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Implementation of vibration feedback in a motion based driving simulator

Master's thesis in Systems, Control and Mechatronics

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DEPARTMENT OF MECHANICS AND MARITIME SCIENCES

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Abstract

Driving simulators are efficient tools for research. However, traditional moving-base platforms have limited ability to reproduce high-frequency vibrations. As a result, the simulation may feel less realistic and lead to increased sensory mismatch. This thesis presents the implementation of a haptic seat actuator in the SimIV driving simulator at the Swedish National Road and Transport Research Institute (VTI), with the aim of strengthening the driver's perception of realistic vehicle motion.

Field measurements were carried out using both internal combustion engine and electric vehicle to try to isolate road and engine-induced vibrations across various speeds. Frequency analysis revealed several characteristics including wheel resonance, longitudinal back and forth body motion, tyre structural modes, and engine harmonics.

A bass shaker (Xcite XBS100-4) was welded onto a plate beneath the simulator seat. A Python algorithm was developed to process live simulator telemetry via UDP. The algorithm continuously blends several vibration sources together, road noise, engine vibration, braking feedback and simulated rumble strips.

A small user validation study suggested that the integration of the haptic vibrations improved the perceived realism of the simulator, the subjective connection to the road, and strengthened the physical feedback of vehicle speed and acceleration.

Keywords: Sim IV, motion-cueing, haptic feedback, vibrations, simulation, simulator, driving simulator

Preface

This report presents the outcome of our master's thesis project carried out at the Department of Mechanics and Maritime Sciences at Chalmers University of Technology during the spring of 2026. The project focuses on enhancing motion cueing and realism in moving-base driving simulators through the implementation of a haptic seat actuator.

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We would like to thank our examiner Fredrik Bruzelius, and our supervisor Smit Saparia, for their guidance and support throughout this project. We also would like to send our thanks to Arne Johansson at VTI and ESAB for the help with mounting the actuator beneath the seat.

Gabriel Arvidsson, Edward Patel, Gothenburg, June 2026

List of Acronyms

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

ABS	Anti-lock Braking System
CSV	Comma-Separated Values
DFT	Discrete Fourier Transform
DOF	Degrees of Freedom
EV	Electric Vehicle
FFT	Fast Fourier Transform
FRF	Frequency Response Function
ICE	Internal Combustion Engine
IMU	Inertial Measurement Unit
ISO	International Organization for Standardization
MADI	Multichannel Audio Digital Interface
PSD	Power Spectral Density
RFV	Radial Force Variation
RMS	Root Mean Square
RPM	Revolutions Per Minute
TRS	Tip-Ring-Sleeve
TWR	Time Waveform Replication
UDP	User Datagram Protocol
VDV	Vibration Dose Value
VNC	Virtual Network Computing
VTI	Swedish National Road and Transport Research Institute
WAV	Waveform Audio File Format

Nomenclature

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

Indices

k	Index for frequency bin in DFT
n	Index for discrete-time samples
m	Index for overlapping segments in Welch's method
j	Index for iterations in Time Waveform Replication

Parameters

f_s	Sampling frequency
g	Gravitational acceleration
L	Length of signal segment
M	Number of overlapping segments
T	Total duration of vibration exposure
R_{eff}	Effective rolling radius of the tyre
m_1, m_2	Unsprung mass of the front and rear wheels
m_3	Sprung mass of the vehicle body
m_4	Mass of the driver seat
c_1, c_2	Damping coefficients of the front and rear suspension
c_s	Damping coefficient of the seat
k_1, k_2	Stiffness coefficients of the front and rear suspension
k_{t1}, k_{t2}	Structural vertical stiffness of the front and rear tyres
k_s	Stiffness coefficient of the seat
b_1, b_2	Distance from vehicle center of gravity to front and rear axles

b_3	Distance from vehicle center of gravity to the seat position
I_3	Pitch moment of inertia of the vehicle body
G_{brake}	Overall amplitude scale factor for braking feedback
p_{max}	Maximum observed brake pedal pressure (14000 kPa)
v_{max}	Maximum vehicle speed reference (100 km/h)
w_{vib}	Empirical mixing gain applied to the RPM engine vibration source
w_{road}	Empirical mixing gain applied to the road noise vibration source
μ	Mean point alignment parameter for the rumble strip Gaussian distribution
σ	Standard deviation parameter for the rumble strip Gaussian distribution

Variables

a_w	Weighted root mean square (RMS) acceleration
$a_w(t)$	Frequency-weighted acceleration signal
B	Accelerometer bias vector (3×1)
M	Accelerometer scale/misalignment matrix (3×3)
R	Combined rotation matrix
u_a, v, V	Vehicle forward velocity
$q(t)$	Stochastic road vertical displacement
$w(t)$	White noise signal
ϕ	Roll angle for tilt estimation
θ	Pitch angle for tilt estimation
ω	Angular velocity of the wheel
f_n	Excitation frequency for the n -th harmonic order
x_1, x_2	Vertical displacements of the front and rear unsprung masses
x_c	Vertical displacement of the vehicle body mass
α	Pitch angle displacement of the vehicle body / Crossfade mixing factor
x_s	Vertical displacement of the driver seat mass
x_{g1}, x_{g2}	Road profiles vertical displacement inputs to the front and rear tyres
A	System state-space matrix
B^T	Transposed input state-space matrix

u	Input vector in state-space representation
$x, \dot{x}$	State vector and its time derivative
$S_{xx}(f)$	Auto-spectral density of the input signal
$S_{xy}(f)$	Cross-spectral density between input and output signals
$\hat{G}(f)$	Frequency Response Function system model estimation
u_j	Updated control drive input signal for TWR iteration j
Q_j	Error convergence gain applied during TWR iteration j
r	Reference signal vector target
y_{j-1}	Measured output signal from the previous TWR iteration
v_{lo}	Nearest lower multiple of 10 of current speed
v_{hi}	Nearest higher multiple of 10 of current speed
p	Applied brake pedal pressure
g_{brake}	Dynamically computed scale gain for braking feedback
p_{tyre}	Position tracking tracking parameter of the tyre on the rumble strip
g_{pos}	Gaussian bell curve position tracking attenuation gain
f_{rumble}	Fundamental excitation frequency generated by a rumble strip
$x_{out}(t)$	Combined final haptic output signal mixed and routed to the shaker

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1

Introduction

1.1 Background

Driving simulators are efficient tools for research, allowing, for example, study of driver behavior, driver training, and vehicle development in a controlled environment. Physical testing is expensive, time consuming, and could be potentially dangerous, which makes realistic driving simulators a beneficial tool. One of the challenges of a realistic simulator is the "Motion Cueing" system, which replicates physical forces to provide the driver with sensory feedback. While moving-base platforms are able to reproduce low-frequency inertial forces such as vehicle acceleration, braking, and turning, they are most likely not able to simulate high-frequency vibrations needed for authenticity. These vibrations, primarily caused by road texture and the vehicle's engine, provide critical cues for speed perception and road texture. With these cues absent, drivers may experience a mismatch between sensory inputs, making the simulator less realistic, affecting driving behavior and in some cases cause motion sickness.

1.2 Purpose

The purpose of this thesis is to help VTI improve the realism of Sim IV driving simulator. By investigating different implementation strategies for road and engine vibration feedback in the driver seat in a driving simulator, the goal is to improve the realism, speed perception and over all driving experience. Better vibration feedback makes the simulator more realistic and immersive, which further helps with vehicle development, safety research, and studying how people drive. Making their virtual testing environment a more reliable and safe option compared to physical testing.

1.3 VTI Sim IV Driving Simulator

This project was conducted using the VTI Sim IV driving simulator, a moving-base simulator located at VTIs office in Gothenburg, shown in figure 1.1. It is used in studies related to road safety, vehicle dynamics, and external research projects. Sim IV was taken into operation in 2011 and is equipped with an eight degrees-of-freedom motion system and a visual system providing 180 degrees field of view [1]. The visual scene is generated by four 4K projectors and displayed on a curved projection

1. Introduction

screen. The motion system consists of a vehicle cabin mounted on a platform, which combines a hexapod with a large linear sled, enabling the reproduction of both rotational and translational motion cues. But these motion cues exist only in the lower frequency band, below 10 Hz. More vibrations are present in real driving, meaning there is a sensory gap between the Sim IV and reality.



Figure 1.1: Existing setup of VTI Sim IV.

It supports cabin exchange, which allows different vehicle cabins to be used depending on the purpose of the study. In this project, a Volvo XC60 car cabin was used. Sim IV can also be equipped with a Volvo FH16 (truck cabin) and a bicycle cabin. Figure 1.2 shows the interior of the simulator equipped with the Volvo XC60 cabin.



Figure 1.2: Interior of Sim IV with Volvo XC60 cabin

1.4 Goals

The objectives of this thesis are to:

- Identify the frequency range and vibration characteristics associated with road surface interaction and vehicle powertrain feedback on driver seat.
- Consider the spatial limitations and determine placement of actuators for effective vibration transmission to the driver.
- Develop a method to relate the input signal to the electrodynamic shakers with the seat measurements, enabling accurate amplitude control across the operating frequency range.
- Evaluate the performance of selected tactile actuators through modelling and experimental testing.

1.5 Limitations

The aim is to clearly establish the boundaries of the work.

The study includes:

- Identification of vibration requirements regarding frequency and amplitudes in the driver seat.
- Selection and evaluation of selected tactile actuators.
- Conceptual design for actuator placement and system integration in the simulator.
- Human subject studies for validating the effectiveness of the proposed solution.

The study does not include:

- Development of actuator hardware.
- Modifications to the simulator's visual or audio systems.
- Full vehicle dynamics modelling.

2

Theory

This chapter introduces the theoretical background needed to understand the implementation of a haptic seat actuator in the Sim IV. It first describes the motion system of the SimIV, followed by human perception of motion, vibration evaluation standards, frequency-domain analysis, motion cueing principles, and relevant vibration sources. Together, these topics provide the basis for understanding how seat-based haptic feedback can be used to complement the simulator’s motion platform and improve the perceived realism of vehicle motion.

2.1 Sim IV

2.1.1 Motion System

The Sim IV combines a sled with 2 DOF together with a hexapod with 6 DOF, resulting in an 8 DOF system, commonly referred to as a 6+2 DOF configuration. This because the sled has 2 DOF, surge and sway that overlaps with the hexapod, but the sled has significantly larger displacement range, allowing motions that would exceed the hexapod’s workspace. The motion system is able to reproduce motions up to 6 Hz. Table 2.1 below showcases the different acceleration, speeds and displacements the motion system is able to handle [1].

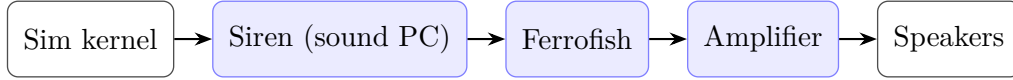
2.1.2 Audio Signal Architecture

In the current sound architecture, telemetry data is sent from the core simulation kernel (sim kernel) to a sound PC running Siren, an in-house built sound generation software based on OpenAL. Siren creates the real-time audio environment, including the sound of the ego vehicle and the surrounding traffic.

From the sound PC these audio signals are sent to a Ferrofisch A16 MK2 via a Multi Digital Audio Interface (MADI) connection, where all incoming audio channels are received as a multichannel digital stream. The Ferrofisch unit then converts these digital MADI channels into analog signals and routes them individually to its Tip Ring Sleeve (TRS) outputs.

These TRS outputs are connected to the Labgruppen C-Series power amplifiers, which drives all the existing sound units in the simulator. A flow chart of the audio architecture is illustrated in figure 2.1.

Motion	Acceleration ([g]) [m/s^2]	Velocity [m/s]	Position [m]
Surge	$\pm(0.65 g)$ 6.374	± 0.8	+0.408 -0.307
Sway	$\pm(0.60 g)$ 5.884	± 0.8	+0.318 -0.318
Heave	$\pm(0.60 g)$ 5.884	± 0.6	+0.261 -0.240
	[deg/s^2]	[deg/s]	[deg]
Roll	± 275	± 40.0	+16.5 -16.5
Pitch	± 275	± 40.0	+15.5 -16.0
Yaw	± 350	± 50.0	+20.5 -20.5
	([g]) [m/s^2]	[m/s]	[m]
X	$\pm(0.50 g)$ 4.903	± 2.0	± 2.5
Y	$\pm(0.50 g)$ 4.903	± 3.0	± 2.3

Table 2.1: The Specifications of the Sim IV Motion System.**Figure 2.1:** Data flow and software architecture of the mixed real-time audio stream.

2.2 Human Perception of Motion

The way humans perceive motion and vibration in a vehicle depends on several sensory systems working together. While visual and auditory cues provide important context, the most critical inputs come from the vestibular and somatosensory systems [2]. Together, these allow a driver to sense everything from a sharp turn to the subtle hum of the engine.

The vestibular system, located in the inner ear, essentially functions as the body's internal accelerometer. It uses semicircular canals, which are essential for maintaining balance and detecting rotational movement of the head, and otolith organs to sense linear acceleration and gravity [3]. This system is most effective at detecting low-frequency movements, such as the initial sensation of braking or cornering. In many cases, these physical cues work alongside what the driver sees to help them understand exactly how the vehicle is moving in space.

As motions increase in frequency, the body relies more on the somatosensory system. This involves a network of receptors in the skin, muscles, and joints that detect physical pressure and high-frequency vibrations. For example, sensors in the muscles provide feedback about body displacement during a turn, while receptors in the

skin pick up the higher-frequency "buzz" from the road surface or engine. In some cases, a driver might even hear a vibration through the car's structure before they consciously feel it through the seat.

