Seat Vibration Simulator for Ride Comfort Evaluation

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A thesis submitted in fulfillment of the requirements for the degree of Master of Engineering in the

Cognitive Robotics
Mechanical, Maritime and Materials Engineering

September 20, 2018
Abstract

Ride comfort is one of the aspects of vehicles that manufacturers are constantly trying to improve, and seat vibration is an important contributing factor to ride comfort. The present study developed a seat vibration simulator for investigating the human perception of seat vibration and its application on ride comfort evaluation.

The simulator was delivered as a fully functional system. The design requirements of the simulator were defined based on the review of proving ground measurements and previous studies. It concluded that the operating frequency range was 10-80Hz with excitation applied in both longitudinal and vertical directions. A compact and cost-effective design was proposed featuring a modular floating vibration platform and compact electrodynamics actuators. The Time Waveform Replication (TWR) process was implemented as the control strategy of the simulator and yielded good accuracy for tracking the target stimuli within the frequency range of 6-80Hz.

The second part of the present study discussed the application of the seat vibration simulator. An experimental measurement method was proposed and validated for measuring the difference threshold of human perception of seat vibration, including the spectrum baseline method for generating reference stimulus from proving ground measurements, the modified A-AFC protocol for reducing the experiment duration, and the software for the fully automated experiment. The proposal also included the potential application of the difference threshold as a new ride comfort indicator.
Acknowledgements

I would like to express the deepest appreciation to my supervisor Barys for his incredible guidance and support through all the entire year. I consider myself very lucky to have a supervisor who had Skype meeting with me on Sunday evening.

My special thanks to my daily supervisor Marco and Xabier for guiding me through the entire project and supporting me on all kinds of matters, especially when I made mistakes.

I got enormous help from my colleagues at Toyota Motor Europe. I would like to thank Mickael for his help on developing the experiment program, supporting some of the crazy ideas that I had, and discourage the other even crazier ones (which is even more important). I would like to thank Dr. Cor-Jacques Kat for letting me participate in his JND experiment, from which I have gained valuable first-hand experience. I would like to thank Wael, Peter, and JeanLeon for their participation in my experiment, which was not an easy task at all. I also got a lot of help from my colleagues at TU Delft, especially from Francesco who was also doing an internship at Toyota Motor Europe. I would not be able to finish my project without the coffee breaks we had together.

I would like to thank my parents, for their unconditional love and support. I would like to thank my wife Xiaorong, for her understanding and encouragement.
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Chapter 1

Introduction

Ride comfort is one of the aspect of vehicles that manufacturers are constantly trying to improve, and seat vibration is an important contributing factor of ride comfort. The present study developed a research tool for human perception of seat vibration and investigated its application for objective evaluation of ride comfort. The present study was conducted at Toyota Motor Europe as an internship project.

The background of this study was introduced in Section 1.1, which explained the need of better understanding of human perception of ride comfort. The challenges of evaluating seat vibration objectively were explained in Section 1.2, which is the research problem of the present study. Finally, the objective of the study is specified in Section 1.3.

1.1 Background

Ride comfort is defined as "subjective state of well-being or absence of mechanical disturbance" according to ISO 5805 [34]. It is generally accepted in the automotive industry that the main sources of such disturbance affecting the ride comfort are "oscillations which reach the vehicle’s passenger compartment and cause noise, vibration, or both" [29].

The importance of ride comfort in passenger vehicles is constantly growing, especially due to the recent development in automated driving technologies. One of the main drivers of higher level of automated driving is "Enable user’s freedom for other activities when automated systems are active" [11]. Recent studies have shown that people will conduct more non-driving activities such as resting, using smartphone, reading, and watching movies [40]. These activities are more demanding regarding the ride comfort of vehicles because:

- drivers spending more time "eyes-off-road" in autonomous vehicles make them more susceptible to motion sickness [16].
- the non-driving activities are more likely to be affected by the vibration of vehicle [74].

There are both mechanical factors and the psychological factors contributing to ride comfort [29]. The mechanical factors are the vibration and noise perceived by the occupants, such as the seat vibration, steering wheel vibration, interior noise, etc. The psychological factors often involves the occupants’ mood, expectations, activities in the vehicle, etc. The interaction between different factors further complicated the problem. For example, the perception of vibration is often found affected by noise
Chapter 1. Introduction

[4]. Investigating all these factors would be very difficult if not impossible. The present study focus on the seat vibration for the following reasons:

- the majority of the human mass is supported by the seat with the reminder supported by the feet support [21]. Therefore the human-seat interface is where most of the energy transfer from vehicle to human body is taking place.

- the vibration causing the discomfort of drivers and passengers is often predominantly the whole-body vibration [22], which is most likely the result of seat vibration. On the other hand, vibration at other locations, such as steering wheel or floor, are less likely to cause whole body vibration.

The knowledge about human perception can be applied to objective indicator of ride comfort. Traditionally, ride comfort of vehicles can only be evaluated subjectively due to the lack of knowledge regarding human perception of vibration. The evaluation is often conducted by highly-trained test drivers who drive the vehicle on selected tracks or roads, then give ratings based on their opinions regarding the performance of the vehicle. The objective evaluation does not rely on judgment of test drivers, instead it is based on the measurements of vehicle, such as the acceleration at various positions. These measurements are processed to give indicators that represent the performance of the vehicle. It is generally accepted that objective evaluation methods have various advantages over subjective methods, such as better efficiency, accuracy, repeatability, and lower cost.

1.2 Problem description

Despite the huge effort that was put into investigating human perception of vibration, it is still far from being fully understood. The researches of this field date back to 1960s, and multiple standards were established regarding the objective evaluation of seat vibration, such as ISO2361 [33], BS6841 [7], etc. These standards were mostly based on the studies in 1980s and haven’t changed much until today. However, they have seen limited usage in automotive industry for the evaluation of ride comfort [50], although they were established for long time. They were often criticized for resulting mismatches with subjective ratings [38] and being unable to accurately differentiate small differences of seat vibration\(^1\). Some new evaluation methods or improvements of the standard methods were proposed in recent publications, but none of them were fully validated to be able to solve these two problems.

Accurately evaluating the human perception of seat vibration is facing various challenges. The first among them is the complexity of seat vibration itself. The vehicle seat vibration has the characteristics of broadband, non-stationary\(^2\), and multidirection [22]. These characteristics make it difficult to describe the vibration with single objective parameter, often more parameters are needed for such complex vibration. On the other hand, multiple sensory systems are involved in the perception of seat vibration, including vision, vestibular, somatic, and auditory systems. The output of these systems are combined to form the judgment regarding the vibration [46]. This psychological process is largely unknown, and the existing researches investigating multiple sensations have already shown interaction between among them [30, 69, 54]. The complexity of human sensations and seat vibration made it

\(^1\) further explained in Section 5.3.
\(^2\) including transient vibration, such as impacts and shocks.
difficult to find the correlation between them. The present study planned to develop the simulator as the research tool for finding the correlation. As further explained in Section 2.5, the simulator provide good control over the vibration stimuli and the environment, which allows the parameters and sensations to be isolated and studied one pair at the time so the correlation can be found easily.

Another challenge is that experiments regarding human perception of vibration often need to deal with large difference among individuals. The variation are in both physical properties (such as height and weight) and psychological properties (such as sensitivity and knowledge regarding vibration), which make it difficult to acquire consistent result from experiments, especially with limited number of samples [4, 1]. Conducting experiments with simulator is more efficient than field tests with real vehicles, therefore the experiments can have more subjects to compensate for the individual differences.

### 1.3 Objective

This research aims to develop a research tool for ride comfort evaluation focusing on two objectives:

- to develop a cost-effective simulator that replicates vehicle seat vibration for the investigating human perception.
- to demonstrate the application of the developed simulator in the measurement of difference threshold with the modified A-AFC method.

### 1.4 Scope

The seat vibration in real vehicles is non-stationary as it often contains transient behavior like impacts and shocks. It can be considered as an combination of stationary and non-stationary content. The stationary content is the stationary vibration that is consistent and does not change over time (assuming driving conditions stay the same), such as the vibration cause by road surface texture or engine vibration. The non-stationary content is the transient vibration that occurs in the form of impacts and shocks, such as the vibration caused by driving over potholes and speed bumps. The discomfort caused by the vibration is dependent on both the stationary and non-stationary content. So far, no previous study has found how much does each of them contribute to the discomfort.

The standard methods do not separate the stationary and non-stationary content. Instead, they provided two averaging methods for vibration with and without shocks. The Root Mean Square (RMS) method was suggested to be applied to vibration without shocks, and the fourth power Vibration Dose Value (VDV) method was suggested when shocks are present. RMS is calculated with Equation 1.1, and VDV is calculated with Equation 1.2, in both equations $T$ is the duration of the measurement, and $a_w$ is the frequency weighted acceleration.

\[
RMS = \left\{ \int_0^T [a_w(t)]^2 dt \right\}^{1/2} \tag{1.1}
\]
\[ VDV = \{ \int_0^T [a_w(t)]^4 dt \}^{\frac{1}{4}} \]  

(1.2)

Apparently, VDV puts more weight on the peak acceleration values than RMS. However, so far no study has validated that balance of stationary and non-stationary content in VDV is consistent with actual human perception. It is very likely that such balance is dependent on many characteristics of vibration, such as the amplitude and duration of both stationary and non-stationary content [1]. Existing studies regarding human perception of non-stationary vibration are rather rare, comparing to the studies of stationary vibration. Most of the studies in this field used stationary stimuli, including the studies that the standards were based on. It was suggested that the stationary and non-stationary content should be evaluated separately, as they require different evaluation methods [73].

The scope of present study is limited on stationary vibration, while the investigation regarding non-stationary vibration is suggested for the future works.

1.5 Structure of thesis

The present study started with a review of the existing researches regarding human perception of vibration in Chapter 2, in which the current understanding and measurement methods were introduced. The design process is introduced in Chapter 3. The control system and strategy was developed and validated in Chapter 4. The experimental method of measuring difference threshold was proposed in Chapter 5 as an example of application. The findings of present study and proposal for future work were given in Chapter 6.
Chapter 2

Human perception of seat vibration

In this chapter, existing studies were reviewed to provide guideline for designing the research tool regarding human perception of seat vibration. The review starts with the effect of exposure to seat vibration (Section 2.1) and the human sensory systems involved in perceiving seat vibration (Section 2.2). Then the methods for modeling human perception are reviewed in Section 2.3. The experimental methods for measuring human perception are introduced in Section 2.4, which are the main application of the research tool developed in the present study. Finally, the advantage of using simulators is introduced in Section 2.5.

2.1 Effect from exposure to seat vibration

Exposing to seat vibration causes whole-body vibration [23], which is defined as a motion transmitted to the human body as a whole through supporting surfaces, as opposed to vibration directed more locally, such as hand-arm vibration. Most standards and researches have shown 2 types of effect of whole-body vibration: discomfort and health risks. The present study focus on discomfort as the seat vibration in modern vehicles, especially passenger vehicles, are not strong enough to raise health risks [22].

Discomfort is a highly subjective and difficult to be measured. Most studies asked the subjects to indicate the discomfort level with numbers (i.e. ratings). These ratings can be used together with measurement during the experiments to find the relation between the physical stimuli and sensation. Some studies also asked the subjects to describe their sensation in words. For example, the discomfort caused by vibration in working environment were described as fatigue, lowered concentration and work performance, drowsiness and irritation [79]. For passengers of the vehicle, the discomfort was found involving inability to fall asleep, and difficulties for reading and writing [60]. There are studies tried to measure discomfort through physiological parameters such as heart rate [32]. Although some correlations were found in these studies, they were not able to provide a reliable measuring method.

2.2 Human sensory systems

Human use multiple sensory systems to perceive vibration. The sensory systems involved are: vision, vestibular, somatic, and auditory systems. Each sensory system has its own operating frequency range. The eyes can easily detect vibration
with large displacement, which normally occurs in relatively low frequencies. The vestibular system of the inner ear senses linear and rotational acceleration from 0.1 to 10 Hz. The somatic system has multiple types of sensors. They sense the vibration through abdomen, receptors in skin, and motion between joints and muscles. Some receptors underneath the skin can detect vibration up to 500 Hz. The auditory system can detect vibration within the audible frequency range (20 - 20,000 Hz) [46].

