

Prominence of Different Directions of Hub Forces and Moments in Structure-Borne Tire Noise

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ABSTRACT

In the automotive industry, tire noise is an important factor for the perceived quality of a product. A useful method to address such NVH problems is to combine recordings with measurements and/or simulations into auralizations. An example of a method to create structure-borne tire noise auralizations is to filter recordings of hub forces and moments through binaural transfer functions experimentally measured from the hub of the car to an artificial head in the car cabin. To create authentic auralizations of structure-borne sound, all six degrees of freedom (DOFs) of hub forces and moments and transfer functions should be included. However, rotational DOFs are often omitted due to measurement difficulty, complexity, time, and cost. The objective was to find which DOF (or DOFs) is perceived as most prominent in structure-borne tire noise. An auralization model of interior structure-borne tire noise was used. An auralization including all DOFs was compared with auralizations lacking one DOF at a time in a listening test. Auralizations lacking either all translational DOFs or all rotational DOFs were also included. Results from the listening test showed that the overturning moment (M_a) was perceived as the most prominent DOF. Removing all rotational DOFs led to a larger audible difference than removing all translational DOFs. Therefore, rotational DOFs should not be disregarded in tire noise auralization. The results suggest which DOFs are important to consider in tire and vehicle sound design.

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INTRODUCTION

In the automotive industry, Noise, Vibration and Harshness (NVH) issues are important factors for the perceived quality of a product. Interior tire noise is an essential part of NVH. Tire noise is generated from the tire/road interaction and transferred into the car cabin through structure-borne and airborne paths. At low frequencies, typically less than 500 Hz [1], the structure-borne contribution dominates. Vibrations caused by the tire/road interaction are transferred through the hub to the suspension and chassis and radiate into the car cabin. At higher frequencies, the contribution from airborne tire noise dominates over the structure-borne tire noise. Although interior tire noise is an essential part of NVH and is highly related to comfort, it is often handled late in the car development process when there is little freedom for modifications of the car. For both car and tire manufacturers, it is desirable to predict tire noise early in the development process to allow time and cost efficient product development. This is achieved by moving field testing to indoor lab environments and using computer-aided engineering tools for early predictions of product performance and quality. A useful method to address NVH problems and to reduce field-testing is to combine recordings with measurements and/or simulations into auralizations. An example of a method to create auralizations is to filter the recordings through airborne and/or structure-borne transfer functions [2, 3]. Transfer functions are obtained from experimental measurements by a method such as Transfer Path Analysis (TPA), from numerical simulation methods such as Finite Element Modeling

(FEM), or from methods combining these two techniques such as Component Mode Synthesis (CMS) and Frequency response function Based Substructuring (FBS) [4]. To create authentic structure-borne auralizations, the structural contributions from all six degrees of freedom (DOFs) - three translational and three rotational DOFs - have to be included. Due to factors such as measurement difficulty, complexity, time, and cost, rotational DOFs are often omitted and ignored [5, 6]. This results in simplified auralizations with varying magnitudes of audible errors. However, the required level of detail in the auralization depends on the stage in the development process. In an early development stage, audible errors and artifacts might be acceptable as long as the main character of the sound is realistic. In cases where auralizations are used for detailed psychoacoustic analysis, it is important to keep the auralizations perceptually equivalent to real sounds [7, 8]. What is considered important for preservation of a sound's main character may be application specific, but a basic requirement should be that listeners' preference ratings for a certain sound should not be altered due to errors and artifacts in the auralization [9].

The aim of this study was to investigate how different DOFs of hub forces and moments affect perceived tire noise in the cabin of a car. An auralization containing all six DOFs was compared with auralizations containing all but one DOF in a listening test to determine which DOFs are perceived as the most prominent in structure-borne interior tire noise. In addition, auralizations lacking

either all translational or all rotational DOFs were compared to the auralization containing all six DOFs in order to determine how large the perceived effects are when all translational or all rotational DOFs are omitted.

