Conference on Modelling Fluid Flow (CMFF'09)
The 14th International Conference on Fluid Flow Technologies
Budapest, Hungary, September 9-12, 2009

INVESTIGATION OF THE VIBRATIONAL AND THE FLOW-INDUCED SOUND DUE TO THE TURBULENT FLOW OVER DIFFERENT PLATE STRUCTURES

Stefan MÜLLER[1], Stefan BECKER[2], Jens GRABINGER[3], Thomas BIERMEIER[4], Denis BLANCHET[5]

[1] Corresponding Author. Institute of Fluid Mechanics (LSTM), Friedrich-Alexander-University Erlangen-Nuremberg. Cauerstr. 4, D-91058 Erlangen, Germany. Tel.: +49 9131 85 29480, Fax: +49 9131 85 29503, E-mail: stefan.mueller@lstm.uni-erlangen.de
[2] Institute of Fluid Mechanics, Friedrich-Alexander-University Erlangen-Nuremberg. E-mail: stefan.becker@lstm.uni-erlangen.de
[3] Department of Sensor Technology (LSE), Friedrich-Alexander-University Erlangen-Nuremberg. E-mail: jens.grabinger@lse.eei.uni-erlangen.de
[4] Institute of Fluid Mechanics, Friedrich-Alexander-University Erlangen-Nuremberg. E-mail: thomas.biermeier@lstm.uni-erlangen.de
[5] ESI GmbH Munich (Germany). E-mail: dbl@esigmbh.de

ABSTRACT

Experimental and numerical investigations of the fluid-structure interaction as well as the sound, induced by the turbulent flow over a flexible plate structure and radiated to the acoustic far field, are presented. To modify the standard inflow condition that is a fully turbulent boundary layer, an obstacle is placed upstream of a flexible plate. In order to investigate the influence of such a plate on the wall-near flow field, the flow-induced vibration of the plate structure is measured using a laser-scanning vibrometer. Furthermore unsteady pressure measurements are performed at the plate surface. Based on the measurements of the vibration of the flexible plate, the radiation of vibrational sound to the acoustic far field is computed employing a finite-element (FE) acoustics solver. The sound transmission into a plenum chamber mounted at the back side of the flexible plate is calculated by use of a Statistical Energy Analysis (SEA) model based on the results of the unsteady pressure measurements. Microphone measurements are carried out to generate a basis of comparison. The experiments are performed in an acoustic wind tunnel. The results show that only tonal sound components are caused by the structural movement.

Keywords: FEM, flexible plate, fluid-structure-acoustic interaction, scanning vibrometry, SEA, sound

NOMENCLATURE

D	$[m]$	square cylinder edge length
E	$[N/m^2]$	modulus of elasticity
Re	[-]	Reynolds number
U_∞	$[m/s]$	free-stream velocity
h	$[m]$	plate thickness
x	$[m]$	main flow direction
y	$[m]$	spanwise direction
ν	[-]	Poisson number
ρ	$[kg/m^3]$	density

Subscripts and Superscripts

g	sheet glass plate
a	aluminium plate

1. INTRODUCTION

In many technical applications one can find constellations where a fluid is flowing over a flexible plate structure. Due to turbulent fluctuations in the fluid, the structure is irregularly loaded with pressure and shear stress forces causing a movement of the plate. Additionally, the plate vibration influences the flow field so that there is a two-sided coupling between fluid and structural mechanics. In this situation, two mechanisms of sound generation arise. On the one hand, vibrational sound is generated by the structural movement of the plate. On the other hand, the turbulent fluctuations in the flow field lead to the production of flow-induced sound. Examples for such a case are coverings and panellings of cars and airplanes or windows of conveyances, where flow-induced as well as vibrational noise is radiated into the passenger cabin. In order to take measures towards a reduction of vibrational and flow-induced noise, a better understanding of the mechanisms leading to the emission of sound in situations with fluid-structure interactions is necessary.