Human sensitivity to these sensations is not uniform across all frequencies. When seated, the body is particularly sensitive to vertical vibrations in the 4-8 Hz range because this is where the spine and internal organs tend to resonate [2]. Understanding these specific sensitivities is vital when designing driving simulators. By focusing the simulator's haptic feedback on the frequency ranges where humans are most sensitive, it is possible to create a realistic sense of motion even if the hardware has physical limits. These biological characteristics lead directly into the international standards used to measure and evaluate vibration exposure.

2.3 Whole-Body Vibration Standards

The evaluation of human exposure to mechanical vibration is primarily governed by the **ISO 2631-1** standard. This framework provides the methodology for quantifying how vibrations affect human health, comfort, perception, and the incidence of motion sickness [4].

2.3.1 Frequency Weighting and Sensitivity

The human body exhibits non-linear, frequency-dependent sensitivity to vibration. ISO 2631-1 defines specific frequency-weighting curves to account for these variations: for vertical vibrations (z -axis), sensitivity peaks between 4-8 Hz due to resonance effects, while for horizontal vibrations (x and y axes), sensitivity is highest between 1-2 Hz. ISO 2631-1 defines the frequency range for health, comfort, and perception as 0.5–80 Hz, while the range associated with motion sickness is restricted to 0.1–0.5 Hz, corresponding to motions such as those commonly experienced on boats.

2.3.2 Evaluation Metrics

To quantify the severity of vibration exposure, the standard defines two primary metrics based on frequency-weighted acceleration $a_w(t)$:

2.3.2.1 Root Mean Square Acceleration

The weighted RMS acceleration (a_w) is the primary measure for steady-state vibration. It represents the average intensity over the duration of exposure T :

$$a_w = \left[\frac{1}{T} \int_0^T a_w^2(t) dt \right]^{\frac{1}{2}} \quad (2.1)$$

To determine if the basic evaluation method is suitable, a crest factor is used. The crest factor is defined as the modulus of the maximum instantaneous peak value of

the frequency-weighted acceleration signal to its RMS value[2].

$$\text{Crest factor} = \frac{\text{peak acceleration}}{\text{r.m.s acceleration}} \quad (2.2)$$

A normal value of the crest factor is seen to be between 3-6 but if the crest factor is less or equal then 9, the basic evaluation is seen as sufficient.

2.3.2.2 Vibration Dose Value

For vibration signals that are less periodic or contain shocks, the RMS value may underestimate the effects of the vibrations. In such cases, the *Vibration Dose Value* (VDV) is used. This is a cumulative measure that is more sensitive to peaks:

$$VDV = \left[\int_0^T a_w^4(t) dt \right]^{\frac{1}{4}} \quad (2.3)$$

2.4 Frequency Domain Analysis

2.4.1 Discrete Fourier Transform

The Discrete Fourier Transform (DFT), transforms a time-domain signal into its frequency-domain, which makes it possible to identify the frequencies. For a discrete signal $x[n]$ of length N , the DFT is defined as [5] :

$$X[k] = \sum_{n=0}^{N-1} x[n] e^{-j2\pi kn/N}, \quad k = 0, 1, \dots, N-1. \quad (2.4)$$

Computing the DFT directly from this equation requires a massive amount of calculations, which is highly inefficient for long measurements. To solve this, the Fast Fourier Transform (FFT) is utilized. The FFT is an efficient algorithm to compute the DFT, which significantly reduces the computational complexity of the operation, making it practical and fast to analyze large amounts of measured data [5].

2.4.2 Power Spectral Density

The *Power Spectral Density* (PSD) represents the signal power is distributed across different frequencies. According to [6], the power spectral density (PSD) of a discrete-time signal $x[n]$ is defined as:

$$S_x(e^{j\omega}) = \sum_{m=-\infty}^{\infty} R_x[m] e^{-j\omega m}, \quad (2.5)$$

where $R_x[m]$ is the autocorrelation sequence. Since this theoretical definition requires an infinite number of samples, the PSD must be estimated in practice. Unlike a standard FFT, which displays the frequency content summed over the entire measurement period, the PSD accounts for both amplitude and duration by normalizing with the observation time [7]. By representing power per unit frequency, the PSD

makes it possible to identify which frequency components contribute most to the overall signal power. In vibration analysis, this is particularly valuable for determining where vibration is concentrated in the frequency spectrum, thereby identifying dominant excitation sources and resonant behavior.

2.4.2.1 Welch Method

The Power Spectral Density (PSD) in this work is estimated using Welch's method [8], a technique designed to improve the reliability of spectral estimates compared to the standard periodogram. While the raw periodogram of a finite signal provides high frequency resolution, its variance does not decrease with increasing signal length, resulting in a "noisy" estimate. Welch addressed this by introducing a method based on time averaging over short, modified periodograms. The Welch PSD estimate is defined as:

$$\hat{P}_{xx}^{\text{Welch}}(f) = \frac{1}{M} \sum_{m=1}^M \left[\frac{1}{ULf_s} \left| \sum_{n=0}^{L-1} w[n]x_m[n]e^{-j2\pi fn/f_s} \right|^2 \right] \quad (2.6)$$

where M is the number of overlapping segments, L is the length of each segment, and f_s is the sampling frequency.

2.5 Driving Simulator Motion Cueing Principles

Accelerations affecting the body when driving is referred to as motion cues, these cues are generally generated by changes in linear and angular acceleration of the vehicle. In a real system, motion cues arise naturally from the interaction of the vehicle and the road [9]. For example, longitudinal acceleration occurs during acceleration and braking, lateral acceleration occurs when cornering and vertical accelerations occur based on the road roughness and the suspension. These physical accelerations provide essential feedback that helps the driver with the situational awareness and control of the vehicle. Driving platforms attempt to reproduce these sensations using motion platforms and tactile transducers. But due to the technical limitations, such as not being able to reproduce sustained acceleration by linear displacement, motion cue algorithms need to be utilized like tilt-coordination to simulate sustained accelerations through gravity.

2.6 Haptic Feedback in Driving Simulators

Haptic feedback can generally be split up into two different modalities: tactile and kinesthetic [10]. Tactile feedback is the tactile perception from the skin, such as vibrations. The kinesthetic feedback is referring to the forces perceived through our muscles, tendons and joints.

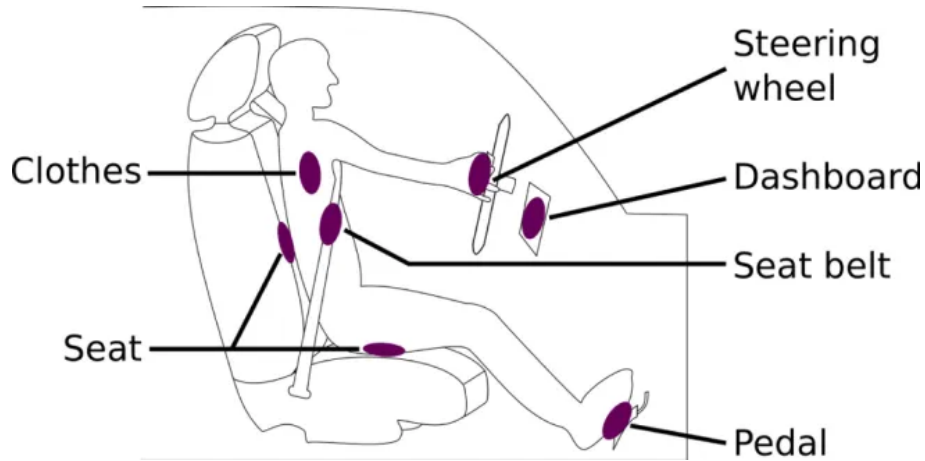


Figure 2.2: Areas for haptic feedback in a car [10]

Kinesthetic feedback, also referred to as force feedback, is mostly related to the steering, by adding a high-torque motor to the shaft of the steering wheel, the forces of a real car can be simulated. The tactile feedback is simulated using tactile transducers, also known as bass shakers. The transducers are normally placed on the floor or attached to the seat. These actuators translate audio-frequency telemetry into physical vibrations.

2.7 Vibration Sources

The vibrations to be reproduced in the simulator environment are primarily categorized into road-induced and engine-induced excitations. These sources provide the tactile cues necessary for a driver to perceive speed and the road surface.

2.7.1 Road-induced Vibrations

Road vibrations are perceived to be stochastic, originating from the vertical displacement of the tires as they interact with the road surface. The road roughness defines the basic vibration level, while the vehicle speed determines how strongly the vibrations are excited. According to Gillespie [11], road roughness acts as an acceleration input at the wheels whose amplitude increases proportionally to the square of the velocity (v^2). As a result of this quadratic relationship, an increase in speed not only transforms the spatial frequency of the road profile into higher temporal frequencies, but also exponentially amplifies the vibration magnitude.

2.7.1.1 Wheel hop resonance

Wheel hop refers to the vertical oscillation of the unsprung mass caused by the resonant interaction between the wheel and axle assembly and the combined stiffness of the tyre and suspension system [11]. The resonance frequency for wheel hop is around 10 Hz for passenger cars, and up to 15 Hz for heavier vehicles [12].

2.7.1.2 Fore–Aft Resonance

The fore–aft mode is a longitudinal oscillation of the wheel assembly relative to the vehicle body, caused by the flexibility of the suspension bushings in the longitudinal direction. Unlike wheel hop, which is a vertical motion, the fore–aft mode involves the wheel moving back and forth along the direction of travel.

This resonance typically occurs at frequencies in the range of 15–20 Hz for passenger cars [13]. Because the natural frequency is determined by the mechanical properties, it remains approximately constant regardless of vehicle speed.

2.7.1.3 Radial Force Variation

Tyre non-uniformities generate periodic vertical forces that vibrate the vehicle at multiples of the wheel rotational frequency. These non-uniformities stems from manufacturing variations such as belt splices and geometric ovalities [14, 11]. First-order vibrations ($n = 1$) are typically associated with mass imbalance, while second-order vibrations ($n = 2$) arise from radial force variation (RFV) and structural variations. The excitation frequency for the n -th order is:

$$f_n = n \cdot \frac{V}{2\pi R_{\text{eff}}}$$

where V is the vehicle speed and R_{eff} is the effective rolling radius. Since the frequency scales linearly with speed, these excitations are readily distinguishable from mechanical resonances, which remain fixed in frequency.

2.7.1.4 Belt Structural Modes

According to the theoretical frameworks provided by Pacejka [14] and Gillespie [11], the tyre exhibits fundamental vibration modes in the vertical, longitudinal, and yaw directions for frequencies below approximately 60 Hz.

These modes are governed by the structural and material properties of the tyre, including the mass of the belt and the effective stiffness arising from both the rubber structure and the internal air pressure. In particular, Gillespie describes the tyre belt as a vibrating mass supported by the stiffness of the sidewalls and internal pressure, where the natural frequencies are determined by the physical construction of the tyre.

As a result, these structural modes are expected to appear as speed-independent resonance frequencies in the vibration spectrum.

2.7.1.5 Suspension Isolation and Damping

Gillespie [11] describes hydraulic shock absorbers as non-linear components, where the damping force depends on piston velocity and valve characteristics. Small road irregularities encountered at higher speeds can therefore generate rapid suspension motions, which may increase the high-frequency vibrations. Because the chassis is too massive to react to these rapid inputs, it becomes dynamically isolated from the road in terms of rigid body motion. Consequently, distinct low-frequency body

motions are suppressed, and the resulting tactile feedback is primarily perceived as continuous buzzing sensation in the cabin.

2.7.2 Engine Vibrations

Engine vibrations are mostly linear and are primarily determined by the engine's speed (RPM). In vibration analysis, these frequencies are categorized into "orders," which represent multiples of the rotational speed of the engine. The 1st order corresponds exactly to the engine's RPM and is typically associated with the mechanical imbalance of rotating masses, such as the crankshaft or the drivetrain.

In a standard four-cylinder, four-stroke engine, two cylinders fire per revolution, making the 2nd order the dominant frequency for combustion-related vibrations [15]. For example, the frequency for the 2nd order vibration is calculated by multiplying the fundamental rotational frequency by the order number:

$$f_2 = 2 \cdot f_1 = 2 \cdot \left(\frac{\text{RPM}}{60} \right) \quad (2.7)$$

3

Methods

3.1 Vehicle modeling

The initial approach was to simulate how vertical vibrations are transferred from the road surface, through the vehicle, and to the driver seat. The model parameters were then intended to be tuned to match real measurements. The purpose of this approach was to obtain seat-vibration signals that could be used to generate a suitable input signal for the haptic seat actuator.

The half-car dynamic model was selected as the modeling approach, shown in figure 3.1. The model includes five states representing the vertical displacement of the front and rear wheels, vertical displacement of the vehicle body, pitch of the vehicle body, and the vertical displacement of the driver seat. The road excitations x_{g2} and x_{g1} are used as input signals to the rear and front wheels, respectively. To generate these signals, the method from [16] was used. In this method, the ISO 8608 road roughness classes are converted into time-domain road excitation signals using the filtering white noise method.

