On Objective Measures for Ride Comfort Evaluation

Katrin Strandemar

TRITA-S3-REG-0505
ISSN 1404-2150
ISBN 91-7178-204-4

AUTOMATIC CONTROL
DEPARTMENT OF SIGNALS, SENSORS AND SYSTEMS
ROYAL INSTITUTE OF TECHNOLOGY (KTH)
STOCKHOLM, SWEDEN, 2005

Submitted to the School of Electrical Engineering, Royal Institute of Technology, in partial fulfillment of the requirements for the degree of Technical Licentiate.
Copyright © 2005 by Katrin Strandemar

On Objective Measures for Ride Comfort Evaluation

Automatic Control
Department of Signals, Sensors and Systems
Royal Institute of Technology (KTH)
SE- 100 44 Stockholm, Sweden
Abstract

An essential tool in the truck development process is the ability to quantify and grade vehicle dynamic behavior. Today this is performed either through subjective or objective tests. Subjective tests have the disadvantage that numerous factors influence test drivers’ opinions while objective measures have the advantage of repeatability. However, objective methods of today are often only able to provide a rough grading of vehicles. The main objective with this thesis is to develop more sensitive objective methods for ride comfort evaluation. An effective test procedure to measure driver perception sensitivity to small differences in vehicle ride is suggested and utilized. The driver sensitivity is tested on dynamic behavior that is typically graded in vehicle development. Cab motions from a truck are first measured and then recreated in a simulator where a test driver is seated. The perception threshold for small changes in typical vehicle motion is established in this way for each test person. The perception sensitivity tests indicate that humans are quite sensitive to transients in vehicle motion. One problem with many common vehicle ride measures is that the impact of transient behavior is small due to the averaging used to condense the measurement data into scalar measures. A new evaluation method for ride comfort, with influences from the well known handling diagram, is suggested. This method has four main advantages: it is fairly simple to interpret, it shows the absolute vibration level, it considers transient events separately and it shows changes in vehicle character with increasing excitation. Promising results from both measurements and simulations are derived.

New technology has made it possible to vary vehicle suspension parameters during vehicle ride. In order to prescribe different damping for different vehicle modes, modal motion estimates are needed. A system identification approach is suggested. It yields improved estimates of vehicle modal motion compared to previous work.
Acknowledgements

First I would like to express my gratitude to my supervisors Professor Håkan Hjalmarsson and Dr. Boris Thorvald. Thank you Håkan for all help with writing this thesis and thank you Boris for all our interesting discussions and for educating me in vehicle dynamics.

This work was partially sponsored by IVSS (Intelligent Vehicle Safety Systems) which is gratefully acknowledged.

Many thanks go to all my colleagues at Scania for helping me with many things. To my boss Carina, thank you for being on my side all the time. Special thanks go to Ulf Olsson who has shared his great experience of vehicle development and helped me with several practical things throughout this work.

I am thankful to the people at the control department at S3 for always making me feel very welcome, you are great!

Finally I would like to thank my parents, Christina and Johan for all their help and support. Henrik, thank you for always being there, at work, at home, through good and bad.
Contents

1 Introduction 1
1.1 Common Ride Evaluation Methods 2
1.2 Thesis Outline and Contributions 14

2 Driver Perception Sensitivity 17
2.1 Introduction 18
2.2 Vehicle Measurements 20
2.3 Generation of Signals for Simulator Tests 21
2.4 Simulator Tests 23
2.5 Proposed Difference Measures 24
2.6 Human Perception Theory 27
2.7 Results 32
2.8 Conclusions 34

3 Description of the Ride Diagram 37
3.1 A new Measure: The Ride Diagram 39
3.2 Limit for Transients 42
3.3 Road Influence 43
3.4 Summary 46

4 Evaluation of the Ride Diagram Using Simulations 49
4.1 Road Profile Used in Simulations 50
4.2 Simulation Model 52
4.3 Suspension Analysis 54
4.4 Transients and Stationary Behavior 60
4.5 Conclusions 61

5 Evaluation of the Ride Diagram Using Measurements 63
5.1 Altering Cab Bushings 64
5.2 Different Damper Settings 65
1 Introduction

The importance of improving vehicles dynamic properties is constantly growing in truck development. Various areas like active safety and driver environment are dependent on enhanced dynamic behavior. An essential tool in this process is the ability to quantify and grade these properties. A vehicle’s dynamic response is often divided into two sub areas, handling and ride, which may be characterized as:

**Ride:**  *Vibrations in vehicle response to e.g. road irregularities.*

*Vibration comfort:* low frequency vibrations (0-20Hz)
*Harshness:* high frequency vibrations (>20Hz)

**Handling:** *Vehicle response to driver input.*

Today the most common method to determine vehicle ride and handling qualities is by subjective rating tests. Test drivers ratings will determine which vehicle that has preferable dynamic behavior.
However, the complex connection between ratings expressed by test drivers and explicit design parameters makes improvement work difficult. Since test drivers change their acceptance level over time, lack of repeatability in subjective testing is a problem. A long time goal is therefore to develop objective measures for driving impressions.

This thesis starts with a description of the most common methods for ride comfort evaluation today in Section 1.1. Problems and shortcomings of these methods, as well as advantages, are discussed to give a background for the following work. Section 1.2 contains an outline of the thesis together with a short presentation of the contributions.

### 1.1 Common Ride Evaluation Methods

This section contains descriptions of common methods to measure ride and handling objectively. First there is an explanation of some essential terminology in this thesis according to an ISO vocabulary in Section 1.1.1. Many different strategies on how objective methods can be evaluated together with subjective ratings are also given. This because it is crucial that the objective measure corresponds to how most drivers perceives the vehicle. Subjective ratings can be collected in several different ways and various rating scales may lead to a very wide spread of ratings. Many problems are connected to evaluations where humans are used as measuring instruments. Humans’ sensitivity and calibration values are relatively unknown and vary between individuals and over time. One additional difficulty is the ability to judge one thing at a time without being influenced by other properties. In Section 1.1.2 several different rating techniques are presented together with important issues to consider when rating tests are performed.

The most important factor of objective evaluation methods is that they correlate well with how most test drivers perceive the vehicle. Objective measures therefore consist of two important building blocks: subjective information from test drivers and measured vehicle properties. Correctly performed rating tests are essential when building meaningful models that generate objective evaluations.
Section 1.1.3 begins with a description of advantages with objective measures followed by descriptions of objective measures divided into transient sensitive and stationary methods.

1.1.1 Vocabulary

The SS-ISO 5805[25] standard describes the meaning of several important concepts used in this thesis.

**Ride:**
measurable motion environment (including vibration shock, translational and rotational accelerations) as experienced by people in or on a vehicle.

**Ride quality:**
degree to which the whole subjective experience (including the motion environment and associated factors) of a journey or ensemble of journeys by vehicle is perceived and rated as favorable or unfavorable by passengers or operators

**Comfort:**
subjective state of well-being or absence of mechanical disturbance in relation to the induced environment (concerning mechanical vibration or repetitive shock)

1.1.2 Subjective Ratings

Today the most common method to evaluate vehicle ride and handling qualities is by subjective rating tests. When subjective rating tests are performed it is important that the methods are carefully constructed. Different rating form designs can result in more or less useful results. In order to either check or construct an objective measure, data from subjective evaluations are needed. This section first discusses some problems with subjective ratings and then describes a number of different rating techniques combined with issues to consider when rating tests are performed.
Common Problems with Subjective Ratings

When subjective rating tests are utilized in vehicle development as decision material for changes affecting dynamic properties, there are a number of issues that must be considered:

- Many issues not directly coupled to the ride or handling influence the test drivers, such as, tiredness and mood.
- Preconceived opinions among test drivers.
- Changes in test drivers’ acceptance levels over time.
- Differences between individuals in vocabulary used for rating.
- Property changes smaller than driver perception.

Even if these types of problems always exist in subjective ratings, a good rating scale may decrease their effect. We next give an overview of different rating techniques.

Design of Rating Scales

A common way to collect subjective ratings is through a number of statements that are given relative a predefined scale. This scale could either have a description of what the scale values correspond to or be anchored to a reference vehicle.

In the SAE recommended practice [28] a subjective rating scale for evaluation of noise and ride comfort characteristics related to motor vehicle tires is described. The method is especially interesting since it uses knowledge about the test driver’s experience. In the first step all test drivers judge the tested vehicle as unacceptable, borderline or acceptable. The raters are divided into groups such as trained observers which is a subgroup of critical observers which is a subgroup of some observers and so on, see the row of observer groups in Figure 1. If the main part of the trained observers answers “acceptable” the final rating will be 8 or 9 regardless of the other ratings. If the main part answers unacceptable the group to the left, in this case critical observers will be checked in the same way until a group with main answers acceptable is found. The grade will then be
set by the two numbers below that group, see Figure 1. A simpler more general subjective rating scale for handling is presented in an earlier version of SAE recommended practice [29].

<table>
<thead>
<tr>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>UNACCEPTABLE</td>
<td>BORDER LINE</td>
<td>ACCEPTABLE</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ALL OBSERVERS</th>
<th>MOST OBSERVERS</th>
<th>SOME OBSERVERS</th>
<th>CRITICAL OBSERVERS</th>
<th>TRAINED OBSERVERS</th>
<th>NOT OBSERVED</th>
</tr>
</thead>
<tbody>
<tr>
<td>INTOLERABLE SEVERE</td>
<td>VERY POOR</td>
<td>POOR</td>
<td>MARGINAL</td>
<td>BARELY ACCEPT.</td>
<td>FAIR</td>
</tr>
<tr>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
</tr>
</tbody>
</table>

Figure 1. SAE recommended practice for subjective rating scale for evaluation of noise and ride comfort characteristics related to motor vehicle tires [28].

Park et al. [15] form a rating test that separates driver’s assessment of a dynamic property from the opinion of whether it is an improvement or not. The idea is that the scale should not inflict opinion values to objective questions nor demand objective answers to opinion questions, since this will confuse test drivers. An example is given in Figure 2 where this is illustrated by two different questions about the damping properties of a vehicle. It is often possible to get more exact and repeatable answers if a reference vehicle is used. The choice of reference is of great importance, since it influences how accurate ratings test drivers are able to give.

| Type of damping: | harder | | | | softer |
|-----------------|--------|--------|--------|--------|
| Satisfaction with damping: | discomfort | | | | comfort |

Figure 2. Subjective rating scale with separated opinion questions and property questions.
Utilizing a reference vehicle can be taken one step further to a method called the couple-by-couple method, used by Zong et al. in [23] to evaluate vehicle handling stability. In this time consuming but interesting subjective rating approach, a number of vehicle configurations are ranked among themselves on a scale from best to worst. Drivers are driving two vehicle configurations at a time, unaware of which. This is possible since a driving simulator is used. They are only asked which configuration that is best of those two. Then they are presented with a new pair according to the schedule presented as the couple by couple method, see Figure 3. If one driver first drives vehicle 1 and 2 as one couple and then drive vehicle 3 and 4 as one couple, the better vehicles from each couple will form a new couple and the worse vehicle another couple and so on. The main advantage with this method is that drivers only compare two vehicles at a time, since it is difficult to remember the character of many vehicles.

