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Development and investigation of a finite-element model for predicting railway vehicle undefloor noise

Diplomarbeit
Likun LUO



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DIPLOMARBEIT

Development and investigation of a finite-element model for predicting railway vehicle underfloor noise

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Abstract

Kurzfassung

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Notations and Symbols

A	m^2	Fläche (<i>area</i>)
c^D	N/m^2	mechanischer Modul bei konstanter dielektrischer Verschiebung (<i>stiffened elastic constant</i>)
c^E	N/m^2	mechanischer Modul bei konstanter elektrischer Feldstärke (<i>elastic constant with constant E field</i>)
c_s	m/s	Schallgeschwindigkeit (<i>wave velocity</i>)
\bar{c}_s	m/s	Steifigkeitsschallgeschwindigkeit (<i>stiffened wave velocity</i>)
c_l	m/s	Longitudinalwellengeschwindigkeit
c_t	m/s	Transversalwellengeschwindigkeit
\vec{T}_{ges}	N/m^2	mechanische Spannung (<i>stress</i>)
\vec{T}	N/m^2	mechanische Wechselspannung
\vec{T}_e	N/m^2	mechanische Wechselspannung der einfallenden Welle
\vec{T}_r	N/m^2	mechanische Wechselspannung der reflektierten Welle
\vec{T}_d	N/m^2	mechanische Wechselspannung der durchgehenden Welle
α	$1/\text{m}$	Absorptionskoeffizient (<i>attenuation constant</i>)
$\tilde{\gamma}$	dB	Absorptionsdämpfung
ρ	kg/m^3	Wechseldichte
τ	s	Zeitverzögerung
Φ	N/V	Wandlungsfaktor
$\frac{\partial \underline{x}}{\partial t}$		partielle Differentiation von \underline{x} nach t
\bar{x}		Mittelwert von x
\vec{x}		Vektor \vec{x}
\vec{x}_n		Normalkomponente des Vektors \vec{x}
$\text{div } \vec{x}$		Divergenz des Vektors \vec{x}
∇		Nabla-Operator
\mathbf{K}		Matrix oder Tensor \mathbf{K}
\mathbf{K}_t		transponierte Matrix \mathbf{K}
\underline{x}		komplexe Zahl \underline{x}
$\text{Re}\{\underline{x}\}$		Realteil von \underline{x}
$\text{Im}\{\underline{x}\}$		Imaginärteil von \underline{x}
\underline{x}^*		konjugiert-komplex von \underline{x}
j		imaginäre Einheit
$J_n(x)$		Besselfunktion n-ter Ordnung

Chapter 1

Introduction

1.1 Background

Modern rail vehicles like trains, metro vehicles or trams have to meet ever increasing acoustic requirements and regulations not only to improve the acoustic comfort of the passengers, but also to reduce environmental noise pollution from railways [1–3].

The transmission of noises is complex procedures because many paths: airborne, structure borne structural vibration

- outline the specific objective of the research The aim of this thesis is ...

This thesis aims to provide/determine/develop/investigate

- Provide overview of thesis structure: The thesis is structured as follow ...

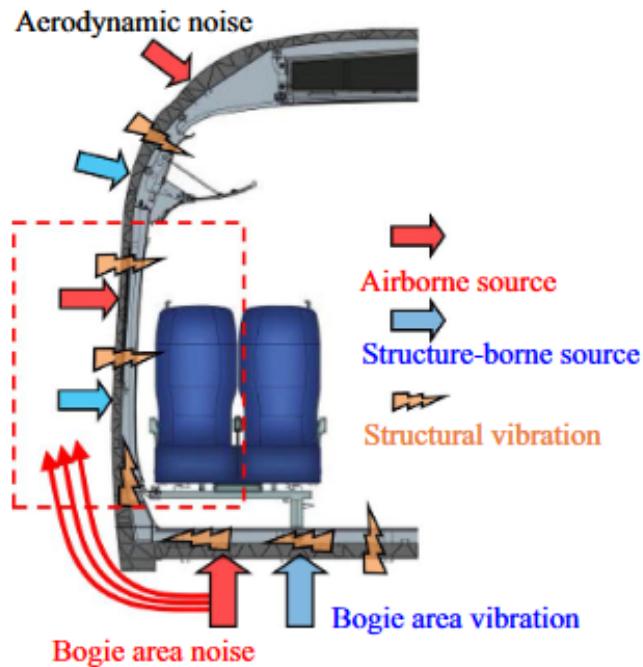


Figure 1.1: Transmission path of exterior noise sources [3]

1.2 Aims and objectives

This thesis, in collaboration with Siemens Mobility Austria GmbH, aims to develop a finite-element model suitable for predicting the sound propagation from train underfloor area into the exterior environment around the carbody.

The thesis is organised as follows. The fundamental background of the thesis is explained in Chap. 2. Chap. 3 describes the outer pressure field measurement

Chapter 2

Fundamentals

2.1 Governing equations and finite element formulation

2.2 Perfectly Matched Layer

2.3 Non-conforming grids

2.4 Fundamentals of noise measurement

2.4.1 Octave-band and fractional octave-band

2.4.2 Sound Level

2.4.3 Weighting

2.5 UBX metro train

Chapter 3

Measurements

In order to be able to validate the results obtained from the latter finite element simulation, specific measurements have been carried out. In this chapter, the acoustic power measurement of sound source and the outer pressure field measurement around carbody are presented. The obtained sound power spectral of the omnidirectional loudspeaker provide simulation input for the finite element analysis in section 4.2. Furthermore, the measurement results of outer pressure field serve as validation for the finite element model in chapter 5.

3.1 Characterisation of sound source

Two important characteristics of a sound source are directivity and sound power. For the validation measurement, a dodecahedron loudspeaker of type Brüel & Kjaer 4292-L is used as excitation, which can be treated as an omnidirectional sound source. Hence, only the sound power emitted by the loudspeaker is of interest. In the following, the acoustic power spectrum of the sound source is measured following ISO 9614-2 [4].

Measurement setup

In fig. 3.1 the measurement setup for the determination of sound power levels is shown. The omnidirectional loudspeaker is placed on a concrete reflective floor and is enveloped by a 1m x 1m x 1m reference box. The loudspeaker is driven by a reference amplifier with pink noise as input signal (fig. 3.2). In order to reduce fluctuation in output power, the amplifier is switched on and warmed up for at least 20 minutes before the start of the measurement.

The measurement is carried out according to the ISO 9614-2 standard, in which the sound power is determined from intensity measurement over measurement surfaces. This method has the advantage that room reflections and any sound sources outside of the reference box do not influence the measurement result.

The intensity measurements are done using the Brüel & Kjaer sound intensity probe kit type 3654, which includes a pair of microphones that are placed face to face to each other, separated by a spacer. That is why this method of intensity measurement is also called two-microphone method or $p-p$ method. Instead of direct measurement, the particle velocity is obtained by a finite-difference approximation to the pressure gradient. [5] [6]

From the euler equation of motion

$$\rho_0 \frac{\partial \vec{u}}{\partial t} = -\nabla p \approx -\frac{p_2 - p_1}{\Delta r} \quad (3.1)$$

where p_1 and p_2 are the measured pressure of the microphone closer and further from the incident sound wave, respectively and Δr the thickness of the spacer.

Hence, the particle velocity can be calculated from the two pressures

$$\vec{u} = \int \frac{(p_1 - p_2)}{\rho \Delta r} \partial t \quad (3.2)$$

The instantaneous sound intensity is obtained by the product of the approximated particle velocity and mean of the two measured pressures

$$\vec{I} = p\vec{u} = \frac{p_1 + p_2}{2} \int \frac{(p_1 - p_2)}{\rho \Delta r} \partial t \quad (3.3)$$

Fig. 3.3 shows the intensity measurement using scan method. For the scanning, the intensity probe is moved over the surface with a constant speed and is kept perpendicular to the scanned area so that only the normal component of intensity is captured:

$$I_n = \vec{I} \cdot \vec{n} \quad (3.4)$$

where \vec{n} is the unit normal vector of the measurement surface. Each partial measurement surface is scanned twice, using vertical and horizontal scan path, respectively. The obtained result is the spatial-averaged intensity. The sound power of a single measurement surface is given by the product of the spatial average intensity and the surface area:

$$P_i = \langle I_{ni} \rangle S_i \quad (3.5)$$

The total sound power of the enclosed sound source is given by:

$$P_{total} = \sum_{i=1}^N P_i \quad (3.6)$$



Figure 3.1: Setup for sound power measurement



(a) Inter-M R300 Plus reference amplifier

(b) NTI Minirator MR-PRO

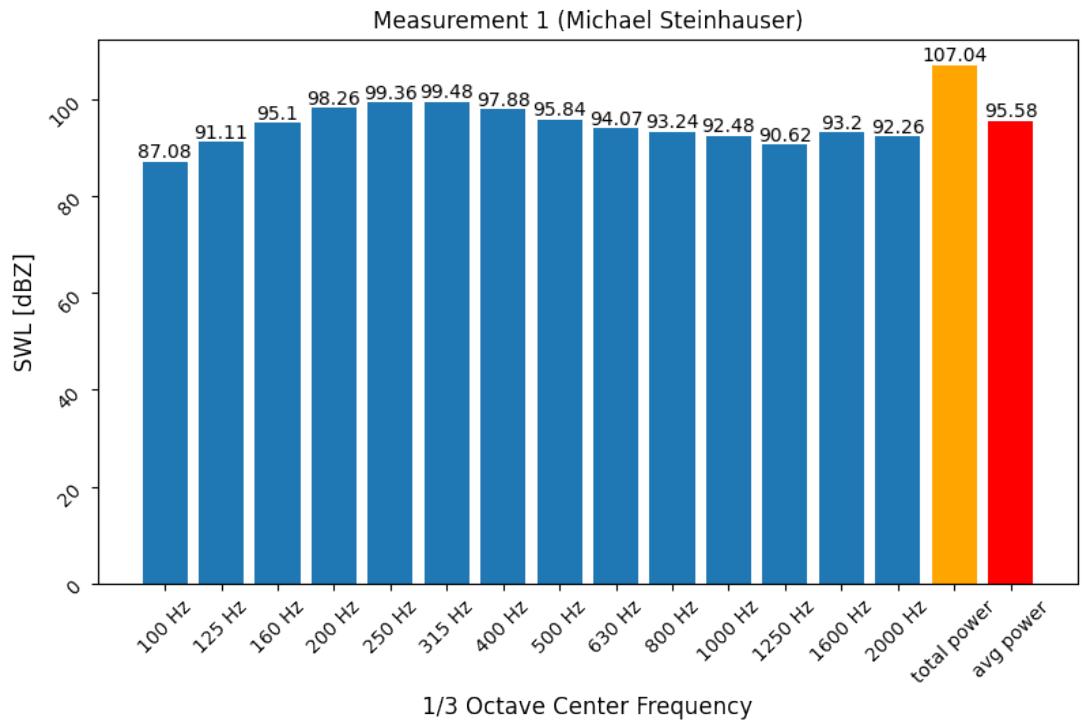
Figure 3.2: Signal generator and amplifier



