Validation of the Time-Domain Boundary Element Method in Acoustics regarding Flexible Multibody Simulations and Acoustic Measurements

Simon Schneider¹, Bernd Graf¹, Timo Giese², Ivaylo Haralampiev²

¹University of Applied Sciences Ulm, 89075 Ulm, Germany
²FunctionBay GmbH, 80339 Munich, Germany

Abstract. With flexible multibody simulation, the dynamical behaviour of vehicle components as well as entire systems can be simulated in an efficient way. In addition to structural vibrations, noise emissions play an important role in terms of comfort in the vehicle industry. While the dynamics of the structures can be simulated directly in the time-domain, the sound radiation is usually calculated in the frequency-domain. This process enables an efficient investigation of steady-state operating conditions but not of transient operating conditions. The time-domain Boundary Element Method (TD-BEM) in acoustics allows to determine the sound pressure directly in the time-domain, without complicated transformation from the frequency-domain. Existing work concentrates on stability criteria and general theoretical applicability. The validation on real test cases with measurement results, however, has rarely been focused. On this work, we aim to close this gap between theory and practical applications with the help of real housing structures and test cases. In order to set up an acoustic validation, meaningful measurements need to be combined with validated structural simulations that match the physical world with regard to the sound radiation as closely as possible. Therefore, several test set-ups are used to measure the real dynamic and acoustic behaviour in an anechoic chamber with multiple accelerometers, a laser vibrometer, various microphones and a sound intensity PU probe. Based on these measurements, multibody and BE-models are created, representing the same scenarios with numerical methods. Derived from the simulated surface accelerations, the acoustic radiation is calculated in the time-domain. Comparisons at a simple oval-shaped housing show that the method generates very promising results. The mean deviation between the measured and simulated sound pressure levels (time-domain) is less than 1 dB and the frequency spectra show a high quantitative correlation as well.

1. Introduction

Nowadays, main aspects such as costs, quality and time must be taken into account during the early product development in order to produce a promising product. This is summarized by the iron triangle of product development [1]. According to the Rule of Ten, error detection also plays an enormous role in the early R&D phase [2]. That is the reason why simulation methods are used more and more in mechanical engineering. In the vehicle industry, driving comfort plays a significant role for the customers choosing an automobile. Therefore machine acoustics is an important part of today’s engineering disciplines [3].
In the field of mechanical engineering, vibrating structures are responsible for sound radiation. This is why acoustic considerations of components are often coupled with dynamic ones. In recent years, flexible multibody simulation is used to investigate the dynamic behaviour of structures. A practicable approach is to employ flexible bodies in order to consider the dynamic behaviour of entire systems directly in the time-domain. This leads to a reasonable amount of computational resources.

If a numerical acoustic analysis is coupled with a structural vibration analysis, the calculation usually takes place in the frequency-domain. This is an efficient way to examine steady-state but not transient operating conditions for exterior acoustic problems.

The time-domain Boundary Element Method enables determining the sound parameters, like the acoustic pressure $p$, directly in the time-domain by solving the boundary integral equation problem (1) [4].

$$\epsilon p(x_p, t)4\pi = \int_{\partial\Omega} \left[ q(\hat{x}, t_{ri}) + \frac{\partial \hat{v}}{\partial n} \left( -\frac{p(\hat{x}, t_{ri})}{r^2} - \frac{1}{cr} \frac{\partial p(\hat{x}, t_{ri})}{\partial t} \right) \right] dS(\hat{x})$$

The first term of the integral equation (1) describes the monopole source of an acoustic radiation. The second term, which derivative the distance vector $\hat{r}$ with respect to the normal direction $\vec{n}$, describes the dipole source. The dipole component considers the reflection of incoming waves. A practicable approach that seems promising for complex real structures is the solution of the discretized equation (1) using the collocation method [4].

In most publications, this method is examined for stability criteria only or there are shown some theoretical applications. The comparison with real test structures usually plays a minor role or no role at all. Therefore, this paper presents a quantitative comparison between results of the TD-BEM method in acoustics and physical measurements.

The workflow to validate an acoustic calculation is shown in Figure 1. In the modal analysis step, an initial comparison between the real test body and the associated FE model is conducted using a numerical (NMA) and experimental (EMA) modal analysis. The aim is the best possible correlation between the physical and the simulation model regarding their modal parameters. The dynamic behaviour of the excited test object is determined by the vibration analysis. This is done on the one hand with measurements from the test bench and on the other hand with the flexible multibody simulation describing the test set up. Similar to the modal analysis, the goal is to ensure the simulation model matches the real measured values as closely as possible. The damping and the boundary conditions such as contacts and excitations play a major role in this process.

Only with a very good validation of the calculated surface vibrations by measurement, it is possible to ensure that the boundary conditions of the sound radiation of the physical and numerical model are equal.

The final step is to use the TD-BEM to compute the acoustic sound pressures directly in the time-domain in order compare these results with the measured signals.
2. Test object and experimental setup
A thin-walled oval principle housing (OPG) made of cast aluminium is used as a test body, shown in Figure 2. The simple design holds the complexity of the geometry and the boundary conditions at a manageable level. For these investigations the OPG represents a simplified gearbox housing.