![Schedule for test drivers using the “couple by couple” method, to rank 12 vehicles among themselves.](image)

**Figure 3.** Schedule for test drivers using the “couple by couple” method, to rank 12 vehicles among themselves.

**Interpretation of Ratings**

Since test-drivers have different opinions, sensitivity ability and interpretation of questions, the rating data is preferably processed
before use. Correlation diagrams over driver’s answers to questions and changing of dynamic properties will reveal these differences. Crolla et al. [1] perform correlation analysis between each test driver’s answer to each question and performed parameter change. From correlation diagrams it is possible to determine if the answer is systematically dependent on parameter changes or not. Random answers could possibly appear due to a number of issues discussed in the beginning of this section.

1.1.3 Objective Measures

Efforts have been made to create objective ride and handling measures. Ride quality is commonly estimated through measurements with accelerometers positioned to measure driver exposed vibrations. Humans will cope with vibrations from certain frequencies better than others, which make frequency spectra interesting to study. However, results from previous work show that it is hard to distinguish the small differences that test drivers perceive.

This section describes a number of methods for objective evaluation of vehicle ride. First some advantages with objective measures are stated and then followed by evaluation methods that are divided into two categories, stationary based and transient sensitive.

Advantages with Objective Measures

In the previous section many problems with the subjective ratings were discussed. Little et al. [10] concludes that the main advantages with objective evaluations are:

-Objectivity.

-Repeatability.

-Problem identification: Characterization of driving impression problems.

-Market positioning: Objective comparison between competitors.

-Scalability: Driving impression scales for comparison.
Objective Measures for Ride Evaluation

Methods Based on Stationary Assumptions

Many methods for ride comfort evaluation are based on the Power Spectral Density (PSD) estimate. When a PSD estimate is calculated in this work the Welch method is used, described by Stoica and Moses in [18]. Let \( x(n) \) denote the measured signal for which we shall estimate the spectrum. In the Welch method, data segments are allowed to overlap and can be represented as

\[
x_i(n) = x(n + iD) \quad n = 0,1,\ldots,M - 1 \quad i = 0,1,\ldots,L - 1
\]  

(1.1)

where the starting point for segment \( i \) is \( iD \) and the segment length is \( M \). Notice that if \( M > D \) there will be overlap e.g. \( M = 2D \) corresponds to an overlap of 50%. The total length of the signal is \( LD \). The \( i \)'th periodogram is calculated as

\[
\hat{X}^{(i)}(f) = \frac{1}{MU} \left| \sum_{n=0}^{M-1} x_i(n) w(n) e^{-j2\pi fn} \right|^2 \quad i = 0,1,\ldots,L - 1
\]

(1.2)

where \( U \) is a normalization factor that corresponds to the power in the window function, given by

\[
U = \frac{1}{M} \sum_{n=0}^{M-1} w^2(n).
\]

(1.3)

The Welch power spectrum \( \hat{X}(f) \) is then defined as the average of the periodograms in (1.2), i.e.

\[
\hat{X}(f) = \frac{1}{L} \sum_{i=0}^{L-1} X^{(i)}(f)
\]

(1.4)

The Hamming window function defined as

\[
w(n) = 0.54 - 0.46 \cos \frac{2\pi n}{M - 1} \quad 0 \leq n \leq M - 1
\]

(1.5)

together with a 50% overlap are used for the power spectral density estimates in this thesis. For statistical properties of this estimate we refer to Stoica and Moses [18].
A commonly used frequency based evaluation method is the *ISO 2631 Evaluation of Human Exposure to Whole-body Vibration* [24]. This standard describes an evaluation method on how vibrations affect health, comfort, perception and motion sickness. The ride quality is determined by the frequency contents in the acceleration signals. In ISO 2631 different weighting curves are used to determine how vibrations in different directions contribute to the level of discomfort. The definition of the directions in the standard is given according to Figure 4. This definition of directions is widely accepted in the area of vehicle dynamics and will be used throughout this thesis.

![Figure 4. Definition of directions in vehicle dynamical theory.](image)

The ISO weighting curves will suppress frequency contents that do not contribute much to discomfort. We now introduce the RMS function of a signal $x$ as

$$RMS(x) = \left[ \frac{1}{N} \sum_{n=1}^{N} x^2(n) \right]^{1/2}$$

(1.6)

where $N$ denotes the total length of the signal. The principal weightings are shown in Figure 5 where $w_k$ is the weighting for vibrations in $z$-direction, $w_d$ for $x$ and $y$ directions and $w_f$ for motion sickness in all directions. The weighted acceleration signal is denoted $a_w$, i.e. $a(t)$ is filtered with the corresponding filter from Figure 5 depending on the direction of $a(t)$. The basic evaluation measure in ISO 2631 is the weighted acceleration root-mean-square value, $RMS(a_w)$. 
This evaluation measure gives better results for stationary vibrations than for vibrations containing transients. The crest factor should be reported together with the weighted RMS value to indicate if it is appropriate to use. The crest factor is defined as the maximum instantaneous peak value of the frequency-weighted acceleration signal divided by the RMS value, i.e.,

\[
\frac{\max(a_w)}{RMS(a_w)}
\]  

(1.7)

Figure 5. Frequency weighting curves for principal weightings.

A similar evaluation method but more focused on ride quality in trucks is presented in SAE Recommended Practice: Measurement and Presentation of Truck Ride Vibrations [27]. Here today’s most common methods to measure truck ride vibrations are presented. This practice thoroughly describes how to perform measurements of truck ride vibrations. Positioning of the accelerometers is presented. Several
existing acceleration measures are described such as RMS, weighted RMS and absorbed power.

**Transient Sensitive Methods**

For signals with crest factors \((1.7)\) above 9, ISO 2631 proposes some additional methods. The maximum transient vibration value

\[
MTVV = \max_{n_0} \left\{ \frac{1}{\tau} \sum_{n=n_0-\tau}^{n_0} [a_w(n)]^2 \right\}^{\frac{1}{2}} \tag{1.8}
\]

is based on a running RMS value with some window length \(\tau\). Another measure that gives more weight to peak values is the fourth power vibration dose value

\[
VDV = \left[ \frac{1}{N} \sum_{n=1}^{N} a_w^4(n) dt \right]^{\frac{1}{4}}. \tag{1.9}
\]

Ushijima and Kumakawa [22] propose a method for objective evaluation of harshness. The method is based on multiple regression analysis to obtain an expression for the objective rating that corresponds to the subjective ratings. Test drivers graded the sense of shock and the sense of damping for different vehicles, driving over an obstacle with rectangular cross section. The analysis was performed on the logarithm of the peak acceleration value resulting from driving over the rectangular obstacle. Nine cars were tested, of which one was selected as reference and five were used for the multiple regression analysis and three for validation. Eight accelerometers were used in each vehicle but only the two correlating best with the subjective ratings were used in the objective rating equation. The authors conclude that more test drives are required to evaluate the method, since this original test did not show that good correlation.

An interesting approach is suggested by Giuliani and Ugo [4] to investigate ride comfort and not loose track of transient behavior. Test drivers drive some cars and trucks at different speeds on a specific test track. Data collected from accelerometers were processed according to existing methods such as RMS, Crest Factor and PSD. The windowed RMS, MTVV (1.8) is calculated and indicated by the dotted line in Figure 6.
Figure 6. The dotted line corresponds to MTVV (1.8). For each transient, a shock indicator is defined as RMS value subtracted from the maximum MTVV value.

Before the vehicle drives over the obstacle, a stationary RMS level is established. Drivers seem to be sensitive to changes in this level rather than the absolute values. To predict the sense of shock the RMS value is subtracted from the maximum value of the MTVV (1.8). The shock indicator is marked in Figure 6 where the dotted line correspond to the MTVV and the dashed line correspond to the RMS value of the signal. Lee and White [9] aim to produce an objective measure on the impact harshness that corresponds to subjective harshness evaluations. Different suspension configurations are tested on a vehicle and wavelet analysis is performed on the measurements instead of PSD. The damping ratios together with the initial peaks of the major vibration modes are used for a multiple regression analysis. Good correlation between subjective ratings from vehicles with similar modifications and objective measures are obtained.

1.1.4 Conclusions

The most important factor of objective evaluation methods is that they should correlate well with how most test drivers perceive the vehicle. All objective methods that use some evaluation formula based on subjective data are dependent on correctly performed rating tests. A weak link from subjective ratings to objective measures could be the
result of poorly performed rating tests. Structures and choices made regarding subjective rating scales must be thoroughly considered. Large differences in vehicle configurations will probably lead to a distinct connection between subjective and objective evaluations. Tests performed on actual vehicles instead of driving simulators tend to have weaker connections. This may be due to the fact that it is easier to accomplish large and evident property differences in driving simulators. Property differences in vehicle development are however often small and require sensitive measures.

In many cases subjective evaluations have to be used to conclude which vehicle that is the most comfortable, since available measures are not sensitive enough.

**Problems with Existing Objective Evaluation Methods**

Most ride and handling measures in this literature survey have problems with the link between objective measures and subjective ratings. In many cases problems with subjective rating evaluations occur and in other cases the objective measures are based on parameters that do not reflect the driving impressions. Due to the lack of accurate and reliable objective measures, ride comfort is today mostly subjectively evaluated with test drivers.

Efforts to measure discomfort by using ISO weighting curves together with PSDs have been quite successful. As long as big differences in the RMS values can be shown and the accelerations are quite stationary, the subjective ratings seem to correlate well with the objective measures. However, as ride comfort in heavy vehicles is constantly improving, there is a growing need for more sensitive objective methods to evaluate ride.

One additional problem is that transients seem to have a large effect on the ride quality and the averaging in these kinds of measures, c.f. (1.4) decreases the sensitivity to transients.

**Demands and Desirable Qualities of Objective Measures**

When an objective measure is developed the following points should be taken into consideration.
Demands on objective measures:

- Agreement with subjective ratings.

- Reproducibility i.e. the measurements should not depend on the actual driver.

Desirable qualities of objective measures:

- No need for a special test track.

- No complex testing procedures with demands like constant velocity.

- Indication of which alterations that will lead to improvement.

1.2 Thesis Outline and Contributions

In Chapter 2 a study on driver’s ability to sense small differences in vehicle behavior is performed. This work does not only focus on how small differences drivers are able to detect but what type of differences that they pay most attention to. This study has been published in:


The work from this sensitivity study shows that transients in the signals play an important role in the subjective rating process. Therefore a new ride comfort evaluation method that considers transient events separately is suggested. This method, called “The Ride Diagram”, is presented in Chapter 3 and more thoroughly
investigated in Chapter 4 and Chapter 5. In Chapter 4 vehicle simulations with different suspension parameters are used to investigate the ride diagram behavior. In Chapter 5 some vehicle evaluations are performed with ride diagrams based on measurement signals from a real truck.