The frequency ranges of these sensory systems have some overlaps, therefore the feeling of discomfort is formed by combining the output of multiple systems. For example, the interaction between somatic and auditory systems is one of the most studied area. Some studies have shown the presence of noise will affect the perception of vibration [4, 30]. The overall combination process of all sensory systems is not fully understood yet.

2.3 Modeling human perception of vibration

Human perception of vibration is a complicated process. The discomfort sensation caused by the vibration depends not only on the characteristics of vibration, but also on the characteristics of environment and human [26, 22, 23]. Most studies regarding human perception of vibration in laboratory conditions in order to minimize the effect of environment (further discussed in subsection 2.5).

Despite the subjective nature of discomfort, many studies tried to use indicators to quantify discomfort in order to correlate it with the physical stimuli. An indicator is a single value representing the discomfort level perceived by the subject. Ideally it should be calculated directly from the measurement without any subjective evaluation, thus enables comparing discomfort caused by different vibrations, or predicting the discomfort from measurements or simulations [22].

Psychophysics studies revealed a general relationship between physical stimuli and the sensations and perceptions they produce. Stevens’s power law is the psychophysics model commonly used in vibration related studies [22], in which the relation between the sensation $\psi$ (i.e. the discomfort due to vibration) and the physical magnitude or intensity $\varphi$ (i.e. the vibration magnitude) is assumed to be:

$$\psi = k\varphi^n$$  \hspace{1cm} (2.1)

where $k$ and $n$ represent the human sensitivity to the vibration. The coefficient $k$ is the proportionality constant that depends on the units used, and the exponent $n$ describes the rate of change of sensation $\psi$ with vibration magnitude $\varphi$ [72].

Many studies suggested that the perceived discomfort $\psi$ depends not only on the physical magnitude or intensity $\varphi$, but also on other physical properties of vibration stimuli such as frequency and direction. For example, human was found to be most sensitive (causing most discomfort) to whole-body vertical vibration at about 5 Hz, and less sensitive to vibration with higher frequencies [48]. Therefore, these physical properties also need to be taken into consideration when evaluating seat vibration.

There are 2 types of methods often used to quantify the influence of physical properties on perception: equivalent comfort contours and sensory thresholds.
2.3.1 Equivalent comfort contours

The equivalent comfort contours consist of a series of stimuli that are different from the reference stimulus but cause the same level of discomfort [23]. They are often used in studies to show how does human sensitivity changes together with multiple physical parameters of the stimuli. Early studies regarding the equivalent comfort revealed the relation between human perception and the frequency, direction, and location of the vibration stimuli [26, 27, 64, 65, 25]. The resulted equivalent comfort contours in these studies later became the frequency weightings and multiplying factors in BS6841 [7], then they were adopted in the ISO2631 [33].

2.3.2 Sensory thresholds

The sensory thresholds are another method to represent the sensitivity of human sensory systems. The absolute threshold is often used to describe the the lowest stimulus level at which a stimulus can be detected, while the differential threshold (also known as Just Noticeable Difference, JND) is the minimum change in stimulus level needed in order for a difference to be noticeable [20, 9]. These thresholds also provide guidance for evaluating the effect of vibration. For example, the absolute threshold can be used to predict whether the vibration can be perceived by the user, while the differential threshold can be used to predict whether an improvement or deterioration can be noticed by the user. The absolute threshold was also investigated in the aforementioned early studies and later was adopted in BS6841 [7] and ISO2631 [33] as well. The absolute threshold was found to have generally the same trend as the equivalent comfort contours [26, 27, 64, 65, 25]. The difference threshold was less often used for evaluation purposes it was not very well understood yet, most of the existing studies focused on measuring the difference threshold [4, 47, 55].

2.4 Measuring human perception of vibration

Both equivalent comfort contours and sensory thresholds are often measured in laboratory with vibration simulators. This section introduces the experimental methodologies for these measurements. For measuring the equivalent comfort contours, the subjects are asked to give ratings indicating the discomfort they felt. On the other hand, subjects are often asked to choose between a few stimuli when measuring the sensory threshold, such as choosing the stimulus that feels strongest or the stimulus that can be perceived.

2.4.1 Method for measuring equivalent comfort contours

The magnitude estimation method is widely used for measuring equivalent comfort contours [72]. For each group of stimuli, a constant reference stimulus is often used to improve the consistency of ratings. In the experiment, the subjects are asked to assign a number representing the discomfort of the test motion relative to the discomfort of the reference motion, assuming the discomfort of the reference motion
corresponded to a certain fixed value [23, 1, 30]. Then the equivalent comfort contours can be plotted by connecting the points representing the stimuli that received the same rating.

### 2.4.2 Method for measuring sensory threshold

The sensory threshold can be measured with the following two methods:

1. Constant Stimuli.

During the experiment, the subjects are asked to indicate which stimuli can be detected (for absolute threshold) or which stimulus feels more uncomfortable/stronger (for differential threshold) during each trial. Both of these methods are based on the concept of psychometric function, models the relationship between a given feature of a physical stimulus (such as the amplitude or frequency of vibration), and the correct rate of forced-choice responses of a human test subject. The thresholds are conventionally defined as the point on the psychometric function that corresponds to the correct rate of 50%.

The method of constant stimuli is a non-sequential procedure, in which various stimulus levels are present to the subject in random order. The range of the varying parameter, such as the amplitude of vibration, is chosen before conducting the experiment and remain the same for all subjects [20] based on experience or previous study. This method needs to measure the full range of psychometric function, so the range of stimulus levels must cover the corresponding correct rate from 0% to 100%. Normally it takes at least 10 points to get a good representation of the psychometric function if the range is set properly. This method often takes a huge amount of time, which could be a serious problem as the human sensitivity might change over time during the experiment.

The method of A-AFC differs from the method of constant stimuli in that the A-AFC method focuses on the threshold point instead of the entire psychometric function [43]. The adaptive procedure determines the level (or other parameters of interest) of each stimulus presented to the subject depends on the response of the subject to the previous stimuli [4], as shown in Figure 2.1. The advantage of A-AFC method is that it only need to measure one single point on the psychometric function, which could save a lot of time. Also A-AFC method does not require prior knowledge to choose the range of the varying parameter since it is determined by the adaptive procedure.

### 2.5 Experiment environment

The aforementioned studies regarding human perception of seat vibration conducted at least part of their experiment in laboratory condition, i.e. the subjects were exposed to vibration stimuli that were created artificially on simulators instead of real vehicles. Field tests with real vehicles were conducted much less often, and only for the purpose of validating the findings of laboratory experiment, such as in [50, 61].
2.5. Experiment environment

Figure 2.1: A schematic representation of the A-AFC method

The main advantage of simulators is that they provide high controllability of the vibration stimuli and the environmental factors, as further introduced in the following sections.

2.5.1 Controllability of the stimuli

Using simulators provides full control over the stimuli applied to the test subject. Any arbitrary vibration can be produced as long as it is within the operating range, regardless of being simple sinusoidal vibration, broadband random vibration, or recordings from real vehicles. On the other hand, the vibration in real vehicles largely depends on the road profile and vehicle properties. Neither of them can be easily manipulated in order to achieve a certain acceleration waveform or spectrum [29, 23], which is often needed for the experiments introduced in Section 2.4.

Using simulators could often save a lot of time comparing to field test with real vehicles. Therefore larger number of subjects can participate in the experiment, which better eliminates the effect of individual differences among test subjects and yields more consistent result. The high efficiency of simulators comes from their capability of rapidly switching between different stimuli, which can be done by simply send another input signal. Switching between different stimuli is very common in the experiments involving comparing 2 or more stimuli (one of them might be the reference), which would be difficult in field test.

Also, it is easier to conduct “double blind test” with simulators, in which neither the subject nor the experiment operator know the stimuli in advance. The common practice for achieving “double blind” is to present the stimuli in random order, and use computer program to control the experiment, thus eliminate the interference of psychological factors such as personal preference and expectation of vehicles [4]. On the other hand, it would be difficult in field tests to keep the subjects from recognizing the vehicles, especially if the vehicles are not identical.
2.5.2 Controllability of the environment

Environmental factors could affect the experiment in two ways. They either directly act on the test subjects to cause changes in their perception, or they affect the equipment to cause changes in the stimuli.

Environmental factors can be easily controlled when using simulators in laboratory condition. The common practice is to eliminate them or keep them constant through the experiment so that the subject’s perception of vibration will not be affect. For example, it has been proved by many studies that the noise will affect the perception of vibration [30, 69, 42, 3]. Most of the simulators were designed to minimize its operating noise. Although they still make audible noise sometimes, the noise levels are generally much lower than the interior noise of vehicles so that it can be easily eliminated by wearing ear plugs or masked by wearing headphones playing white noise [4, 52].

There are some environmental factors that cannot be controlled, such as weather (temperature), and wear and degradation of simulator/vehicle. These factors might affect the experiment indirectly by changing the dynamics characteristics of the simulator/vehicle thus distorted the stimuli. The influence of these factors can be mitigated by implementing real-time close-loop control strategy to ensure the accurate replication of vibration thus providing high repeatability of the experiment [13, 75], which is not possible on real vehicles.

2.6 Summary

This chapter reviewed existing studies of human perception. The main effect of exposure to seat vibration was found to be discomfort. Multiple human sensory systems were involved in the perception of seat vibration, therefore a well controlled experiment environment is needed for investigating human perception of seat vibration. The modeling and measuring methods of human perception were reviewed to provide guidelines for the design of research tool. The experiment environment of existing studies were reviewed by comparing the laboratory and field experiments, and the simulators were found to be the best research tool because they provide better control of the stimuli and environment.
Chapter 3

Design

The design process of the seat vibration simulator is introduced in this chapter. The design process started with reviewing proving ground measurements (Section 3.1) to find the general characteristics of seat vibration in passenger vehicles. The design requirements were defined based on these characteristics and requirements from Toyota Motor Europe as introduced in Section 3.2. With the design requirements clarified, the existing design of simulators in previous studies were reviewed in Section 3.3. The mechanical design choices of the simulator were explained in detail in Section 3.4. The selection and installation of accelerometers were explained in Section 3.5.

3.1 Review of proving ground measurements

The main function of the simulator is replicating vehicle seat vibration in real-world driving conditions. Therefore the general characteristics of seat vibration in passenger vehicles must be well understood before setting the design requirements. These characteristics are found by reviewing proving ground measurements. The measurements include multiple tracks that represented the typical road surfaces in EU. The data contains 3-axes acceleration measurements at seat cushion, seat back, and seat rail. For each track, the measurement was made at various speed. Six different passenger vehicle models within the same market segment were measured. Their performances in ride comfort are considered comparable.

The raw data files were in the format of the data acquisition system used at the proving ground. A program was developed to import the data files into MATLAB environment and indexed them based on the track, vehicle model, speed, and measurement location (channel). Then the files can be selected based on these indices. The Power Spectral Density (PSD) of the selected acceleration measurement were calculated with Welch’s method [78] with frequency resolution of 0.5Hz. An example of such PSD is shown in Figure 3.1. These PSD reveal the frequency characteristic of vehicle seat vibration.

The acceleration of seat rail was measured at all 4 mounting points of the seat in longitudinal, lateral, and vertical directions. The seat rail is mounted on the vehicle floor and moves with 6 degrees of freedom (DOF). A rigid body transformation was conducted to reconstruct the 6 DOF motion of the seat at the center of 4 mounting points. The rigid body transformation is further explained in Section 4.3.
Previous studies have pointed out that the main contributing factor of discomfort is the vibration at the seat cushion and seat back [22]. The review of the proving ground data focused on these two locations, plus the measurement at seat rail as additional reference. Frequency weightings were applied to the acceleration measurement according to ISO2631 [33]. The resulted weighted RMS values represent the contribution of discomfort of vibration in each direction and location, and can be used to make rough comparisons.

The following observation was made from reviewing the data.

1. at seat rail, the vibration in vertical direction has the highest amplitude among the 3 translational directions. The vibration in longitudinal direction is lower than the vertical direction. The lateral direction direction has the least amplitude. According to ISO2631, the contribution of lateral direction was negligible in most cases.

