the amplitude of these peaks. Possible reasons for this can be errors in the simulation, with for instance imperfections in the track but could also be due to the vehicles response over the undulation. Adding the results in Appendix A, one can see that both the passive cars have the significantly larger peak compared to both the semi-active cars. One would expect the road profile in both cases to be more similar but this difference could be from the different responses of the vehicles since the road profile is estimated from the vehicle, as described in section 4.1.1.

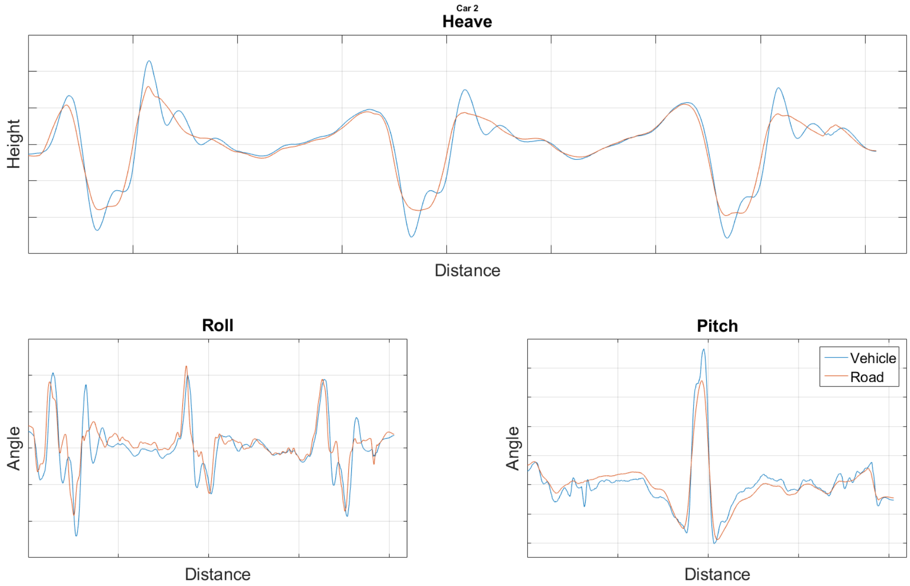

Figure 5.1: Simulation results of heave, pitch and roll for car 2 (passive) vehicle on primary ride large amplitude track.

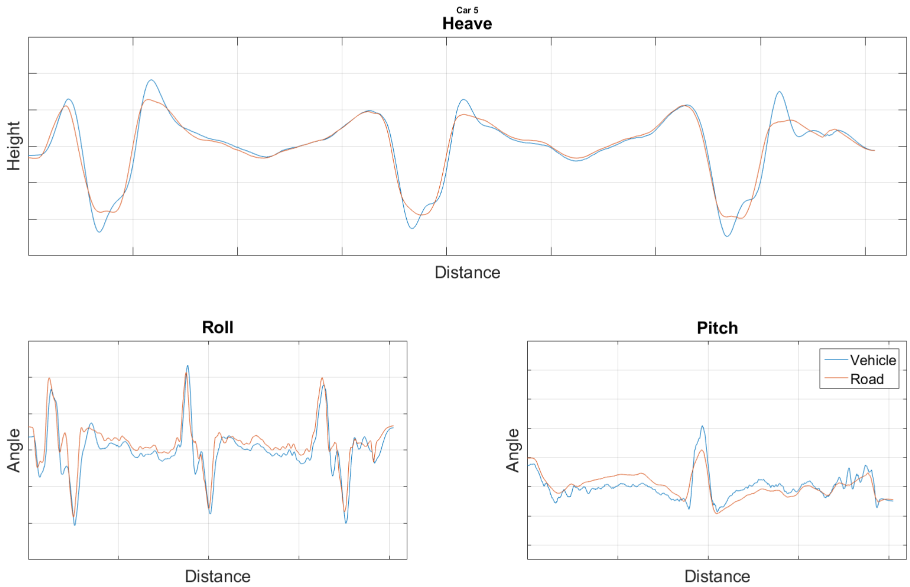

Figure 5.2: Simulation results of heave, pitch and roll for car 5 (preview) vehicle on primary ride large amplitude track.

The results for the two semi-active cars (4 & 5) differs from the passive cars in how they are able to handle the motions. There is still a peak of the primary heave motion, as seen in figure 5.2, after a undulation but especially for the model of the preview car it is not as pronounced. There is also a significant reduction in the overshoot motions, where the preview car eliminates close to all overshoot motion while the skyhook car still has some, although very little. Similar changes can be observed from the roll angle with the skyhook car, keeping the vehicle in closer connection to the road and the preview car doing this even better. For the pitch angle the most significant difference is that the tall peak seen in both passive cars is not as pronounced in either of the semi-active ones.

This behaviour can also be seen in the metrics when looking at the primary ride large amplitude part in table 5.1. The bounce, roll and pitch metrics all show that cars 4 & 5 are more controlled and thus deviate less from the movement of the road. This pattern is not shown when looking at the small amplitude primary ride, where all cars are much closer to each other in terms of bounce, roll and pitch. However, the values for peak jerk are clearly higher for the semi-active cars which can be caused by the rapid change of damping coefficient.

Table 5.1: Selection of metrics from simulation.

Metric	**Unit**	**Car 1**	**Car 2**	**Car 4**	**Car 5**
Primary ride - small amplitude					
Bounce	mm	5.49	4.53	4.80	4.70
Roll	deg	0.148	0.106	0.127	0.119
Pitch	deg	0.135	0.094	0.104	0.095
Peak jerk	m/s^3	2095.8	1549.8	2406.5	2718.2
RMS acc.	m/s^2	1.52	1.257	2.18	2.22
Peak acc.	m/s^2	15.90	14.19	18.91	19.05
Primary ride - large amplitude					
Bounce	mm	23.36	26.03	22.52	21.19
Roll	deg	0.359	0.437	0.412	0.358
Pitch	deg	0.316	0.301	0.273	0.275
Peak jerk	m/s^3	1031.6	1255.3	1487.2	1325.5
RMS acc.	m/s^2	2.18	2.54	2.68	2.51
Peak acc.	m/s^2	9.71	11.98	12.91	11.29
Secondary ride - smooth surface					
Choppiness	m/s^2	0.335	0.228	0.196	0.216
Shake	m/s^2	0.746	0.521	0.510	0.537
Harshness	m/s^2	0.191	0.160	0.259	0.217
Secondary ride - rough surface					
Choppiness	m/s^2	0.653	0.615	0.444	1.015
Shake	m/s^2	0.756	0.756	0.634	1.050
Harshness	m/s^2	0.194	0.198	0.211	0.586

5.1.2 Physical

The passive cars (1, 2 & 3) in physical tests all show similar behaviour, a pronounced peak in heave followed by multiple overshoots. This is also the case for the roll angle. In figure 5.3 and 5.4 a clear reduction of the peak and the overshoots can be seen from car 2 to car 3, this as the cars are increasingly stiffer. As for the semi-active cars, car 5 (preview) presented in figure 5.5, produces low peaks in heave and very little overshoot is observed. Furthermore, it especially follows the heave movement of the road without creating as sharp peaks. Car 5 does have peaks and overshoots when looking at the roll angle, which car 4 has very little of. The results for car 1 and 4 are presented in Appendix A.

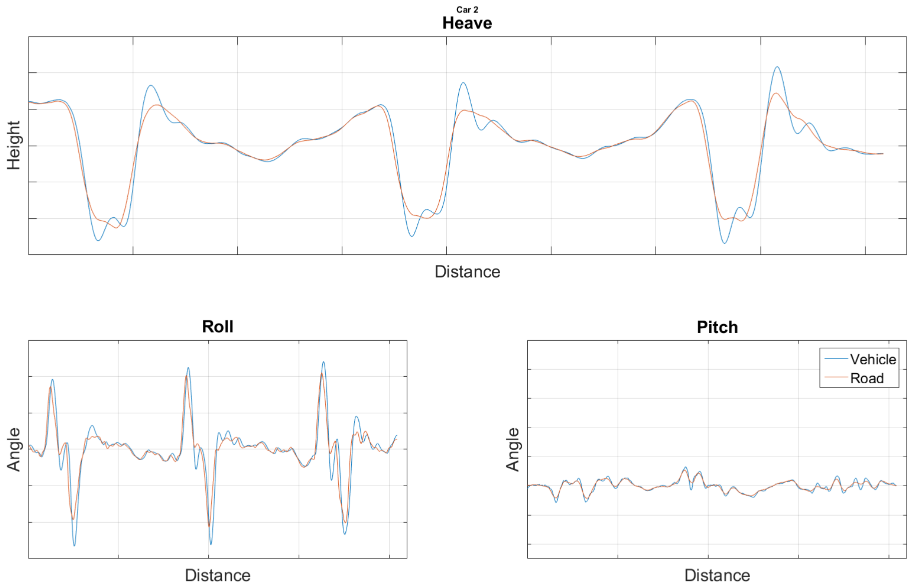

Figure 5.3: Heave, pitch and roll of car 2 (passive) from physical test.

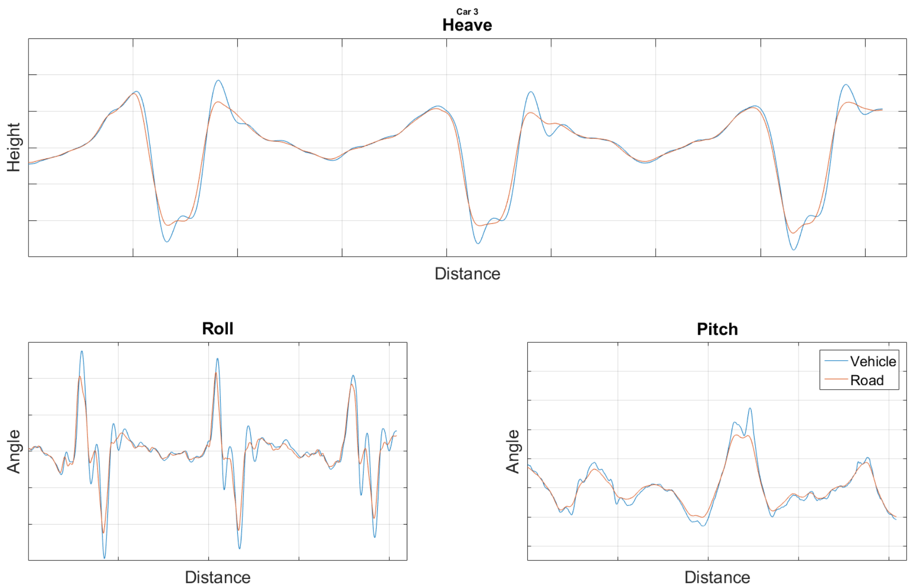

Figure 5.4: Heave, pitch and roll of car 3 (passive) from physical test.

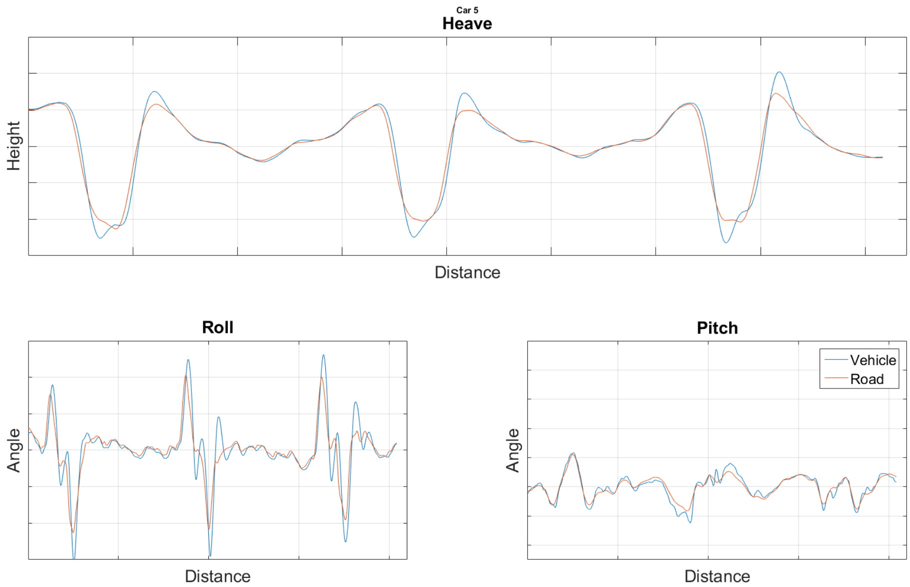

Figure 5.5: Heave, pitch and roll of car 5 (preview) from physical test.

Similar observations can also be made from the figures when looking at the metrics in table 5.2. For the small amplitude primary ride the bounce of car 3 is clearly lower than for cars 1 & 2, the other passive cars. For the semi-active cars this drops even lower. This lower bounce for car 3 comes at the cost of a higher RMS and peak of the vertical acceleration. This is not the case for car 4, where the bounce is lower while also keeping the RMS and peak values in line with car 2, clearly showing the impact a semi-active system can have. Car 5 does have a lower bounce compared to car 4 but at the cost of higher RMS and peak accelerations. Another difference for car 5 is the significantly increased jerk, which is an effect of the system taking control actions due to roll inputs causing abrupt changes in the force. For the large amplitude track the lowest bounce is instead found for car 3, while car 3, 4 & 5 all have significantly lower bounce values compared to car 1 & 2. Again, the much increased jerk for car 5 is observed, this while it produces the lowest acceleration values.

Moving over to the secondary ride over smooth surface, car 3 is observed to have by far the highest values out of any car. This is the downside of the increased stiffness which gives it very good primary ride capabilities. The semi-active cars also produce higher values than cars 1 & 2 but not as high as car 3 while still being very capable in primary ride. Similar trends are seen for the rough surface, but the differences are not as large.