The ride diagram was first presented in:


and a simulation study was presented in:


When vehicle ride comfort is optimized for a single driving condition, then the resulting parameters will probably not be optimal for other load cases. There have however emerged new technical solutions where parameters, e.g. damping, may be varied adaptively. For these cases estimation of the motions to control are crucial. Chapter 6 discusses estimation of modal coordinates from damper displacements using system identification. This work was presented in:


Finally in the last Chapter 7 a summary of this work is given together with some proposals for future work.
This chapter investigates driver sensitivity to changes in vehicle ride response. A test procedure is presented where the objective is to quantify driver perception limit for small changes in typical vehicle behavior. The procedure starts with measured cab motions from a tractor semi trailer combination with different cab suspension settings. The measured motions are then recreated in a motion simulator that has a platform where the test person is seated. Changes are made in small steps by combining measured signals from different settings and the test driver has to determine whether he/she could sense a difference between two vehicle motions or not. Several combinations and intensity levels are tested. The outcome of the answers from several test persons is finally used for quantification of driver perception sensitivity using time and frequency domain difference measures.
An important part of this study is to investigate the validity of only considering the PSD spectra as a tool to determine whether test drivers feel differences between two vehicles or not.

2.1 Introduction

Since decisions regarding vehicle dynamics, e.g. chassis and cab suspension tuning, are based upon subjective evaluation, test driver perception sensitivity is one important factor that will indicate how small changes that are meaningful to test. If the test driver’s sensitivity to changes in vehicle response were well known, effort could be spared in the iteration process for the best vehicle suspension setting. In most existing objective evaluation methods e.g. linear regression models for ride and handling, subjective ratings are one building block, [9,17]. This because it is crucial that the objective measures correspond to how most drivers perceive the vehicle. A high correlation between vehicles with large dynamic differences and subjective ratings is easy to obtain but of little value in vehicle development where differences between design iterations often are small. Furthermore, if the subjective ratings do not originate from vehicle characteristics due to driver sensitivity being lower than the induced difference in response, problems with the objective measures will arise.

Human vibration sensitivity has, when it comes to fictitious signals, been investigated in earlier research by Griffin [3]. This work is however difficult to apply in the case of complex vehicle motion since it only considers people’s lowest threshold for sensing one acceleration signal in one direction and one frequency at a time. The work in this thesis will therefore focus on sensitivity to changes in typical vehicle behavior.

One of the main objectives of the work in this chapter is to establish an effective test procedure to measure driver perception sensitivity to parameter changes affecting vehicle ride. An overview of the different steps in the methodology is presented in Figure 7.
The measurement procedure on trucks with different cab suspension settings is briefly described in Section 2.2. The measured data is processed to work as input data in motion simulator tests. The motion simulator provides means to test many different settings accomplishing very small differences, even smaller than the differences between the measured trucks. Test drivers are placed in the motion simulator moving as a real truck and asked questions where they have to determine whether they can sense a difference between similar motions. The pre-processing of measured data is described in
Section 2.3 and the simulator tests are further described in Section 2.4. To quantify differences between motions of three degrees of freedom, both a time domain measure and a frequency domain measure are suggested in Section 2.5. Test drivers’ answers from the simulator tests are processed according to human perception theory. This is discussed in Section 2.6. Results of the study are presented in 2.7 where the driver perception sensitivity is evaluated as functions of the two difference measures. Conclusions are given in Section 2.8.

### 2.2 Vehicle Measurements

In this section a short description of the measurement procedure is given. The measurements are performed with accelerometers placed under the driver seat. Measurements of cab accelerations are performed while driving a tractor semi trailer combination on a special track at Scania’s test course at 85km/h. Each recording is 20 seconds long and consists of a number of transients with relatively high acceleration level during the whole sequence. Some approximate properties of the signals are shown in Table 1 where the notation of directions is according to Figure 4.

<table>
<thead>
<tr>
<th>Directions</th>
<th>X [m/s²]</th>
<th>Y [m/s²]</th>
<th>Z [m/s²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS</td>
<td>1.9</td>
<td>1.9</td>
<td>1.5</td>
</tr>
<tr>
<td>Maximum peak</td>
<td>8</td>
<td>10</td>
<td>7.5</td>
</tr>
</tbody>
</table>

Table 1. Approximate signal intensity of the motion signals.

The sensors, three accelerometers and a gyroscope, are positioned under the seat to measure the 6-degree of freedom motion of the cab. This measurement point is chosen since it corresponds to the motion generation point in the simulator and makes it possible to make corresponding measurements in the simulator, eliminating
recalculation for comparison. Four trucks with different cab suspension bushings were measured.

2.3 Generation of Signals for Simulator Tests

Signals from the vehicle measurements are used as input to the vehicle simulator. The acceleration signals are integrated twice since the reference signals to the simulator are positions. The simulator environment makes it possible to accomplish arbitrary small differences in a controlled way. This is very useful when evaluating driver sensitivity to small differences in vehicle response. However, the manipulation of data must be performed with care since the simulator must emulate important dynamics of a real truck. Section 2.3.1 describes the first method of achieving motions with small differences based on a convex combination of time signals. In Section 2.3.2 an alternative method is suggested where two signals are created with equal phase information but different PSD spectra. The latter type of signal is useful when evaluating the influence of the phase information on the drivers’ sensitivity to motion differences.

2.3.1 Combining Time Signals to Generate Small Differences

A simple way to achieve small differences in a signal would be to simply scale it with a constant. But this is not preferable since it does not correspond to how vehicle motion changes with different parameter settings. Design iterations rarely decrease amplitudes for all frequencies, but amplitude distribution change e.g. when reducing a single disturbing frequency. Sometimes it is just the transient’s curve form or the phase lag between motions in different directions that make a vehicle more or less comfortable. Therefore another approach is used where signals are varied using different combinations of two measured vehicle configurations, cf. Figure 8. The resulting signal $s(t)$ is given by a convex combination of two signals, $s_1(t)$ and $s_2(t)$, originating from two different vehicle configurations, i.e.

$$s(t) = \lambda s_1(t) + (1 - \lambda) s_2(t) \quad 0 \leq \lambda \leq 1.$$  

(2.1)
The utilized procedure enables continuous transition from the first to the second vehicle configuration and is possible to apply when time signals are synchronized.

Figure 8. Position signals from two trucks with different cab suspension, changed in small steps.

2.3.2 Generation of Phase Equivalent Signals (with different energy)

A commonly used measure to quantify driver comfort is by way of the Power Spectral Density (1.4) of measured accelerations. In order to investigate to what extent PSD spectra capture differences in vehicle behavior, a virtual time signal, $s_{\text{virt}}(t)$, is created. Starting with measured signals from two different vehicles, the virtual signal is created by combining the amplitude contents from Signal 2, with the phase information from Signal 1. This is achieved by scaling the Fourier transform of Signal 1 by the frequency dependent amplitude factor

$$H(f) = \frac{S_1(f)}{\sqrt{S_2(f)}} \quad (2.2)$$

where $S_1(f)$ and $S_2(f)$ are the PSD of Signal 1 and Signal 2, respectively. The virtual signal will thus have a PSD spectra close to Signal 2, i.e.

$$S_{\text{virt}}(f) \approx S_2(f) \quad (2.3)$$
and be phase equivalent with Signal 1. This new signal will be interesting to use later on to determine whether it is only the amplitude differences that test drivers pay attention to or not.

2.4 Simulator Tests

As illustrated in Figure 9, the measured cab motions are recreated in a motion simulator. Test drivers are seated in the same position as in the actual vehicle with the same driver seat. A steering wheel is also used to ensure the correct sitting position. Throughout this test, driver exposed vibrations are measured on the simulator in the same way as in the vehicle.

Figure 9. A Scania tractor semi trailer combination together with the servo electrical motion platform used to generate motion for sensitivity measures.

2.4.1 Test Procedure

Four test persons, two trained test drivers, one inexperienced driver and one driver with long experience are used. Perception sensitivity of each person is investigated separately. All test persons start at the
same difference intensity. The difference intensity was increased if they did not feel the difference end decreased if they had many correct answers. All test drivers judge signals generated according to both Sections 2.3.1 and 2.3.2.

Table 2. Typical test sequence with random order of pairs used at one intensity level.

<table>
<thead>
<tr>
<th>PAIR</th>
<th>ANSWER*</th>
<th>RESULT</th>
</tr>
</thead>
<tbody>
<tr>
<td>2,2</td>
<td>no</td>
<td>corr. rej</td>
</tr>
<tr>
<td>1,1</td>
<td>no</td>
<td>corr. rej</td>
</tr>
<tr>
<td>2,1</td>
<td>yes</td>
<td>hit</td>
</tr>
<tr>
<td>2,1</td>
<td>no</td>
<td>miss</td>
</tr>
<tr>
<td>1,1</td>
<td>no</td>
<td>corr. rej</td>
</tr>
<tr>
<td>1,2</td>
<td>yes</td>
<td>hit</td>
</tr>
<tr>
<td>2,2</td>
<td>yes</td>
<td>false al.</td>
</tr>
<tr>
<td>1,2</td>
<td>yes</td>
<td>hit</td>
</tr>
</tbody>
</table>

*Did you feel a difference? (yes/no)

Test persons compare signals in eight pairs at each intensity level, see Table 1. Four out of eight pairs consist of two different signals and the remaining four consists of identical signals. The eight signal pairs are presented in random order.

2.5 Proposed Difference Measures

In order to quantify differences in vehicle response for different configurations, two different measures are used. One difference measure is defined in the time domain and focuses mostly on transients. This measure is described in Section 2.5.1. The other measure, described in Section 2.5.2 is defined in the frequency domain. This enables us, for example, to study whether these types of small differences in vehicle behavior are better detected by frequency or time domain measures.

All differences in this work are generated around a working point. Both utilized measures include a division with the signal intensity to make them more applicable to other operating points according to Weber fractions [13]. This has however not yet been evaluated. For a
brief illustration of Weber fractions, assume that a person can barely detect the difference between something with intensity 10 and 10.1 then the detection level around 20 will be 0.2, i.e. the sensitivity scales linearly with the operating point.