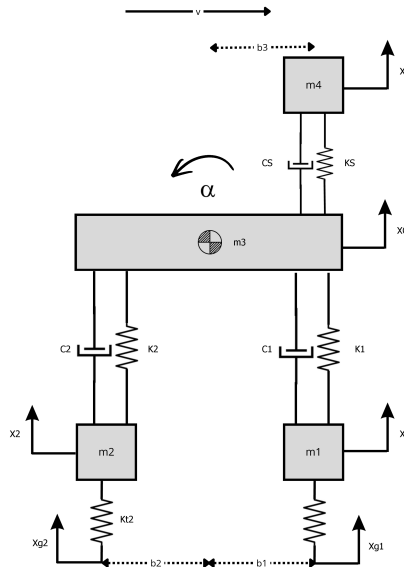


Figure 3.1: Half-car model with added seat (m4)

The state space representation of the model is given by:

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u}$$

$$\mathbf{x}^T = \begin{bmatrix} x_c & \dot{x}_c & \alpha & \dot{\alpha} & x_1 & \dot{x}_1 & x_2 & \dot{x}_2 & x_s & \dot{x}_s \end{bmatrix} \quad \mathbf{u} = \begin{bmatrix} x_{g1} \\ x_{g2} \end{bmatrix}$$

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -\frac{k_1+k_2+k_s}{m_3} & -\frac{c_1+c_2+c_s}{m_3} & \frac{k_2b_2-k_1b_1-k_sb_3}{m_3} & \frac{c_2b_2-c_1b_1-c_sb_3}{m_3} & \frac{k_1}{m_3} & \frac{c_1}{m_3} & \frac{k_2}{m_3} & \frac{c_2}{m_3} & \frac{k_s}{m_3} & \frac{c_s}{m_3} \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{k_2b_2-k_1b_1-k_sb_3}{I_3} & \frac{c_2b_2-c_1b_1-c_sb_3}{I_3} & -\frac{k_2b_2^2+k_1b_1^2+k_sb_3^2}{I_3} & -\frac{c_2b_2^2+c_1b_1^2+c_sb_3^2}{I_3} & \frac{k_1b_1}{I_3} & \frac{c_1b_1}{I_3} & -\frac{k_2b_2}{I_3} & -\frac{c_2b_2}{I_3} & \frac{k_sb_3}{I_3} & \frac{c_sb_3}{I_3} \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ \frac{k_1}{m_1} & \frac{c_1}{m_1} & \frac{k_1b_1}{m_1} & \frac{c_1b_1}{m_1} & -\frac{k_{t1}+k_1}{m_1} & -\frac{c_1}{m_1} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_2}{m_2} & \frac{c_2}{m_2} & -\frac{k_2b_2}{m_2} & -\frac{c_2b_2}{m_2} & 0 & 0 & -\frac{k_{t2}+k_2}{m_2} & -\frac{c_2}{m_2} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ \frac{k_s}{m_4} & \frac{c_s}{m_4} & \frac{k_sb_3}{m_4} & \frac{c_sb_3}{m_4} & 0 & 0 & 0 & 0 & -\frac{k_s}{m_4} & -\frac{c_s}{m_4} \end{bmatrix}$$

$$\mathbf{B}^T = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & \frac{k_{t1}}{m_1} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{k_{t2}}{m_2} & 0 & 0 & 0 \end{bmatrix} \quad (3.1)$$

All model parameters were initially copied from [17], and initialized in MATLAB. The model mainly produced vibration in low frequency bandwidth, and according to Gillespie [11], rigid-body motion is a characteristic of low-frequency vehicle dynamics, with upper limit of 10–15 Hz. Vibration above this frequency band requires modeling of structural flexible modes. Consequently, this type of model is unable to reproduce vehicle's vibration in the higher frequency range. Going forward, the model is not something that is used. The vibrations that are to be reproduced lies outside the span of which the model can simulate.

3.2 Field measurements

No existing measurements were found for a single vehicle across varying speeds that would enable analysis of the relationship between speed and vibration characteristics. Such measurement data was needed to provide a realistic reference. Therefore, measurements were conducted as part of this study. The measurement data is needed to capture the vibrations in the seat during different speeds, with the measurement data it was also possible to identify different sources of vibration.

3.2.1 Equipment

Seat vibrations were measured using an AdaFruit MMA8451 triple-axis accelerometer [18], interfaced with a Raspberry Pi for data acquisition. The accelerometer was run with a sampling rate of 500 Hz, for it to be able to measure confidently up to 250 Hz. The Raspberry Pi was remotely controlled from a PC via VNC, enabling convenient operation and monitoring during the measurements.

The accelerometer was mounted in two configurations depending on the measurement location. For measurements underneath the seat, the sensor was attached

using magnetic fixtures to ensure a rigid coupling between the sensor and the seat frame, shown in figure 3.3 and 3.4. For measurements on the seat surface, a small 3D-printed housing was designed to hold the accelerometer, seen in figure 3.2 allowing the driver to sit on top of it during the measurements.

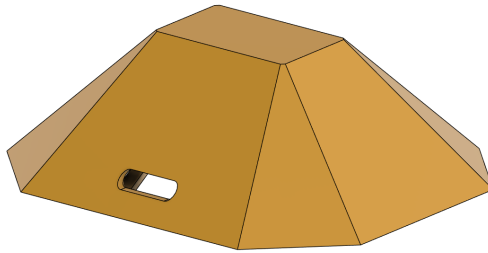


Figure 3.2: Housing for the accelerometer for measurement on seat



Figure 3.3: Accelerometer attached with magnets under Volvo EC40 driver seat

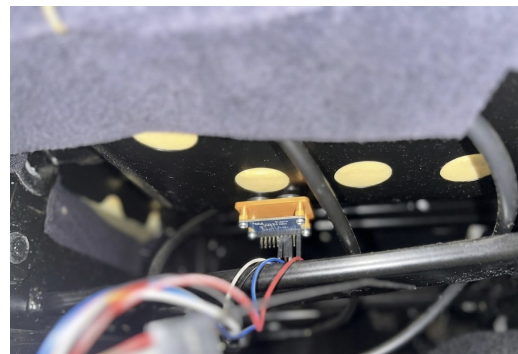


Figure 3.4: Accelerometer attached with magnets under Ford Escort driver seat

3.2.2 Accelerometer Calibration

3.2.2.1 Linear calibration

The accelerometer was calibrated to correct for sensor biases and scale and misalignment errors using a linear ellipsoid fitting method. The calibration is based on the assumption that when the IMU is stationary, the magnitude of the measured acceleration equals $1g$ according to [19].

$$1g = \sqrt{a_x^2 + a_y^2 + a_z^2} \quad (3.2)$$

By collecting measurements in multiple known orientations, the raw sensor data forms an ellipsoid due to bias and scaling errors. Using these measurements, calibration parameters can be estimated to transform the ellipsoid into a unit sphere. The measurements were processed using the calibration program Magneto [20], which estimates the bias vector $\mathbf{B}$ (3×1) and the scale and misalignment matrix $\mathbf{M}$ (3×3). This is consistent with the standard sensor error model, where the calibration is expressed as [19]:

$$\mathbf{A}_{\text{calibrated}} = \mathbf{M}(\mathbf{V}_{\text{raw}} - \mathbf{B}) \quad (3.3)$$

where $\mathbf{M}$ accounts for scale factors and axis misalignment, $\mathbf{B}$ represents the sensor bias, and $\mathbf{V}_{\text{raw}}$ represents the raw sensor measurements. This linear (affine) transformation compensates for systematic measurement errors and maps the distorted ellipsoid of raw data into a sphere of radius 1g.

3.2.2.2 Orientation calibration

The primary objective of the calibration is to ensure that acceleration measurements are consistent with respect to gravity, making them independent of the sensor's tilt (roll and pitch), regardless of whether the sensor is mounted perfectly straight or at a slight angle. To achieve this orientation-independent output, the sensor's static tilt relative to the Earth's gravity field is estimated. Then it is used to construct a rotation matrix that transforms the sensor's local coordinate system into a standardized global reference frame. In which the Z -axis is aligned with gravity. However, this transformation does not account for rotation around the gravity vector (yaw), which remains undefined when using accelerometer data alone. In the present application, this limitation is acceptable since only the Z -axis component is of interest, corresponding to the direction of the only applied vertical force described in section 3.3. When the accelerometer is at rest, it ideally measures only the gravitational acceleration $\vec{g}$, as defined in Equation (3.2). The distribution of $\vec{g}$ across the sensor's axes ($[a_x, a_y, a_z]^T$) allows for the estimation of the sensor's roll (ϕ) and pitch (θ) angles. Following the aerospace rotation sequence (R_{xyz}) described in [21], these angles are computed as:

$$\phi = \text{atan2}(a_y, a_z), \quad \theta = \text{atan2}(-a_x, \sqrt{a_y^2 + a_z^2}) \quad (3.4)$$

The aerospace rotation sequence defines a rotation matrix $R_{\text{local}}^{\text{global}}$ that transforms a vector from the global reference frame into the sensor's local frame. Using passive rotation matrices for the X -axis and Y -axis [21], this is expressed as:

$$R_{\text{local}}^{\text{global}} = R_x(\phi) \cdot R_y(\theta) \quad (3.5)$$

However, the desired transformation is the inverse, from the sensor's local frame to the global frame. Since rotation matrices are orthogonal, the inverse equals the transpose. Applying the matrix transpose identity $(A \cdot B)^T = B^T \cdot A^T$ gives:

$$R_{\text{global}}^{\text{local}} = (R_{\text{local}}^{\text{global}})^T = (R_x \cdot R_y)^T = R_y^T \cdot R_x^T \quad (3.6)$$

Evaluating the transposed matrices with the estimated roll and pitch angles gives:

$$R_x^T(\phi) = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \phi & -\sin \phi \\ 0 & \sin \phi & \cos \phi \end{bmatrix}, \quad R_y^T(\theta) = \begin{bmatrix} \cos \theta & 0 & \sin \theta \\ 0 & 1 & 0 \\ -\sin \theta & 0 & \cos \theta \end{bmatrix} \quad (3.7)$$

The combined rotation matrix R for the local-to-global transformation is therefore:

$$R = R_{global}^{local} = R_y^T \cdot R_x^T \quad (3.8)$$

This matrix is used to transform local sensor data ($\vec{v}_{local}$) into the global coordinate system ($\vec{v}_{global}$):

$$\vec{v}_{global} = R_{global}^{local} \cdot \vec{v}_{local} \quad (3.9)$$

This transformation ensures that acceleration measurements can be accurately analyzed in a gravity-aligned reference frame, regardless of the physical mounting angle of the sensor.

3.2.3 Measurement Procedure

The measurements were recorded in Gothenburg, Sweden, on motorway asphalt in order to capture representative excitation from typical highway driving conditions. To accurately distinguish between different vibration sources, two test vehicles were utilized: a traditional internal combustion engine (ICE) vehicle (Ford Escort 3-door 1.6 Manual, 90hp, 1994) and a battery electric vehicle (EV) (Volvo EC40, 250hp, 2025).

On the Ford vehicle, measurements were performed both beneath the seat and on top of the seat pad. In the electric vehicle, measurements were only performed underneath the seat. Due to the high noise of the accelerometer combined with the low amplitude vibrations measured on top of the seat, these were seen as not optimal for use. Forward in the report the only measurements referenced were from under the seat where the attachment was more rigid.

Including a pure electric vehicle in the test setup proved to be an important methodological step in order to isolate the road-induced vibrations. In the conventional internal combustion engine (ICE) vehicle, the measured vibration signal was dominated by excitations from the engine, making it difficult to separate engine harmonics from broadband vibrations caused by road roughness. In contrast, the electric vehicle, which lacks the periodic excitations associated with a combustion engine, provided a much cleaner baseline. This allowed the vibration response due to road input to be more clearly identified and used as a reference for comparison. Measurements were performed at speeds ranging from 10 to 100 km/h, with increments of 10 km/h.

3.2.4 Field measurement analysis

In this section, the aim is to identify the types of vibrations present in the measurements and determine their likely origins. By analyzing the frequency content and behaviour of the signals, the vibrations can be linked to their physical sources, providing insight into the vehicle's dynamic response for different inputs, and observe

measurement patterns. As also identified in [22], dominant harmonics occurred at very low frequencies around 12 Hz, and 18 Hz.

3.2.4.1 Wheel Hop

The highest peak in almost every PSD is located around 12 Hz, regardless of velocity. This was identified as wheel hop resonance (see section 2.7.1.1).

The measured data from the Volvo EC40 in Figure 3.5 reveals a clear correlation between the wheel's rotational frequency and the wheel hop resonance. Assuming a typical tyre circumference of approximately 2.26 m (rolling radius 0.359 m), the 1st order rotational frequency of the tyre coincides with the wheel hop natural frequency at speeds around 90–100 km/h, where the wheel rotates at approximately 12–14 Hz. At these speeds, the periodic excitation from the rotating wheel aligns with the natural frequency, driving the system into resonance. This effect is visually evident in Figure 3.5, where the marker size (proportional to PSD amplitude) grows progressively larger as the green line (1st order wheel rotation) approaches and crosses the shaded wheel hop band.

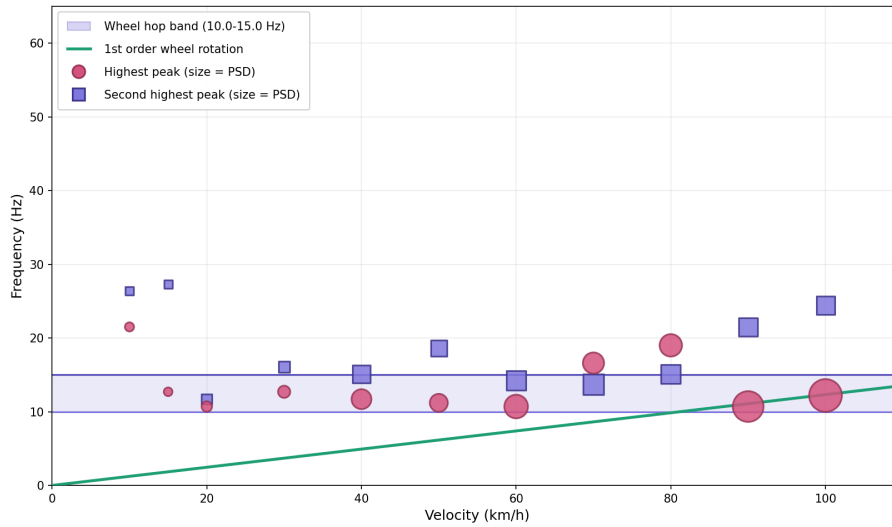


Figure 3.5: Shows correlation when the rotation of the wheel is approaching the resonance of the wheel hop, the vibration is amplified. Measurements from Volvo EC40

3.2.4.2 Fore aft vibration

A distinct peak between 15–18 Hz is visible at lower speeds, particularly at 30, 40, and 50 km/h, where it appears as a clear peak in the PSD. The centre frequency does not shift with speed, confirming that it originates from a mechanical resonance rather than a rotational order. This was identified as the fore–aft resonance (see section 2.7.1.2), which appears just above the wheel hop frequency range [13].

