AN EFFICIENT MODEL FOR OFFSHORE PILE DRIVING NOISE TAKING INTO ACCOUNT THE HAMMER CHARACTERISTICS

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Abstract: *Pile driving of state-of-the-art piles for offshore wind farms requires the application of noise mitigation systems to assure that sound pressure levels fulfill official regulations. Currently applied are mainly sound mitigation systems reducing the propagation of the emitted sound, e.g., bubble curtains, rather than reducing the sound generation itself. However, rapidly increasing dimensions of wind turbines with even higher pile diameters demand additional measures to keep sound pressure levels within the defined limits. Therefore, the design of the hammer regarding its acoustic characteristics has recently gained attention.*

To optimize the hammer design, it is required to model the underwater sound pressure caused by the hammer impact. Here, the computation time is crucial to the overall optimization time. However, currently applied models to estimate the underwater sound pressure are either computationally expensive (finite element models) or do not allow for a detailed hammer design (analytical models).

Within this contribution, a computationally low-cost model, which is able to take modifications of the hammer into account, is presented. The model consists of two steps: In the first step, a finite element model is applied to compute the pile head acceleration. In the second step, a transfer function is used to obtain the sound pressure level based on the pile head acceleration.

Keywords: *Offshore pile driving, underwater noise, impact-hammer design, computationally low-cost model*

1. INTRODUCTION

Offshore wind energy is an important source of renewable energy. However, every construction of a new offshore wind park causes high underwater noise levels, threatening to harm marine mammals. Especially crucial is the piling of commonly applied monopiles that are driven into the sea bed using an impact hammer.

In order to protect the marine fauna, several countries have defined official limits for sound pressure levels during pile driving. Noise mitigations systems are usually applied to assure that sound pressure levels do not exceed these limits. However, the capacity of offshore wind turbines is increasing and so are pile diameters. To drive larger piles, more energy is required, thus more noise is emitted and sound mitigation measures at the pile driving location have to be extended. In view of this development, the modification of the sound source itself, i.e. hammer and pile, in order to decrease sound emission, has recently gained attention.

In order to design a hammer with improved acoustic characteristics, it is necessary to be able to estimate the sound pressure levels accurately. Several approaches, to model sound emission and propagation caused by offshore pile driving, exist. An overview can be found for example in [1]. In the context of hammer optimization, it is especially important that the chosen modelling approach includes the detailed hammer design. Furthermore, the calculation of the sound pressure levels has to be fast, to be suitable for an optimization. Considering these requirements, neither analytical models nor a detailed Finite-Element (FE) analysis can be applied here: While analytical models do not allow for a detailed consideration of small changes of the excitation signal, FE models for pile driving noise are simply too time-consuming to be used for multiple computations within an optimization. The present contribution aims to present an alternative model that is both, accurate and fast. The proposed model consists of two steps. The first step constitutes a FE analysis of the short hammer impact. In the second step, results of the FE analysis are applied to estimate sound pressure levels using a transfer function (TF) in the frequency domain.

2. ACOUSTIC OFFSHORE PILE DRIVING MODEL

To simulate the sound generation, emission, and propagation, a two-step model is applied. First, a FE model of hammer and pile is used to evaluate the hammer impact. The axial pile head velocity is then passed to the second model, used to obtain the underwater sound pressure caused by the vibrations of the pile. The second model may be a detailed FE model of the pile, water and sea bed or a TF based on the FE model. The modelling process is shown in Fig. 1. All three models will be shortly introduced in the following sections.

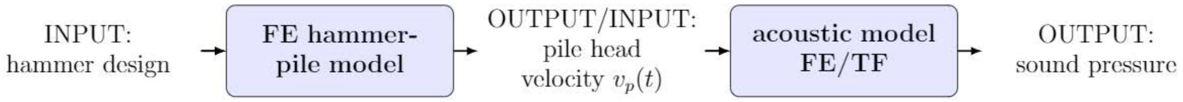

Fig. 1: Two steps to model the sound generation, emission and propagation caused by offshore pile driving. For the second step, either the FE model or the TF may be applied.

