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MEASUREMENT, SIMULATION AND SYNTHESIS OF TURBULENT-BOUNDARY-LAYER-INDUCED VIBRATIONS OF PANEL STRUCTURES

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The characterization of the vibro-acoustic behavior of structures excited by a turbulent boundary layer (TBL) is an expensive task and thus raises the demand for less expensive alternative solutions. This contribution focuses on the synthesis of TBL-typical structural vibration by means of a loudspeaker array placed in front of a panel structure. The derived method can be applied to any flat or curved panel structure (e. g. an aircraft fuselage section). The work is subdivided into three main parts. Firstly, experiments in an aero-acoustic wind tunnel are performed to capture the characteristics of the TBL excitation and the induced vibrations of a rectangular aluminum plate. Secondly, a finite element model of the plate and a statistical excitation model are derived and validated with measurement data. Despite the use of simple simulation models, a high agreement between measurement and simulation is achieved. It is concluded that realistic values for the vibrations of panel structures can be derived. In the third part of this work, the aluminum plate is mounted in the test opening of a transmission loss facility. A loudspeaker array located in the reverberant sending room of the facility is placed in front of the panel and the frequency response functions (FRFs) from the loudspeaker array to the structural vibration of the plate are derived. The structural vibration is measured with a scanning laser Doppler vibrometer placed in the semi-anechoic receiving room of the facility. The filtering of the simulated target values through the inverse FRFs yields the required control signals for the loudspeaker array. Twelve loudspeakers, corresponding to 60 loudspeakers per square meter, are used to synthesize the TBL-induced structural vibration up to 500 Hz. The synthesized structural vibration is consistent with the target values obtained from simulation.

1. Introduction

The turbulent boundary layer (TBL) belongs to the most relevant external disturbance sources for aircraft interior noise [1]. Therefore, flight or wind tunnel tests are conducted to assess the vibro-acoustic behavior and the sound transmission of fuselage structures. Many efforts have been made in the past to replace flight or wind tunnel tests by simulation or by laboratory experiments with targeted structural actuation. Fahy [2] examines a simulative approach using suitable scaled models in order to avoid full scale testing of flight vehicles. It is concluded that the extend of structures which must be simulated may vary from several interframe sections of the complete cylinder to a limited number of panels bays between two frames. Many researchers, however, focus on the investigation of simple plates. Early simulation results of the turbulent-flow-excited vibration of a rectangular flat plate are reported by Strawderman and Brand [3]. Maury et al. [4, 5] apply a wavenumber-frequency formulation to describe the response of a plate to a TBL excitation. A different approach based on the finite element method (FEM) and the boundary element method (BEM) ist described in Montgomery [6]. There, a finite element (FE) model of a plate is excited by a number of distributed forces having proper spatial and temporal correlation. This approach, which is also followed by Schiller [7], is adopted in this work. Unlike in Montgomery, the TBL pressure field is described by the well-known Corcos [8] model. This approach is validated by means of measurement data from an aeroacoustic wind tunnel [9].

Some research has been done in the past regarding the laboratory synthesis of TBL pressure fields or induced structural vibrations by using loudspeakers [10, 11, 12, 13, 14, 15]. In principle, the TBL pressure field could be synthesized directly from the TBL model by using a sufficient number of independent sound sources [11]. This, however, requires a very high source density that can generally not be achieved in the experiments. In particular, the synthesis of TBL excitations induced by low speed flows is very difficult even at low frequencies. This is due to the increasing number of correlation lengths to be synthesized over the panel surface for increasing frequencies [13]. Therefore the filtering effect of the structure must be exploited in order to reduce the complexity. If the vibratory response is known from measurement or simulation (as described above), a number of loudspeakers can be used to actuate the structure and to induce the prescribed vibration field. This approach, which is used in this work, renders the synthesis problem feasible but requires exact knowledge of the structural dynamics which might be a severe drawback.

2. Acquisition of flow data

Experiments in an aeroacoustic wind tunnel are performed in order to capture flow data for the parameterization of the Corcos TBL model. Furthermore the structural dynamics and the vibration response of a flat panel are measured. This data is used for model updating and validation. Further details on the surface pressure and vibration measurements in the wind tunnel can be found in Hu and Misol [9]. The configuration of relevance for this paper is the aluminum panel without riblet foils excited by a turbulent flow of $62.4\,\mathrm{m/s}$. In addition to the measurements documented in [9], the panel vibration response to an impulse hammer excitation is captured and the frequency response functions (FRF) from the force excitation to the 35 accelerometers are evaluated.