Table 5.2: Selection of metrics from physical testing.

Metric	Unit	Car 1	Car 2	Car 3	Car 4	Car 5
Primary ride - small amplitude						
Bounce	mm	1.73	1.76	1.65	1.59	1.53
Roll	deg	0.042	0.041	0.047	0.038	0.034
Pitch	deg	0.056	0.027	0.049	0.036	0.034
Peak jerk	m/s^3	443.8	292.2	485.5	263.4	675.1
RMS acc.	m/s^2	0.262	0.258	0.351	0.257	0.286
Peak acc.	m/s^2	1.10	1.10	1.52	1.07	1.27
Primary ride - large amplitude						
Bounce	mm	29.48	24.17	18.22	19.21	19.75
Roll	deg	0.422	0.317	0.310	0.256	0.382
Pitch	deg	0.173	0.057	0.157	0.1222	0.141
Peak jerk	m/s^3	1565.2	403.5	551.0	429.5	2227.3
RMS acc.	m/s^2	3.36	2.66	2.29	2.11	1.87
Peak acc.	m/s^2	13.79	9.05	8.15	9.13	7.62
Secondary ride - smooth surface						
Choppiness	m/s^2	0.099	0.126	0.260	0.163	0.134
Shake	m/s^2	0.228	0.232	0.789	0.335	0.351
Harshness	m/s^2	0.131	0.112	0.177	0.148	0.143
Secondary ride - rough surface						
Choppiness	m/s^2	0.849	1.205	1.383	0.855	0.930
Shake	m/s^2	1.52	1.68	2.01	1.52	1.44
Harshness	m/s^2	0.523	0.779	0.529	0.430	0.350

5.1.3 Comparison

Comparing the simulated results with the physical there are differences and similarities between them. The most outstanding differences are found in the small amplitude primary ride test. For instance the values for the simulated bounce are more than a factor two larger compared to the physical ones and the accelerations differs with a factor around 10. This indicates that the track used in simulation is not the same as the one used for the physical tests. The track is still of small amplitude and can be seen as such but the physical tests and subjective assessments are not made on the same track as the simulations. Another major difference is found in the performance of car 1. Comparing its results from simulation and physical tests it can be seen that there is less significant overshoots in the simulation as opposed to the physical tests. This is also seen in the metrics where the bounce, RMS and peak accelerations are significantly lower and car 1 produces lower values than car 2 in simulation, which is not the case for the physical tests. Again, this suggests an error in the simulation model used for car 1.

For most of the large amplitude primary ride the results appear comparable, this in both absolute and relative terms. There is a difference between the simulated and physical values but it is for the most part not that large. For the bounce the difference is around 10%. There are clear differences between some metrics, where the peak jerk is one of them. In the simulation all cars are on a fairly equal level while in the physical tests there are major differences between them. This can be due to imperfections in the simulation causing some sharp edges in the road or similar which can lead to a peak in the jerk. Important for the results, however, is the fact that most of the same trends seem to appear in both sets of results. For the higher frequency secondary ride the results are expected to differ, which they also do for both smooth and rough surface. The measurement for the smooth surface secondary ride are made at the same track as the small amplitude primary ride which means that these simulation results are not directly comparable to the physical results either.

5.2 Subjective

The results from the subjective tests can, as previously mentioned, be divided per car, driver or as a comparison between the cars. Depending on how it is divided different aspects can be analyzed to understand the full picture.

5.2.1 Divided per car

When dividing the subjective scores per car the overall score for said car is seen and also how the different drivers rated it relative to each other. Figures 5.6 and 5.7 shows the results summarized per car for cars 3 and 5, results for the remaining car can be found in Appendix B. Car 2 was only driven by 4 of the drivers, which is seen in figure B.2. Driver 1 also rated the secondary ride of the car as overall much poorer than the other drivers due to imbalance in a wheel originating from a crack in the rim. Because of this cracked rim the car had to have its wheels swapped which reduced the available time to drive it resulting in only four of the drivers having the time to do so.

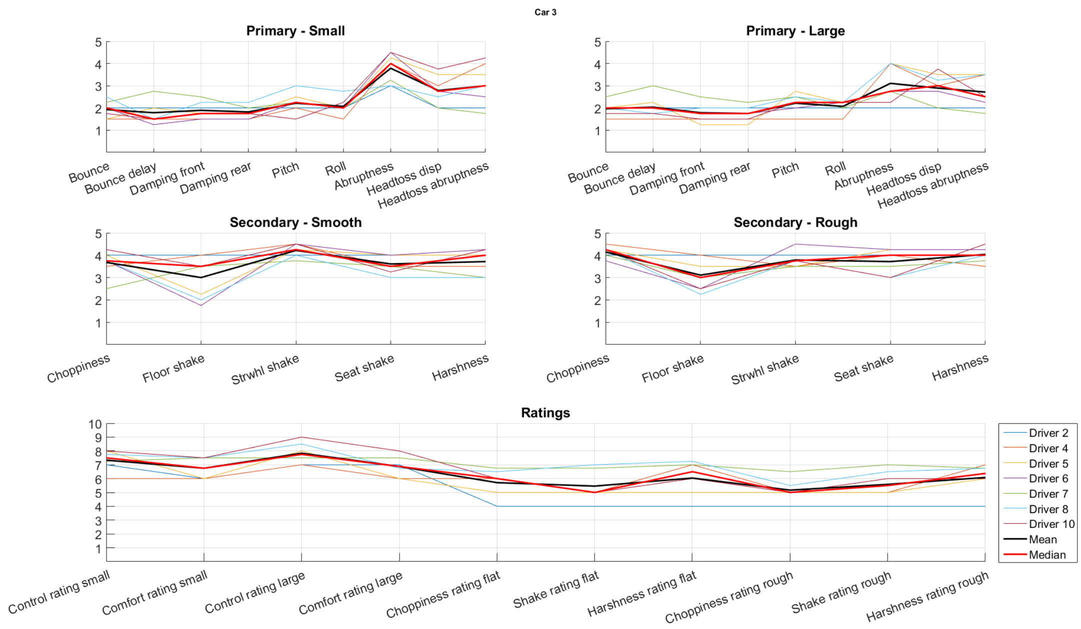

Figure 5.6: Subjective scores for car 3.

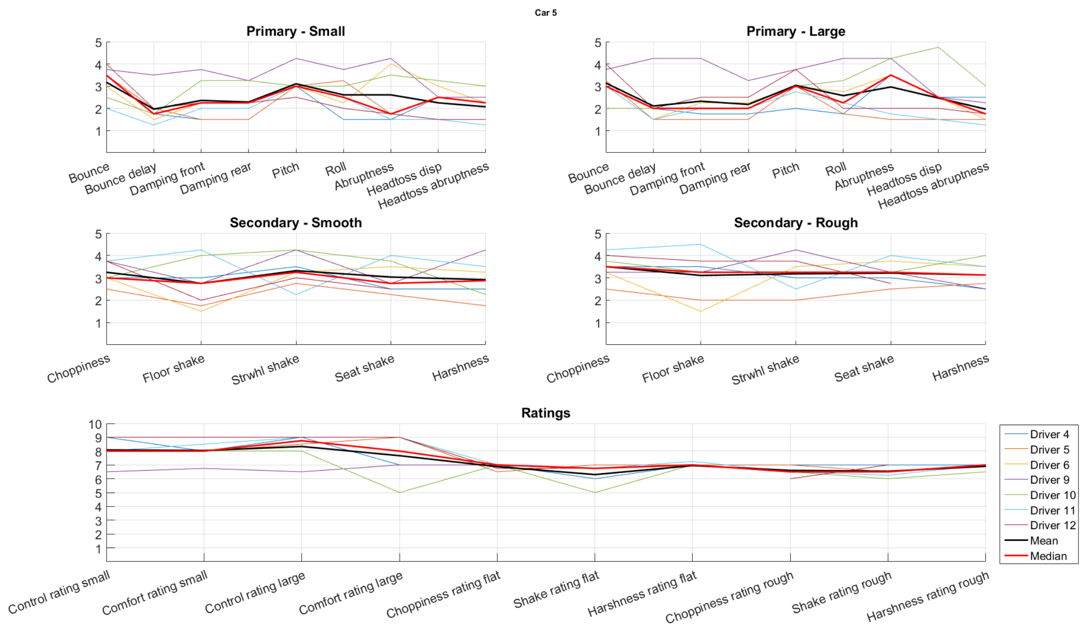

Figure 5.7: Subjective scores for car 5 (preview).

Common for the passive cars (1, 2 & 3) is that most of the drivers score them fairly similar relative to each other creating a trend between them meaning that the mean value gives a fair representation. These trends are overall not as clear for the semi-active cars. This especially for car 5 (preview) where the standard deviations, found in table 5.3 & 5.4, are significantly larger than for any of the other cars. The reason behind this could be that the drivers have an expectation of what should happen when they drive over a certain undulation, in terms of the character of the response of the vehicle. With the passive cars, expectations are close to what happens in reality, while for a semi-active car the expectations and reality are further apart. Adding the preview system moves the response of the vehicle further away from the driver's expectations and hence it might be harder to make a definitive assessment.

Looking at the primary ride scores the passive cars, as well as the skyhook car, keep very similar trends and a low spread. This except for scores given to abruptness, headtoss displacement and headtoss abruptness (metric 7-9) where the passive cars have a significantly larger spread than primarily the skyhook car, as seen in table 5.3. It can also be seen that the preview car has a large spread on these metrics but it does not have the same relative increase in the standard deviation on these compared to the other primary ride metrics (metric 1-7) as the passive cars have.

Moving over to the secondary ride scores a similar trend emerges, such that the passive cars have overall lower standard deviations. This disregarding car 2 as one reading is significantly flawed, which can be seen in the standard deviations in table 5.4. Similar here as well is the fact that the skyhook car (Car 4) is closer to the passive cars than the preview one. Another interesting point to look at is the floor shake metric. As seen in all of the figures this metric has a particularly large spread across all cars, which is also seen in the standard deviation in table 5.4.

This trend is not present when looking at ratings, there the largest spread is found in car 3. Even more interesting is the fact that car 3 is the one with the overall lowest standard deviations in the other scores. This means that the drivers judge its performance to the same levels but how they see that convert into good or bad differs significantly between them. This shows on different preferences between different drivers and that the same car, which everyone says performs the same way, is perceived as better or worse depending on who drives it, which is what makes objectively quantifying ride very difficult.

Table 5.3: Standard deviation mean values in primary ride subjective scores.

Car	small 1-6	small 7-9	large 1-6	large 7-9
Car 1	0.516	0.657	0.477	0.639
Car 2	0.401	0.519	0.434	0.584
Car 3	0.397	0.779	0.377	0.779
Car 4	0.544	0.455	0.481	0.400
Car 5	0.723	0.885	0.771	0.975

Table 5.4: Standard deviations in secondary ride subjective scores.

Car	flat: -mean	-floor shake	rough: -mean	-floor shake
Car 1	0.403	0.476	0.597	0.822
Car 2	0.950	0.747	1.020	0.851
Car 3	0.559	0.968	0.446	0.734
Car 4	0.651	0.894	0.589	0.741
Car 5	0.788	1.080	0.705	1.029

5.2.2 Divided per driver

When dividing the results per driver, as in figure 5.8 and 5.9, the relative perceived performance of the cars can be studied. Subjective scores for all drivers can be found in Appendix B.

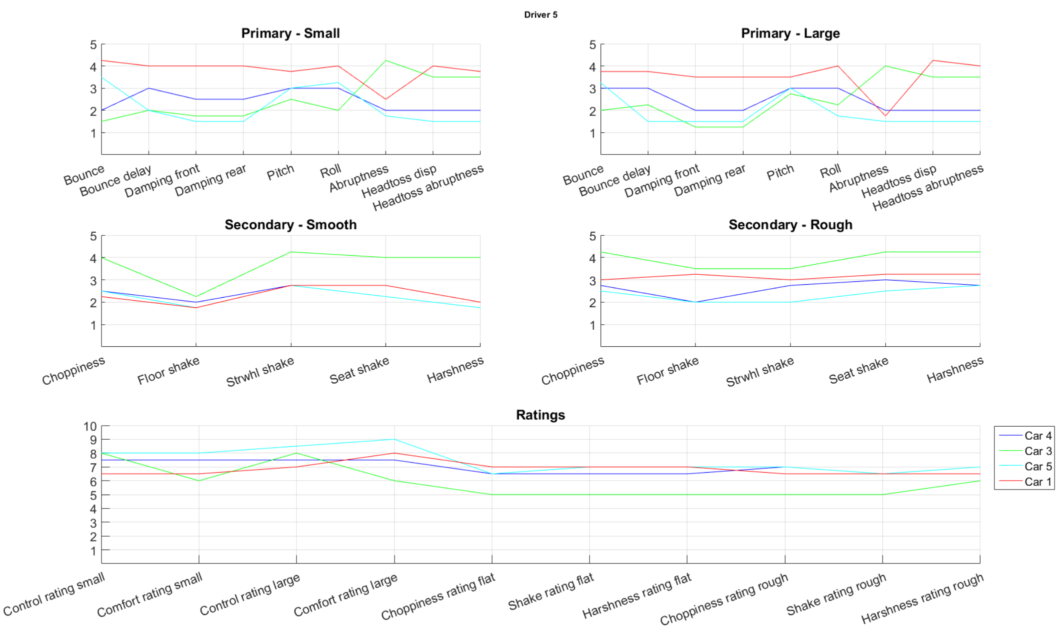

Figure 5.8: Subjective scores set by driver 5.