Separating the analysis of the impact and the sound propagation into two steps, is a common approach, as used in different ways for example in [2], [3]. In the latter, also transfer functions were discussed. However, in contrast, the present contribution focuses on replacing the second FE model, i.e., the actual acoustic model, with the TF to reduce simulation time during an optimization of the impact hammer.

2.1 FE model for hammer and pile

The first FE model serves to compute the pile head velocity caused by the hammer impact based on the hammer design. It includes only the main hammer components, anvil and impact weight, and the pile. A simplified illustration of exemplary hammer components can be found later on in Fig 3. The influence of water and sea bed on the pile head deformations are negligible and therefore not included in the model.

During pile driving, the anvil rests on the pile head and the impact weight is lifted up to fall afterwards on the anvil which transmits the impact to the pile. This process is repeated several hundred times for each pile, however, only one isolated impulse is modeled. For the purpose of modelling, the ram energy is defined via the initial velocity of the impact weight, here set to 1000kJ.

2.2 FE model for the acoustic propagation

The second FE model serves to compute the acoustic pressure based on the pile head velocity. It includes the pile, water, and the sea bed. The sea bed is modelled with layers with different sound velocities and densities, originally provided in the context of the project BORA for the offshore wind park BARD Offshore I. Further specifications of both FE models can be found in [4], [5].

2.3 Transfer function

Here, we assume that the dependency of sound emission on pile head deformation is at least approximately linear considering its representation in the frequency domain. The proposed TF relates the underwater sound pressure p at one specific position to the axial pile head acceleration a_p obtained from the FE hammer pile model, i.e.

$$TF(r,z,f) = \frac{p(r,z,f)}{a_p(f)}, \tag{1}$$

Here r refers to the distance to the pile, y to the z to the height over the sea bed, and f to the frequency.

To define the TF, the sound pressure values for at least one hammer design have to be known. Hence, although the TF is here presented as an alternative to the acoustic FE model, it is always based on the same and can therefore never fully replace it. The ram impulse with 1000kJ ram energy of the existing hammer MHU3500S, produced by MENCK, was simulated to obtain reference pressure values to define the TF.

3. COMPARISON OF MODELS

In order to discuss the accuracy of the proposed model, two case studies are presented in the following. In both cases, several simulations of the sound emission due to the hammer impact for different hammer designs were performed using the acoustic FE model and the TF to estimate the underwater sound pressure. The difference of the resulting sound pressure levels, ΔSEL and $\Delta\mathrm{SPL}_{\mathrm{Peak}}$, is then used to evaluate the performance of the TF.

For both case studies, a cylindrical pile with 70m length, 6.5m diameter and 80mm wall thickness serves as an example. The embedded length of the pile is 35m and the water depth is

30m. The TF was evaluated in 200m distance to the pile and 2m above the sea bed. The two following sound pressure levels were applied: The sound exposure level (SEL) defined as

$$\mathrm{SEL} = 10\log_{10}\left(\frac{1}{T}\int_0^T \frac{p(t)^2}{p_0^2}\,dt\right)[dB] \tag{2}$$

and the peak sound pressure level ($\mathrm{SPL_{Peak}}$) defined as

$$\mathrm{SPL_{Peak}} = 20\log_{10}\left(\frac{\max(|p(t)|)}{p_0}\right)[dB], \tag{3}$$

where p_0 is the reference pressure of p_0 = 1µPa and T is the reference time of 1s.

3.1 Case study: Variation of the material of the hammer components

The first case study includes 300 variations of the material of the hammer components, i.e. the density ρ and Young's modulus E of both hammer components. Every hammer design was thus defined via the parameter vector: $\theta = [\rho_{Anvil}, E_{Anvil}, \rho_{Weight}, E_{Weight}]$. The samples were randomly distributed and the parameter space was restricted via $\rho_{min} = 0.5\rho_0$, $\rho_{max} = 1.5\rho_0$ and $E_{min} = 0.5E_0$, $E_{max} = 1.5E_0$. The shape and the original material values ρ_0, E_0 of the hammer MHU3500S were used.

The distribution of the error of the TF is shown in Fig. 2. The results show good agreement of both models. Even the largest error of the SEL is smaller than 0.5dB. Errors in the $\mathrm{SPL_{Peak}}$ are larger, up to 2.1dB, but still small considering the difficulty in predicting the $\mathrm{SPL_{Peak}}$. However, since the shape of the hammer components in the case study was the same as used to define the TF, the error might be unrealistically small for other hammer designs.