### 2.5.1 Time Domain Difference Measure

Human experience seems to put a lot of weight on transients, cf. Giuliano and Ugo [4]. The difference in time domain between two signals may therefore be calculated considering only the part of signals above a certain magnitude. First define

\[
b(t) = \begin{cases} 
1 & \text{RMS}(s_1) + \text{RMS}(s_2) < |s_1(t) + s_2(t)| \\
0 & \text{otherwise} 
\end{cases} \tag{2.4}
\]

as an indicator to define the limit where transients occur for the signals \(s_1(t)\) and \(s_2(t)\). By using (2.4) it is possible to define a time domain difference measure for each direction of acceleration by integrating the signal difference when the transients occur. For example assume that \(s_1(t)\) and \(s_2(t)\) correspond to acceleration signals in the \(x\)-direction, then the corresponding difference measure is defined by

\[
\Delta^t_x = \frac{\int_0^T |s_1(t) - s_2(t)| b(t) dt}{\int_0^T \frac{1}{2} |s_1(t) + s_2(t)| b(t) dt} \tag{2.5}
\]

where \(T\) denotes the total signal length.
Figure 10. Difference calculated from time domain capturing the transient differences and differences in curve form. Solid line corresponds to \( s_1(t) \) and dashed line corresponds to \( s_2(t) \).

In Figure 10 it is illustrated how the difference between signal \( s_1(t) \) and \( s_2(t) \) is calculated. The total time difference measure, \( \Delta' \), is defined as

\[
\Delta' = \sqrt{\Delta'_x^2 + \Delta'_y^2 + \Delta'_z^2}
\]  

(2.6)

where \( \Delta'_x \), \( \Delta'_y \) and \( \Delta'_z \) are calculated according to (2.5) for each direction of motion.

### 2.5.2 Frequency Domain Difference Measure

A similar measure as (2.5) is also defined in the frequency domain where the difference measure is based on PSD spectra of motion accelerations. Let \( S_1 \) and \( S_2 \) be the PSD in the x-direction, then the frequency measure is defined as

\[
\Delta'_x = \frac{\max \left\{ |S_1(f) - S_2(f)| \right\}}{\frac{1}{2} \left( S_1(f) + S_2(f) \right)}
\]  

(2.7)

where \( F_{\text{max}} \) is half the sampling frequency, i.e. Nyquist frequency. The difference measure is illustrated in Figure 11 as the marked area.
frequency domain difference measure will increase when the energy level changes or when the energy shifts from one frequency to another.

This measure will typically reflect changes in stationary vibration levels and not be sensitive to changes in transient response. As for the time domain measure one difference is calculated in each direction according to (2.7) and then combined as

\[ \Delta^f = \sqrt{\Delta_x^f{}^2 + \Delta_y^f{}^2 + \Delta_z^f{}^2} . \]  

(2.8)

### 2.6 Human Perception Theory

Human perception theory Heeger [5] is a part of the science branch psychophysics that deals with human’s ability to sense external stimuli in the form of e.g. audio or visual signals. Sensory thresholds are established by presenting a stimulus of different intensity to test persons while asking whether they can sense it or not. Two types of thresholds may be established, \textit{absolute threshold}, which is the lowest intensity that can be detected and \textit{difference threshold}, which is the intensity difference required for discrimination of two stimuli.

When presenting a stimulus of certain intensity at different occasions the test subject will typically sense it in some, but not all cases. The
perception threshold is therefore commonly defined as the intensity when the test subject can detect or discriminate the stimulus in 50% of the cases. However, each time the test subject indicates detection or discrimination without stimulus being present must also be considered. Otherwise a strategy to give positive answers in all cases would result in illusive low perception threshold. Depending on if stimulus is present or not and the test subject’s answer, there are four possible outcomes, see Figure 12.

![Figure 12. Possible outcomes from a trial in a detection test.](image)

### 2.6.1 Internal Response

In the process when the test subject takes decision to answer yes or no there are two factors contributing to uncertainty, internal and external noise. The surrounding environment defines external noise. Internal noise comes from the fact that the neural response in the brain (internal response) is noisy and variable. Identical stimulus intensity will thus result in different internal response at different occasions. Some internal response will also result when no stimulus is present. Assuming that internal response for noise alone and stimulus plus noise follow Gaussian distributions, Figure 13 may be used to illustrate the decision making process.
When the internal response is above a certain level, test subject will answer yes. Different test subjects however use different criteria for this decision. This is illustrated by the position of the vertical line denoted criterion response. The ratio between hits and false alarms determines how separated the two distributions are, see Figure 14.
If the distributions for noise alone and stimulus plus noise overlap each other, no choice of criterion can result in 100% hits and 0% false alarms. A stimulus with higher intensity will shift the stimulus plus noise distribution to the right, thus separating the two curves. Only when the two distributions are fully separated a 100% hit rate combined with 0% false alarm is possible.

Receiver Operating Characteristic (ROC) illustrates possible choices of criterion for the test subject in a single graph. The ROC curve is generated by shifting the criterion response line from right to left while looking at the ratio between hits and false alarms, see Figure 15.
When the two distributions only have small overlap, Case a in Figure 15, the ROC curve will have a steep beginning and a flat ending. It is thus possible to combine high percentage hits with low percentage false alarms. When the two distributions coincide, Case c in Figure 15, the hit rate will be equal to the false alarm rate, independent of criterion choice. Case b in Figure 15, describes the case when test persons begin to have difficulty separating the stimulus.

2.6.2 Psychometric Function

A common way to illustrate sensitivity thresholds is by psychometric functions. These functions illustrate the percent of cases for which a certain stimulus intensity is detected. In this work the percent detections is defined as the percent of hits when less than 5% false alarms occur, see Figure 16. This means that the ability to detect differences is based on the distribution alone and not the choice of criterion response position. Comparison may thus be made between different test persons.
2.7 Results

We will now evaluate the results from the test person’s answers together with the time difference measure (2.6) and the frequency domain difference measure (2.8). The rate of false alarms and hits for each stimulus intensity and each person will determine the detected differences in percent according to Section 2.6.2. This information is combined with either the frequency- or the time difference measure and used to create psychometric functions describing each person’s sensitivity threshold to changes in vehicle response. A good difference measure should be high if humans are sensitive to the applied difference and low if humans are insensitive to the motion difference.

2.7.1 Time Domain Difference Measure

The result will now be shown as psychometric functions where we have difference size on the x-axis and percent detection on the y-axis. The detection threshold is often defined as the value when test persons have over 50% detections. The result from all four test drivers on all tested signals is shown in Figure 17. This result in four different psychometric functions, one curve for each test person, see Figure 17. Each point in the diagram corresponds to one difference intensity level tested by eight pairs as described in Section 2.4.1. The difference in
each point could be of different character like time combinations, phase equivalents etc.

Figure 17. Psychometric functions based on time difference measure. Each curve corresponds to one test driver.

Independent of difference source the time difference measure increase when people’s ability to sense differences increase. One exception is the solid line, which corresponds to the inexperienced driver. The time domain difference measure seems to increase independent of the type of difference. It does not seem to matter if the signals in one point are phase equivalents or time combinations. Each line in Figure 17 consists of three time combinations and one point with phase equivalents.

2.7.2 Frequency Domain Difference Measure

In Figure 18, the dashed line represents signals generated with the phase equivalent method presented in 2.3.2. The solid line corresponds to signals generated as time combination in 2.3.1
Figure 18. Psychometric functions based on frequency difference measure.

From this test it is possible to see that test persons have more difficulties separating signals with the same phase information, in spite of equally large differences in the PSD-spectra. The percent detections do not increase with increased difference according to the frequency difference measure. This result show that the frequency measure is not suited for this purpose. Somehow the phase information as well as the amplitude information has to be considered both in sensitivity measures and objective evaluation methods.

2.8 Conclusions

The presented methodology provides measures to investigate and quantify driver perception sensitivity to changes in vehicle response. In this work the smallest perceived difference starting from measured typical vehicle motion is examined. Presented measures for driver perception sensitivity are suited for differences in real vehicles, first to be used in the development process when tuning vehicle parameters, secondly as information when objective evaluation methods are developed.

One advantage with the utilized methodology to recreate motion in a simulator is that vehicle measurement strategies may be validated against simulator subjective ratings. This will make subjective rating tests and vehicle measurements both more accurate and reliable, and
also contribute to the search for objective measures of driving impressions. Signals with different types of differences were created and used in the detection tests. Some had similar amplitude spectra and different phase information and others had the opposite. Regardless of which signal type the time domain measure increased then test drivers ability to detect differences increased. Results in Section 2.7.2 from tests with phase equivalent signals show that PSD spectra do not capture enough information to use as base for objective evaluation of vehicle response. From the above observation we conclude that the proposed time difference measure is better suited than the frequency difference measure to measure human’s ability to detect differences. Perception threshold information needs to be used in vehicle development since it is only meaningful to test design iteration steps larger than driver sensitivity.
3 Description of the Ride Diagram

For an objective ride measure one fundamental difficulty is to condense vehicle measurement data but preserve relevant information. Within handling evaluation, a common way to graphically visualize vehicle properties is using handling diagrams of the type developed by Pacejka [14], where vehicle steering effort is plotted versus lateral acceleration.

The handling diagram in Figure 19 is an example of illustrating differences in vehicle character. Since a handling diagram is calculated from the difference in front and rear wheel slip angles it does not give the complete picture. The vehicle body slip angle as a function of the lateral acceleration is in this case the corresponding “intensity level”.
The basic idea with the handling diagram is to illustrate changes in vehicle handling by studying the understeer gradient, which is defined by

\[ K_{us} = \frac{\partial}{\partial a_y} (\delta - \delta_A) \]

where \( \delta_A \) is the Ackerman angle and \( \delta \) is the front wheels steering angle. Figure 19 shows a handling diagram for two trucks with and without tag axle steering. The purpose of the tag axle steering is to improve low speed maneuverability. Tag axle steering is proportional to front axle steering but designed with a dead zone so that it does not steer for typical highway driving. It can be seen that both trucks are understeered, \( K_{us} > 0 \) for small \( a_y \). However during evasive maneuvers on highways the tag axle steering may be activated due to larger steering angles. This causes the tag axle steered truck to become oversteered. The above example is used to illustrate the following. Not only is the absolute (or average) value of the understeer gradient of interest. How the understeer gradient changes with increased lateral acceleration is also important. Most drivers react negatively to
vehicles with sudden changes in the understeer gradient, i.e. changes in $K_{us}$.

Within ride, both RMS and PSD are commonly used as evaluation measures. While a PSD diagram gives more detailed information than a single RMS value, both methods have the disadvantage of smoothing out transient events by averaging. We saw in Chapter 2 that this is undesirable.

With increased excitation amplitude many vehicle systems initially respond in a linear fashion i.e. with proportional increased response amplitude. Due to limitations in possible displacements for various suspension elements, e.g. springs and bushings, a further increase in excitation eventually results in a nonlinear response and thus a change in character. A hypothesis in this thesis is that humans react negatively to sudden changes in vehicle character. The change in vibration level may thus be as important as the absolute vibration level. A good measure for any vehicle property should therefore capture and display both changes in intensity and character.

In this chapter we propose a new evaluation method for ride comfort, the ride diagram. This ride diagram is defined in Section 3.1 and a limit used for separating transient and stationary behavior is presented and motivated in Section 3.2. The shape of the ride diagram will be the result of the road character and the vehicle character. This will be illustrated in Section 3.3. A more thorough investigation is performed in Chapter 4 and Chapter 5.

