PROCEEDINGS of the
23rd International Congress on Acoustics

9 to 13 September 2019 in Aachen, Germany

Modal Analysis for damage detection in structures by non-contact measurements with a commercial microphone array

Olaf BÖLKE[1]; Jan HEIMANN[1]; Joaquin GARCIA[2]

[1] gfai tech GmbH, Berlin, Germany

[2] Universidad Nacional de Tres de Febrero (UNTREF), Buenos Aires, Argentina

ABSTRACT

When monitoring critical structures, fatigue fracture, deformations, holes and much more are cases of failure which must be detected at an early stage. Changes of the modal parameters (eigenfrequencies, attenuation ratios, and mode shapes) of the structure give information about the extent and the location of the deterioration. Conventional measurement methods (i.e. acceleration sensors, laser vibrometers, etc.) for vibration analysis have the disadvantage that they can either "detune" the vibration modes due to their own weight and/or require a long measurement time due to punctual measurements. In contrast, the use of a suitable microphone array allows the high-resolution acquisition of the entire surface vibration covered by the array. Thus, the modal parameters of interest are determined by measuring the pressure fluctuations in the near field of the structure. A commercial acoustic camera with 120 microphones (Fibonacci120, gfai tech GmbH) is used for this purpose. On the basis of artificially generated failure cases (load fracture, inhomogeneities, etc.) on application-oriented, large-area structures, a method for the detection of failure cases using a microphone array is demonstrated.

Keywords: Microphone Arrays, SONAH, Modal Analysis

1. INTRODUCTION

The dynamic response of a structure is an important aspect of understanding the structure's behavior. Strong vibrations can generate not only unwanted noises but also structural wear and thus material weakness. The risk of rupture in structures under load conditions leads to necessary understand the structure's dynamic response when a vibration is present. This can be achieved by monitoring the natural frequencies or magnitudes of frequency response functions to detect faults and mechanical failures [1]. Typically, experimental modal analysis (EMA) is carried out to obtain the modal parameters such as mode shapes, eigenfrequencies, and damping ratios. This consist in applying a defined force (e.g by a shaker or an impact hammer) and record the structural vibration response using different transducers such as accelerometers and laser Doppler vibrometers (LDV). However, the laser scanning of several points can be time-consuming when large surfaces are under study. On the other hand, attached external sensors on the structure can "detune" its natural response due to the added mass. Thus, recent studies proved that non-contact measurements can be used, especially for the analysis of very light and fragile structures [1].

This paper is focused on detecting failure cases on large-area structures. The EMA is used to determine the eigenmodes by measuring the sound pressure with a contactless microphone array. Artificial inhomogeneities (additional mass and notches) are applied to a plate surface to test the detection of typical structural failure. In current studies, such as [2], qualitative investigations of the determined modes have already been carried out, whereby the used algorithms usually provide unsatisfying results. In this study the measured pressure fluctuations are assigned to the surface vibrations using the near-field holography method SONAH („Statistically Optimal Near-field Acoustical Holography") [3]. Furthermore, the analyses of the data collected by a commercial

[1] boelke@gfaitech.de [2] joaquin.garcia1900@hotmail.com

microphone array (Fibonacci120 AC Pro, gfai tech GmbH) are carried out with the software „NoiseImage“. The Modal Assurance Criterium (MAC) is used to evaluate the change in modal shape[4]. The MAC values are usually interpreted as a statistical indicator for the comparability of two modal forms.

2. THEORY

2.1 Experimental Modal Analysis

The response of a structure to a punctual force can be described by modal superposition [5]. Considering a plate with mode shapes $\phi_r(x,y)$ for the eigenfrequencies ω_r and modal damping ratios ξ_r, then the transverse displacement η of the surface is given by:

$$\eta(x,y,t) = \sum_{r=1}^{\infty} \frac{\tilde{F}}{M_r} \frac{\phi_r(x,y)\,\phi_r(x_f,y_f)}{\omega_r^2 - \omega^2 + 2i\xi_r\omega_r\omega}\, e^{i\omega t}, \tag{1}$$

for a harmonic force $\tilde{F}$ at position (x_f, y_f), generalized mass M_r and angular frequency ω for all modes r.