METHOD

An auralization model of structure-borne interior tire noise has been developed and validated in a previous study [10]. A benefit of the auralization model is that the structure-borne tire noise is separated from other noise sources in the car. Furthermore, the noise contribution from each DOF can be separated and listened to individually. In this study, the auralization model is used to determine the influence of different DOFs of hub forces and moments on structure-borne interior tire noise. For the structure borne tire noise, the hub acts as the coupling element between the wheel and the car. By measuring the forces and moments in six DOFs at the hub during operational conditions, the full structure-borne contribution was obtained. The recorded hub forces were the inputs for the auralization model and were filtered through binaural transfer functions (BTFs) measured from the hub to an artificial head placed in the cabin of the car. With BTFs, the spatial information in the sound is preserved when the auralization is reproduced using headphones [11]. The auralizations lacking one or more DOFs were compared with the auralization including all six DOFs in a listening test. The schematic of the experiment is shown in Figure 1.

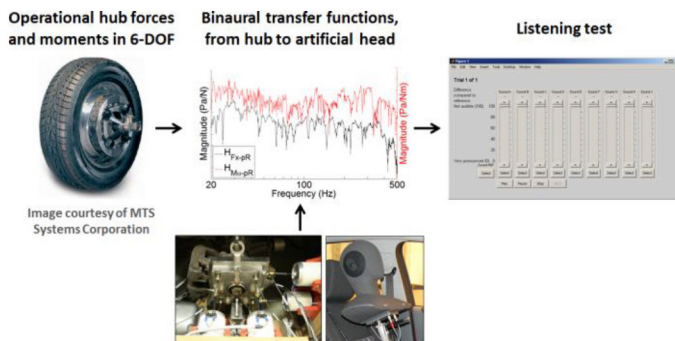


Figure 1. Schematic illustration of the experiment to produce and evaluate auralizations of structure-borne tire noise.

Creation of Auralizations

Auralizations were created in a three-step process, including recording the operational hub forces and moments, measuring the BTFs, and convolving the hub forces and moments with the BTFs.

Recording the Operational Hub Forces and Moments

An MTS SWIFT 20A wheel force transducer was used to record operational hub forces and moments in six DOFs. The SWIFT system consisted of a load cell mounted on the outside of a slightly modified tire rim. A wheel hub adapter was used to mount the load cell to the hub of the car. The total mass of the system was comparable to a standard wheel. According to information from the manufacturer of the load cell, the deviation from a flat frequency response was approximately 2 dB in the frequency range 20-500 Hz in all directions. This frequency range covers the typical range of structure-borne tire noise. Although the measurement rim was slightly

modified, it was considered to have similar dynamic properties as a conventional rim. This implies that the measurement rim might not be considered to be a rigid body below 500 Hz in all DOFs, but still has a similar influence on the measured hub forces and moments as would be expected from a conventional rim. The accuracy of the system has been validated previously [12, 13]. The MTS SWIFT 20A was mounted on a station wagon (2011 Audi A4 Avant) on the front right side (passenger side). Friction winter tires (no studs) with dimension 205/55R16 were used. Hub forces and moments were recorded at 70 km/h in a straight line on dry asphalt. The sampling frequency was 12.8 kHz.

Measurement of Binaural Transfer Functions (BTFs)

Because the recorded hub forces and moments were determined at the center of the hub, BTFs in six DOFs had to be measured from the center of the hub to an artificial head. The system of BTFs is shown in Figure 2.

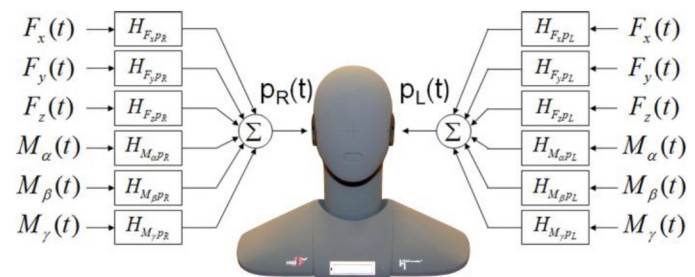


Figure 2. Binaural transfer functions (H) in six degrees of freedom (translational directions x, y, z and rotational directions α, β, γ) from hub forces (F) and moments (M), which are convolved to create the auralized signal (p) at time (t) for each ear (R, L) of an artificial head.