### 3.1 A new Measure: The Ride Diagram

According to the sensitivity study in Chapter 2, test persons base a lot of their ride impression on the highest peaks in the acceleration signals. Jerk is furthermore sometimes considered as the main source of vibration discomfort, cf. Quanan and Huiyi [16]. The main objectives with this new measure are that it should condense large amounts of ride measurement data, capture transient events and show vehicle response to increased excitation.

A new time domain measure based on peaks in the time signal is therefore proposed. The time domain measure results in two values for each excitation level and these values are then used to create a diagram.
Objective Measures for Ride Evaluation

Definition of the Ride Diagram
Calculating the curves which define the ride diagram is basically a three step process. First the signal is divided into segments separated by sign changes in the derivative of $a(t)$, according to Figure 20. We define the peak samples of the acceleration signal of interest, $a(t)$ as
\[
\Omega = \{n \mid a(n-1) > a(n) < a(n+1) \text{ or } a(n-1) < a(n) > a(n+1)\} \tag{3.1}
\]
and denoting these samples by $n_k$, $k=1,2,\ldots,N_k$, where $N_k$ is the total number of peaks. The $k$'th segment can be written formally as
\[
a_k = \{a(n)\}_{n=n_k}^{n_{k+1}} \quad k=1,2,\ldots,N_k-1 \tag{3.2}
\]
It is now possible to define one peak-to-peak value for each segment as
\[
ptp(k) = \left| \max(a_k) - \min(a_k) \right|, \quad k=1,2,\ldots,N_k-1. \tag{3.3}
\]
We will set $ptp(0)=0$.

Since the suspension travel is limited, high $ptp$ values only occur when high jerk is combined with high accelerations.

The second step consists of classifying each segment as either “stationary” or “transient” depending on if its $ptp$ value is less or
greater than a limit later defined in Section 3.2. The transient sample points are now defined as

\[ a^{k}_{\text{trans}} = \begin{cases} 
\{a(n)\}_{n=n_k}^{n_{k+1}} & \text{ptp}(k) > T_{\text{limit}} \land \text{ptp}(k-1) \leq T_{\text{limit}} \\
\{a(n)\}_{n=n_k}^{n_{k+1}} & \text{ptp}(k) > T_{\text{limit}} \land \text{ptp}(k-1) > T_{\text{limit}} \\
0 & \text{otherwise}
\end{cases} \quad (3.4) \]

where \( k=1,2,..,N_k - 1 \). In Figure 21 an acceleration signal where segments have been classified as either transient or stationary is shown.

![Figure 21. Segment of an acceleration signal, where the dotted line segments are sorted as transient.](image)

Calculating the mean square values, \( MS \), is the third step. With the transient samples of the acceleration signal separated it is possible to write the \( MS \) value for the transient part of the signal as

\[ MS_{\text{trans}} = \frac{1}{N} \sum_{k} \sum_{n} \left| a^{k}_{\text{trans}}(n) \right|^2 \quad (3.5) \]
where \( N \) is the total number of samples of \( a(n) \). The stationary part is defined as the part of the signal which is not classified as transient, and for this part we define the stationary \( MS \) value as

\[
MS_{\text{stat}} = \frac{1}{N} \sum_{i=1}^{N} a(n)^2 - MS_{\text{trans}}.
\]  

(3.6)

Mean square values are calculated for each excitation level (vehicle speed) and the resulting \( MS_{\text{trans}} \) and \( MS_{\text{stat}} \) values are finally plotted versus vehicle speed, as shown in Figure 22.

Figure 22. Ride Diagram based on measurements from a Scania tractor semi trailer combination with two different cab suspension settings.

Note that the total \( MS \) value at a certain excitation level is the distance between the transient \( MS \) and the stationary \( MS \), cf. Figure 22.

3.2 Limit for Transients

The limit \( T_{\text{limit}} \) in (3.4) determines which segments that are classified as transient and stationary, respectively. A limit dependent on the signal RMS is chosen for separating transients from stationary in the acceleration signal based on the results from Chapter 2 which show that a similar floating limit worked well picking out signal parts that humans pay most attention to. It is defined as the signals energy equivalent amplitude.
\[ T_{\text{limit}} = 2\sqrt{2}RMS(a). \] (3.6)

It is important that the limit is linked to the total energy of the signal. The limit for the ride diagram is designed to work as it does in Figure 23. If a pure sinusoid is considered all data ends up on the stationary side.

![Diagram](image)

Figure 23. Example on how the chosen limit separates transient and stationary in two cases. One pure sinusoid and one with an increased amplitude for one period.

### 3.3 Road Influence

When comparing different vehicles by utilizing a ride diagram, all measurements should be performed on the same road if speed is used to vary excitation. The shape of the ride diagram will depend both on the type of road and the vehicle.

In order to investigate how the floating limit for separating transients works, some additional plots are presented for two different roads, denoted Road A and Road B. Road A is a really bad road on the Scania test track consisting of a number of concrete blocks with sharp edges, Road B is a highway like road with smooth asphalt and very small and few transients. In Figure 24 the percent of peak to peak distances is plotted versus the size in a histogram.
Objective Measures for Ride Evaluation

Figure 24. Histogram over the peak to peak amplitude distribution. The arrows mark where the transient limit is for each signal. The histogram is calculated with a resolution of 0.1 m/s².

It can be observed that the solid line corresponding to Road A has a much more flat distribution. The vertical lines correspond to the limit between what is classified as transient and stationary segments for each signal. Figure 24 only shows this distribution on one vehicle configuration at one speed. But in Figure 25 the percentage of ptp’s that are classified as transient for Road A and Road B respectively, are plotted versus speed for a vehicle with 5 different suspension settings. Each symbol corresponds to one setting and the vehicle is driven over the two different tracks at speeds between 25-85 km/h.
From these results one can observe that the percentage of the signal considered transient increase if a rougher road is considered and that it also increases with excitation (speed).

3.3.1 Main Character Depending on Road

The ride diagram shown in Figure 22 was obtained on a track with many obstacles (quite high excitation) resulting in a wide separation between the curves corresponding to the transient and stationary parts (rapidly increasing $MS$) and with a fairly symmetrical V-shape around the speed axis. Driving on a smoother road with few obstacles would result in a lower $MS$ i.e. a narrower V-shape tilted towards the stationary side. In Figure 26 it is illustrated how the V-shape widens and tilts for two different road types.

Figure 25. Percentage of peak to peak segments considered transient for Road A (black) and Road B (gray). The different symbols correspond to different vehicle configurations.
Figure 26. Principal sketch of how the road affect the main shape of the Ride Diagram.

Figure 26 shows a principal sketch of how the road input will affect the main shape of the ride diagram. A rough road that contains many obstacles will typically result in a wide V-shape, Figure 26, # 1. The narrower V-shape slightly tilted towards the stationary side denoted # 2 will typically originate from a smooth highway like road with a few obstacles.

3.4 Summary

From discussions in Chapter 1 it is clear that there is a need for more sensitive objective evaluation measures. Common objective methods have the disadvantage that transient events in the ride experience are not captured. The sensitivity tests in Chapter 2 show that test drivers pay attention to the transients in the signal. A new evaluation method called the ride diagram has therefore been suggested in this chapter. It has some similarities with the handling diagram described in [14] since it shows vehicle behavior as a function of excitation. But it also condenses large amounts of data without losing track of transient behavior. This new evaluation method will be investigated with data
from both simulations and measurements in Chapter 4 and Chapter 5, respectively.
4 Evaluation of the Ride Diagram Using Simulations

When objective evaluation methods such as the ride diagram are suggested there are two main properties that should be investigated. A first step is to examine how the objective evaluation results depend on typical vehicle parameters. To be able to improve the tested vehicle based on results displayed in the ride diagram, they must be linked to design parameters in a known manner. A second step is to make subjective evaluations that show how test driver’s opinions are linked to the ride diagram shape. Investigating the first step with simulations enables easy change of parameters in a noise free and controlled environment. This is the objective of the present chapter.

In Section 4.1 the road used for simulations is presented. Section 4.2 describes the vehicle model used for simulations. This section is
followed by Section 4.3 where suspension analysis is performed by varying different parameters in the model and plotting the resulting ride diagrams. Finally, the conclusions from the simulations are presented in the last in Section 4.4.

4.1 Road Profile Used in Simulations

A synthetic road is generated and used as excitation input in the simulations. The road is built up by combining two types of road unevenness sources, sinusoids and steps, which may be considered as stationary and transient road events respectively. Road generation is done as follows.

4.1.1 Generation of Sinusoid Road Input

This section describes how a road is built up to have an amplitude spectrum with slope according to ISO 8608 [26]. This road is basically generated from a sum of sinusoids with different spatial frequency and random phase. Special consideration is taken to enable good frequency coverage, despite the use of finite number of sinusoids, within the defined wavelength range. In this way a more continuous distribution of frequencies is achieved.

4.1.2 Generation of Transient Road Input

A second road consisting of positive steps with a random height between 0.03 and 0.06 m is also generated. The positive steps are realized as short and steep ramps, see dash dotted line in Figure 27.
Figure 27. Solid line shows the resulting road profile with sinusoids and superimposed step. The dash dotted line shows only the steep ramp signal.

This implementation of transients does not affect the road spectrum since a ramp has practically the same frequency distribution as an ISO road. Furthermore the impact will increase with increasing speed for a ramp. A selected number of 20 positive steps are uniformly distributed over the road stretch of 1 km.
Figure 28. Amplitude spectra for sinusoid road, step road and combined road. The dotted line for the combined spectra is on top of line for sinusoid road spectra.

The sinusoid road is finally combined with a version of the step road where the linear trend has been removed. A road stretch of 1 km length is generated and Figure 27 shows a representative part of the combined road profile where a step occur. The diagram in Figure 28 shows corresponding road amplitude spectra. The road generation approach above enables arbitrary division of the road total mean square value between stationary and transient excitation, with the extreme cases being roads consisting solely of sinusoids or steps.

4.2 Simulation Model

The vehicle simulations are performed with the quarter car model shown in Figure 29 which is implemented in Matlab. The quarter car model is a two mass model where the mass \( m_a \) corresponds to the axle mass and \( m \) corresponds to the chassis mass. \( k_t \) and \( c_t \) denotes the tyre stiffness and damping. In this particular study only the chassis suspension parameters \( k \) and \( c \) are varied. A more accurate model of the truck dynamics should of course include cab and seat suspension. The objective with this model is however not to accurately describe a
truck, but instead to investigate how the ride diagram shape is affected by varying vehicle parameters in general.

\[
F = f(z - z_a)
\]

Figure 29. Quarter car model used for modeling vertical vibrations.

To enable study of non linear behavior, the spring in the secondary suspension is modelled with bump stop character outside the linear range, see Figure 30. The maximum compression distance without reaching the bump stops is denoted \(\delta\), and effects from this are studied in Section 4.3.2.