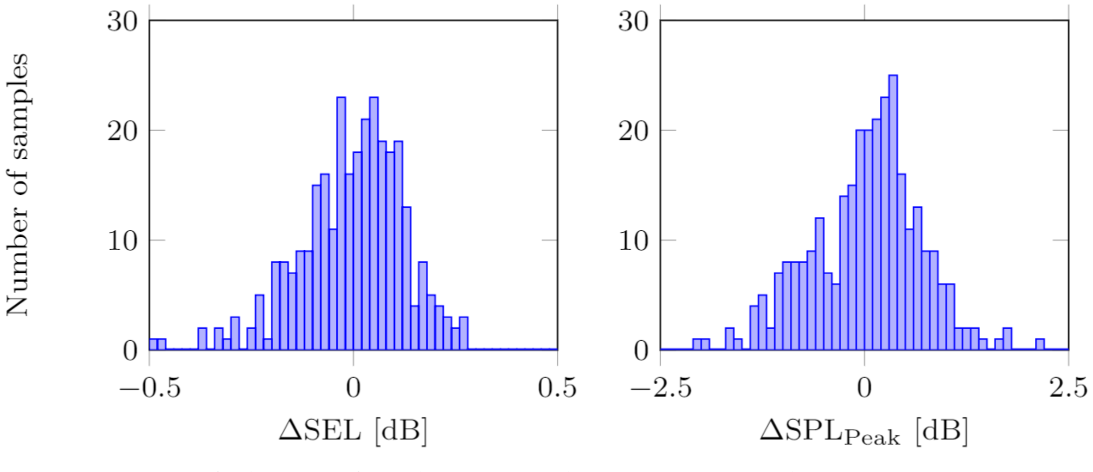

Fig. 2: Error of the TF for the material parameter study: $\Delta SEL = SEL_{FEM}$ - SEL_{TF} $\Delta SPL_{Peak} = SPL_{Peak,FEM}$ - $SPL_{Peak,TF}$

3.2 Case study: Variation of the shape of the hammer components

In contrast to the previously presented case study, this section discusses the performance of the TF for different shapes of the hammer components. For this purpose, the cross section of the shape of the hammer components was defined via a polygon. A visualization of exemplary hammer components defined as polygons is shown in Fig. 3. Axis symmetry was assumed. The shape was described by 13 parameters. 198 samples were randomly distributed. The parameter space was restricted by the maximum height h and diameter d for each component, i.e. $h_{anvil} = 2m$, $d_{anvil} = 8m$ and $h_{weight} = 15m$, $d_{weight} = 4m$. The corresponding differences in sound pressure levels are shown in Fig. 4.

