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RESEARCH ARTICLE

Squeak and rattle noise prediction in vehicle acoustics

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Abstract

Squeak and rattle belong to unintended noise audible by occupants of a vehicle. This noise negatively affects the perceived built quality, leading to nonbuying decisions, high warranty costs, and poor brand reputation. Therefore, vehicle manufacturers seek to prevent the emergence of such noise, preferably in the early development phase and during production at the latest. This causes substantial monetary and temporal expenditure, which must be minimized. Rattle is defined as repeated impact. The underlying physical phenomenon is impulsive short-duration contact normal to the surface. Squeak, in contrast, originates from an in-contact motion in the tangential direction and is a friction-induced stick-slip phenomenon. State-of-the-art numerical squeak and rattle prediction is based on linear analysis resulting in an empirical noise risk index. However, quantification of noise and assessment of audibility is not possible. The reason is the nonlinearity of the contact forces. Hence, the actual system response is not calculable with a linear approach. Mathematically, the nature of both excitation events is nonlinear due to frictional contact and nonsmooth due to short impulsive behavior in time. This makes linear simplification and a solution process in the time domain challenging. Both events appear periodically, leading to an oscillation characterized by fundamental and higher harmonics. Due to this periodic character, squeak and rattle phenomena fulfill the prerequisite for applying the harmonic balance method (HBM) to solve the governing nonlinear equation of motion. The more harmonics considered, the more precise the modeling and the dynamic response prediction. In addition, the alternating frequency/time domain method (AFT) allows for switching between the frequency and the time domain during the iterative solution process. Thus, the nonlinear contact forces can be evaluated in the time domain. The equation-solving process results in the calculation of surface velocities. This gives way to determining a proxy for the emitted sound power of the oscillatory system. The simulation method based on combining HBM and AFT was validated on test rigs for squeak and rattle noise. The industrial applicability of this simulation approach was demonstrated numerically and experimentally on an actual vehicle part showing promising results. Thereby, important steps in nonlinear structural dynamics and vehicle acoustics were made toward a calm, smooth, and enjoyable ride for vehicle occupants.

1 | INTRODUCTION

Besides visual impression, auditory perception is an essential way humans recognize and evaluate their surroundings. Sounds audible inside a vehicle strongly affect the occupant's impression of a car—positively and negatively. Such sounds can be divided into pleasant and wanted sounds, like the one of an engine of a sports car, into informative and warning sounds, like a distance warning beep, and into unpleasant and unwanted noise. The latter, among others, include noise from wind turbulence, rolling tires, the drive train, and squeak and rattle (SAR) [1]. In recent years, significant advancements in vehicle acoustics were made to lower noise levels inside passenger vehicle compartments. The further reduction comes from the evolving electrification of drive trains. In such an environment of low noise level, the auditory sense of the vehicle occupant adapts and becomes more sensitive. Besides the absence of further masking noise, this effect leads to the passenger hearing the random transient SAR noise all the more.

SAR noise stemming from interior vehicle components, which is associated with loose parts, poorly built quality, or even loss of functionality, negatively affects the impression of a car. Therefore, SAR noise noticed by vehicle occupants often leads to warranty claims [2]. This subsequently causes warranty costs and image deterioration for vehicle manufacturers. Thus, audible SAR noise must be prevented in a vehicle interior.

Squeak noise and rattle noise originate from contact interaction, that is, stick-slip and impact [3]. The origin of squeak noise is a frictional in-contact motion in a tangential direction characterized by the stick-slip effect. The repeated transition between the sticking and slipping state of motion excites the contact partners into vibration, leading the surfaces to emit a squeak noise. Rattle noise, in contrast, originates from repeated impact, that is, short-duration opening and closing contact between the contact partners in the normal direction to the surface. This leads to an impulsive broadband excitation resulting in the emission of rattle noise.

