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Objective tuning of semi-active dampers

Method development for ride metric optimisation utilizing a four poster rig

Master's thesis in Mobility Engineering

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DEPARTMENT OF MECHANICS AND MARITIME SCIENCES
CHALMERS UNIVERSITY OF TECHNOLOGY

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Cover: Rig setup for optimisation of damper settings, complete vehicle connected to computer.

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Abstract

Active systems in premium passenger cars are gaining popularity which in turn leads to higher demands on these systems to perform. One such system are semi-active dampers which can replace passive dampers and is a big part in how the suspension performs. Traditionally, semi-active dampers are tuned by experienced engineers with focus on subjective assessment and understanding of vehicle dynamics. Test driving and analysing data lets the engineers tune the specific system to a desired functionality. To achieve an overall better result for semi-active damper systems it would be effective to extend the objective capabilities of the tuning process. This is exactly where this project finds its purpose. This project aims to create a method that makes it possible to objectively tune semi-active dampers utilizing a four poster rig. Focus was on primary ride comfort, looking at the level of damper control and ride abruptness. To do so a metric for primary ride control was found and validated on the four poster rig. Optimisation of semi-active damper parameters was investigated in CAE and transferred to the four poster rig. A complete vehicle was setup on the rig and the semi-active damper ECU was connected to a laptop. The laptop measured and updated the vehicle according to patternsearch optimisation while the vehicle got excited by a recorded. Different target functions and initial guesses were investigated, which resulted in three interesting outcomes. The optimisation worked well on the rig and it ran continuously without human input. To conclude the project, a metric was found and validated and an objective tuning method was found. With that conclusion the aim of the project was satisfied.

Keywords: Semi-active, damper, tuning, subjective, objective, four, poster, rig, active, systems, primary, ride, control, abruptness.

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Jesper Ramsberg, Gothenburg, June 2023

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

CCD	Continously controlled damper
COG	Center of gravity
DOF	Degree of freedom
GA	Genetic algorithm
PS	Pattern search
PSO	Particle swarm optimisation

Nomenclature

Below is the nomenclature of indices, sets, parameters, and variables that have been used throughout this thesis.

Indices

i,j	Indices for parameter position
t	Index for time step
$\ddot{x}$	Indicates time derivative

Parameters

M	Sprung mass
m	Unsprung mass
C	Spring coefficient
D	Damper coeffcient
ζ	Level of critical damping
l	Longitudinal length of vehicle
l_f	From cog to front wheel
l_r	From cog rear wheel
w	Width of vehicle

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Introduction

1.1 Background

Active systems in premium passenger cars are gaining popularity. These are systems that actively adapt to the driving scenario and environment. One such system are semi active dampers used in vehicle suspension, sometimes called Continuously Controlled Dampers(CCD). A CCD damper will continuously change its characteristics to perform as well as possible for the current driving scenario. This makes it possible to have a comfortable car while not comprising as much in handling as could have been traditionally required. During development of a chassis with CCD experienced vehicle dynamicists will use their vehicle dynamics understanding together with subjective input from test driving to design the damper characteristics. An objective method to tune the characteristics of CCD can complement the subjective method in such a way that the overall result becomes better. Such a method could also be implemented on a test rig where engineers can judge the behaviour of the vehicle and tune in a strictly controlled environment.

1.2 Aim

This project aim to develop a method to objectively tune CCD which can be applied on a four poster test rig. This aim may be broken down into sub aims.

1. Understanding vehicle behaviour in terms of primary ride comfort. Thus understanding primary ride control and abruptness.

2. Establish a metric that describes the amount of primary ride control.

3. Validate this metric objectively and subjectively using a four poster rig.

4. Establish a method that tunes the semi active dampers according to this metric.

1.3 Limitations

The project will limit its focus to comfort and ride quality, specifically primary ride motion and abruptness. CCD can be used to maximise road to tyre grip and handling dynmaics, but that is outside the scope of this project.

Since the aim is method development, the project is not expected to output a production ready tuning but rather a framework for finding a tuning set according to objective data. This means that the amount of parameters that are taken into account will be reduced compared to a production ready vehicle. This is done to keep focus on the method development and speed up iterations when designing the method.

The tuning found when validating the method will be tested on a real life vehicle on a test track but on road testing will not be part of the method development. Simulations will help cover larger data sets on simpler model and rig tests will investigate full vehicle behaviour but with smaller data sets.

The studies in this master thesis has a focus on the relative differences when it comes to parameter settings, subjective ratings and vehicle behavior. This removes the need to make simulation models that match the real world vehicle used in tests. It is the relative results that are of interest and it was understood during the project that the use of absolute results did not add sufficient value. The focus can be kept on creating a good method while keeping simulation models simpler.

1.4 Ethical and Ecological aspects

One interesting topic to tackle with different levels of active suspension is overspeeding. As the suspension becomes more and more sophisticated the driver will feel a greater sense of confidence in the car. Normal sized speed bumps can be driven over with ease at higher speeds than what passive cars can ever manage. This could lead to overspeeding where it was earlier not possible due to bumps. Over speeding is often regarded extra dangerous in residential or school areas where visibility often is low, short distances to react and a lot of people moving around. And it is in these areas where active damping can be effictive in improving comfort due to the low speed and uneven road quality. More comfort could lead to the driver being less hesitant about the vehicle speed and therefore overspeeding.

On the other hand some negative health effects could be minimised by active suspension. ISO have standardised vibration and shock impacts on humans in ISO 2631 (Jacobsson, 2021b). Exposure time and frequency range have impact on human comfort. With active or semi active suspension it could be possible to avoid these harmful frequency ranges to a further extent, leading to less pain and health problems.

2

Theory

This chapter will introduce useful tools and information for this project. Firstly two vehicle models are introduced. They make it possible to simulate parts of the vehicle behavior, which important for ride comfort in heave, pitch and roll. Secondly some information about how semi-active dampers work and how they can be used to improve ride quality.

2.1 Quarter car model

The quarter car model is useful when analysing vertical motion on a passenger car. It represents the vehicle in a simple manner while still describing suspension characteristics in a good way.

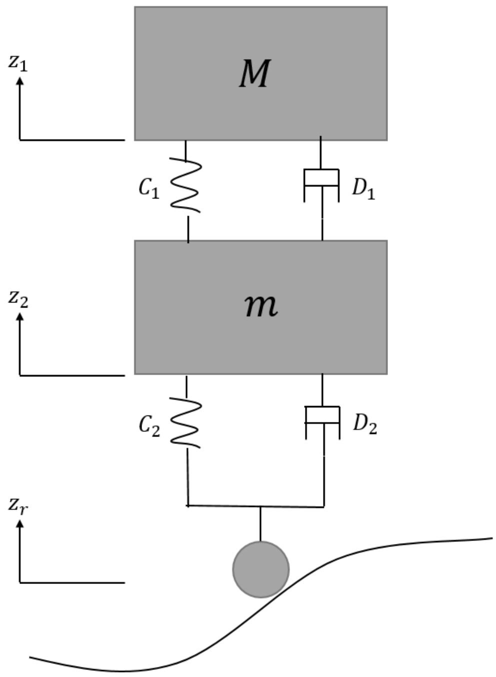


Figure 2.1: Quarter car model

The mass M represents the sprung mass, the small mass m represents the unsprung mass. C_1 and D_1 are suspension spring and damper coefficients. C_2 and D_2 are tyre spring and damper coefficients. z_1, z_2 and z_r are the vertical dislpacements of sprung mass, unsprung mass and road.

$$M\ddot{z}_1 = c_1(z_2 - z_1) + d_1(\dot{z}_2 - \dot{z}_1) \tag{2.1}$$

$$m\ddot{z}_2 = c_1(z_1 - z_2) + d_1(\dot{z}_1 - \dot{z}_2) + c_2(z_r - z_2) + d_2(\dot{z}_r - \dot{z}_2) \tag{2.2}$$

The quarter car model is a relatively simple model that can be simulated quickly in for example Python or Matlab/Simulink. This makes it perfect for comparing many different settings of the parameters as well active control. It enables good relative studies, but lack the detail that would be needed to capture the full characteristic of real car, for example the pitch motion.

2.1.1 Natural frequency

Choosing a spring stiffness that achieves a good model behaviour can be a difficult task, therefore it is better to choose stiffness according to a decided natural frequency of the system. By doing so, the behaviour of the system is decided and the spring stiffness follows from that. Trying to get the behavior right by altering spring stiffness without a particular natural frequency in mind might require more testing to get the desired behavior.

$$w_n = \sqrt{\frac{c}{M}}\ rad/s \tag{2.3}$$

$$f_n = \frac{w_n}{2\pi} = \frac{\sqrt{c/M}}{2\pi}\ Hz \tag{2.4}$$

The two stiffnesses are then decided according to equation (2.5) and (2.6)

$$c_1 = (2\pi f_{n,sprung})^2 \cdot M \tag{2.5}$$

$$c_2 = (2\pi f_{n,unsprung})^2 \cdot (M + m) \tag{2.6}$$

2.1.2 Critical damping

After the spring stiffness is decided, the level of damping can be decided. This is done by deciding on a level of critical damping ζ.

$$\zeta = \frac{d_{actual}}{d_{critical}} = \frac{d_{actual}}{2\sqrt{c \cdot M}} \tag{2.7}$$

$$d_1 = \zeta \cdot 2\sqrt{c_1 \cdot M} \tag{2.8}$$

$$d_2 = \zeta \cdot 2\sqrt{c_2 \cdot (M + m)} \tag{2.9}$$

Setting the damper level of the suspension damper according as a level of critical damping becomes especially interesting when looking at the skyhook principle.

2.2 7 Degree of freedom vertical model

A 7 degree of freedom(DOF) model for vertical dynamics is a powerful tool that takes much more detail into account compared to the quarter car model without increasing the complexity. The fact that heave, pitch and roll is captured makes it useful for ride comfort analysis.

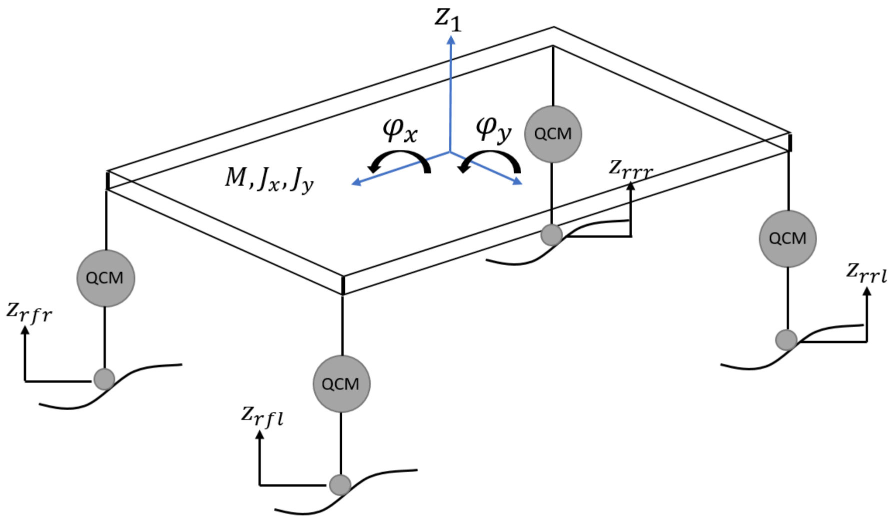


Figure 2.2: 7 Degree of freedom vertical model

The model presented in figure 2.2 presents the 7 DOF model in a way that is useful for this report. Each corner of the model includes a full quarter car model, presented i section 2.1. That is because all the aspects of the quarter car model is interesting while also adding on pitch and roll behavior.