Experimental modal analysis (EMA) is used to find the modal properties $(\phi_r, \omega_r, \xi_r)$ in the model given in (1). This is done by fitting frequency response functions (FRFs), which are determined from the measured force excitation and the measured system response. The modal parameters are estimated using the Polyreference Least Squares Complex Frequency (POLY-LSCF) algorithm [6]. The POLYLSCF algorithm uses (weighted) least square approaches of multiple-input-multiple-output frequency response functions. The stability diagrams determined during the analysis allow an easy analysis even of complex systems with highly damped modes and/or large modal overlap. Further information on mathematical implementation and validation using experimental and numerical models can be found in the work of Peeter et. all [7] and Phillips et. all [6].

2.2 Modal Assurance Criterion

The Modal Assurance Criterion (MAC) is defined as the normalized scalar product of two modal vectors v_i, v_j:

$$MAC_{ij} = \frac{\left|v_i^H\, v_j\right|^2}{\left(v_i^H\, v_i\right)\left(\, v_j^H\, v_j\right)}, \tag{2}$$

where $(\cdot)^H$ is the complex conjugate transpose (Hermitian) of a matrix.

The MAC is a statistical indicator that can be used as a measure of the quantitative comparability of two modal forms. A MAC value of 0 indicates that the modes are not consistent and a value of 1 represents a fully consistent mode shape [4]. Therefore, the MAC matrix (cross correlation of all modes in a given range) obtained by EMA and/or FEA consists of zeros, except in the diagonal where the values are close to 1. This least squares-based form of linear regression analysis yields an indicator that is most sensitive to the largest difference between comparative values and results in modal assurance criterion that is insensitive to small changes or small magnitudes. Under certain conditions, such as a stationary, linear system state, the MAC can also be taken as a criterion for the orthogonality of the compared modes. However, the MAC values must be interpreted according to the measurement situation: On the one hand an insufficient local resolution of two orthogonal mode shapes can lead to high MAC values, while on the other hand lower values can be achieved e.g. by noisy measurement signals [4].

2.3 SONAH

Acoustic near-field holography describes a method of estimating the sound field on the source surface by measuring the acoustic quantities (usually sound pressure or sound velocity) at a small distance from the source surface. The requirements on the microphone array when using traditional nearfield acoustical holography (NAH) are very high. A uniform microphone arrangement is required, where the array covers the entire sound source. In addition, the minimum microphone distance must be less than half the wavelength at the highest frequency of interest. However, SONAH does not have these restrictions: It can not only work with irregular arrays, it also gives good results when the array is smaller than the source [3]. The latter is due to the fact that, unlike traditional NAH, the strong spatial window effects do not occur [8].

The publication of Puhle et. al [9] describes, among other things, the mathematical implementation of two advanced NAH methods such as SONAH and HELS (Helmholtz Equation Least-Squares). In addition, Puhle et. al. also qualitative validates the experimentally determined mode shapes from acoustic measurements using SONAH and HELS with LDV-measurements. This work continues the investigations by concentrating not only on the amplitudes of the calculated surface oscillations, but also on the complex amplitude including the spatially distributed phase. Thus, the MAC is used to quantify the quality of certain mode shapes using FEM, LDV and SONAH.

3. MEASUREMENT SETUP

The vibration response of a stainless-steel plate (600 mm x 600 mm x 4mm) to a force excitation was investigated. The plate was supported by four rubber bands in a metal frame, which were guided through corresponding holes in the corners of the plate (diameter 7 mm). Thus, free boundary conditions could be realized for the relevant frequency range (100 to 1000 Hz). Using a Shaker-Stinger setup (PCB SmartShaker with an integrated power amplifier, model K2007E01), the plate was excited perpendicular to the surface, whereby the force was measured using a force sensor (PCB type 208C02), see Fig. 1 (left). White noise with frequencies from 100 to 1000 Hz was used as an excitation signal. The plate response was recorded by measuring the sound pressure oscillation with a commercially available microphone array (acoustic camera "Fibonacci120", gfai tech GmbH). In order to improve the signal-to-noise ratio, the measurements were performed in an anechoic chamber. The microphone array used is a multifunctional array and is well suited for both SONAH and conventional beamforming. Preliminary investigations have shown that the best results can be obtained with a microphone distance of approx. 14 mm to the measuring surface, see Fig. 1 (right). The pressure oscillations were recorded using a data acquisition system (mcdRec 721B, gfai tech GmbH) with a sampling rate of 48 kHz and averaged over 16 seconds. The particle velocity on the surface of the plate was determined using the nearfield hologram module of the Noise Image software (gfai tech GmbH), which is based on the SONAH algorithm.

