Hybrid NVH Simulation for Electrical Vehicles III – Acoustic Model

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Introduction

The creation of a fast simulation tool for the NVH in-cabin auralization of electrical drive vehicles demands short calculation times throughout the whole simulation chain. As part of the FVA research project No. 682, the Institute of Electrical Machines (IEM) and the Institute for Machine Elements and Machine Design (IME) of the RWTH Aachen University have implemented fast simulation methods for the force excitation of the electrical machine ([1]) and its subsequent structural dynamic transmission ([2]) for frequencies up to 5kHz. These two simulation stages calculate the resulting velocities and forces on the surface of the drive train and at its coupling points with the car structure based on the electrical input of the machine. It is the task of the acoustical model to generate a fast binaural in-cabin auralization from those calculated values. Fig. 4 depicts the structure of the research project, starting with the force excitation simulation and ending with a binaural auralization.

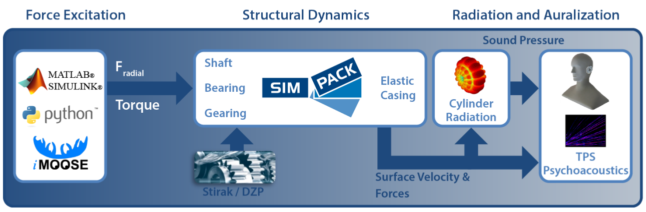

Figure 1: Structure of the FVA research project No. 682.

The forces and surface velocities are transformed into radiated sound pressure and structural velocities and fed into binaural transfer paths to generate the in cabin signals. While conventional methods such as BEM simulations deliver accurate results for the calculation of the radiated sound pressure from the surface velocity of a body, they usually require long computational times and thus are not suitable for a fast simulation chain. As a solution to this problem, the extension of an analytical approach for the radiation calculation ([3]) is developed.

The complete and accurate numerical simulation of all transfer paths in a vehicle is also a slow and error-prone procedure. Since the focus of the project is on the variability of the drive train simulation and not of the adjacent vehicle structure, measuring the required air and structure borne transfer paths between the engine compartment and the passenger cabin is the preferable alternative. The resulting binaural signals offer the full possibilities of objective and subjective audio signal analysis. The simulation methods, measurements and examples for the analysis are presented in this paper.

Interfaces

The preceding stages of the simulation chain yield the forces and velocities on the surfaces of the vibrating bodies of the drive train and at their coupling points with the vehicle structure. The data is forwarded to the acoustical model through two separate interfaces: one for airborne and one for structure borne sound. The interfaces are described in the following.

Airborne Sound

The maximum regarded frequency of the simulation chain is 5kHz. On cylindrical bodies, only velocity modes of the same or larger wave length contribute to the radiated sound pressure [4]. Therefore the regarded structural wave length can also be limited. Two sampling points or more per wave length are required for the correct description of the velocity modes, resulting in 216 radial and axial surface velocity sampling points on the exemplary electrical drive train, as depicted in Fig. 4.

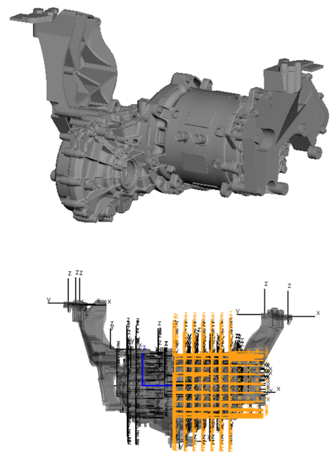

Figure 2: Electrical drive train model (top) with sampling point distribution (bottom).

Structure Borne Sound

The drive train has 3 coupling points with the vehicle structure. Considering a coupled mechanical system as shown in Fig. 3, the source velocity can be described as

$$\mathbf{v_f} = \mathbf{Y_s} \cdot \mathbf{F_b} \tag{1}$$

using the source admittance Y_s and the blocked force F_b exercised by the source. The velocity in the adjacent structure can be described as

$$\mathbf{v_r} = \mathbf{Y_r} \cdot \mathbf{F_r} \tag{2}$$

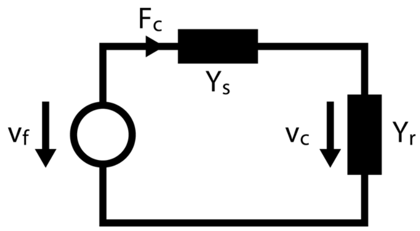

Figure 3: Coupled mechanical system.

with the admittance of the structure Y_r and the force F_r acting on it. The force F_c of the coupled system is then described by ([7])

$$\mathbf{F_c} = (\mathbf{Y_s} + \mathbf{Y_r})^{-1} \cdot \mathbf{v_f} = (\mathbf{Y_s} + \mathbf{Y_r})^{-1} \cdot \mathbf{Y_s} \cdot \mathbf{F_b} \quad (3)$$

where the blocked force F_b is the result of the structural dynamics simulation and F_c is the input for the subsequent structure borne transfer paths. As a consequence, the admittances Y_s and Y_r have to be measured. Given the fact that the elastomer dampeners between the drive train and the vehicle structure could not be characterized in the course of the project, $Y_s \to \infty$ assumed, resulting in $F_b = F_c$. Thus, the simulated forces are directly fed into the structure borne transfer paths.