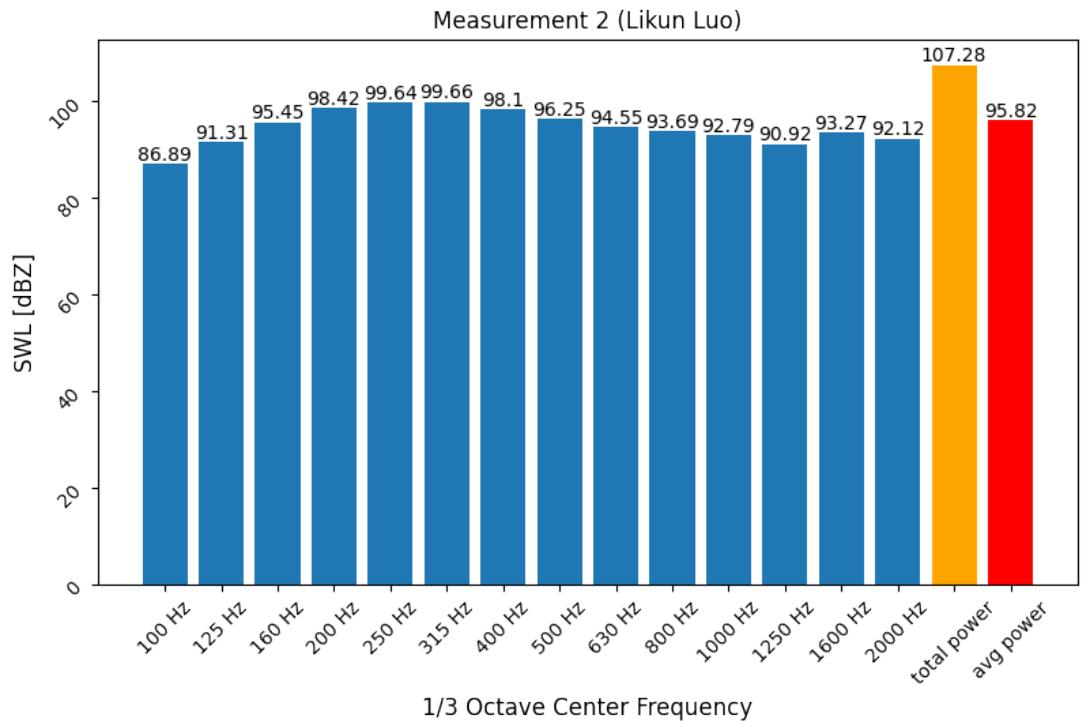
Figure 3.3: Scan method according to ISO 9614-2 [4]

Measurement results

The measurement is post-processed using the Brüel & Kjaer type 2270 hand-held analyzer and the result is shown in one-third octave band spectra. Fig. 3.4 shows the measured nonweighted (Z-weighting) sound power level (SWL ref 1e – 12 W) from 100 Hz to 2000 Hz. The measurement is carried out twice, the deviation of both measurements is within 0.3 dB in total power. For the finite element simulation, the averaged value of both measurements as shown in fig. 3.4c will be used.



(a) Measurement series 1



(b) Measurement series 2

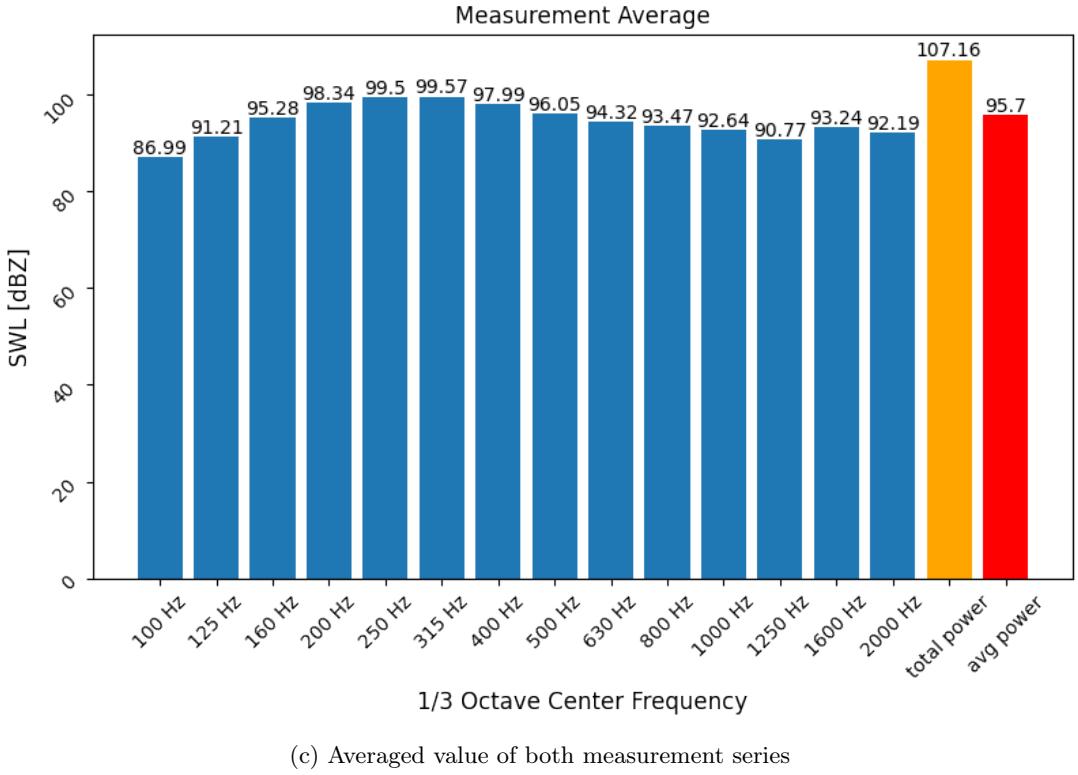


Figure 3.4: Sound power spectra in 1/3-Octave band

3.2 Pressure field measurement on a standing metro train

In this section, the sound pressure measurement on a standing UBX metro with controlled excitation has been carried out. The aim to perform measurements under standstill condition is to avoid the uncertainties introduced by the complex real sources (e.g. wheel-rail interaction) in dynamic conditions. In such wise, the quality of the finite element model can be better assessed.

Measurement setup

The measurement was performed on October 4, 2022 at company premises of Siemens Mobility Austria in 1110 Vienna. The measurement setup is shown in fig. 3.5. Fig. 3.5a shows the front car of the measurement vehicle of type Siemens U-Bahn Wien X. In order to approximate free field conditions as closely as possible, the measurement vehicle is parked in a relatively empty area of company premises avoiding environmental reflections as much as possible.

To evaluate the outer pressure field close to the carbody, 10 pressure microphones are arranged in a line array order, allowing simultaneous measurement along the height direction. As shown in fig. 3.5b, the microphones are attached on a tripod at a distance of half meter to each other, the position of the microphones starts at 0.5 m and ends at 5 m above ground.

The acoustic excitation comes from an omnidirectional loudspeaker with pink noise as input signal, using the identical settings as in the previous acoustic power measurement. The loudspeaker is placed beneath the car underframe in the empty space between the bogie axle and the wheel axle, at about 55 cm above ground (Fig .3.6). Measurements have been performed with 2 different loudspeaker locations. One at the front of bogie (Position A), and the other at the rear (Position B), as marked in fig. 3.5b. The aim is to evaluate the possible asymmetric effect caused by the anti-symmetric layout of the bogie

components (e.g. brake discs), also it provides a validation for the latter finite element model, which will exploit the symmetry using image-source technique.

Due to the limited availability of the measurement vehicle, the measurement is concentrated on only one side of the vehicle, near the front bogie. In fig. 3.7, all measurement positions of the microphone array are shown and labeled. Position a is the starting position of measurement, it lies on the centerline of the bogie frame, 10 cm away from the carbody edge (Fig. 3.7b). The full description of measurement positions is found in the table below:

Position Label	Description
a	on bogie centerline , 10 cm away from carbody edge
b	50 cm out of bogie centerline in front direction, 10 cm away from carbody edge
c	100 cm out of bogie centerline in front direction, 10 cm away from carbody edge
d	50 cm out of bogie centerline in rear direction, 10 cm away from carbody edge
e	100 cm out of bogie centerline in rear direction, 10 cm away from carbody edge
f	on bogie centerline , 50 cm away from carbody edge
g	on bogie centerline , 100 cm away from carbody edge
h	on bogie centerline , 150 cm away from carbody edge
i	on bogie centerline , 200 cm away from carbody edge

Table 3.1: Description of measurement positions

At each measurement position, the multi-channel time-varying signal of the pressure field is captured by the Müller-BBM PAK MKII data acquisition system. The duration of single measurement run is set to 15 seconds. The captured time signal will be post-processed using the provided Müller-BBM PAK software suit.