The body motion equations describes how the sprung mass M will move when excited by the suspension forces $F_{s,ij}$. Pitch torque is calculated as the torque around the y-axis using the pitch inertia J_y, the distance from CoG to the front axle l_f and the rear axle l_r. Roll torque is calculated around the x-axis with the track width w and roll inertia J_x. Heave force is calculated from the sum of all suspension forces where positive is upwards.

$$M\ddot{z}_1 = \sum F_{s,ij} \tag{2.10}$$

$$J_y\ddot{\varphi}_y = \sum F_{s,rj} \cdot l_r - \sum F_{s,fj} \cdot l_f \tag{2.11}$$

$$J_x\ddot{\varphi}_x = \sum F_{s,il} \cdot \frac{w}{2} - \sum F_{s,ir} \cdot \frac{w}{2} \tag{2.12}$$

Each quarter of the car will produce a suspension force according to road input and sprung mass corner height.

$$z_{1,fl} = z_1 - l_f\varphi_y + \frac{w}{2}\varphi_x \tag{2.13}$$

$$z_{1,fr} = z_1 - l_f\varphi_y - \frac{w}{2}\varphi_x \tag{2.14}$$

$$z_{1,rl} = z_1 + l_r\varphi_y + \frac{w}{2}\varphi_x \tag{2.15}$$

$$z_{1,rr} = z_1 + l_r\varphi_y - \frac{w}{2}\varphi_x \tag{2.16}$$

These equations make up the position for each corner and are together with their respective derivatives used to calculate the forces in each quarter of the model. z_1 is the position of the sprung mass in all four corners. As shown in the quarter car model, the suspension force F_s is calculated from the difference in sprung mass and unsprung mass.

2.3 Semi active damper

Semi active dampers adjust their own damper curve according to the driving scenario. A good CCD strategy for a semi active damper can therefore generate good comfort without compromising to much on handling dynamics.

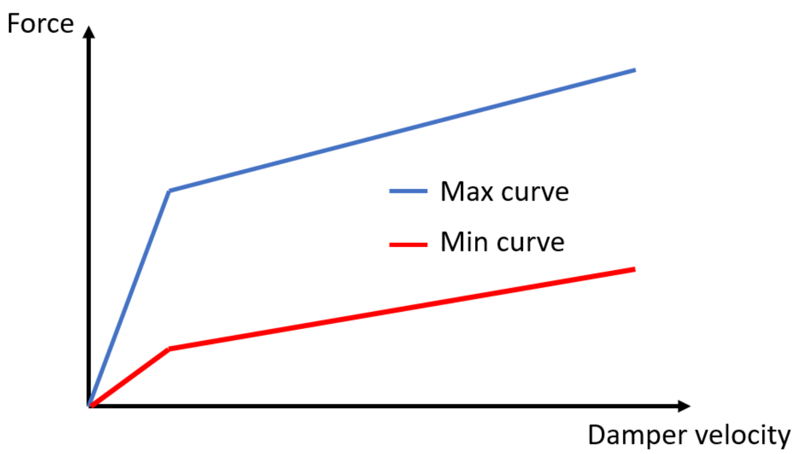


Figure 2.3: Principle picture of semi active damper bandwidth

A semi active damper can use any damper curve as long as it is within the bandwidth of the damper hardware. A principle sketch of such limits is presented in figure 2.3. Compare this to a passive damper which has only one damper curve which is fixed over time, and it becomes clear that the possibilities are much greater for a semi active damper. These systems are generally fast, up to 500 Hz sensor sampling frequency (van der Sande et al., 2022). The speed allows these systems to quickly adapt to vibrations and bigger undulations in the road.

2.4 Skyhook control

Ideal skyhook control can be modelled as an extension of the quarter car model. The idea is that the vehicle body, the sprung mass M is connected to the sky with a damper. The ideal skyhook controller would control the vehicle sprung mass only with sky as reference and not impact the unsprung mass, the suspension damper is replaced with the shyhook (Jacobsson, 2021a). Referring to equation (2.1) and (2.2) and altering them with the skyhook, the following is achieved:

$$M\ddot{z}_1 = c_1(z_2 - z_1) - d_{sh}\dot{z}_1 \tag{2.17}$$
$$m\ddot{z}_2 = c_1(z_1 - z_2) + c_2(z_r - z_2) + d_2(\dot{z}_r - \dot{z}_2) \tag{2.18}$$

In reality the shyhook will replace the normal damper and therefore it will also impact the unsprung mass.

$$M\ddot{z}_1 = c_1(z_2 - z_1) - d_{sh}\dot{z}_1 \tag{2.19}$$
$$m\ddot{z}_2 = c_1(z_1 - z_2) + d_{sh}\dot{z}_1 + c_2(z_r - z_2) + d_2(\dot{z}_r - \dot{z}_2) \tag{2.20}$$

This skyhook setup is however only possible with fully active suspension. Then it would be possible to generate force without suspension movement aswell as force that is opposite to the passive damper force. A semi-active damper is based on a passive damper and can only dissipate energy from the suspension system during damper movement. What the semi active CCD system can do however is work within its limit and minimise damping when required and maximise when required.

One such method of skyhook control that is possible with semi-active damper are what you might call a "bang-bang controller". The thought process of such controller is that four different damper curves are modelled. A maximum and a minimum curve of both compression and rebound. The controller will then choose which curve to apply according to the scenario the vehicle is in.

Table 2.1: Skyhook control logic

	Heave	¬Heave
Comp	Min. compression curve	Max. compression curve
¬Comp	Max. rebound curve	Min. rebound curve

These four damper curves are seen as the limits of what the damper can deliver and they are set according a factor of critical damping. A useful setup is then to use a controller that switches the damper between these limit levels actively according to the logic presented in table 2.1. Of course one could argue that an arbitration between the limit levels of the damper would be more useful and it probably is, but the presented logic makes it possible to analyse skyhook effectively. Effects on the vehicle can be simulated and analysed without making the problem to complex and hard to understand. The logic is only presented for heave but could be applied for pitch and roll as well. An arbitration method would then be necessary for the system to work properly.

To understand what the possibilities with skyhook are, figure 2.4 is presented. In that picture there are three different damper setups visualised along with the road that the quarter car model travels over. The passive system gets the highest overshoots and has more oscillations. The Ideal skyhook has the smoothest movement without any overshoot. The bang-bang controller falls somewhere in between. This highlights that it is possible to achieve a smoother ride without a fully active system. It should be mentioned that all these three are setup with reasonable damper values for each respective system type.

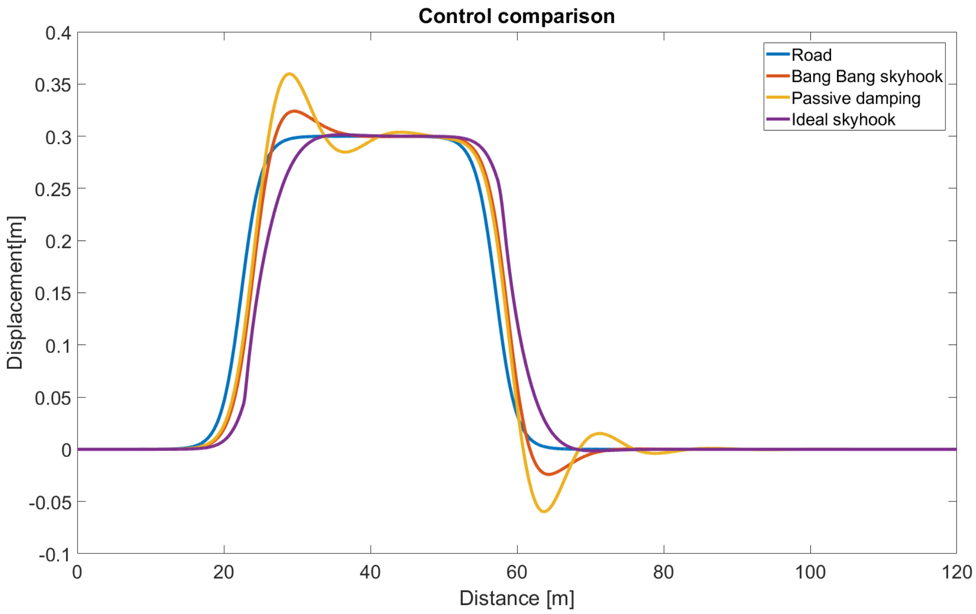


Figure 2.4: Comparison of damper control alternatives

An ideal skyhook controller is fully active and has access to actuators that can add energy to the suspension system. The bang-bang controller can still only dissipate energy from the suspension system. This means that it can only generate a force which is opposite to the damper movement.

2.5 Ride quality

When investigating ride quality for a passenger car there are multiple of ways of looking at it. One approach is looking at the primary ride motion of the vehicle. Primary ride is the motions on the vehicle caused by low frequency, high amplitude disturbances, usually in the range of 1-5 Hz (Arvidsson and Mar Runolfsson, 2019). A primary ride metric aims to capture how the vehicle handles these disturbances.

Another interesting property is the abruptness of the primary ride motion. Abruptness in primary ride can be described as how sharp the switch from positive to negative heave, pitch and roll velocity feels to the vehicle occupants. With too stiff damper tuning the vehicle might behave abrupt in its primary ride motion transitions (W. Neal et al., 2015). An abruptness metric would aim to measure how abrupt the transitions in primary ride movement are.

The combination of these two characteristics is interesting when describing comfort for low frequency, high amplitude body movements. Balancing the two characteristics is an important task in vehicle comfort.

2.6 Patternsearch

Patternsearch is a tool for finding a local minimum of a function without the use of function gradient. By removing the need for function gradient the method will work for non-smooth functions and parameter searches where the parameters are not adding up to a direct function.

Patternsearch optimisation is sometimes described as a family of methods but in this thesis the type applied by Matlab in their optimisation toolbox is used. An initial guess is altered step by step to find a better solution. The alogrithm is explained very well by Mathworks on their website (Mathworks, 2023a).

3

Methods

This chapter will cover how a primary ride metric was chosen and the process for validating that metric. The design process for the rig optimisation method will be presented along with how an on track study concluded the outcome from the rig optimisation.

To firstly establish a useful metric it is important to understand what the goal of the metric is. As the project focuses on primary ride and skyhook design it is useful to see what characterises the ideal skyhook. That was done through simulation of a quarter car model connected to an ideal skyhook controller. It was also compared to a passive damper and a bang-bang setup. The bang-bang setup uses the logic presented in table 2.1. The main primary ride motions that can be examined in a quarter car model is heave displacement and heave velocity. Acceleration is usually considered when looking at comfort at higher frequencies, outside the primary ride region.

The goal was to find a metric that describes primary ride control of the car body. This would be a metric that had a clear correlation with the amount of damping that was implemented in the simulation model. Meaning more damping would either increase or decrease the metric value. The same damping level should not give two different metric results or the other way around.