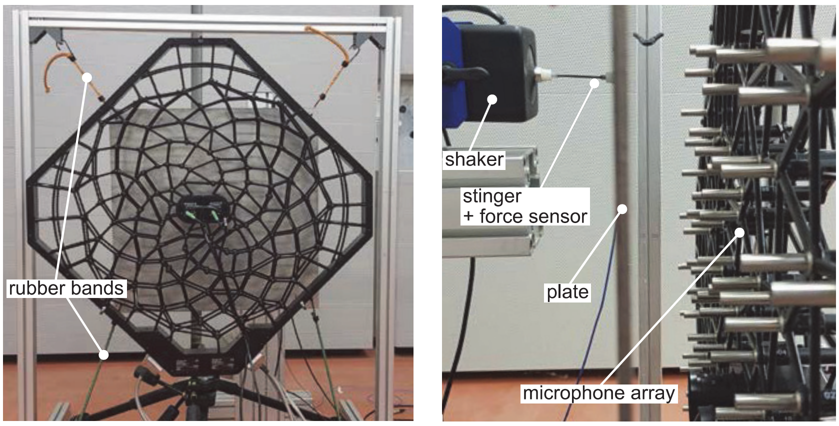

Figure 1 – Setup for determination of the plate mode shapes with a commercial microphone array (Fibonacci120 AC Pro, gfai tech GmbH) in an anechoic chamber. Left: rubber mounted plate with microphone array. Right: Alignment of shaker, plate, and parallel microphone array.

Three measurements were carried out to investigate the effects of a geometric change on the mode shapes: 1) plate without changes, 2) plate with simulated fatigue fracture (ca. 2 mm deep and 30 cm long notch, horizontal on the lower third of the plate) and 3) plate with additional mass (300 mm x 25 mm x 5 mm brass beam, horizontal on the lower third of the plate).

In a preliminary study [10], the mode shapes were simulated and compared with the results of measurements using laser Doppler vibrometry (LDV) and SONAH to verify the measurement methodology. Figures 2 and 3 show representative mode shapes determined by FEM (left column), LDV (middle column) and SONAH (right column). The normalized amplitude (upper row) and the corresponding phase (lower row) of the plate are presented. Both the mode shapes based on LDV and SONAH correspond qualitatively well to the values estimated by FEM. It is noticeable, however,

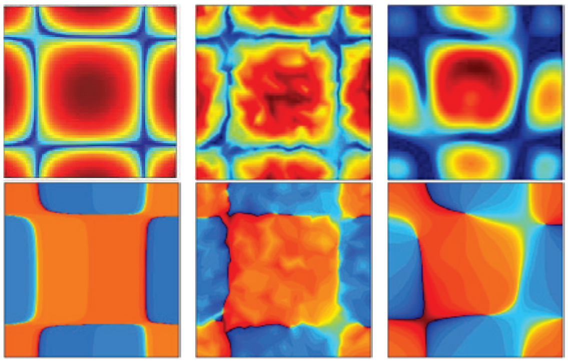

Figure 2: Amplitude (top) and phase (bottom) of mode No 8; FEM (left), LDV 170 Hz (middle), SONAH 172 Hz (right) [10]