(a) Front view



(b) Side view

Figure 3.5: Outer pressure field measurement setup

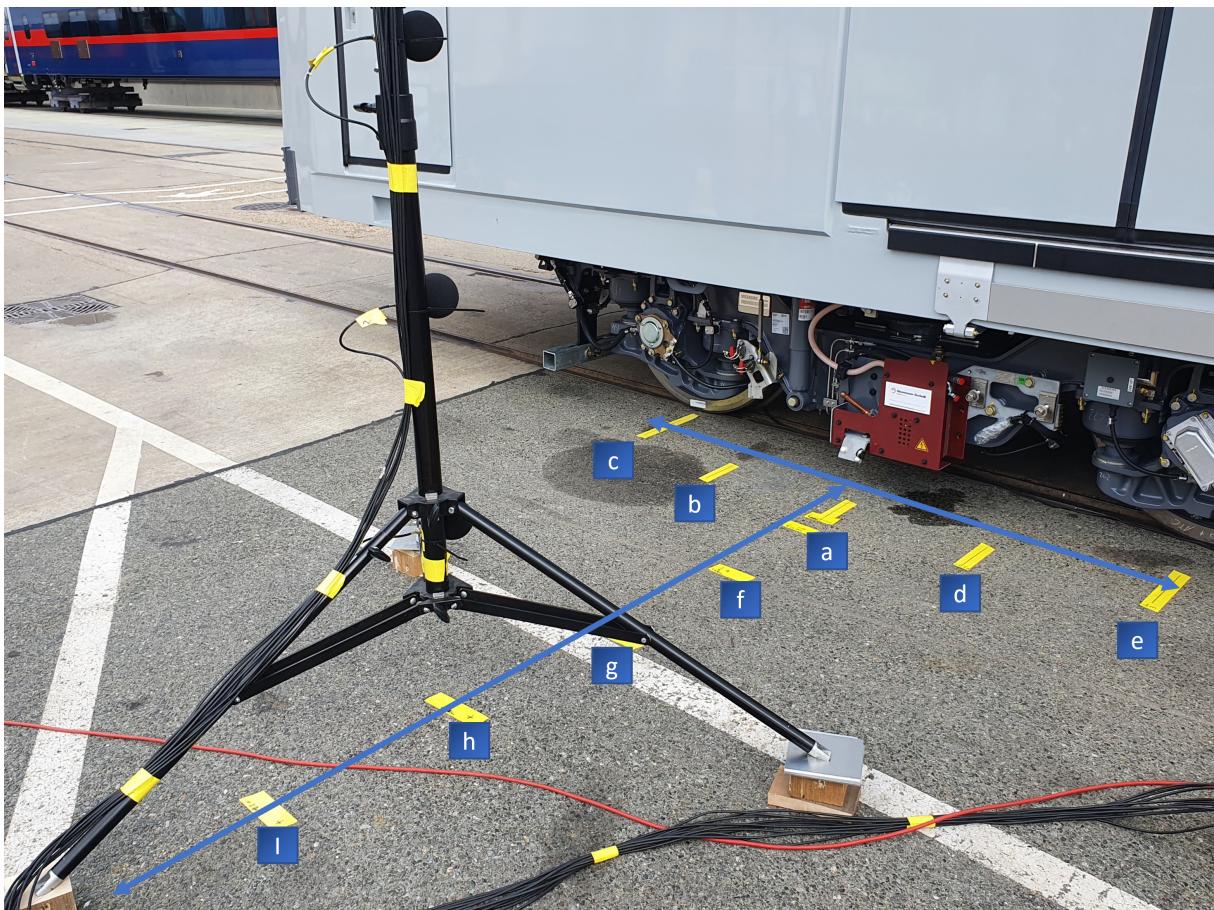


(a) Position A: front of bogie



(b) Position B: rear of bogie

Figure 3.6: Loudspeaker locations



(a) Measurement positions a to i



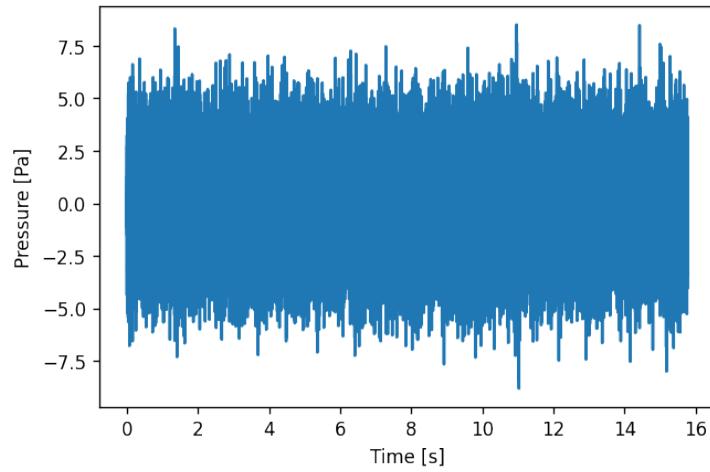
(b) Postition a: centerline of the bogie, 10 cm away from carbody edge

Figure 3.7: Measurement positions of microphone array

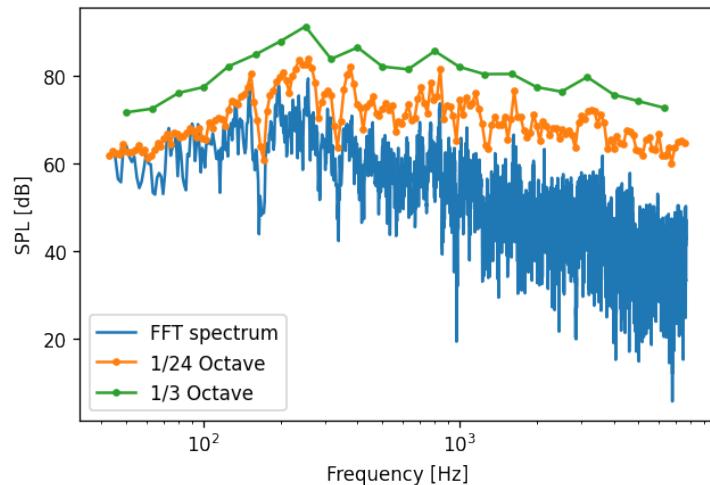
Measurement results

In fig. 3.8a, the measured time signal of microphone 1 (0.5 m above ground) at measurement position a (front bogie centerline, 10 cm away from carbody) with loudspeaker placed at the front of bogie can be seen. The signal is converted into frequency domain using Fourier transformation as shown in fig. 3.8b. The post-processing can be done directly in the provided software suit Müller-BBM PAK 6.x, fig. 3.9 shows a screenshot of the GUI and the FFT parameter used is circled.

In fig. 3.8b, the blue curve shows the amplitude of acoustic pressure in narrow band resolution. The narrow band data is converted to 1/n octave form by summing up the amplitude of narrow-band spectral lines contained within the corresponding frequency bandwidth. The advantage of post-processing the data into octave bands is that it provides clearer information about the frequency composition of the noise signal. For example, it can be observed that in the 1/3 octave curve in fig. 3.8b, the peak of the SPL appears at 315 Hz, which also matches the peak in the SWL spectrum of the sound source as shown in fig. 3.4c.



(a) Time domain



(b) Frequency domain

Figure 3.8: Measurement data at position a, microphone 1 (0.5 m), loudspeaker front

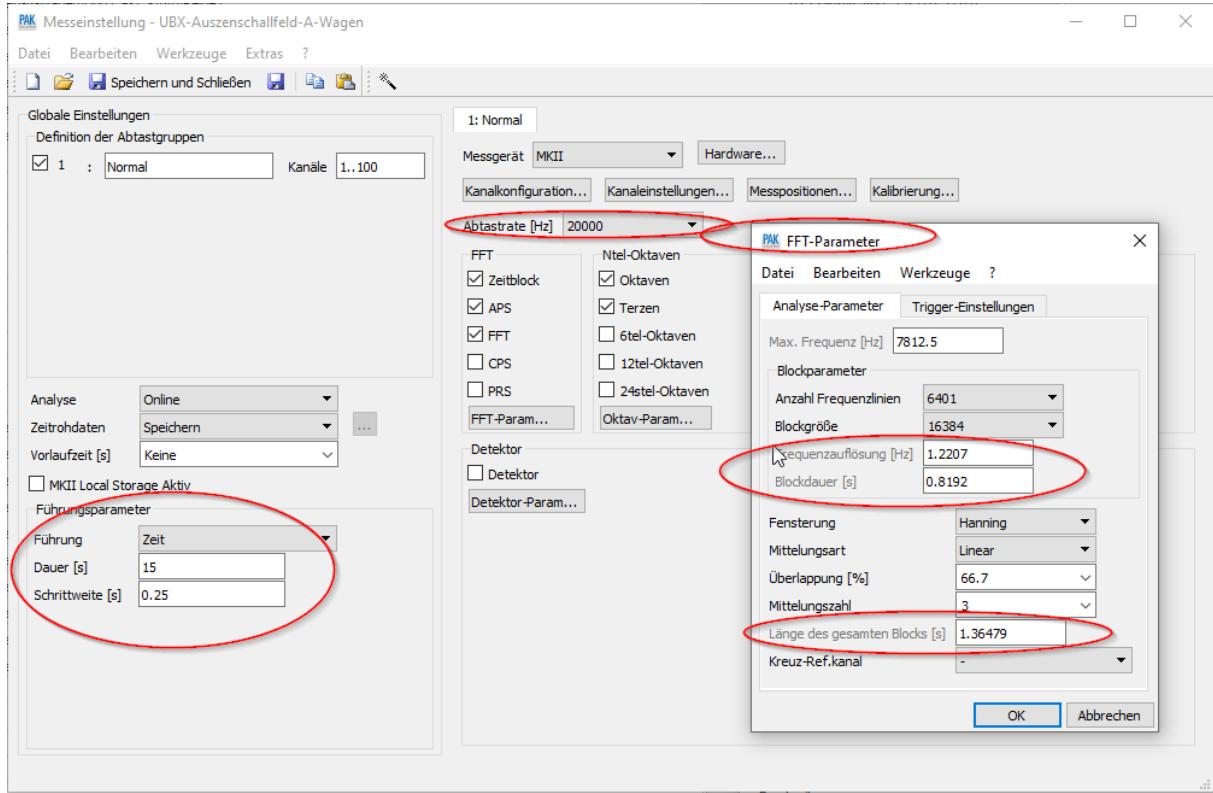


Figure 3.9: FFT parameter used

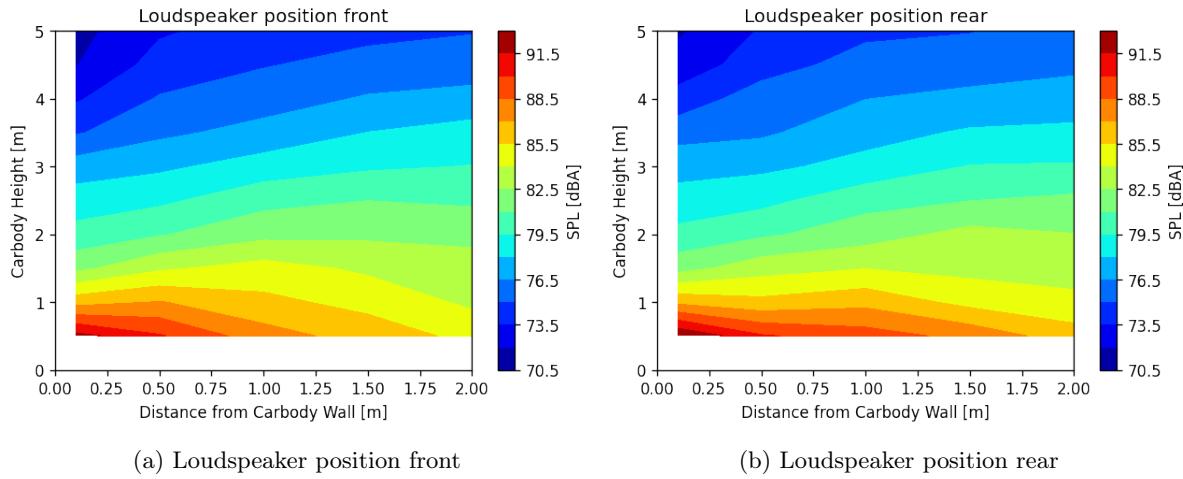


Figure 3.10: Pressure field around carbody

In fig. 3.10, a visualization of the total pressure field around the carbody is shown. The horizontal axis represents the distance from carbody wall, the vertical axis the height above ground and the color the a-weighted total pressure level, respectively. The white space in the plot is due to the missing data in the measurement since the measurement positions start at 10 cm away from carbody wall and half meter above ground. Comparing the pressure field of the two different loudspeaker locations, the asymmetric effect introduced by the brake disc can be observed. The brake disc of the front wheel axle standing in the transmission path of the loudspeaker seems to block a part of the acoustic wave.