Up to now there is very little work towards a comprehensive investigation of the fluid flow, the structural mechanics and the acoustics as well as the interaction of the different fields (see, e.g., Vergne et al. [1]).

In the present work, the interaction of a fluid flow with a flexible plate structure and the resulting acoustic field was studied in detail. For this purpose, a test case was developed which represents a simplified model of a car side window. A description of the test case is given in the following section. Both experimental and numerical investigations of the test case were carried out. The experiments were performed in a low-noise wind tunnel using microphone measurements and unsteady pressure measurements. Flow-induced structural vibrations were determined by a laser-scanning vibrometer. Based on the dataset of the vibrometer measurements, the sound radiation to the acoustic far field was computed employing a finite-element acoustics solver. Additionally, the sound transmission in a plenum chamber mounted at the back side of the flexible plate is calculated with an SEA model. Further details of the numerical and the experimental methods are given below. Finally, the results of the study concerning the flow field, the vibration of the flexible plate and the generated sound are presented.

2. MODEL SETUP

To understand the physics of the fluid-structure-acoustic interaction in the flow over a flexible plate, a simplified model was created which covers the major features of relevant technical application cases. The basic setup consists of a flexible plate structure which is part of an otherwise rigid wall. The streamwise and spanwise extension of the whole setup amounts to 1000 *mm* × 660 *mm*, respectively. The flexible plate is made of sheet glass with a thickness of $h_g = 3$ *mm*. The density ρ is 2300 kg/m^3, the modulus of elasticity E is 6.2×10^{10} N/m^2, and the Poisson number ν is 0.24. For the SEA a frequency-independent damping loss factor of 6 % was used. The streamwise and spanwise extension of the flexible plate amounts to 170 *mm* × 170 *mm*, respectively. The plate is joined over a length of 10 *mm* at the four edges on a massive aluminium construct (45 *mm* thickness) to avoid coupling between plate and mounting. This results in an effective area of the flexible plate of 150 *mm* × 150 *mm*. Both the top and the bottom surface of the flexible plate are freely accessible. In order to modify turbulence characteristics of the inflow, an obstacle was mounted upstream of the flexible plate. A sketch of the setup is depicted in Figure 1.

In this paper, the results of two different configurations are presented, one case with a square cylinder obstacle in front of the flexible plate and the other without any obstacle. The edge length of the square cylinder is $D = 20$ *mm*. The obstacle was positioned spanwise directly in front of the flexible part of the glass plate and just on top of the rigid aluminium plate. In the presented investigations, a free-stream velocity of $U_\infty = 40$ *m/s* was applied, corresponding to a Reynolds number of $Re = 52\,000$ based on U_∞ and D.

To generate a basis of comparison, additional investigations were carried out with a rigid aluminium plate in place of the glass plate. The thickness of the aluminium plate amounts to $h_a = 10$ *mm*. For validation of the SEA, measurements were conducted with a plenum chamber with sound-reflecting walls mounted on the back side of the massive aluminium plate, to model, e.g., the interior of a vehicle. The streamwise and spanwise extension of the chamber with a depth of 800 *mm* is 570 *mm* × 570 *mm*, respectively.

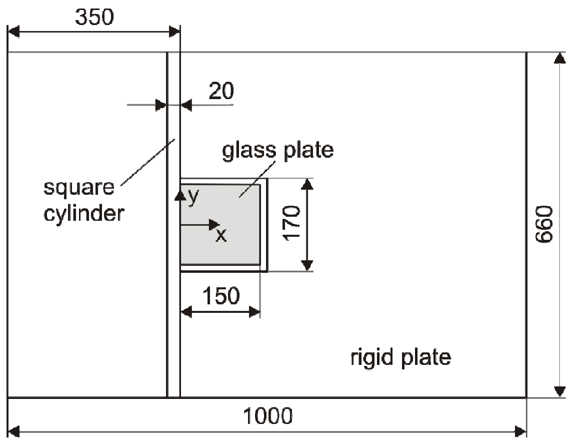

Figure 1. Schematic drawing of the setup, top view (units in *mm*)

3. EXPERIMENTAL METHOD

The measurements were performed in the acoustics wind tunnel of the University of Erlangen-Nuremberg, which is equipped with sound absorbers (anechoic chamber condition). A description of the wind tunnel is given by Becker et al. [2]. Figure 2 shows a sketch of the measurement setup with the equipment employed.