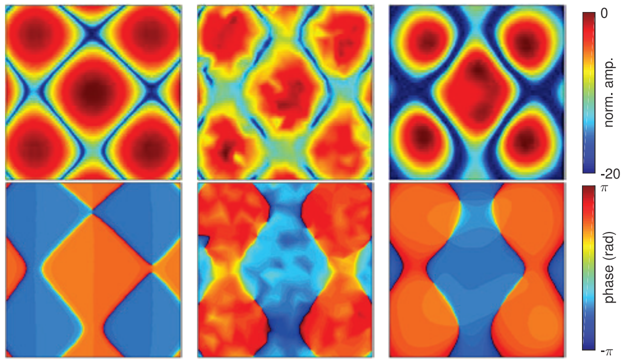

Figure 3: Amplitude (top) and phase (bottom) of mode No 14; FEM (left), LDV 324 Hz (middle), SONAH 324 Hz (right) [10]

that the vibration amplitudes in the outer edge of the plate deviate from the LDV results or the calculated values, respectively. It can be assumed that this is related to the microphone density decreasing with the distance to the center of the plate. However, it should be noted that the phase of the vibration mode can be determined with less noise by SONAH than with the comparable LDV, whose quality depends strongly on the surface quality.

4. RESULTS

The three different panel designs are compared to assess the suitability of the microphone array for error detection based on the mode shapes determined. Due to the minor changes in the geometry, only slight shifts of the resonance frequency (<30Hz) in the transfer function between force and determined sound pressure fluctuations can be determined. However, it was found that some modes are no longer excited by the fix driving point position due to the asymmetric change in geometry. In the following, therefore, only the mode shapes that could be determined for all three plate configurations are compared.

Figures 4 to 11 show the amplitude (upper row) and phase (lower row) of the mode shapes based on SONAH. The columns correspond to the three different plate configurations (from left to right): 1) plate without changes, 2) plate with a notch, and 3) plate with an additional mass. The determined Auto-MAC values of the presented mode shaped showed exclusively high values in the diagonal, which can be interpreted as a strong linear independence from the determined modes. Where the changes in the oscillation mode due to the simulated fatigue break (case 2) in the frequency range investigated here (100 to 600 Hz) shows only minor changes in the mode shape (see e.g. Fig. 6), the additional mass (case 3) shows a more significant change compared to the reference case (case 1). Due to the additional mass in the lower part of the plate at case 3, a stiffening occurs, which causes a significantly changed mode shape beginning from the 2nd mode shown here (Figure 5, right). Due to the horizontal orientation, a higher influence on modes that would have an oscillation node in the region of mass can be seen here (compare, for example, Figures 10 and 11).

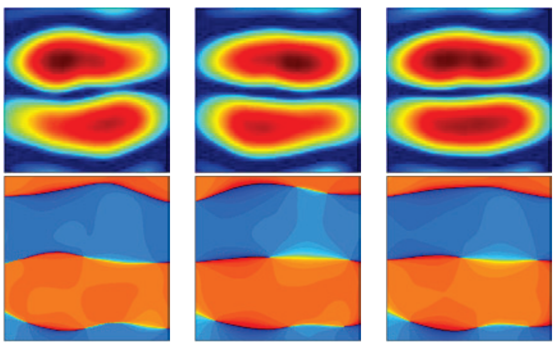

Figure 4: Amplitude (top) and phase (bottom) of the plate mode No 1; Left: 163 Hz (Case 1); Middle: 158 (Case 2); Hz; Right: 163 Hz (Case 3)

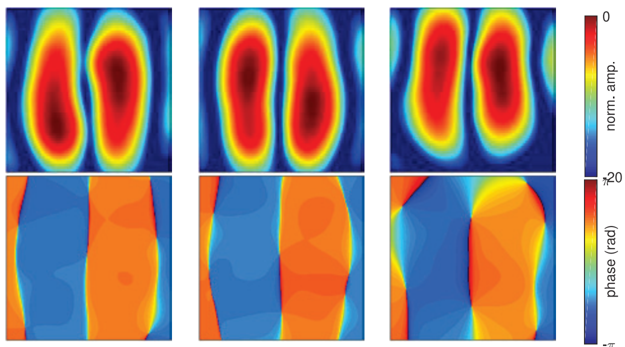

Figure 5: Amplitude (top) and phase (bottom) of the plate mode No 2; Left: 165 Hz (Case 1); Middle: 164 (Case 2); Hz; Right: 173 Hz (Case 3)