2. the PSD curves of acceleration in vertical direction have shown high similarities among different vehicle models, while the curves of longitudinal and lateral directions have more variations.

3. The vehicle seats effectively filtered the vibration in vertical direction above 30Hz, and the PSD curves approaches zero above 80Hz. This observation is consistent with the previous studies regarding Seat Effective Amplitude Transmissibility (SEAT) [77, 80]. The vibration in longitudinal direction is also mostly under 30Hz but have slightly more high frequency content above 30Hz.

### 3.2 Design requirements

The design requirements of this project was formulated based on the characteristics of vehicle seat vibration found in the previous section, and additional requirements from Toyota Motor Europe. The design requirements include operating frequency range, excitation directions, budget, and space.

The lower limit of the operating frequency range was defined by the requirement from Toyota Motor Europe. They already have access to a hexapod motion platform prior to this project. The hexapod is capable of generating 6 DOF motion with the frequency range of 0-10Hz, which could not cover the full range of seat vibration. Therefore, a new research tool was needed for investigating the seat vibration above
3.3 Review of existing seat vibration simulators

10Hz. As discussed in further details in Section 3.4, giving up the low frequency capability below 10Hz would allow the usage of more compact actuators with less stroke and force capabilities. This could bring the cost down for two order of magnitudes. And the simulator can be placed within the existing building thus eliminate the cost of new construction. Therefore, the lower limit of the operating frequency range was set to be 10Hz.

The higher limit of the frequency range related to ride comfort is 80Hz according to ISO2631. The observations in the previous section were consistent with the standard. Based on these information, the higher limit of the operating frequency range was set to be 80Hz.

The motion of vehicle seat has 6 DOF. Previous studies have found that the vibration in these 6 directions have different contribution to ride comfort. The effect of the 3 rotational directions is often negligible, especially in the frequency range of 10-80Hz [50]. In addition, the contribution of the lateral direction is negligible, as explained in Section 3.1. Based on these reasons, it was concluded that having two directions of excitation would provide the best balance between functionality and complexity. The frequency ranges of vertical and longitudinal directions were chosen based on the observation of proving ground data: 10-30Hz for the vertical direction and 10-80Hz for the horizontal direction.

This project was a pilot project that aimed to provide information and experience regarding the simulation of seat vibration, which may contribute to future investment in more advanced seat vibration simulators. The goal of this project is to provide a cost effective solution for simulating seat vibration. The simulator also needs to be operated in a designated room at Toyota Motor Europe. The entire setup (including the desk for the operator) should fit into the room and without interfering with existing equipment.

3.3 Review of existing seat vibration simulators

With the design requirements clarified, the seat vibration simulators used in previous studies was reviewed to provide understanding of the existing solutions. Different types of simulators were compared and analyzed in this section.

Seat vibration simulator generally consist of two parts: a vibration platform and a seat. The vibration platform provide the desired excitation to the seat, while the seat transmit the excitation to test subject. Accelerometers were installed on the vibration platform and/or at the contact surface between the seat and human body for measurement and control.

3.3.1 Seat

The seat support the weight of the test subject and transmit the vibration generated by the vibration platform to the subject. There were two types of seat often used on seat vibration simulators: rigid seat and vehicle seat. Rigid seat were commonly used in the early studies regarding human perception of whole-body vibration [26, 27, 64, 65, 25]. The results of these studies were later adopted by ISO2631 [33] and BS6841 [7]. Some recent studies also use rigid seats for measurement of perception
Chapter 3. Design

threshold [4] and equivalent comfort contour [57]. The rigid seats were made out of wood and/or metal, and were designed to have no resonance within the operating frequency range of the simulator. These studies aimed to find the universal principle of human perception of whole-body vibration, which should hold true for all kinds of seats. And the rigid seat was the easiest solution to implement because using it would avoid the nonlinear dynamics of the vehicle seat thus reduce the complexity of control. It also simplified the measurement of acceleration as the accelerometers did not have to be placed at the contact surface between human body and the seat, which requires special kind of accelerometers [23]. In addition, using the rigid seat could reduce the affect of individual differences among test subjects and yields more consistent result, because the dynamics of vehicles seats are dependent on the body properties of the occupant such as weight and body sizes [45, 44, 77].

Many recent studies used vehicle seat to create more realistic simulation of seat vibration in real vehicles. The studies regarding the dynamics of vehicle seats revealed that the seat plays an important role in the transmission of vibration from floor to human body. Vehicle seat has its own dynamics that affect the vibration transmitted from vehicle to human body [44, 45]. For example, the seat could significantly attenuate the vertical vibration of the floor at high frequency, so that the vertical seat acceleration is predominantly at frequencies below about 20 Hz [24]. The physical properties of the seat, such as the stiffness and damping could also affect the discomfort when the vibration at the contact surface remain the same [28, 31, 63]. The seat effective amplitude transmissibility (SEAT) was proposed as an indicator of dynamic seat comfort performance [23], and it has been validated in multiple studies[59, 77, 80, 10]. In addition to the dynamics, the sitting posture of vehicle seats could also affect the discomfort caused by vibration. Studies have shown the seating posture could affect the transmissibility of vibration from seat to human body, thus affect the perceived discomfort [59, 39, 49, 58, 8].

3.3.2 Vibration platform

The vibration platform support the weight of the seat and apply excitation to the seat. It is consist of 3 major parts: 1) a rigid platform that holds the payload; 2) a suspension system that supports the weight of platform and allows it to have one or multiple degrees of freedom; 3) one or multiple actuators providing the driving force to the platform. The existing vibration platforms can be divided into 3 categories based on their suspension systems.

The first type of vibration platform has the suspension system that allows only one degree of freedom. The first simulator introduced in [4] was an example of this type of vibration platform. The schematic of this simulator is shown in Figure 3.2. The suspension system consists of air springs to support the weight of the platform, and linear guides (steel rollers) as constrains that allows motion only in the vertical direction. This type of simulator were widely used in studies investigating human perception of vibration in single direction [1, 38, 30]. Some actuators (shakers) can also function as a suspension system with a single degree of freedom, such as the actuators used in [27] (shown in Figure 3.3a) and [36] (shown in Figure 3.4). These two studies also introduced variations of single degree of freedom vibration platforms.

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1 The first simulator introduced in this study.
2 Compared to the aforementioned rigid seats, which often required the subject to sit in an upright posture.
that combined 2 suspension systems and actuators to provide an additional degree of freedom. For example, the simulator used by [27] (Figure 3.3b) added a sliding table on top of the vertical vibration table shown in Figure 3.3a and another actuator acting in horizontal direction, so that it can produce vibration in both vertical and horizontal direction. The simulator in [36] added another vibration platform to simulate the floor vibration, as shown in Figure 3.4.

The second type of vibration platforms can fully replicate the 6 DOF rigid body motion of the seat. Some of them arranged 6 actuators into 3 pairs to form a hexapod, such as the Multi-Axial Simulation Table from MTS Systems Corporation and eM6-400-1500 motion system from E2M Technologies B.V. (Figure 3.5). Simulators based on hexapod style vibration platforms were used in [51, 50]. Others simulators mounted the actuators in orthogonal directions, such as the MINI MAST system from MTS Systems Corporation (Figure 3.6b) and the Maintis system (Figure 3.6a). Simulator based on orthogonal actuators were used in [35, 71]. The Cube system from Team Corporation (Figure 3.7a) also has actuators mounted in orthogonal directions, but it has a more compact design than MINI MAST and Maintis, as the 6 actuators were installed in a cubic case, hence the name Cube. It was initially developed for Ford Vehicle Vibration Simulator (Figure 3.7b) in 1997 [52]. On Ford Vehicle
Chapter 3. Design

Figure 3.4: The simulator from [36]. Two actuators with integrated suspension system were used for the seat and floor. Vibration Simulator, the seat was mounted on the top side of Cube, while the floor was mounted on the front side with an additional single degree of freedom actuator. The steering wheel and dash board was mounted on another vibration platform with 4 degrees of freedom and independent from the Cube. This simulator was capable of replicating seat, steering wheel, and floor vibration at once [52]. It is a very versatile simulator and was used in the studies regarding virtual evaluation of ride comfort [53], vehicle seat dynamics [77], idle vibration [6], and human perception of vibration [2].

Figure 3.5: Hexapod vibration platform

The third type is floating vibration platform which do not have any constrains of its motion. Instead, the platform is placed on a flexible pad to isolate it from the ground and allows it to move freely in all 6 directions with minimal resistance. The second simulator introduced in [4] gave an example of this type of vibration platform. The schematic of this simulator is shown in Figure 3.2. The flexible pad is made of polyurethane foam (represented as mass-spring system in Figure 3.2) with specifically selected stiffness and damping character, so that the resonance frequencies of the platform in all 3 transnational directions are below the operating frequency range of the platform. The actuators of this simulator were a special kind of electrodynamic shakers. Their main difference to the actuators of first and second types of platforms is that they were installed directly on the platform, instead between the platform and the base. These shakers were also called inertia shakers because their force output come from the reaction force of the electromagnetic force generated by the coil.
3.4 Mechanical design

As introduced in Section 3.3, a seat vibration simulator consists of two parts: vibration platform and seat. The design process of these two parts is explained in this section. The design choice regarding the seat was made first, as the vibration platform needed to be designed around the seat. In the present study, a front seat from a Toyota passenger vehicle was selected. As explained in Section 3.3.1, using vehicle seat provides more realistic simulation of the discomfort caused by seat vibration.

There were 3 types of vibration platforms used in previous studies as they were reviewed in Section 3.3.2. The third type of the vibration platforms was concluded to be the most cost-effective solution for the design requirements defined in Section 3.2. Especially the required operating frequency range fits the characteristic of the electrodynamic actuators very well. The first type of vibration platform has a suspension system with single DOF, which does not meet the design requirement of having two directions of excitation. The requirement can be achieved by combining two suspension systems, but the additional suspension system will add additional weight and complexity to the system, which is contradictory to the goal of developing a cost-effective simulator. The second type of vibration platform can apply...
excitation in all 6 DOF and some of the commercial models can cover the full frequency range of ride comfort (start at 0Hz). However, such performances exceeded the design requirement thus will result in over-investing. For example, the cost of the Cube system (Figure 3.7) is over 700,000 euro. As explained in Section 3.2, the abilities of operating at frequency below 10Hz and applying excitation in all 6 DOF were already achieved by the hexapod motion platform. The second type simulators are also relatively bulky comparing to the third type so that they could not fit into the designated room at Toyota Motor Europe.

The actuators of the present design are compact electrodynamic shakers, also known as tactile transducers, which have simpler structure than the typical electrodynamic shakers. The main difference between them is that the inner pole of the compact type is made by permanent magnet and is fully enclosed by the case, as shown in Figure 3.9, while the inner pole (piston) of typical electrodynamic shaker has its own coil and is directly connected to the payload. These differences made the compact electrodynamic shakers easier to manufacture, thus have much lower cost comparing to the typical ones. Also due to these differences, the compact type is mounted directly to the vibration platform since the excitation force is applied through the case of the shaker, whereas the typical electrodynamic shakers are often mounted between the vibration platform and a stationary base (often the ground).

![A sectioned view of a typical electrodynamic shaker](image1)

![The coil and inner pole of the compact electrodynamic shaker](image2)
3.4. Mechanical design

There are two models of electrodynamics shakers selected for the present design, as shown in Figure 3.10. They were selected mainly based on their frequency range specifications. The Q10B shaker is low frequency oriented (5-40Hz) and has higher force rating, therefore it was used in the vertical direction. The LFE shaker has wider frequency range (5-400Hz), so it was used in the longitudinal direction.

![Shaker Images](A) The Q10B shaker from Earthquake Sound (B) The LFE shaker from Guitammer

**FIGURE 3.10:** The electrodynamics shakers used in the present design

After the actuators were selected, the next step in the design process is packaging, in which the positions of every main component were selected. The overall layout of is similar to the the simulator introduced in [4] (Figure 3.8). The seat is located on the top. A rigid frame is used as the chassis, located at the bottom. The isolation layer is placed between the chassis and ground, so that the vibration of the chassis is isolated from the ground.