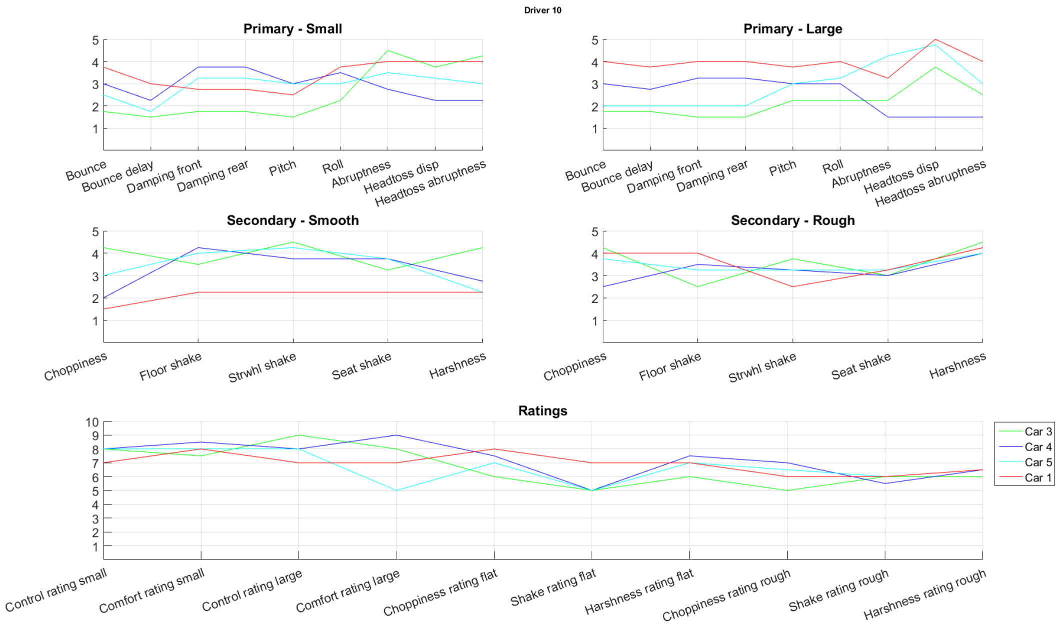

Figure 5.9: Subjective scores set by driver 10.

5.2.3 Comparison between cars

When plotting the average values for all cars together as in figure 5.10 it is possible to see the consensus judgment of the cars relative to each other. For the small amplitude primary ride scores there are two cars deviating from the rest. These are cars 1 and 3 which score higher respectively lower than the remaining cars. The ratings related to small amplitude primary ride of these cars however are both lower than for the other three. This indicates that a more loose, car which is experienced as more floating and disconnected, as car 1, or a car which is more firm and copying of the road, as car 3 is, are both undesired. Instead the remaining cars which are more absorbing and presumably not too firm nor too loose are scored higher, thus indicating that none of the extremes are desired but rather these in between. Cars 1 & 3 also get a lower comfort rating than the others, since both of them are scored worse for abruptness, headtoss displacement and headtoss abruptness. This is similar for the large amplitude primary ride judgments but here car 3 is rated higher in control rating despite still being judged as copying and firm.

Looking at the secondary ride for both flat and rough surfaces it is clear that the cars judged to have the most choppiness, shake and harshness which corresponding to the highest score also score the lowest ratings. The opposite is also true, or close to, the cars which scored the highest ratings were judged to have the lowest levels of choppiness, shake and harshness.

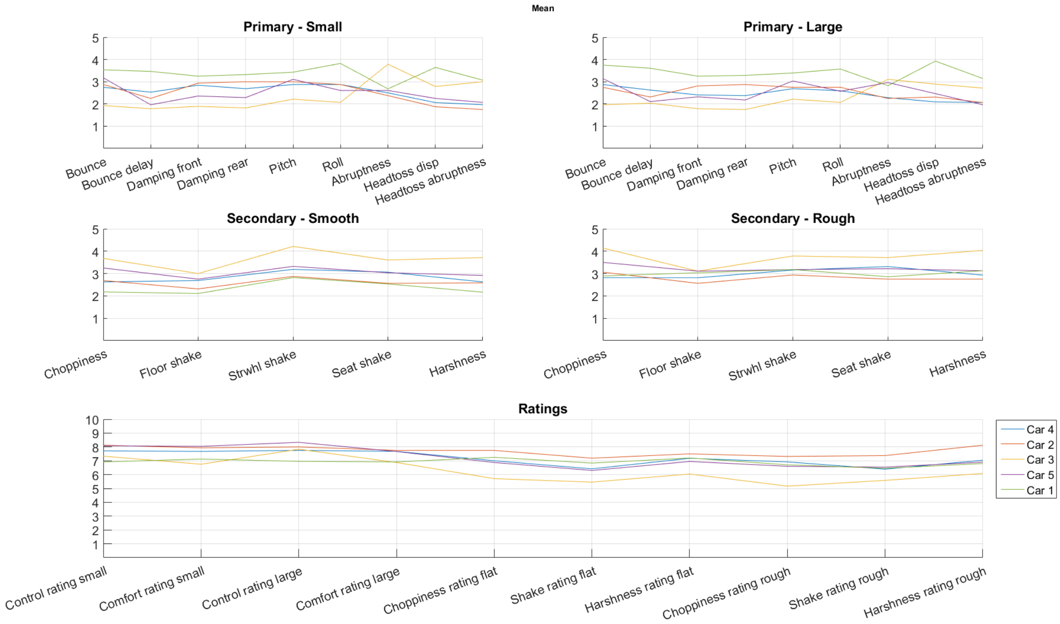

Figure 5.10: Mean subjective scores for all cars.

5.3 Correlation

In order to find which objective metric gives the best representation of the ride, as experienced in the car, a correlation study is carried out. The most important correlation is the one between the physical objective tests and the subjective scores. This since these are the same types of cars on the same tracks which can, subsequently, give an understanding of what attributes are valued as important. This can then be checked if it also correlates to the simulated data or if the same trends are valid. If the same trends are valid then the metric can be used from an early stage in the CAE process.

5.3.1 Primary ride

The correlation matrices for primary ride are presented in figures 5.11 to 5.14, where points 1-10 are the subjective ratings and the remaining are the objective metrics, described in section 4.1.2. The marked correlations all have a correlations coefficient of at least 0.5. For the small amplitude the subjective scores of interest are number 1 and 2, since these are the control and comfort ratings for small amplitude road. This means that the vertical lines number 1 and 2 are where relevant correlations are found. For the large amplitude the control and comfort metric are instead number 3 and 4 respectively.

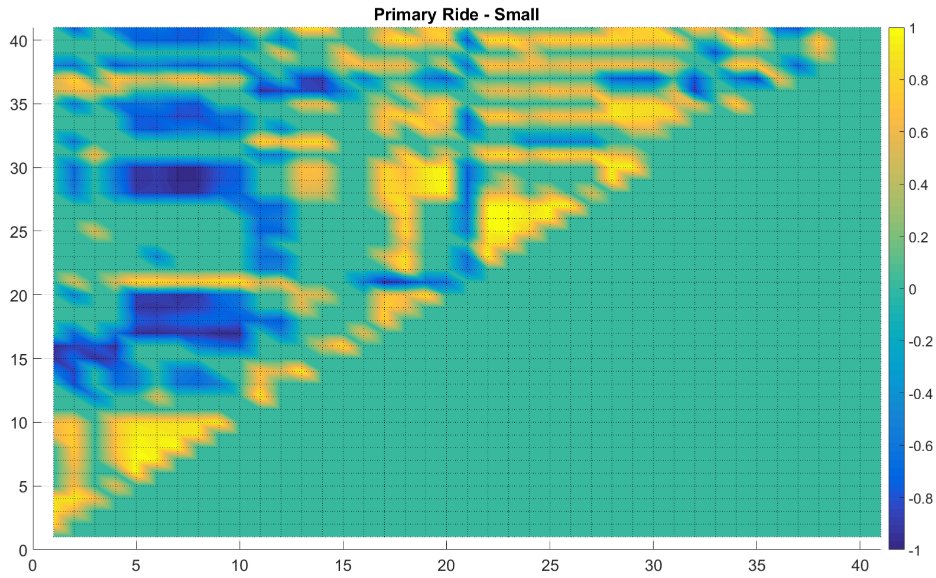

Figure 5.11: Correlation between subjective ratings and physical objective metrics for primary ride small amplitude road.

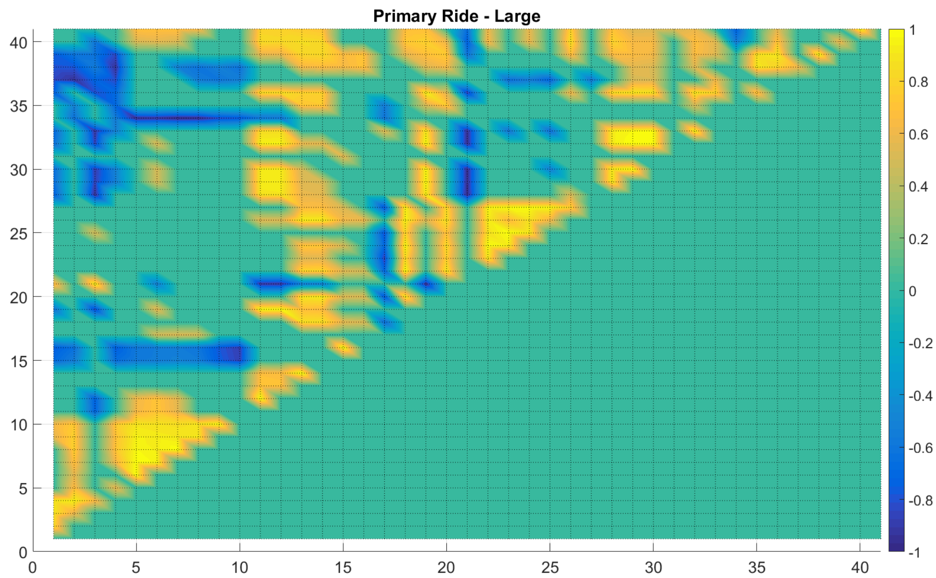

Figure 5.12: Correlation between subjective ratings and physical objective metrics for primary ride large amplitude road.

Figure 5.13: Correlation between subjective ratings and simulated objective metrics for primary ride small amplitude road.

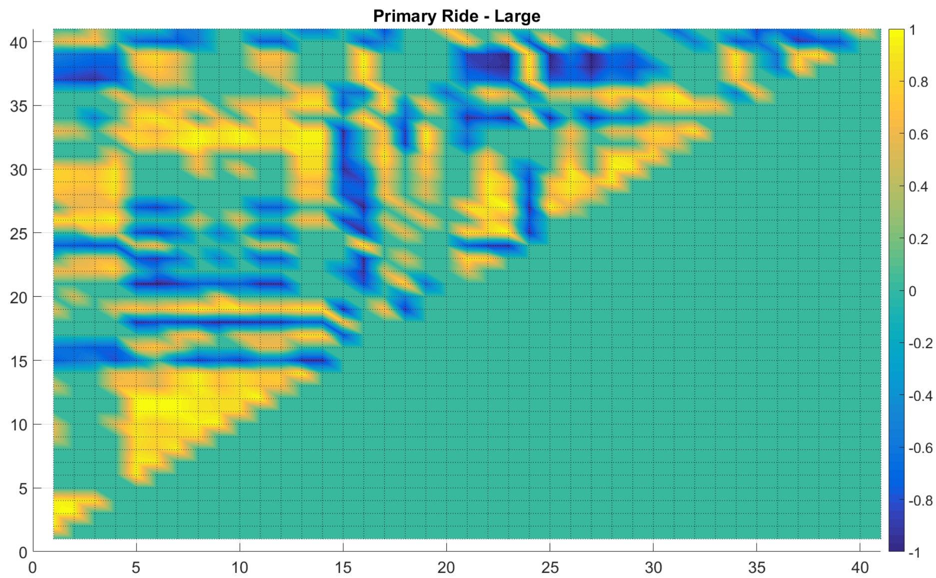

Figure 5.14: Correlation between subjective ratings and simulated objective metrics for primary ride large amplitude road.

When comparing the correlation matrices between physical and simulated, figures 5.11 to 5.13 and 5.12 to 5.14, it is clear that they differ significantly. As mentioned the first priority is to find correlations between the physical objective metrics and the subjective scores, therefore the strongest correlations from both small and large amplitude are summarized in table 5.5. Also here can the differences between simulated and physical results be seen.

Table 5.5: Correlation coefficients for primary ride.

Subjective	**Objective**	**Phys. corr.**	**CM corr.**
Small amplitude			
Control	Roll	-0.973	-0.994
Comfort	Roll	-0.858	-0.982
Comfort	Head displacement	-0.600	-0.970
Comfort	Peak head jerk	-0.803	-0.914
Large amplitude			
Control	Peak vertical acc.	-0.947	0.598
Control	RMS vertical acc.	-0.865	0.724
Control	Peak vertical vel.	-0.925	-
Control	Peak heave	-0.718	-0.935
Comfort	Roll	-0.749	-0.730
Comfort	Peak heave	-0.826	-0.772
Comfort	Head displacement	-0.933	-0.764

The strongest correlation found for the small amplitude control rating is to the roll metric. This also appears valid for the simulated data. The linear approximation of subjective control rating versus objective roll metric for small amplitude primary ride can be seen in figure 5.15 for the physical data and in figure 5.16 for the simulated. Here the scattered dots mark each driver's score, the star markings the mean score, the red lines the 95th percentile confidence level for the mean value and the black lines mark the 95th percentile for the scores of the population. The 95th percentiles are added as a basic statistical measure to give a rough estimate on how well the approximation fits the data. According to the linear approximation in figure 5.15 can the mean subjective control rating be approximated to about 0.5 points accuracy with a 95% accuracy if the physical roll metric is known and even more accurate if the simulated roll metric is known. However, it is important to remember that the simulated track is not directly comparable to the physical, which can be seen on the values of the objective metric on the x-axis. It is also important to remember that it is a small sample size. For the comfort rating on the small amplitude road the roll, again, has the strongest correlation but also the peak head jerk, shown in figure 5.17, along with a strong correlation in the head displacement.