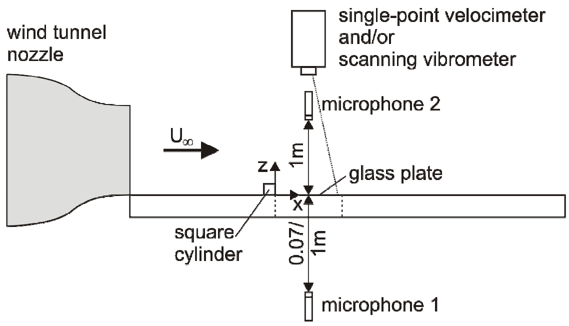

Figure 2. Sketch of the measurement setup, side view

Two 1/2" free field condenser microphones (B&K 4189), each at a distance of 1 *m* and perpendicular to the overflown and the non-overflown side of the flexible plate, are employed for the sound measurements. Additionally, the distance of the microphone averted from the turbulent flow is reduced to 0.07 *m*. This is approximately the distance, where a car driver would perceive the sound. To compare the results of the SEA with sound measurements, a microphone is flush placed in the centre point of the wall of the plenum chamber opposite to the glass plate.

Phase-resolved laser scanning vibrometer measurements are carried out to identify the vibration modes of the flexible plate. For this purpose, a Polytec single-point laser-Doppler velocimeter (OFV-303) is used together with a Polytec scanning vibrometer (OFV-056) to observe the flexural modes of the plate at different frequencies. The single-point vibrometer constantly detects the vibrations of the centre point of the plate while the surface of the plate is scanned with the scanning vibrometer. This measurement yields the amplitude and the phase of the plate vibration at each node of the measurement grid, which involves 31 × 31 points (5 *mm* spatial resolution) in streamwise and spanwise direction, respectively. The measurement duration is chosen in such a way to achieve a frequency resolution of 1.6 *Hz* up to a cut-off frequency of 5 *kHz*.

To investigate the structure-acoustic interaction, correlation measurements of the single-point velocimeter and the microphones are carried out. Simultaneous recording of the radiated sound in the far-field and the velocity of the midpoint of the plate surface is guaranteed by the data acquisition card employed (NI 4472).

For the measurement of unsteady pressure fluctuations at high sample rates, a piezoresistive pressure transducer (Kulite XCQ-093-5SG) is applied. The measurements are made at different positions of the glass plate and the aluminium plate, respectively. At each location, a measurement time of 30 *s* is used at a sample rate of 44 *kHz*.

4. FINITE ELEMENT SCHEME

For the acoustics computations, the in-house finite-element multiphysics solver CFS++ [3] is applied. This program is based on a finite-element discretization of the wave equation which describes the propagation of sound to the acoustic far field. Structure-acoustic coupling is incorporated by applying appropriate boundary conditions to the wave equation. The acoustic particle velocity at the surface is set equal to the wall-normal velocity of the plate.

Based on the measured velocity field of the plate surface at discrete frequencies, the propagation of sound was calculated in the frequency domain in order to compute the time-harmonic sound pressure at the microphone position [4]. These simulations allow for the separation of vibrational and flow-induced sound by comparing the results with measured sound pressure spectra.

5. STATISTICAL ENERGY ANALYSIS

In contrast to the classical computer based analysis of vibrational problems, which are focused on quantities like force or displacement, the modelling approach of Statistical Energy Analysis only considers energetic quantities. The main advantage of this method is the small amount of degrees of freedom which significantly reduces the necessary computation time and therefore allows for supplementing time consuming FE calculations at higher frequency ranges. Furthermore, SEA does not work with deterministic quantities, but with statistical ones. All calculations are done with frequency band limited and spatially averaged variables.