SAR noise prevention during vehicle development and elimination during production is time- and resource-consuming. The state-of-the-art industry approach to predict SAR noise in an early development stage includes numerical simulation based on linear theories, that is, linear finite element models without contact definition and linear modal analysis with corresponding postprocessing. This linear approach results in an empirical SAR noise risk index calculated for interior vehicle assemblies such as instrument panels, center consoles, or door panels. The index indicates the risk for contact interaction at adjacent or attached parts, which can lead to SAR noise emission. The SAR risk index shows the design engineer where to modify the corresponding component to prevent SAR noise. Also, it allows one to distinguish between more and less critical contact areas. SAR noise, however, originates from nonlinear contact interaction. Therefore, nonlinear contact forces must be considered for a realistic prediction of the deflection shapes under vibratory excitation. Only considering the contact interaction force using a nonlinear simulation approach gives way to quantify SAR noise and assess its audibility realistically.

2 | THEORY

This section provides an overview of the methods proposed to predict SAR noise in a vehicle interior in a quantifiable manner. The governing equations of motion describe the nonlinear behavior of the contact interaction at squeaking or rattling spots and the resulting dynamic behavior of an automotive interior subsystem, for example, instrument panel, door trim panel, or center console. The HBM, in combination with the AFT, is exploited to approximately solve the corresponding equations and thereby find the physically backed-up state of motion of such a squeaking or rattling system. With the surface velocity computed, the equivalent radiated power (ERP) density can be calculated as a proxy to the sound power of a squeak or rattle noise emitted by an interior vehicle component.

2.1 | Harmonic balance method

The HBM is a frequency-domain method that finds approximations for steady-state solutions of differential equations [4–8]. The governing differential equation of motion describing the dynamic behavior of a potentially squeaking or rattling system discretized with n degrees of freedom is

$$\mathbf{M}\ddot{\mathbf{u}}(t) + \mathbf{D}\dot{\mathbf{u}}(t) + \mathbf{K}\mathbf{u}(t) + \mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t) = \mathbf{f}_{\mathrm{e}}(t). \tag{1}$$

Herein, $\mathbf{M}$, $\mathbf{D}$, and $\mathbf{K}$ denote the mass, damping, and stiffness matrix of the system, respectively. The displacement $\mathbf{u}(t)$, velocity $\dot{\mathbf{u}}(t)$, and acceleration $\ddot{\mathbf{u}}(t)$ describe the sought state of motion of the mechanical system. The vector $\mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t)$ collects the nonlinear forces, which arise from contact interaction in squeaking or rattling configurations. The vector $\mathbf{f}_{\mathrm{e}}(t)$ represents the external excitation force driving the system into vibration. Under the assumption that the excitation force,

$$\mathbf{f}_{\mathrm{e}}(t + T) = \mathbf{f}_{\mathrm{e}}(t), \tag{2}$$

the nonlinear force,

$$\mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t + T) = \mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t), \tag{3}$$

and the vibration response of a system represented by the displacement,

$$\mathbf{u}(t + T) = \mathbf{u}(t), \tag{4}$$

are periodic with respect to the same oscillation period T with the fundamental angular frequency ω,

$$T = \frac{2\pi}{\omega}, \tag{5}$$

the time-dependent quantities in Equation (1), that is, $\mathbf{u}(t)$, $\dot{\mathbf{u}}(t)$, $\ddot{\mathbf{u}}(t)$, $\mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t)$, and $\mathbf{f}_{\mathrm{e}}(t)$, can be approximated by truncated Fourier series,

$$\mathbf{u}(t) \approx \mathbf{u}_{\mathrm{h}}(t) = \frac{\mathbf{u}_{0,\mathrm{c}}}{2} + \sum_{k=1}^{h} \left(\mathbf{u}_{k,\mathrm{c}} \cos(k\omega t) + \mathbf{u}_{k,\mathrm{s}} \sin(k\omega t)\right), \tag{6}$$

$$\dot{\mathbf{u}}(t) \approx \dot{\mathbf{u}}_{\mathrm{h}}(t) = \sum_{k=1}^{h} k\omega\left(-\mathbf{u}_{k,\mathrm{c}} \sin(k\omega t) + \mathbf{u}_{k,\mathrm{s}} \cos(k\omega t)\right), \tag{7}$$