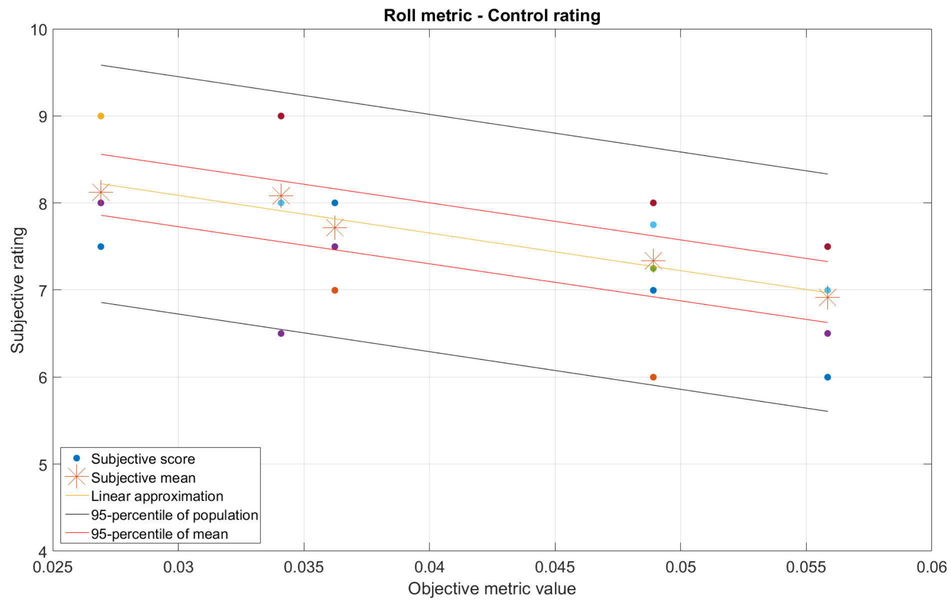

Figure 5.15: Linear approximation of subjective control rating and physical roll metric for primary ride small amplitude road.

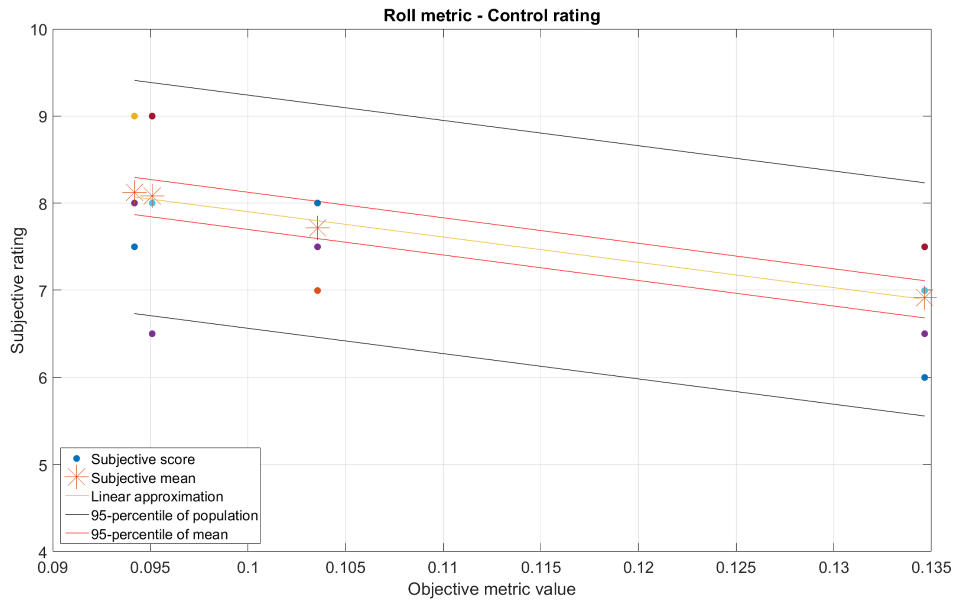

Figure 5.16: Linear approximation of subjective control rating and simulated roll metric for primary ride small amplitude road.

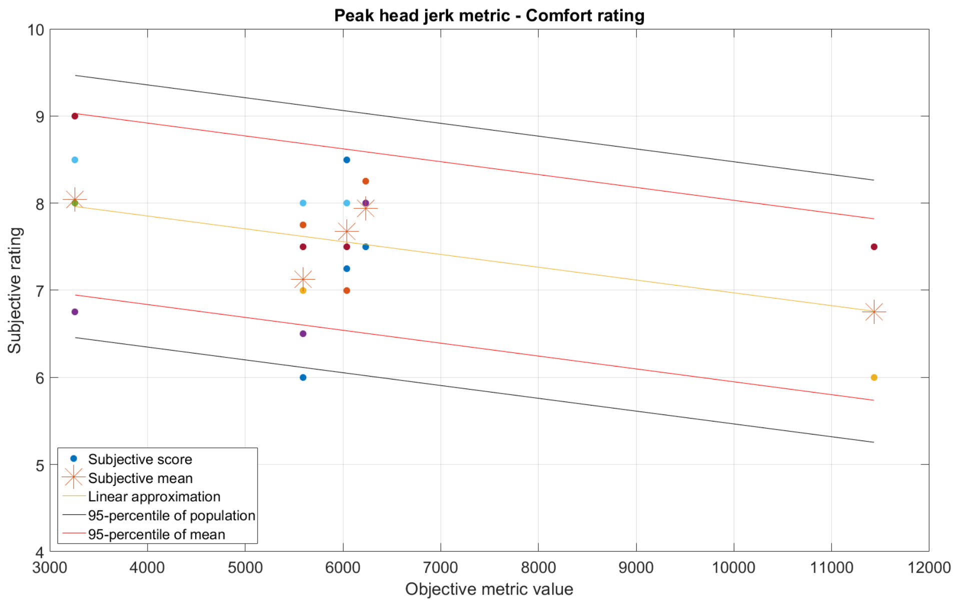

Figure 5.17: Linear approximation of subjective comfort rating and physical peak head jerk metric for primary ride small amplitude road.

For the large amplitude track the strongest correlation to the subjective control rating is found in the peak vertical acceleration. This linear approximation is shown in figure 5.18 and given a value for the peak vertical acceleration the average subjective control rating can be estimated to a 1 point range with 95% accuracy. Similar is true for the peak vertical velocity and also the RMS of the vertical acceleration correlates well. Interesting for both the peak and RMS related to the vertical acceleration is that the correlation from the simulated results is very different. From these results it shows on a positive correlation between the metrics and the subjective rating instead of a negative. Therefore the peak heave metric can be more interesting to use as it has a negative correlation for both the physical and simulated results as seen in table 5.5. For the subjective comfort rating the best correlation is found for the head displacement, shown in figure 5.19 but there is also a strong correlation to the peak heave metric. For these there are similar, although weaker, correlations also for the simulation results. The linear approximations for max velocity to control rating and peak heave to comfort rating are shown in Appendix C.

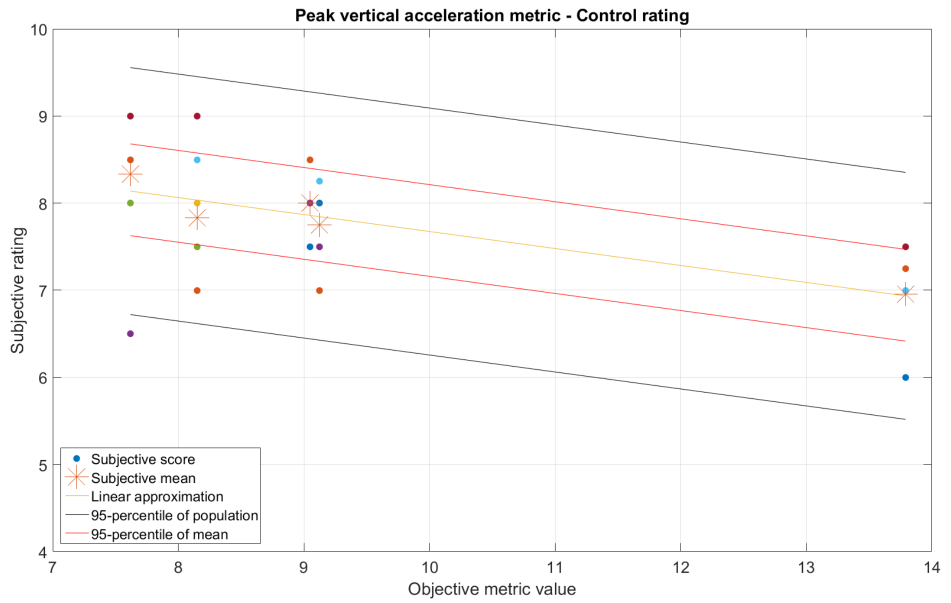

Figure 5.18: Linear approximation of subjective control rating and physical peak vertical acceleration metric for primary ride large amplitude road.

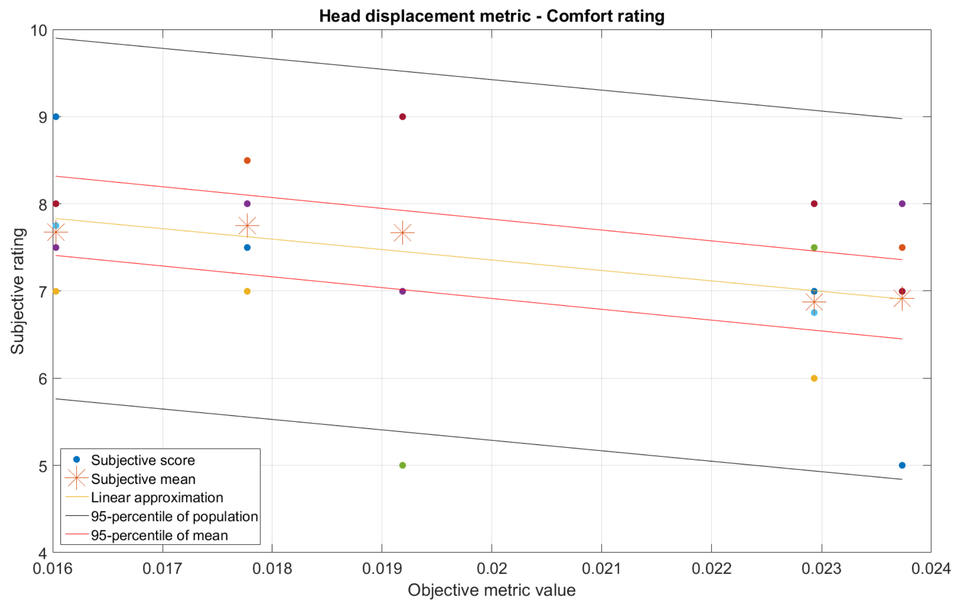

Figure 5.19: Linear approximation of subjective comfort rating and physical head displacement metric for primary ride large amplitude road.

5.3.2 Secondary ride

The subjective ratings connected to the secondary ride, namely choppiness, shake and harshness are number 5 through 10, where 5-7 are for smooth surface and 8-10 for rough surface. Correlations for physical data are shown in figure 5.20 and

5.21 and for the simulation results in figure 5.22 and 5.23. Since the track used for smooth surface secondary ride is the same one used for small amplitude primary ride, it needs to be remembered that the cars were not subjectively evaluated on a track with the same properties as the track used to find the simulation results.

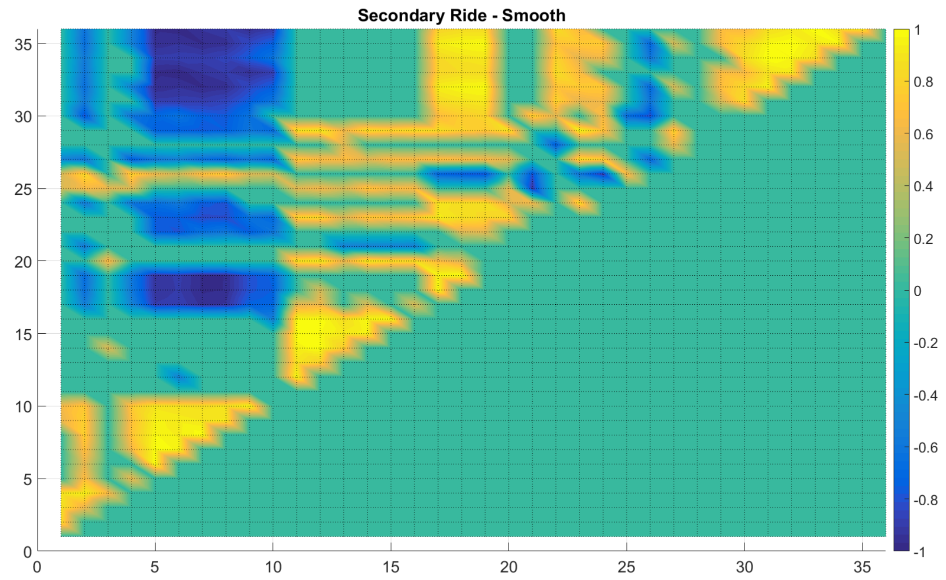

Figure 5.20: Correlation between subjective ratings and physical objective metrics for secondary ride flat road.