According to the basic theory behind this technique, the system of interest is decomposed into sub domains which show similar mode shapes. Using the fact that vibrational energy transfers over system boundaries from higher to lower levels (decisive is the energy per mode, not the overall level) and provided that the boundary conditions are well known, it is possible to infer from subsystems with known energy contents to unknown energies in physically or energetic adjoining subsystems. The energy content of a subsystem is directly connected with its level of vibration and is therefore a measure for the radiated sound [5].

In the following, it is shortly depicted, how the SEA module of the commercial software VA One (formerly AutoSEA2, ESI Group) can be used to predict the sound pressure level in the centre of a plenum cavity behind an overflown glass structure. The simulation model is based on investigations of wind noise in cars by Richard DeJong [6, 7] and therefore accordingly set up. He proofed in his work that in SEA simulations, broad band components of flow noise can be accounted for by modelling two additional, fictional acoustic cavities, which represent the turbulent flow on the side window and the acoustic near field. Using this approach, these cavities are excited via the vibrating glass plate and partially reflect this energy back into the cavity behind the excited glass structure. Consequently, the developed model for the overflown glass plate consists of four subsystems: A glass plate, an acoustic plenum cavity at the back side of the glass plate and two acoustic cavities on the other side of the flexible structure. As suggested there are four junctions in between them:

- the near field cavity, the glass plate and the plenum cavity
- the turbulent flow, the glass plate and the plenum cavity
- the near field cavity and the turbulent flow
- the glass plate and the plenum cavity

They all are energetically connected. For the excitation of the plate the turbulent flow over the structure is used. Therefore, unsteady pressure measurements are conducted at the different typical pressure areas on the overflown side of the glass plate. Afterwards, a weighted average pressure spectrum is calculated and used as excitation for the glass plate.

6. RESULTS

6.1 Fluid-Structure-Acoustic Interaction

In Figure 3 to 5, frequency spectra of the measured plate velocity, the unsteady wall-pressure and the sound pressure are provided for the rigid aluminium plate and the flexible glass plate, respectively. The square cylinder is acting as obstacle.

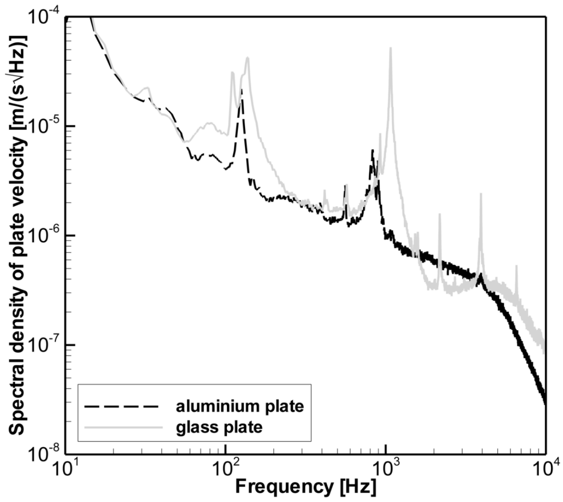

Figure 3. Frequency spectra of the measured vibration velocity in the centre point of the plates

Measurements of the surface velocity of the glass plate were performed with the single-point velocimeter. The frequency spectrum of the vibrational velocity in the centre point of the glass plate shows several peaks between 1 *kHz* and 6 *kHz*, which corresponds to the eigenfrequencies of the glass plate obtained by modal analysis. In comparison to this, there are no peaks in the spectrum of the rigid aluminium plate, which is in accordance with expectation. The sharp tonal components at approximately 100 *Hz* and 900 *Hz* are not relevant since they are present in both spectra and induced due to vibrations by the wind tunnel. But the spectrum of the glass plate differs from the spectrum of the aluminium plate between 50 *Hz* and 400 *Hz* as well as between 6 *kHz* and 10 *kHz* up to half an order of magnitude corresponding to a broadband excitation by the turbulent wall-bounded flow (Fig. 3).

