inter.noise 2014
MELBOURNE AUSTRALIA
16-19 NOVEMBER

Using a psychoacoustic criterion for the actuator placement in an active structural acoustic control system

Veatriki PAPANTONI[1]; Christian HESSE[2]; Michael ROSE[3]; Hans Peter MONNER[4]

[1,2,3,4] Institute of Composite Structures and Adaptive Systems, German Aerospace Center (DLR), Braunschweig, Germany

ABSTRACT

Active structural acoustic control (ASAC) systems for flexible structures with enclosed cavities are normally designed with the goal of minimizing the acoustic potential energy in the cavity. This goal is also taken into account during the search for an optimal actuator placement for the ASAC system. This paper is concerned with the change in the acoustic response of a structurally excited cavity when a psychoacoustic criterion is used for the placement of the actuators of the ASAC system instead. The placement of actuators on a flexible structure enclosed by a cavity is optimized using a genetic algorithm. Two optimization criteria are used: the acoustic potential energy in the cavity and the psychoacoustic loudness in the cavity. Numerical results regarding the acoustic potential energy reduction and the loudness reduction in the cavity of the coupled system in the case of an optimal feedforward control are presented for the two different actuator placement criteria.

Keywords: ASAC, actuator placement, loudness I-INCE Classification of Subjects Number(s): 38.3, 38.5.1

1. INTRODUCTION

The design of an ASAC system with the goal of attenuating an acoustic measure (e.g. sound pressure level, acoustic potential energy) normally involves finding an optimal actuator placement as part of the problem solution. The actuator placement can take place following various criteria, e.g. maximization of the modal forces applied by the actuators or maximization of the controllability degree, as listed in (1). In connection with applications such as aircraft interior noise, the criterion used for the determination of an optimal actuator placement is usually the minimization of the acoustic potential energy transmitted into the aircraft interior.

In comparison to systems of structures radiating into a free field, where the formulation of a control strategy using structural sensors can be realized with the help of radiation modes (2), systems with enclosed cavities are more complex. In this case the interaction between structure and fluid has to be taken into account. Various theoretical studies have dealt with this problem mainly using a modal interaction model of the coupled structure-fluid system (3, 4). However, very few experimental studies based on this theory can be found in literature, e.g. in (5), where an optimal placement of structural sensors is found in order to actively control the noise inside a cylindrical shell with a floor partition.

In lots of the application fields of active noise reduction systems, especially in passenger transportation, the aspect of how the noise is subjectively perceived by the passengers is gaining importance. Most of the approaches used are restricted to an evaluation of the system with regard to a psychoacoustic metric before and after the use of the active noise reduction system. There are however still very few attempts to directly link a psychoacoustic metric to the active control system, e.g. (6). The present study investigates whether the inclusion of a psychoacoustic criterion, i.e. loudness, in the optimization process of the actuator placement in an ASAC-system with a rectangular cavity, can effectively influence this psychoacoustic metric.

This paper is structured as follows. First, the theory regarding the modal interaction model for coupled structure-fluid systems is presented and the acoustic potential energy in the enclosure of such a system is expressed as a function of the velocities on the structure. Next, the principle of the ASAC pre-design tool,

[1] veatriki.papantoni@dlr.de
[2] christian.hesse@dlr.de
[3] michael.rose@dlr.de
[4] hans.monner@dlr.de

which is used in this study for the placement of actuators on a structure to be actively controlled, is described. Finally, the system used for the simulations as well as the simulation results are presented and discussed.

2. THEORY

2.1 Modal interaction model of a coupled structure-fluid system

When the system to be actively controlled is a coupled enclosure, the global error criterion used is the time-averaged frequency-dependent acoustic potential energy (APE) in the enclosure, which can be expressed as

$$E_p(\omega) = \frac{1}{4\rho c^2} \int_V |p(\mathbf{r},\omega)|^2 \,\mathrm{d}V\,, \tag{1}$$

where ρ is the density and c the speed of sound within the acoustic fluid enclosed by the cavity, $p(\mathbf{r},\omega)$ the acoustic pressure at some point $\mathbf{r} = (x,y,z)$ in the enclosed fluid, and the integral is evaluated over the cavity volume V (7). The dependency of the pressure on the frequency ω is omitted below for the purpose of brevity.