3. Modeling and simulation

This section describes the TBL model and its use for the derivation of correlated pressure signals on a predefined FE grid. The obtained nodal forces serve as excitation signals for a harmonic analysis of the FE model. The resulting structural response is compared to the measurement data obtained in the wind tunnel.

3.1 Excitation model and pressure field synthesis

According to the Corcos model, the cross-power spectral density (CPSD) of two pressure signals x_i and y_j measured in a fully developed TBL is given by

$$S_{ij}(\omega) = S_{ii}(\omega) e^{-|r_{sp}|/L_{sp}(\omega)} e^{-|r_{st}|/L_{st}(\omega)} e^{-j\omega r_{st}/U_c}. \tag{1}$$

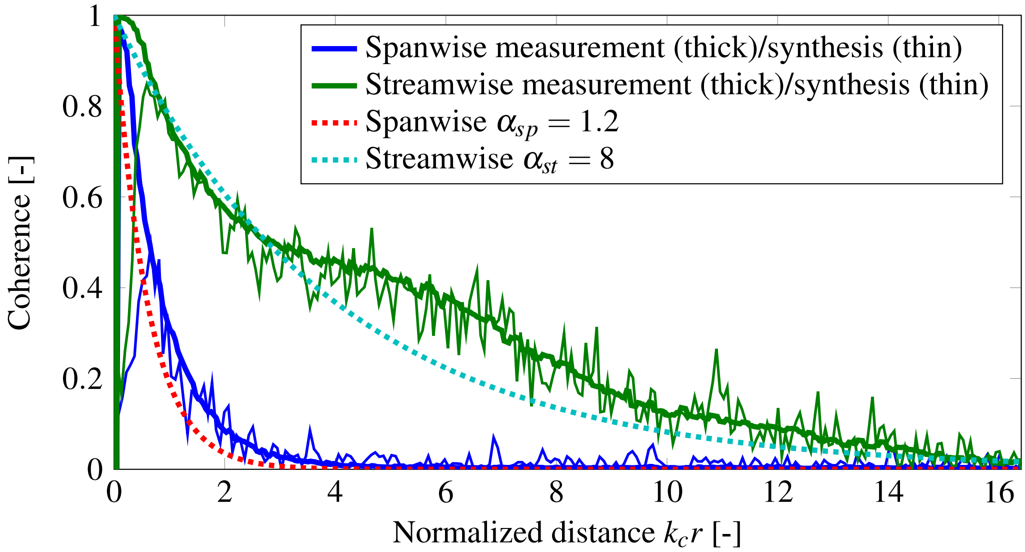

Figure 1: Spatial coherence of measured and synthesized pressure fields for $r_{sp} = 0.12\,\mathrm{m}$ and $r_{st} = 0.11\,\mathrm{m}$ compared to typical values from literature [11].

In this formula S_{ii} describes the power spectral density of x_i which is assumed to be identical at any point in the TBL pressure field. The measured PSD that is used here can be found in [9, Fig. 5]. The distances between the two points in the span- and streamwise directions are r_{sp} and r_{st} and the corresponding correlation lengths are L_{sp} and L_{st}. U_c is the convection velocity which is approximated as $42\,\mathrm{m/s}$ for the investigated flow speed of $62.4\,\mathrm{m/s}$. The correlation lengths are defined as $L_{sp/st} = \alpha_{sp/st} U_c/\omega$. Typical values for α are $\alpha_{sp} = 1.2$ and $\alpha_{st} = 8$ [11]. As can be seen in Fig. 1 the measurement results are similar but show some deviations especially in the streamwise direction for normalized distances $k_c r = \omega/U_c r$ between 4 and 14. In order to capture these deviations in the synthesis, a frequency dependence of α is assumed that follows the characteristics of the measurement data. The frequency range considered in this work is $50-500\,\mathrm{Hz}$.