Similar to the design in [4], the actuators are mounted on the chassis. Further consideration was taken regarding the placement of actuators in order to achieve the optimal design. The excitation forces need to be symmetrical around the center of mass in order to minimize the rotational torque caused by uneven excitation forces. Therefore the actuators need to be placed either on the center line, or in pairs around it. After evaluating multiple possible combinations, the best system layout was found to be having 2 actuators at each mounting point of the seat, one in vertical direction and the other one in longitudinal direction. This layout enables the modular design of the bracket and actuator assembly, which consist of a bracket and 2 actuators, as shown in Figure 3.11 and 3.12. The modular design reduced the number of different components and the cost of manufacturing. All of the components shown in Figure 3.11, except the fans and actuators, were designed specifically for the simulator. The drawings of these components can be found in Appendix A.

The position of the bracket can be adjusted in both longitudinal and lateral directions in order to adapt to different seats. The bracket and the chassis were made of aluminium alloy to keep the weight of the simulator as light as possible.

An air mattress was put underneath the rigid frame. Comparing to the PU foam used in the previous studies [4], the air mattress is more flexible thus allows larger amplitude of vibration. In addition, the pressure of the air mattress can be easily controlled so that the stiffness can be tuned accurately. A pressure sensor was installed at the valve of the air mattress to monitor the pressure, as shown in Figure 3.14. An overview of the simulator is shown in Figure 3.13.
3.5 Measurement of vibration

The vibration generated by the actuators needs to be measured. Similar to the proving ground measurements reviewed in Section 3.1, the acceleration needs to be measured at seat rail, seat cushion, and seat back.

The accelerometers were selected based on their number of axes, dynamic range, frequency range, and compatibility with the data acquisition device. The accelerometers should ideally be able to measure translational acceleration in all 3 directions, otherwise multiple accelerometers might need to be installed at the same location, which is often more expensive. The dynamic range of was set to $\pm 2g$ because the proving ground data measurements have $\pm 2g$ dynamic range. The frequency range of accelerometers should cover the operating frequency range of the simulator (10-80Hz). The output voltage of the accelerometer should be compatible with the data acquisition device, which accept voltage signals with variable range from $\pm 1V$ to $\pm 10V$. The model 4030 accelerometer from TE Connectivity was found to meet all of these requirements with a budget-friendly price. It is a 3-axes MEMS type DC accelerometer with dynamic range of $\pm 2g$ and frequency range of 0-200Hz. It has build-in amplifier with output signal at 0.5-4.5V.

The mounting positions of accelerometers were planned out during the design phase.
The seat rail connector (shown in Figure 3.11) was designed according to the dimension of the accelerometer so that it has a large enough surface for attaching the accelerometer. The accelerometers were attached with petro wax, then secured with zip ties, as shown in Figure 3.15. For the seat cushion and seat back, the accelerometers were installed beneath the fabric of the seat to avoid causing additional discomfort to the test subject. Two indentations were made on the foam at the corresponding locations to keep the accelerometers from protruding out of the seat surface.

3.6 Summary

This chapter presented the design process of seat vibration simulator. The design requirements were defined based on the findings of reviewing proving ground measurements and the other requirements from Toyota Motor Europe. It concluded that the operating frequency range was 10-80Hz with excitation applied in both longitudinal and vertical directions. A compact and cost-effective design of seat vibration simulator was proposed featuring a modular floating vibration platform and compact electrodynamics actuators mounted in longitudinal and vertical directions.
Chapter 4

Control

An important challenge in vibration-related experiment is to calculate the proper input signals for the actuators in order to get accurate replication of the desired vibration. The simulator often have complicate nonlinear dynamics that are at least partly unknown, which causes large distortions between the output and the target. This issue is further complicated on the simulators that have multiple degrees of freedom, as the interaction between actuators in different directions is often inevitable [13]. The control system of the simulator needs to compensate such distortions by correcting the input signal so that the output can track the target with minimum error. Current practice in the industry to solve the multi-variable tracking problem on vibration simulators is to calculate the input signal with an iterative, off-line algorithm, which is often referred to as Time Waveform Replication (TWR) process. The TWR process is based on a linear model of the system. Multiple open-loop iterations are performed to compensate for the modeling error [13].

The TWR process was introduced in Section 4.1. Then a control system was built to implement the TWR process on the simulator (Section 4.2). Section 4.3 introduced the signal processing method, which prepares the measurements for the TWR process. The implementation of the TWR process was introduced in Section 4.4. The performance of the control system was tested in Section 4.5.

4.1 Review of control strategy of vibration simulators

Dodds [18] was the first to propose solving the multi-variable tracking problem of vibration simulators in the frequency domain. His approach, later known as the TWR process, is widely accepted and used in industry for replicating vibration on simulators [17]. The TWR approach consists of two phases as shown in Figure 4.1: a system identification phase followed by the target simulation phase. The system identification phase gives the Frequency Response Function (FRF) model of the system and its inversion. The target simulation phase correct the remaining error in the output by conservatively correct the input in the frequency domain in an iterative way [13].

The system identification phase models the system with a linear non-parametric frequency domain model: the FRF model [5]. Then the FRF model is inverted at each frequency. Some studies proposed using linear time domain model, such as [67, 37, 12], but did not find significant improvement over the FRF model.

The linear models used in TWR process could not fully capture the nonlinear behavior of the system, thus resulted in modeling error. Larger modeling error takes more
iterations at the target simulation phase to correct. To minimize the modeling error, the excitation level during the system identification phase should be as close as possible to the target excitation level [66]. However, a good model is needed in the first place in order to get the correct excitation level. The solution to this “Chicken or the egg” problem is to do the identification in an iterative way, start with white noise or pink noise as input signal, and use the resulted model to calculate the new input signal [13].

The iteration of the target simulation phase consists of two steps [13]:

1. calculating a new input signal with Equation 4.1.

2. sent the input signal to the simulator to measure the output. Check the error between the output and target for convergence.

Equation 4.1 shows the calculation of the new input signal, in which \( u, y, r, \) and \( \hat{G} \) are in frequency domain. \( u \) is the input signal, \( y \) is the output signal, \( \hat{G}^{-1} \) is the inverted FRF model, \( r \) is the reference signal. The superscript \( j \) refers to the iteration number \( (j = 1, 2, 3, ...) \). The iteration gain matrix \( Q_j \in \mathbb{R}^{n \times n} \) is diagonal with all of its diagonal elements \( q_j \) satisfy \( 0 \leq q_j \leq 1 \). Initially \( u^0 \) and \( y^0 \) are zero vectors of appropriate sizes. The reason to perform this calculation in the frequency domain is that the FRF model \( \hat{G} \) is in the frequency domain. Discrete Fourier Transformation (DFT) is used to make the necessary transitions from time to frequency domain and vice versa [13].

\[
u^j = u^{j-1} + \hat{G}^{-1}Q_j(r - y^{j-1})
\]  \hspace{1cm} (4.1)

An important advantage of the off-line TWR process is that it allows controlling linear plants which are non-minimum phase or linear plants with a delay with arbitrary accuracy (ignoring noise of measurement), because it can compensate completely for delays or non-minimum phase zeros [15]. While the online feed forward approach requires approximating the inverse of a non-minimum phase model with methods such as “zero phase error tracking control algorithm” [76], or “stable dynamic inversion” [12].

![Figure 4.1: A schematic representation of the TWR process [13]](image-url)
Another advantage of TWR process is its simplicity. It was incorporated into various commercial software and requires only some basic signal processing background to conduct the process [13].

The main limitation of TWR process is that the iterations takes a lot of time. Depending on the properties of payload and actuators, it usually takes 10 to 30 iterations to achieve the target acceleration, which needs to be repeated for every input. The fact that the calculation of Equation 4.1 has to be done in the frequency domain make it not possible to implement correction to input signal in real-time. The current measurement $y^i$ needs to be fully acquired and transformed to the frequency domain before the next input $u^{i+1}$ can be calculated. So the second step needs to be completed before new input can be generated, hence the feedback is off-line [13].

### 4.2 Control system layout

A computer with analog input and output interface was needed to implement the TWR process. The overall layout of the system is shown in Figure 4.2. The output interface is the Audio Interface (TASCUM US-16x08), which send analog audio signal to the power amplifiers to drive the actuators. The input interface is a data acquisition board (USB-1608GX-2AO), which acquire voltage signal from the accelerometers. The control algorithm was implemented in MATLAB environment as it provides easy access to the input and output devices.

![Figure 4.2: System layout](image)

### 4.3 Signal processing

The acquired voltage signal needs to be processed first to remove offset voltage and noise. The accelerometers has an offset voltage about 2.5V when no acceleration is applied. And the sensitivity (ratio between output voltage and acceleration) is about 1V/g. Individual calibrations of offset voltage and sensitivity of all accelerometers were provided by the manufacturer. The offset voltage is affected by gravity. The accelerometers at the seat rail are aligned with the ground reference frame. Their
vertical axes of measure approximately 1g when standing still, and other axes are also affected due to the small positioning errors. Therefore the real-world offset voltages are re-calibrated every time before running the simulator. The real-world offset voltages are measured by taking the average of 1 second of recording when the simulator is still (no excitation), which was found long enough to yield consistent results.

The accelerometers at the seat cushion and seat back are not aligned with the ground reference frame due to the shape of the seat. The angles between the axes of accelerometers and the reference frame were calculated with the real-world offset voltages and the individual calibrations. Then the measurements of these two accelerometers were transformed into the ground reference frame with rotation matrices.

A Butterworth low pass filter with cutoff frequency of 200Hz is applied to the voltage signals since the frequency range of the accelerometers are 0-200Hz. Then the offset voltages are subtracted from the signal and then converted into acceleration by dividing the sensitivity.

The acceleration measurement measured at the seat rail was taken by 4 accelerometers mounted near the 4 mounting points, which consist of 3 axes translational acceleration measurements at 4 locations, 12 channels of data in total. Preliminary result (Figure 4.3) have shown that the acceleration measurements were not exactly the same at all 4 locations. This mismatch indicated that the simulator was moving in both translational and rotational directions. Since the rotation cannot be directly measured by the accelerometers, it will cause error in the measurements of translational acceleration. The rigid body transformation was conducted to reconstruct the 6 DOF acceleration of the platform, so that the interference of rotational acceleration is minimized. Two rigid body transformation methods were implemented and compared.

\[
\vec{a}_B = \vec{a}_A + \vec{\omega} \times \vec{r}_{AB} + \vec{\omega} \times (\vec{\omega} \times \vec{r}_{AB})
\]

\( (4.2) \)
Here it was assumed that the simulator has very low angular velocities so the third term on the right hand side $\vec{\omega} \times (\vec{\omega} \times \vec{r}_{AB})$ is negligible. And the second term can be rewritten in matrix form, as show in Equation 4.3.

$$\dot{\vec{\omega}} \times \vec{r}_{AB} = -\left(\vec{r}_{AB} \times \dot{\vec{\omega}}\right) = \begin{bmatrix} 0 & r_{ABz} & -r_{ABy} \\ -r_{ABz} & 0 & r_{ABx} \\ r_{ABy} & -r_{ABx} & 0 \end{bmatrix} \begin{bmatrix} \dot{\omega}_x \\ \dot{\omega}_y \\ \dot{\omega}_z \end{bmatrix}$$ (4.3)

Thus Equation 4.2 can be rewritten in matrix form as Equation 4.4.

$$\begin{bmatrix} a_{Bx} \\ a_{By} \\ a_{Bz} \end{bmatrix} = \begin{bmatrix} a_{Ax} \\ a_{Ay} \\ a_{Az} \end{bmatrix} + \begin{bmatrix} 0 & r_{ABz} & -r_{ABy} \\ -r_{ABz} & 0 & r_{ABx} \\ r_{ABy} & -r_{ABx} & 0 \end{bmatrix} \begin{bmatrix} \dot{\omega}_x \\ \dot{\omega}_y \\ \dot{\omega}_z \end{bmatrix}$$ (4.4)

The 2 terms on the right hand can be combined into one term as shown in Equation 4.4, which can be seen as a transformation from the acceleration of point A into point B. The matrix on the right hand side is named transformation matrix $T$, as defined in Equation 4.6.