3.1 Metric validation

A process of metric validation was necessary to understand how the metric behaved and what it captured.

Both simulation and four poster rig testing was used in an attempt to capture both a wide range of settings and the subjective correlation. To capture the behaviour of the simulations and rig testing a variety of road profiles were used. These are presented in table 3.1

Table 3.1: Rig test schedule

Load case	Character	Velocity	Roll	Pitch
1	Long bumps	90 km/h	No	No
2	Long and short bumps combined	50 km/h	No	Yes
3	Long and short bumps combined	50 km/h	Yes	Yes
4	Recorded road	50 km/h	Yes	Yes

All of the test excited the vehicle in heave, but not all excited in pitch and roll. Bumps are simple and result in a predictable vehicle motion while the recorded road is a more realistic country road scenario. Both was expected to be useful for understanding the metric.

3.1.1 Simulation

Simulation of quarter car models with a bang-bang CCD controller was done. Such simulation enabled sweeps over a wide range of damper settings, from 0% critical damping up to 200% critical damping. These sweeps were used to study the metric characteristics for 50 different damper setting. Both for the min and the max curve of the bang-bang controller. It was then possible to see if the new metric had a clear trend correlated to damper setting.

3.1.2 Four poster rig validation

To complement the high resulotion tesing on a quarter car model in simulation, a validation test on four poster rig were performed. These test used a Volvo Cars vehicle with a tunable CCD controller. It was possible to emulate the bang-bang logic in this vehicle, which enabled the same test as in simulation to be done only with lower resolution in damper settings. These settings are presented in table 3.2. Average critical damping is in the used range of damper velocities and in both compression and rebound.

Table 3.2: Damper setting for rig session 1

Setting name	Average critical damping factor, ζ
A	0.26
B	1.73
C	0.39
D	0.23
E	0.74

Six vehicle dynamicists rode in the car, two at a time. Their task was to rate the different settings according to primary ride and abruptness. Setting C was always the reference and the other settings were rated relative to C and each other. The rating protocol is presented in figure 3.1, instructions were to put C in the middle in all load cases and metrics.

Rig test protocol Rate setting A-E on the different metrics

Primary ride control

Load case	Low	Heave	High
1			
2			
3			
4			
	Low	Pitch	High
1			
2			
3			
4			
	Low	Roll	High
1			
2			
3			
4			

Abrubtness

Load case	Smooth	Heave	Abrupt
1			
2			
3			
4			
	Smooth	Pitch	Abrupt
1			
2			
3			
4			
	Smooth	Roll	Abrupt
1			
2			
3			
4			

Figure 3.1: Rating protocol for metric validation

The motivation for using fewer settings than in simulation is that it that it reduced the run time per person and keeping the amount of data to a reasonable level. Five settings resulted in 5x6x4 = 120 data sets, which was considered to be top of what is reasonable for this study. The aim was to qualitatively see if the metric characteristics seem to match, if the same trends could be seen in subjective, objective and simulated tests. The goal was not to map the subjective rating to an exact metric value.

To do so, all the settings were also measured to get the objective metric data from each setting. This makes it possible to compare the trends in the ratings with the trends in the objective metrics. Out on the road it is difficult to know how the road excites the vehicle, it would require extra sensors that measure the road. In the rig it is possible to use the input signal for the excitation as the road signal. This makes it possible to calculate with the road in the primary ride metric.

3.2 Method development

The fourth aim of the thesis is maybe the most comprehensive target as well, to develop a method for objectively tuning CCD dampers. This target is split in to two parts

1. Create a framework for optimising the CCD controller in simulation.
2. Alter the framework to make it possible to apply on the four poster rig.

The goal was to optimise the CCD controller in order to achieve good metric values in heave, pitch and roll. To do so the heave, pitch and roll parameters can be altered. Dependent on the level of detail that is desirable the amount of parameters can span between 15 and 100. The problem is therefore split into two different approaches. One simulation method that allows tuning of all parameters and one rig approach that focuses on tuning one or two specific modal velocities at a time. The rig approach would then have 15-30 parameters to handle. The rig optimisation used the CCD controller from Volvo Cars, a controller for their semi-active damper system. This is a more complex controller with more customisation compared to the bang-bang controller. This makes it possible to try the tuning method on a system that is used in production cars.

3.2.1 Simulation

With simulation there is a possibility go through a wide range of setting quickly. With these possibilities a genetic algorithm seemed appropriate.

Matlab's built in Genetic Algorithm (GA) function was setup to find all the parameters with the goal to minimise a cost function consisting of abruptness and primary ride. The capabilities of GA and the speed of simulation makes it possible to search for all the heave, pitch and roll parameters at the same time. With this setup it is possible to find a global minimum for the combination of these parameters. For more information about the GA used, see (Mathworks, 2023b).

The aim of the GA would be to find a global optimum of all parameters were the primary ride comfort becomes as good as possible. It would result in a setting that has as low primary ride control metric value and as low abruptness metric value as possible, and at the same finding a balance between the two metrics.

The limits for the optimisation can be set to the hardware limit of the dampers. This will be a large search space but since GA can find an optimum despite many variables and large search space it is still feasible to you use this approach.

3.2.2 Rig tuning

Rig tuning is more time consuming and difficult which requires the problem to be smaller. Therefore a known parameter set is used as an initial guess. Patternsearch was then used to tweak each parameter to achieve an overall better parameter set. The initial guess can be found by the GA method in simulation, by subjective tuning or simply a random guess within the damper limitations.

Simulation and rig tuning can be combined by finding a global optimum with simulation for a specified cost balance and then use rig tuning to find a setting that works slightly better in the real vehicle.

3.2.3 Rig optimisation approaches

To be able to conduct a study over what might be possible to learn from the rig optimisation the idea was to have different damper settings as a result of different rig optimisation approaches. The different settings could then be compared on the same piece of road as the recorded which the rig setup used to optimise, the recorded road load case. A reminder is that the optimisation worked with finding good values for the metric measuring primary ride(car body movement) and abruptness(how jerky the primary ride movement is). These two metrics have three parts each, heave, pitch and roll. Pitch and heave metrics were normalised to one according to a base setting and added together.

3.2.3.1 Baseline to balanced cost approach

The global optimum approach weighted primary ride and abruptness equally and added them together. The cost function became:

$$Cost = Primary\ Ride + Abruptness \tag{3.1}$$

The approach used a baseline setting as initial guess and adapted the baseline within the limits.

3.2.3.2 GA to balanced cost approach

This approach has the same cost function as the baseline to balanced cost approach. What is interesting with this approach is that the initial guess is found by using genetic algorithm and a simulink simulation model, the 7 DOF model presented in figure 2.2. This is motivated by the interest in seeing what a purely objective approach with CAE and optimisation can achieve.

3.2.3.3 Focused cost function

In an attempt to aim at either primary ride control or abruptness two new cost were tried, one primary ride focused and one abruptness focused:

$$Primary\ ride\ cost = Primary\ ride\ target + 2 \cdot Abruptness \tag{3.2}$$
$$Abruptness\ cost = 2 \cdot Primary\ ride + Abruptness\ target \tag{3.3}$$

Where the two targets were taken from the result of the "Baseline to balanced" approach. The distance to the best primary ride of that approach became the primary ride target and distance to the best abruptness value became the abruptness target.

3.2.4 Rig optimisation on track study

After the rig optimisation it was concluded which setting had an interesting outcome and could be worth testing on track. The most interesting settings were then driven by eight vehicle dynamicists and rated according to primary ride and abruptness. The test was conducted on two pieces of road. Firstly on the recorded road used during the optimisation, secondly on a handling track. This handling track also have characteristics of a country road but allows for slightly higher speed without exciting the car body too much compared to the recorded road section.

4

Results

The result chapter will cover the chosen metric, the validation of that metric and the rig optimisation method along with its outcome.

4.1 Metric choice

To understand what might work well as a primary ride metric a comparison between passive, CCD bang-bang and fully active ideal skyhook was done. This comparison is presented in figure 4.1.

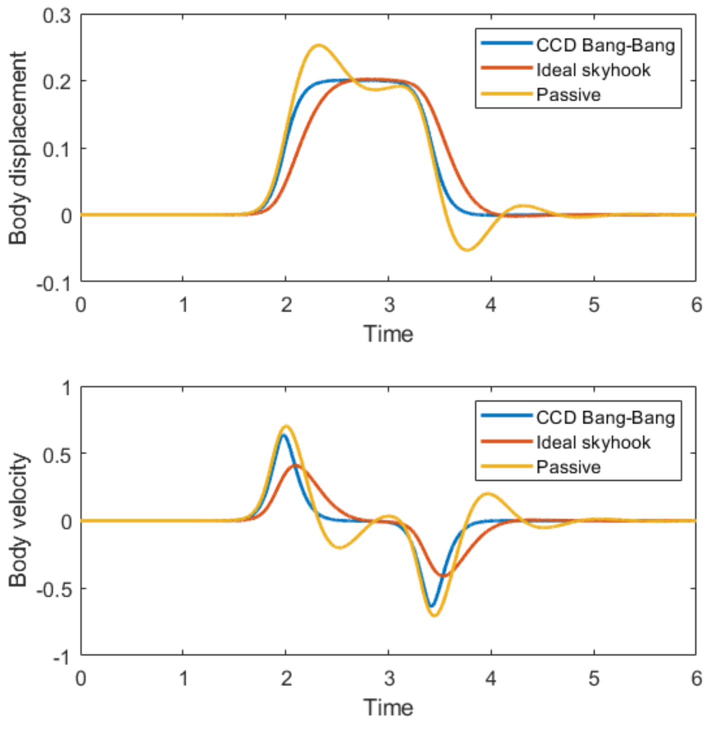


Figure 4.1: Control levels compared

The displacement is much smoother for both ideal skyhook and bang-bang CCD compared to the passive damper, with no overshoot. This is true also for velocity. The ideal skyhook decreases the velocity drastically while the bang-bang makes the velocity much smoother. A partial goal of the ideal skyhook could therefore be to decrease the overall body velocity. It should be true for heave, pitch and roll even if this experiment only excited in heave.

A good proposal for a primary ride metric that captures what more control does for CCD and ideal skyhook is to base it on body modal velocities.

$$Primary\ ride\ metric = \frac{RMS(v_{body})}{RMS(v_{road})} \tag{4.1}$$

This captures the overall velocity trends over a specific load case while also normalising the result to the road. The aim with road normalisation is to see if the metric will produce the same value even if the road amplitude is adjusted.

The metric in equation 4.1 together with an abruptness metric available from Volvo Cars will describe the primary ride comfort. The primary ride metric will show how fast the body is moving, where slow movements are desirable. Abruptness will describe how abrupt or sharp the transition from positive to negative heave, pitch and roll motion is. A ping pong ball is a good example. It is abrupt when the ball bounces on a table. To instead have ramp that the ball can roll through would be much less abrupt.

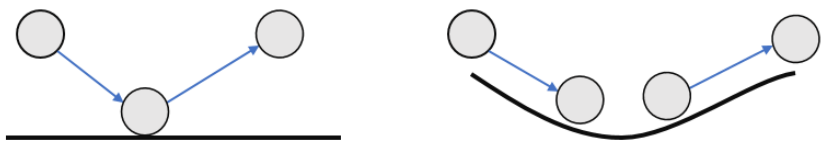


Figure 4.2: Abruptness example

The comparison with the ping pong ball is shown in figure 4.2. The left side visualises something that would be very abrupt. If the car body behaved liked on the left, the switch in heave velocity would be aggressive. If the car body moved like on the right, the transition would be less abrupt.