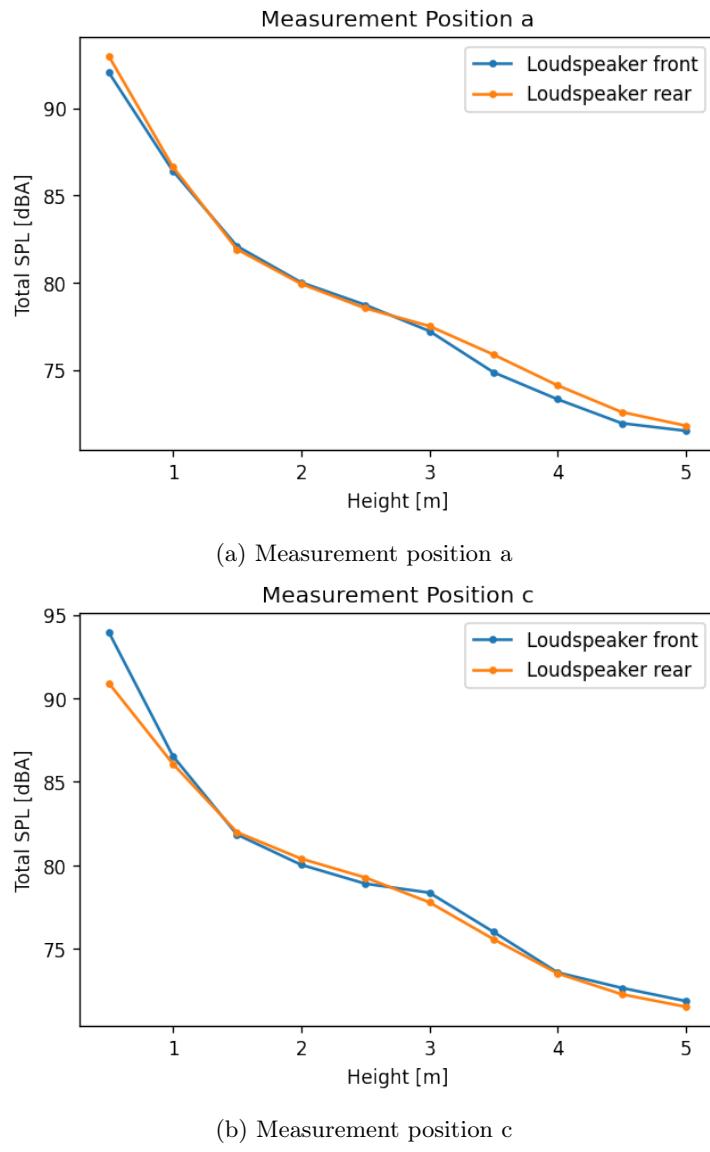
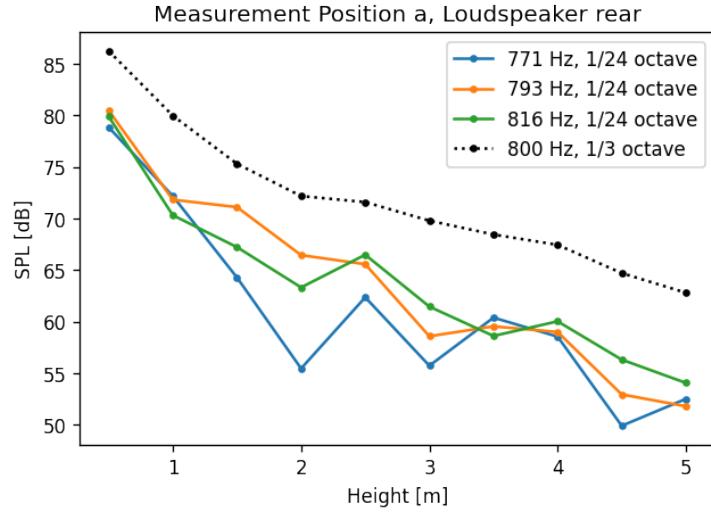
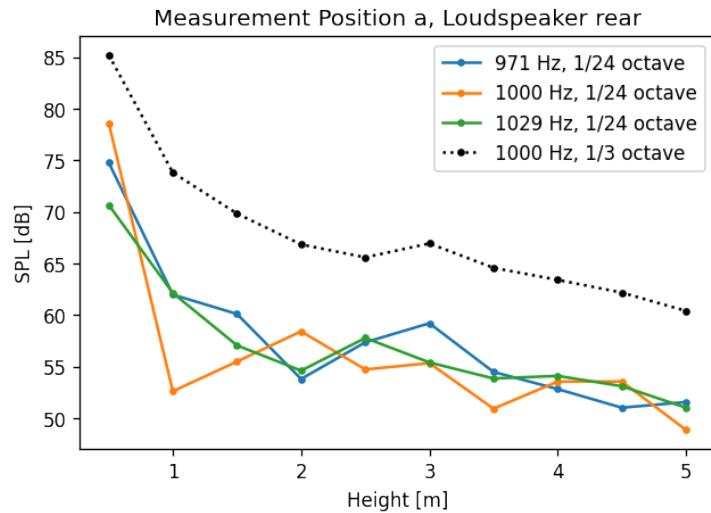


Figure 3.11: Total SPL as a function of the height above ground

In order to compare the pressure field quantitatively, one can plot the acoustic pressure as function of height for different measurement positions, which can be seen in the figures above. In 3.11a it can be observed that at measurement position a, the pressure curve caused by loudspeaker at different locations shares similar shape, and the pressure is strictly decreasing over carbody height.



(a) 800 Hz



(b) 1000 Hz

Figure 3.12: SPL as a function of the height above ground, single-frequency, measurement position a, loudspeaker rear

In fig. 3.12, the sound pressure level over height for different $1/n$ octave band center frequencies is displayed. In the $1/24$ octave resolution, several local minima in the curve shape can be observed, e.g. for 771 Hz in fig. 3.12a or for 1000 Hz in fig. 3.12b, which are caused by the destructive interference of the acoustic wave. The destructive interference can also be observed in the pressure field of the single frequency band as shown in fig. 3.13, sinks in acoustic field are to be found at about 1 m and 3.5 m height, respectively, which correspond to the position of the local minima in fig. 3.12b.

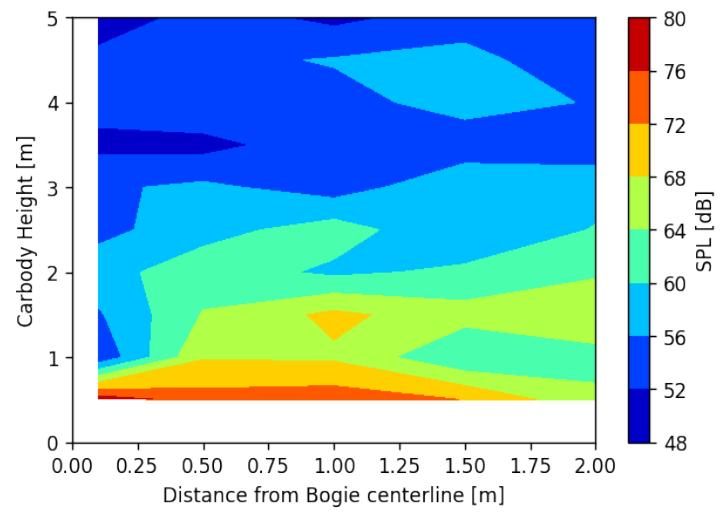


Figure 3.13: Pressure field of 1/24-octave frequency 1000 Hz, loudspeaker rear

Chapter 4

Finite Element Modeling

In this chapter, the development of the finite element model of the UBX metro and the simulation setup are described. First, in section 4.1, the design of the model geometry and the mesh of the model are shown. Then, in section 4.2, the incorporation of boundary conditions into the finite element model is described. Finally, in section 4.4, a parametric study is carried out to investigate the influence of different simulation parameters on the solution of the initial finite element model. All these simulations are done with the acoustics module of the open-source FEM software openCFS [7].

4.1 Geometry and mesh

As shown in sec. 2.5 Due to the large dimension of the car body
a representative model has to be