$$\ddot{\mathbf{u}}(t) \approx \ddot{\mathbf{u}}_{\mathrm{h}}(t) = \sum_{k=1}^{h} k^2\omega^2\left(-\mathbf{u}_{k,\mathrm{c}} \cos(k\omega t) - \mathbf{u}_{k,\mathrm{s}} \sin(k\omega t)\right), \tag{8}$$

$$\mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t) \approx \mathbf{f}_{\mathrm{nl,h}}(\dot{\mathbf{u}}, \mathbf{u}, t) = \frac{\mathbf{f}_{\mathrm{nl},0,\mathrm{c}}}{2} + \sum_{k=1}^{h} \left(\mathbf{f}_{\mathrm{nl},k,\mathrm{c}} \cos(k\omega t) + \mathbf{f}_{\mathrm{nl},k,\mathrm{s}} \sin(k\omega t)\right), \quad \text{and} \tag{9}$$

$$\mathbf{f}_{\mathrm{e}}(t) \approx \mathbf{f}_{\mathrm{e,h}}(t) = \frac{\mathbf{f}_{\mathrm{e},0,\mathrm{c}}}{2} + \sum_{k=1}^{h} \left(\mathbf{f}_{\mathrm{e},k,\mathrm{c}} \cos(k\omega t) + \mathbf{f}_{\mathrm{e},k,\mathrm{s}} \sin(k\omega t)\right). \tag{10}$$

Herein, h is the number of harmonics taken into account to approximate the time-dependent quantities. The higher the number of harmonics considered, the more precise the approximation. $\mathbf{u}_{k,\mathrm{c}}$ and $\mathbf{u}_{k,\mathrm{s}}$ with $k \in [0, 1, \ldots h]$ are the Fourier coefficients for the cosine and sine part, respectively. The Fourier coefficient of the cosine part with index $k = 0$ considers the static load case and higher orders reassemble the corresponding vibration response.

Inserting the Fourier ansatz, that is, Equations (6)–(10), into Equation (1) transfers the equation from time into the frequency domain. Collecting the cosine and sine part yields

$$\sum_{k=0}^{h} \underbrace{\left[\left(\mathbf{K} - k^2\omega^2\mathbf{M}\right)\mathbf{u}_{k,\mathrm{c}} + k\omega\mathbf{D}\mathbf{u}_{k,\mathrm{s}} + \mathbf{f}_{\mathrm{nl},k,\mathrm{c}} - \mathbf{f}_{\mathrm{e},k,\mathrm{c}}\right]}_{\mathbf{R}_{\mathrm{c},k}} \cos(k\omega t) +$$

$$\sum_{k=1}^{h} \underbrace{\left[\left(\mathbf{K} - k^2\omega^2\mathbf{M}\right)\mathbf{u}_{k,\mathrm{s}} + k\omega\mathbf{D}\mathbf{u}_{k,\mathrm{c}} + \mathbf{f}_{\mathrm{nl},k,\mathrm{s}} - \mathbf{f}_{\mathrm{e},k,\mathrm{s}}\right]}_{\mathbf{R}_{\mathrm{s},k}} \sin(k\omega t) = 0. \tag{11}$$

The expressions in brackets must vanish to satisfy Equation (11) for all indices $k \in [0, 1, \ldots h]$. Meeting this requirement allows for formulating the residual equations for the cosine part $\mathbf{R}_{\mathrm{c},k}$,

$$\mathbf{R}_{\mathrm{c},k} = \left(\mathbf{K} - k^2\omega^2\mathbf{M}\right)\mathbf{u}_{k,\mathrm{c}} + k\omega\mathbf{D}\mathbf{u}_{k,\mathrm{s}} + \mathbf{f}_{\mathrm{nl},k,\mathrm{c}} - \mathbf{f}_{\mathrm{e},k,\mathrm{c}} = 0 \qquad \forall \qquad k \in [0, 1, \ldots h], \tag{12}$$

and the sine part $\mathbf{R}_{\mathrm{s},k}$,

$$\mathbf{R}_{\mathrm{s},k} = \left(\mathbf{K} - k^2\omega^2\mathbf{M}\right)\mathbf{u}_{k,\mathrm{s}} + k\omega\mathbf{D}\mathbf{u}_{k,\mathrm{c}} + \mathbf{f}_{\mathrm{nl},k,\mathrm{s}} - \mathbf{f}_{\mathrm{e},k,\mathrm{s}} = 0 \qquad \forall \qquad k \in [1, 2, \ldots h]. \tag{13}$$