The main task of the pressure field synthesis, which delivers the nodal forces for the harmonic analysis, is to calculate the CPSD described by Eq. 1 for any combination of FE nodes. This can be a numerically expensive task since it scales quadratically with the node number. The final CPSD matrix for a FE grid with N nodes is given by

$$\mathbf{S_{yy}}(\omega) = \begin{bmatrix} S_{11}(\omega) & S_{12}(\omega) & \cdots & S_{1N}(\omega) \\ S_{21}(\omega) & S_{22}(\omega) & \cdots & S_{2N}(\omega) \\ \vdots & \vdots & \ddots & \vdots \\ S_{N1}(\omega) & S_{N2}(\omega) & \cdots & S_{NN}(\omega) \end{bmatrix}. \tag{2}$$

Therein S_{ij} is the CPSD of the pressure signals at nodes i and j according to Eq. 1. By definition, $\mathbf{S_{yy}}$ results from the expectation $\mathbb{E}$ of the outer product of the vector of pressure signal spectra $\mathbf{Y} = [Y_1\, Y_2\, \cdots\, Y_N]^T$.

$$\mathbf{S_{yy}}(\omega) = \mathbb{E}\left\{\mathbf{Y}(\omega)\mathbf{Y}^H(\omega)\right\}. \tag{3}$$

Hence, in order to get the desired pressure signals $\mathbf{Y}$ from the known CPSD matrix $\mathbf{S_{yy}}$, the outer product operation described in Eq. 3 must be reversed. This is accomplished by an eigenvalue decomposition of $\mathbf{S_{yy}}$ with the orthogonal eigenvector matrix U and the diagonal and real eigenvalue

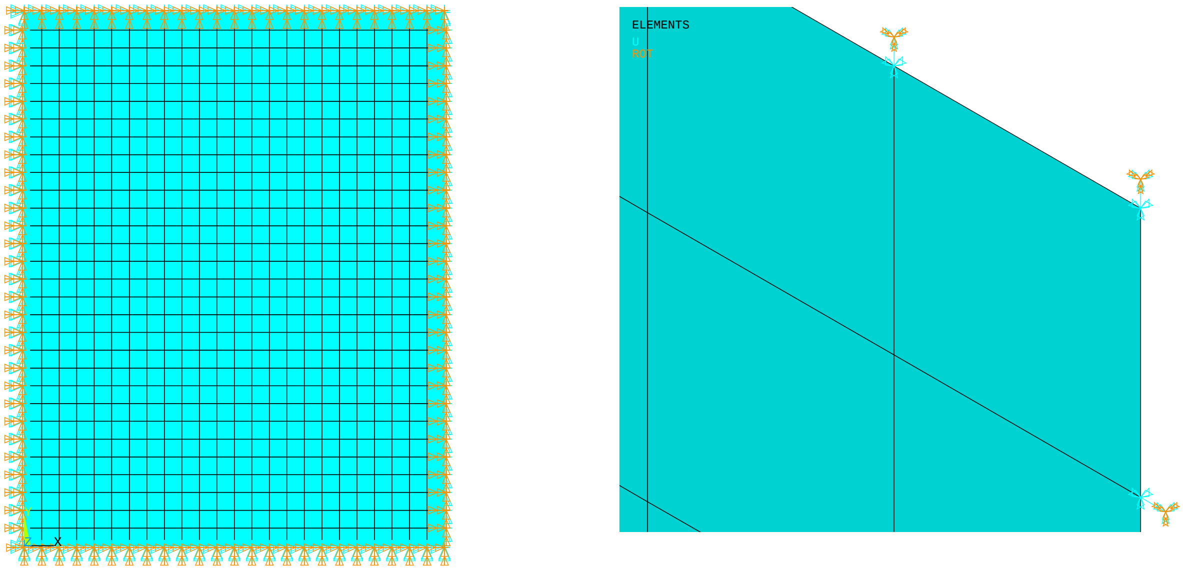

Figure 2: Structural model with FE grid (black), elements (green) and boundary conditions (green/brown). Complete model (left) and close-up of the upper right part (right).

matrix D.

$$\mathbf{S_{yy}}(\omega) = \mathbf{U}(\omega)\mathbf{D}(\omega)\mathbf{U}^H(\omega) = \underbrace{\mathbf{U}(\omega)\mathbf{D}^{1/2}(\omega)}_{\mathbf{G}(\omega)}\mathbf{D}^{1/2}(\omega)\mathbf{U}^H(\omega) = \underbrace{\mathbf{U}(\omega)\mathbf{D}^{1/2}(\omega)[\mathbf{U}(\omega)\mathbf{D}^{1/2}(\omega)]^H}_{\mathbf{G}(\omega)\mathbf{G}^H(\omega)} \tag{4}$$