According to the modal interaction model (8), the acoustic pressure in the fluid volume within rigid boundaries can be expressed as a sum of the pressure distributions in the acoustic modes ϕ_i :

$$p(\mathbf{r}) = \sum_{i=1}^{\infty} p_i \phi_i(\mathbf{r})\,, \tag{2}$$

with p_i the pressure amplitude of the ith acoustic mode of the cavity.

By substituting Eq. (2) into Eq. (1) and due to the orthogonality of the acoustic modes in the cavity, the APE taking into consideration N_a acoustic modes can be written as

$$E_p = \frac{1}{4\rho c^2} \sum_{i=1}^{N_a} \Gamma_i |p_i|^2 = \mathbf{p}^H \mathbf{L} \mathbf{p}\,, \tag{3}$$

with $\mathbf{p}$ the $(N_a \times 1)$-dimensional vector of the acoustic modal amplitudes and $\Gamma_i = \int_V \phi_i^2(\mathbf{r})\mathrm{d}V$ the volume normalization factor of the ith acoustic mode. $\mathbf{L}$ is a diagonal weighting matrix with the diagonal terms

$$\mathbf{L}(i,i) = \frac{\Gamma_i}{4\rho c^2}\,. \tag{4}$$

In order to express the APE in terms of the structural vibration a modal structural-acoustic transfer function matrix $\mathbf{Z}_a$ is introduced, such that

$$\mathbf{p} = \mathbf{Z}_a \mathbf{v}\,, \tag{5}$$

where $\mathbf{Z}_a$ has the dimensions $(N_a \times N_s)$ and $\mathbf{v}$ is the $(N_s \times 1)$ vector of the N_s structural velocity modes taken into consideration.

The transfer function can be obtained from the Kirchhoff-Helmholtz integral equation for the case of a rigid-walled enclosure without any sources in the field

$$p(\mathbf{r}) = j\rho\omega \int_S G(\mathbf{r}_s|\mathbf{r}) v(\mathbf{r}_s)\,\mathrm{d}S\,, \tag{6}$$

with $j = \sqrt{-1}$, $\mathbf{r}_s = (x_s, y_s, z_s)$ a point on the vibrating surface of interaction and $G(\mathbf{r}_s|\mathbf{r})$ the Green's function for the rigid-walled cavity modes (8):

$$G(\mathbf{r}_s|\mathbf{r}) = \sum_{i=1}^{N_a} \frac{\phi_i(\mathbf{r}_s)\phi_i(\mathbf{r})}{\Gamma_i(\kappa_i^2 - k^2)}\,. \tag{7}$$

Here κ_i is the wave number of the ith acoustic mode with the resonant frequency ω_i and k is the acoustic wave number at the frequency of excitation ω.

By using the modal formulation for the structural velocities in Eq. (6), substituting Eq. (6) in Eq. (5), and solving for $\mathbf{Z}_a$, we obtain:

$$\mathbf{Z}_{a_{ij}} = \frac{j\rho\omega S}{\Gamma_i(\kappa_i^2 - k^2)} B_{ij}\,. \tag{8}$$

The modal coupling matrix B_{ij} is calculated as the integral of the acoustic modes ϕ_i and the structural modes ψ_j over the surface of interaction S of the vibrating structure:

$$B_{ij} = \int_S \psi_j(\mathbf{r}_s)\phi_i(\mathbf{r}_s)\,\mathrm{d}S\,. \tag{9}$$

Now that the transfer function $\mathbf{Z}_a$ can be calculated, the acoustic pressure from Eq. (5) can be substituted in Eq. (3)

$$E_p = \mathbf{v}^H \mathbf{Z}_a^H \mathbf{L} \mathbf{Z}_a \mathbf{v} = \mathbf{v}^H \Pi \mathbf{v}\,, \tag{10}$$

so that the APE can be expressed as a function of the structural velocity amplitudes $\mathbf{v}$ and the so-called error weighting matrix (EWM) Π (3, 4, 9). This formulation is used for the calculation of the APE throughout this study.

2.2 The ASAC pre-design tool

In order to design a suitable ASAC-system for a radiating structure, a so-called ASAC pre-design tool (10), which was developed in the DLR, is used. This tool was initially designed for plain structures without enclosures and was extended in the course of this study in order to describe structures with enclosures. With the help of this tool, a suitable actuator placement can be derived and the system's broadband performance in noise reduction is estimated.