Figure 4 shows the spectra of the unsteady wall-pressure measurements 92 *mm* downstream to the leading edge of the plates at the centreline of the spanwise direction. One can observe higher amplitudes between 50 *Hz* and 400 *Hz* and 6 *kHz* and 10 *kHz*, respectively, for the case with the glass plate, which is in good agreement with the results of the vibration velocity measurements. Contrary to Figure 3, there are no prominent peaks as a result of the natural vibrations of the glass plate.

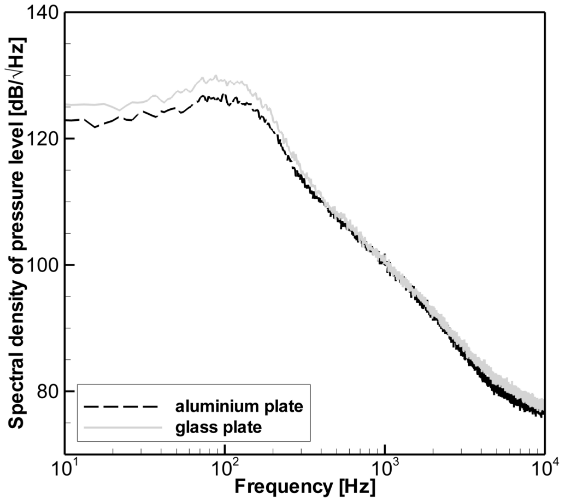

Figure 4. Frequency spectra of the wall-pressure fluctuations at $x = 92$ *mm* **and** $y = 0$ *mm*

For glass and aluminium as plate material, the spectra of the sound pressure level at a 0.07 *m* distance perpendicular to the back side of the plates are very similar. The main differences are the tonal components for the glass plate and the higher level between 6 *kHz* and 10 *kHz* in this case, respectively (Fig. 5).

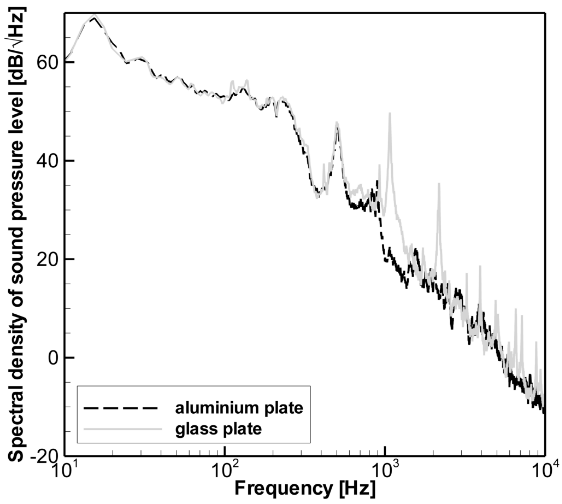

Figure 5. Sound pressure level at a 0.07 *m* **distance perpendicular to the back side of the plates**

To sum up, one can say that the flow over a flexible plate is influenced by the flow-induced broadband vibrations of a flexible plate structure but there is no effect of the natural vibrations. Contrary, the modified wall-bounded turbulent flow causes no difference in the sound radiation up to 1 *kHz* and only the eigenfrequencies distinguish the examined spectra.

6.2 Separation of Vibrational and Flow-Induced Sound

To separate vibrational sound from flow-induced sound in the near field and the far field, detailed investigations of the structural movement and the radiated sound were conducted. The following results refer to the glass plate with the square cylinder used as obstacle, as described in section 2.