The desired pressure signals $\mathbf{Y}$ result from filtering a vector $\mathbf{X}$ of N uncorrelated white noise signals of unit variance through the filter matrix $\mathbf{G}$. Equation 5 validates the filtering process $\mathbf{Y} = \mathbf{GX}$.

$$\mathbf{S_{yy}}(\omega) = \mathbb{E}\{\mathbf{Y}(\omega)\mathbf{Y}^H(\omega)\} = \mathbb{E}\left\{\mathbf{G}(\omega)\mathbf{X}(\omega)\mathbf{X}^H(\omega)\mathbf{G}^H(\omega)\right\} = \mathbf{G}(\omega)\mathbf{S_{xx}}(\omega)\mathbf{G}^H(\omega) = \mathbf{G}(\omega)\mathbf{G}^H(\omega). \tag{5}$$

The CPSD matrix $\mathbf{S_{xx}}$ equals the identity matrix because the white noises are uncorrelated and of unit variance. The required nodal forces are obtained from the pressure spectra $\mathbf{Y}$ by multiplication with the FE element size.

3.2 Structural model

The structure is modeled with FEM (Ansys) using shell elements for the plate and torsional springs to describe the fixture at the edges. The translation of the edge nodes of the plate is set to zero while the rotational degrees of freedom are unconstrained (see Fig. 2). The rotation axis of the torsional spring element is in parallel to the corresponding plate edge. The dimensions of the aluminum plate are $0.47 \times 0.37 \times 0.0011\,\mathrm{m}^3$ and the FE element size is $0.47/30\,\mathrm{m} \approx 1.6\,\mathrm{cm}$. The number of plate nodes is equal to 775. A frequency dependent stiffness of the torsional springs is defined to better approximate the boundary conditions. The spring stiffness is constant for all springs and a scaling factor is used to account for changes in the FE element size. The model updating is done iteratively by changing the stiffness values according to the relative locations of measured and simulated eigenfrequencies. A good agreement is achieved in the frequency range from 0 Hz to 500 Hz. The structural damping is set to 1 %.

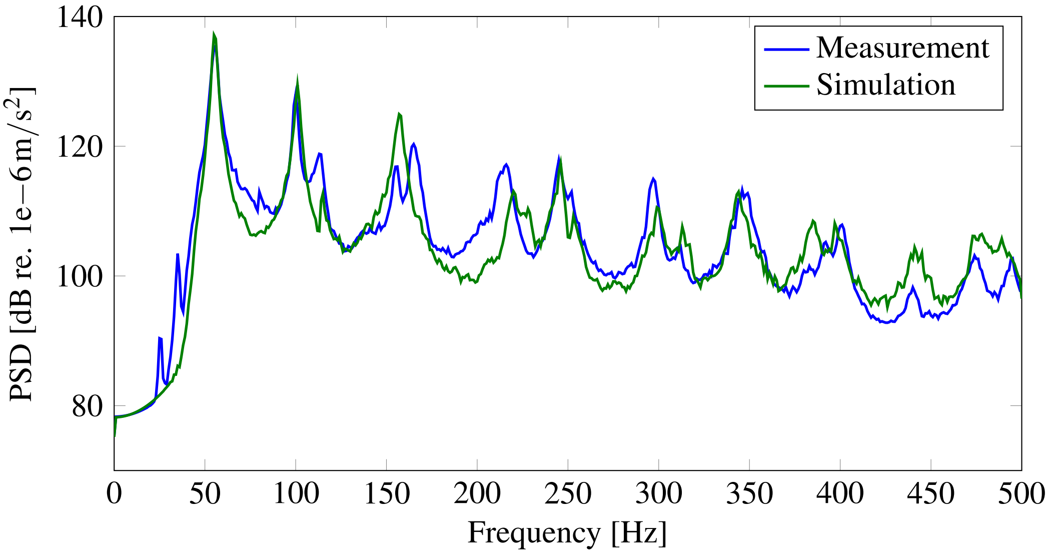

Figure 3: Mean Power spectral density of normal surface acceleration of the plate. Evaluated at 35 positions for a flow velocity of 62.4 m/s.