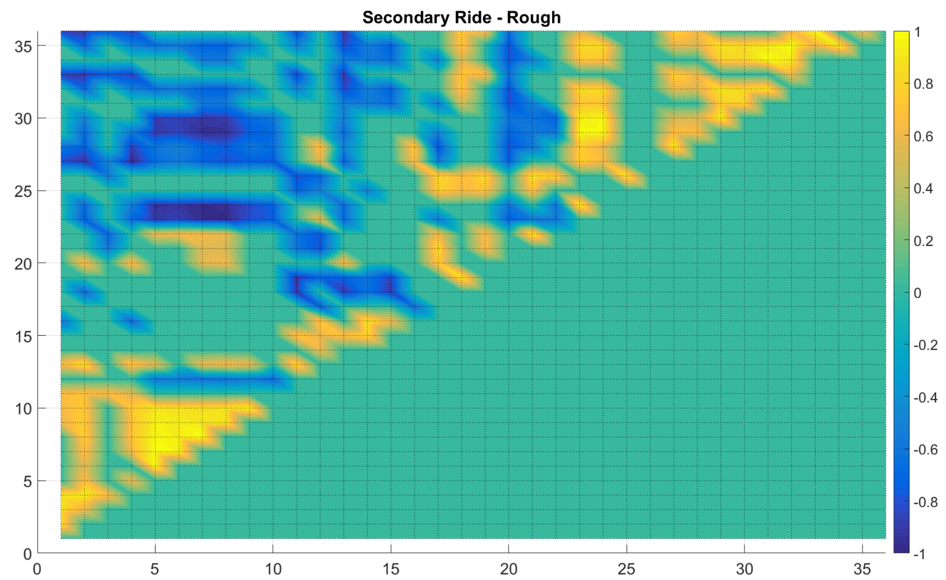

Figure 5.21: Correlation between subjective ratings and physical objective metrics for secondary ride rough road.

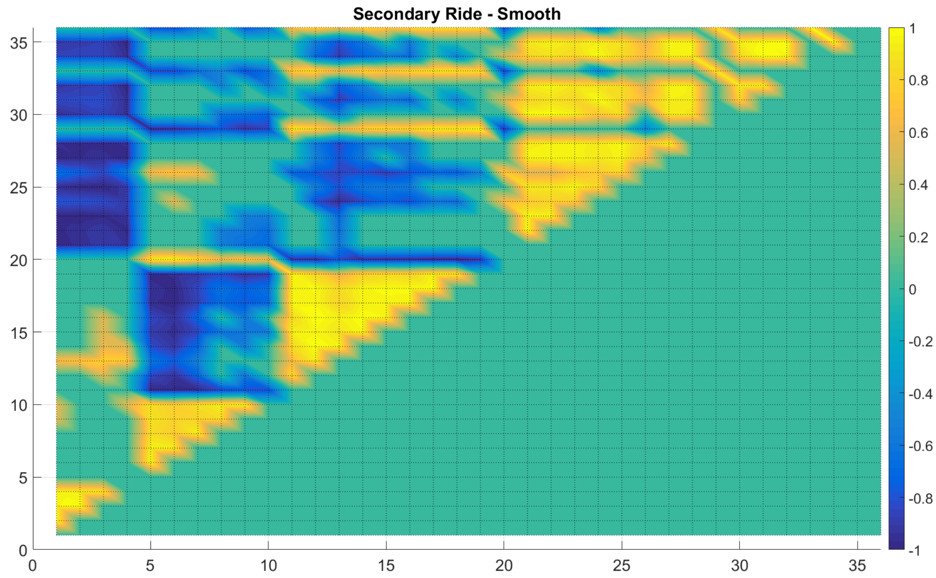

Figure 5.22: Correlation between subjective ratings and simulated objective metrics for secondary ride flat road.

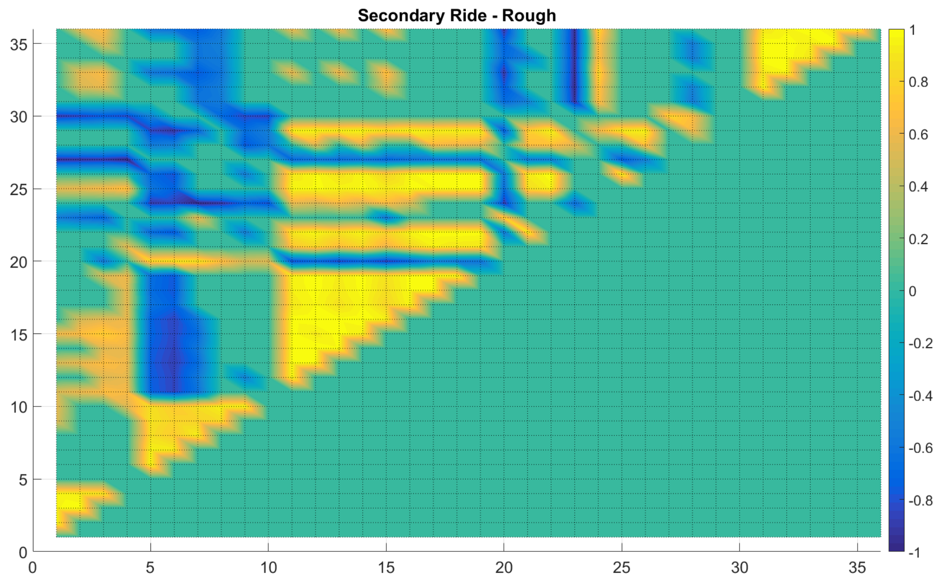

Figure 5.23: Correlation between subjective ratings and simulated objective metrics for secondary ride rough road.

It can be seen in figure 5.20 that there are many different objective metrics all showing strong correlations to the subjective ratings for smooth surface. Many show correlation to all of the subjective ratings and not one in particular, such as the shake rating (6) correlating well to the metrics for weighted choppiness (34), shake (35) and harshness (36). For the rough surface, in figure 5.23, there are less and not as strong correlations but there are still significant correlations.

Table 5.6: Correlation coefficients for secondary ride.

Subjective	**Objective**	**Phys. corr.**	**CM corr.**
Smooth surface			
Choppiness	Choppiness	-0.894	-
Choppiness	RMS vertical acc.	-0.935	-0.940
Shake	Shake	-0.938	-
Shake	RMS vertical acc.	-0.901	-0.987
Harshness	Harshness	-0.932	-
Harshness	RMS vertical acc.	-0.976	-0.848
Rough surface			
Choppiness	Choppiness	-0.582	-0.557
Choppiness	Pitch horizon	-0.957	-0.932
Shake	Shake	-0.504	-
Shake	Pitch horizon	-0.825	-0.725
Harshness	Pitch horizon	-0.739	-0.811

Looking at table 5.6 the stronger correlations can be seen. Common for the smooth surface is that the rating correlates well to the rating for the same metric, choppiness rating to weighted choppiness metric for instance. However, the RMS of the vertical acceleration also correlates well to all of said subjective ratings. The correlation to the RMS is also found from the simulation results, but as mentioned it is not the same track and the model does not capture the higher frequency movements correctly. For the rough surface the ratings does not correlate as well to their respective metric as for the smooth surface. Instead the strongest correlation found is to the absolute pitch metric and this correlation is also found from the simulation results. Linear approximations with confidence intervals as previously stated are shown in figure 5.24 and 5.25 as well as in Appendix C.

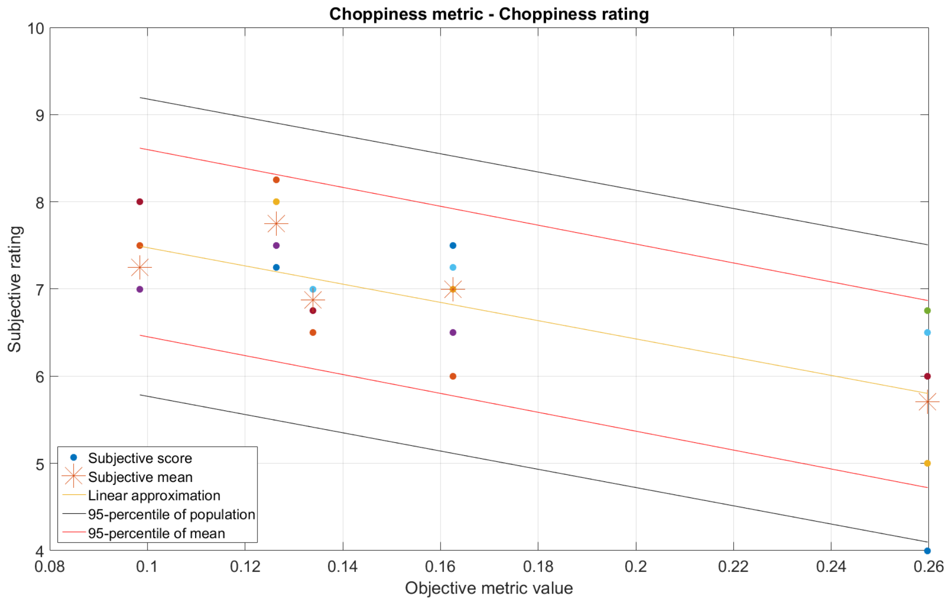

Figure 5.24: Linear approximation of subjective choppiness rating and physical choppiness metric for secondary ride smooth surface road.

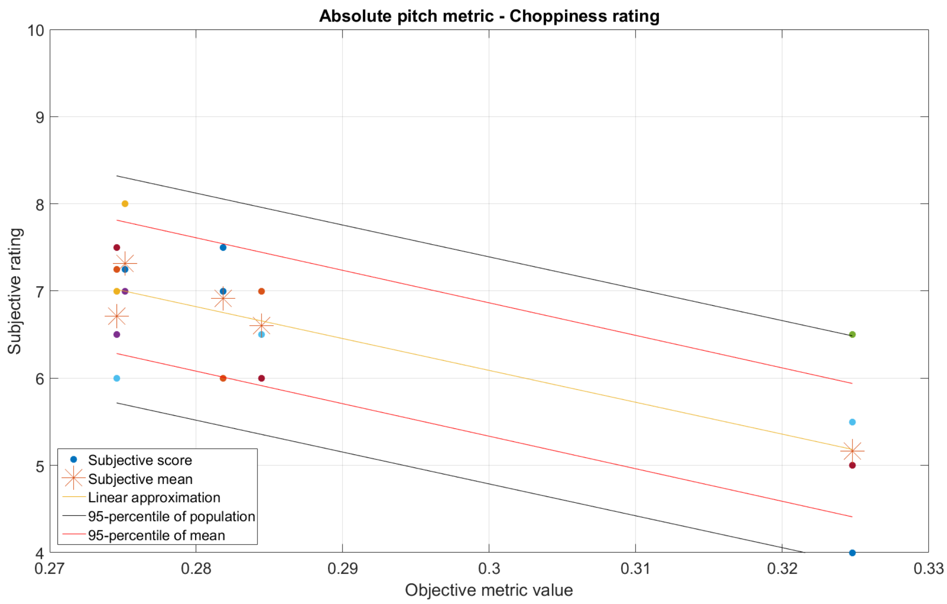

Figure 5.25: Linear approximation of subjective choppiness rating and physical absolute pitch metric for secondary ride rough surface road.

6 Discussion

Contrary to initial expectations, utilizing a car equipped with a fully active suspension proved not to be possible at all throughout the duration of the project. Therefore, achieving the initial objectives and, subsequently, the title of the thesis itself is ultimately challenging. Had the fully active car been available it would naturally have been preferred to do the same evaluations and another correlation study with it and compare it to the semi-active and passive cars. Further development of the method would be to combine ride and handling attributes to increase passenger comfort even further. This can include highly specific procedures such as preventing an open cup of coffee from spilling while cornering, meaning that ride criteria might change when including lateral dynamics. In this scenario the car would ideally not stay flat but instead tilt inwards, similar to a motorcycle leaning, thus balancing out the relevant forces in order to keep the coffee in the cup. This is the opposite of a passive car which would tilt outwards. Other specific tests could include countering pitching movements while accelerating and braking and driving over typical speed bumps and potholes, as commonly found in urban environments with the occupants, ideally, not noticing the obstacles.

Only expert drivers (vehicle dynamics engineers) drove the cars during the physical testing along with both thesis students which can be considered amateurs. While not having a professional experience with ride evaluation, the students are experienced drivers with academic knowledge and understanding of vehicle dynamics. Due to license restrictions at the Hällered Proving Ground there was no possibility of including amateur drivers with limited or no engineering background to represent a general road user and customer. This means that the results reflect the opinions of those professionals and not the opinions of the average customer. To find the opinions of the customers another subjective evaluation could be made with non-professionals which represent these better. A question was raised in the latter stages of the project to include "blind" tests of each car, i.e. not informing drivers which type of car, chassis and suspension was being tested at each time. Since 10 out of the 12 test drivers were full-time employees and vehicle dynamics engineers at Volvo Cars it was considered irrelevant as they could figure out which car they were driving with little difficulties.

Although the majority of the test procedures went smoothly some deviations were discovered at various stages. Some could be mitigated while others could not. Objective tests on Car 5 (Preview MPC) were performed while still on winter tyres rather than summer tyres as the other cars, which affects secondary ride results due to their differences in structure and compound. In order to achieve consistent results the subjective ride evaluation was also performed with winter tyres on Car 5 and summer tyres on the remaining cars. By the time the third test driver drove Car 2 it was reported that abnormal wheel imbalance was felt at a certain speed. That test run was aborted and the car taken to a tyre work shop. Upon inspection a crack in the right rear wheel was discovered, generating a significant amount of radial force variation, which explains the vibrations felt by the driver. Replacement wheels were fitted and tests then continued, but due to the time it was unavailable fewer drivers had the chance to drive it. With a vibration like this the primary ride scores are still useful but the secondary ride evaluation is compromised.