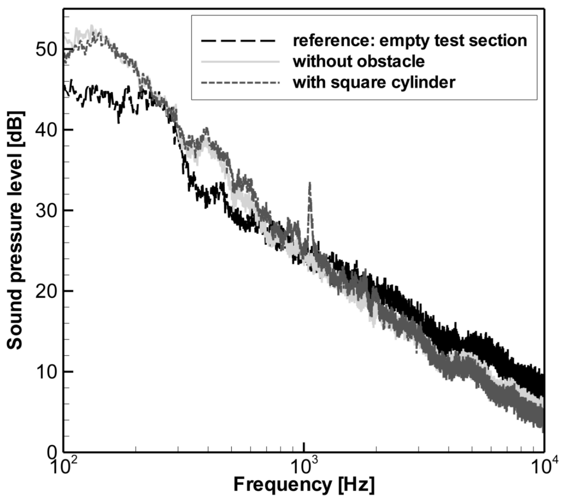

Figure 6. Sound pressure level at a 1 *m* distance perpendicular to the back side of the glass plate

Figure 6 shows frequency spectra of the measured sound pressure level in 1 *m* distance to the back side of the glass plate. Spectra obtained for the empty test section, for the glass plate without obstacle and for the glass plate with obstacle are provided. Both spectra for the glass plate are very similar with little higher amplitudes in the case with square cylinder under 2 *kHz* and little higher amplitudes in the case without obstacle over 3 *kHz*, respectively. Due to the stronger excitation of the glass plate in the case with obstacle, the corresponding spectrum shows a prominent peak at 1 *kHz* (first eigenfrequency). Both spectra differ between 100 *Hz* and 400 *Hz* from the reference measurement with an empty test section. This results from the turbulent fluctuations in the flow field over both plates.

In Figure 7 and 8 the spectra of measured and simulated sound pressure level are depicted for a distance of 0.07 *m* and 1 *m* perpendicular to the back side of the glass plate, respectively. Results of the phase-resolved scanning vibrometer measurements acted as data base for the calculation of the sound propagation to the near and the far field with CFS++.

For the case with 0.07 *m* distance one can observe a very good agreement between measurement and simulation regarding the tonal components of the spectra between 1 *kHz* and 5 *kHz*. The good result of the computation of the sound pressure level based on the velocity field of the plate surface indicates that the sound generation at these discrete freqeuncies is dominated by the vibration of the flexible structure and not by flow-induced noise due to the turbulent fluctuations. Between the tonal peaks, the tendency is predicted correctly by the computational method. This indicates that the turbulent flow leads to a weak broadband excitation of the glass plate but the sound radiation is dominated by the turbulent fluctuations. The same arguments are valid for the frequency domain under 1 *kHz*.

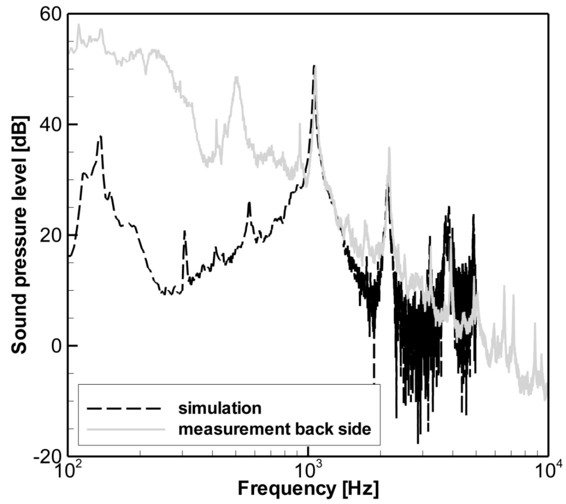

Figure 7. Mesured and simulated sound pressure level at a 0.07 *m* distance perpendicular to the back side of the glass plate

In the far field at 1 *m* distance a strengthening of the described phenomena can be observed. Only the peak at 1 *kHz* is correctly predicted by the Finite Element Scheme. Hence, at the front side as well as at the back side of the glass plate, the detected noise is induced by the turbulent flow, with the exception of the tonal component at the back side.