Airborne Radiation

To connect the simulated velocity values with the airborne transfer paths the radiated sound pressure caused by the surface vibration of the drive train needs to be calculated. As can be seen in Fig. 4 the surface of the drive train mostly consists of two surface classes: cylindrical surfaces and piston-like surfaces. As a means to bypass BEM simulations analytical radiation models for cylinder elements and piston-like plates are implemented. The resulting sound pressure values of both models are superposed to calculate the resulting total sound pressure.

Cylinder Model

Based on the solution of the wave equation in cylindrical coordinates the radiated complex sound pressure of a cylindrical body at a given point in cylindrical coordinates

$$\underline{p}(r,\varphi,z) = \frac{1}{2\pi}\sum_{n=-\infty}^{\infty} e^{jn\varphi}\int_{-\infty}^{\infty} \overline{\underline{V}}_n(r_0,k_z)\frac{\underline{H}_n(k_r r)}{\underline{H}'_n(k_r r_0)}e^{jk_z z}dk_z \quad (4)$$

can be calculated ([4]). Herein,

$$\overline{\underline{V}}_n(r_0,k_z) = \int_{-\infty}^{\infty} \underline{v}(r_0,z)e^{-jk_z z}dz. \quad (5)$$

is the surface velocity spectrum ([4]), which is a two-dimensional transformation of the normal velocity $v(r_0, z)$ on the cylindrical surface at the surface radius r_0. Using the stationary phase approximation ([5] the integral in Eq. 4 can be approximated in cylindrical coordinates. The result is the complex sound pressure

$$\underline{p}(R,\theta,\alpha) \approx \frac{\rho_0 c}{\pi}\frac{e^{jkR}}{R}\sum_{n=-\infty}^{\infty}(-j)^n e^{jn\alpha}\frac{\overline{\underline{V}}_n(r_0,k\cdot\cos(\theta))}{\underline{H}'_n(kr_0\sin(\theta))\sin(\theta)} \quad (6)$$

for points on a spherical surface with the radius R around the source. Comparisons with BEM simulations as depicted in Fig. 4 show a good compliance for the cylinder model, considering the required computation time.

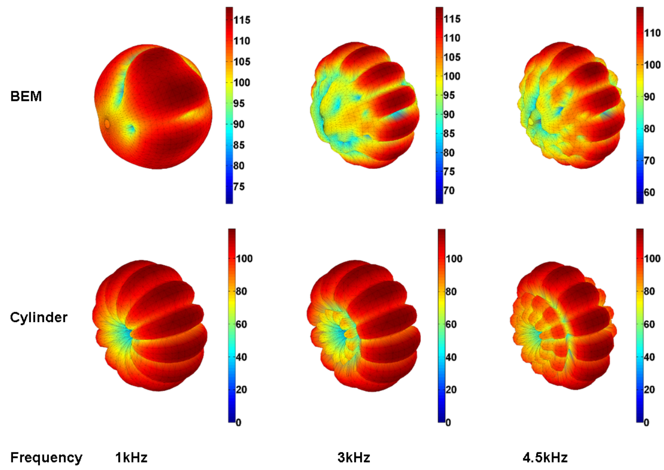

Figure 4: Comparison between BEM (top) and cylinder model (bottom) results (left) for the 6th circular and 0th axial mode at various frequencies.

Piston Model

The on axis radiated sound pressure by a circular piston moving with the velocity $\hat{v}$ at a given distance r is denoted as ([6])

$$p(r,t) = \rho_0 c\hat{v}e^{j\omega t}\left(e^{-jkr} - e^{-jk\sqrt{r^2+a^2}}\right). \quad (7)$$

Using the directivity factor

$$R(\theta) = \frac{2J_1(ka\sin\theta)}{ka\sin\theta} \quad (8)$$

with the angle θ relative to the main axis it is possible to calculate the sound pressure at any given point in front of the piston.

Transfer Paths

The goal of the simulation chain is to generate binaural sound pressure signals at the driver position for the assessment of the drive train noise. Therefore, the calculated sound pressure, forces and velocities are fed into measured binaural transfer paths.