4.2 Metric validation

The validation data for the primary ride metric presented in equation (4.1) is presented in this chapter. It includes both simulation data and four poster rig data.

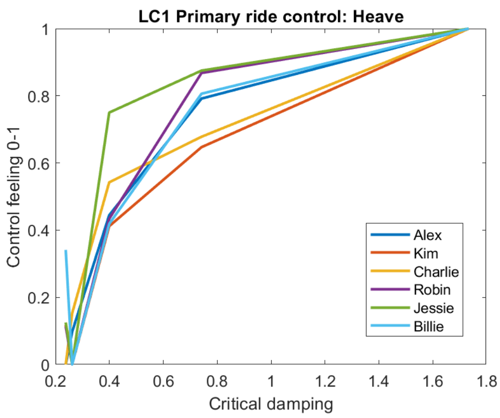
4.2.1 Rig test for validating simulations

In the following figures the ratings and measurements from the metric validation session on the rig. It showcases how the new metric behaves and how it corresponds to the subjective feeling of the car. The names of the participant have been kept anonymous.

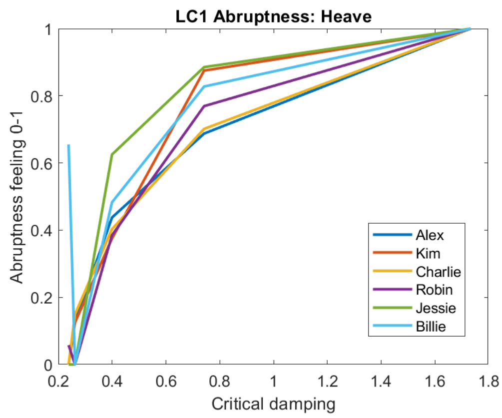
4.2.1.1 Load case 1: Long bump 90

Load case 1 as mentioned is a pure heave excitation, a long bump in 90 km/h but with no phase delay between front and rear axles. The bump is both up and down. The car however have some pitch behaviour in these scenario, hence that is also recorded.

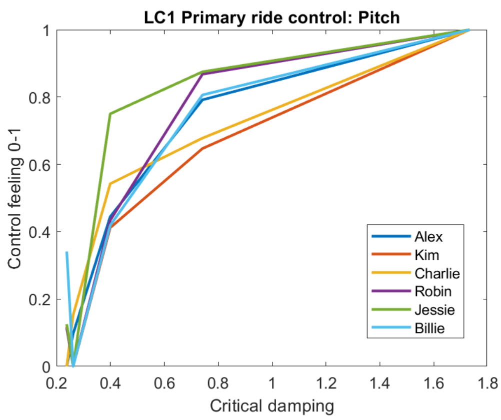
4.2.1.1.1 Ratings The six occupants rated both heave and pitch and this is visualised in figure (4.3).



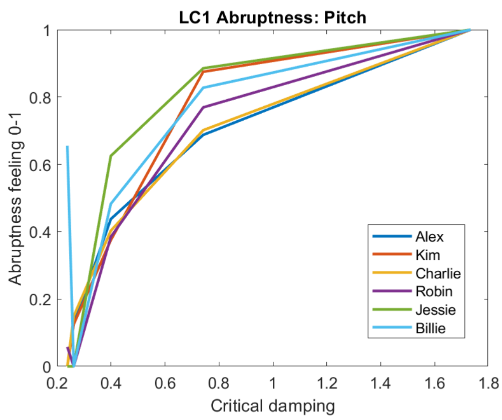
(a) Primary ride: Heave



(b) Abruptness: Heave



(c) Primary ride: Pitch

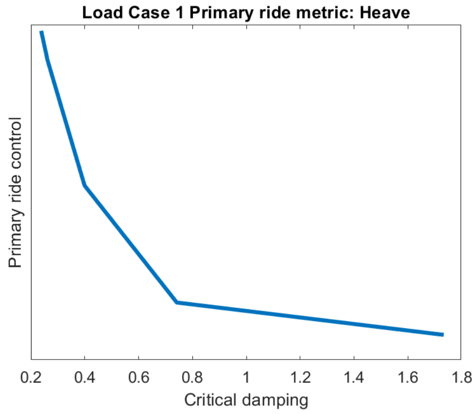


(d) Abruptness: Pitch

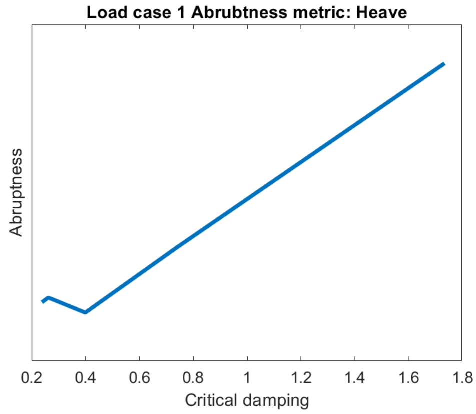
Figure 4.3: Load case 1 rated

All occupants rated more critical damping as more control and more abruptness, though to different extent. Half of the occupants switched the two lowest setting in abruptness, in control four of the occupants did that switch. This is probably due to the fact that those settings are close and low damping.

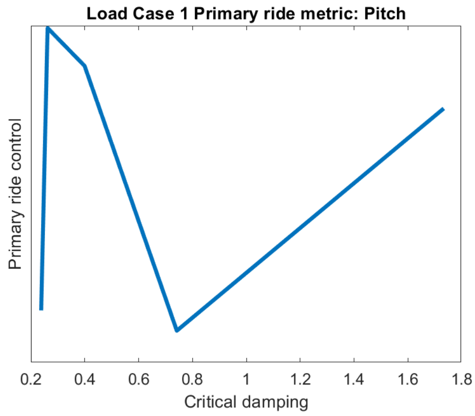
4.2.1.1.2 Measurements During the test on the rig the vehicle was also measured to see how the subjective ratings correlated to the actual movement of the vehicle.



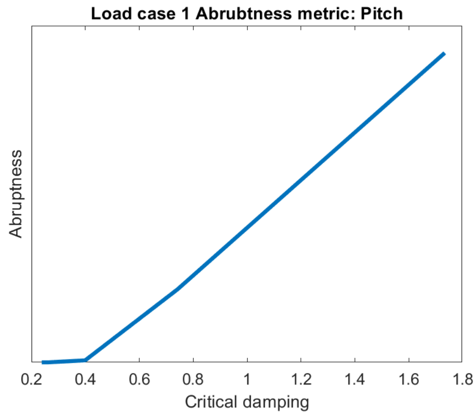
(a) Primary ride: Heave



(b) Abruptness: Heave



(c) Primary ride: Pitch



(d) Abruptness: Pitch

Figure 4.4: Load case 1 measured

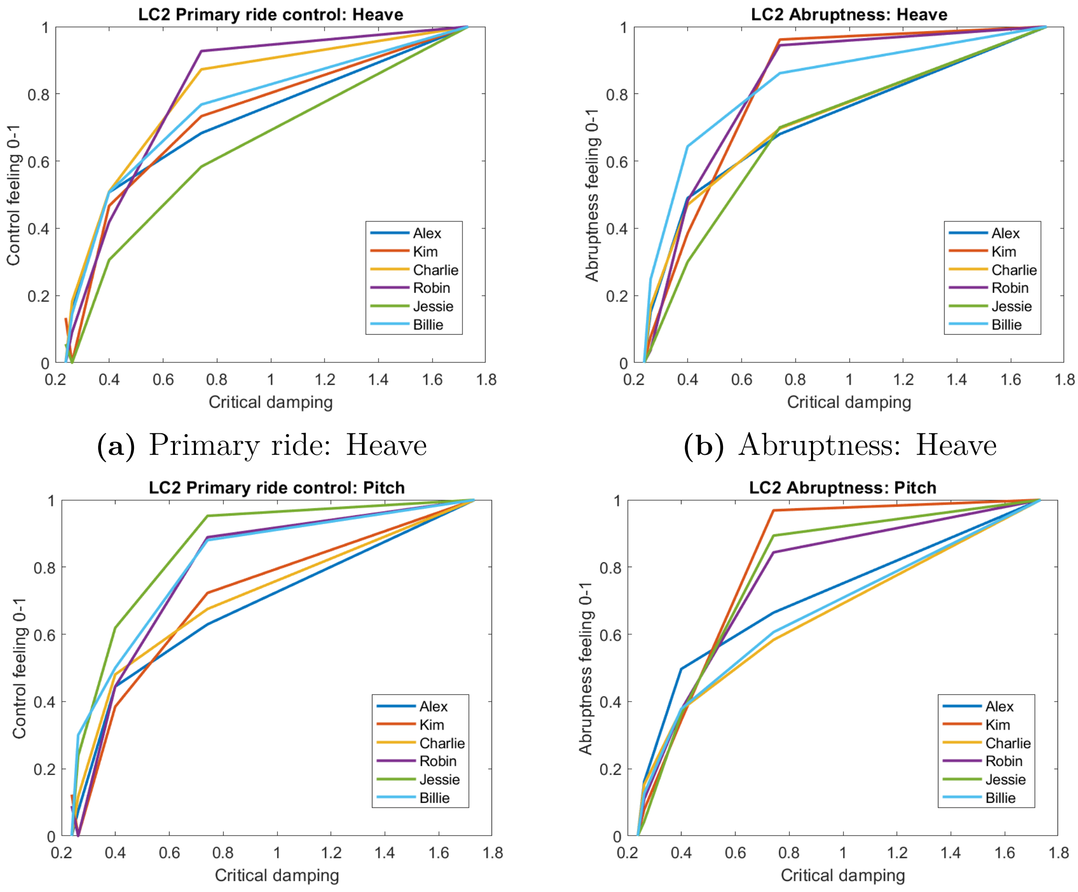
More damping clearly slows the car body down in heave while at the same time the measured abruptness increases. The primary ride change is similar to the rated one in the sense that more damping gives more control. That was the case for the rated abruptness as well but the abruptness change is measured to be more linear.

Pitch is difficult in this load case due to that the excitation signal should not have any phase shift. Therefore there should not be any road pitch angle. The division in the primary ride metric is therefore $\frac{RMS(Pitch\ Velocity)}{0}$ and the pitch angular velocity itself is small. That would be the main reason that the primary ride metric for pitch is not following a clear trend. The abruptness does however have a clear trend even in pitch.

4.2.1.2 Load case 2: Long short bump 50 without roll

Load case is composed of the same bumps as load case 1 followed by smaller steeper bumps and the test is simulated to be in 50 km/h. This load case has a phase shift representing the vehicle wheel base, thereby exciting a pitch motion.