Additionally, measurements of the correlation between the acoustic pressure at the microphone position and the velocity in the centre point of the vibrating plate were conducted. The corresponding coherence spectra are provided in Figure 9 for microphone distances of 0.07 *m* at the back side of the glass plate and 1 *m* at the front side of the glass plate. The spectra confirm the results in Fig. 7 and 8. In the near field a strong correlation is found at the frequencies of the tonal vibrational noise. In between the tonal components, at lower frequencies and in the far field there is a small coherence between the sound pressure signal and the plate velocity, but this is due to the flow-induced vibration of the plate. At these frequencies the flow-induced noise is generated by the turbulent fluctuations.

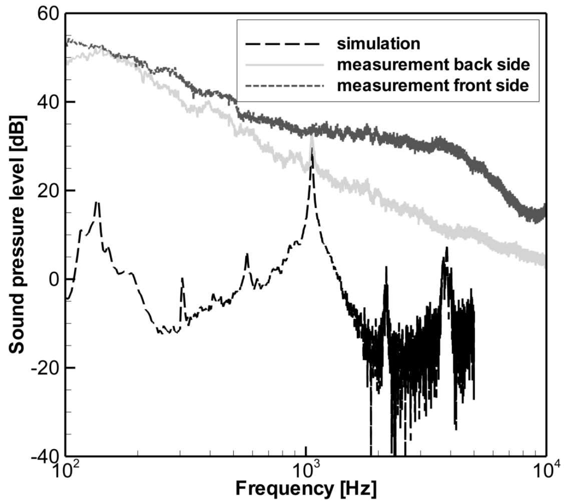

Figure 8. Mesured and simulated sound pressure level at a 1 *m* **distance perpendicular to the back side of the glass plate**

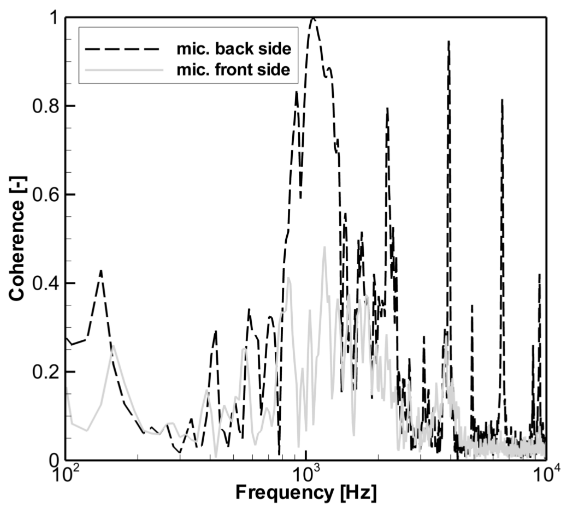

Figure 9. Spectra of the coherence between sound pressure signal (back side: 0.07 *m* **distance, front side:** 1 *m* **distance) and plate velocity signal (centre point)**

6.3 High-Frequency Broadband Noise

Figure 10 shows a comparison of the measured and the simulated sound pressure levels in the plenum chamber mounted at the back side of the glass plate. The sound radiation into the plenum cavity was calculated with the commercial software VA One based on unsteady pressure measurements.

Due to the fact that at least three eigenmodes of the structure have to lie in the chosen frequency bandwidth, results can only be presented starting from 5 *kHz*. In addition to that fact, one can understand that no sharp tonal peaks can be seen from the results of the simulation. The results of the standard Statistical Energy Analysis are in good qualitative agreement with the results of the measurement. Considering not only the application of energy to the glass plate by the turbulent flow, but also the flow-induced sound by the approach of DeJong, an almost perfectly match of results of measurement and simulation can be achieved. This is in good agreement with the results presented above and validates the theory that high-frequency broadband noise is flow-induced and caused by turbulent fluctuations.