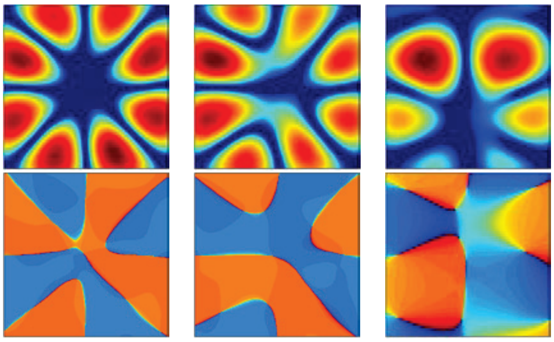

Figure 6: Amplitude (top) and phase (bottom) of the plate mode No 3; Left: 186 Hz (Case 1); Middle: 184 (Case 2); Hz; Right: 196 Hz (Case 3)

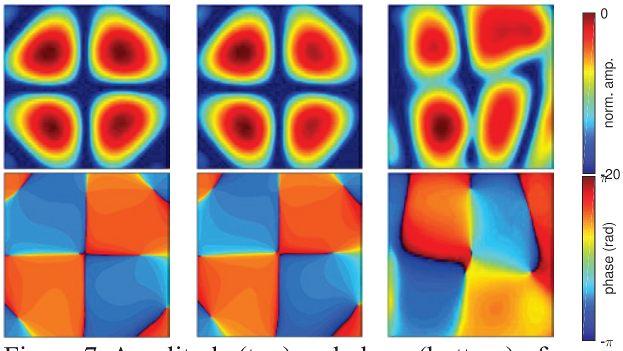

Figure 7: Amplitude (top) and phase (bottom) of the plate mode No 4; Left: 206 Hz (Case 1); Middle: 204 (Case 2); Hz; Right: 237 Hz (Case3)

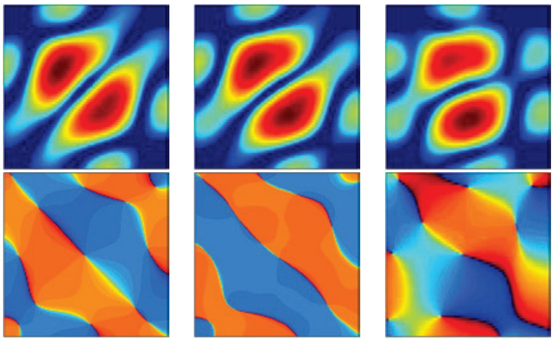

Figure 8: Amplitude (top) and phase (bottom) of the plate mode No 5; Left: 280 Hz (Case 1); Middle: 277 (Case 2); Hz; Right: 283 Hz (Case 3)

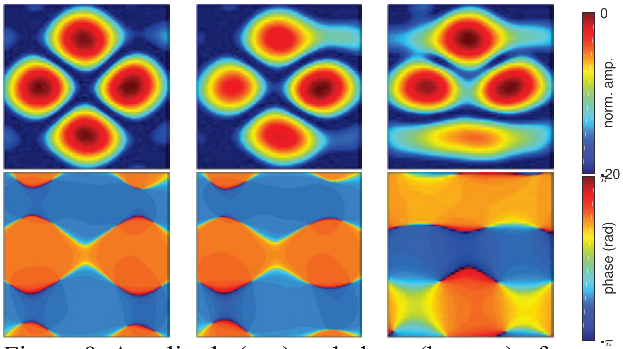

Figure 9: Amplitude (top) and phase (bottom) of the plate mode No 6; Left: 313 Hz (Case 1); Middle: 306 (Case 2); Hz; Right: 314 Hz (Case 3)