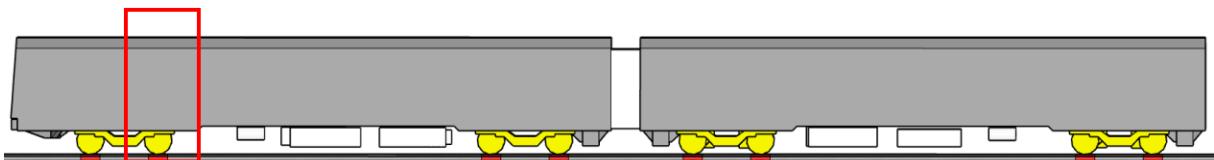


Figure 4.1: Side view of UBX, red box marked the modeling area

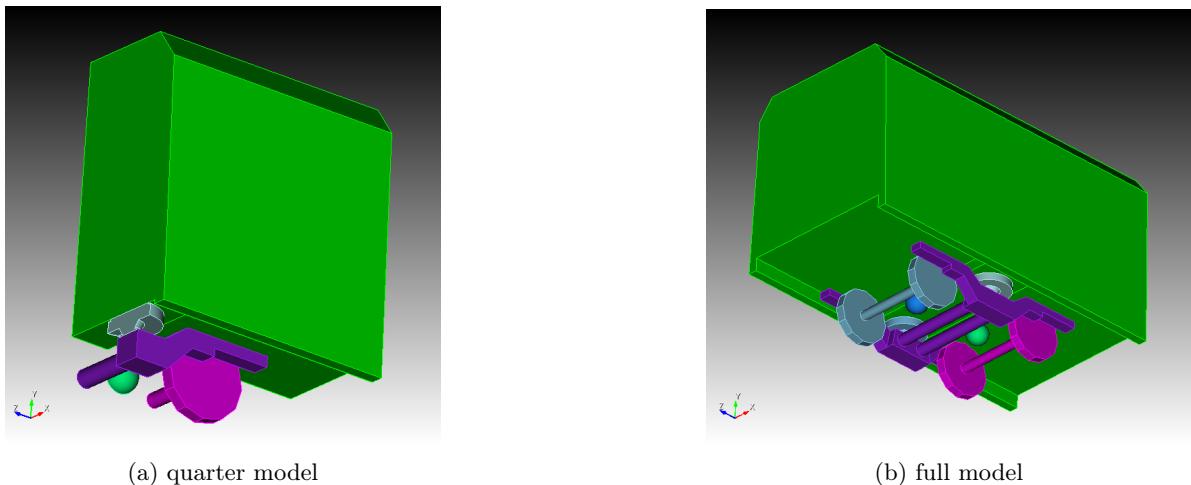


Figure 4.2: Geometry of model

4.1.1 Determination of simulation domain

4.1.2 Computational Effort

The aim of this section is

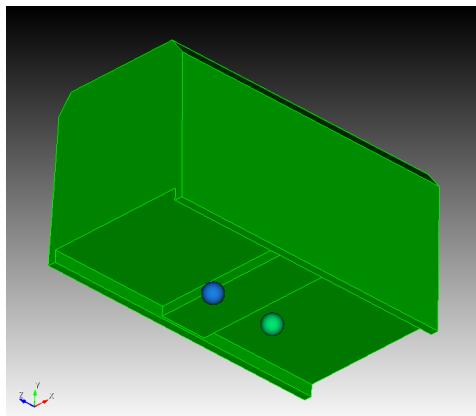
4.2 Boundary conditions and loads

4.2.1 Modeling of sound source

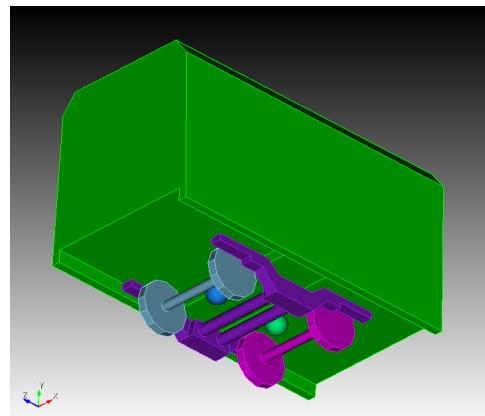
4.3 General simulation setup

4.4 Parametric study

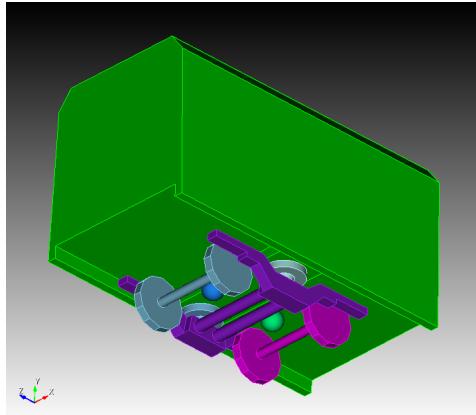
4.4.1 Variation of underfloor geometry



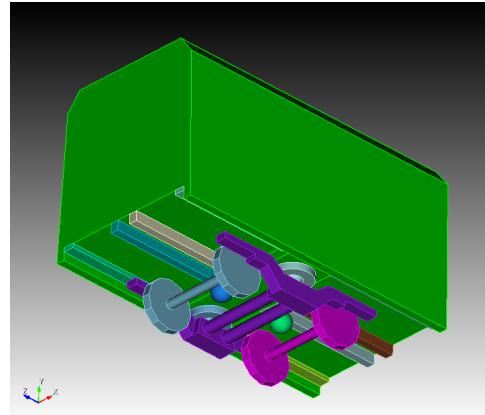
(a) No underfloor components



(b) No air suspension



(c) Initial model



(d) Additional structures

Figure 4.3: Used geometry

4.4.2 Inclusion of ground absorption

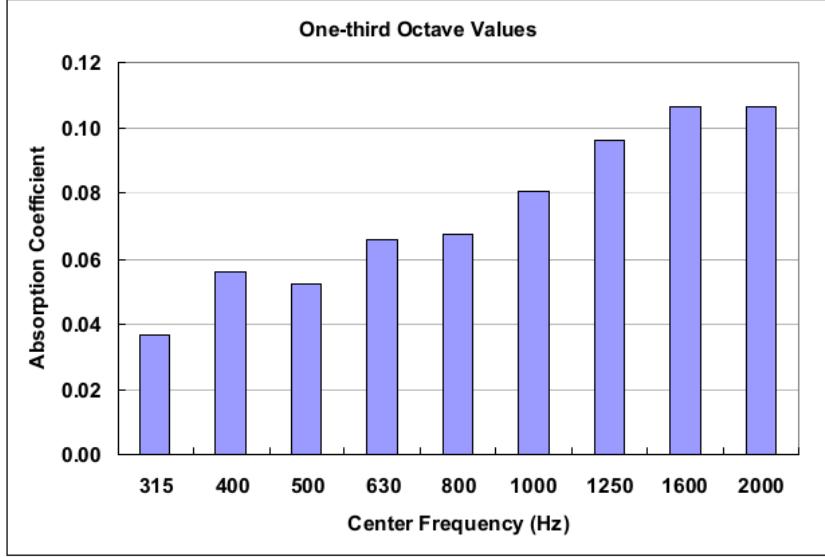


Figure 4.4: Absorption coefficient in one-third octave bands [8]

Table 4.1: Estimated absorption coefficient from fig. 4.4, the values for frequency lower than 315 Hz are chosen arbitrary

Freq (Hz)	100	125	160	200	250	315	400
α	0.02	0.02	0.02	0.02	0.02	0.035	0.055
Freq (Hz)	500	630	800	1000	1250	1600	2000
α	0.05	0.065	0.065	0.08	0.095	0.105	0.105

$$\alpha = 1 - |r^2| \quad (4.1)$$

$$r = \frac{Z_s - Z_0}{Z_s + Z_0} = \frac{\frac{Z_s}{Z_0} - 1}{\frac{Z_s}{Z_0} + 1} \quad (4.2)$$

$$\tilde{Z}_s = \frac{Z_s}{Z_0} = \tilde{R} + j\tilde{X} \quad (4.3)$$

$$\begin{cases} \frac{4\tilde{R}_s}{\tilde{R}_s^2 + \tilde{X}_s^2 + 2\tilde{R}_s + 1} = \alpha \\ \arctan \frac{\tilde{X}_s}{\tilde{R}_s} = \varphi \end{cases} \quad (4.4)$$

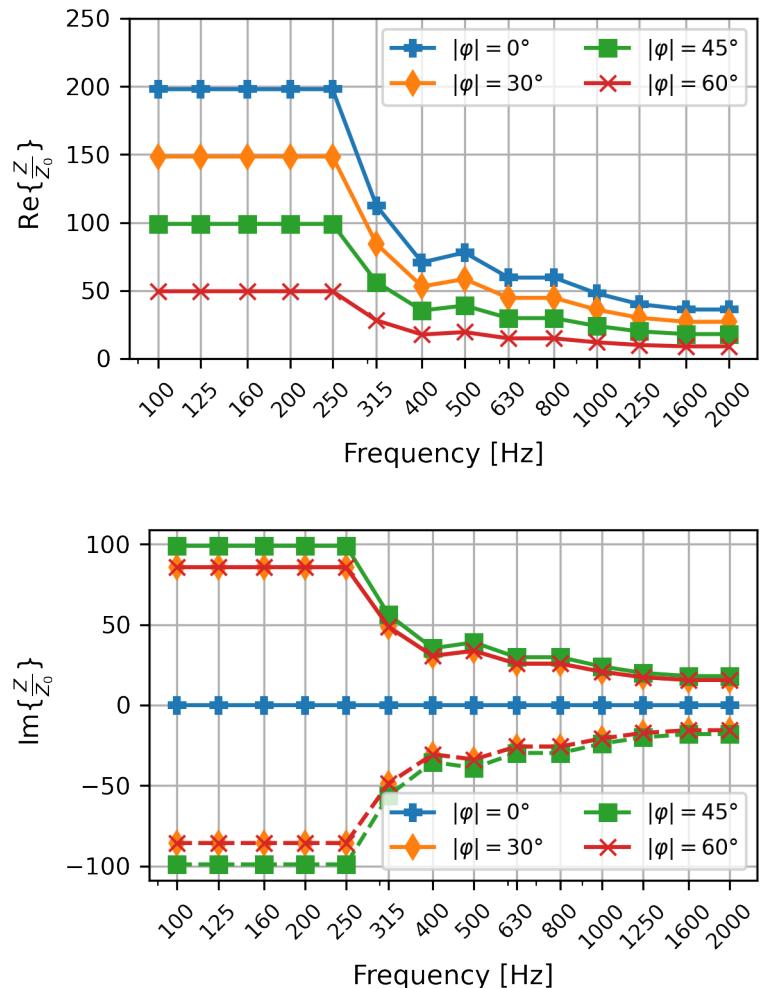


Figure 4.5: Input impedance in one-third octave bands

4.4.3 Variation of frequency steps per 1/3-octave band

Chapter 5

Results and Discussions

In the following, the results obtained from the finite element simulation using different setups are discussed and compared to the outer pressure field measurement.

5.1 Comparison of simulation and measurement results

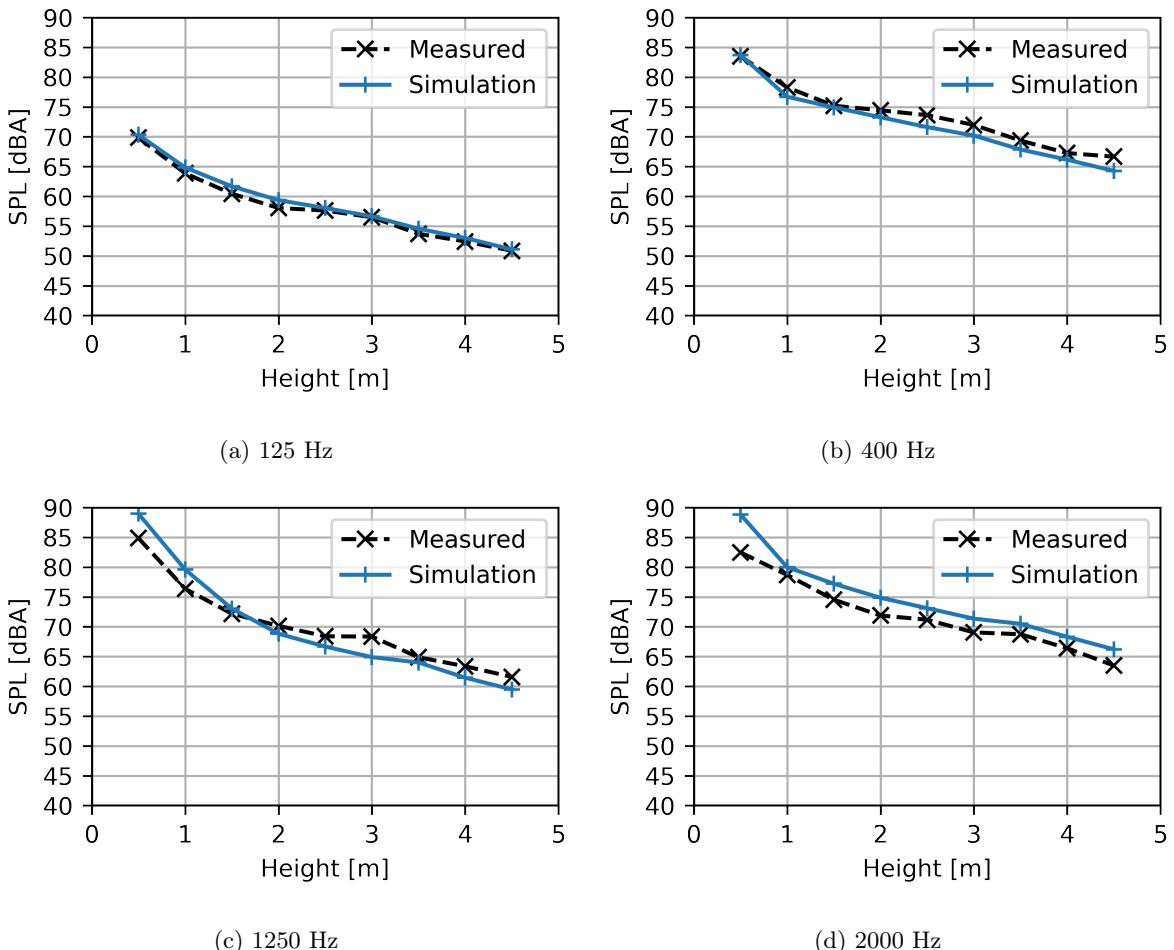


Figure 5.1: Sound distribution at measurement position a, comparison between predictions obtained from the initial finite element model with the measurements. A-weighted SPL in one-third octave bands, dBA ref 20 μ Pa.