To validate the operational vibration and acoustic analysis, a test bench is set up in a state-of-the-art anechoic chamber. In the anechoic chamber, it is possible to achieve free field conditions in the frequency range under survey and external acoustic disturbances are prevented.

The OPG is placed on a foam with very low stiffness. Below the foam is an aluminium plate, which first eigenfrequency is above the highest OPG natural frequency to be examined. This plate is fixed with rubber elements on a steel frame with sand filled tubes. The foam, the aluminium plate and the steel frame decouple the test object from the environment.

Five uniaxial miniature accelerometers are attached at the housing to measure the dynamic behaviour during operation in normal direction at different structure points. The sensor positions are selected in a way that the structural modes and the fluid modes of the air inside can be recorded well.

During the measurements of the OPG the sound pressure near the surface as well as in the far field is recorded with 12 microphones. In addition, the measurement points of the microphones are chosen in a way that as many modes as possible in the frequency range of interest can be detected.
3. Modal analysis
To validate the simulation model at the level of natural structural vibrations, the modal model of the OPG was got by an experimental modal analysis under free-free boundary conditions and the roving hammer method.

In order to minimize possible geometrical deviations of the simulation model, the real test object was digitized using a 3D scanning process. Based on this, the scan data was expanded to an FE mesh and the numerical modal analysis of the digital mock-up was carried out. The correlation between these two modal models is shown in the MAC matrix in Figure 3.

The first 39 natural dynamic modes in the frequency range up to 2500 Hz were examined. In this region, the maximum frequency deviation is about 2 % and the mean frequency deviation is reduced to 0 % by optimizing the E-modulus. The correlating mode pairs with an average MAC-Value of 92% [5] are a good basis for further steps of the analysis.

4. Vibration analysis
The aim of the vibration analysis is to validate the dynamic behaviour of the numerical model with the help of measurements from the physical test body. In this step, we evaluate the surface accelerations of the oval principle housing which represent the boundary conditions of the acoustic analysis.
4.1. Experimental transient vibration analysis

During the experimental transient vibration analysis, the housing is excited by an impact hammer, which includes an integrated uniaxial force sensor.

This results in a pure normal oriented force, which doesn’t change the boundary conditions of the OPG.

The polyamide tip of the hammer primarily excites the lower frequency range (up to approx. 2000 Hz). The recorded force signal of the measurement is shown in Figure 4.

![Figure 4: Force-Signal of stimulation](image)

To ensure a strong dynamic answer over the first 39 modes, the excitation point is located in an area where most of the mode shapes show a high amplitude. The modal damping values are extracted for this test set-up by a modal analysis of the frequency response functions of the accelerometers and forces. These modal damping factors are used to update the simulation model (see Figure 1) taking into account the test bench boundary conditions.

4.2. Flexible multibody simulation

Based on the FE model [5], a reduced flexible body (RFI) is created for a multibody simulation in the MBS-software RecurDyn. The RFI includes the foam, representing the only boundary condition of the OPG.

To consider the foam, 25 circular node sets were created evenly distributed over the flange, which are connected to CBush elements via RBE3 elements, shown in Figure 5. The ends of the CBush elements are fixed to the ground and their stiffness depends on the measured modal results. The sensor masses of the acceleration sensors are also taken into account with mass points on the structure's outer surface.

This modelling technique allows to apply the individually measured damping ratios of the rigid body modes of the RFI.
The modal damping values of the measured elastic mode shapes were also assigned directly to the correlating numerical ones. Poorly correlated or unassigned analytical modes got their damping from a linearly extrapolated damping function as shown in Figure 6.

A spline curve, based on the measured force signal from Figure 4, is used as the analytical force signal of the MBS model.

4.3. Correlation of the transient dynamic behaviour
To evaluate the vibration simulation results, the calculated accelerations at the sensor points are compared with those of the measurement.

In the time-domain (see Figure 7), the experimental and numerical results fit very well over the complete simulation time of 1 s. For a more detailed insight, the observed frequency spectra are also of great interest. Figure 7 shows a superb correlation until 2 kHz. Above, the results show slight deviations, but still match the general trend of the amplitudes.

Figure 5. Flexible Body of the OPG

Figure 6. Modal damping of the RFlex body
The comparison of the simulation regarding the remaining four acceleration sensors and the laser vibrometer are of the same high quality. The absolute mean deviations with regard to the level values in the time range between measurement and simulation are noted in Table 1.

### Table 1. Mean deviation of the accelerations between measurement and simulation

| Type     | Point 56 | Point 116 | Point 152 | Point 184 | Point 241 | Point 282 |
|----------|----------|-----------|-----------|-----------|-----------|-----------|
| Direction | X+       | X+        | Y-        | Z-        | X-        | X-        |
| Mean dev. [dB] | 0.6    | 0.2       | 0.4       | 0.1       | 0.5       | 0.4       |

Due to the high level of accordance, the dynamic multibody simulation is considered to be validated successfully. This basis can be used for the following acoustic simulation with respect to the surface vibrations i.e. the boundary conditions of the acoustic radiation.