4.2.1.2.1 Ratings The trends in the ratings are similar to load case 1 which is shown in figure 4.5.



(a) Primary ride: Heave

(b) Abruptness: Heave

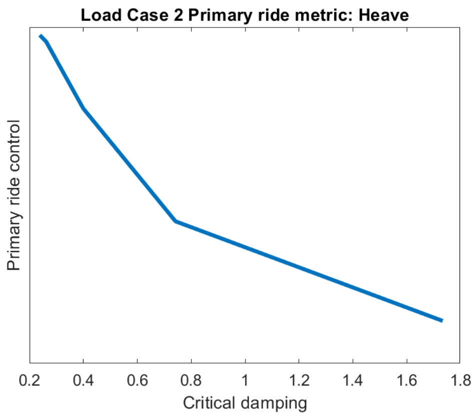
(c) Primary ride: Pitch

(d) Abruptness: Pitch

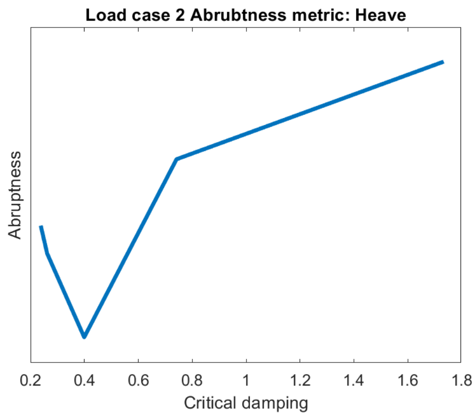
Figure 4.5: Load case 2 rated

For this load case the rating differences between the two highest setting seem to be smaller than the same difference in load case 1. The controller in the car is not designed for a bang-bang setup in mind and must do an arbitration between heave and pitch. Since the bang-bang controller asks for max from the damper in some way it is difficult to know how the vehicle prioritises the different modes of motion.

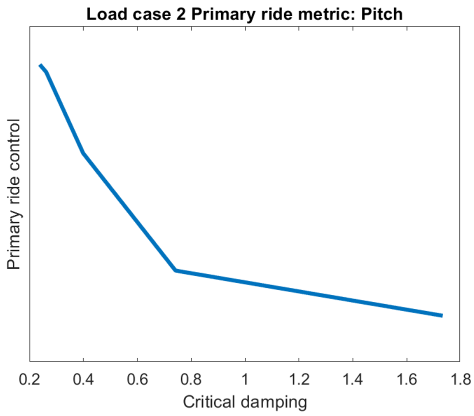
4.2.1.2.2 Measurements are presented in figure 4.6



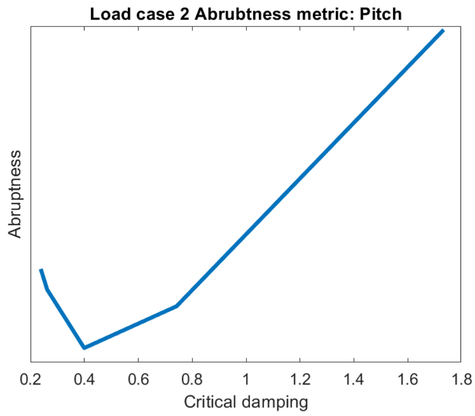
(a) Primary ride: Heave



(b) Abruptness: Heave



(c) Primary ride: Pitch



(d) Abruptness: Pitch

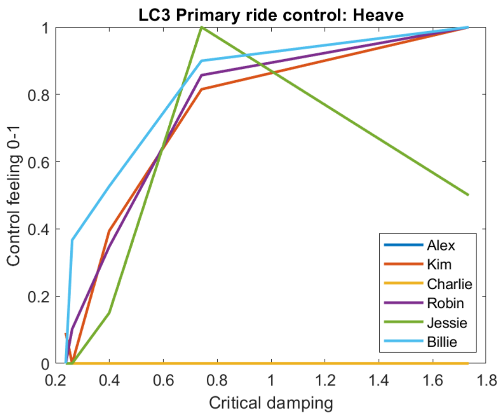
Figure 4.6: Load case 2 measured

The primary ride metrics showcases that more damping gives more control. On the other hand it seems like to little damping gives more abruptness. It could be because the vehicle is in its bump stop range of the damper travel. After some damper travel a bump stop/spring aid kicks in which creates an increase in stiffness. This stiffness could lead to increased abruptness.

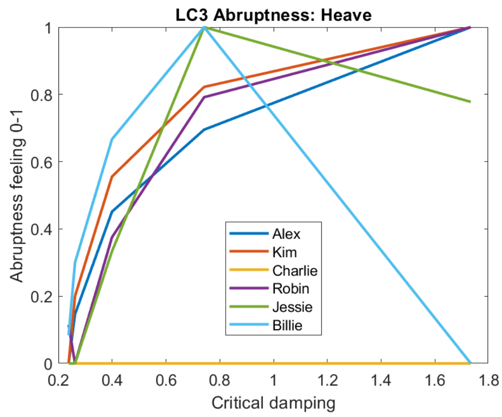
4.2.1.3 Load case 3: Long short bump 50 with roll

Load case 3 represents the same load case as load case 2 but only excites one side of the car at a time, thus inducing roll and pitch into the car.

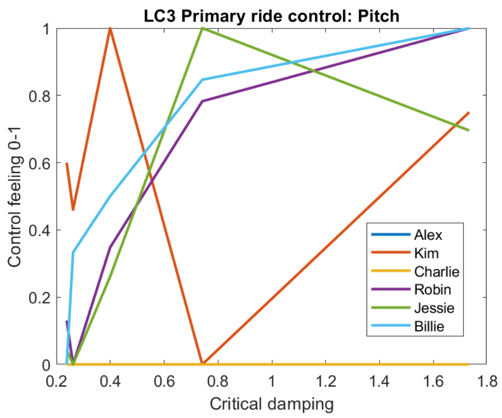
4.2.1.3.1 Ratings from load case 3 are presented in figure 4.7.



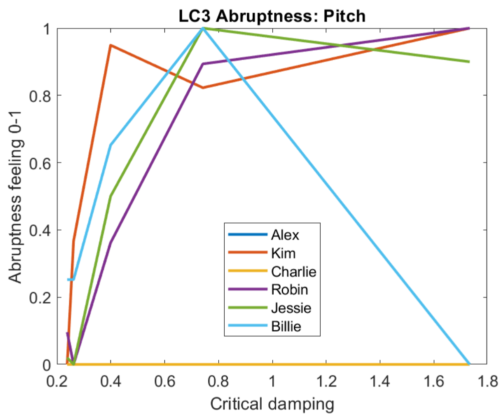
(a) Primary ride: Heave



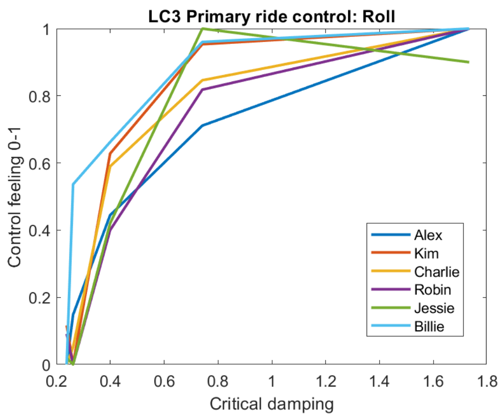
(b) Abruptness: Heave



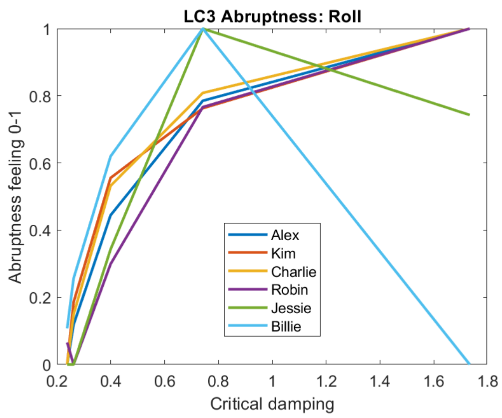
(c) Primary ride: Pitch



(d) Abruptness: Pitch



(e) Primary ride: Roll

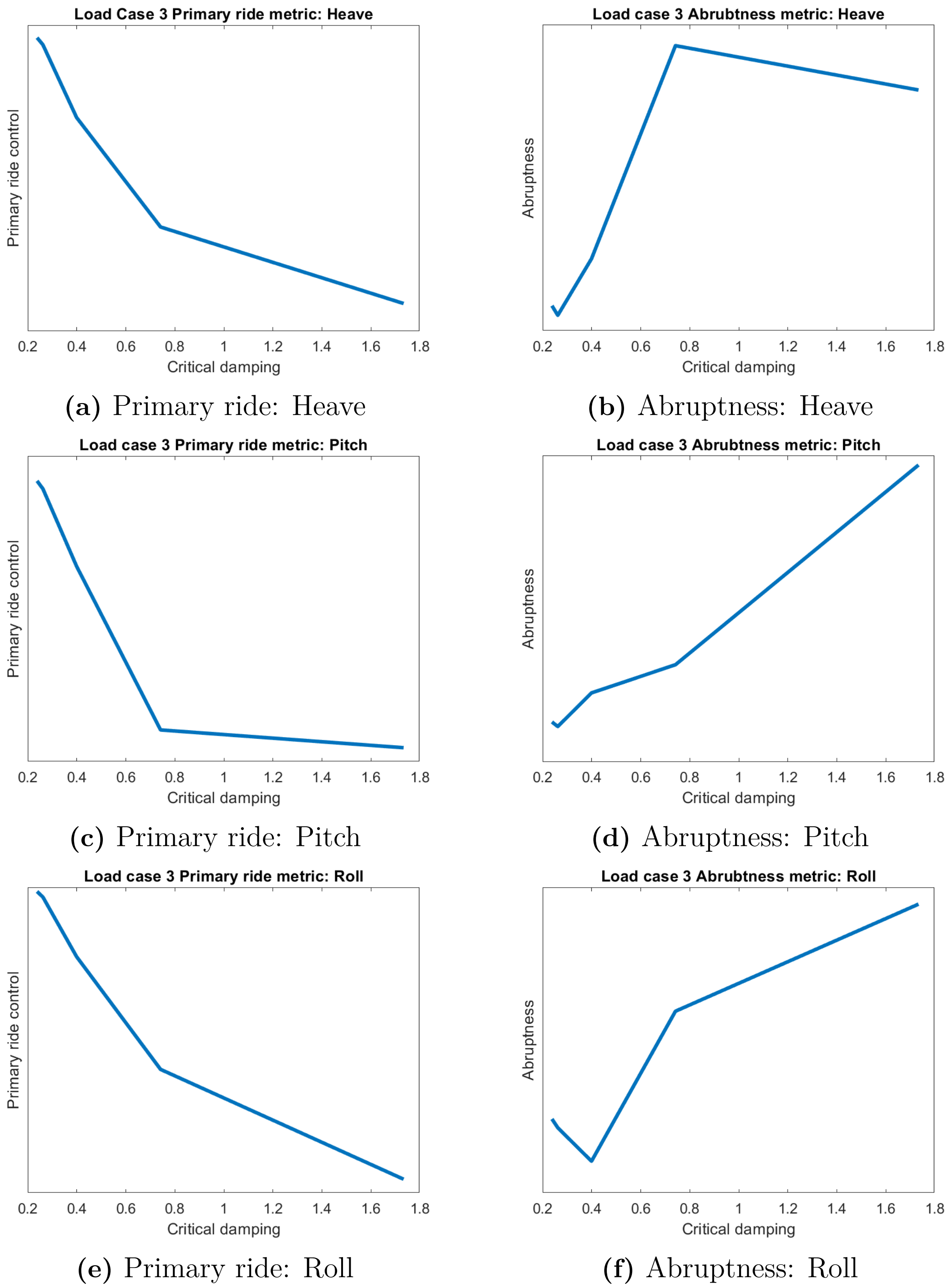


(f) Abruptness: Roll

Figure 4.7: Load case 3 rated

The trends seem somewhat similar to the other load cases. On the other hand there are ratings in pitch and heave that differ quite drastically from the usual trends. Heave, roll and pitch are all present in this load case which might make it difficult to isolate the modes of motion. However this is not what is seen in load case 4, which is a real road that also includes all modes of motion.