The peak is sharp and well-defined at low speeds but becomes progressively flatter as speed increases, which is most evident at 90 km/h (Figures 3.6 and 3.7). This behavior is consistent with the transition from discrete to overlapping road excitations: at low speeds, individual surface irregularities act as distinct inputs, allowing

the resonance to develop coherently. As speed increases, excitations overlap in time, reducing the system's ability to sustain a narrow band oscillation and resulting in a broader spectral peak.

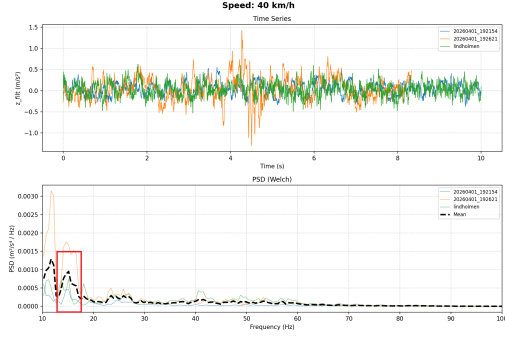


Figure 3.6: Peak below 20 Hz

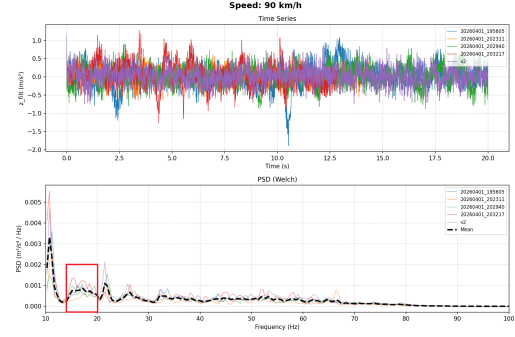


Figure 3.7: Peak above 20 Hz

3.2.4.3 Radial Force Variation

A clear vibration peak that increases linearly with velocity was observed in the measurements. The excitation first becomes apparent at approximately 50–60 km/h and reaches about 25 Hz at 100 km/h. The linear scaling with speed indicates a rotational order rather than a mechanical resonance. Following figures highlights the peak:

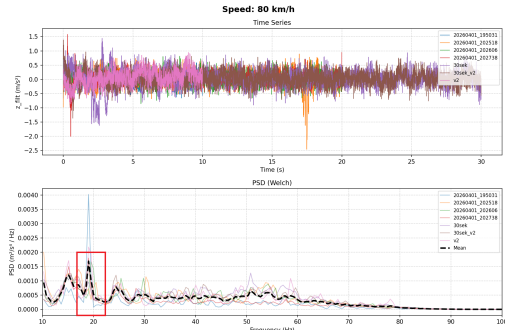


Figure 3.8: Peak observed at 80 km/h, Volvo EC40

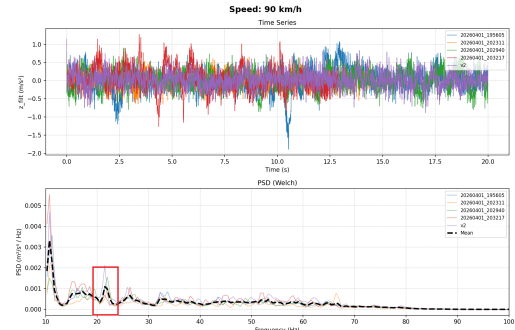


Figure 3.9: Peak observed at 90 km/h

For the Volvo EC40, equipped with either 235/50R19 or 255/45R19 tyres [23], the effective rolling radius is approximately $R_{\text{eff}} = 0.359$ m. Using the second-order frequency relation from Section 2.7.1.3, the predicted frequencies are $f_2 \approx 19.7$ Hz at 80 km/h and $f_2 \approx 22.2$ Hz at 90 km/h. These values agree well with the measured peaks shown in Figures 3.8 and 3.9, confirming the identification as radial force variation.

3.2.4.4 Belt Structural Modes

Tyre vibrations are known to influence the vehicle structure in the frequency range of 50–200 Hz [24]. This is consistent with the energy concentrations that were ob-

served in the 40–60 Hz range across all measurements. In contrast to the rotational orders, these frequencies remain approximately constant as vehicle speed increases, although their amplitude grows significantly with velocity. This behavior indicates structural resonances of the tyre rather than rotational effects. The observed frequency range aligns well with the theoretical prediction that tyre belt modes occur below approximately 60 Hz (Section 2.7.1.4), supporting the identification of these peaks as structural belt vibrations.

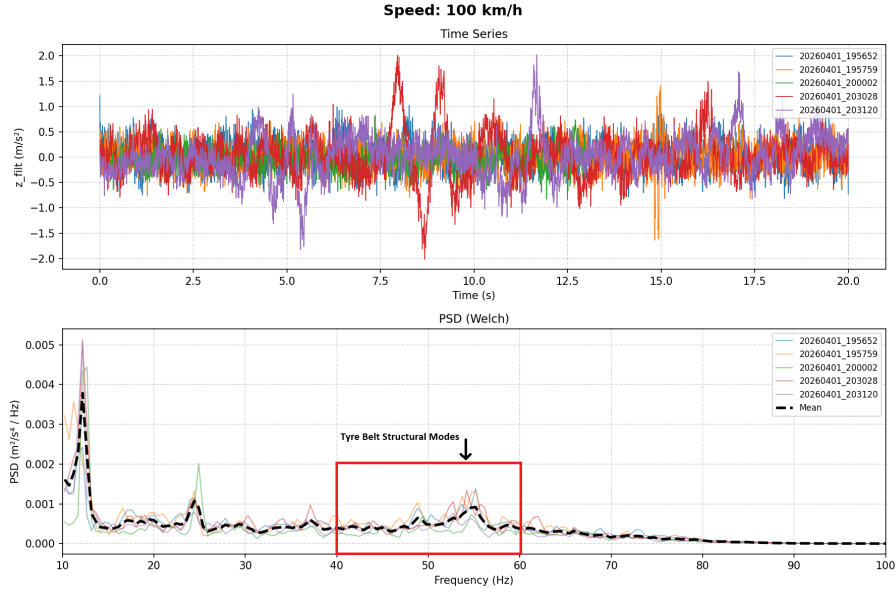


Figure 3.10: Tyre belt vibration observed at 100 km/h, Volvo EC40

The excitation of these modes is driven by road-induced inputs, which increase in magnitude with vehicle speed. As a result, the structural resonances are excited more strongly at higher velocities, leading to increased amplitude in the corresponding frequency bands. Since the natural frequencies are determined by the tyre’s physical properties, the peak locations remain unchanged while their magnitude increases, as shown in Figure 3.10.

3.2.4.5 Engine induced vibration

To identify which vibrations a combustion engine induces on the car seat, measurements were taken at various engine speeds to observe the correlation between RPM and seat vibration.

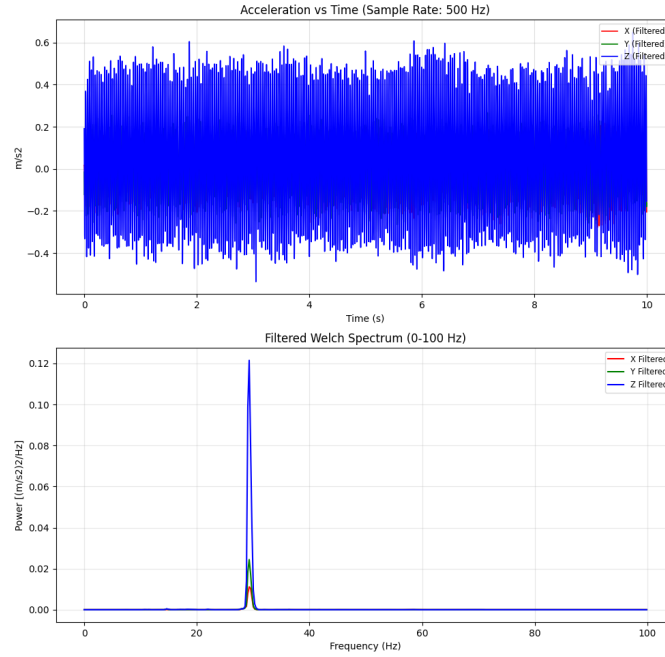


Figure 3.11: Idle engine vibrations at about 1000 RPM

Our measurements of the idle engine vibrations came out to something around 30 Hz, shown in figure 3.11. This lines up with the theory in section 2.7.2, because:

$$f = \frac{1000 \text{ rpm}}{60 \text{ s}} \times 2 = 33.3 \text{ Hz} \quad (3.10)$$

Additional RPM:s were recorded to investigate whether the first- and second-order engine vibration components appear in the measurements.

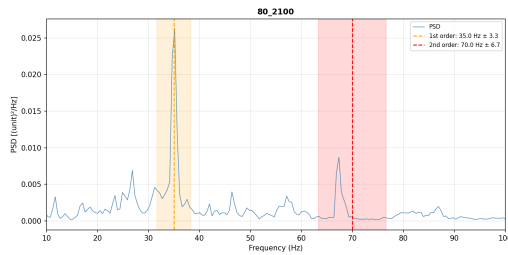


Figure 3.12: PSD at 80 km/h and 2100 rpm

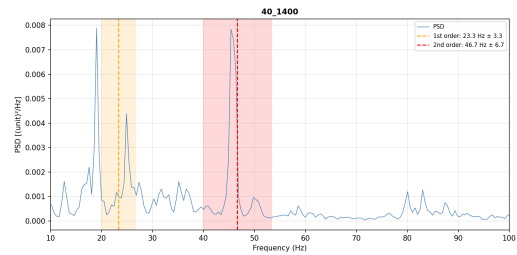


Figure 3.13: PSD at 40 km/h and 1400 rpm

The first- and second-order engine vibration frequencies corresponding to the measured engine RPM are indicated in the PSD plots. A ± 200 RPM interval is highlighted around each order to account for uncertainty in the estimated engine speed, since the RPM values were manually read from the RPM gauge. The observed behaviour is also consistent with the theory presented in Section 2.7.2.

3.3 Haptic Actuator Hardware

A tactile actuator was selected to produce the required seat vibrations. Two general mounting configurations were considered: a floor-mounted vibration platform and a seat-mounted actuator. The floor-mounted approach was ruled out due to the spatial constraints of the car cabin, insufficient force transmission through the seat structure, and the risk of exciting unintended resonances in surrounding panels. Mounting the actuator directly beneath the seat was identified as the more suitable solution, providing a compact and structurally direct excitation path. Although distributing multiple actuators across the seat would allow greater spatial control of vibration patterns, space constraints limited the design to a single unit. The primary constraint was the available vertical space under the seat, which demanded a compact and slim shaker. According to human perception of vibration [4], the vibration perception range lies within 0.5 - 80 Hz, the motion platform is already able to produce vibrations of up to 6 Hz. So the span the shaker is set to mainly cover is 10 - 80 Hz.



Figure 3.14: Xcite XBS100-4 [25].



Figure 3.15: Dayton Audio BST-300EX [26].

The two chosen tactile actuators are shown in Figure 3.14 and Figure 3.15. The Xcite XBS100-4 was selected based on its slim profile of 48 mm, peak force output of 324 N, and operational frequency range of 11 Hz - 1 kHz [25], which encompasses the vertical vibration frequencies relevant to automotive road excitation as identified in the literature review and filed measurements. A Dayton Audio BST-300EX was also acquired as a secondary unit to provide redundancy in case the primary actuator failed during testing. With a thickness of 63.5 mm, peak force output exceeding 300 N and frequency range of 10-200 Hz[26].

3.3.1 Mounting

The primary objective of the mounting strategy was to ensure maximum transmissibility of the identified frequencies. High-frequency vibrations can easily be dampened by yielding materials or compliant structures, therefore, the mounting required to be of a high mechanical stiffness. To achieve this, a custom mounting solution was developed. A metal plate was constructed to mount the shaker to the seat structure. The plate was designed to be compatible with both the XBS 100-4

and the BST-300EX. The plate was then permanently attached by welding it beneath the seat. Welding was preferred over other fastening methods to eliminate the risk of mechanical rattling or screws coming loose under continuous use. The finalized assembly is shown in Figure 3.16, where the rigid coupling ensures that the actuator's force is efficiently converted into vertical seat acceleration.