In order to measure the BTFs in six DOFs, the measurement setup had to be capable of exciting all DOFs without changes in the setup. This prevents inconsistencies due to displacements, alterations, or changes in the loading of the structure. For this study, a station wagon (2005 SAAB 9-3) was used as a test object. Without the tire or rim mounted on the hub, all equipment was attached directly to a specially made brake disc, as shown in Figure 3.

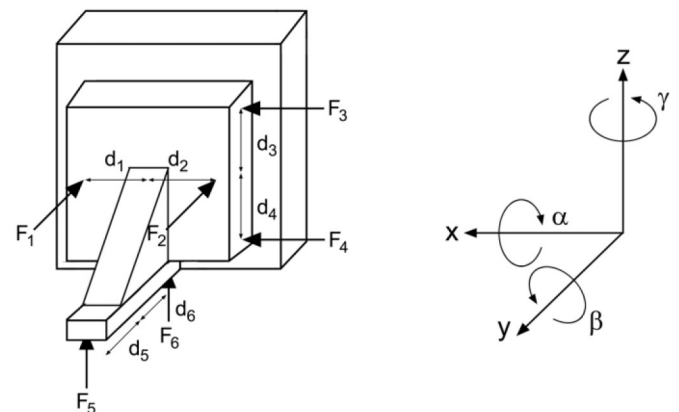


Figure 3. Square-shaped measurement disc (left) and tire axis system (right) according to ISO 8855:2011 [14]. The locations of the measured forces F_{1-6} and the distances d_{1-6} between forces are shown on the measurement disc.

The same setup was used in a previous study where the mobility of the hub was measured in six DOFs [15]. This setup has been validated by listening test comparison of auralizations with artificial head recordings made under the same conditions [10]. The left front hub was excited in six DOFs simultaneously, shown in Figure 4, using white noise from 0-1600 Hz.

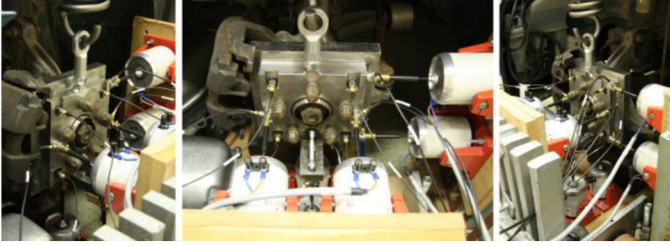


Figure 4. Measurement setup.

The artificial head (Head Acoustics HMS IV) was placed in the driver's position in the car. The response at each force transducer and at the artificial head was recorded. The recorded responses from the force transducers were summarized in the time domain to represent the net force (F_x , F_y , and F_z) or moment (M_α , M_β , and M_γ) in each direction according to equations (1, 2, 3, 4, 5, 6). Locations of forces F_{1-6} and distances d_{1-6} are shown in Figure 3, where $d_{1-4} = 0.070$ m, $d_5 = 0.066$ m, and $d_6 = 0.034$ m.

$$F_x(t) = F_3(t) + F_4(t) \quad (1)$$

$$F_y(t) = -F_1(t) - F_2(t) \quad (2)$$

$$F_z(t) = F_5(t) + F_6(t) \quad (3)$$

$$M_\alpha(t) = d_5 F_5(t) - d_6 F_6(t) \quad (4)$$

$$M_\beta(t) = d_3 F_3(t) - d_4 F_4(t) \quad (5)$$

$$M_\gamma(t) = -d_1 F_1(t) + d_2 F_2(t) \quad (6)$$

To determine the BTFs from the summarized forces and moments to each ear of the artificial head, the system is considered as two MISO systems (multiple input single output) [16, 17], as shown in Figure 2. The BTFs for each ear are expressed as

$$\mathbf{S}_{\mathbf{Fp}} = \mathbf{S}_{\mathbf{FF}} \cdot \mathbf{H}_{\mathbf{Fp}} \quad (7)$$