The Fourier ansatz does not fully satisfy the governing equations of motion. Therefore, the HBM applies the Galerkin method, that is, the method of weighted residuals. It uses the same base functions for the ansatz and the weighting functions to satisfy the governing equations over the integral of one oscillation period. This yields the governing equations of motion in the frequency domain,

$$\underbrace{\left[\tilde{\mathbf{M}}(\omega) + \tilde{\mathbf{D}}(\omega) + \tilde{\mathbf{K}}(\omega)\right]}_{\stackrel{!}{=}\,\tilde{\mathbf{S}}(\omega)} \tilde{\mathbf{u}} + \tilde{\mathbf{f}}_{\mathrm{nl}}(\tilde{\mathbf{u}}) \stackrel{!}{=} \tilde{\mathbf{S}}(\omega)\tilde{\mathbf{u}} + \tilde{\mathbf{f}}_{\mathrm{nl}}(\tilde{\mathbf{u}}) = \tilde{\mathbf{f}}_{\mathrm{e}}. \tag{14}$$

The matrices describing the system in the frequency domain, that is, the dynamic mass matrix $\tilde{\mathbf{M}}(\omega)$,

$$\tilde{\mathbf{M}}(\omega) = \begin{bmatrix} 0 & 0 & 0 & \cdots & 0 & 0 \\ 0 & -1^2\omega^2\mathbf{M} & 0 & \cdots & 0 & 0 \\ 0 & 0 & -1^2\omega^2\mathbf{M} & \cdots & 0 & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & -h^2\omega^2\mathbf{M} & 0 \\ 0 & 0 & 0 & \cdots & 0 & -h^2\omega^2\mathbf{M} \end{bmatrix}, \quad \tilde{\mathbf{M}} \in \mathbb{R}^{(2h+1)n \,\times\, (2h+1)n}, \tag{15}$$

the dynamic damping matrix $\tilde{\mathbf{D}}(\omega)$,

$$\tilde{\mathbf{D}}(\omega) = \begin{bmatrix} 0 & 0 & 0 & \cdots & 0 & 0 \\ 0 & 0 & -1\omega\mathbf{D} & \cdots & 0 & 0 \\ 0 & 1\omega\mathbf{D} & 0 & \cdots & 0 & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & 0 & -h\omega\mathbf{D} \\ 0 & 0 & 0 & \cdots & h\omega\mathbf{D} & 0 \end{bmatrix}, \quad \tilde{\mathbf{D}} \in \mathbb{R}^{(2h+1)n \,\times\, (2h+1)n}, \tag{16}$$

and the dynamic stiffness matrix $\tilde{\mathbf{K}}(\omega)$,

$$\tilde{\mathbf{K}}(\omega) = \begin{bmatrix} \mathbf{K} & 0 & 0 & \cdots & 0 & 0 \\ 0 & \mathbf{K} & 0 & \cdots & 0 & 0 \\ 0 & 0 & \mathbf{K} & \cdots & 0 & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & \mathbf{K} & 0 \\ 0 & 0 & 0 & \cdots & 0 & \mathbf{K} \end{bmatrix}, \quad \tilde{\mathbf{K}} \in \mathbb{R}^{(2h+1)n \times (2h+1)n}, \tag{17}$$

are permutations of their corresponding time-domain counterpart according to the harmonics considered and degrees of freedom n of the system. The dynamic system matrix $\tilde{\mathbf{S}}(\omega)$,