Certain discrepancies between simulated and physical data can be observed. Some can be caused by minor differences in vehicle specifications between the actual vehicles and the CarMaker models such as wheel and tyre sizes, engine and trim configurations and in some cases chassis and suspension tuning. The tuning of the software controlling the semi-active suspension systems, including the preview system, is in constant development and is not always identical to the tuning of the simulated systems. This causes differences between the simulated and the objective results.

Different correlation methods could have been used instead of just the linear regression model. For instance could a method including a saturation limit towards a full subjective score, a 10/10, be explored as it is increasingly more difficult to achieve a perfect score and since the score cannot go any higher than that. Other non-linear or more complex correlation approaches can also be explored. One approach which can be used to find correlations is through the use of neural networks, for instance a self-organizing map could be used. This has been done previously in a study relating to steering and handling [19] but not in studies regarding ride. This is a subject for further investigation in order to improve and create an even more complete method. Another aspect is to improve the simulations to produce results closer to the physical world. This would decrease the discrepancies between the two sets of objective results and thus improve the accuracy in the method. In simulation a model of the human body could also be added to attempt to predict the subjectively experienced comfort more accurately than from measurements taken from the vehicle.

As ride evaluation is generally categorized in three categories: primary ride, secondary ride and impact harshness, an obvious extension of this research would be to examine impact harshness as only the primary and secondary ride were taken into account here. Furthermore, the physical test procedures can be made more precise to allow for more accurate correlation analysis. Instead of giving the drivers a combination of roads to drive on for the evaluations, a single stretch of road can be chosen for the driver to rate a specific attribute. Its subjective rating can then be directly compared to the objective measurements taken on the exact same stretch of road. This would mean that all drivers would have a very similar input from the road to the car which means they all rate very close to the same behaviour in the car. This can, in turn, reduce the spread in the ratings which gives a lower standard deviation and thus more conclusive results. The subjective evaluation form can also be changed to include more and different evaluation points. The one used was limited in order to keep the time of each test at a suitable level, but for instance an overall ride rating could be added. A reference vehicle which all drivers were to drive first could also be added to give all drivers the same baseline.

Another factor affecting the large standard deviations and the correlations is the fact that the data set used is small. It is small both in terms of amount of drivers having driven each car and in terms of the amount of cars. The more evaluations per car the more weight the mean value of the ratings carries. Similar for the cars, where it would have been desired to include more cars to increase the certainty levels of the correlations. This can be done by including cars which have a good spread in the objective metrics, but the amount of data used can still catch the overall trends.

7

Conclusion

A functional method has been developed to objectively quantify vehicle's ride quality and thus the performance of the suspension system. Fundamentally, it helps translate people's subjective assessments of vehicle's ride quality to objective, quantifiable metrics. Some correlation was found for comfort and control rating between the objective and subjective tests, thus it is possible to estimate a subjective rating using data from objective tests. According to these the roll metric should minimized to achieve good control rating for small amplitude primary ride and minimize peak head jerk for comfort rating. For the large amplitude the peak vertical acceleration should be minimized from physical testing in order to score well on control rating. For a good comfort rating the head displacement should be kept low. For the secondary ride, especially on smooth surface, the aim should be to minimize the choppiness, shake and harshness metrics and on a rough surface the pitch horizon metric is to be kept as low as possible to achieve a good rating. Also one can see that the RMS value of the vertical acceleration does not give the best correlations to the subjective ratings which means it is not the best metric to use for ride evaluation. Although not fully complete, the method is still in development and merits further work. A larger data set of drivers and test vehicles would be ideal as well as ensuring that every driver would drive each car in order to increase certainty of the results. Combining that with focusing the test procedures even further, as mentioned in the discussion chapter, stronger correlation for more metrics could be expected. What is clear is that based on these results the preview system ensures improved primary ride whereas its benefits are not as clear in terms of the secondary ride.

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A

Appendix 1

A.1 Simulation results

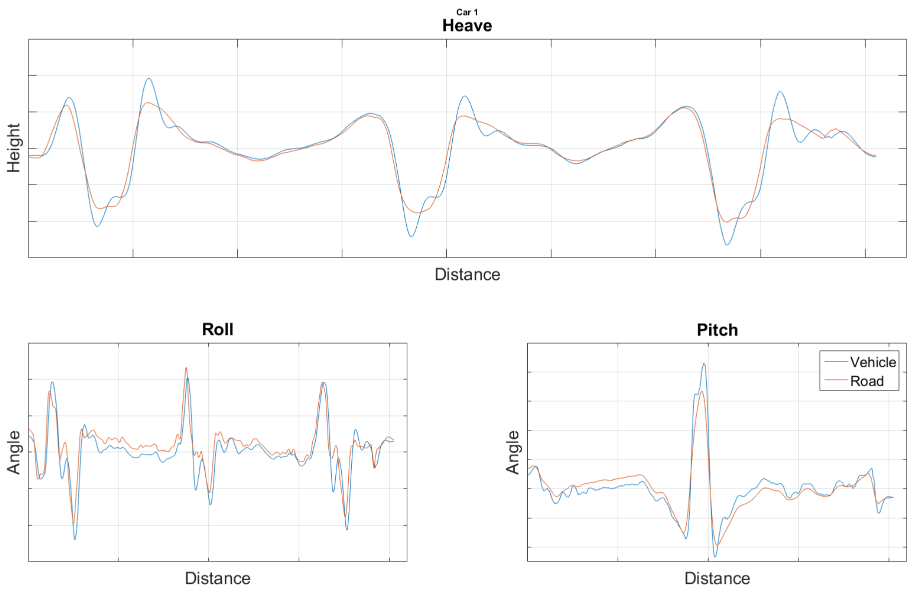

Figure A.1: Simulation results of heave, pitch and roll for car 1 (passive) vehicle on primary ride large amplitude track.

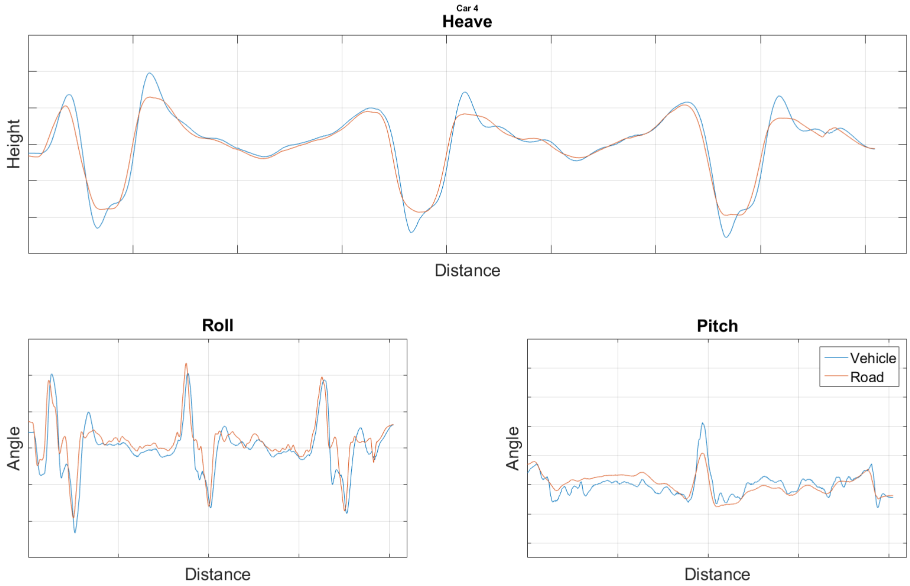

Figure A.2: Simulation results of heave, pitch and roll for car 4 (skyhook) vehicle on primary ride large amplitude track.

A.2 Physical results

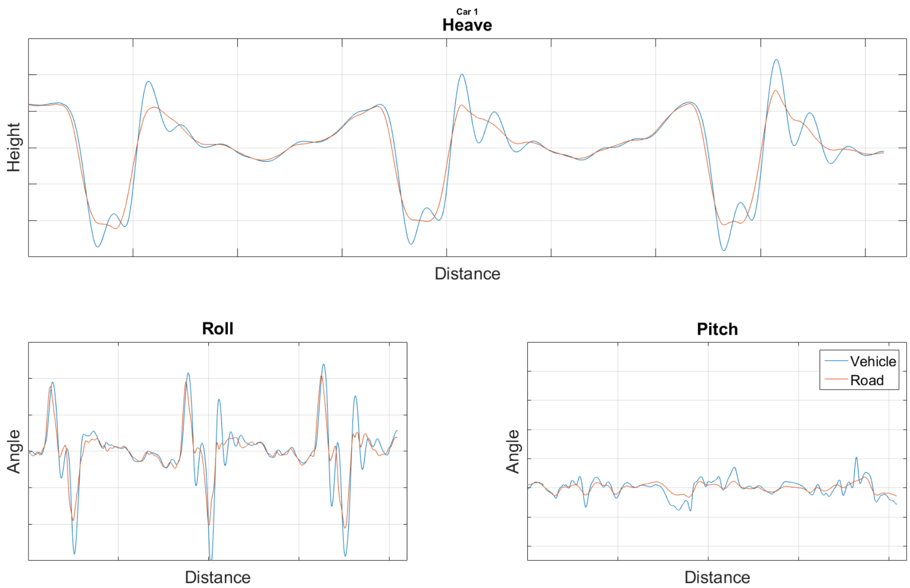

Figure A.3: Heave, pitch and roll of car 1 (passive) from physical test.

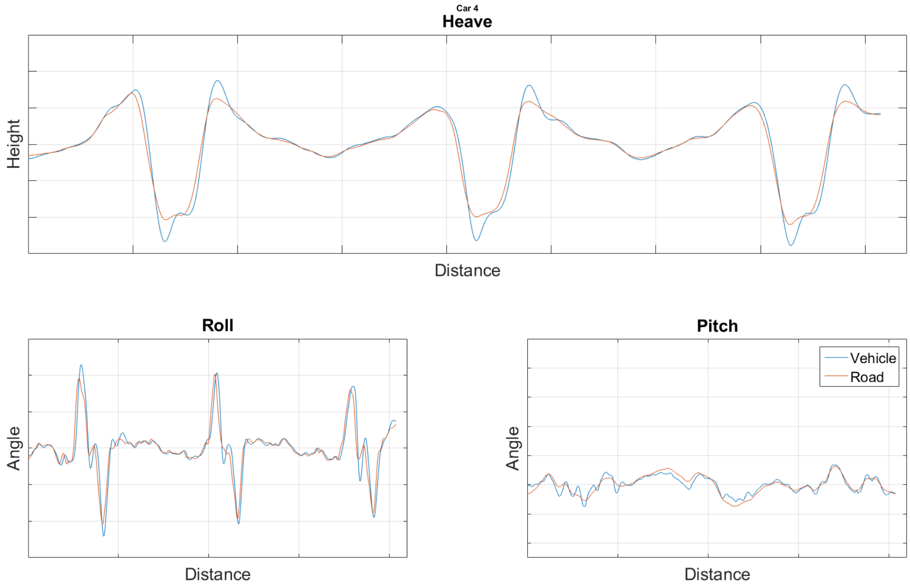

Figure A.4: Heave, pitch and roll of car 4 (skyhook) from physical test.

B

Appendix 2

B.1 Divided per car

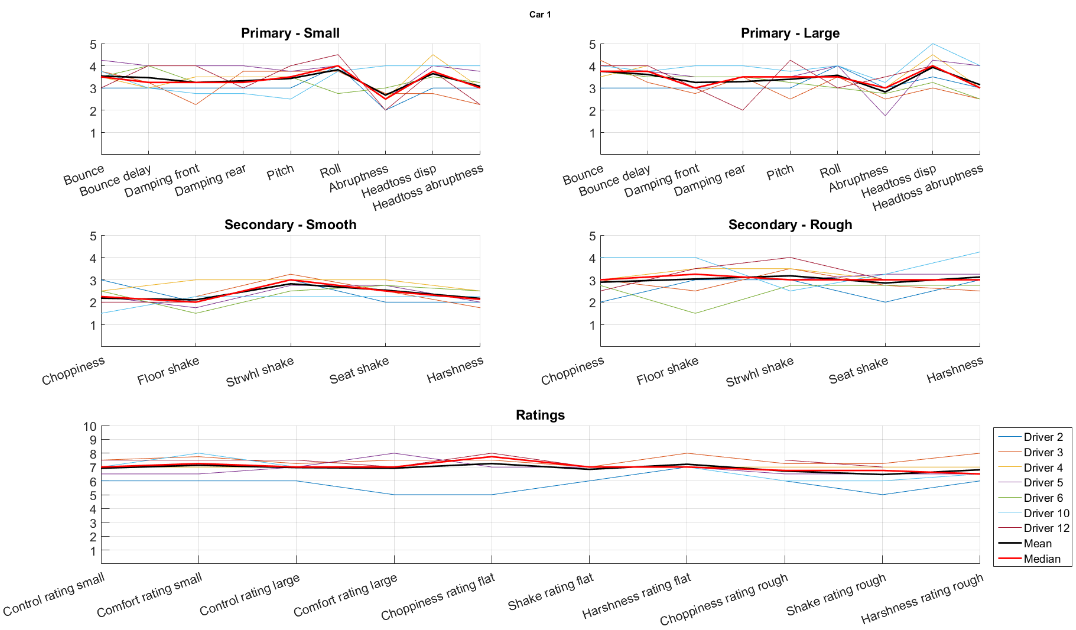

Figure B.1: Subjective scores for car 1.