where the matrix vector $\mathbf{S}_{\mathbf{Fp}}$ represents the cross-spectral density functions from each force and moment to one of the ears of the artificial head. The matrix $\mathbf{S}_{\mathbf{FF}}$ corresponds to the auto and cross-

spectral density functions between the net forces and moments. $\mathbf{H}_{\mathbf{Fp}}$ contains the transfer functions from each force and moment to one ear of the artificial head. Equation (7) can be expressed for the left ear as

$$\begin{bmatrix} S_{F_x p_L} \\ S_{F_y p_L} \\ S_{F_z p_L} \\ S_{M_\alpha p_L} \\ S_{M_\beta p_L} \\ S_{M_\gamma p_L} \end{bmatrix} = \begin{bmatrix} S_{F_x F_x} & S_{F_x F_y} & S_{F_x F_z} & S_{F_x M_\alpha} & S_{F_x M_\beta} & S_{F_x M_\gamma} \\ S_{F_y F_x} & S_{F_y F_y} & S_{F_y F_z} & S_{F_y M_\alpha} & S_{F_y M_\beta} & S_{F_y M_\gamma} \\ S_{F_z F_x} & S_{F_z F_y} & S_{F_z F_z} & S_{F_z M_\alpha} & S_{F_z M_\beta} & S_{F_z M_\gamma} \\ S_{M_\alpha F_x} & S_{M_\alpha F_y} & S_{M_\alpha F_z} & S_{M_\alpha M_\alpha} & S_{M_\alpha M_\beta} & S_{M_\alpha M_\gamma} \\ S_{M_\beta F_x} & S_{M_\beta F_y} & S_{M_\beta F_z} & S_{M_\beta M_\alpha} & S_{M_\beta M_\beta} & S_{M_\beta M_\gamma} \\ S_{M_\gamma F_x} & S_{M_\gamma F_y} & S_{M_\gamma F_z} & S_{M_\gamma M_\alpha} & S_{M_\gamma M_\beta} & S_{M_\gamma M_\gamma} \end{bmatrix} \begin{bmatrix} H_{F_x p_L} \\ H_{F_y p_L} \\ H_{F_z p_L} \\ H_{M_\alpha p_L} \\ H_{M_\beta p_L} \\ H_{M_\gamma p_L} \end{bmatrix} \quad (8)$$

The transfer functions are then solved with

$$\mathbf{H}_{\mathbf{Fp}} = \mathbf{S}_{\mathbf{FF}}^{-1} \cdot \mathbf{S}_{\mathbf{Fp}} \quad (9)$$

Both the auto spectral density functions of the input signals and the cross spectral density functions between input and output signals were calculated in B&K PULSE in the frequency range 0-1600 Hz with 0.25 Hz frequency resolution. A previous study has shown that auralizations should have a frequency resolution of at least 1 Hz in order to avoid audible errors [8]. A Hanning window with 50% overlap was used, and each measurement consisted of 28 averages. A high-pass filter with a cut-off frequency of 0.7 Hz was used for all force transducers, and the artificial head had a high-pass filter with a cut-off frequency of 22.4 Hz.

Creation of stimuli

Auralizations were created by convolving each DOF of the recorded forces and moments with the impulse response of the corresponding DOF of the BTF, as shown in Figure 2, according to

$$p_k(t) = \sum_{i=x}^z (\text{ifft}[H_{Fipk}](t) * F_i(t)) + \sum_{j=\alpha}^{\gamma} (\text{ifft}[H_{Mjpk}](t) * M_j(t)) \quad (10)$$

where

$i = x, y, z$

$j = \alpha, \beta, \gamma$

$k = \text{Right, Left}$

where $p_k(t)$ is the auralized time signal at the k^{th} ear (right or left), $\text{ifft}[H_{Fipk}](t)$ is the impulse response of the BTF in the i^{th} translational DOF (x, y, z) for the k^{th} ear, $\text{ifft}[H_{Mjpk}](t)$ is the impulse response of the BTF in the j^{th} rotational DOF (α, β, γ) for the k^{th} ear, F_i is the recorded hub force in the i^{th} direction, and M_j is the recorded hub moment in the j^{th} direction.