![Characteristics for spring and bump stop in secondary suspension](image)

Figure 30. Characteristics for spring and bump stop in secondary suspension.

The nominal model parameter values, shown in Table 1, are chosen to represent a typical truck with a chassis eigen frequency at 1 Hz and
axel frequency at 10 Hz. The resulting frequency behavior is shown in Figure 31.

<table>
<thead>
<tr>
<th>NOMINAL PARAMETER VALUES</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k$</td>
</tr>
<tr>
<td>$c$</td>
</tr>
<tr>
<td>$m$</td>
</tr>
<tr>
<td>$k_t$</td>
</tr>
<tr>
<td>$c_t$</td>
</tr>
<tr>
<td>$m_a$</td>
</tr>
</tbody>
</table>

Figure 31. Transfer function properties of the quarter-car-model with parameters set according to Table 1.

4.3 Suspension Analysis

To illustrate how the ride diagram may be used for suspension analysis, the quarter car model is simulated on the synthetic road described in Section 4.1. The simulations are performed in Matlab version 7 using the ODE45 solver. Four cases are investigated and a ride diagram is generated for each case. The first case in Section 4.3.1
4 Evaluation of the Ride Diagram Using Simulations

investigates relative damping for the sprung mass. The second case in Section 4.3.2 investigates limited suspension travel for the secondary suspension. The third case described in Section 4.3.3 investigates different eigen frequencies for the sprung mass with a constant limited suspension travel. In Section 4.3.4 a fourth case with a resonant road input spectra is used together with varied sprung mass eigen frequency.

4.3.1 Varying Relative Damping

The ride diagram in Figure 32 shows the result from variation of sprung mass relative damping $\xi$, which is approximated by

$$\xi \approx \frac{c}{2\sqrt{k \cdot m}}$$  \hspace{1cm} (4.1)

This approximation is made by looking only at the transfer function from axle mass position to chassis mass acceleration. This transfer function has a pole polynomial given by

$$s^2 + \frac{c}{m} s + \frac{k}{m}$$  \hspace{1cm} (4.2)

The relative damping has been varied from 90 to 110% of its nominal value in steps of 10 %. This has been realized by scaling the damping constant $c$ around its nominal value given in Table 1. As seen in Figure 32, in this case higher damping decreases the stationary part, $MS_{\text{stat}}$, and at the same time the transient part is approximately kept constant. For this load case a completely linear model was utilized.
Figure 32. Ride diagram with varying relative damping for sprung mass.

### 4.3.2 Varying Suspension Travel

Three different suspension travel distances are simulated, the compression travel limit $\delta$ is set to 0.12, 0.1 and 0.08m respectively. The resulting ride diagram is shown in Figure 33. From this figure it can be seen that as the stationary part, $MS_{stat}$, of the mean square value is approximately the same for the three parameter settings. The transient side in the ride diagram on the other hand clearly reveals a difference, with the smallest free suspension travel resulting in highest $MS_{trans}$ value.
4 Evaluation of the Ride Diagram Using Simulations

4.3.3 Varying sprung mass eigen frequency

Figure 34 shows results from simulations performed with different sprung mass eigen frequency, \( \omega_0 \) and a constant limited suspension travel. The nominal eigen frequency for the chassis mass is approximated according to (4.2) by

\[
\omega_o \approx \sqrt{\frac{k}{m}}. \tag{4.3}
\]

\( \omega_0 \) is then varied by setting

\[
k = \alpha^2 \cdot k_{nom} \tag{4.4}
\]

\[
c = \alpha \cdot c_{nom} \tag{4.5}
\]

with \( \alpha \) changing valued from 80 to 110% in steps of 10% maintaining the same relative damping.

Four different suspension settings are simulated. The stationary vibration level increase with stiffer suspension as can be expected. The softest suspension may however not always be the best choice since it for increased excitation will be the first to reach the suspension limit travel. This type of reasoning is every day work for test engineers when tuning vehicle suspension. In this case the ride diagram illustrate how much better (i.e how much softer) it is possible

Figure 33. Ride diagram with varying free suspension travel.

<table>
<thead>
<tr>
<th>Speed [km/h]</th>
<th>MSstat</th>
<th>MStrans</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>35</td>
<td>35</td>
<td>35</td>
</tr>
<tr>
<td>45</td>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>55</td>
<td>55</td>
<td>55</td>
</tr>
<tr>
<td>65</td>
<td>65</td>
<td>65</td>
</tr>
</tbody>
</table>

\[ Less\ suspension\ travel \]

\[ ([m/s^2]^2) \]
to set the suspension without reaching the limit for the suspension travel for a specific excitation level.
An increase of the transient part might be considered worse than an increase of stationary part. Depending on how the two different parts contribute to the total ride impression, it may not be optimal to minimize the total mean square value.

![Ride Diagram](image)

**Figure 34.** Ride diagram resulting from varying eigen frequency of sprung mass.

### 4.3.4 Resonant Road Input

In reality there exist no roads with a road amplitude spectrum completely following a straight line as in Figure C.2-Classification of roads in the ISO standard [26]. All real roads have some overrepresented frequencies. If such a “resonant” road is considered the ride diagram will alter its shape. The overrepresented spatial frequency will for some speed be transformed to a time domain input frequency that corresponds to the eigen frequencies of the truck, see the
resonant peaks in Figure 31. When speed is used as excitation increase the MS value will change in a nonlinear fashion for resonant roads. In Figure 35, the ride diagram, obtained from simulations of three vehicles with different $\omega_0$, are shown. The road used in this simulation has one dominant frequency. The nonlinear vibration increase, due to the resonant frequency, mainly occurs on the stationary side even if some acceleration increase also ends up at the transient side. The vehicle with the stiffest suspension will have the highest peak occurring at the highest speed. The hypothesis is that even if this nonlinear vibration increase is not the result of a nonlinear component, test drivers will find it uncomfortable anyway. In Section 5.1.1 test drivers opinion on two vehicles with this kind of behavior is presented.

![Ride Diagram for three vehicles with different $\omega_0$. Simulated on a road with one resonant frequency.](image)

Figure 35. Ride Diagram for three vehicles with different $\omega_0$. Simulated on a road with one resonant frequency.
4.4 Transients and Stationary Behavior

When utilizing the ride diagram together with the quarter-car model it is also of interest to study the time domain signals and to observe what kind of events that are selected as transients. This is not always obvious since the sorting algorithm utilizes a floating limit which depends on the signal $RMS$ value. There are two different cases when signals typically are considered transient: when transient road obstacles occur and when the suspension travel reaches its limits. These events are of course linked since the model will have large suspension travel over obstacles. Effects from this kind of nonlinearity occur at the transient side. The limit proposed in Section 3.2 to discriminate transients from stationary behavior, seems to work as intended since the transients that originate from road transients and the nonlinear behavior in the suspension, are selected as transient, see Figure 36.

![Figure 36](image_url)

Figure 36. Time signals from a simulation with the quarter car model showing transient segments. The upper plot shows how the suspension compresses when the ramp in the road profile (solid line) occurs at 76.8s. This result in an acceleration peak in the lower plot that is classified as transient.
4.5 Conclusions

The proposed way of evaluating and visualizing vehicle ride objectively by way of the ride diagram has four main advantages:

-It is fairly simple to interpret.

-It shows the absolute vibration level.

-It considers transient events separately.

-It shows changes in vehicle character with increasing excitation.

Since the ride diagram shows the combined vehicle-road interaction effect, it may be used as indicator of what vehicle settings that best fit specific road conditions. The ride diagram has the advantage over conventional ride measures such as RMS and PSD spectrum in that it considers transient events separately. Simulations with a quarter car model show that the ride diagram in several examples is able to separate transients due to nonlinear behavior, e.g. limited suspension travel, from the stationary vibrations. Furthermore, the effects from choosing a softer suspension with a limited suspension travel were studied and the pros and cons were clearly visualized. A better understanding on how changes in vehicle properties affect the ride diagram is thus provided. This insight is essential to enable ride comfort improvements by shaping the ride diagram. It looks promising to utilize the ride diagram for ride comfort evaluations.
In this chapter we pursue the evaluation of the ride diagram using measurements from real trucks. Measurements are performed on a Scania tractor semi-trailer combination. Characteristics of shock absorbers and bushings are altered between the different sets of measurements. In this chapter there are two parameter changes tested, first there is cab suspension bushings in Section 5.1 and then there is chassis suspension dampers in Section 5.2. The applied parameter changes in Section 5.1 are the result of altered cab bushings and are small. They correspond to typical decisions that have to be taken by test drivers when tuning ride comfort in vehicle development. Generally these changes are difficult to detect for the average driver. The applied parameter changes in Section 5.2 that concerns the
chassis suspension dampers and is large. The results from measurements and interviews with test drivers are summarized in section 5.3.

5.1 Altering Cab Bushings

The idea with the ride diagram is to characterize trucks by studying changes in ride comfort with excitation. When ride comfort is concerned, the excitation input is mainly defined by road profile and vehicle velocity. In this experimental study the excitation intensity is changed by driving over the same road profile at different speeds. A special test track at the Scania proving ground is driven at different constant speeds, starting from 25 up to 85 km/h with 5 km/h increments. The track is built up by obstacles of various heights and lengths, differing on left and right sides. This corresponds to a very rough road, offering broad band excitation. Accelerometers are used to measure 6 degree of freedom cab motion. Since excitation depends both on the road profile and the vehicle speed, driving slow on an uneven road in some way corresponds to driving faster on a more even road.

5.1.1 Test Drivers Opinions

The intention with the ride diagram is to describe ride comfort qualities in a more detailed way than e.g. a single RMS value and be a complement to the PSD spectra. The presented simulations in Chapter 4 show that the method behaves as intended separating transient events effectively. In the ride diagram that was presented in Figure 22, measurements on a truck with two different suspension settings are shown. A general opinion among test drivers is that the setting corresponding to the solid line is the most comfortable. It can be observed that the dotted line has a steeper slope change than the solid line at 55 km/h. Even if this change of slope is not caused by limitations in suspension travel for the chassis mass it is possible that other suspension elements on this complex multi degree of freedom mechanical system, such as bushings, behave in a nonlinear manner. A hypothesis is however, that people react negatively to sudden slope
changes since they expect a linear behavior. An interesting observation from test drivers that supports this statement is that after driving over this test stretch at 25 km/h with 5 km/h speed increments at each round, 60 km/h is considered to be the most uncomfortable speed. If drivers expect a linear increase of vibration, 60 km/h would result in the most unexpected impression.

5.2 Different Damper Settings

In order to investigate ride diagrams from real measurements a Scania tractor with semi trailer was used. Accelerometers were placed in the cab and three different damper settings were tested on two different public roads. The tested damper settings are described in Figure 37.