$$\tilde{\mathbf{S}}(\omega) = \begin{bmatrix} \mathbf{K} & 0 & 0 & \cdots & 0 & 0 \\ 0 & -1^2\omega^2\mathbf{M} + \mathbf{K} & -1\omega\mathbf{D} & \cdots & 0 & 0 \\ 0 & 1\omega\mathbf{D} & -1^2\omega^2\mathbf{M} + \mathbf{K} & \cdots & 0 & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & -h^2\omega^2\mathbf{M} + \mathbf{K} & -h\omega\mathbf{D} \\ 0 & 0 & 0 & \cdots & h\omega\mathbf{D} & -h^2\omega^2\mathbf{M} + \mathbf{K} \end{bmatrix},$$
$$\tilde{\mathbf{S}} \in \mathbb{R}^{(2h+1)n \times (2h+1)n},$$

collects the frequency-domain system matrices. The vector $\tilde{\mathbf{f}}_{\mathrm{e}}$,

$$\tilde{\mathbf{f}}_{\mathrm{e}} = \begin{bmatrix} \mathbf{f}^{\mathrm{T}}_{\mathrm{e},0,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{e},1,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{e},1,\mathrm{s}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{e},2,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{e},2,\mathrm{s}} & \cdots & \mathbf{f}^{\mathrm{T}}_{\mathrm{e},h,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{e},h,\mathrm{s}} \end{bmatrix}^{\mathrm{T}}, \quad \tilde{\mathbf{f}}_{\mathrm{e}} \in \mathbb{R}^{(2h+1)n \times 1}, \tag{19}$$

consists of the Fourier coefficients describing the excitation force in the frequency domain. Equally, the vector $\tilde{\mathbf{f}}_{\mathrm{nl}}(\tilde{\mathbf{u}})$,

$$\tilde{\mathbf{f}}_{\mathrm{nl}} = \begin{bmatrix} \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},0,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},1,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},1,\mathrm{s}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},2,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},2,\mathrm{s}} & \cdots & \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},h,\mathrm{c}} & \mathbf{f}^{\mathrm{T}}_{\mathrm{nl},h,\mathrm{s}} \end{bmatrix}^{\mathrm{T}}, \quad \tilde{\mathbf{f}}_{\mathrm{nl}} \in \mathbb{R}^{(2h+1)n \times 1}, \tag{20}$$

contains the Fourier coefficients describing the nonlinear forces. The vector $\tilde{\mathbf{u}}$ contains the Fourier coefficients $\mathbf{u}_{k,\mathrm{c}}$ and $\mathbf{u}_{k,\mathrm{s}}$ with $k \in [0, 1, \ldots h]$ describing the state of motion of the system according to Equation (6),

$$\tilde{\mathbf{u}} = \begin{bmatrix} \mathbf{u}^{\mathrm{T}}_{0,\mathrm{c}} & \mathbf{u}^{\mathrm{T}}_{1,\mathrm{c}} & \mathbf{u}^{\mathrm{T}}_{1,\mathrm{s}} & \mathbf{u}^{\mathrm{T}}_{2,\mathrm{c}} & \mathbf{u}^{\mathrm{T}}_{2,\mathrm{s}} & \cdots & \mathbf{u}^{\mathrm{T}}_{h,\mathrm{c}} & \mathbf{u}^{\mathrm{T}}_{h,\mathrm{s}} \end{bmatrix}^{\mathrm{T}}, \quad \tilde{\mathbf{u}} \in \mathbb{R}^{(2h+1)n \times 1}. \tag{21}$$

The HBM allows for setting up the governing equations of motion in the frequency domain, as denoted in Equation (14). As shown above, this set comprises a number of $(2h + 1)n$ equations and an equal number of unknown Fourier coefficients describing the state of motion of the squeaking or rattling system in the frequency domain. However, to solve this set of algebraic equations, the nonlinear forces evolving from contact interaction at the squeaking or rattling spots still need to be estimated.