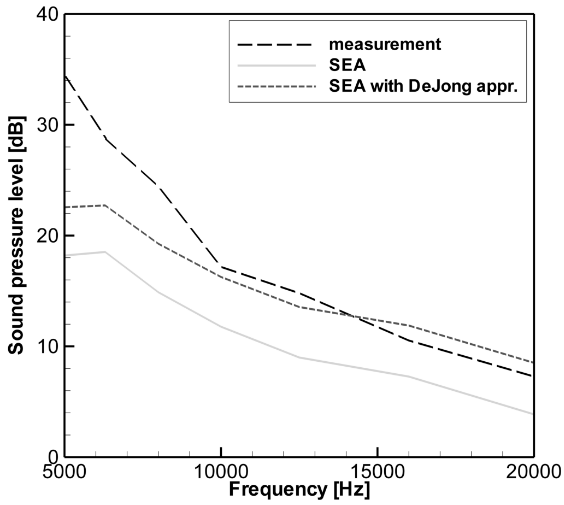

Figure 10. Measured and simulated sound pressure level in the plenum chamber at the back side of the glass plate

7. CONCLUSION

Experimental and numerical investigations of the fluid-structure interaction as well as the sound, induced by the turbulent flow over a flexible plate structure and radiated to the acoustic far field, were presented. Inflow conditions differing from the turbulent boundary layer were generated by a square cylinder. To investigate the influence of the flow-induced vibration of a plate structure on the wall-bounded flow, measurements of the vibrational velocity and the unsteady wall-pressure were performed at an aluminium plate and a glass plate. Based on measurements of the movement of a glass plate by laser scanning vibrometry, the sound radiation to the acoustic far field was computed employing the finite-element based multiphysics solver CFS++. The sound radiation in a plenum cavity at the back side of the glass plate was calculated with the commercial SEA software VA

One based on unsteady pressure measurements. Microphone measurements were performed to generate a basis of comparison.

The flow over a flexible plate is influenced by the flow-induced broadband vibrations of the flexible glass plate but there is no effect on the radiated sound. Both in the acoustic near field and in the far field the tonal noise is generated by structural vibrations and the broadband noise is flow-induced due to turbulent fluctuations in the flow field. For a conveyance this means that the turbulence characteristics have to be modified in order to reduce the excitation of eigenmodes. The results provide information which are helpful to better understand the generation of noise, which is the basic requirement for noise reduction in many technical applications similar to the present model setup.

ACKNOWLEDGEMENTS

Financial support by the Bavarian Research Foundation (BFS) and ESI Group for their support and the supply of the software VA One are gratefully acknowledged.

REFERENCES

[1] Vergne, S., Auger, J.-M., Périé, F., Jacques, A., and Nicolopoulos, D., 2007, "Aeroelastic Noise", *Large-Eddy Simulation for Acousitcs*, edited by Wagner, C., Hüttl, T., Sagaut, P., Cambridge University Press, Cambridge, pp. 272-293.

[2] Becker, S., Hahn, C., Kaltenbacher, M., and Lerch, R., 2008, "Flow-Induced Sound of Wall-Mounted Cylinders with Different Geometries", *AIAA J*, Vol. 46, No. 9, pp. 2265-2281.

[3] Kaltenbacher, M., Hauck, A., Triebenbacher, S., Link, G., and Bahr, L., 2007, "CFS++: Coupled Field Simulation", *Technical Report, Department of Sensor Technology, University of Erlangen-Nuremberg*.

[4] Grabinger, J., Müller, S., and Becker, S., 2009, "Separation of Aero- and Vibroacoustic Noise from a Flexible Structure", Proc. *International Conference on Acoustics, 23 - 26 March 2009*, Rotterdam, Netherlands.

[5] DeJong, R. G., Bharj, T. S., Booz, G. G., 2003, "Validation of SEA Wind Noise Model for a Design Change", *Technical Report* SAE Paper No. 2003-01-1552.

[6] DeJong, R. G., Bharj, T. S., Lee, J. J., 2001, "Vehicle Wind Noise Analysis Using a SEA Model with Measured Source Levels", *Technical Report* SAE Paper No. 2001-01-1629.

[7] Lyon, R. H., and DeJong, R. G., 1995, *Theory and Application of Statistical Energy Analysis*, Butterworth-Heinemann.