![Figure 37. Illustration of three vehicles with different cab suspension settings. The white truck will later on, correspond to dotted lines.](image)

The public roads were, old E20 between Nykvarn and Mariefred and ordinary E20 on the same stretch. Old E20 is a quite bad Swedish countryside road with several parts of broken asphalt, ride diagrams from this road is presented in Section 5.2.1. The ordinary E20 is a highway with good asphalt quality, ride diagrams from this road is described in Section 5.2.2.

5.2.1 Countryside Road (bad road)

In Figure 38, ride diagrams for three vehicles with different damper settings are shown driving on a countryside road. This road is
considered a bad road. The “white” vehicle, corresponding to the dotted line, is worst at both the stationary side and the transient side. The stationary side is large because the low damping results in large low frequency motions since the resulting oscillations are poorly damped. The transient side is also large since it reaches the suspension travel limit for the worst transients.

The gray and black vehicle have similar behavior even though the difference in damping is very large between them. A problem with the black vehicle configuration is that the chassis damper becomes so hard that the tires and cab suspension move instead of the chassis suspension. Chassis and axle mass thus start to behave more as one mass. The probable effect from this will be lower cab accelerations but higher forces on the chassis.

5.2.2 Highway (smooth road)

In Figure 39, ride diagrams for three vehicles with different damper settings are shown driving on smooth highway. This ride diagram could be compared with the ride diagram from simulations in Section 4.3.1. In both ride diagrams the damping is varied.
It can be observed that the black vehicle behaves as if it had less damping than the gray vehicle, which is explained in Section 5.2.1. Observe that all the ride diagram curves from the highway measurements fit into the gray marked triangle area in Figure 38. This illustrates the difference in excitation between the two roads.

A truck is a complex mechanical system with many suspension elements. From these two measurement cases in Section 5.2.1 and Section 5.2.2 it can be seen that it is difficult to exactly explain what happens in the ride diagram when parameters are varied. This is one of the reasons for investigating the general behavior of the ride diagram with simulations.

However, test drivers opinions seem to correlate with the shape of the ride diagram. The black vehicle and the grey vehicle behave approximately the same on the bad country road. The white vehicle is considered the worst and at 90km/h it reaches the suspension travel limit and is found to be very uncomfortable.

On the highway it is hard to sense the differences between vehicles. Further work will focus on determining how test driver perceive stationary and transient vibrations differently.
5.3 Conclusions

The ride diagrams from measurements show promising results so far. When the ride diagram was investigated through simulations in Chapter 4 some basic behavior as the result of changes in vehicle parameters were established. These shape changes in the ride diagram were the result of parameter changes of about 10 to 20%. The established effect from parameter changes is probably valid for parameter changes of the same size around the nominal values. If very large parameter changes are made other phenomena can occur.
6 Estimating Vehicle Modal Motion

One main feature with the ride diagram is that it can in some cases show advantages and disadvantages with different suspension settings. If vehicle parameters are optimized for one driving condition, then they will probably not be optimal for another load case. In a truck, chassis suspension damping is commonly included by individual passive hydraulic dampers. A drawback that follows from individually mounted dampers is that damping of different vehicle modes may not be selected directly, but instead results from the geometrical positioning and the setting of the dampers. Better adjustment possibilities are obtained by using electrically controlled dampers, which enable distributing the damping modally. Utilizing modal damping commonly means prescribing damping for sprung mass eigen modes in bounce $z$, pitch $\theta$ and roll $\varphi$ (see Figure 4 for notation).
With electrically controlled dampers it is possible to build suspension systems where the damping can be varied during ride through adaptive control. For these systems, estimation of the motions to control is crucial. This chapter contains such an example where there is a need for improved estimation of modal motions.

In this chapter, system identification is suggested as tool to obtain estimates of the modal coordinates only based on measurements of the damper displacements. Another solution for this would simply be to put sensors on the chassis and directly obtain measurements of the modal coordinates. But sensors are costly and the displacement sensors are already used in the air suspension to measure and control the position of the chassis and cab.

The background to this work is described in Section 6.1 where some previous work together with the main idea is described. Different sources for modal motion are discussed in Section 6.2. The system identification approach is applied on data from a MBS model in Section 6.3. Based on the experiences from the simulation environment, the applicability of the estimation method is investigated on measurements from a real truck in Section 6.4. Some of the properties of the obtained models are discussed in Section 6.5 and the chapter is concluded in Section 6.6.

6.1 Background

In this study modal coordinates are calculated for a two axle vehicle with only the four relative damper displacement velocities denoted, $\dot{\delta}_1$, $\dot{\delta}_2$, $\dot{\delta}_3$ and $\dot{\delta}_4$, available as inputs. The front left damper is denoted with subscript index 1, front right 2, rear left 3 and rear right 4. In previous work by Holen and Thorvald [6] geometric relations are derived for modal coordinate velocities for a tractor. The relations are derived under the assumptions that the road is a flat surface and that the chassis body move. This results in the following geometric relations

$$
\ddot{z}_{GR} = \frac{(\dot{\delta}_1 + \dot{\delta}_2)}{2}(1 - \lambda) + \frac{\dot{\delta}_3 + \dot{\delta}_4}{2} \lambda,
$$

(6.1)
\[ \dot{\theta}_{GR} = -\frac{(\dot{\delta}_1 + \dot{\delta}_2)}{2L} + \frac{(\dot{\delta}_3 + \dot{\delta}_4)}{2L}, \]  
\[ \dot{\phi}_{GR} = \frac{(\dot{\delta}_1 - \dot{\delta}_2)}{t_{d1}} (1 - \kappa) + \frac{(\dot{\delta}_3 - \dot{\delta}_4)}{t_{d2}} \kappa, \]

where \( L \) denotes the vehicle wheel base, \( t_{d1} \) and \( t_{d2} \) are the front and rear axle damper distances, \( \lambda \) is the relative position of the mass center and \( \kappa \) is a weighting factor for different front and rear roll damping, cf. Figure 40. Subscript GR denotes estimation through geometrical relations. In this previous work pitch center is approximated to be at the center of mass position.

Simulations using a non-linear MBS (Multi Body System) model of a two axle tractor semi-trailer combination were presented in Holen and Thorvald [7]. Results from these simulations show that although modal damping works sufficiently well for low frequency sprung mass motions, problems occur for transient road inputs. When the road surface change the assumptions in (6.1)-(6.3) are not valid.

Figure 40. An overview of the notation used in (6.1)-(6.3).

Further analysis in Holen and Thorvald [8], based on the same vehicle model, indicate that the geometric modal coordinate relations of bounce, pitch and roll (6.1-6.3) show discrepancies to actual motion. This depends on the fact that the road surface is not flat, chassis is not
a rigid body and possibly there are probably small errors in e.g. the damping factor \( \kappa \) and the relative position of the mass center \( \lambda \) etc. From the results in [6-8], it may be concluded that better estimates of the modal coordinates are needed as inputs to modal damping systems. Therefore, we propose to utilize a system identification approach to improve estimation of modal coordinates.

System identification deals with the topic of obtaining mathematical models for dynamic systems based on measured data, see, e.g., Ljung [11]. Applications that use identification are commonly found in various engineering disciplines where good models for analysis and control are desired. Recent examples in vehicle modeling are Elahina et al. [2] and Milanese et al. [12] where identification is used to model heavy truck frame and passenger car vertical dynamics.

The initial step in an identification process is to select a model structure. We will use linear models of low order with the damper displacement velocities as inputs and one of the modal coordinates as output. Measured data for the system of interest, i.e. input and output signals, is then used to obtain parameters for the chosen model structure. These parameters are identified on one part of the data, the estimation data, and are verified on another set of data, the validation data. More details on the identification procedure are provided in Section 6.3 and Section 6.4. But first different sources that induce modal motion will be described.

### 6.2 Excitation Sources for Modal Motion

There are two main sources for modal motion: first there are road induced motions and secondly there are driver induced motions.

#### 6.2.1 Road Induced Motions

Road induced motions result from different types of unevenness in the road. Obstacles and other unevenness give rise to tyre forces that affect the vehicle. Driving over a plank will typically result in a bounce and pitch motion. If a one sided obstacle is encountered, also a roll motion will occur. In this thesis the focus is on road induced motions since these motions define ride comfort.
6.2.2 Driver Induced Motions

Driver induced motion means different types of motions that are the result of driver inputs. Driver input could for example be steering, braking, gear shifting etc. These maneuvers generate forces in lateral and longitudinal directions that make the chassis sprung mass move. When these types of motions are considered for modeling it would be preferable to use additional signals closely connected to the driver input such as steering wheel angle, brake pedal position etc. Since the aforementioned sources of motions are fundamentally different, separate models should be used for each source. This is due to the limited flexibility of the linear model structure to capture motions originating from different types of excitation. It is however possible to have several models, e.g. one for each type of excitation, where switching between models could be triggered by signals such as brake pedal position. The main focus in this chapter will however be on road induced excitations, and not driver induced inputs such as braking or steering.

6.3 Modal Coordinate Estimation on MBS Model

In an attempt to better estimate the modal coordinates, one low order MISO (Multi Input Single Output) model is identified for each modal coordinate of the studied vehicle system. Each model is identified with the four relative damper displacement velocities as inputs and one of the three modal coordinate velocities, bounce, pitch or roll as output. The utilized MISO models are on state-space form, which is a common model structure for identification of mechanical systems. With state-space form, the relationship between input \( u(t) \) and output \( y(t) \) at the discrete time step \( t \) is written as a system of first-order differential equations using an auxiliary state vector \( x(t) \):

\[
\begin{align*}
    x(t+1) &= Ax(t) + Bu(t) \\
    y(t) &= Cx(t) + e(t)
\end{align*}
\]

The values of the A, B and C matrices in (6.4) and (6.5) are in this work identified using the MATLAB® System Identification Toolbox. The estimation command was used with the Prediction Error Method.
(PEM) was used with focus set to ‘Simulation’ (see the system identification toolbox for more details). Since these MISO systems have the same input and output as the geometric relations they are easily compared with the results from these relations.

### 6.3.1 MBS Simulations

The initial analysis using the system identification approach for modal coordinate estimation is performed on simulation data from the MBS model previously described in Holen and Thorvald [8]. This model of a 36100kg tractor semi trailer combination is composed of several sub models such as axels, frame, cab, fifth wheel and semi trailer (totally 49 DOF). Although the model has been validated versus measurement data in [7], its behavior is typical for the vehicle type rather than an exact description of the specific vehicle later used for modal coordinate estimation based on real measurements. MBS models generally provide a noise free developing environment with full repeatability. New design improvements are often easily implemented which makes them suitable for evaluating various damping control strategies, such as modal damping. A further benefit is the possibility to evaluate a fairly wide range of load cases before planning any truck measurements. This to gain knowledge of what type of excitation that is preferable as estimation data.