4.2.1.3.2 Measurements from load case 3 is presented in figure 4.8.



(a) Primary ride: Heave

(b) Abruptness: Heave

(c) Primary ride: Pitch

(d) Abruptness: Pitch

(e) Primary ride: Roll

(f) Abruptness: Roll

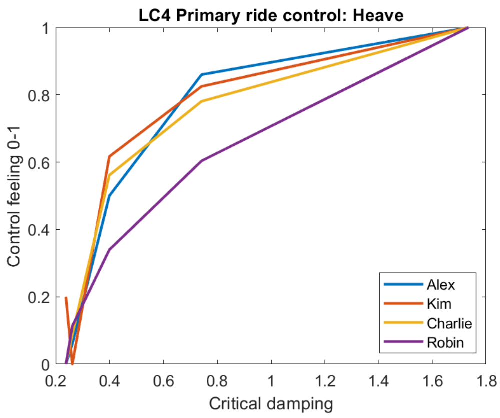
Figure 4.8: Load case 3 measured

More damping is clearly more controlled in this load case according to the metric. Abruptness for heave does not correlate directly with the level of damping. Pitch and roll do have a clear correlation.

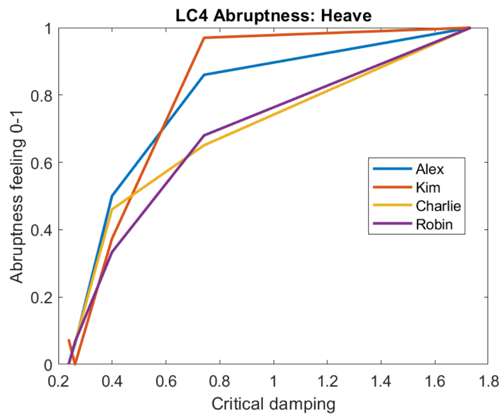
4.2.1.4 Load case 4: Recorded road

The recorded is a road section is a used to create a more realistic scenario for the vehicle. It should excite the vehicle in primary ride but also be realistic in its shape.

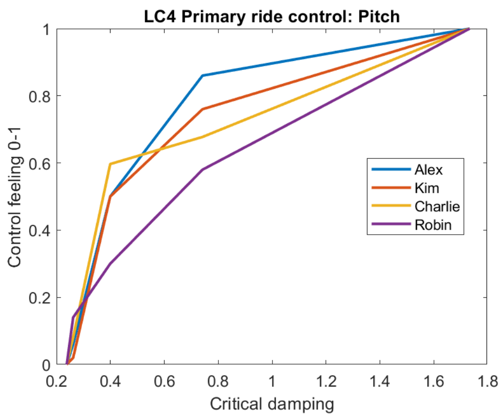
4.2.1.4.1 Ratings for this load case is presented in figure 4.9.



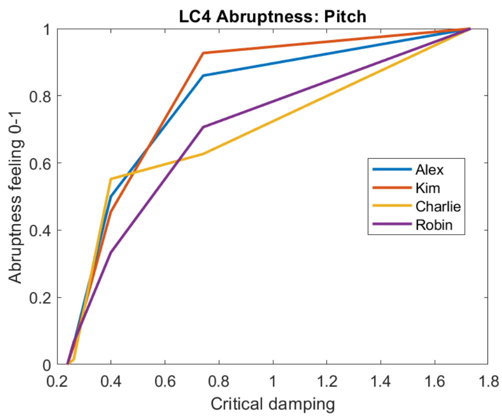
(a) Primary ride: Heave



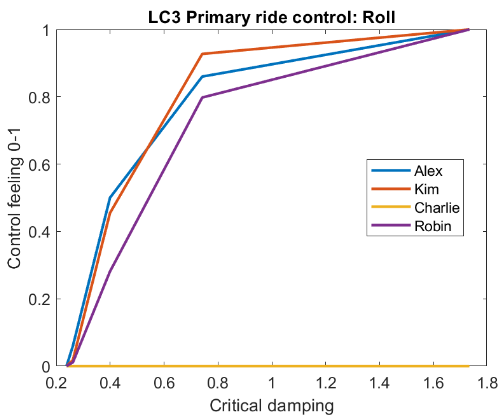
(b) Abruptness: Heave



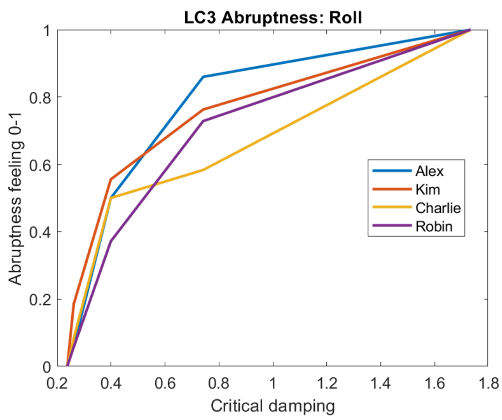
(c) Primary ride: Pitch



(d) Abruptness: Pitch



(e) Primary ride: Pitch



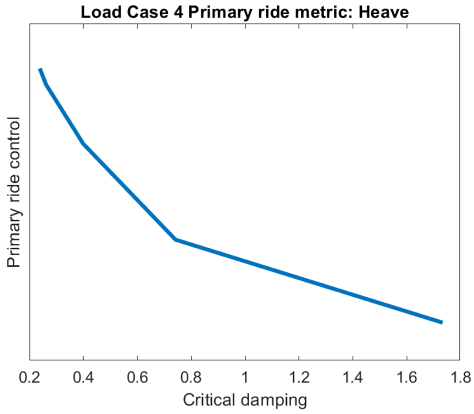
(f) Abruptness: Pitch

Figure 4.9: Load case 4 rated

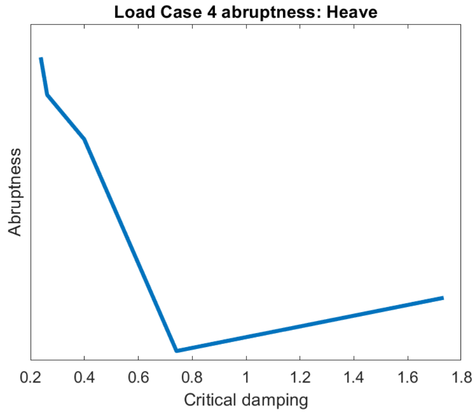
The trends are overall the same as the previous load cases, which is a promising sign that the metric might capture what it is intended to capture. Yet again the problem with arbitration with the three different modes of motion requests is an issue, even in this case it would be difficult to know exactly how the three modes are prioritised by the controller. That said the metrics will still describe the vehicle body behaviour regardless of how the controller handles the arbitration.

One interesting thing was the primary ride feeling in the recorded road load case. That road is a rough road that dramatically excited the vehicle, while also being the load case that resembles a real road the most. When using maximum damping levels, setting the critical damping as high as possible, some occupants described it as "very good". This could could be motivated by the heave velocity and travel of the suspension being kept to the lowest. When going the softest setting the occupants described a feeling of being tossed around and feeling a lot of movement.

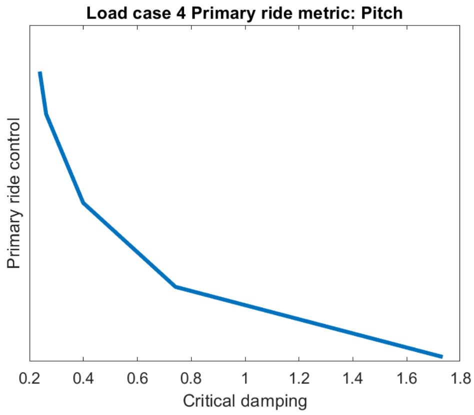
4.2.1.4.2 Measurements from the recorded road is presented in figure 4.10. This load case excited the vehicle in heave, pitch and roll.



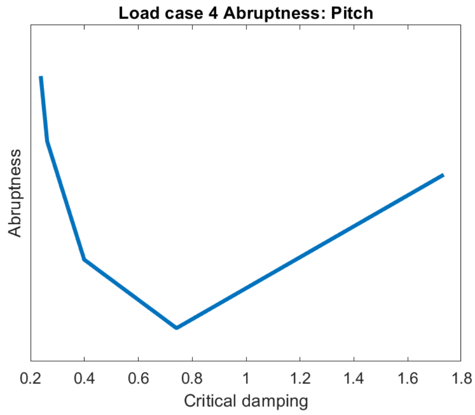
(a) Primary ride: Heave



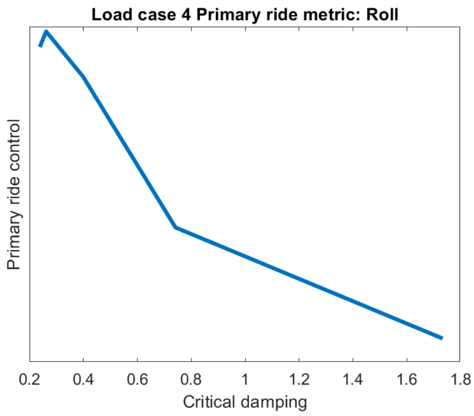
(b) Abruptness: Heave



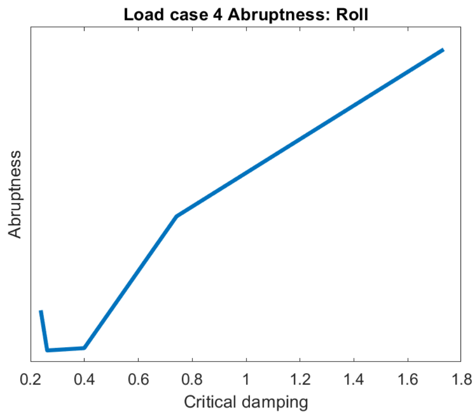
(c) Primary ride: Pitch



(d) Abruptness: Pitch



(e) Primary ride: Roll



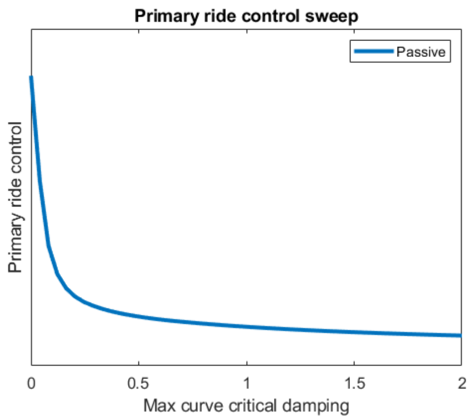
(f) Abruptness: Roll

Figure 4.10: Load case 4 measured

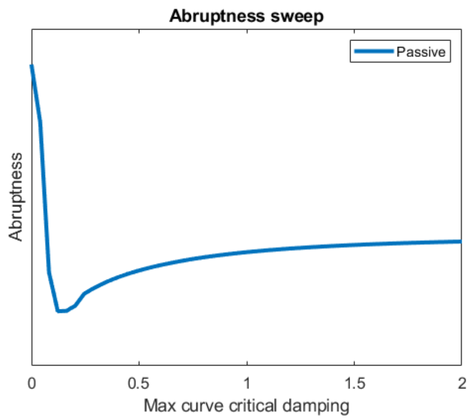
For all three modes of motion the RMS of the velocity is reduced greatly, especially in heave and pitch. If the feeling is very good it validates to some extent that lower velocity is desirable. The abruptness has initial peaks in low damping before the usual upwards trend begins. This is probably due to the fact that the damper is in spring aid territory. The corner that seemed to have the largest travel was the rear right. This impression was confirmed true when looking at the suspension travel data. It could be seen that the travel for the settings below 0.5 in critical setting is in the range of the bump stops and spring aids. This will result in a jump in spring stiffness which would contribute to abruptness. The large travel will therefore be a contributing factor to the large abruptness in low damping settings in load case 4, the recorded road.