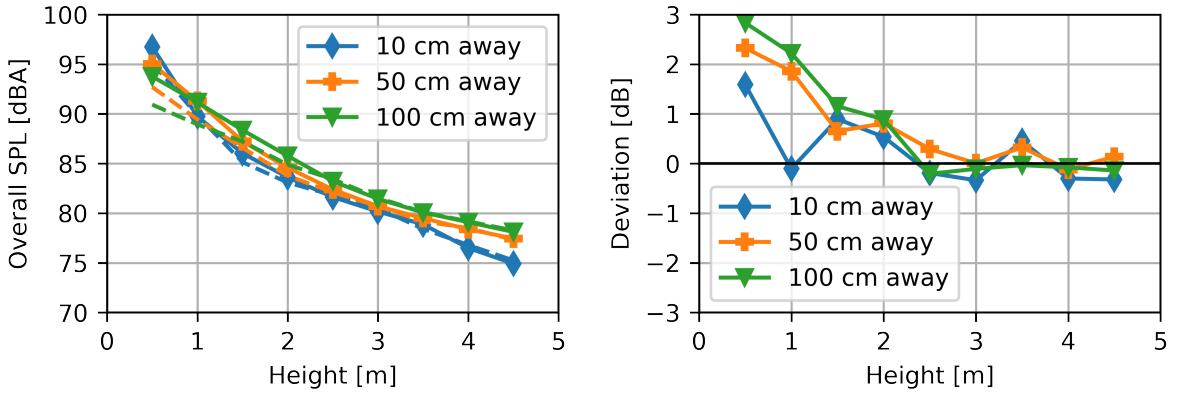


Figure 5.2: Comparison of overall SPL in dBA ref $20 \mu\text{Pa}$. Left: overall SPL as a function of height at various measurement positions (dashed curves: measurement; solid curves: simulation) Right: deviation between simulation and measurement results

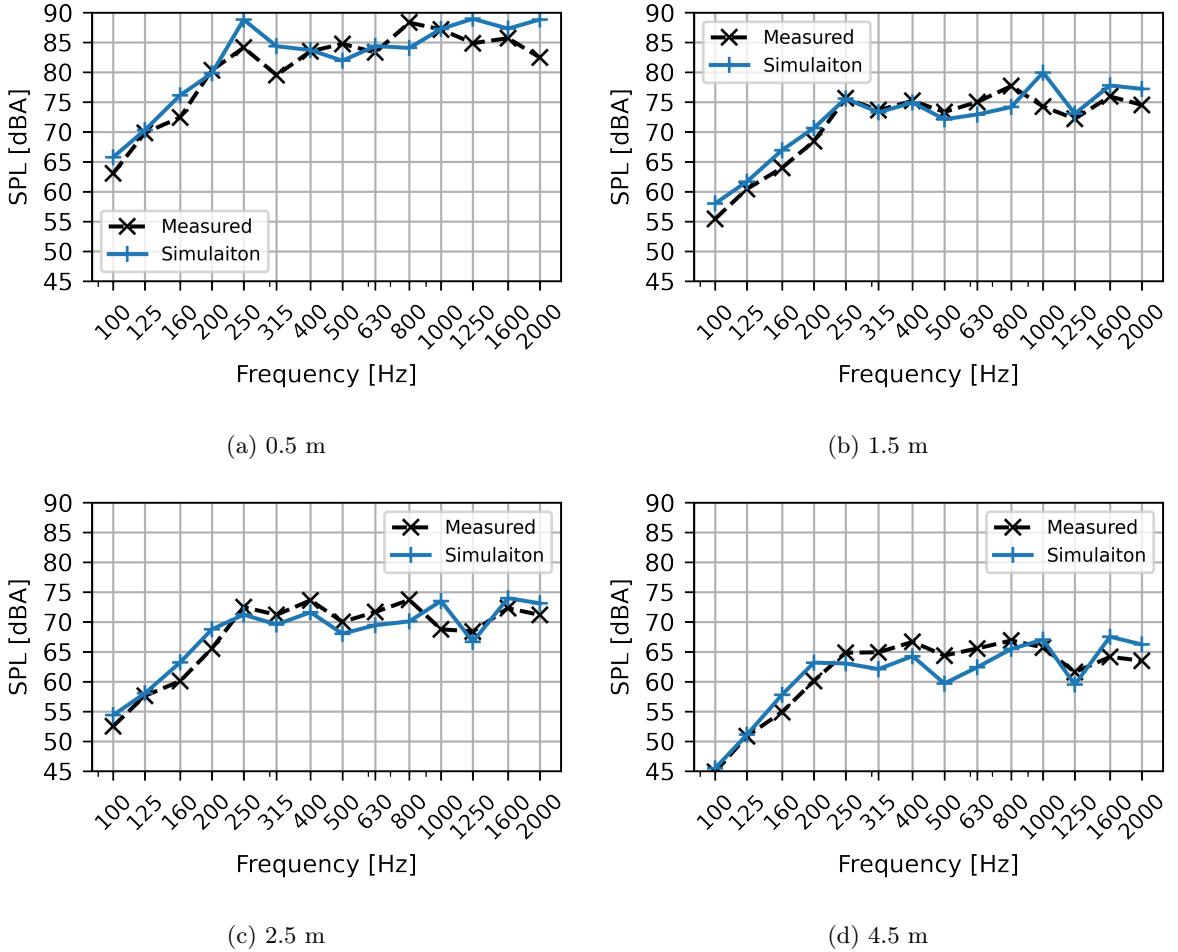


Figure 5.3: Comparison of overall SPL in dBA ref $20 \mu\text{Pa}$. Left: overall SPL as a function of height at various measurement positions (dashed curves: measurement; solid curves: simulation) Right: deviation between simulation and measurement results

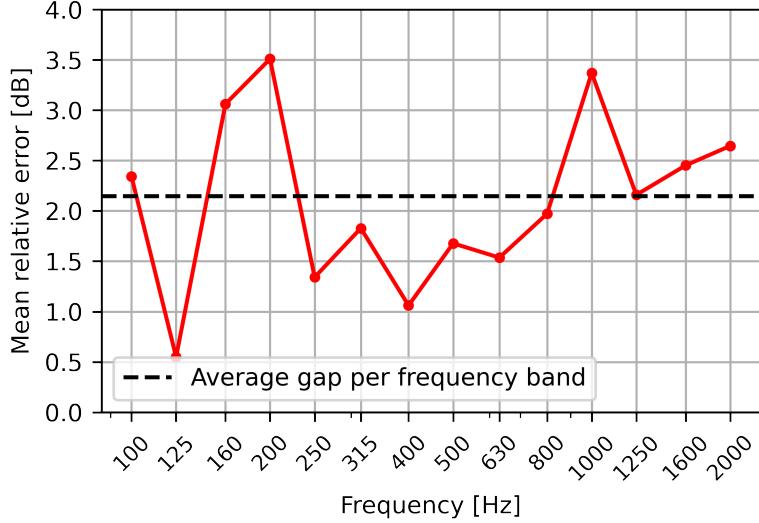


Figure 5.4: Mean relative error of 1/3-octave frequency

The sound pressure level is first converted to linear scaling:

$$p(\text{Pa}) = p_0 \cdot 10^{\frac{\text{SPL}}{20}} = 2 \cdot 10^{-5} \text{ Pa} \cdot 10^{\frac{\text{SPL}}{20}} \quad (5.1)$$

Mean relative error:

$$\text{MRE}(p_{\text{measured}}, p_{\text{simulation}}) = \frac{1}{N} \sum_{i=0}^{N-1} \frac{|p_{\text{simulation},i} - p_{\text{measured},i}|}{|p_{\text{measured},i}|} \quad (5.2)$$

The mean relative error can be converted to decibel:

$$\text{MRE(dB)} = 20 \cdot \log_{10}(1 + \text{MRE}) \quad (5.3)$$

Table 5.1: Mean relative error of 1/3-octave frequency spectrum over all microphone positions

Freq (Hz)	100	125	160	200	250	315	400	500	630	800	1000	1250	1600	2000
MRE (dB)	2.34	0.55	3.06	3.51	1.34	1.82	1.06	1.67	1.53	1.97	3.36	2.15	2.45	2.65

Table 5.2: Average gap per frequency band and mean relative error of overall SPL of initial model

Average gap per frequency band (dB)	Mean relative error of overall SPL (dB)
2.14 ± 1	0.73 ± 0.86

5.2 Effect of geometric variation

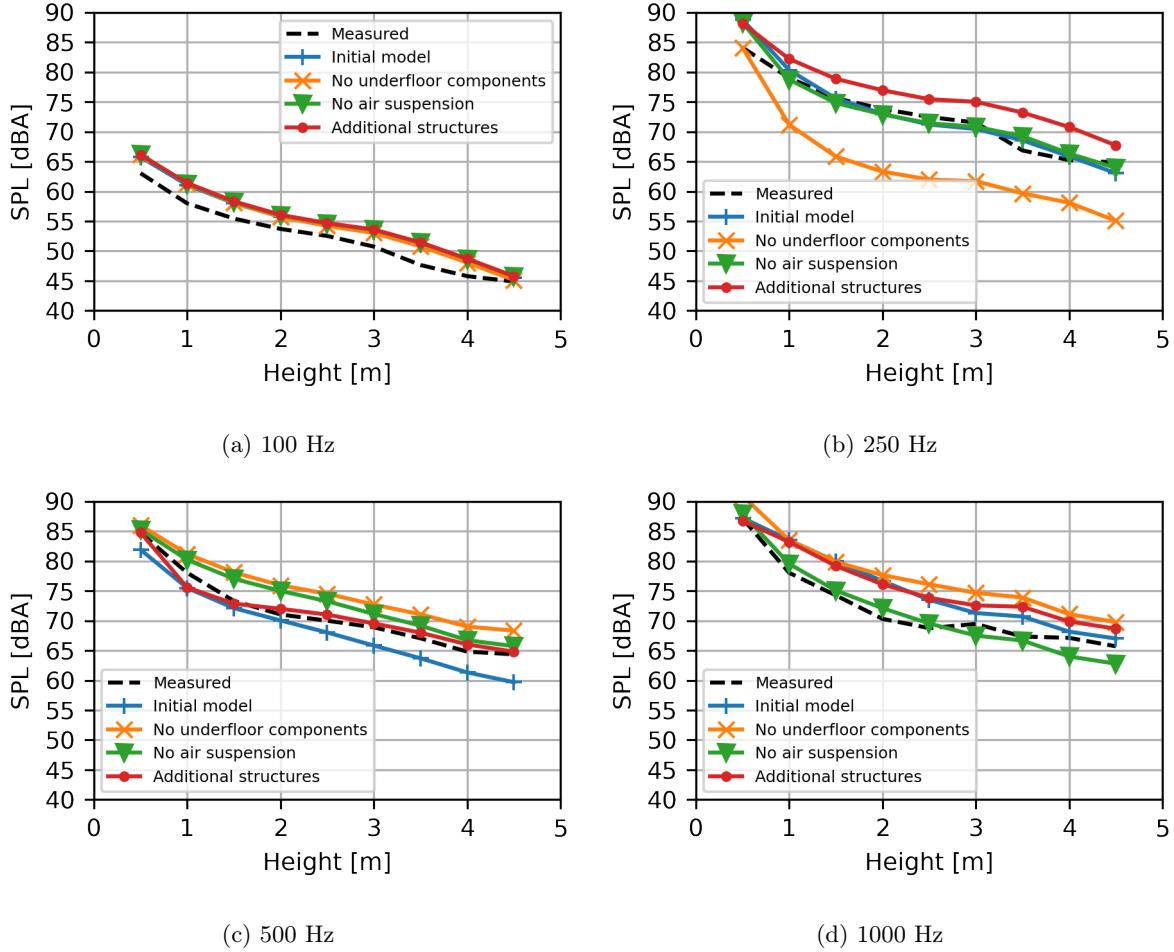


Figure 5.5: Sound distribution at measurement position a, comparison between predictions obtained from the initial finite element model with the measurements. A-weighted SPL in one-third octave bands, dBA ref 20 μ Pa.