To be able to identify good models it is necessary that the estimation data contain input and response information from various road types and vehicle maneuvers. Several road files were generated and evaluated for this purpose. A general observation is that estimation data that contains both individual and simultaneous damper excitation generates the most accurate models. When estimation data contains such information, the identification algorithm is given the best possibilities to determine how the input signal affects the vehicle modes, both individually and weighted together. Figure 41 shows a road profile with a series of road bumps generated for this purpose. From the MBS model, estimation data are saved as output steps with 0.002s sampling interval and include both the relative damper displacement velocities and the modal coordinate velocities.
6.3.2 Identification on Simulation Data

A third order state-space model was estimated from data collected driving over the obstacles in Figure 41. The model was then validated on a number of load cases including, positive steps (double and one-sided), pot holes and an ISO road as the one described in Section 4.1. A very good model fit is obtained for the estimation data. Figure 42 shows simulations from one of the validation stretches, the ISO road. It can be seen that the identified model is able to capture the true bounce velocity quite well whereas the geometrical relations perform much worse.
Figure 42. MBS simulation data of bounce velocity driving over an ISO road. The solid line corresponds to the measured bounce velocity and the dash dotted line corresponds to the estimated bounce velocity from the identified third order model. The dashed line corresponds to the geometrical relations obtained from (6.1)-(6.3).

6.4 System Identification on Data from Measurements

In Section 6.3.1 it was shown in the simulation environment that the estimation of the modal coordinates can indeed be improved by using system identification cf. the curves in Figure 42. To verify that the method also is applicable to real vehicles we will in this section apply identification of modal coordinates based on measured data from a real truck. Section 6.4.1 describes the measurement procedure and data collected from different types of roads are used for identification in Section 6.4.2.

6.4.1 Vehicle Measurements

A two axle tractor with semi-trailer, was equipped with a measurement system and data from various load cases were obtained. The relative damper displacement velocities were derived by taking
the derivative of the damper displacements measured with cable position transducers (CPT). The bounce velocity was derived by integrating the vertical acceleration signals, measured with accelerometers on the chassis. Pitch and roll velocities were derived by taking the derivative of the chassis roll and pitch angles measured with a gyroscopic platform. All data were sampled with 0.005 s time interval and some signals were low pass filtered to remove noise and make it possible to calculate the derivative.

### 6.4.2 Identification on Measurement Data

In this section one model for estimating vehicle pitch velocity is presented. The model is estimated on data corresponding to a series of bumps somewhat similar to the best working estimation load case in the simulation environment, see Figure 41. The model is then validated against two additional sets of measured data, one with low excitation and one with medium excitation.

A state space model of third order is identified in the same way as described in Section 6.3. The obtained MISO model can be used to estimate pitch velocity from damper displacement velocities.

**Estimation Data with High Excitation**

The road bumps in the estimation data are of different heights but are considered as high excitation resulting in large damper displacements. The output from a simulation based on the identified model for the pitch velocity is close to the measured pitch velocity for the estimation data. This is illustrated in Figure 43 where a part of the estimation data including a bump is shown.
Figure 43. Signals from estimation data. The solid line corresponds to the measured pitch velocity and the dash dotted line corresponds to the estimated pitch velocity from the third order system identification model. The dashed line corresponds to the geometrical relations obtained from (6.1)-(6.3).

It could probably be possible to estimate better models if the input excitation for the identification data is chosen more carefully.

**Validation with Low and Medium Excitation**

The validation of the identified model is performed on two different roads. The first road is considered as medium excitation. This road is a typical Swedish countryside road and is also used in Section 5.2.1. One section with data from this road is shown in Figure 44 together with simulated pitch velocity based on the identified model and geometrical relations. The system identification shows improved estimates of pitch velocity.
6 Estimating Vehicle Modal Motion

Figure 44. Validation data for medium excitation on ordinary Swedish country road. The solid line corresponds to the measured pitch velocity and the dash dotted line corresponds to the estimated pitch velocity from the third order system identification model. The dashed line corresponds to the geometrical relations obtained from (6.1)-(6.3).

A corresponding validation is also performed for a smooth Swedish highway, considered as a road with low excitation. The results of this validation are shown in Figure 45.
Figure 45. Validation data for low excitation on highway. The solid line corresponds to the measured pitch velocity and the dash dotted line corresponds to the estimated pitch velocity from the third order system identification model. The dashed line corresponds to the geometrical relations obtained from (6.1)-(6.3).

From Figure 44 and Figure 45 it is clear that the identified model perform much better than the geometrical relations in (6.1)-(6.3) not only on estimation data, but also on these validation data. This validation shows a great improvement of the pitch velocity estimate based on identification compared to the geometrical relations. The main improvement is that the 10 Hz disturbance from the axle motions is filtered out.

6.5 Model Properties

The properties of the obtained linear MISO models are examined below. Since they all have similar character, only one of them is shown. We will consider the frequency responses of the third order pitch model used in Section 6.4. The model has a typical low pass character, which preserves frequencies up to 1-2 Hz and gradually removes frequency contents above that, see Figure 46. These properties seem reasonable from a mechanical point of view, since the
chassis pitch mode has an eigen frequency around 1 Hz. The relative damper displacement velocities used as inputs have a significant amount of energy around 10 Hz originating from the axle motions. The vehicle suspension will absorb this motion and it will not result in any chassis pitch motion at 10 Hz. This can be compared with the geometrical relations, (6.1)-(6.3), where the 10 Hz contents appear in the modal chassis coordinates estimates.

Figure 46. Frequency response function of the 3rd order pitch model used in Section 6.4 where the front left damper is the input.

The selection of a suitable model order is a known difficulty within system identification and there are many different approaches. It is possible to use different penalty functions such as e.g. the Akaike information theoretic criterion [11], but just using some physical insight about the real system and investigation of the retrieved models can be just as useful.
6.6 Conclusions

The results show that low order MISO models obtained with a system identification approach can be used to estimate modal coordinate velocities with considerably higher accuracy than geometric relations. The identified models have been evaluated on data both from vehicle simulations and from vehicle measurements. The new estimation approach is shown to be applicable both for analysis with MBS models and for on-board vehicle estimation.

Although good models were obtained for all tested load cases, their validity is somewhat limited to road induced excitation. This since driver induced maneuvers such as braking and steering wheel action excite the vehicle differently compared to road induced excitations from obstacles and road bumps. By utilizing additional signals such as brake pedal position or steering wheel angle in the identification procedure and by further model refinements, improved estimates can be obtained also for driver induced maneuvers.
7 Summary and Future Work

In this final chapter we will summarize our results and provide an outlook on possibilities for further work.

7.1 Summary

It is very important that objective evaluation methods to determine vehicle ride quality correlate well with how most test drivers perceive the vehicle. Developing evaluation methods that fit with subjective data for large differences in vehicle configurations is easy. But in vehicle development, differences are however often small and sensitive measures are needed. All ride comfort measures have to condense large amounts of data into a representative format, and in this process transient differences often disappear due to averaging.
In Chapter 2 a method to determine drivers’ perception sensitivity to small changes in vehicle motion is proposed. In this work, two different ways of quantifying the differences in a three degree of freedom motion is suggested. One measure is based on differences in transient behavior and the other is based on differences in PSD (Power Spectral Density) spectra. These difference measures are then combined with the answers from test drivers that had been processed through human perception theory. The results from this test show that test drivers’ reactions to changes in vehicle motion were more connected to the transient behavior than the PSD.

The need for more transient sensitive measures combined with a wish to condense vehicle data without losing information lead us to develop the ride diagram. The ride diagram has influences from the handling diagram, which presents vehicle character as a function of excitation. It also has an additional feature which separates transient from stationary, while still showing the total energy.

Two aspects of this newly proposed objective evaluation method have to be investigated before it can become useful in vehicle development, namely:

-How is the shape of the ride diagram affected by typical changes in vehicle suspension parameter?

-How is the ride diagram shape connected to test driver opinions?

This thesis has focused on the suspension parameter aspect which was investigated first in a simulation environment and then on data from measurements. Results from this showed that the ride diagram behaves as expected, and in addition reveals transient behavior where other common methods do not.

This thesis ends with a chapter on vehicle modal motion estimation through a system identification approach. New technology has made it possible to vary vehicle suspension parameters during vehicle ride. In order to prescribe different damping for different vehicle modes, modal motion estimates are needed. The modal motions are estimated with the relative damper displacements used as inputs. The system identification approach show improved estimates compared to previous methods.
7.2 Future Work

A new evaluation method for ride comfort evaluation has been suggested in this thesis. Three approaches to continue the work with the ride diagram are suggested in this section. First we propose to investigate the link between subjective opinions and the shape of the ride diagram. Secondly, a new interesting approach to utilize road measurements to keep track of different excitation levels is suggested. Finally, it could be interesting to use data from measurements on a road simulator to create ride diagrams.

7.2.1 Linking the Ride Diagram Shape to Subjective Opinions

Simulations and measurements have been performed with various vehicle parameter settings. The data from these tests have been plotted in ride diagrams and how parameter changes affect the ride diagram shape has been established. However, it is also essential to link the shape of the ride diagram to test driver opinions.

One idea towards this objective is to collect subjective opinions live during measurements, i.e. let test subjects hold a button in their hand that they press when they feel uncomfortable. This signal could then be utilized to determine what people find uncomfortable in the vibration signals. The following issues could, for example, be studied:

- How much of the uncomfortable vibrations will be classified as transient vibrations?

- Which types of transients are considered uncomfortable by test drivers?

7.2.2 Utilizing Road Measurements to Derive another Form of Excitation Increase

In this thesis all ride diagrams are made with speed as the variable by which the excitation is increased. This has, in some cases, the
disadvantage that the evaluation becomes very connected to a certain road and that road’s resonant frequencies determine the shape of the ride diagram, cf. Section 4.3.4. If the road profile was known it would open up new possibilities in determining response to road irregularities. The signals could for example be divided into different excitation levels depending on the road profile input. A road profile section containing many bumps would typically be sorted as high excitation. An alternative to using speed on the excitation axle could be found.

If road measures were utilize many other investigations would be possible, a linear model could for example be identified for low excitation. The model could then be simulated with road measurements as input and the output from the model could be compared with measured accelerations from the truck. Nonlinear vehicle response may in this way be detected when output signals from the model and measurements differ.

### 7.2.3 Utilizing a Road Simulator to Derive another Form of Excitation Increase

Using speed as excitation increase has some disadvantages as discussed in Section 7.2.2. But the only possibility to increase excitation linearly in a systematic way when looking at vehicle response to road irregularities is by increasing speed. However in a road simulator it would be possible to manipulate the input signals. A road simulator does not correspond to real driving conditions in all senses, e.g. no longitudinal forces are generated, but it might still be very useful.
8 Bibliography


[26] Mechanical vibration-Road Surface profiles-Reporting of measured data, ISO 8608, INTERNATIONAL STANDARD

[27] Measurement and Presentation of Truck Ride Vibrations SAE Recommended Practice, September 1999. SAE J1490