$$T = \begin{bmatrix} 1 & 0 & 0 & 0 & r_{ABz} & -r_{ABy} \\ 0 & 1 & 0 & -r_{ABz} & 0 & r_{ABx} \\ 0 & 0 & 1 & r_{ABy} & -r_{ABx} & 0 \end{bmatrix}$$ (4.6)

Each row of $T$ corresponds to an element of $a_B$, so that the transformation can be applied to measurements of other points by adding corresponding rows into $T$. An example is given in Equation 4.7.

$$\begin{bmatrix} a_{Bx} \\ a_{By} \\ a_{Bz} \\ a_{Cx} \\ a_{Cy} \\ a_{Cz} \\ \vdots \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & r_{ABz} & -r_{ABy} \\ 0 & 1 & 0 & -r_{ABz} & 0 & r_{ABx} \\ 0 & 0 & 1 & r_{ABy} & -r_{ABx} & 0 \\ 1 & 0 & 0 & 0 & r_{ACz} & -r_{ACy} \\ 0 & 1 & 0 & -r_{ACz} & 0 & r_{ACx} \\ 0 & 0 & 1 & r_{ACy} & -r_{ACx} & 0 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \end{bmatrix} \begin{bmatrix} a_{Ax} \\ a_{Ay} \\ a_{Az} \\ a_{Ax} \\ a_{Ay} \\ a_{Az} \end{bmatrix}$$ (4.7)

With enough measurement points, the vector on the right hand side of Equation 4.7, i.e. the translational acceleration at point A and the angular accelerations can be approximated by the dot product of pseudoinverse of transformation matrix $T$ and the vector containing measured accelerations, as shown in Equation 4.8. For this specific application, point A was selected to be the center of the 4 mounting points.
This method has made 2 approximations during the process:

1. $\vec{\omega} \times (\vec{\omega} \times \vec{r}_{AB})$ was assumed to be negligible in Equation 4.3.

2. the use of pseudoinverse was a least square approximation process to solve Equation 4.7. Therefore, a second method of reconstructing 6 DOF rigid body motion was used to check the accuracy of these approximations.

The second method [62] uses 9 accelerometers\(^1\) that are placed along the x, y, and z axes of the body-fixed frame, as shown in Figure 4.4. At point 0, the acceleration is measured at x, y, and z directions ($a_x^0$, $a_y^0$, $a_z^0$). At point 1, the acceleration is measured at x and z directions ($a_x^1$, $a_z^1$). At point 2, the acceleration is measured at y and z directions ($a_y^2$, $a_z^2$). At point 3, the acceleration is measured at x and y directions ($a_x^3$, $a_y^3$). Thus, 9 translational accelerations were measured in total. The front right accelerometer at the seat rail was used as point 0, the front left one was used as point 1, and the rear right one was used as point 2. An additional accelerometer was installed on the chassis right beneath the front right accelerometer to take measurement at point 3.

This specific configuration of accelerometers allows the angular accelerations $\dot{\omega}_x$, $\dot{\omega}_y$, and $\dot{\omega}_z$ to be calculated directly from the aforementioned 9 translational accelerations [62], as shown in Equation 4.9, in which $\rho_1$, $\rho_2$, $\rho_3$ are the distance between point 1, 2, 3 and 0, as shown in Figure 4.4. Then the angular velocities $\omega_x$, $\omega_y$, and $\omega_z$ can be calculated by integrating $\dot{\omega}_x$, $\dot{\omega}_y$, and $\dot{\omega}_z$, then substitute into Equation 4.2 to calculate the translational acceleration at the desired point.

\(^1\)9 accelerometers were counted with signal axis accelerometers, the actual number of accelerometers used on the simulator was 4, since they were 3-axes accelerometers.
\[ \dot{\omega}_x = \frac{(a_{z1} - a_{z0})}{2\rho_1} - \frac{(a_{y3} - a_{y0})}{2\rho_3} \]
\[ \dot{\omega}_y = \frac{(a_{x3} - a_{x0})}{2\rho_3} - \frac{(a_{z2} - a_{z0})}{2\rho_2} \]
\[ \dot{\omega}_z = \frac{(a_{y2} - a_{y0})}{2\rho_2} - \frac{(a_{x1} - a_{x0})}{2\rho_1} \]

Equation (4.9)

A comparison between these two methods were made with a 10 seconds white noise signal. Both methods were used to reconstruct the angular acceleration. The results of both methods were very close to each other (Figure 4.5), indicating the approximations made in the first method did not cause large errors.

![Figure 4.5: Reconstruction of angular acceleration by two methods](image)

Although these two methods yielded very similar result, the first method was considered more suitable for future applications. The main benefit of the first method is that its requirement regarding the placement of accelerometers is more flexible. It only requires the accelerometers to have sufficient distance from each other and one of them to be not in the same plane with the others. The second method requires the accelerometers to be placed exactly along the axes of the body-fixed reference frame, which might lead to some locations that are not accessible.

## 4.4 Implementation of Time Waveform Replication process

After reliable measurement of acceleration was acquired, a band-pass filtered white noise signal was used to test the open loop response of the simulator. The PSD of the input signal is shown in Figure 4.6a, and the measured output in the vertical direction is shown in Figure 4.6b. Large distortion can be found between the output and input, indicating proper control algorithm was needed.

![Figure 4.6: Open loop response in the vertical direction](image)
Chapter 4. Control

As introduced in Section 4.1, the TWR process was the standard control algorithm for vibration simulations. The TWR process provides good accuracy for tracking the target acceleration, and the convergence rate was found to be acceptable for the application introduced in Chapter 5.

System identification is the first phase of TWR process. Two types of linear models were identified and then compared in this phase: frequency response function (FRF) and transfer function. The identification of both models follows the same procedure:

1. generate a random signal (white noise), then apply a band-pass filter to remove the low and high frequency content that is out of the operating frequency range.
2. send the band-pass filtered signal to the actuators and take measurements.
3. calculate the linear model with the measured output and the input signal (forward prediction).
4. validate the model by comparing the measured output to the predicted output from the model.
5. invert the model and validate the inverted model by comparing the input signal and the predicted input from the model (backward prediction).

The FRF model $H(f)$ is calculated by:

$$H(f) = \frac{S_{xy}(f)}{S_{xx}(f)}$$  \hspace{1cm} (4.10)

where $S_{xy}(f)$ is the Cross Spectral Density in the frequency domain of input $X(f)$ and output $Y(f)$, and $S_{xx}(f)$ is the Auto Spectral Density in the frequency domain of $X(f)$. The quality of FRF model was checked with the spectral coherence $C_{xy}(f)$:

$$C_{xy}(f) = \frac{S_{xy}(f)^2}{S_{xx}(f)S_{yy}(f)}$$  \hspace{1cm} (4.11)

where $S_{yy}(f)$ is the Auto Spectral Density in the frequency domain of output $Y(f)$. The value of spectral coherence $C_{xy}(f)$ should always satisfy $0 \leq C_{xy}(f) \leq 1$. For an ideal constant parameter linear system, the coherence $C_{xy}(f)$ will be equal to one. The FRF model of the simulator resulted in low $C_{xy}(f)$ at multiple frequency within the operating frequency range, indicating the simulator has strong nonlinearity at these frequencies.

The result of system identification phase has shown that the inverted transfer function model has smaller modeling error than the inverted FRF model. The model validation results of the transfer function model are shown in Figure 4.7. It can be seen that the transfer function and the inverted transfer function have shown good matching in the frequency domain for both forward and backward predictions. The error in the time domain is larger in the backward predictions, which is due to the fact that the transfer function was non-minimum phase so its inversion can only be approximated. Multiple existing studies suggested that the Shannon-Bode method [81] has been used for inverting vibration platform models and yielded accurate result [70, 19].
4.5 Control system performance

4.5.1 Accuracy

The performance of the simulator was validated by first testing its accuracy and iteration time. Proving ground recordings from various tracks and vehicle models

After the system model was identified and inverted, the initial input signal was generated from the inverted model to be used the starting point of the TWR iteration. In each iteration, the new input is calculated with Equation 4.1 and sent to the simulator. The new output is measured and compared to the target to calculate the spectrum error, which is used to check for convergence and to calculate the new input.

The $Q^j$ in Equation 4.1 represent the iteration gain. The purpose of $Q^j$ is to make sure the iterations take conservative correction steps. The classic TWR process uses single scalar values as weightings for each input channel. The weightings may depend on the iteration number $j$ so that the step size is large initially and gets smaller as the iteration approaches the target. It was found in the practices that, when $Q^j$ is too large, the output would diverge or oscillate during the iteration. However, if $Q^j$ is set to be very small in the beginning, then the number of iterations needed to converge would increase significantly. Further investigation revealed that these 2 problems were frequency dependent, i.e. at certain frequency ranges oscillating behavior was very likely to occur while at some other frequency ranges the correction steps seem to be too small. A frequency and iteration number dependent $Q^j(f)$ was used to solve this problem.

The convergence condition of iteration was defined with spectrum error $\varepsilon(f)$:

$$
\varepsilon(f) = \frac{|P_o(f) - P_t(f)|}{P_t(f)}
$$

(4.12)

in which $P_o(f)$ is the PSD of the output measurement, $P_t(f)$ is the PSD of the target, and $f$ is the frequency. The convergence is considered achieved when the 97th percentile of $\varepsilon(f)$ within the operating frequency range is less than 5%.
were tested and the simulator successfully replicated these vibration signals with 10-15 iterations. The system was found to be able to track the target signal accurately within the frequency range of 6-80Hz. The same level of accuracy can be achieved for either seat rail or seat cushion, but not both locations at the same time, which is due to the seat used on the simulator was different from the vehicles measured at the proving ground. A typical result of TWR process targeting the seat rail is shown in Figure 4.8, which shows good matching between the target PSD and the measured PSD in both longitudinal (X) and vertical (Z) directions.

![Output acceleration, X direction](image1.png)

![Output acceleration, Z direction](image2.png)

![Normalized Error, X direction](image3.png)

![Normalized Error, Z direction](image4.png)

**Figure 4.8:** Result of the TWR process. The figures in the first row have shown the target PSD and measured PSD. The figures in the second row shown the spectrum error $\varepsilon(f)$

### 4.5.2 Replay error

After getting the corrected input signal from the TWR process, the simulator also needs to repeat the same excitation multiple times with minimum replay error. A calibrated white noise signal targeting the seat rail was used to test the replay error in the entire operating frequency range. The following 4 scenarios were tested:

1. comparison between two test subjects with different body weight and size. Subject 1 was 178cm and 70kg, while subject 2 was 170cm and 80kg. The replay error caused by two different subjects was over 50% at 10Hz in both longitudinal and vertical directions(Figure 4.9a). This is the largest replay error among the four tested scenarios.

2. comparison between slightly different sitting postures. The test subject was told to stand up and sit down again in the same posture between each run. The replay error was about 18% in the vertical direction and 14% in the longitudinal direction, as shown in Figure 4.9b.

3. comparison between same sitting postures. The test subject was told to keep the same sitting posture as best as he could, which resulted in about 5% replay error in the longitudinal direction and 2% in the vertical direction, as shown in Figure 4.9c.
4. the replay error of the simulator by itself. The simulator was tested without any subject and the resulted replay error was less than 1% in the longitudinal direction and less than 1.5% in the vertical direction, as shown in Figure 4.9d

![Graphs showing replay error variation](image)

(A) Two different subjects  
(B) Different sitting postures  
(C) Same sitting posture  
(D) No occupant

**FIGURE 4.9:** Variation of seat rail acceleration

The results of these experiments have shown that the replay error was mainly caused by the disturbance of human body. It was estimated that the replay error during actual operation would very likely be around 5%-20%, as it is not possible to keep the subject sitting very still for long period of time.

4.5.3 Crosstalk

Another example of the disturbance caused by human body is the crosstalk between the longitudinal direction and vertical direction. Crosstalk is a phenomenon by which the excitation in one direction causing vibration in the other direction. In the example shown in Figure 4.10, the excitation in the vertical direction below 18Hz caused vibration in the longitudinal direction, despite the input signal in the longitudinal direction was cutoff below 18Hz. The crosstalk made the TWR process unable to converge to the target.