4.2.2 Simulated objective characterisation

Simulations were done with sweeps of the max critical damper curves, here referring to the logic presented in table 2.1. A sweep was done for max damper curve side. The sweep is between 0% critical damping and 200% critical damping.



(a) Primary ride metric sweep



(b) Abruptness metric sweep

Figure 4.11: Max curve sweep in skyhook

The rig testing lack detail in the amount of damper levels, especially between 80% and 200% critical damping. With simulation that level of detail is extracted. With these simulation it can be seen that more critical damping in CCD increases the amount of primary ride control but also increase abruptness. The peak abruptness at low damping is from reaching the end of travel even in simulation. The endstops are modelled as a very stiff spring, like hitting a metal wall.

4.3 Method development

To solve the problem for global optimisation in simulation and limited optimisation on the rig it had to have a starting point. That was on looking at just altering the heave table.

4.3.1 Optimisation proof of concept

To find out if the Genetic algorithm can work, the parameters heave in the CCD controller was optimised and compared to a baseline tuning and an in-car tuning example. The recorded road was chosen as appropriate load case, it is a real road and it has a wide spread of excitation frequencies and amplitudes.

Data	Primary ride	Abruptness
Baseline	2.56	5.77
Tuning example	2.27	5.22
GA Optimisation	2.17	4.85

Table 4.1: Metric comparison for testing optimisation

Table 4.1 shows a comparison between settings produced for the semi-active CCD controller. The point is to show that an optimisation algorithm is a helpful tool for finding good tuning sets. GA finding is produced by an optimisation run over 140 generations and 64 individuals. Baseline and tuning example are tuning sets that once was set by tuning subjectively in car. When tuning subjectively an engineer will take the broad vehicle behavior into account, not just primary ride and abruptness, but these two metrics are a big part of it. The main takeaway from this is that it is possible to find good tuning sets with optimisation if the target is well defined, maybe even better than what is subjectively possible. 140 generations and 64 individuals for this 11 seconds of recorded road simulation equals to 27 hours of real time running, which is not feasible for the rig.

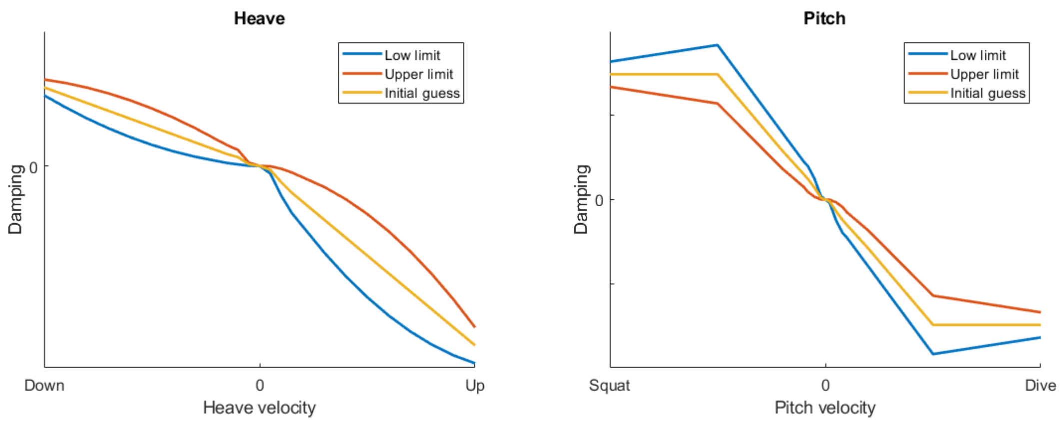
4.3.2 CAE Investigation of rig method

Rig testing is limited by real time and on/off loading of parameters to the ECU of parameter to the vehicle. Because of this it is reasonable to limit the total run time to less than on working day. With more confidence in the rig setup the method could run much longer without monitoring. Table 4.1 is extended with an optimisation method that might be possible to pursue on the rig.

Data	Primary ride	Abruptness
Baseline	2.56	5.77
Tuning example	2.27	5.22
GA Optimisation	2.17	4.85
Rig pattern search	2.45	5.47

Table 4.2: Metric comparison for testing optimisation

Patternsearch was used in a narrow search space around an initial guess, in this the setting called 'Baseline' in table 4.2. Figure 4.12 showcases how the limits were set to achieve this narrow search space. The same strategy was first tested in CAE and then transferred to the rig.



(a) Heave with limits

(b) Pitch limits

Figure 4.12: Limits for the patternsearch optimisation

The total primary ride improvement is almost 5% but the heave primary ride improvement is closer to 7%. Abruptness improvement is around 6% and heave abruptness improvement is almost 11%. The test itself was done in 1000 function runs, which corresponds to 1000 real life runs. 1000 runs would take approximate 5 hours. Some margin for loading parameters on the ECU network must be considered in order to make it work on the rig. The improvements seem to be in range that the test occupants were able to distinguish between during the metric validation.

4.3.3 Rig optimisation method

Since the aim of the project was to create an objective tuning method that is applied on a four poster rig, the description of the rig optimisation method is a part of the result.

A complete vehicle with functioning CCD was in the optimisation loop on the rig. Between every excitation of the vehicle a computer analysed the results and updated the CCD parameters according to the Patternsearch optimisation method. This was all done in an automatic setup which allowed for optimisation running on its own over a long enough time.

For Patternsearch to be effective it is good to limit the search space. Choosing limits is therefore done in the same way as in CAE, which is presented in figure 4.12. More space is given around the low velocities, percentage wise. Without that the algorithm would not be able to edit those parameters. Roll was removed from the scope of this analysis to reduce the amount of parameters, resulting shorter run time required on the rig. This ensured that the problem is kept much smaller on the rig compared to simulation studies.

Figure 4.13: Complete vehicle setup on rig

Figure 4.13 shows the setup on the four poster rig. The yellow posters under each wheel will excited according to a recorded road file. Due to the vehicle standing still, the dampers heated up quickly during excitation to a point where they behaved different. Because of this, cooling fans were installed close to the dampers to keep them cool.

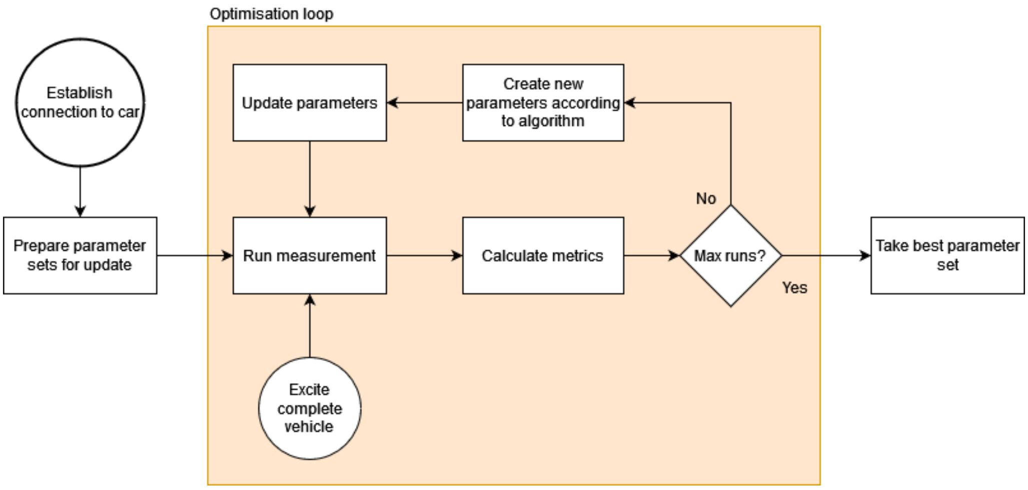


Figure 4.14: Flow for the rig optimisation setup

Figure 4.14 show the overall process of the rig optimisation. A laptop measured and updated the vehicle ECU controlling the CCD according to the patternsearch algorithm but the laptop could not communicate with the rig. This required a synchronisation between the rig and the laptop, so that the MATLAB optimisation and the rig started at the same time. The optimisation was set to have a maximum number of evaluations runs, thus limiting the maximum time it could work on the problem. If that limit was reached the optimisation stopped and the best parameter set yet was chosen.

With this setup it was possible to run optimisation without human interaction. The vehicle was excited and updated continuously for up to three hours without interruptions. The different cost functions presented in section 3.2.3 were investigated but not all resulted in useful outcomes.

The method itself has a wider range of application than just primary ride control and abruptness. If the measurement options, the parameter choice and the cost function is chosen in consideration with a different target in mind it could still be effective. This could be for example grip optimisation or secondary ride comfort.

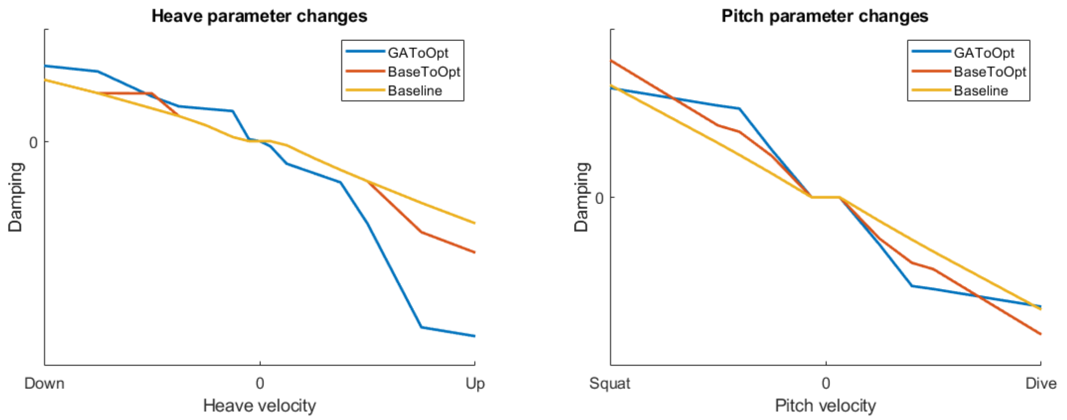
4.3.4 Rig optimisation results

Three useful settings were found from the rig optimisation along with the metric values for these three settings. These are shown in table 4.3. The important result is the order in which the three settings ended up in in the different metrics. For example in primary ride heave, 'GAToOpt' most controlled, 'BaseToOpt' second most constrolled and 'Baseline' was least controlled. Roll is measured but not part of the optimisation. Roll has worsened in both settings from the optimisation.