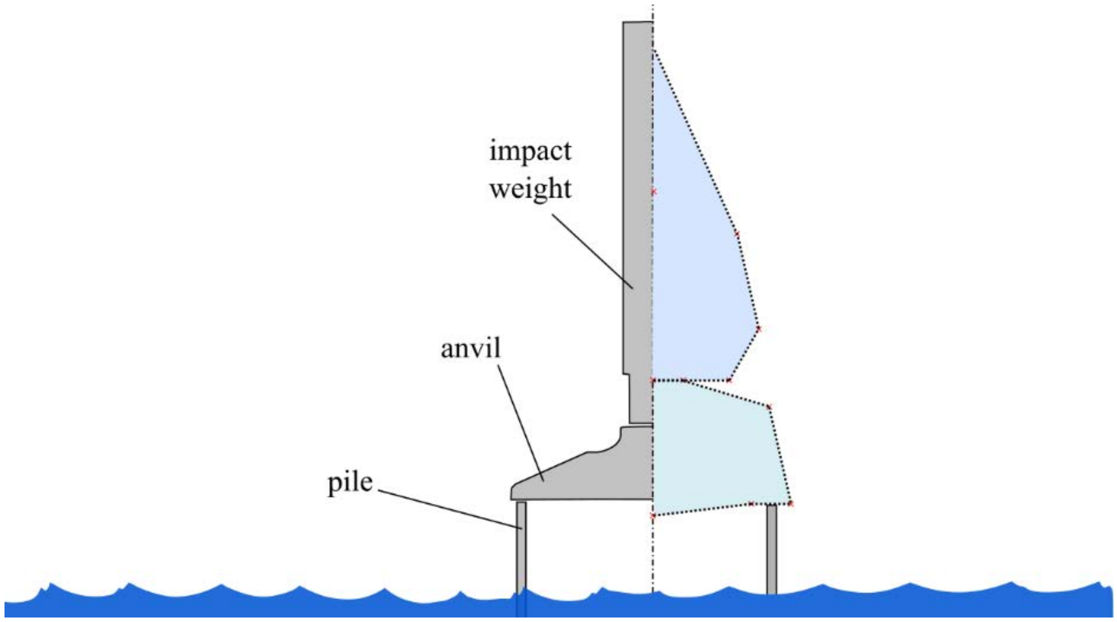

Fig. 3 A simplified illustration of the axis symmetric original hammer components at the left side and one example of the polygon parameterization at the right side. The pile is here approximated as a hollow cylinder.

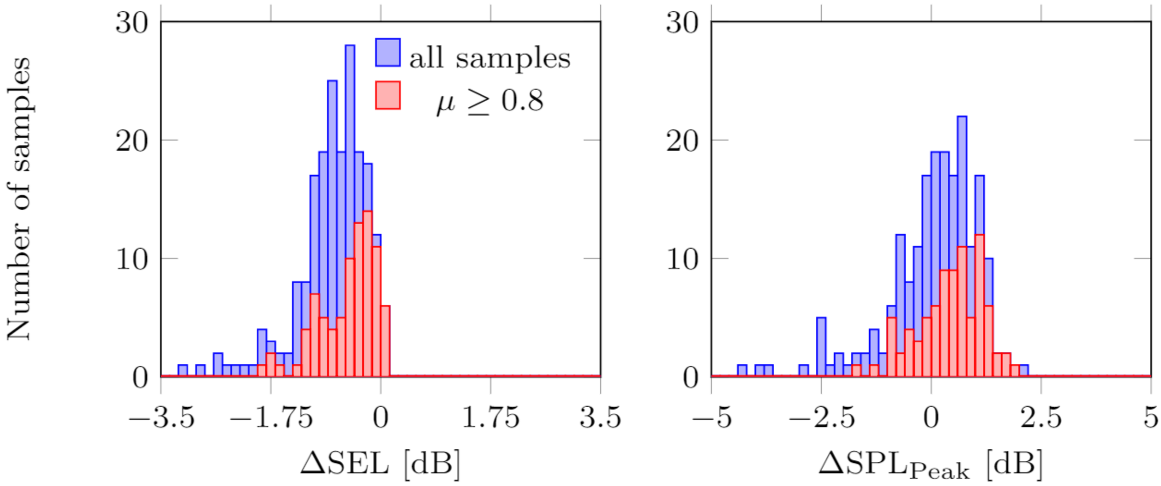

Fig. 4: Error of the TF for the parameter study of the shape of the hammer components: $\Delta SEL = SEL_{FEM} - SEL_{TF}$ $\Delta SPL_{Peak} = SPL_{Peak,FEM} - SPL_{Peak,TF}$*. Red bars refer to the error distribution, considering only hammers with an efficiency* μ *of at least 0.8.*

In comparison to the previous case study, errors are larger, but still indicate general good agreement of the TF and FE model as no extreme outliers occurred. If only hammers with a driving efficiency of at least 0.8 are considered, the maximum error is even less than 2dB for both sound pressure levels, the SEL and the $\mathrm{SPL}_{\mathrm{Peak}}$. However, even deviations of only 2dB might hinder an optimization, if it affects the comparison of the acoustic characteristics of hammer designs.

4. CONCLUSIONS

The estimation of sound pressure levels based on a linear TF in the frequency domain were discussed. The use of the TF for two exemplary case studies showed overall promising results and therefore support the approach of a linear TF in the frequency domain. However, the results of the second case study indicate that the here applied reference pressure based on an existing hammer might cause larger errors if the hammer of choice is too different from the original one. Future work will be directed to model the sound emission and propagation in the frequency domain to obtain a more accurate TF.

5. ACKNOWLEDGEMENTS

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