2.2 | Alternating frequency/time domain method

The governing equations of motion (14) are solved iteratively to find approximations for the state of motion of the squeaking or rattling system. Therefore, the Fourier coefficients of the nonlinear contact forces $\tilde{\mathbf{f}}_{\mathrm{nl}}(\tilde{\mathbf{u}})$ are required. These contact forces, however, depend on the system's past and present states of motion and therefore are evaluated in the time domain, according to [9]. Cameron and Griffin proposed the AFT to solve such a system of equations in the frequency domain containing nonlinear forces, which need to be evaluated in the time domain [9].

The AFT is a scheme that allows for switching between the frequency and the time domain by applying the discrete Fourier transform and its inverse counterpart. The state of motion must first be transferred from the frequency to the time domain to evaluate the nonlinear contact forces. The nonlinear contact interaction forces can then be evaluated over one period. As the HBM is a frequency domain method, these contact forces have to be transformed into the frequency domain. This is done by numerically solving the time integrals for the Fourier coefficients of the nonlinear forces,

$$\mathbf{f}_{\mathrm{nl,c}} = \frac{2}{T}\int_0^T \mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t)\cos(k\omega t)\,dt, \qquad k \in [0, 1, \dots h], \tag{22}$$

and

$$\mathbf{f}_{\mathrm{nl,s}} = \frac{2}{T}\int_0^T \mathbf{f}_{\mathrm{nl}}(\dot{\mathbf{u}}, \mathbf{u}, t)\sin(k\omega t)\,dt, \qquad k \in [1, 2, \dots h]. \tag{23}$$

Thereby, once all Fourier coefficients and components of the governing equations of motion (14) are determined, the residual equation

$$\tilde{\mathbf{R}}(\tilde{\mathbf{u}}) = \tilde{\mathbf{S}}(\omega)\tilde{\mathbf{u}} + \tilde{\mathbf{f}}_{\mathrm{nl}}(\tilde{\mathbf{u}}) - \tilde{\mathbf{f}}_{\mathrm{e}} = 0, \tag{24}$$

can be tested for convergence to find the sought state of motion $\tilde{\mathbf{u}}$ of the squeaking or the rattling system.

2.3 | Nonlinear contact forces

The AFT solves the governing equations of motion in the frequency domain. This solution procedure includes the evaluation of the contact forces in the time domain. Therefore, one oscillation period T with fundamental angular frequency ω is discretized by a defined number of sampling points.

A penalty formulation is exploited to evaluate the contact forces in case of rattle, that is, repeated change between contact and separation in the normal direction. The normal force at the node-to-node contact is evaluated at any time increment based on the initial gap or overclosure between and the current location of the contact nodes. Thereby, two cases of contact can be determined. In case of a positive distance value in the normal direction, no normal force is applied, that is, separation. In case of contact, that is, zero or negative distance value, the penalty contact stiffness defines the applied force.

Tangential in-contact movement, characterized by stick-slip transitions, leads to the emission of squeak noise. A dry friction law with variable normal force evaluates the underlying contact force. A contact element is defined including a Coulomb friction element and a spring in a tangential direction, and a spring in the normal direction. It distinguishes three states of contact: separation, sticking, and sliding.

2.4 | Equivalent radiated power

One goal of the numerical prediction of SAR noise is to distinguish between squeaking and nonsqueaking or rattling and nonrattling system configurations. Once noise-emitting configurations are identified, the aim is to obtain a measure for the emitted sound to quantify and assess the severity of the noise. As the surface velocity $\dot{u}_n(t)$ normal to the surface area A of the squeaking or rattling system is known from applying the HBM with AFT, the ERP,

$$P_{\mathrm{eq}} = \frac{1}{2}\rho_f c_f \int |\dot{u}_n(t)|^2 \mathrm{d}A, \tag{25}$$

with the density ρ_f and the speed of sound c_f of the surrounding fluid can be computed. P_{eq} represents an approximate measure. Moreover, it is an upper limit to the emitted noise of a squeaking or rattling system.

3 | VALIDATION

The HBM, in combination with the AFT, allows for setting up the governing equations of motion and numerically predict an approximation for the vibration response of squeaking or rattling systems, that is, it provides the theoretical foundation. To prove that the method can predict the vibration response of SAR noise emitting components, a validation by comparison of measured quantities from experimental results with real squeaking or rattling systems is needed.