Airborne Transfer Path

The airborne transfer paths are measured in a half-anechoic chamber, using a directional loudspeaker in the engine compartment and a dummy head on the driver seat. The drive train is removed for the measurements. The directional loudspeaker is moved along a spherical surface around the original position of the drive train directed in the surface normal direction, matching the simulated positions of the radiation simulation.

Structure Borne Transfer Path

The structure borne transfer paths are also measured in a half-anechoic chamber. The drive train is removed for the measurements. An impulse hammer is used to excite the three coupling points in all three axes. Triax-accelerometers at the input positions record the velocity, a dummy head at the driver position records the binaural sound pressure. Fig. 5 shows the setup with the dummy head in the passenger cabin, as well as some accelerometer positions and the impulse hammer excitation.

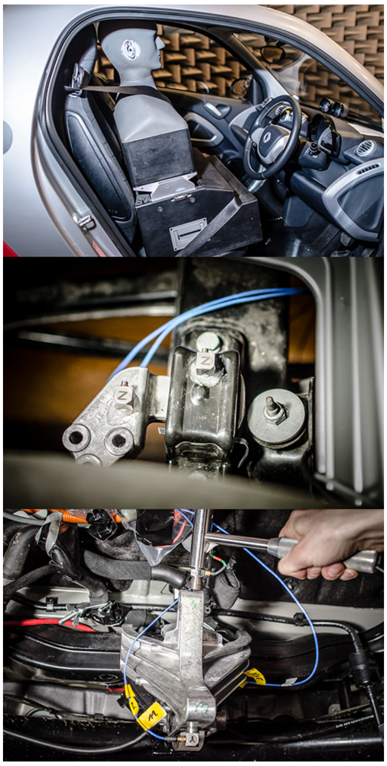

Figure 5: Dummy head in passenger cabin (top), triax-accelerometer in engine compartment (middle), impulse hammer excitation (bottom).

Simulation Performance and Drive Train Evaluation

Fig. 6 shows the left ear results for an exemplary partial load run-up with standard parameters for the simulated machine and drive train structure. The calculation of the binaural time signals in the passenger cabin based on the velocities and forces of the structural simulation takes about 1.7s per simulated signal minute. Thus, the model fulfills the criterion of a fast simulation. The binaural time signal can be evaluated objectively and subjectively. The combination of several methods allows electrical drive developers to get a quick analysis of the effects of changes to drive train components.

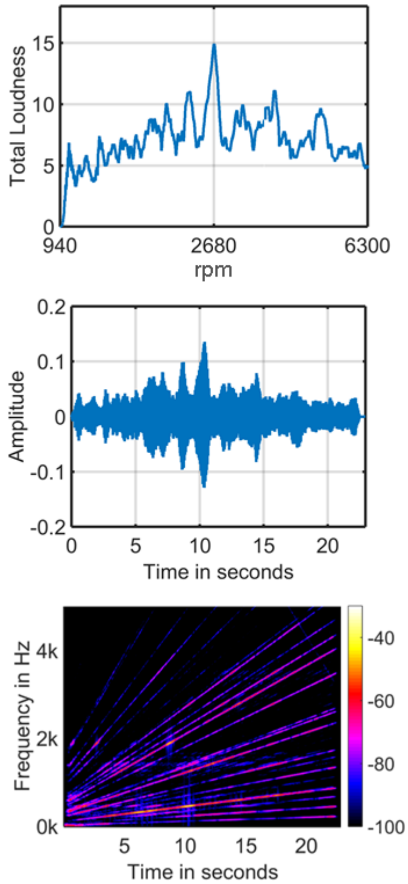

Figure 6: Exemplary partial load run-up, results shown for the left ear. Time signal (top), total loudness over time ([8], middle), spectrogram (bottom).

Fig. 7 shows an example for the simulation of a variant with additional elastic decoupling in the drive train structure. The goal is to diminish the dominant resonances in Fig. 6 for machine speeds around 3000 rpm. As it can be seen in the comparison between the original (red) and new (blue) loudness curve in Fig. 7, the goal has been accomplished. Due to resonance shifts caused by the decoupling, new dominant resonances appear at machine speeds that correspond to vehicle velocities of about 70 km/h, which are a very frequent use-case, resulting in a clear contraindication for this variant. This demonstrates how the application of a fast integrative simulation helps to identify unexpected effects that only appear in the coupled systems.

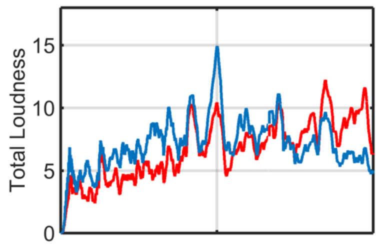

Figure 7: Exemplary partial load run-up, results shown for the left ear. Reference run-up (red), additional elastic decoupling (blue). Total loudness over time.

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