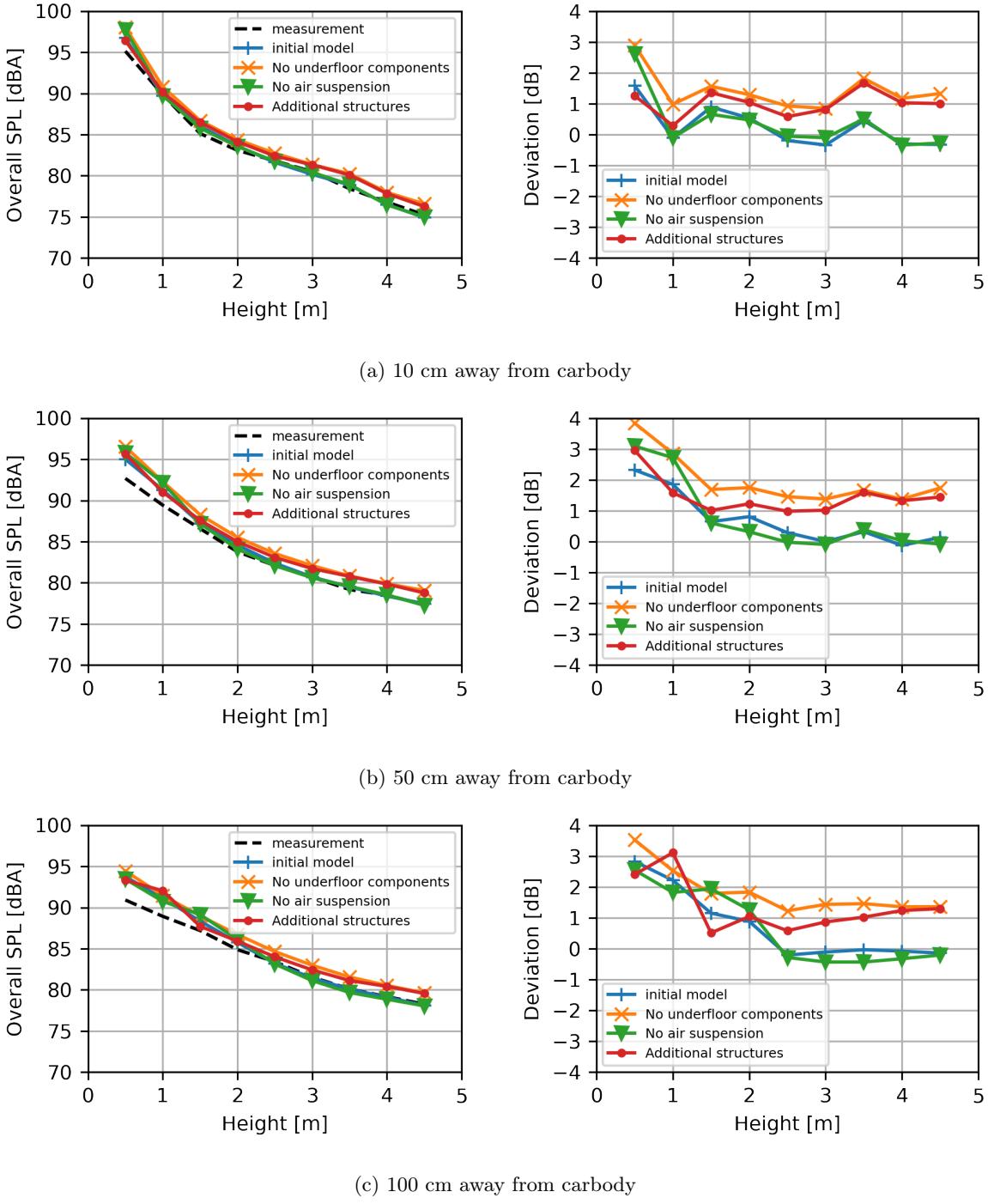


Figure 5.6: Comparison of overall SPL in dBA ref $20 \mu\text{Pa}$. Left: overall SPL as a function of height at various measurement positions (dashed curves: measurement; solid curves: simulation) Right: deviation between simulation and measurement results

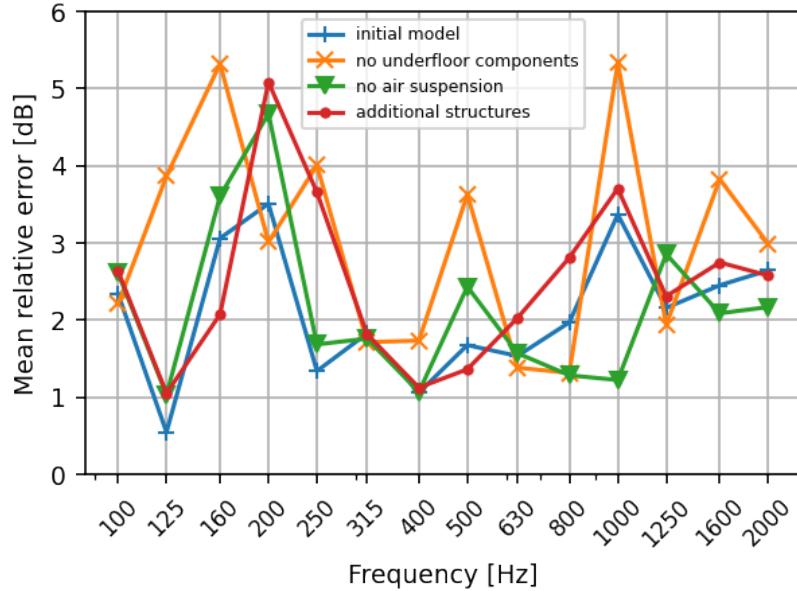


Figure 5.7: Mean relative error of 1/3-octave frequency

Table 5.3: Average gap per frequency band and mean relative error in overall SPL for different geometry variations

Geometry name	Average gap per frequency band (dB)	MRE overall SPL (dB)
No underfloor components	3.12 ± 1.74	1.78 ± 0.91
No air suspension	2.21 ± 1.27	0.85 ± 1.05
Initial model	2.34 ± 1.00	0.73 ± 0.86
With additional structures	2.57 ± 1.38	1.30 ± 0.76

5.3 Effect of ground absorption

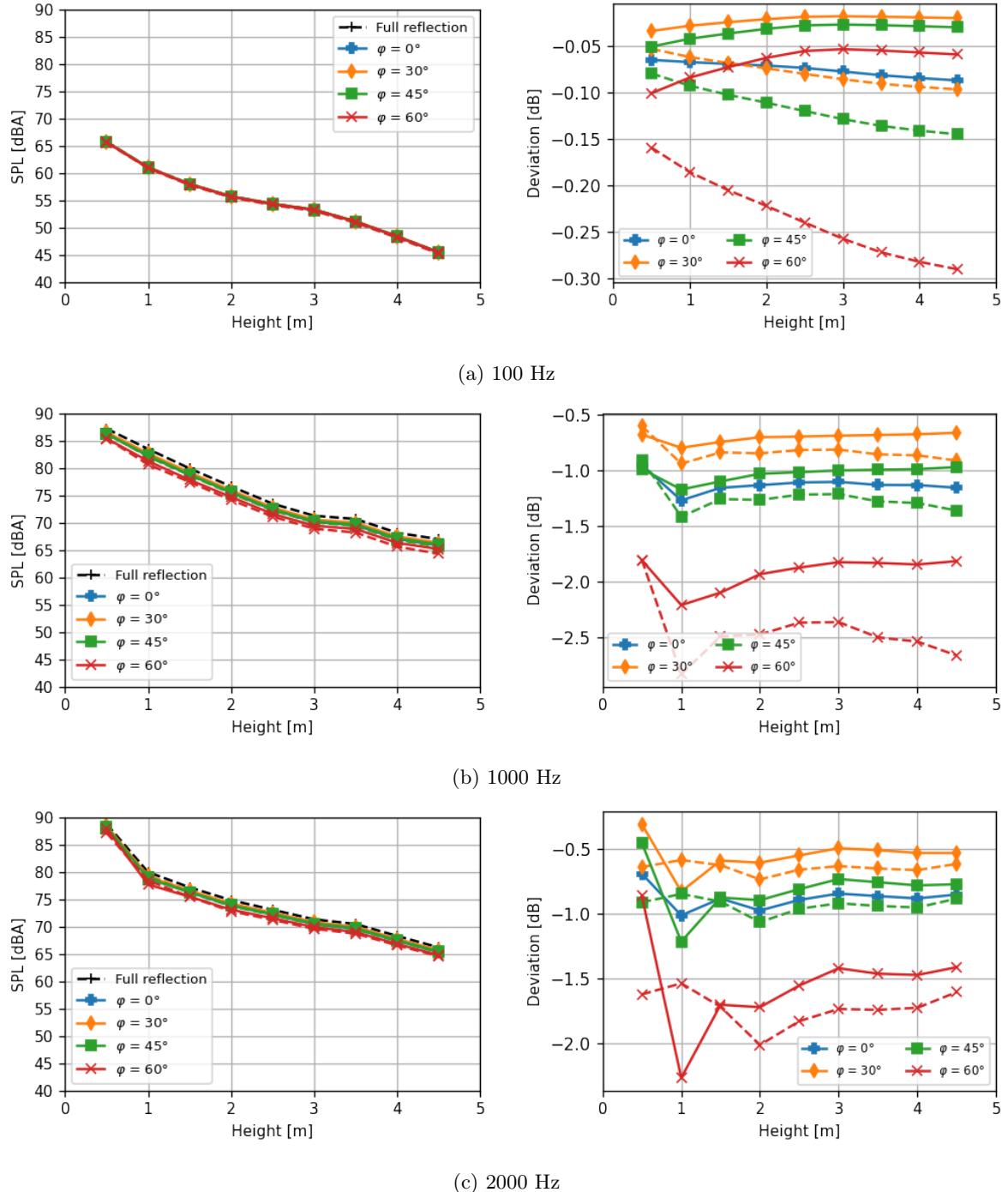
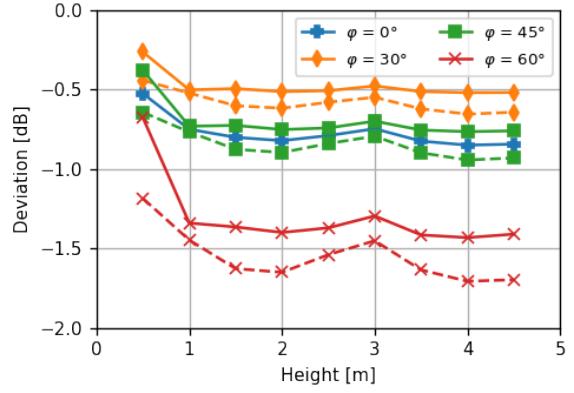
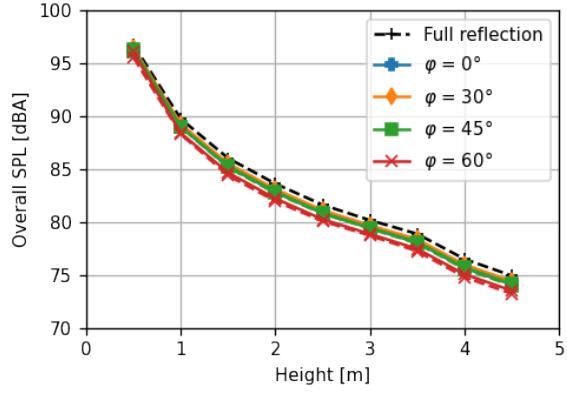
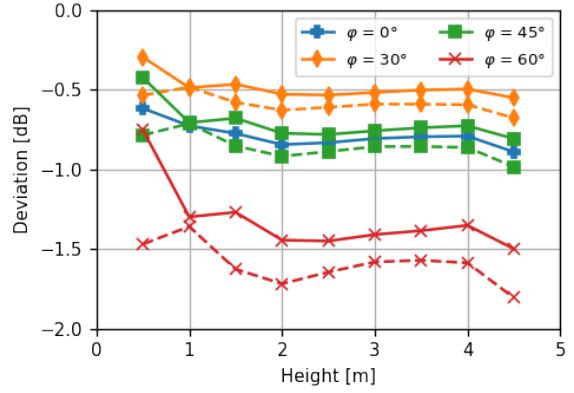
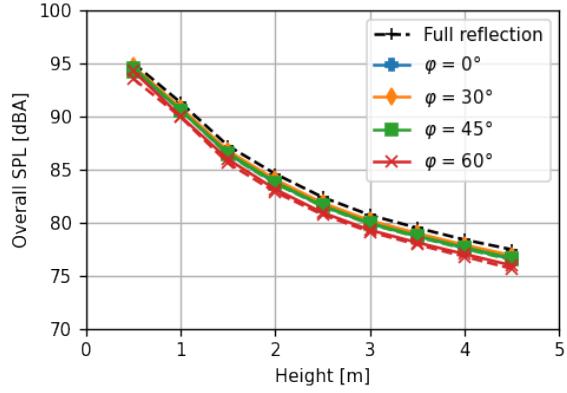