3.1 | Squeak test rig

To prove the applicability of HBM to predict the vibration response of squeaking systems, a squeak test rig was developed [10–12]. As mentioned above, squeak noise originates from an in-contact tangential motion of the two squeaking components characterized by stick-slip transitions. The test rig therefore consists of a rigid frame in u-shape. The frame on the one end holds the stationary stick-slip partner. On the other end, the oscillating cantilever beam is clamped, which holds the second stick-slip partner with convex curvature at the stick-slip interface. The cantilever beam is excited into vibration by an electrodynamic shaker. A piezoelectric force transducer measures the excitation force input. Further force transducers measure the contact force in the normal direction of the stick-slip interaction on the stationary side. On the oscillating beam, the vibration response of the squeaking system is measured with accelerometers and a laser Doppler vibrometer.

This setup could prove that the fundamental and higher harmonics of the excitation frequency dominate the vibration response of a squeaking system configuration to a harmonic excitation [10]. Therefore, the essential prerequisite of periodicity for the usage of HBM, shown in Subsection 2.1, is met to predict the vibration response of a squeaking system.

Further improvements provide the implementation of a variable normal force at the squeaking contact interaction [13]. This allows one to predict not only the even harmonics, with constant normal force considered, but also the odd harmonics of the vibration response of a squeaking system configuration.

3.2 | Rattle test rig

To research the applicability of HBM to predict the vibration response of rattling system configurations, a rattle test rig was developed [14, 15]. The rattle test rig consists of two support frames. One holds the stationary rattle partner, that is, a square plastic plate clamped at all edges. The second support frame clamps the oscillating cantilever beam. At the free end of the cantilever beam, the second rattle contact partner is mounted, that is, a metallic tip. To excite the cantilever beam into vibration, an electrodynamic shaker is applied. A piezoelectric force transducer measures the excitation force. A second force transducer is mounted between the cantilever beam and rattling tip to measure the contact interaction force in the normal direction. The vibration response of the rattle test rig is measured using 3D laser Doppler vibrometers.

The evaluation of the experimental measurements of the vibration response of the rattling test rig demonstrated that the vibration response of a rattling system configuration is dominated by the base and higher harmonics of the excitation frequency [14]. Hence, the fundamental prerequisite of periodicity, as described in Subsection 2.1, for using the HBM to predict the vibration response of a rattling system, is met.

4 | APPLICATION

Subsections 3.1 and 3.2 show the HBM to be able to predict the vibrational response of squeaking as well as of rattling system configurations. The need for using such a nonlinear approach to numerically predict SAR noise due to its origin in stick-slip and impact contact interaction is discussed in [16]. The application of the HBM to predict squeak noise on an actual real vehicle assembly is shown in [17]. Here, a door trim panel in a semianechoic chamber was excited into vibration by an electrodynamic shaker. The electrodynamic shaker excited the structure harmonically close to a contact interface of two panel components. There, a relative movement characterized by stick-slip behavior could be generated. The vibration response of the door trim panel was measured by accelerometers for squeaking and nonsqueaking configuration states. The comparison of the measured results from the experiment showed to be in accordance with the results from the nonlinear numerical simulation approach based on HBM.

5 | CONCLUSION

This work summarizes the advancements made in the numerical prediction of SAR noise in a passenger vehicle's interior. Due to its origin in contact interaction, SAR noise only is quantifiable considering a nonlinear simulation approach. Therefore, HBM with AFT was proposed and utilized. It was shown that HBM is applicable to predict the vibration response of squeaking and rattling system configurations due to fulfilling the periodicity requirement. The method was validated by comparing simulation and experimental results for SAR test rigs. Finally, the application of the simulation approach on an actual vehicle part and the comparison with experimental findings was described.

While the classical linear approach only allows for qualitative SAR noise prediction, the nonlinear approach of HBM with AFT helps quantify the noise emitted by interior vehicle components. Thus, more precise predictions can be made in the early stages of vehicle development, impeding costly hardware experiments later.

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