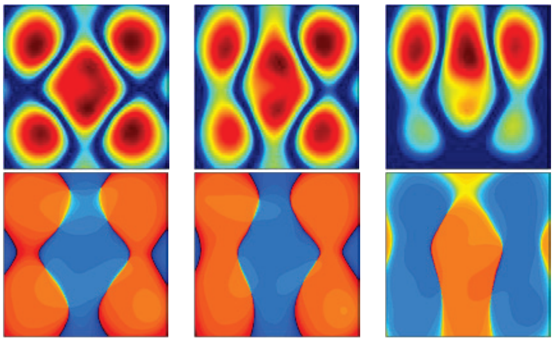

Figure 10: Amplitude (top) and phase (bottom) of the plate mode No 7; Left: 324 Hz (Case 1); Middle: 322 (Case 2); Hz; Right: 329 Hz (Case 3)

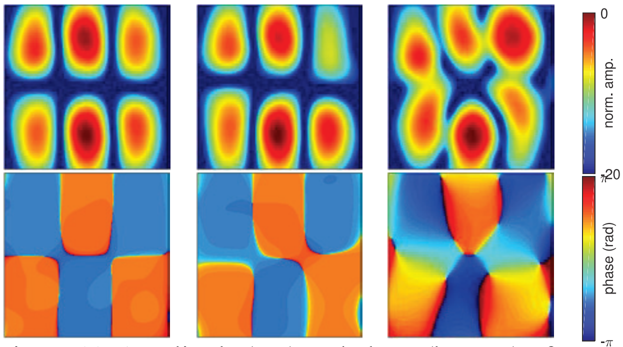

Figure 11: Amplitude (top) and phase (bottom) of the plate mode No 8; Left: 350 Hz (Case 1); Middle: 346 (Case 2); Hz; Right: 433 Hz (Case 3)

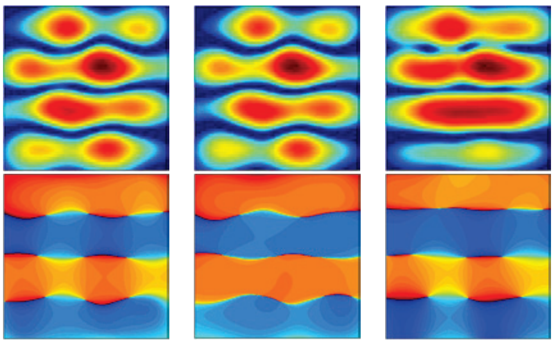

Figure 10: Amplitude (top) and phase (bottom) of the plate mode No 9; Left: 523 Hz (Case 1); Middle: 520 (Case 2); Hz; Right: 517 Hz (Case 3)

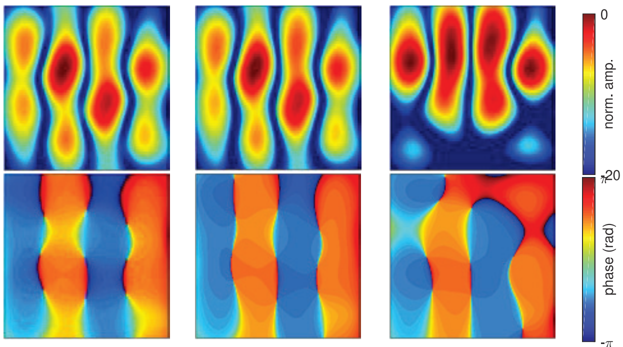

Figure 11: Amplitude (top) and phase (bottom) of the plate mode No 10; Left: 528 Hz (Case 1); Middle: 527 (Case 2); Hz; Right: 532 Hz (Case 3)

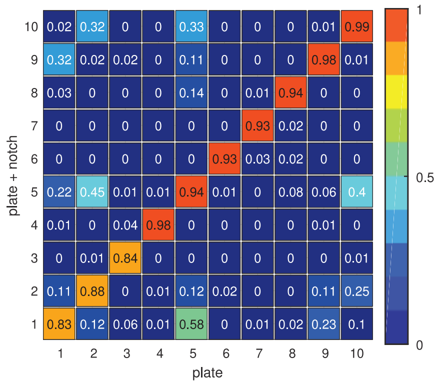

Figure 12: MAC-Values by Eq. (2) between plate (case 1) and plate with notch (case 2)