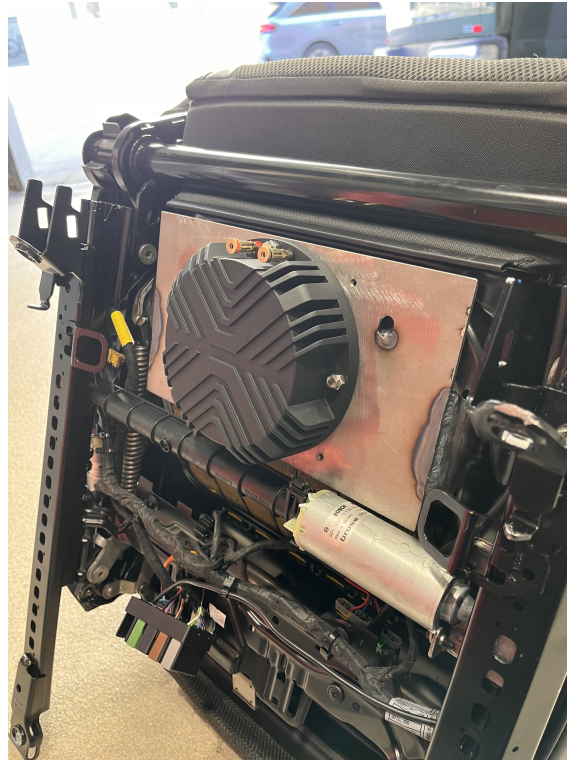


Figure 3.16: Xcite XBS100-4 rigidly mounted to the welded steel plate beneath the simulator seat

3.3.2 Audio

The car and truck cabin already includes buttkickers that are currently not in use. But these buttkickers never induced any vibrations into the seat, the transmissible energy was not sufficient. They are placed under the cabin and when running they produce more noise than vibrations. The cause for this is likely due to the fact that the cabin is seated on the platform metal to metal, and the vibration energy dissipates out into the entire platform. So, in this project, the existing audio signal cables routed from the amplifier to the simulator is used, effectively “hijacking” this pre-existing signal path at the amplifier stage rather than redesigning the upstream routing system.

Originally, the intention was to modify the existing software (siren) that sends the audio signal to the XBS100-4. However, this approach was not regarded as it was later seen to be too time consuming and complex within the project time slot.

Instead, we bypass the existing Ferrofisch processing chain by connecting a raspberry pi to an external sound card, Alesis iO4. The raspberry pi generates a digital signal, which is then converted to an analog TRS output by the sound card. This analog

signal is sent directly to the power amplifier, allowing us to inject our own signal while still using the existing cable route. Figure 3.17 illustrates the signal path used to integrate the seat actuator with the existing simulator hardware. To clarify there is only one signal that is used by the raspberry pi, all the other signals from the sound pc is still going into the amplifier.

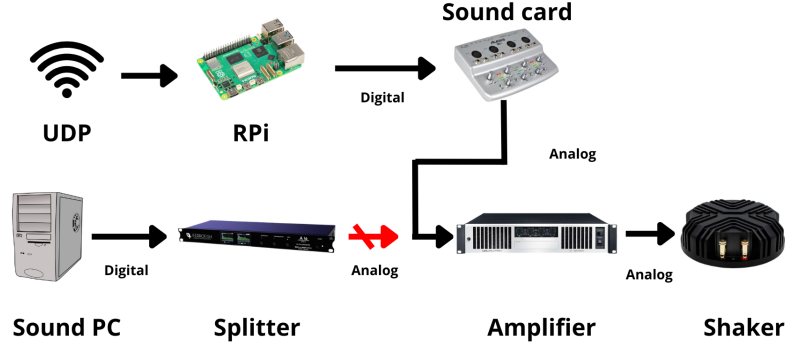


Figure 3.17: Schematic overview of the hardware architecture and signal paths for integrating the haptic seat actuator with the simulator system.

3.4 Signal Generation Process

3.4.1 System identification

The goal of the signal generation process is to produce a suitable input from the raspberry pi for the tactile actuator such that the resulting seat vibration matches a given reference acceleration signal measured in a real vehicle. In this case, the input signal is a WAV file, while the output signal corresponds to measurements obtained using an accelerometer mounted underneath the seat on the welded plate. Since both the input and output signals are originally recorded in the time domain, they must both be transformed into the frequency domain using DFT. The system identification was performed using Frequency Response Function (FRF), and is estimated as [27]:

$$H(f) = \frac{S_{xy}(f)}{S_{xx}(f)},$$

where $S_{xy}(f)$ is the cross-spectral density between the input signal $x(t)$ and the output signal $y(t)$, and $S_{xx}(f)$ is the auto-spectral density of the input signal. The following figure shows the system response when white noise is applied through the Xcite actuator:

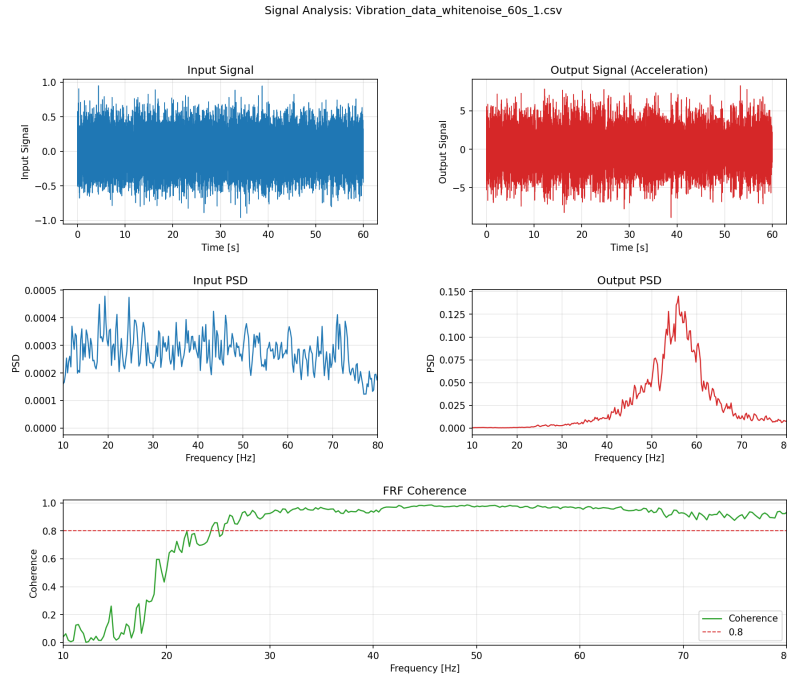


Figure 3.18: Comparison between the input signal and the output acceleration

From figure 3.18 it can be noted that even though the white noise is played evenly along the 10-80 Hz range, the actual vibrations in the seat are centralized with the most force around 60 Hz, suggesting that 60 Hz corresponds to a natural frequency of the system. It also shows that the coherence is low below 20 Hz. At these low frequencies, the excitation energy from the shaker is insufficient, and the accelerometers operate close to their lower measurement limit, causing the response to be dominated by noise. As a result, the FRF becomes unreliable below 20 Hz. Because of this, the half car model was not suitable for the final analysis.

3.4.2 Road Vibration

Time Waveform Replication (TWR) was used to reproduce the measured road induced vibration in different speeds. The TWR procedure is as follows [28]:

1. **FRF:** Calculate frequency response function, $\hat{G}(f)$, between the actuator input and the measured seat acceleration.
2. **Input Signal Estimation:** The established $\hat{G}(f)$ from FRF, is inverted and applied to the real road-vibration reference measurements to estimate the initial guess input signal required by the actuator.

$$u_0 = \hat{G}^{-1}r \quad (3.11)$$

3. **Iterative Error loop:** The drive signal is played back through the actuator, the response is measured, and the error relative to the target is used to update the drive signal. This loop is repeated until the error converges.

$$u_j = u_{j-1} + \hat{G}^{-1}Q_j[r - y_{j-1}] \quad (3.12)$$

Where $u_j(f)$ is the updated input signal vector in the frequency domain for iteration j , $u_{j-1}(f)$ is the frequency-domain input signal vector from the previous iteration, $\hat{G}^{-1}(f)$ is the inverted FRF matrix, Q_j is the diagonal error convergence gain matrix, $r(f)$ is the target reference vector in the frequency domain, and $y_{j-1}(f)$ is the measured output vector from the previous iteration in the frequency domain. For each iteration, the computed frequency-domain input signal is transformed into the time domain via an Inverse Fast Fourier Transform (IFFT). This time-domain signal is then played back through the actuator, enabling the resulting seat acceleration to be measured by the accelerometer and transformed back to the frequency domain to determine the error for the following loop

In this project the iterative matching was performed on the measured PSD rather than the time-domain waveform, as the perceived vibration comfort is primarily influenced by the spectral energy distribution rather than the exact time history. As there is low coherence below 20 Hz, regularization was applied by setting the inverse FRF gain to zero in this frequency range, limiting the control bandwidth to 20–80 Hz where the system model is reliable. The following figure shows the measured output PSD of the simulator seat for several TWR iterations. The blue line represents the target PSD measured in the Volvo EC40 during real road driving at 40 km/h.

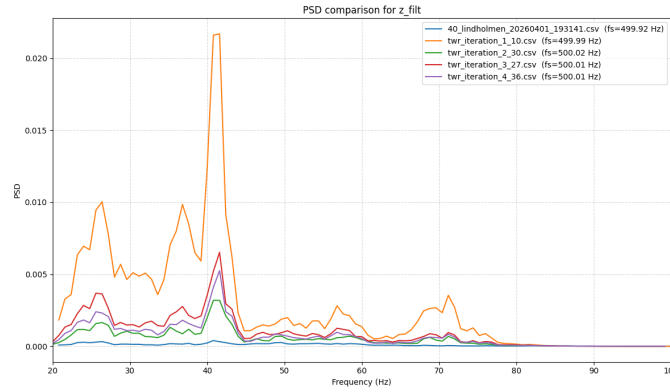


Figure 3.19: Measured output PSD under the simulator seat across four TWR iterations at 40 km/h compared with measurement from Volvo EC40

As seen in Figure 3.19, after each iteration the reproduced PSD progressively converges towards the target spectrum. Since the accelerometer is mounted directly on the actuator plate, the measured amplitudes are much larger at the source. In practice, the vibrations from the final iteration, were too weak to be recognized in the simulator. But by applying a gain factor of 3 to the target PSD, the spectral shape of the TWR followed the same structure, shown in the following figure:

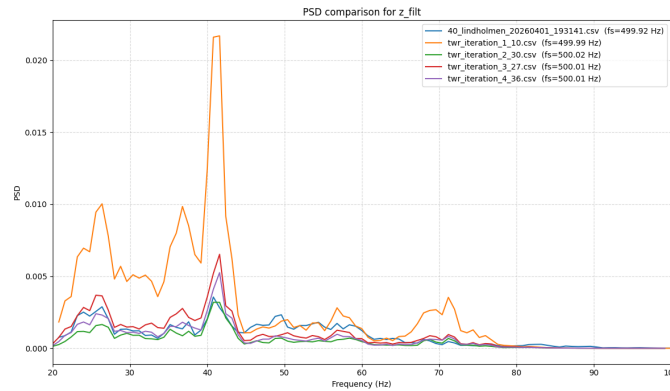


Figure 3.20: Measured output PSD on the simulator seat across four TWR iterations at 40 km/h compared with measurement from Volvo EC40 scaled by a factor of 3

This was done for all the recorded speeds between 10 and 100 km/h, with jumps of 10. From this we get 10 different wav-files which produces correct vibration structure for each speed. The original WAV-files did not produce a clear noticeable difference between speeds, a gain was added which is scaled quadratically with vehicle speed and normalised with respect to 100 km/h, based on Gillespie [11]. For speeds between the recorded speed levels, linear cross-fade is used. For velocities above the measured range, the same gain is applied to the WAV file corresponding to 100 km/h. But as the car increases in velocity, higher frequencies becomes more apparent, as described in section 2.7.1.5. To reflect this in the simulator, the low-frequency vibration were damped for speeds above 100 km/h.

3.4.3 Engine Vibration

From theory in section 2.7.2 and our own measurements, it suggests that the vibrations from the engine are predominantly the first- and second order vibrations of the rpm. To induce these vibrations in the seat, all that is required is the rpm from the simulator. In figure 3.22 the rpm was set to 2100, and as the formula states:

$$f_1 = \frac{2100 \text{ rpm}}{60 \text{ s}} = 35 \text{ Hz} \quad f_2 = \frac{2100 \text{ rpm}}{60 \text{ s}} \cdot 2 = 70 \text{ Hz}$$

Which is exactly what is happening in the rpm conversion, and as you can see in figure 3.21 this closely relates to the real measurements from the combustion engine vehicle.

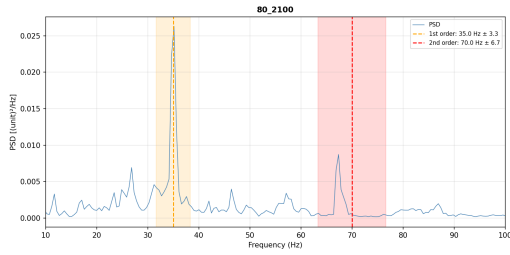


Figure 3.21: PSD at 80 km/h and 2100 rpm

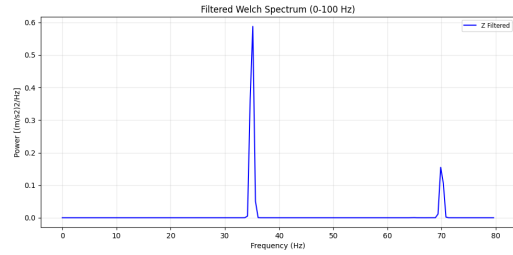


Figure 3.22: Measured under simulator seat when rpm is 2100

To increase the realism different RPMs need to have different amplitude in the signal. The gain for different RPMs were based on the measurements at first, the ratios between the peaks surrounding the first and second order were used. The field study were lacking in this department, as it was only measurements up until 2300 rpm, so everything above 2300 were estimated based on the earlier values. After the first baseline the gains were changed based on subjective opinions on how it was believed to be perceived.

3.4.4 Additional Vibrations

To add more inputs and add more realism there was also some additional vibrations that was included to be played by the shaker.