	Baseline	BaseToOpt	GAToOpt
Primary Ride	2.27	2.13	2.05
Heave	1.593	1.470	1.364
Pitch	0.225	0.199	0.190
Roll	0.451	0.457	0.499
Abruptness	19.73	19.79	20.08
Heave	2.586	2.519	2.418
Pitch	8.243	8.212	8.419
Roll	8.905	9.059	9.243

Table 4.3: On track metric values

'Baseline' is a tuning set created by vehicle dynamicists at Volvo cars through subjective tuning. It works as a reference to compare the settings found by the optimisation method. It was also used as a starting guess in 'BaseToOpt'. 'GAToOpt' uses a starting guess generated purely in CAE with the 7 DOF model and gen



(a) Heave tables from optimisation **(b)** Pitch tables from optimisation

Figure 4.15: Lookup tables as a result of rig optimisation

In total figure 4.15 shows that both the optimisation generated settings uses overall more damping, in both heave and pitch. The difference between 'BaseToOpt' and 'GAToOpt' is not as big in pitch as it is in heave. 'BaseToOpt' follows 'Baseline in pattern in pitch while 'GAToOpt' has an agressive ramp up from zero that eases off.

4.3.5 On track evaluation of rig outcome

The three settings from the rig optimisation were rated according to primary ride control and abruptness. This resulted in an average rating over the eight drivers for heave and pitch on both the recorded road section a the handling track (HT2). Figure 4.16 and 4.17 shows the results from the rating with the corresponding amount of control and abruptness. The control metric value has been inverted in this plot, meaning a higher bar equals more control in both rating and metric.

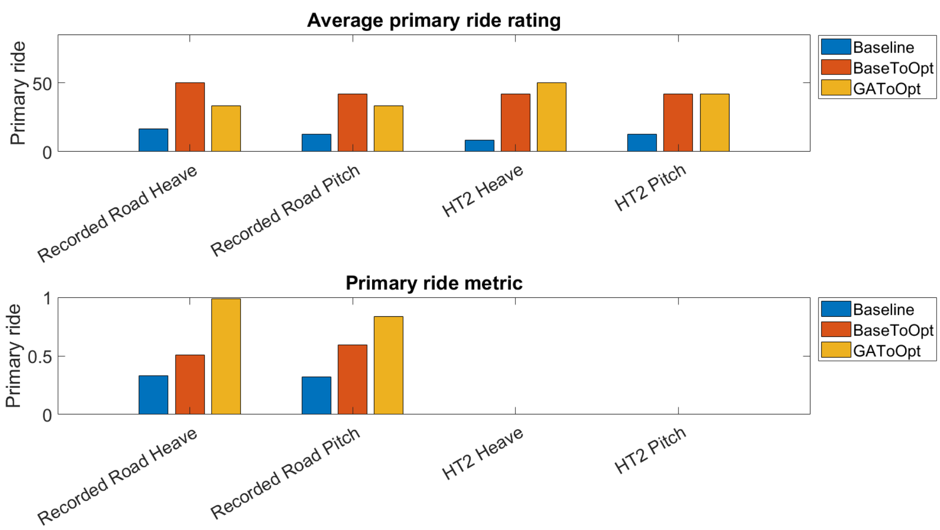


Figure 4.16: Primary ride rating and corresponding metric

Figure 4.16 shows the primary ride results. The top plot is the rating and the lower plot is objective measurement. There is no objective measurement from the handling track since that load case was not tested on the rig. Abruptness is shown in the same manner in figure 4.17. The data is presented for heave and pitch seperated, roll is not included since it was not included in the optimisation.

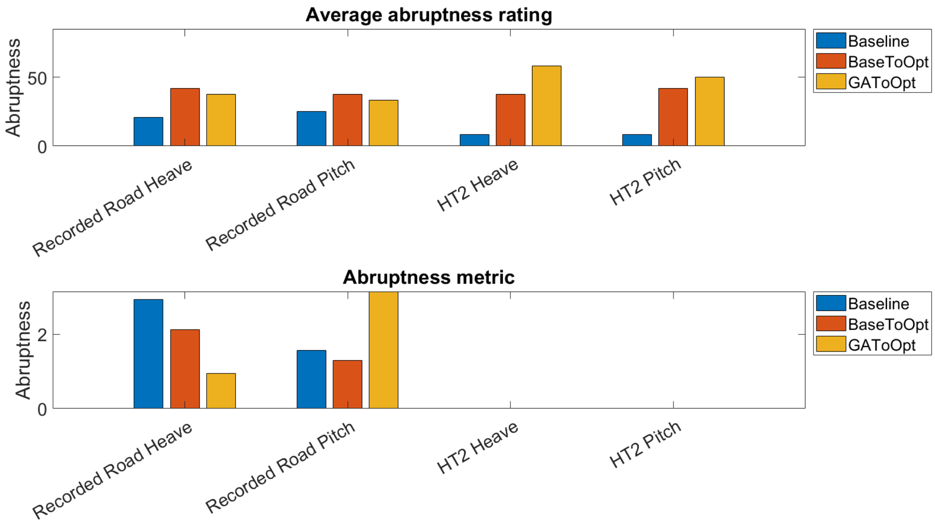


Figure 4.17: Abruptness rating and corresponding metric

'GAToOpt' is most controlled according to the metric measurement, 'BaseToOpt' is second most controlled. This creates a discrepancy between the ratings and the measured metric value for the two rig generated settings since the 'BaseToOpt' is rated most controlled. This discrepancy is not present on the handling track.

The two rig generated settings are less abrupt compared to baseline in heave. In pitch, 'GAToOpt' is more abrupt compared to the other settings according to the metric. As seen in table 4.3, if roll is included, 'BaseToOpt' and 'GAToOpt' is more abrupt.

5

Discussion

While doing the metric validation on the rig the test were done blindly, meaning that the participants did not know which setting was which when the test started. Most of the participants did however figure out which setting was which during the course of trying the different load cases. Some participants tried to spot the particular setting based on the car behavior instead of rating that specific setting. It is hard to know how big the impact is on the result but it does mean than the settings are rated very similar over the different load case. To counter this it would be good if all settings would have names that changed between setting for every load case, meaning the participant would never know what setting it is. That said, the participants were experienced engineers and test driver that most likely would have recognized the settings either way.

There are certain discrepancies between the primary ride control metric values and the test drive ratings done on track. To understand this it is important to know what people are actually rating when they are rating primary ride control. The metric works well when the car body get excited in scenarios with large damper travels. In such a scenario the damper settings will highly impact the body velocity range and therefore also the metric. In those cases it might also be that the feeling of control is dominated by the movement and the speed of the car body. That is maybe why the trends are so clear in the bumps tested in the metric validation sessions. When driving out on track the primary ride control feeling might be impacted by how the car feels compared to the road that the car is passing over, which might result in a different rating. What might also be the case is that the most abrupt settings is rated most controlled by some. A setting that feels abrupt must by intuition be controlled since these metrics often go opposite each other. When that is the case the most abrupt settings is also rated most controlled even if that is not necessarily the case.

Setting up the rig for optimisation was an interesting learning experience that took more time than expected. One big issue was overheating the dampers as a result of exciting them repeatedly. After around 30 repetitions the optimisation starting behaving oddly as a result of overheating dampers, the metric values just got worse and worse. Normally there is air flowing past the dampers as the car travels forward, this was not the case on the rig since the car is stationary. To compensate this fans were installed and pointed at the dampers to keep them cool. Since it was not possible to control the rig with Matlab it was important to know how long a rig repetition was and how long it took for Matlab to run each optimisation step. With this information Matlab could be instructed to wait a certain time so minimise the drift due to the lack of sync.

There was no possibility to measure the car and the rig at the same time, meaning that the metric was calculated without the root mean square of the road. A way around this is to do a test run at first, measure the road, calculate the rms of road velocity and use that in the rest of the optimisation. However, the trends of the metric is the same with or without the road velocity but it would have been good to use it as it was intended.

Some kind of sensor noise also became an issue on the rig. Some updates to the vehicle had almost no effect, for example parameter change close to zero body velocity. Such changes are important when tuning according to the engineers at Volvo Cars. But since the change was so little in actual force it was hard for the measurement to capture it. The metric appeared the same even if changes were made. This noise problem could also be seen when running the same load case with the same setting. Metric values could differ by up to 3% despite their being no change in the test. This was in the end countered by using three runs of the same road as one excitation. This made it somewhat easier to trust the metric values for each iteration. It is probably good to extend the number of excitation that was used but it is a trade of between speed and precision.

One important aspect when drawing conclusions from the on track test driving is the vehicle used. Almost all participants stated the settings from the optimisation was either too controlled or too abrupt. This could be because a Volvo SUV was used for the test. Some participants stated that a car like that should feel plush and smooth, therefore low abruptness is important. Too high control is also not desirable. If a sporty sedan would have been used instead of the SUV, the participants might have thought the settings used too little control.

6 Conclusion

The primary ride metric used gives an overall good indication of level control that is used in the vehicle. In almost all cases were it was tested on the rig it showed that the use of more control in the damper parameters also gave a smaller number in the primary ride control metric, which was the aim of the metric. In the metric validation this could also be seen in the way the different damper levels were subjectively rated. On the other hand, out on track, it is not certain that it captures full feeling of primary ride control since the ratings in the test track study were not fully in line with the amount of control in the parameters or the metric value. This was true also for abruptness.

An effective optimisation loop was setup on the rig. The vehicle got measured and updated continuously according the algorithm and the outcome was clearly dependent on the cost function and the initial guess. This concludes that the setup is working. That said, not all cost functions that were tried had resulted in a useful outcome. Thus only three settings were tried on the test track. The feeling is that the setup also can be used for other goals such as vibration comfort or grip optimisation. For use in different goals the measurements and goal functions must be altered to gather the correct information about the problem.

Rig optimisation generated settings that were an improvement with regards to metric values. Heave and pitch were included in the optimisation and they were improved in both control and abruptness. Only metric that was not improved was the pitch abruptness for one of the settings. With that said it can be concluded that the rig optimisation process will improve the parameters with respect to the metrics it is given.

6.1 Future work

After concluding the match and mismatch of the control and abruptness metrics it could be interesting to extend the work around the metric. It would be interesting to do a larger study regarding the metrics to fully understand what they cover and they do not cover. This study should then include people with other backgrounds than vehicle dynamics. With a bigger study it would be possible to map the objective metric value to a subjective rating scale, which would have been interesting.

Regarding the optimisation loop, to cover a wider range of ride comfort work must be done to load cases and the target function. The target function must include the wide range of ride frequencies, from slow primary ride motions, up to fast secondary ride vibrations. A study with wider range might end up showing that this type of method found in this project can be useful in tuning semi-active dampers, maybe even fully active dampers for ride comfort.

Lastly the focus could be shifted entirely. Ride comfort is only a part of the vehicle dynamics that the semi-active dampers must cover. Another subject is handling, where confidence and grip is important. This could include minimising body roll in the vehicle and maximising road-tyre contact force. It would indeed be interesting to try the method on a problem like this.

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