The crosstalk below 18Hz was found related to the feet position of the test subject. Putting the feet on the platform as shown in Figure 4.11a resulted in 35% of crosstalk (calculated by the ratio of PSD). A simple feet rest was used to move the feet away from the platform. One or two boxes, depending on the height of the subject, are put on the air mattress to support the feet, as shown in Figure 4.11. The feet rest significantly reduced the crosstalk to 3-10% (individual dependent), which no longer cause convergence issues for the TWR process. This example further demonstrate the influence of human body on the motion of the simulator.
Chapter 4. Control

No input signal below 18Hz in X direction
Yet still have vibration in this frequency range

Figure 4.10: The crosstalk between the longitudinal direction and vertical direction

Figure 4.11: Feet positions

4.6 Conclusion and discussion

A control system for the simulator was built around a standard PC with MATLAB environment. TWR process was successfully implemented into the system and yielded good accuracy. However, the replay error of the output was heavily affected by the human body, which made it necessary to calibrate the signal for every test subject individually, and the subject needs to remain the same sitting posture throughout the experiment.

The lightweight design of the simulator might be part of the reason of low replay error. The weight of the simulator including the seat is about 80kg, which is close to human body weights. Therefore, the simulator is sensitive to changes in body weights and postures.

A possible solution for this problem is introducing real-time feedback control. Previous studies have experimented multiple types of feedback control strategy such as linear feedback [13], $\text{H}_\infty$ feedback [14], fuzzy control with feedback [68]. However, most of these studies aimed to accelerate the convergence of the TWR process instead of compensating the disturbance from human body. In fact, most of the vibration experiments introduced in the aforementioned studies did not involve human test subject. The application of real-time feedback control for compensating the distortion of human body would be worth investigating.
Chapter 5

Application: measurement method of difference threshold

In this chapter, an experimental method is proposed and validated for measuring the difference threshold of human perception of seat vibration with the seat vibration simulator.

The difference threshold, also known as Just Noticeable Difference, was introduced in Section 2.3.2. It is one of the method for describing human sensitivities of vibration. The advantage of difference threshold over the equivalent comfort contour is that it can be easily measured based on realistic vehicle seat vibrations, instead of the simple sinusoid stimuli that are often used for the measurement of equivalent comfort contour [57, 56]. This advantage made it promising to develop new ride comfort indicators based on the difference threshold of seat vibration, which is further discussed in Section 5.3.

The difference threshold need to be measured for many different vehicle types, road types, and driving conditions, in order to develop the ride comfort indicator, so the measurement need to be highly efficient. A novel measurement method is proposed in Section 5.1, which use stimuli generated from proving ground measurements and a modified A-AFC method.

The measurement method was tested with trail runs and was found to be effective and efficient, as introduced in in Section 5.2. The trail run exposed a problem of large replay error of the stimuli. The mitigation plan of calibrating / re-calibrating the stimuli before every run is proposed and discussed.

The application of difference threshold is introduced in Section 5.3. The drawbacks of current standard methods are reviewed. Then a new ride comfort indicator based on difference threshold is proposed to improve these drawbacks.

5.1 Method

The experiment of measuring difference threshold consist of 2 steps: preparation of stimuli and measurement. The first step is introduced in Section 5.1.1, in which all of the stimuli, are generated and calibrated for the test subject. The second step is the measurement of difference threshold with the modified A-AFC protocol, which is introduced in Section 5.1.2 and 5.1.3. The experiment was fully automated with a MATLAB GUI, as introduced in Section 5.1.4.
5.1.1 Stimuli generation

There are two types of stimuli used in the experiment: reference and test stimuli. The most important stimuli is the reference because it will be used in every run of the experiment. Test stimuli are generated by applying spectrum manipulation to the reference stimulus.

Reference stimulus The reference stimulus represents the spectral characteristics of seat vibration of the selected road and vehicle types, which is further explained as the “linearization point” in Section 5.3. It is generated from the proving ground measurement reviewed in Section 3.1. The first step is generating the spectrum baseline from proving ground measurements. Then the reference stimulus is created by filtering a random signal to make its spectrum identical to the spectrum baseline.

The spectrum baseline represent the general spectrum characteristic for the selected vehicle type and driving conditions. The review of proving ground measurement in Section 3.1 has shown that the vibration of the same type of vehicle are rather similar when subject to same road input and driving conditions. Inspired by this similarity, the spectrum baseline was defined as the median of the PSD curves of measurements from all these vehicles, as shown in Figure 5.1. Assuming the PSD curves of these $n$ vehicles are $p_{x1}(f), p_{x2}(f), ..., p_{xn}(f)$ for the longitudinal direction, and $p_{z1}(f), p_{z2}(f), ..., p_{zn}(f)$ for the vertical direction, then they can be combined into 2 matrices $P_x(f)$ and $P_z(f)$:

$$ P_x(f) = \begin{bmatrix} p_{x1}(f) & p_{x2}(f) & \cdots & p_{xn}(f) \end{bmatrix} $$

$$ P_z(f) = \begin{bmatrix} p_{z1}(f) & p_{z2}(f) & \cdots & p_{zn}(f) \end{bmatrix} \tag{5.1} $$

Then the spectrum baseline in longitudinal direction $b_x(f)$ and vertical direction $b_z(f)$ are calculated by taking the median of each row of $P_x(f)$ and $P_z(f)$.
The resulted spectrum baseline does not represent any real vehicles, instead it represent the “average” performance of the selected vehicle type, road type, and driving conditions.

The reference stimulus is created as random signal that has the same PSD as the spectrum baseline. The random signal is initially created as a stationary white noise signal. Then its spectrum is manipulated to match with the spectrum baseline, and a band-pass filter is applied to remove the low and high frequency content that is out of the operating range of the simulator. The resulted signal remains stationary, and has the desired spectrum, as shown in Figure 5.2. The spectrum of the reference stimulus is not a perfect match to the spectrum baseline due to its limited length.

The stimuli need to be long enough so that the vibration can be considered stationary for the operating frequency range of the simulator. On the other hand, they should not be too long as it will significantly increase the duration of the experiment. Test subjects have reported that they often have difficulties comparing the two stimuli if they are too long, because they often forgot the feeling of the first one during the playback of the second one. It was found that the stimuli length of 10 seconds provided the best trade off on this issue.

![Reference stimulus, X direction](image1)

(A) Longitudinal direction

![Reference stimulus, Z direction](image2)

(b) Vertical direction

**Figure 5.2:** PSD of the reference stimulus

The reference stimulus generated in this method was verified by professional evaluators at Toyota Motor Europe and was concluded to be realistic. One of the evaluators even identified the track that the reference stimulus was based on.

**Test stimuli** The test stimuli are created from the reference stimulus with spectrum manipulations at the selected frequency bands. Each frequency band corresponds to a difference threshold measurement. Therefore, the selection of frequency bands decides the frequency resolution of the measurement. More frequency bands give better frequency resolution, at the cost of longer experiment duration.

For a given reference stimulus $a_{\text{ref}}(t)$, it is converted into the frequency domain $A_{\text{ref}}(f)$ with Discrete Fourier Transform (DFT). Then $A_{\text{ref}}(f)$ correspond to the frequencies within the selected frequency band $[f_1, f_2]$ are scaled by a scaling factor $k$ ($k > 1$), giving the test stimulus $A_{\text{test}}(f)$. Finally $A_{\text{test}}(f)$ is converted back to time domain $a_{\text{test}}(t)$ with Inverse Discrete Fourier Transform (IDFT). The scaling factor $k$ is also called the level of the test stimuli.
\[ A_{test}(f) = \begin{cases} kA_{ref}(f), & f \in [f_1, f_2] \\ A_{ref}(f), & \text{otherwise} \end{cases} \] (5.2)

### 5.1.2 Classic A-AFC experiment protocol

For measuring the difference threshold in vibration level (amplitude), the Adaptive Alternative Forced Choice (A-AFC) protocol was considered to be more efficient than the Constant Stimuli method, as introduced in Section 2.4.2.

The experiment protocol controls the scaling factor \( k \) of the test stimuli. In every run of the experiment, two stimuli are presented to the test subject in random order, one of them is always the reference stimulus, and the other one is the test stimulus. The subject is asked to indicate which of the two stimuli feels more uncomfortable (stronger), i.e. to identify the the test stimuli. \( k \) in the next run depends on the responses from subject.

The \( x \)-up \( y \)-down characterizes the the adapting strategy of \( k \), where \( x \) is the number of consecutive incorrect answers and \( y \) is the number of consecutive correct answers. The scaling factor \( k \) is increased (moved up) when the subject has made \( x \) consecutive incorrect answers, and is decreased (moved down) when the subject has made \( y \) consecutive correct answers. This process is illustrated in Figure 5.3. The result of \( x \)-up \( y \)-down strategy is that \( k \) will converge on the level where the probability of increase and decrease is equal [43].

For example, for 1-up 2-down strategy, the probability of move up \( P_{up} \) and move down \( P_{down} \) can be expressed by the probability of correct response \( P(X) \):

\[ P_{up} = [P(X)]^2 \]
\[ P_{down} = P(X)[1 - P(X)] + [1 - P(X)] \] (5.3)

Then the value of \( P(X) \) at convergence can be solved by setting \( P_{up} = P_{down} \):

\[ P_{up} = P_{down} \]
\[ [P(X)]^2 = P(X)[1 - P(X)] + [1 - P(X)] \]
\[ [P(X)] = 0.707 \] (5.4)

An example of a typical result of 1-up 2-down strategy is given in Figure 5.3. The value of \( P(X) \) at convergence for other \( x \)-up \( y \)-down strategy can be solved in similar ways. For quick reference, \( P(X) = 0.5 \) for 1-up 1-down strategy, and \( P(X) = 0.794 \) for 1-up 3-down strategy.

There are 3 possible outcomes for \( k \) in each run. It either increases or decreases if the condition for move up or move down is met, or stays unchanged if neither of the conditions is met. These outcomes are called “up”, “down”, and “straight” accordingly. The “straight” step often occurs when the subject has made a correct response but has not yet reached the \( y \) consecutive correct responses, such as the step 5 and 10 in Figure 5.3.

The common practice is to start the experiment with a high enough initial value of \( k \) so that the differences can be easily detected by the subjects. This will help the subjects to get familiarized with the experiment and the stimuli.
5.1. Method

Since the initial level is well above the difference threshold, it is desirable to have adaptive step size for $k$, i.e. the step size $\delta k$ should be initially large and gradually reduces as $k$ approaches convergence. The initial and final (minimum) step sizes $\delta k_{\text{min}}$ are selected prior to the experiment. The step size is halved on every appearances of “upper reversal” [4] in the classic A-AFC strategy. The upper reversal is defined as an “up” step followed by some “straight” step (depend on the selection of $y$), then followed by a “down” step. For example, the upper reversal occurred at step 12, 17, and 20 and are marked with blue squares in Figure 5.3.

The convergence condition of the experiment is defined by the number of “upper reversals” $n_c$ after the minimum step size $\delta k_{\text{min}}$ is reached. The difference threshold is the median of the values of $k$ that corresponds to the $n_c$ “upper reversals” [4]. In the example given in Figure 5.3, $n_c$ was select to be 3, and the result of the experiment was the level corresponding to index 3. It is worth pointing out that although the last “upper reversal” is at the 20th run, the experiment was actually terminated at the 19th run, because the responses of the first 19 runs were already sufficient to determine the convergence, therefore no response was needed from the subject at the 20th step.

There are two extreme cases during the experiment that will not leads to convergence: 1) the subject is very insensitive to vibration and cannot distinguish any difference, then the subject will keep giving incorrect responses and stay at the maximum level $k_{\text{max}}$ all the time; 2) the subject is very sensitive and can distinguish even the smallest difference, then the subject will quickly reach and stay at the minimum level $k_{\text{min}}$. The “upper reversal” will not occur in either of these two cases, therefore the experiment will never meet the convergence condition. To solve this issue, the maximum number of runs at the maximum and minimum level $n_{\text{out}}$ were defined as the additional termination conditions. The experiment is terminated and considered “out of limit” once this conditions is met at either $k_{\text{max}}$ or $k_{\text{min}}$.