3.3 Response analysis

The response of the plate to the TBL excitation is simulated in Ansys with a full harmonic analysis. For this, the derived nodal forces and the FE model of the plate are used. The frequency resolution of the harmonic analysis is set to 5e−2 Hz. This equals a sample length of 20 s in the time domain resulting in a final frequency resolution of 1 Hz after 20 spectral averages. The numerical result is validated with the help of measurement data from 35 accelerometers equally distributed over the plate surface (see [9, Fig. 8]). Figure 3 compares the measured and simulated power spectral density averaged over all acceleration signals. The signals are high-pass-filtered because the first eigenfrequency of the plate is located around 55 Hz. Peaks in the measured response below 50 Hz are attributed to vibrations of the experimental setup (see [9, Fig. 1]). According to Fig. 3 a good agreement between simulation and measurement is achieved in the relevant frequency range from 50 Hz to 500 Hz. It is concluded that the synthesis method is able to produce valid data.

4. Vibration field synthesis

The vibration field synthesis makes use of the simulated structural response to a TBL excitation. It tries to emulate the desired vibration field by a targeted actuation of the structural dynamics. Twelve loudspeakers, corresponding to 60 loudspeakers per square meter, are used to synthesize the TBL-induced structural vibration up to 500 Hz.

4.1 Experimental setup

Figure 4 shows the experimental setup in the sound transmission loss facility. Apart from the vertical mounting of the plate and the thinner wooden base plate, the boundary conditions are comparable to the wind tunnel experiments. The TBL excitation is substituted by twelve loudspeakers positioned in front of the panel (see Fig. 4 (b) and (c)). Only the speakers co-located to the panel are selected (three rows with four speakers each). The structural response is measured on the opposite side in the semi-anechoic room (see Fig. 4 (a)). The calculation of the loudspeaker control signals $\mathbf{C}$ requires the measurement and inversion of the FRF matrix $\mathbf{F}$ from the loudspeakers to the structural vibration. For this task the structural response to each loudspeaker (swept sine excitation) is measured

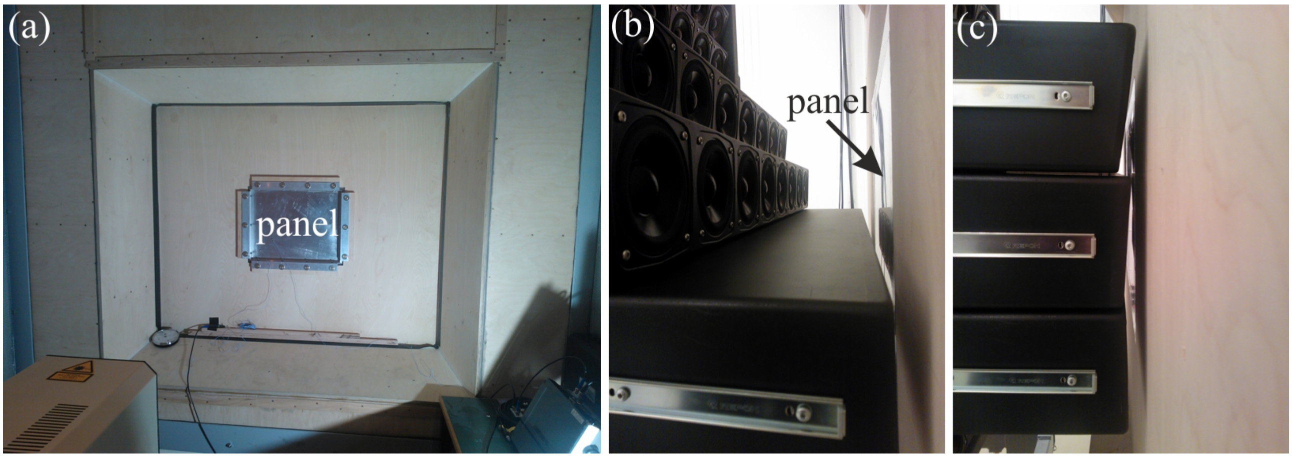

Figure 4: Setup in the transmission loss facility seen from the semi-anechoic room (a) and from the reverberation room (b) and (c).