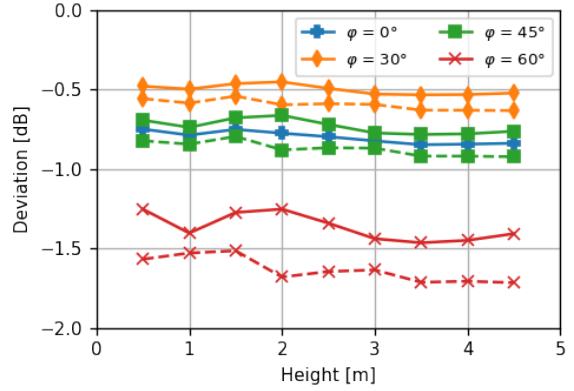
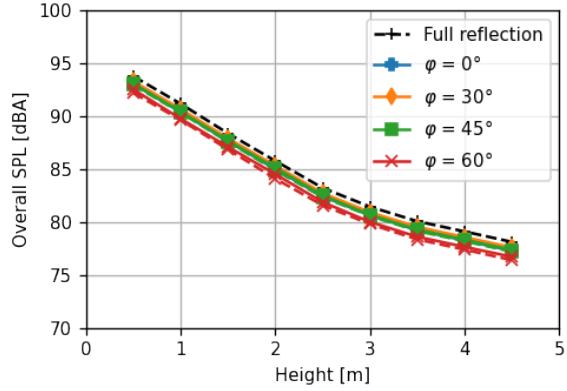
Figure 5.8: Sound distribution at measurement position a, comparison between predictions obtained from the initial finite element model with the measurements. A-weighted SPL in one-third octave bands, dBA ref 20 μ Pa.



(a) 10 cm away from carbody



(b) 50 cm away from carbody



(c) 100 cm away from carbody

Figure 5.9: Sound distribution at measurement position a, comparison between predictions obtained from the initial finite element model with the measurements. A-weighted SPL in one-third octave bands, dBA ref 20 μ Pa.

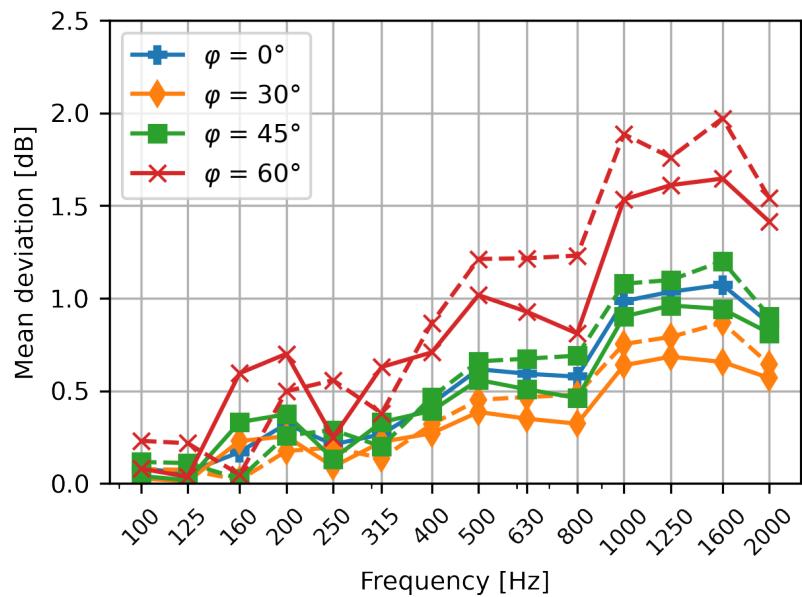


Figure 5.10: Mean relative error compared to full reflective model in 1/3-octave band

5.4 Effect of varying frequency steps per 1/3-octave band

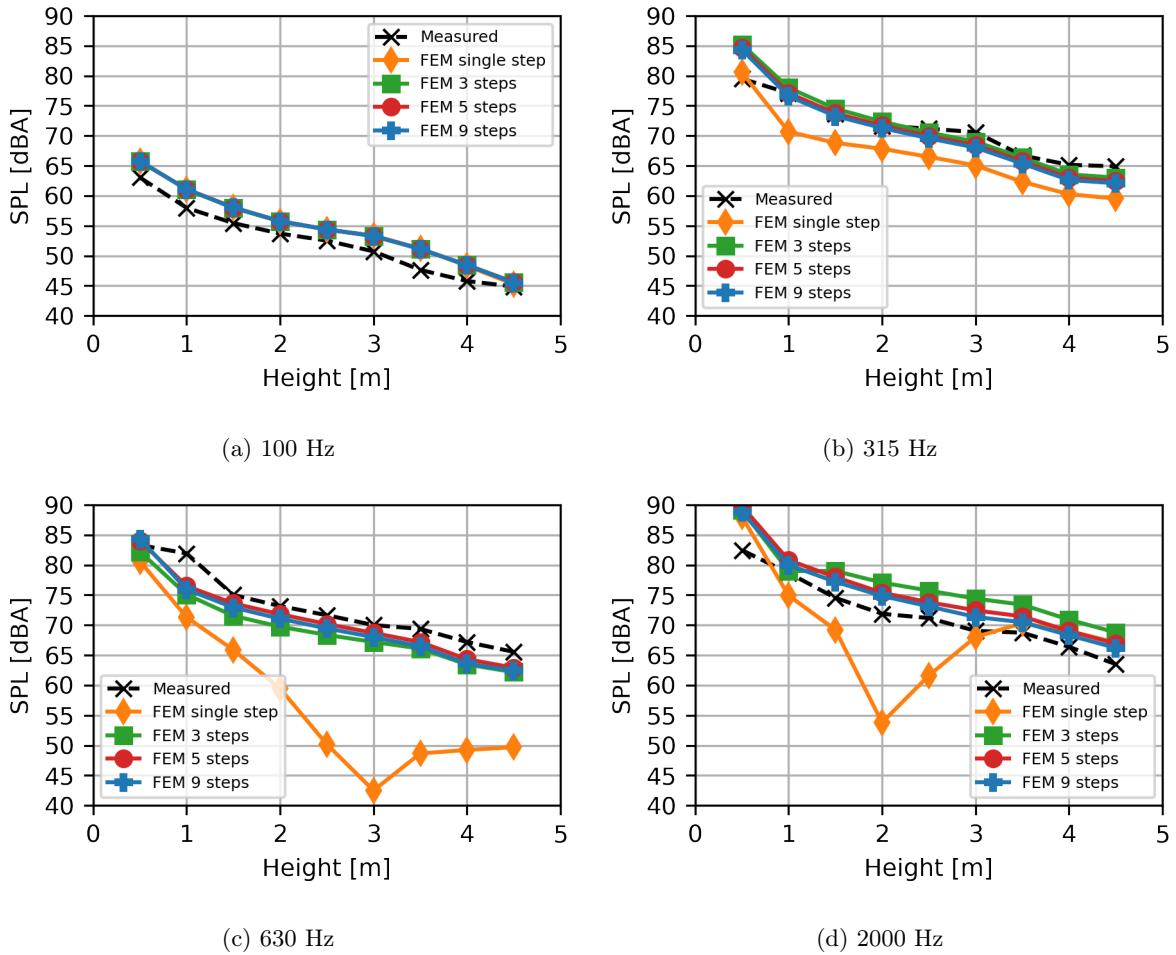
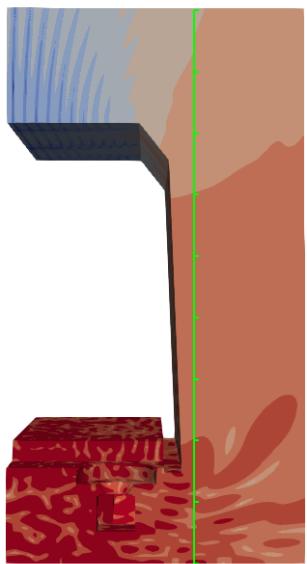
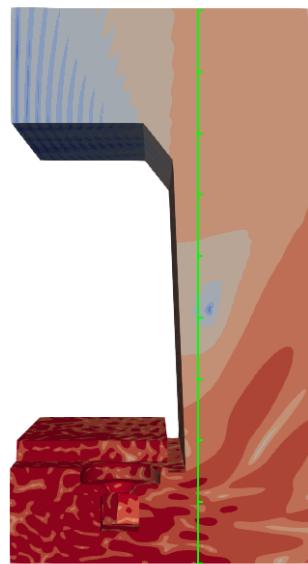


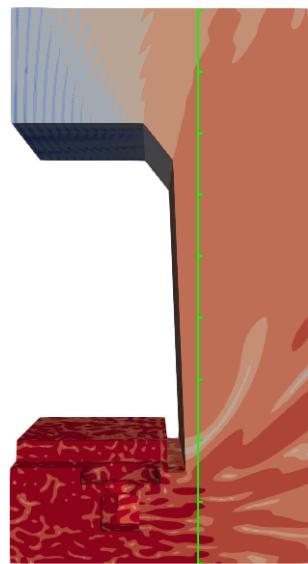
Figure 5.11: Sound distribution at measurement position a, comparison between predictions obtained from the initial finite element model with the measurements. A-weighted SPL in one-third octave bands, dB(A) ref 20 μ Pa.



(a) 1781 Hz



(b) 2000 Hz



(c) 2245 Hz

Figure 5.12: Pressure field of single frequency

Chapter 6

Summary and future works

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