3.4.4.1 Braking

The previous motion cueing algorithm provides limited haptic feedback during braking events, with the exception being a slight pitch forward at the end of the braking event when coming to a standstill. This was apparent when testing the simulator. To account for this a braking vibration is synthesized based on the braking pressure applied and the speed of the vehicle.

The frequency of the vibration increases linearly with speed, with a span of 15 Hz at the lower speeds and up to 40 Hz at the higher speeds. This to keep the sensation within the tactile frequency band and ensure that the vibration feels proportional to the driving situation. The amplitude of the vibration is dependent on the brake pressure applied and a speed-scaling gain, faster driving and more force on the brake pedal will produce more vibrations.

Another aspect of the braking vibration was ABS activation. The brake pressure from the simulator was normalized, this to get a metric of how much the driver is pressing the brake pedal. A threshold was set on the normalized braking pressure signal to activate ABS pulsing. When the ABS is active, the cycle of brake application and release occurs roughly with a frequency of 15 Hz [29]. The threshold for activation was set to 80% of the maximum braking pressure, to ensure that the ABS only activates when the user exerts great force onto the pedal. But since the shaker isn't optimal on lower frequencies, instead of playing 15 Hz directly, the ABS is replicated by amplitude-modulating the existing braking with a frequency of 15 Hz, so in other words the gain is changed 15 times a second for the ABS. In figure 3.23 the braking can be seen in action during a test drive in the simulator.

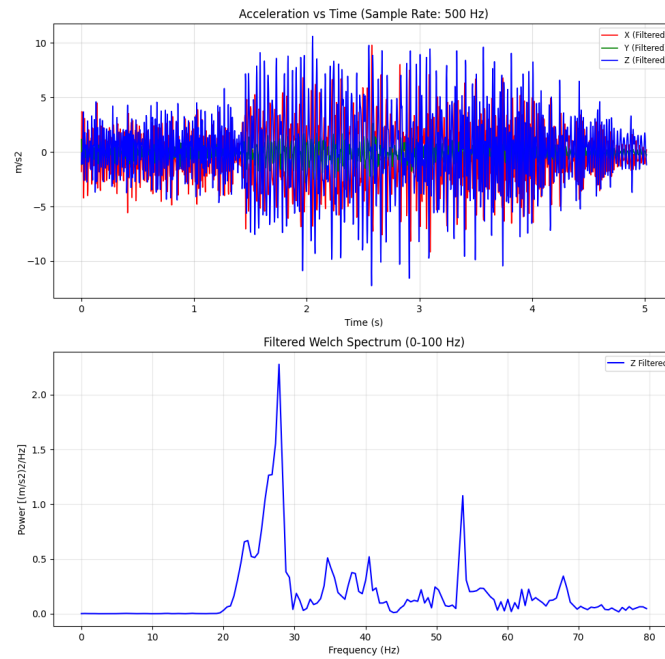


Figure 3.23: Measurement under the simulator seat during a braking event from 120 to 0

3.4.4.2 Rumble Strips

Rumble strips are grooves cut into the asphalt at the road shoulder to alert drivers that they are leaving the driving lane. To simulate the tactile warning, simulation telemetry is used to indicate when the vehicle tire is on the strip. Studies show that there are different distances between the grooves but for the simulator in this case the chosen distance was 0.3 m [30]. The frequency is dependent on the speed of the vehicle, there are three components that make up the vibration. Firstly there is the first order harmonic, which is just the speed divided by the distance between the strips.

It can be extracted from the simulation where the vehicle tire is positioned on the strip with a value from 0-100. Zero being the left edge of the patch, 100 the right edge of the patch laterally. Instead of just playing the same amplitude along the entirety of the strip, the amplitude of the signal is changed using a bell curve distribution. Figure 3.24 depicts the measurement from the simulator of a maneuver of driving in 80 km/h, driving over the rumble strip from the left, and then back onto the road.

3. Methods

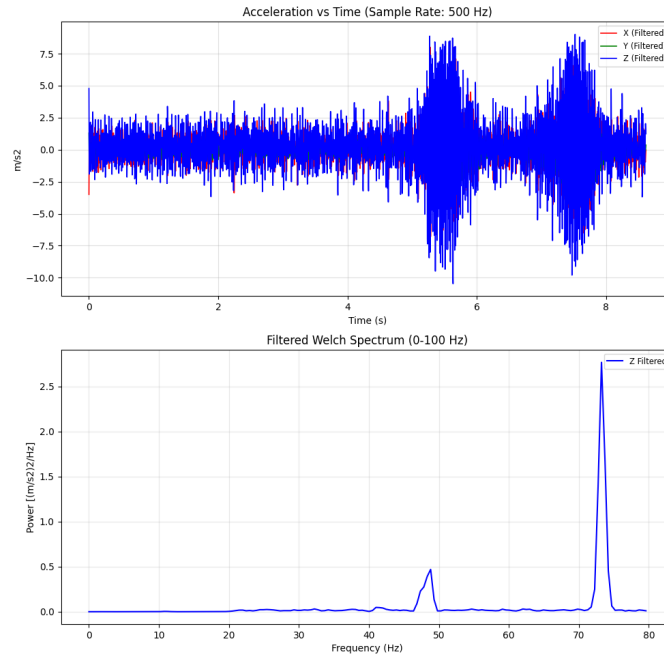


Figure 3.24: Simulator seat measurement of driving 80 km/h and driving on the rumble strip

The peak frequency observed in the measurement is approximately 74 Hz, which aligns with the expected output. At 80 km/h the vehicle speed is approximately 22.2 m/s, the predicted frequency is given by:

$$f = \frac{22.2}{0.3} \approx 74 \text{ Hz} \quad (3.13)$$

which is consistent with the measured result.

4

Results

4.1 Algorithm

The complete algorithm was implemented in Python.

4.1.1 UDP

Information about the simulator and the simulation is extracted using UDP. In table 4.1 below the structure of the UDP-packet that is transmitted can be viewed. The variables are then used in the vibration generation script to produce the correct vibrations for the correct situation.

Variable Name	Description	Type	Unit/Format
iteration	Index of the packet being sent	int	–
v	Vehicle velocity	int	m/s
engine_rps	Engine rotational speed	double	rad/s
brake_pedal_pressure_kpa	Pressure applied to the brake pedal	double	kPa
engine_status	If engine is on or off	double	State
shakerON	Flag to determine if vibrations should be activated	bool	Boolean
activateRumbleStrip	Flag that activates when car is positioned on the rumble strip	bool	Boolean
tyrePos_on_rumblePatch	Position tracking value of the tire on the rumble strip	double	0 – 100

Table 4.1: The structure of the UDP-packet being sent

4.1.2 Structure

Following flowchart shows the structure of the code that receives all necessary parameters from the simulator, updates the shared system state, and sends the corresponding vibration signal to the shaker.

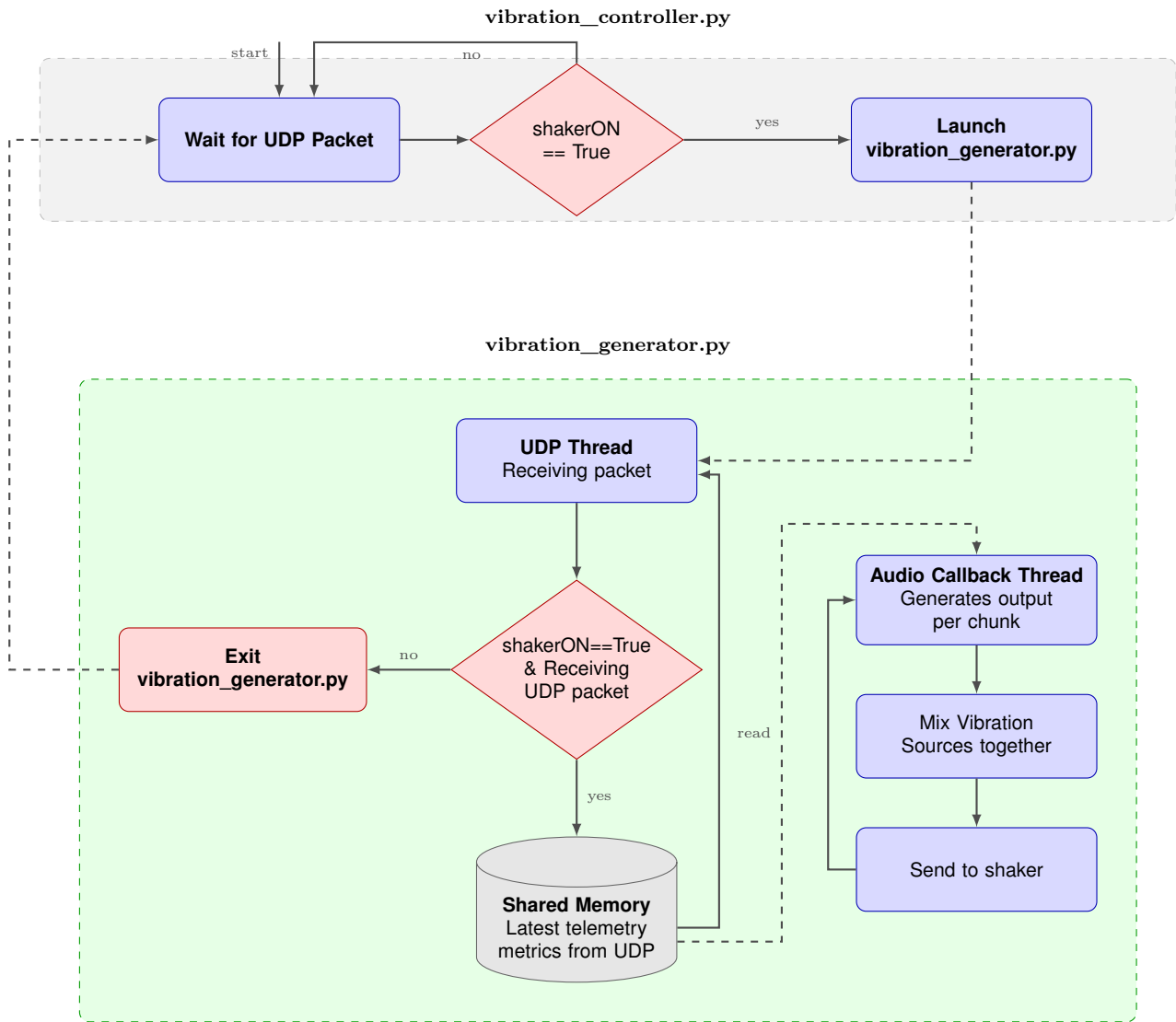


Figure 4.1: Flowchart to describe the flow of the algorithm

4.1.3 Vibration generation and mixing

The vibration output consists of three parallel generator classes, each responsible for a set of vibrations. All three are driven by the UDP listener thread, and their outputs are summed every audio block inside a real-time callback. Figure 4.2 provides an overview of the signal routing from the telemetry packet to the seat shaker.

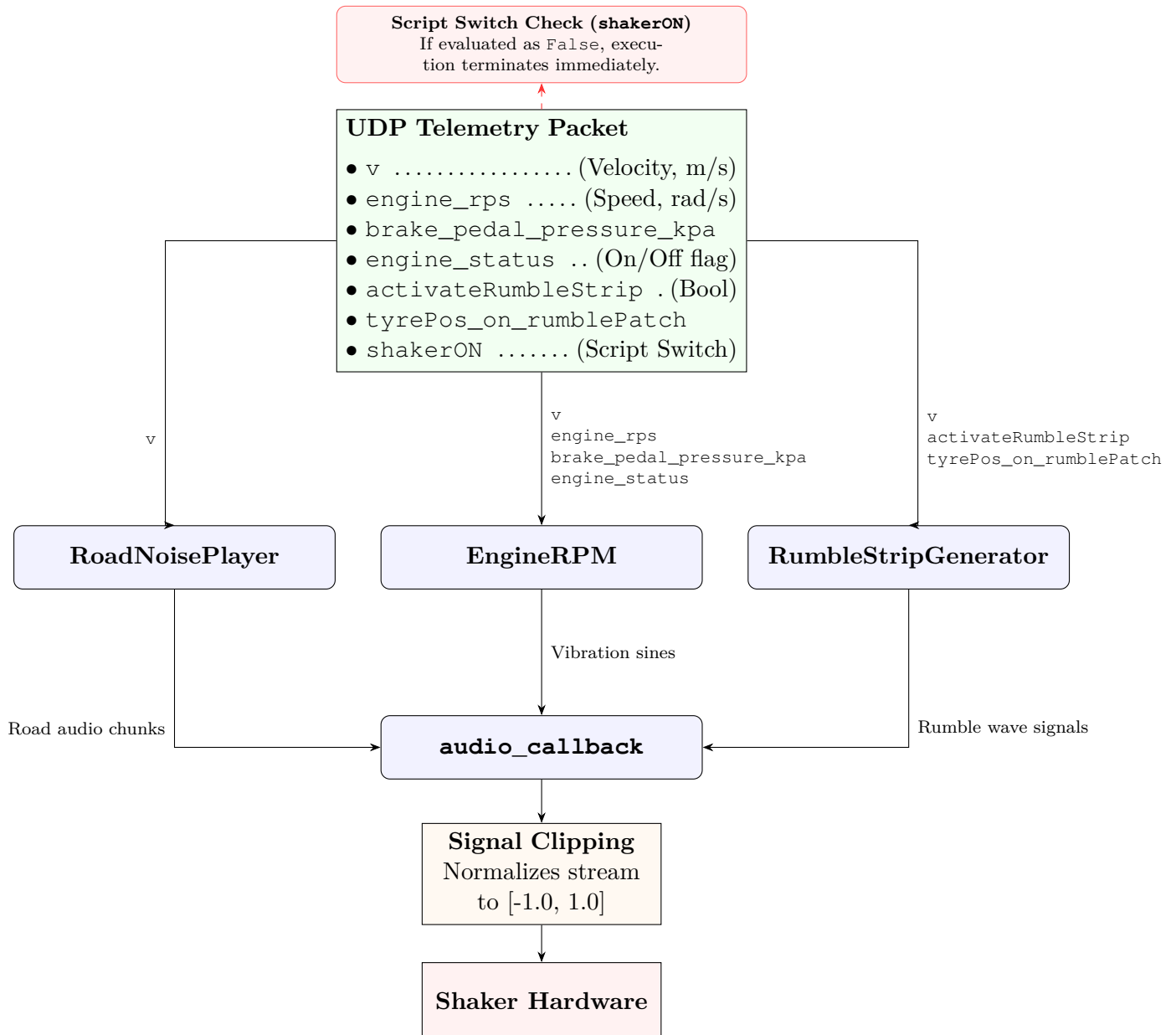


Figure 4.2: Flowchart mapping the distribution and processing loop of UDP telemetry variables into haptic cues.