It was observed in the early trail runs of the experiment that, the subject might become more sensitive to changes of stimuli if the subject was aware of which frequency band is being tested. The interleaved measuring method was implemented to avoid this effect, as suggested by [4]. This method runs multiple experiments for different frequency bands in parallel while sharing the same reference stimulus. The
Chapter 5. Application: measurement method of difference threshold

Table 5.1: Parameters of the reference simulation

<table>
<thead>
<tr>
<th>Name of parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum level $k_{min}$</td>
<td>0%</td>
</tr>
<tr>
<td>Maximum level $k_{max}$</td>
<td>100%</td>
</tr>
<tr>
<td>Minimum step size of $\delta k_{min}$</td>
<td>1%</td>
</tr>
<tr>
<td>Number of consecutive incorrect answers for move up $x$</td>
<td>1</td>
</tr>
<tr>
<td>Number of consecutive correct answers for move down $y$</td>
<td>2</td>
</tr>
<tr>
<td>Number of “upper reversals” for convergence $n_c$</td>
<td>3</td>
</tr>
<tr>
<td>Maximum number of runs at the maximum and minimum level $n_{out}$</td>
<td>8</td>
</tr>
</tbody>
</table>

test runs of different experiments are present in random order. Therefore, subjects would not know which frequency band is being tested.

5.1.3 Reducing the duration of experiment

Another problem that was exposed in the trail runs was the duration of the experiment was too long. It was found that looking for small differences among two vibration stimuli requires huge mental effort. Most subjects got exhausted after 1-2 hours of full concentration. And their sensitivity were very likely to be affected by the fatigue. Therefore, this section aimed to make modifications of experiment protocol for reducing the duration of experiment.

As mentioned in Section 5.1.1, the number of frequency bands and length of stimuli will directly affect the duration of experiment. However, reducing them could also leads to compromises of measurement quality. Therefore these parameters were not the best options.

The modification of experiment protocol was tested with simulated responses, which were generated based on the psychometric function measured in [4] with a hypothetical difference threshold of 30%, as shown in Figure 5.4. The standard 1-up 2-down A-AFC experiment protocol was implemented in the simulation as the reference. All parameters of the simulation were shown in Table 5.1.

![Figure 5.4: The psychometric function used in the simulation](image-url)
The initial result indicated that the distribution of convergence level over large number of experiments (10,000 experiments) is similar to a normal distribution, as shown in Figure 5.5. The simulation result was evaluated by the average number of runs till convergence $n_{\text{avg}}$, the percentage of “out of limit” cases $P_{\text{out}}$, the mean of convergence levels $X_c$, and the standard deviation of convergence levels $s_c$. $n_{\text{avg}}$ is the indicator of experiment duration. $X_c$ and $s_c$ are the indicator of experiment quality. A high quality experiment should have $X_c$ close to the threshold defined in the psychometric function, and $s_c$ as small as possible.

Another indicator of the experiment quality is the out of limit rate. The simulation has shown that there are some rare cases that all the simulation responses were correct and the experiment was terminated as “out of limit”. An example of such cases is shown in Figure 5.6. Clearly, at least part of the correct responses were purely by chance since the minimum level was well below the difference threshold defined in the psychometric function.

The parameters of the experiment protocol that are directly related to duration were tested first, including the number of “upper reversals” for convergence condition $n_c$, and the number of correct response needed before “move down” $y$. These two
parameters were initially set to \( y = 2 \) and \( n_c = 3 \), which were the lowest values possible. As shown in Table 5.2, both increasing \( n_c \) and \( y \) will cause significant increase in \( n_{avg} \), with only minor reduction of \( s_c \). Therefore it was concluded that additional duration caused by increasing \( n_c \) and \( y \) is not worth the benefit.

<table>
<thead>
<tr>
<th></th>
<th>( n_{avg} )</th>
<th>( P_{out} )</th>
<th>( X_c )</th>
<th>( s_c )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference</td>
<td>53.9</td>
<td>1.0%</td>
<td>29.2</td>
<td>4.8%</td>
</tr>
<tr>
<td>Change ( n_c ) from 3 to 5</td>
<td>67.2</td>
<td>1.1%</td>
<td>28.9</td>
<td>4.4%</td>
</tr>
<tr>
<td>Change ( x ) from 2 to 3</td>
<td>75.3</td>
<td>0.3%</td>
<td>32.6</td>
<td>3.3%</td>
</tr>
<tr>
<td>New step size strategy</td>
<td>35.9</td>
<td>0.6%</td>
<td>26.2</td>
<td>6.4%</td>
</tr>
<tr>
<td>New step size strategy and “pass” option</td>
<td>34.4</td>
<td>0%</td>
<td>29.1</td>
<td>3.8%</td>
</tr>
</tbody>
</table>

The simulation results have shown the following two modifications provided reduction of duration without compromising the measurement quality. The first modification was the strategy of reducing step size. Previously, the step size is halved only at “upper reversals”. The new strategy proposed to halve the step size at both “upper reversals” and “lower reversals”, i.e. the step size is halved whenever the current step is not “straight” and is different from the previous step that is also not “straight”. This modification allows the step size to reduce more rapidly, which accelerated the convergence because \( n_c \) is counted only after the minimum step size \( \delta k_{min} \) is reached. This change reduced \( n_{avg} \) by about 30% with slight increase of \( s_c \).

The second modification was the addition of “pass” option to the test subject, which is intended to be used when the subject could not distinguish the difference between the two stimuli. Previously, the subject was forced to choose one of the stimuli, even randomly. There is 50% of probability that the correct answer is chosen by chance, then the experiment would spend more runs at this level or even lower levels, which is unlikely to contribute to convergence. The “pass” option works as an 100% incorrect response that always cause the level of the next step to move up to avoid wasting time at lower levels. However, there was a difficulty of simulating the “pass” option because the probability of subjects to use this option is unknown. A speculated probability function was created to approximate it, as shown in Figure 5.7. The real-world usage of “pass” option was later found to be highly inconsistent among individuals. The simulation result has shown that introducing the “pass” option completely eliminated the “out of limit” cases, and reduced \( s_c \) by almost 50% (comparing to the result of new step size strategy only).

### 5.1.4 Automated experiment

A MATLAB program was developed for fully automated experiment, including generating the test stimuli, conducting the experiment according to the protocol, recording measurements during the experiment, and post-processing and visualizing the results.

The experiment protocol introduced in Section 5.1.2 is very suitable to be conducted by computer program because it is consist of clear and fixed rules that can be programmed easily. In addition, the protocol has two random procedures: the random order of reference and test stimuli, and the random order of parallel experiments required by the interleaved measuring method. Executing the protocol manually
would be time consuming and prone to human error, especially since the experiment often has rather long duration.

Another advantage of automated experiment is that it eliminated the influence of experiment operator, i.e. the experiment becomes “double-blind”, since the subjects do not need to interact with the operator during the experiment. It also helps to reduce the mental stress of the subjects because they can take their time to make the decision, knowing that there is no operator expecting his/her response. An additional “replay” option was given to the subjects for replaying the two stimuli, which was later found to be used very often by the subjects. The subjects would probably be much less willing to use this option if the experiment is operated manually also due to the mental stress of asking the operator to replay the stimuli.

An Graphical User Interface (GUI) was developed for this program, as shown in Figure 5.8. The two windows on the left side is displayed on the monitor facing the operator, for setting up the experiment and monitoring the progress. The two windows on the right side is displayed on the other monitor facing the test subject, for the giving instructions to the subjects and taking responses.

A Xbox controller was re-mapped to keyboard for taking input from the subjects. It is more comfortable to used than a normal keyboard due to its ergonomic design and smaller size. All subjects had no difficulty learning to use it. Additionally, holding the controller with both hands helped to keep their arms at a stable position, i.e.
naturally resting on their laps, which helped to keep their sitting posture consistent and reduce the replay error as explained in Section 4.5.

5.2 Trail run results and Discussion

A few trails runs have been made to validate the measurement method of difference threshold proposed in Section 5.1. The method of generating stimuli based on proving ground measurement was found to give good representation of the proving ground and vehicle type. The modification of experiment protocol was found effective and efficient.

There are 4 frequency band and direction combinations tested in the trail run: 10-20Hz longitudinal, 20-40Hz longitudinal, 10-20Hz vertical, and 20-40 vertical. For each of these combinations, 6 different stimuli levels $k$ were prepared based on previous experience. The other parameters of the experiment protocol are shown in Table 5.3.

<table>
<thead>
<tr>
<th>Table 5.3: Parameters of the trail runs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Name of parameters</td>
</tr>
<tr>
<td>Levels for 10-20Hz longitudinal</td>
</tr>
<tr>
<td>Levels for 20-40Hz longitudinal</td>
</tr>
<tr>
<td>Levels for 10-20Hz vertical</td>
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<tr>
<td>Levels for 10-20Hz vertical</td>
</tr>
<tr>
<td>$x$</td>
</tr>
<tr>
<td>$y$</td>
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<tr>
<td>$n_c$</td>
</tr>
<tr>
<td>$n_{out}$</td>
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</table>

The usage of the added “pass” and “replay” option was found to be highly individual dependent. The inexperienced test subjects were more likely to choose “pass” option, while the professional evaluators are less willing to give up and would use the “replay” option more often.

The preliminary results are shown in Figure 5.9, 5.10, and 5.11. The correct responses are marked by green circles, the incorrect responses are marked by red circles, and the “pass” are marked by red diamond. The vertical axes represent the scaling factor $k$ of the test stimuli, and the horizontal axes represent the number of runs. It can be seen that most of the experiment reached convergence within about 20-25 runs except two experiment terminated due to “out of limit”, and one experiment was terminated early due to the schedule of the subject (first experiment of subject A). The resulted difference threshold measurements are summarized into Table 5.4.

<table>
<thead>
<tr>
<th>Table 5.4: Preliminary results of difference threshold</th>
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<tr>
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<tr>
<td>Subject A</td>
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<tr>
<td>Subject B</td>
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<tr>
<td>Subject C</td>
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</table>
Large variation can be found among subjects, which might due to the fact that subject A and C are professional evaluators, while subject B has no experience regarding evaluating ride comfort.

Although the number of subjects is too small for giving the exact value of the difference threshold, relative comparison can be made between different frequency bands. The results have shown the difference thresholds at 10-20Hz for both directions are very likely to be smaller than 20-40Hz, indicating the subjects are more sensitive to changes at 10-20Hz. The differences in sensitivities could be due to the reference stimulus contains more energy within 10-20Hz than 20-40Hz, as can be seen from Figure 5.1.

There are some inconsistency in the result. The second experiment of subject B (Figure 5.10) is a good example. The subject has shown a sudden change in the sensitivity after 12th run, which caused the experiment to converge at a much higher level. This change might due to the perception of the subject was affected by fatigue.

The replay error of the stimuli after the calibration was found to be the main reason of the “out of limit” cases. For example, the first and third experiments of subject C (Figure 5.11) got all correct responses from the subject, which led to the termination of experiments. The measurements during the experiment showed that the actual
stimuli presented to the subject during these experiments were different from the calibration, i.e. larger replay error occurred. The replay error was larger or more perceivable than what the differences between the test and reference stimuli should be. Therefore, it was possible that the response of the subject was given by identifying the replay error instead of the differences between the test and reference stimuli.

Further investigation revealed the replay error was caused by changes of sitting postures after the calibration, as discussed in Section 4.5. All the stimuli used in the experiment were calibrated in advance before the experiment, which normally took 0.5-1 hour. And the experiment took another 1-1.5 hour. The changes of sitting postures during the experiment were almost inevitable since it was not practical to keep the test subject maintain the exact same sitting posture for over 2 hours. The replay error of the stimuli were not checked or controlled during the experiment.

The mitigation plan for reducing the replay error is calibrating / re-calibrating the stimuli before every run. This calibration method has multiple advantages over the previous one. Since the stimuli are calibrated right before the replay, the time between calibration and test is minimized, thus minimizing the probability of subject making posture changes. Even if the subject changes the sitting posture in between the runs, the calibration before the next run will compensate for it. This strategy
also allowed the subject to pause the experiment and resume it later. Then the subjects will be less affected by fatigue since they are free to take breaks. On the other hand, the calibration of all stimuli at the beginning of the experiment will no longer be needed, which could partly compensate for the time spent on calibrating / recalibrating. This also allows wider range of stimuli levels $k$ and smaller final step size $\delta k_{\text{min}}$ to be used, which would improve the measurement quality and eliminate the requirement of prior knowledge regarding the measurement.