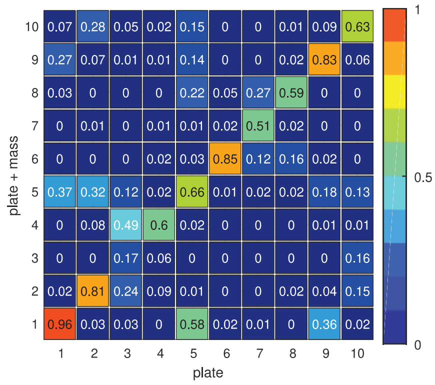

Figure 13: MAC-Values by Eq. (2) between plate (case 1) and plate with mass (case 3)

By means of the MAC values according to Eq. (2), the different mode shapes can be compared qualitatively. Figures 12 and 13 show the MAC values for the cases 2 and 3, respectively, with respect to the plate without modification, case 1 (see Figs. 4 to 11). As expected, the largest values are shown on the diagonal. Only the modes 1 and 5 show MAC-values above 50%. It can be assumed that this is due to the low microphone density near the edges of the plate, so that the vibration field can no longer be resolved high enough to clearly separate the mode shapes.

In accordance with the previous observations, the MAC values in Fig. 12 show much lower values than in Fig. 13, which highlights the much stronger influence of the additional mass on the mode shapes compared to the notch. However, it can be shown that for the first three modes determined in this investigation, slight deviations of the MAC values can also be observed for the plate with the notch.

5. CONCLUSIONS

It has been shown that with a microphone array using SONAH the eigenmodes of a vibrating plate can be quantitatively well estimated. The preliminary investigations [10] showed a good agreement between the simulated and LDV vibration modes, both in amplitude and phase distribution. When comparing the test cases examined here, the influence of an additional mass on the test structure and the influence (within certain limits) of a notch from the certain mode shapes could be determined by contactless microphone array measurements.

The changes in the mode shapes indicate, among other things, a stiffening in the area in which the additional mass is located. Therefore, comparisons of the specific mode shapes allow conclusions to be drawn about the position and type of geometry changes. With an appropriate learned algorithm, e.g. via pattern recognition, a control algorithm based on a microphone array could be created, which can distinguish between different error or failure cases. The advantage of the microphone array measurement technology over the usual LDV-based measurement methods lies in the simplified and less time-consuming measurement method, which measures the entire surface of the array. Future works are focused on further investigations to fully clarify the limitations of the array design used in this investigation.

REFERENCES

1. Farshidi, R., Trieu, D., Park, S. S., and Freiheit, T. Non-contact experimental modal analysis using air excitation and a microphone array. Measurement, 43 (6), 755–765, (2010).
2. Velsen, A. L. van, Moers, E. M. T., Arteaga, I. L., and Nijmeijer, H. Plate mode identification using modal analysis based on microphone array measurements. Euronoise. 2015.
3. Brüel & Kjær. Technical Review No. 1., 2005.
4. Pastor, M., Binda, M., and Harcarik, T. Modal Assurance Criterion. Procedia Engineering, 48, 543–548, 2012.
5. Lars B., Liingaard, M. Experimental modal analysis Ibsen. Aalborg Universitet. Denmark; 2006.
6. Phillips, A.W. and Allemang, R. J. A Low Order Implementation of the Polyreference Least Squares Complex Frequency (LSCF) Algorithm. International Conference on Noise and Vibration Engineering. 2004.

7. Peeters, B., Guillaume, P., Auweraer, H. V. D., Cauberghe, B., Verboven, P., and Leuridan, J. Automotive and aerospace applications of the PolyMAX modal parameter estimation method. IMAC. Vol. 22. 2004.
8. Steiner, R. and Hald, J. Near-field Acoustical Holography without the Errors and Limitations Caused by the Use of Spatial DFT. International Journal of Acoustics and Vibration, 6 (2), 83–89, 2001.
9. Puhle, C. On the sunflower spiral: acoustical holography results. InterNoise. 2016.
10. Bölke, O., Garcia, J., Heimann, J. Experimental Modal Analysis based on non-contact measurements with a commercial microphone array. ICSV26. 2019