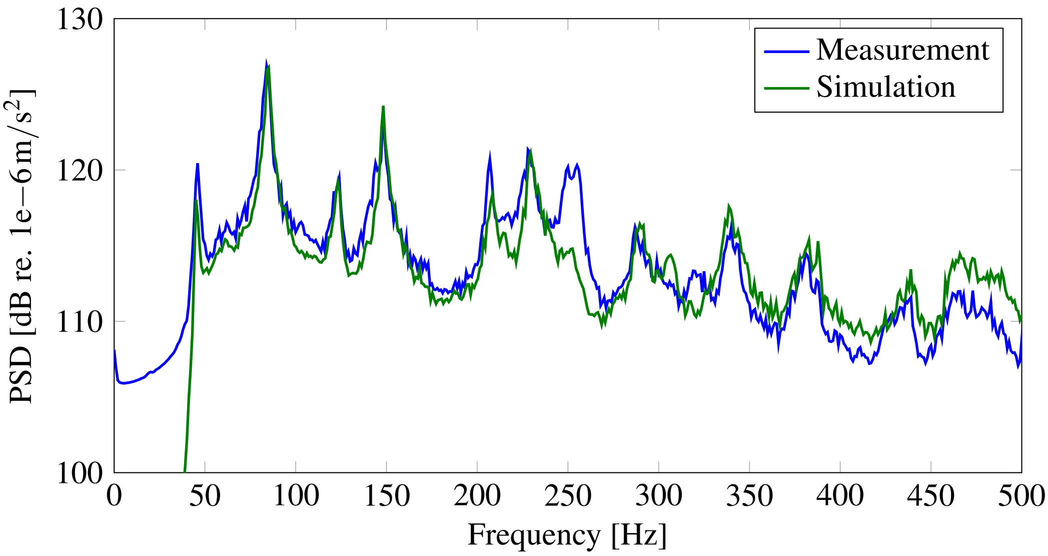

Figure 5: Mean Power spectral density of normal surface acceleration of the plate. Evaluated at 35 positions synthesized for a flow velocity of 62.4 m/s.

sequentially on a predefined grid with a scanning laser Doppler vibrometer. The target vibration field results from a harmonic analysis as described in subsection 3.3. The FRF matrix **F** and the vector of target vibration spectra **Z** is used for the calculation of the loudspeaker control signals according to Eq. 6.

$$\mathbf{C}(\omega) = \mathbf{F}^{+}(\omega)\mathbf{Z}(\omega) \tag{6}$$

Therein, $\mathbf{F}^{+}$ is the Moore-Penrose pseudoinverse of **F**. The vector of control signal spectra **C** is transformed to the time domain and saved as a wave file with twelve tracks. This file is imported in the audio software cubase which controls the sources of the loudspeaker array.

4.2 Results

Figure 5 compares the measured and the simulated power spectral density averaged over all acceleration signals. In this case the measurement data is gathered in the transmission loss facility with a loudspeaker excitation. The loudspeakers are controlled to emulate the structural response to a TBL having the characteristics described in subsection 3.1. Due to deviations in the boundary conditions of the wind tunnel and the transmission loss facility, the first eigenfrequency of the plate is shifted from 55 Hz to 46 Hz. This required an updating of the FE model of the plate to the new eigenfrequencies followed by a harmonic analysis. To prevent an overloading of the loudspeakers, the control signals are high-pass-filtered with a cut-off-frequency of 60 Hz. Therefore, the fundamental mode is not excited as strong as in the wind tunnel. Figure 5 shows a good agreement between simulation and measurement in the relevant frequency range from 50 Hz to 500 Hz. It is concluded that the synthesis method and the experimental setup in the transmission loss facility permit a good approximation of the structural response to a TBL excitation.

5. Conclusion

This contribution describes a method for the synthesis of TBL-induced structural vibration which does not require flight or wind tunnel tests. An integral part of this fully self-contained method is the identification and inversion of the FRF matrix from the loudspeakers to the structural vibration. Prerequisites for this method are a model of the structural dynamics of the test specimen, a suitable TBL excitation model and a laboratory equipped with a scanning laser Doppler vibrometer and a loudspeaker array. The method is validated for a simple plate but can also be applied to more complex structures. The main benefit of this method is the relatively small amount of acoustic sources compared to the complexity of a direct pressure field synthesis. This, however, implies a severe drawback, which is the required knowledge of the structural dynamics of the test specimen. If the structure has many lightly damped modes, a slight temperature shift might induce major shifts of the eigenfrequencies which renders the simulated vibration response invalid. Any modification of the structural system, e.g. the application of sensors and/or actuators, might have an influence that must be incorporated in the structural model. Due to its thinness the utilized plate was especially sensitive to slight changes in the environmental or boundary conditions.

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