5.3 New ride comfort indicator

The studies about human perception of seat vibration in the 1980s [26, 27, 64, 65, 25] were summarized into BS6841 [7], which was adopted later by ISO2631 [33] with minor modifications. Until today, ISO2631 is still the most relevant standard regarding objective evaluation of seat vibration.

The first step of the standard methods is taking acceleration measurements at the locations that the body is exposed to vibration, such as seat, steering wheel, and floor. For seat vibration, it should be measured by accelerometers installed between human body and the seat. The acceleration signals should then be filtered (apply
frequency weightings) and averaged (with Root Mean Square or other averaging methods). Finally averaged acceleration values at different directions and locations are combined with corresponding multiplying factors. The result is a single value indicator representing the discomfort caused by the exposure to vibration.

5.3.1 Problems with current standards

Despite these standards have been introduced for decades, they have seen only limited usage in the automotive industry. They are often criticized for having mismatches with the subjective evaluation results, and being unable of differentiating small differences in ride comfort.

Most ride comfort evaluation in automotive industry are aimed to identify small differences between very similar vehicles, such as benchmarking against competitive models. The standard methods often struggle to differentiate such small difference. An example of this problem was given by applying ISO2631 to the proving ground measurements reviewed in Section 3.1, as shown in Figure 5.12. It can be seen that the indicator values of these 6 vehicles were very close to each other. Although it is possible to tell if one vehicle performed better than the other, it would be very difficult to quantify the differences, which was supposed to be one of the main benefits of objective evaluation.

The underlying assumption of the frequency weightings and multiplying factors is that the discomfort caused by vibration with different frequencies, directions, and locations can be combined linearly. With this assumption, the discomfort caused by any vibrations can be evaluated and compared, regardless of whether the vibration is narrowband or broadband, single-directional or multi-directional, acting on single location or multiple locations of the body. However, this assumption might have over-simplified the human perception process. The assumption of linear combination was never validated, especially for vehicle seat vibration [50]. It was found that 3 signals with identical indicator values (ISO2361 method) but different frequency spectrum received different subjective ratings, and the mismatches grew with the amplitude [38]. The masking effect could also affect the perception of seat vibration [32]. These studies suggested that human perception of vibration is very likely not...
5.3. New ride comfort indicator

a linear process, and treating it as a linear process will inevitably cause linearization error.

The concept of “linearization point” in mathematics can also be applied to the linear assumption of the standard methods. The “linearization point” represent a reference vibration stimulus. It was at this point that the human perception was measured. Similar to the linearization in mathematics, the “linearization point” should be as close to the “point of interest” as possible in order to minimize the linearization error. The “point of interest” is the vibration of real vehicles for the application of evaluating ride comfort. Therefore, the “linearization point” should be close to the vibration of real vehicles.

The standards did not specify their “linearization point” since they assume human perception was linear for any vibration. However, the studies of equivalent comfort contours [26, 27, 64, 65, 25], which were the base of the the standards, did specify their reference stimuli. According to these studies, their reference stimuli were single frequency sinusoidal or narrow-band random signals (i.e. 1/3 octave), which were very different from the characteristic of real-world vehicle seat vibrations and resulted in the aforementioned linearization error. The experiment of difference threshold proposed in this chapter enables the measurement of human sensitivity with the “linearization point” placed very close to the real-world vibration, thus the linearization error can be minimized.

Another reason for the larger mismatches at higher amplitude in [38] might be that Stevens’s power law (equation 2.1) was not fully adopted in the standards. The standards have no coefficient regarding the amplitude of vibration, thus they assume the exponent \( n \) in equation 2.1 to be 1, i.e. a linear relation between the indicator value (such as weighted RMS) and the discomfort. Recent studies have found that \( n \) is very like not 1 and it dependent on the amplitude, frequency, and direction of vibration. For example, \( n \) tends to be greater with low-frequency vibration than with high-frequency vibration [57, 56, 82]. However, there is no study so far has fully investigated this issue and found the optimal value of \( n \). So Stevens’s power law probably will not be incorporated in the evaluation of ride comfort in the near future, which added another reason to place the “linearization point” close to the real-world operating condition.

5.3.2 Proposal for the new ride comfort indicator

The difference threshold measured with the method proposed in this chapter can be used for developing new ride comfort indicators in order to address the aforementioned issues of the standards. It can be converted into frequency weightings and multiplication factors that are specific for each track and vehicle type. The new indicator will have a similar structure as ISO2631, with the main differences being:

1. a separation step of stationary and non-stationary content will be added at the beginning. The non-stationary content of the acceleration will be removed and treated separately with a different method. Only the stationary content will be evaluated in the later steps.

2. the spectrum difference between the measured acceleration and the spectrum baseline will be calculated. The spectrum baseline represents the “linearization
point”, which is the general behavior of a certain type of vehicle on a certain type of road, as introduced in Section 5.1.1.

3. the frequency weightings and multiplying factors based on the difference threshold will be applied to the spectrum difference, instead of universal frequency weighting and multiplying factors.

This indicator can be validated by checking the matching between the objective indicator and subjective ratings. The validation will be conducted on the simulator with stationary vibration stimuli generated with the method introduced in Section 5.1.1.

The next stage of the study will develop indicators for non-stationary vibration stimuli. The method for separating the stationary and non-stationary vibration will need to be developed first. Then the indicator for non-stationary vibration can be developed based on difference threshold of non-stationary stimuli. The experiment protocol introduced in Section 5.1.2 can also be applied to non-stationary stimuli with minor modifications since the parameters controlled by the protocol should reflect the characteristics of non-stationary stimuli, such as peak acceleration, damping, etc. The indicator for non-stationary vibration will also be validated first on the simulator, then applied to real-world measurements after combined with the stationary method.

It is worth pointing out that the proposed new indicators will be vehicle type and road type specific, therefore it will be necessary to conduct measurement of difference threshold for all the combinations that are of interest in order to develop the corresponding indicators. A database can be developed based on the difference threshold measurements and ride comfort indicators. The dependency of ride comfort on road and vehicle type can be investigated once enough data has been collected, which may contribute to developing a new ride comfort indicator that is valid for multiple vehicle and road types.
Chapter 6

Summary and future work

The present study is consist of two parts. The first part introduced the development of seat vibration simulator, and the second part discussed the application of simulator for measuring difference threshold of seat vibration. The study is summarized in Section 6.1. The main contribution is stated in Section 6.2. The suggestion for future work is given in 6.3.

6.1 Summary

The first part of the study started with the review of existing researches regarding human perception of seat vibration. From the review, it became clear that simulators are the optimal research tool for this topic as they provide good control over the vibration stimuli and experiment environment. The review provided background knowledge regarding modeling human perception of vibration and corresponding measurement methods. The review also provided guidelines for specifying the design requirements of the simulator.

The design requirements were defined based on the review of proving ground measurements and the other requirements from Toyota Motor Europe. It concluded that the operating frequency range was 10-80Hz with excitation applied in both longitudinal and vertical directions. A compact and cost-effective design of seat vibration simulator was proposed featuring a modular floating vibration platform and compact electrodynamics actuators acting in longitudinal and vertical directions.

The existing control strategy of vibration simulators was reviewed and the Time Waveform Replication (TWR) process was found to be the standard solution for replicating vibration on simulators. A control system was built to implement the TWR process. Two different rigid body transformation methods were implemented for reconstructing the 6 DOF motion of the seat. The TWR process was implemented and yielded good accuracy of tracking the target stimuli within the frequency range of 6-80Hz, exceeding the design requirement of 10-80Hz. The replay error of the stimuli was found to be heavily affected by the human body and sitting posture. Introducing real-time feedback control was suggested to further reduce the replay error.

The second part of the study discussed the application of the seat vibration simulator. An experimental measurement method was proposed and validated for measuring the difference threshold of human perception, which can be used to develop new
ride comfort indicators. The spectrum baseline method was developed for generating the stimuli from proving ground measurements. Two modifications were made to the classic A-AFC experiment protocol for reducing the experiment duration:

1. new policy for reducing the step size.
2. adding the “pass” option.

These modifications reduced the experiment duration by about 30% without compromising measurement accuracy. The proposed method was tested with a small number of trial runs. The method was found effective and efficient. The replay error was found causing “out of limit” terminations in a few cases. The mitigation plan of calibrating / re-calibrating the stimuli before every run was proposed.

The study also proposed a new ride comfort indicator based on the difference threshold, which aimed to improve some of the drawbacks of current standard methods. The new indicator has two key improvements over the standards:

1. separation of stationary and non-stationary vibration.
2. using difference threshold as the indicator of human sensitivity, which is measured with stimuli that represent real-world vehicle seat vibration.

The future work of measuring the difference threshold and developing ride comfort indicators for non-stationary vibration was proposed.

6.2 Main contributions

The main contributions of the present study are:

- the definition of design requirements of the seat vibration simulator based on the review of proving ground measurements and previous studies.
- the delivery of fully functional simulator that meets all the requirements. The design of the simulator featuring a modular floating vibration platform and compact electrodynamics actuators. The TWR process was implemented on the simulator which enabled accurate replication of vibration stimuli in both longitudinal and vertical directions with the frequency range of 6-80Hz.
- the proposal and validation of measurement method of difference threshold, including the spectrum baseline method for generating reference stimulus from proving ground measurements, the modified A-AFC protocol for reducing the experiment duration, and the software for the fully automated experiment.
- the proposal of potential application of the difference threshold as a new ride comfort indicator.

6.3 Future work

The immediate future work of the present study is to implement the calibrating / re-calibrating protocol of stimuli and then continue the measurement of difference threshold. Once the measurement is finished, the new ride comfort indicator proposed in Section 5.3 can be developed and validated, as discussed in Section 5.3.2.
The real-time feedback control suggested in Section 4.6 might able to further reduce the replay error, if the calibrating / re-calibrating protocol is proven to be not effective. As explained in Section 4.5, the replay error is about 5% when the subject tried to keep the same sitting posture, which might still be too large for the measurement. Thus the real-time feedback control will be necessary for the difference threshold measurement.

The system identification method introduced in Section 4.4 could be improved. Current method has resulted in large modeling errors, and the process of identifying and inverting the transfer function model was complicate and time-consuming. One of the possible reason for large modeling error might be the system identification was conducted with a white-noise signal, which is very different from the stimuli used later in the experiment. Identifying the model with the experiment stimuli was attempted but did not succeed, because the stimuli cannot provide enough excitation to the entire operating frequency range of the simulator. One of the options for future studies for reducing modeling error could be the iterative identification method introduced in [13].

It would be helpful to further investigate the usage and effect of the “pass” option. The present study was not able to verify the effect of the “pass” option due to the small number of test subjects, and the simulation in Section 5.1.2 did not provide reliable estimation regarding the effect because the probability of selecting this option in the real world is still unknown.

Another issue with the simulator is that its capability of the maximum stimuli amplitude was not clear due to lacking safe and reliable measuring method. The most common behavior for exceeding the capability is the “bottoming” of actuators, which is caused by the piston of the actuator hitting the case at the end of the cylinder. The “bottoming” issue is most likely to occur with large amplitude low-frequency stimuli. However, it is very difficult to measure the position of the piston since it was fully enclosed by the case. Therefore, it was not possible to know how much distance is left between the piston and the case during operation. The only indicator available was the impact noise caused by “bottoming”, which was obviously too late for purpose of preventing it from happening. Future work should develop a method to predict the possibility of “bottoming” from the input signal. A model of the actuator might be needed so that the piston motion can be simulated before actually sending the signal to the actuator.
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Appendix A

Drawings of components

Figure A.1: Seat connector stem
Figure A.2: Seat connector base
Figure A.3: Seat connector assembly
Appendix A. Drawings of components

Figure A.4: Base
Figure A.5: Bracket assembly
FIGURE A.6: Bracket main
Figure A.7: Bracket side
FIGURE A.8: Fan cage