### 5. Acoustic analysis

To solve the discretized form of the boundary integral equation (1) Stütz’s approach [4] is followed by using the collocation method. The housing surface is meshed with linear two-dimensional elements. Further, their centres are defined as collocation points and the systems of equations at the collocation points are solved with the help of the mesh geometry [4].

#### 5.1. Acoustic BEM-Mesh

The mesh for the numerical acoustic analysis is derived directly from the outer surfaces of the FE body. With the approach of the linear elements it is possible to eliminate the strong singularity part in the discretized form of (1). The choice of the average element edge length is also based on Stütz. He
describes the advantages in terms of stability if the mesh fulfills a Helmholtz number (2) smaller than 0.2 and a relationship (3) between the time and space discretization higher than 0.8.

\[ H_e = \frac{l_{el}}{c_{air}/f_{max}} \]  \hspace{1cm} (2)

\[ \beta = \frac{c_{air} \times \Delta t}{l_{el}} \]  \hspace{1cm} (3)

The surface, shown in Figure 8, is discretized with an average element edge length of \( l_{el} = 12 \) mm. The highest frequency to be examined is \( f_{max} = 2500 \) Hz, the simulation step size \( \Delta t = 1/25600 \) s and the speed of sound \( c_{air} = 343 \) m/s. From that, the following mesh parameters can be derived:

- \( H_e = 0.09 \)
- \( \beta = 1.1 \)

5.2. Boundary conditions

The vibrations of a structure’s outer surface represent the boundary conditions of the acoustic simulation. These accelerations are required for each BEM element (i.e. collocation point) with respect to the structures normal direction. The element accelerations are responsible for the monopole radiation part in the boundary integral equation (1).

Since the aluminium plate under the OPG (Chapter 3) is designed to be overcritical, its vibrations are negligible. Thus, the elements of the closed lower boundary surface consider no dynamic behaviour in the numerical acoustic analysis. The accelerations of the outer surface (the green region in Figure 8) are defined with the calculated results from Chapter 4.

Finally, the acoustic pressure at the outer surface and at the observation points (i.e. at the collocation points) can be calculated with the computed accelerations (Chapter 4) of the structure.

6. Validation of the time-domain Boundary Element Method

At the validation step, the measured and calculated sound pressures of the microphones are compared and evaluated. The results for microphone 1 are depicted in Figure 9. This individual measurement point is located 0.5 meters away from the test object.
It is impressive how the signals match in the time-domain as well as in the frequency-domain. The RMS - sound pressure level curves are almost identical over the entire observed time range. For a more detailed insight, the data is transformed into the frequency-domain. Especially the peaks at the natural frequencies are highly similar as a result of the high correlation degree of the time functions. More significant differences only exist in the frequency range above approximately 1800 Hz. This is justified by the weak excitation of the hammer in this region. Hence, the sound pressure levels are completely below 40 dB in this frequency range.

With respect to the mean deviations of the RMS values in the time-domain, the correlations of the remaining 11 microphones are comparatively good, as shown in Figure 10.

In a further experimental and numerical investigation, ten of the twelve microphones are positioned at least 1 meter away from the OPG. This separate experiment focuses on the validation of the acoustic results in the far field. Equally good correlations of the signals can be observed. It turns out, however, that in these investigations the random noise of the microphones has a very strong influence on the measured sound pressures. The reason is the low SPL, which declines quadratic to the distance reducing...
the signal and noise ratio. Therefore, higher excitation levels with a shaker are planned for far field analysis.

Note, that the acoustic simulation is set up with a relatively fine BEM mesh, because this is a first comparison to assess the method in a practical application. This choice is useful in terms of the complexity of the method. For application in development processes, possibilities will be examined in order to reduce the complexity of the mesh.

7. Conclusion
This paper demonstrates the complex process up to the acoustic validation of the TD-BEM calculated results with the help of physical measurements. The numerically calculated sound pressure values of this method are very similar to the measured ones, in the time-domain as well as in the frequency-domain. Within an examining time of one second, the absolute mean error between the calculated and the measured sound pressure level ranges between 0.1 and 0.65 dB. Above all, the curves as well as the peaks of the numerical and the physical sound pressure level in the frequency range are highly similar.

With the TD-BEM method in acoustics, both the radiation from structural vibrations and the reflection of incoming waves are considered. Therefore, the acoustic signal at one point is a superposition of all acoustic signals of the structure. Thus, it is assumed that the validation of the acoustic signals at the analysed observation points are representative for the complete structure. Final considerations of the results shows that the TD-BEM method coupled with a validated flexible multibody simulation delivers good results.

Further investigations will show the applicability of the TD-BEM at steady-state and critical operating points such as the influence of a complex ribbed structure geometry. According to Stütz, the numerical eigenfrequencies of the BEM mesh can lead to instability of the method. The next examinations will show the influence of this effect on practical structures and vibrations with regard to the calculated sound radiation.

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