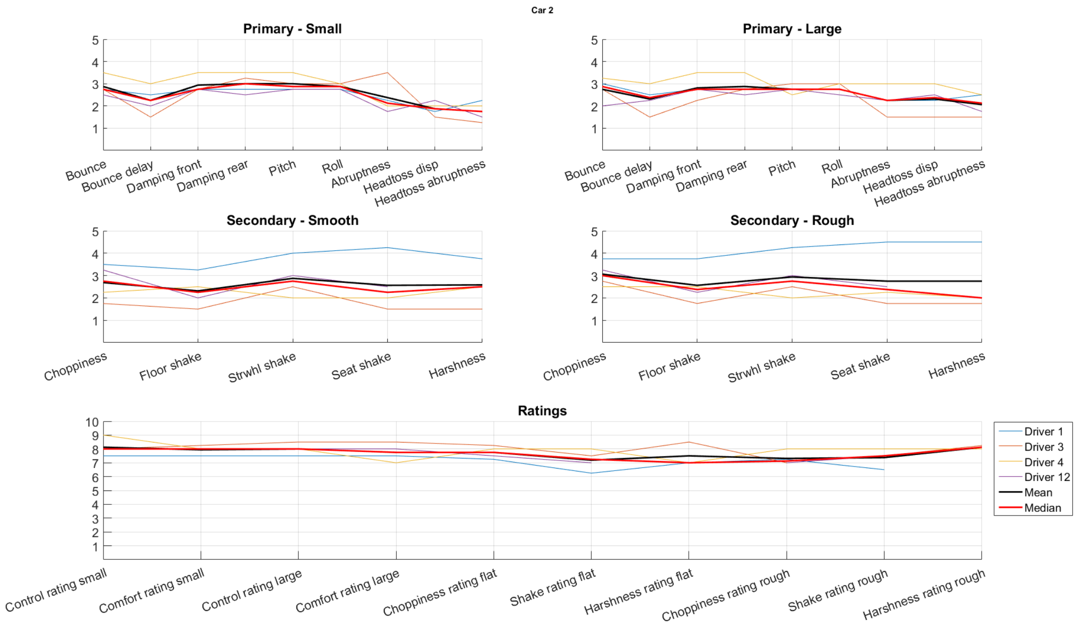

Figure B.2: Subjective scores for car 2.

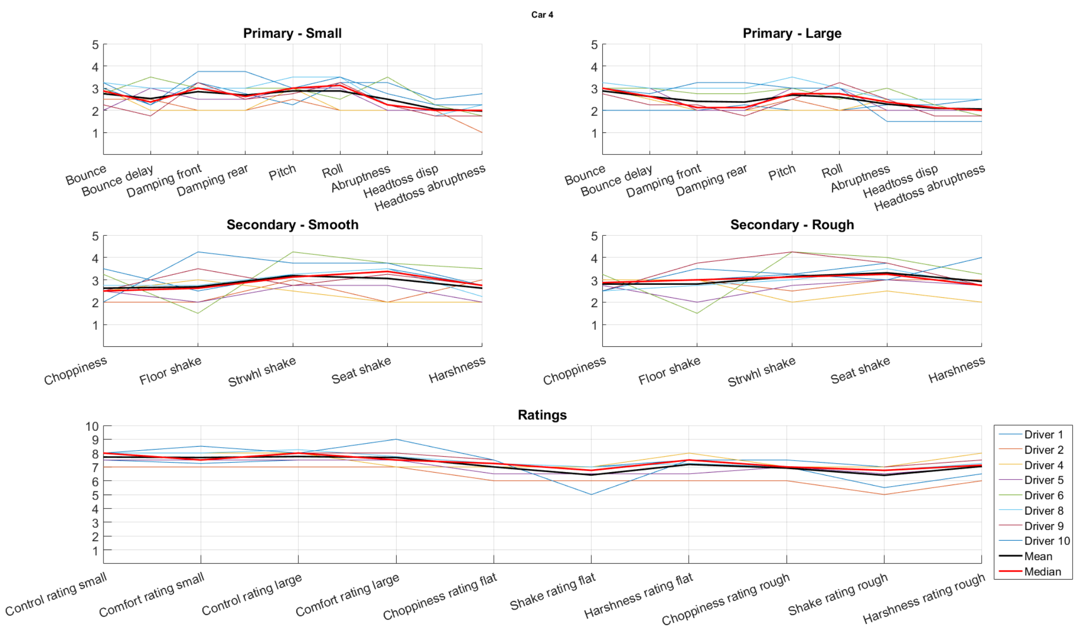

Figure B.3: Subjective scores for car 4 (skyhook).

B.2 Divided per driver

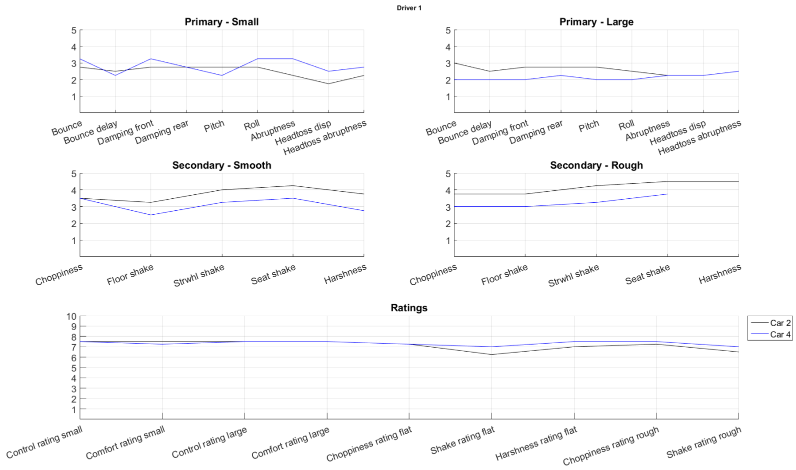

Figure B.4: Subjective scores set by driver 1.

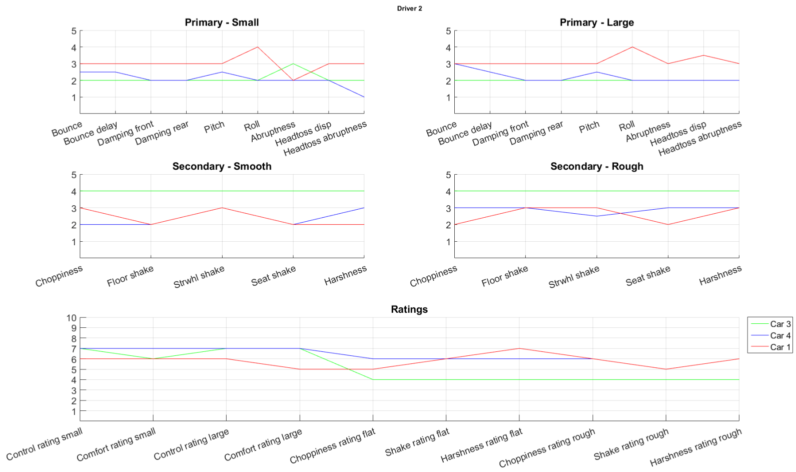

Figure B.5: Subjective scores set by driver 2.

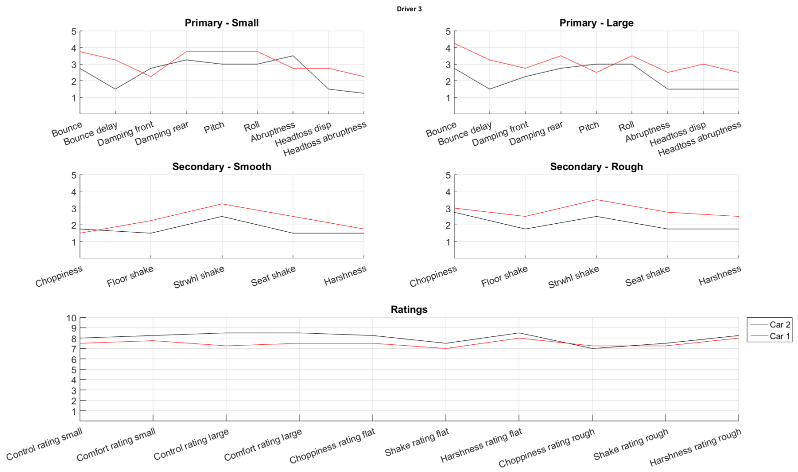

Figure B.6: Subjective scores set by driver 3.

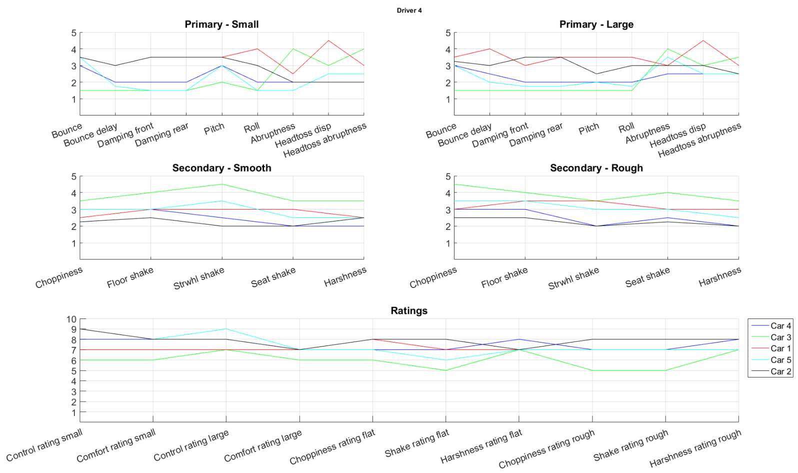

Figure B.7: Subjective scores set by driver 4.

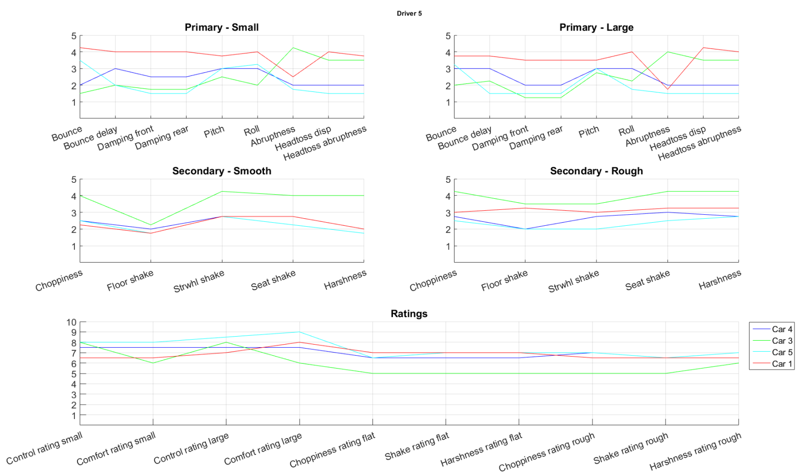

Figure B.8: Subjective scores set by driver 5.

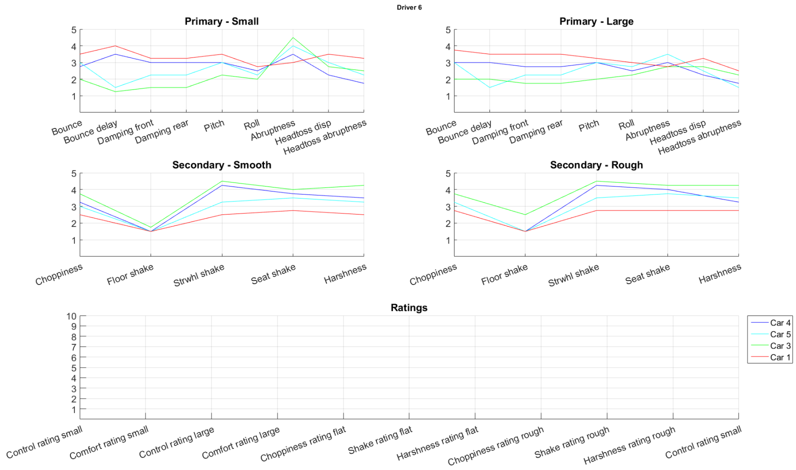

Figure B.9: Subjective scores set by driver 6.

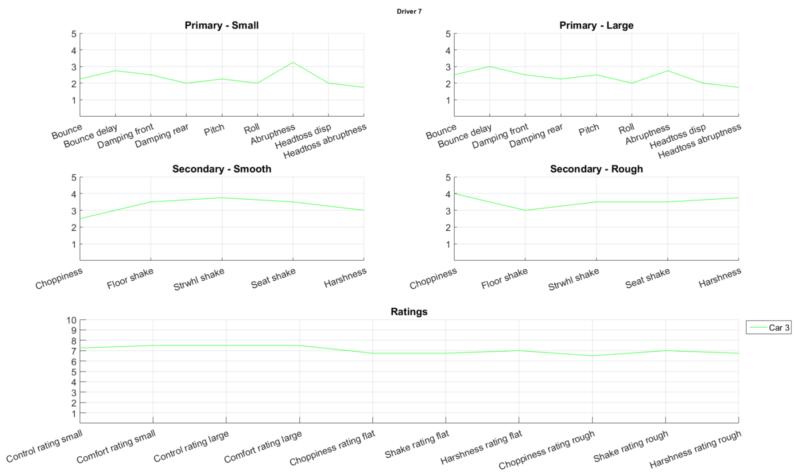

Figure B.10: Subjective scores set by driver 7.