The RoadNoisePlayer, RPMGenerator, and RumbleStripGenerator each generate audio using a fixed chunk size of 2,048 samples. These chunks are generated simultaneously and mixed into a single audio stream during each callback cycle. Operating at a standard sampling rate of 44,100 Hz, a chunk size of 2,048 samples corresponds to a duration of approximately 46.4 ms per buffer ($\frac{2,048}{44,100}$). This chunk size was chosen as an optimal tradeoff. It ensures low latency for real-time responsiveness while maintaining CPU efficiency on limited hardware, like the Raspberry Pi. By processing 2,048 samples at a time, the audio callback runs about 21.5 times per second, which keeps the CPU workload down. The chunk size of 2,048 was also chosen because it is a power of two, which makes internal signal processing (FFT, filtering, and interpolation) much more efficient. Finally, the synchronized generation of all

three audio sources within each callback ensures perfect alignment when mixing the vibration components, resulting in a smooth and distortion-free combined output to the shaker.

4.1.3.1 RoadNoisePlayer

The method for calculating this can be seen in section 3.4.2. But in short, on each audio chunk, `RoadNoisePlayer` reads the velocity v (converted to km/h), identifies the two nearest recordings, and crossfades between them:

$$x_{\text{road}}(t) = (1 - \alpha) x_{v_{\text{lo}}}(t) + \alpha x_{v_{\text{hi}}}(t), \quad \alpha = \frac{v - v_{\text{lo}}}{v_{\text{hi}} - v_{\text{lo}}} \quad (4.1)$$

4.1.3.2 EngineRPMGenerator

This class synthesises engine and braking vibrations. How engine vibrations and braking vibrations are introduced can be seen in sections 3.4.3 and 3.4.4.1. On each audio chunk, it reads the `engine_rps` field (rad/s), converts it to RPM, and looks up the first- and second-order harmonic gains from a CSV table. Two phase-continuous sine waves, meaning the phase is tracked and carried over between audio chunks to ensure a seamless signal without discontinuities, are generated and weighted by their respective harmonic gains. These are then summed to produce the final engine vibration signal, according to the relationship::

$$f_n = n \cdot \frac{\text{RPM}}{60}, \quad n \in \{1, 2\} \quad (4.2)$$

The brake vibration is decided by:

$$f_{\text{brake}} = f_{\text{min}} + \frac{v}{v_{\text{max}}} \cdot (f_{\text{max}} - f_{\text{min}}), \quad f_{\text{min}} = 15.0 \text{ Hz}, \quad f_{\text{max}} = 40.0 \text{ Hz}$$

The current speed of the simulator is v and v_{max} is 200. Gain of brake vibration is derived from `brake_pedal_pressure`, normalized against the observed maximum of 14 000 kPa, and scaled by a speed-dependent factor:

$$g_{\text{brake}} = \frac{p}{p_{\text{max}}} \cdot \left(\frac{v}{v_{\text{max}}} \right)^{0.5} \cdot G_{\text{brake}} \quad (4.3)$$

where p is the applied brake pressure, v is the vehicle speed, and $G_{\text{brake}} = 3.5$ is an overall amplitude scale factor, subjectively chosen upon feeling. This formulation ensures that a given pedal force produces stronger feedback at higher speeds than at lower speeds. The `engine_status` flag acts as a gate: if the engine is off, the generator bypasses synthesis and returns a block of zeros.

4.1.3.3 RumbleStripGenerator

This class plays the tactile feedback of when a tyre crosses a rumble strip on the road-shoulder. Also further explained in 3.4.4.2. It is activated by the boolean

`activateRumbleStrip` flag and modulates its output amplitude through a Gaussian bell curve centered on the midpoint of the strip ($\mu = 50$, $\sigma = 25$), computed from the tracking parameter `tire_pos_on_rumble_patch` (valued 0 to 100).

$$g_{\text{pos}} = \exp\left(-\frac{1}{2} \left(\frac{p_{\text{tyre}} - 50}{25}\right)^2\right) \quad (4.4)$$

With these bell curve parameters, it is rather wide and produces a smooth entry onto the rumblestrip. The fundamental frequency is derived dynamically from vehicle speed and the structural groove pitch of 0.3 m:

$$f_{\text{rumble}} = \frac{v \text{ [m/s]}}{0.3 \text{ [m]}} \quad (4.5)$$

4.1.3.4 `audio_callback` and Mixing

The `audio_callback` function is generating the final signal that is sent to the shaker. It is called repeatedly by the `sounddevice` output stream, once for every block of 2048 samples, at a sampling frequency of 44 100 Hz. Which means that one chunk (2048 samples) corresponds to a time window of approximately 46.4 ms.

For each block, the callback requests a new signal from each vibration source, the RPM generator, the road-noise player, and the rumble-strip generator. These signals are then scaled and added together to create one combined output signal:

$$x_{\text{out}}(t) = \text{clip}(w_{\text{vib}}x_{\text{RPM}}(t) + w_{\text{road}}x_{\text{road}}(t) + x_{\text{rumble}}(t), -1, 1) \quad (4.6)$$

where $w_{\text{vib}} = 0.6$ and $w_{\text{road}} = 5.0$ are empirically chosen mix gains.

The final signal is clipped to the range $[-1, 1]$ before playback. Clipping prevents the output values from exceeding the allowed range, but too much gain can still cause distortion.

4.2 Validation

To validate the implemented vibrations, a user study was conducted to see if the implemented vibrations would affect the realism. A total of eight participants took part in the study, including one female. The participants were between 21 and 56 years old. Due to the limited sample size, the result should be taken with a grain of salt, as it is not enough to draw statistically reliable conclusions. Individual differences have a disproportionate influence on the averages, meaning a single outlier can change the data considerably. Future work should aim to recruit more participants and do a more in-depth study. The questionnaire used is presented in Appendix A. Table 4.2 shows how the procedure of the driving test would look like for a participant that started without vibrations. Between participants the order of which condition started alternated, this was tracked with participant ID. If the number was uneven vibrations were off at the start and vice versa.

Table 4.2: Driving test procedure

4. Results

Description	Shaker	Speedometer	Traffic	Driving Time	Driving Objective
Trial run 1	OFF	ON	ON	5-10 mins	Drive normally as one may wish. Get a feel for the simulator under current conditions.
Fill questionnaire for Trial run 1	-	-	-	-	
Trial run 2	ON	ON	ON	5-10 mins	Drive normally as one may wish. Get a feel for the simulator under changed conditions.
Fill questionnaire for Trial run 2	-	-	-	-	
Test run 1	OFF	OFF	OFF	5 mins	Objective test: The driver will drive in three different speeds, 0-70 km/h, 0-110 km/h and 0-50 km/h. Between each speed test the driver will brake down to a standstill. The driver will notify when they think they are driving the desired speed and it will be noted down.
Test run 2	ON	OFF	OFF	5 mins	Objective test: The driver will drive in three different speeds, 0-70 km/h, 0-110 km/h and 0-50 km/h. Between each speed test the driver will brake down to a standstill. The driver will notify when they think they are driving the desired speed and it will be noted down.
Fill rest of questionnaire	-	-	-	-	Complete subjective feedback evaluation.

To minimize participant bias, information regarding the specific system modifications was withheld. Prior to the experiment, participants were informed that they would perform a speed test, but they were kept blind to the exact velocities. This to prevent any prior practice and ensuring that participants relied purely on their direct perception and intuition.

4.2.1 Study Results

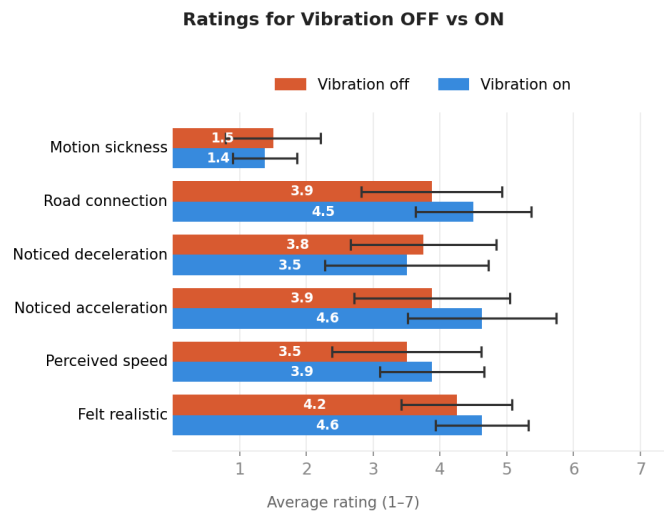


Figure 4.3: Comparison between vibration ON vs OFF (1=Strongly Disagree, 7=Strongly Agree)

In figure 4.3 the comparison between how the participants perceived the simulator can be viewed. This data suggests that in general the implementation of the vibrations had a positive influence on the feeling of the simulator. Judging from the data the area it mostly enhanced was the feeling of the acceleration of the vehicle.

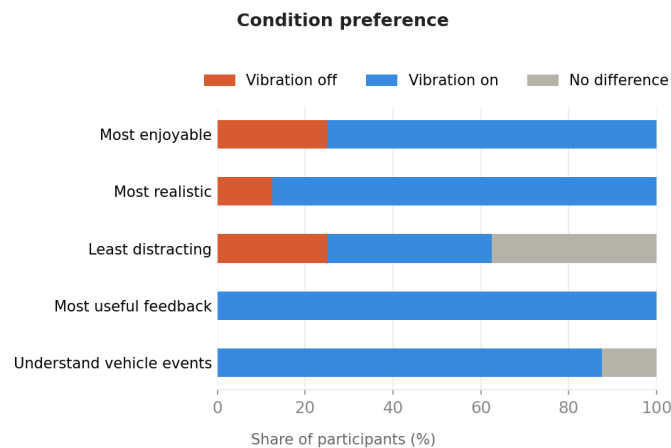


Figure 4.4: Preference between vibration ON vs OFF (1=Strongly Disagree, 7=Strongly Agree)

In figure 4.4 the participants were asked to select the preferred condition based on certain statements. Here it is suggested that having vibrations on were the condition that was mostly preferred.

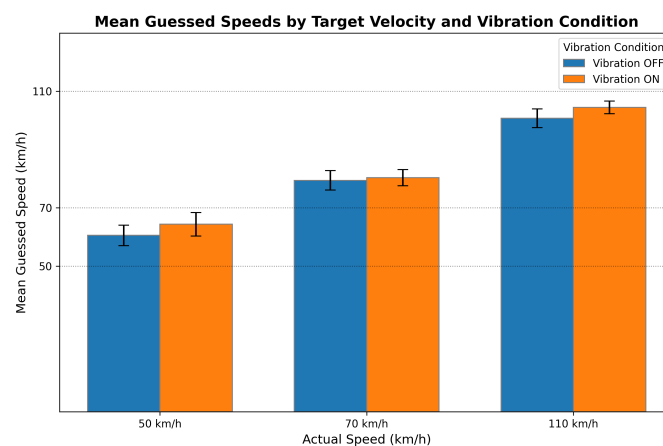


Figure 4.5: Guessed speeds with Vibration ON and OFF

In figure 4.5 the mean guesses during speed test can be viewed. Across all three speeds participants consistently misjudged their speed. Participants themselves noted that the task was challenging, with several describing a tendency to try to use the visual cues to estimate speed. Although the increase in perceived speed was small, the result suggests that seat vibrations may influence how participants judge vehicle speed.

It could be expected that, for the same visual speed, the presence of seat vibrations would lead participants to drive slower, while the absence of vibrations would lead to them to drive faster. However, the data visualize the opposite.

4.2.2 User Feedback

After the simulator tests, participants were asked a small number of open-ended questions regarding the perceived vibration feedback. The purpose was not to perform a formal user study, but to collect general comments about realism, immersion, and possible improvements. A summary of the responses is presented in Table 4.3.

Table 4.3: Summary of participant feedback from the simulator evaluation.

Question	Summary of responses
What differences did you notice between the two conditions?	In general the feeling off speed and acceleration felt better.
Did the vibrations help you understand any specific vehicle events?	Participants mainly mentioned speed changes, acceleration, braking, and road feel.
Were there any vibrations that felt unnecessary, distracting, or poorly timed?	Some participants felt the idle and low speeds vibration were too harsh.
Any additional comments or suggestions?	Better brake pedal feedback, more varied road scenarios, improved deceleration tuning, and additional vibration feedback in the backrest.

5

Conclusion

The aim of this thesis was to implement a tactile transducer to produce vibration feedback in a driving simulator to increase the immersiveness. The result shows that the Time Waveform Replication method is able to recreate road-induced vibrations and engine vibrations can be created directly from simulated vehicle's engine rotation. The user study suggests that the vibration feedback overall improved the perceived sensation of driving, but also showed that the perception of the vibrations varied between participants. As this is a perceptual system, the experienced realism is not determined by the vibration feedback alone. The simulator model, visual environment, audio feedback, motion cueing, and the sensitivity of the driver, everything contributes to the overall driving experience. Several stated that the vibrations in general helped, but the vibrations alone is not enough, other parts would still require more attention.

The biggest limitation of the implemented system was the low coherence below 20 Hz, which was shown in the field measurements to be the frequency range where most of the vibration energy was concentrated. If these low-frequency vibrations could be reproduced, the perceived realism experience would likely improve.

5.1 Future Work

Based on the limitations identified in this work and the feedback received from participants, several directions for future work have been identified. The initial step is to remove the Raspberry Pi and get the vibration script integrated directly into the existing system. Since the vibrations now are limited to above 20 Hz, there is a gap in where there exists no vibrations between the motion platform and up to 20 Hz. This was also the area where most of the energy was located according to the field measurements. Missing this area can greatly reduce the perception of realism in the simulator. Related to this, the vibration amplitudes should be tuned and different vibration profiles for different driving conditions investigated. Now it is not possible to switch between how rough the road is, being able to change vibrations depending on friction and road type could help with realism. The road vibration generation could become more modular, instead of playing different WAV-files and crossfading. Investigating playing white-noise directly and doing a relevant transfer function depending on the speed will make it more stochastic and not as hard-coded.

Additionally, alternative actuator placements should be explored, such as in the backrest or at multiple positions within the seat, in order to create more spatially

detailed haptic feedback. To get the most realistic vibrations, actuators should vibrate the entire cabin so the vibrations are transferred into the seat more natural. But this would require very strong actuators and the cabin would most likely need to be "disconnected" from the rest of the platform. Since this has already been tested but unsuccessfully. Finally, the vehicle model should be improved, as several participants noted that the car behaved unnaturally in response to driver input. For instance, there is no special response when exerting max force on the gas pedal, in a real automatic vehicle it would downshift and be more aggressive. And several people described the car as slow and unresponsive.

Bibliography

- [1] J. Jansson, J. Sandin, B. Augusto, M. Fischer, B. Blissing, and L. Källgren. *Design and Performance of the VTI Sim IV*. Proceedings of the Driving Simulation Conference 2014, Paris, France, 2014. Available at: https://www.researchgate.net/publication/266686614_Design_and_performance_of_the_VTI_Sim_IV (accessed May 19, 2026).
- [2] Griffin, M. J. (1990) Handbook of Human Vibration. Academic Press.
- [3] Bruck, L., Haycock, B., and Emadi, A. A Review of Driving Simulation Technology and Applications. *IEEE Open Journal of Vehicular Technology*, vol. 2, pp. 1–16, 2020. Available at: <https://ieeexplore.ieee.org/document/9250565> (accessed March 10, 2026).
- [4] International Organization for Standardization (1997) ISO 2631-1: Mechanical vibration and shock - Evaluation of human exposure to whole-body vibration - Part 1: General requirements. ISO, Geneva.
- [5] Kong, Q., Siau, T., and Bayen, A. Python Programming and Numerical Methods: A Guide for Engineers and Scientists – Fast Fourier Transform (FFT). *Python Numerical Methods (UC Berkeley)*, 2020. Available at: <https://pythonnumericalmethods.studentorg.berkeley.edu/notebooks/chapter24.03-Fast-Fourier-Transform.html> (accessed April 8, 2026).
- [6] GeeksforGeeks. Power Spectral Density. *GeeksforGeeks*. Available at: <https://www.geeksforgeeks.org/electronics-engineering/power-spectral-density/> (accessed April 9, 2026).
- [7] Vibration Research. *FFT vs PSD: What's the Difference?*, 2025. Available at: <https://vibrationresearch.com/blog/fft-psd-difference/> (accessed April 8, 2026).
- [8] Welch, P. The use of the fast Fourier transform for the estimation of power spectra: A method based on time averaging over short, modified periodograms. *IEEE Trans. Audio Electroacoust.*, vol. 15, pp. 70-73, 1967. Available at: <https://ieeexplore.ieee.org/document/1161901> (accessed April 9, 2026).
- [9] Technical University of Munich (TUM). Motion Cueing Algorithms. Available at: <https://www.mec.ed.tum.de/en/am/research/completed-projects/motion-cueing/> (accessed March 10, 2026).
- [10] Petermeijer, S. M., Abbink, D. A., Mulder, M., and de Winter, J. C. F. The use of haptic and tactile information in the car to improve driving safety: A review of current technologies. *Frontiers in ICT*, vol. 5, 2018. Available

- at: <https://www.frontiersin.org/journals/ict/articles/10.3389/fict.2018.00005/full> (accessed March 10, 2026).
- [11] T. D. Gillespie. *Fundamentals of Vehicle Dynamics*. Warrendale: Society of Automotive Engineers, Inc., 1992. Available at: https://www.researchgate.net/publication/239666431_Fundamentals_of_Vehicle_Dynamics (accessed April 22, 2026).
 - [12] Statens väg- och transportforskningsinstitut (VTI). Betydelsen av olika karakteristika hos beläggningsytan för trafik och omgivning (VTI notat 71-2000). Available at: <https://www.vti.se/> (accessed April 21, 2026).
 - [13] Wang, X. (ed.). *Vehicle Noise and Vibration Refinement*. Woodhead Publishing, 2010. ISBN: 978-1-84569-497-5. Available at: <https://www.sciencedirect.com/book/edited-volume/9781845694975/vehicle-noise-and-vibration-refinement> (accessed April 22, 2026). <https://www.sciencedirect.com/topics/engineering/sprung-mass>
 - [14] H. B. Pacejka. *Tyre and Vehicle Dynamics*, 2nd ed. Oxford: Butterworth-Heinemann, pp. 375–384, 452, 2006. Available at: <https://www.sciencedirect.com/book/9780750669122/tyre-and-vehicle-dynamics> (accessed April 21, 2026).
 - [15] Vibration Research. *What is Order Analysis?*. Vibration Research Blog. Vibration Research Corp. Available at: <https://vibrationresearch.com/blog/what-is-order-analysis/>
 - [16] Wang, X.; Cheng, Z.; Ma, N. Road Recognition Based on Vehicle Vibration Signal and Comfortable Speed Strategy Formulation Using ISA Algorithm. *Sensors* 2022, 22, 6682. Available at: <https://doi.org/10.3390/s22176682> (accessed May 25, 2026).
 - [17] González, A.; O'Brien, E. J.; Li, Y. Y.; Cashell, K. *The use of vehicle acceleration measurements to estimate road roughness*. *Vehicle System Dynamics*, 46(6): 483–499, 2008. Available at: <http://dx.doi.org/10.1080/00423110701485050> (accessed May 25, 2026).
 - [18] Adafruit Industries, *Adafruit Triple-Axis Accelerometer - MMA8451*, Available: <https://www.adafruit.com/product/1714> [Accessed: 30-Apr-2026].
 - [19] I. Frosio, F. Pedersini, and N. A. Borghese. Autocalibration of MEMS Accelerometers. *IEEE Transactions on Instrumentation and Measurement*, 2009. Available at: https://www.academia.edu/4784326/Autocalibration_of_MEMS_Accelerometers (accessed April 10, 2026).
 - [20] Sailboat Instruments. *Magneto v1.2: Calibration Tool for Magnetometers and Accelerometers*. Hosted by Atadiat. Available at: <https://atadiat.com/en/download/magneto-v1-2/> (accessed April 10, 2026).
 - [21] M. Pedley. *Tilt Sensing Using a Three-Axis Accelerometer*. Application Note AN3461, Rev. 6. NXP Semiconductors, 2013. Available at: <https://www.nxp.com/docs/en/application-note/AN3461.pdf>
 - [22] J. Plouzeau, D. Paillot, B. Aykent, och F. Merienne. *Vibrations in dynamic driving simulator: Study and implementation*. CONFERE 2013, Jul 2013, Frankrike. Tillgänglig på: <https://hal.science/hal-00844571v1>

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- [23] Wheel-Size.com. *Volvo EC40 (2025) Wheel and Tyre Sizes*. Available at: <https://www.wheel-size.com/size/volvo/ec40/2025/> (accessed April 22, 2026).
- [24] V. Yordanov et al. *Study of the vibrational behaviour of the components of a tyre-suspension-body system*. MATEC Web of Conferences, BulTrans-2018, 2018. Available at: https://www.matec-conferences.org/articles/matecconf/pdf/2018/93/matecconf_bultrans2018_02005.pdf (accessed April 16, 2026).
- [25] XCITE Audio. XBS100-4: Xtreme Power, Immersive Bass, Ultra Slim. *XCITE Audio Product Catalog*, 2026. Available at: <https://www.xcite-audio.com/product/xbs100-4/> (accessed April 8, 2026).
- [26] Dayton Audio. BST-300EX Extreme High Power Pro Tactile Bass Shaker 300 Watts. *Dayton Audio Product Catalog*, 2026. Available at: <https://www.daytonaudio.com/product/1953/bst-300ex-extreme-high-power-pro-tactile-bass-shaker-300-watts> (accessed April 8, 2026).
- [27] Siemens Digital Industries Software. *What is a Frequency Response Function (FRF)*. Siemens Community. Available at: <https://community.sw.siemens.com/s/article/what-is-a-frequency-response-function-frf>
- [28] Cornelis, B., Toso, A., Verpoest, W. and Peeters, B. (2014) Improved MIMO FRF estimation and model updating for robust Time Waveform Replication on durability test rigs. In: *Proceedings of ISMA2014 including USD2014*, Leuven, Belgium, pp. 760-762.
- [29] M. F. Abdullah, G. A. A. Qasem, M. Farid, H. S. Lim, C. P. Lee, and N. Alsakkaf. Adaptive control techniques for improving anti-lock braking system performance in diverse friction scenarios. *International Journal of Electrical and Computer Engineering (IJECE)*, vol. 15, no. 1, pp. 260–279, 2025. Available at: <http://ijece.iaescore.com> (accessed April 21, 2026).
- [30] Vägverket. *Vägar och gators utformning: Säker framkomlighet – Preliminära riktlinjer för utformning, reglering och drift*. VV Publikation 2006:37, 2006. Available at: <https://trafikverket.diva-portal.org/smash/get/diva2:1364398/FULLTEXT01.pdf> (accessed May 19, 2026).
- [31] Vibration Research Corporation. What Is Order Analysis? *Vibration Research Blog*. Available at: <https://vibrationresearch.com/blog/what-is-order-analysis/> (accessed April 13, 2026).
- [32] Signal Processing Systems Group, TU/e. Power Spectral Density. *SPS Education*, 2021. Available at: https://signalprocessingsystems.netlify.app/disciplines/statistical/statisticalsignalprocessing_signals_psd/ (accessed April 8, 2026).

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Appendix 1

The survey was distributed via Google Forms. The complete questionnaire is reproduced below.

Study in Driving Simulator

Please answer based on your experience during the driving session. There are no right or wrong answers.

* Anger obligatorisk fråga



1. Participant ID *

2. Age *

3. Gender *

Markera endast en oval.

- ☐ Male
- ☐ Female
- ☐ Other

4. Driving experience *

Markera endast en oval.

- ☐ No driving license
- ☐ Beginner (Less than 2 years)
- ☐ Intermediate (2–5 years)
- ☐ Proficient (5-10 years)
- ☐ Experienced (More than 10 years)

5. First run (1=Strongly Disagree, 7=Strongly Agree) *

Markera endast en oval per rad.

	1	2	3	4	5	6	7
The driving experience felt realistic.	☐	☐	☐	☐	☐	☐	☐
I could easily perceive the speed.	☐	☐	☐	☐	☐	☐	☐
I could notice when the car accelerated.	☐	☐	☐	☐	☐	☐	☐
I could notice when the car decelerated.	☐	☐	☐	☐	☐	☐	☐
I felt a clear connection with the road.	☐	☐	☐	☐	☐	☐	☐
I experienced motion sickness.	☐	☐	☐	☐	☐	☐	☐

6. If you experienced motion sickness, please describe it briefly.

7. Second run (1=Strongly Disagree, 7=Strongly Agree) *

Markera endast en oval per rad.

	1	2	3	4	5	6	7
The driving experience felt realistic.	☐	☐	☐	☐	☐	☐	☐
I could easily perceive the speed.	☐	☐	☐	☐	☐	☐	☐
I could notice when the car accelerated.	☐	☐	☐	☐	☐	☐	☐
I could notice when the car decelerated.	☐	☐	☐	☐	☐	☐	☐
I felt a clear connection with the road.	☐	☐	☐	☐	☐	☐	☐
I experienced motion sickness.	☐	☐	☐	☐	☐	☐	☐

8. If you experienced motion sickness, please describe it briefly.

9. For each item below, select the condition you preferred or found more effective. *

Markera endast en oval per rad.

	Vibration OFF	Vibration ON	No difference
Which condition helped you understand what the vehicle was doing?	☐	☐	☐
Which condition provided the most useful physical feedback overall?	☐	☐	☐
Which condition felt the least distracting?	☐	☐	☐
Which condition felt the most realistic or authentic to a real-world driving experience?	☐	☐	☐
Which condition did you find the most enjoyable and fun?	☐	☐	☐

10. What differences did you notice between the two conditions?

11. Did the vibrations help you understand any specific vehicle events?

12. Were there any vibrations that felt unnecessary, distracting, or poorly timed?

13. Any additional comments or suggestions?

Det här innehållet har varken skapats eller godkänts av Google.

Google Formulär

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