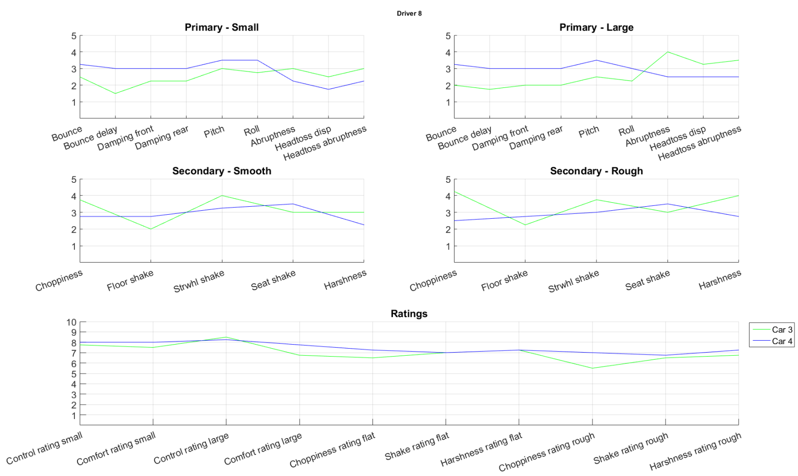

Figure B.11: Subjective scores set by driver 8.

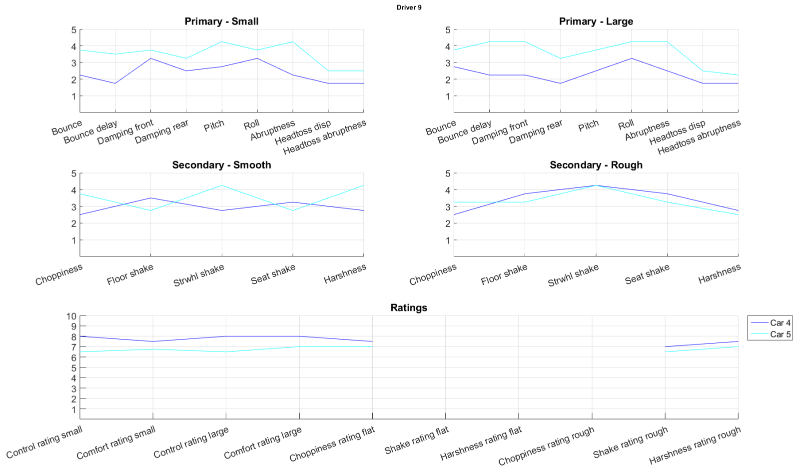

Figure B.12: Subjective scores set by driver 9.

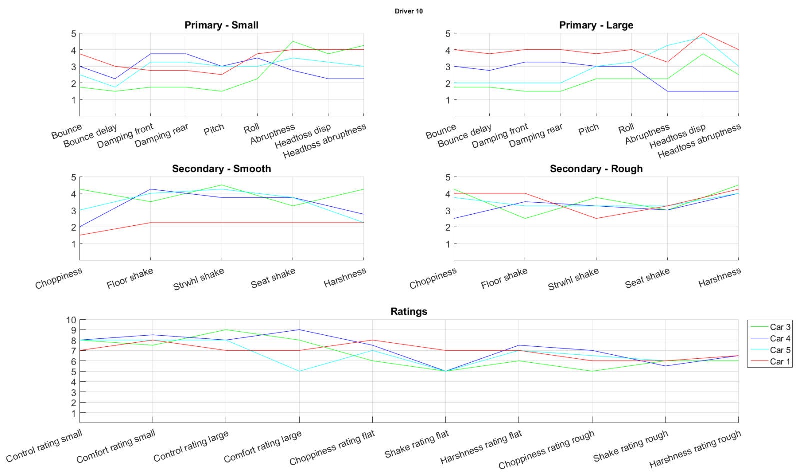

Figure B.13: Subjective scores set by driver 10.

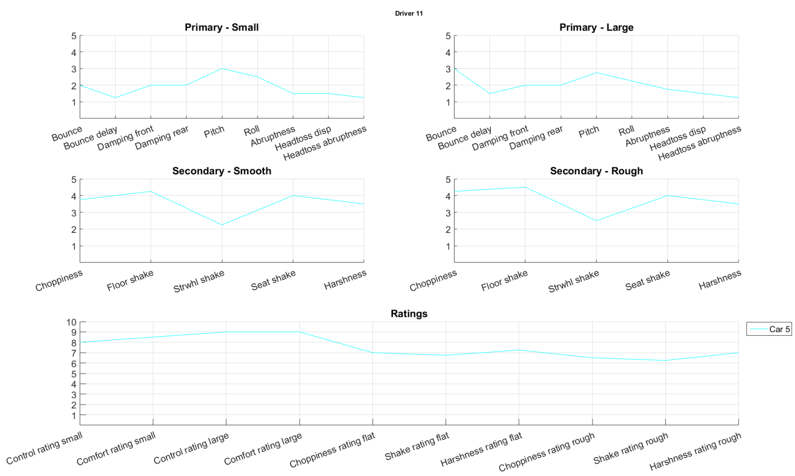

Figure B.14: Subjective scores set by driver 11.

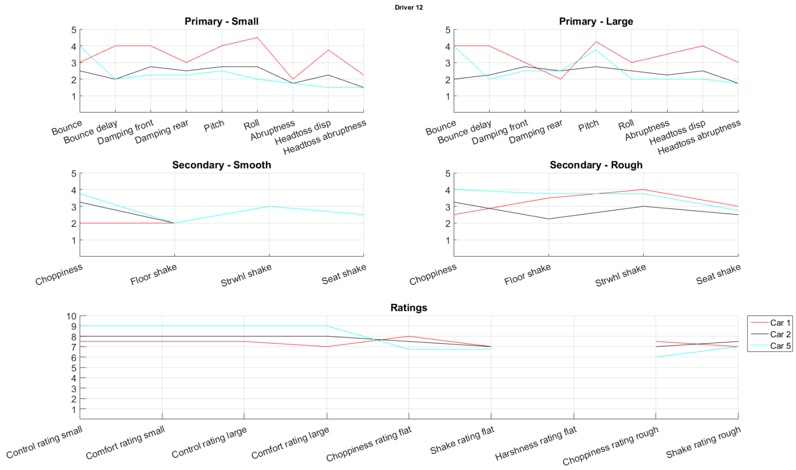

Figure B.15: Subjective scores set by driver 12.

C

Appendix 3

C.1 Linear approximations for primary ride

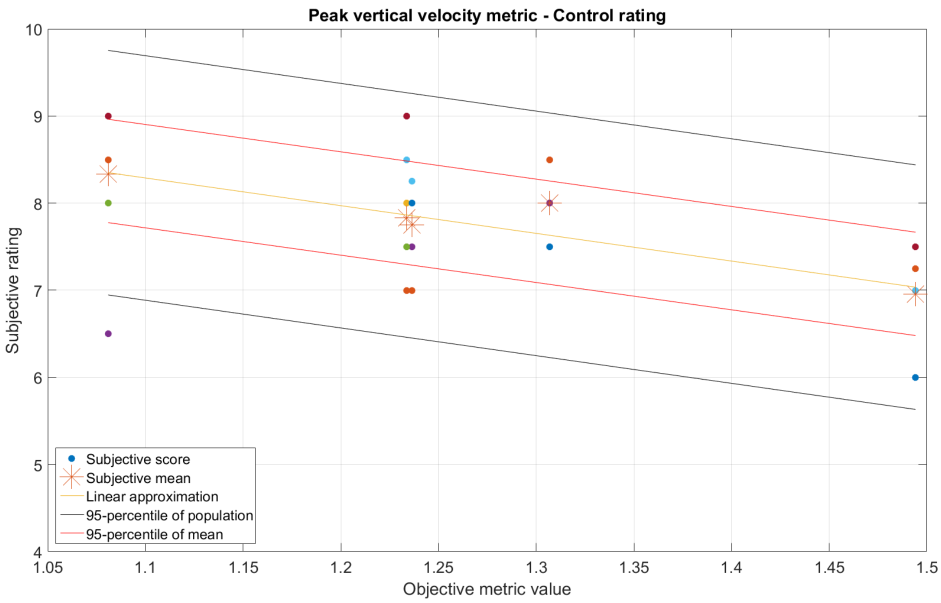

Figure C.1: Linear approximation of subjective control rating and physical peak vertical velocity metric for primary ride large amplitude road.

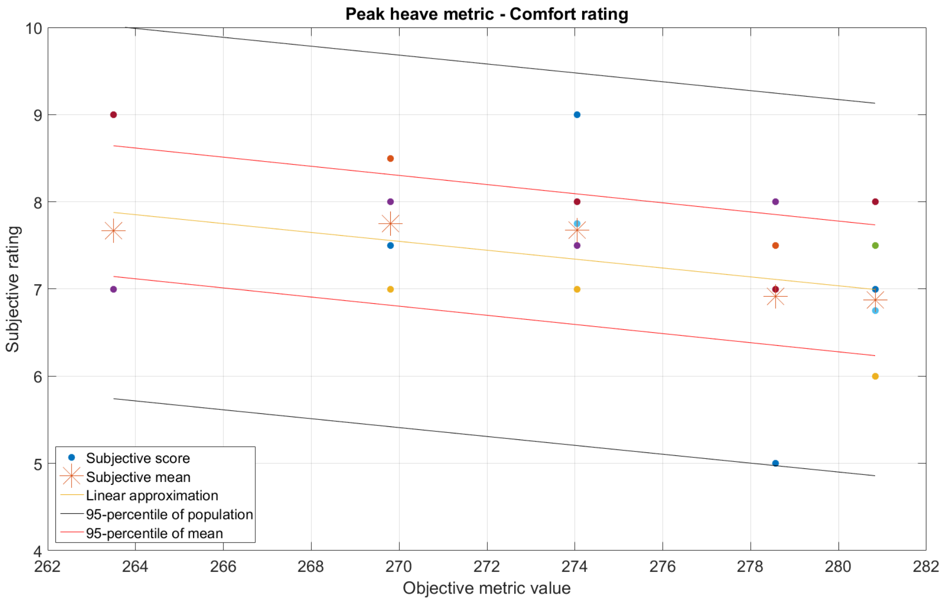

Figure C.2: Linear approximation of subjective comfort rating and physical peak heave metric for primary ride large amplitude road.

C.2 Linear approximations for secondary ride

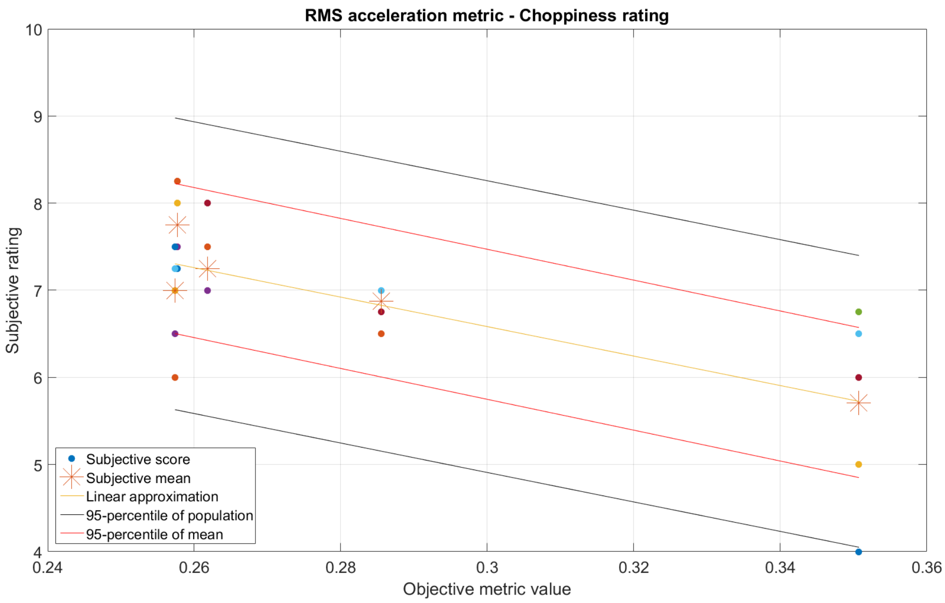

Figure C.3: Linear approximation of subjective choppiness rating and physical RMS acceleration metric for secondary ride smooth surface road.

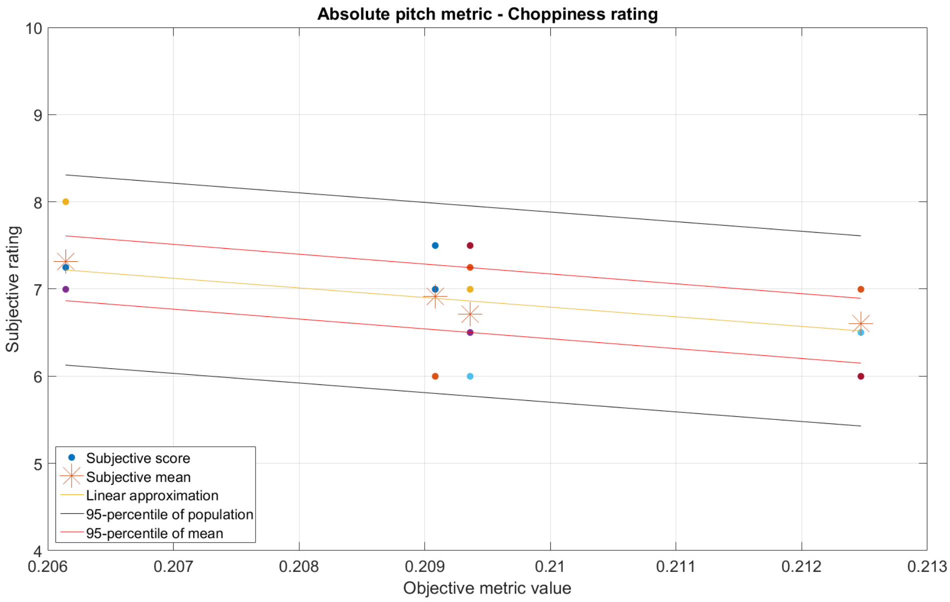

Figure C.4: Linear approximation of subjective choppiness rating and simulated absolute pitch metric for secondary ride rough surface road.