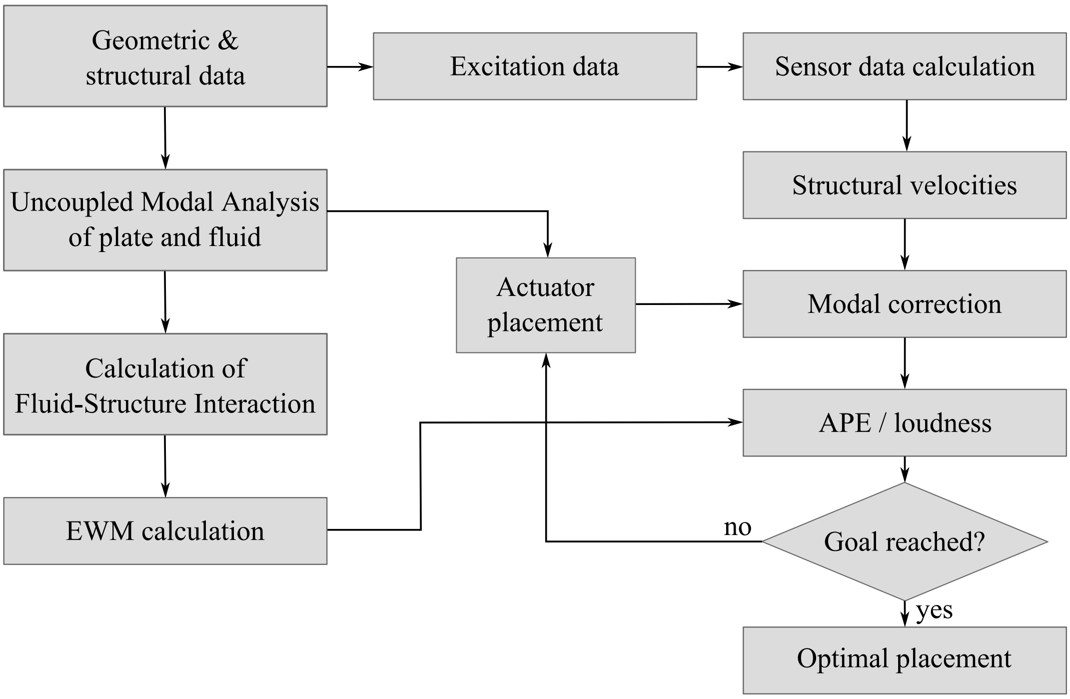


Figure 1 – Schematic representation of the steps followed within the ASAC pre-design tool.

Figure 1 shows the main stages followed in this process. First, the uncoupled natural mode shapes of the structure (plate) and of the fluid (air) in the cavity are computed by the finite element (FE) software ANSYS®. Next, the force excitation resulting from a typical acoustic excitation is calculated for a virtual sensor net on the plate surface. The modal coupling matrices are calculated as described in section 2.1 and the error weighting matrix (EWM) is used to determine the APE in the cavity. An initial actuator placement for a given number of piezoceramic patches is calculated with respect to the areas of maximum strain on the vibrating plate. Finally, a genetic algorithm varies the positions of the piezoceramic patches with respect to a specific goal.

In order to avoid building a different finite element model for every placement of the actuators and subsequently evaluating this model to get the necessary frequency response data, a more efficient approach is used. The model of the host structure (plate) is stored and corrected for each actuator placement with effect of the actuators by using the modal correction method (11).

The APE reduction is estimated for a frequency-domain optimal feedforward controller. This estimated reduction is an optimistic prediction of the system's performance, as the optimal controller does not take into account the causality constraints of the real-time implementation of such a system on a DSP. This estimate can however be used as an indicator of the maximally possible attenuation in the case of a causal system.

The genetic optimization of the actuator placement in this study uses two goals: the APE in the cavity, and the psychoacoustic loudness according to ISO 532B (12). The loudness is calculated for two different cases: (a) for one point in the middle of the cavity, and (b) the mean loudness over a number of points homogeneously distributed inside the cavity.

3. SIMULATION

3.1 Simulation parameters

Figure 2a shows the panel-cavity system which is modelled in this analysis. The cavity is 0.9 m long (L_x), 0.6 m high (L_y), and 0.9 m long (L_z), and one of its side surfaces consists of a 5-mm-thick, baffled aluminium panel. The constants used in the simulation are $c_0 = 343\,\mathrm{m/s}$ for the sound speed in the air filling the cavity, $\rho_0 = 1.204\,\mathrm{kg/m^3}$ for the density of the air, and $\rho_p = 2700\,\mathrm{kg/m^3}$ for the panel density. The cavity walls are modelled as acoustically rigid surfaces.

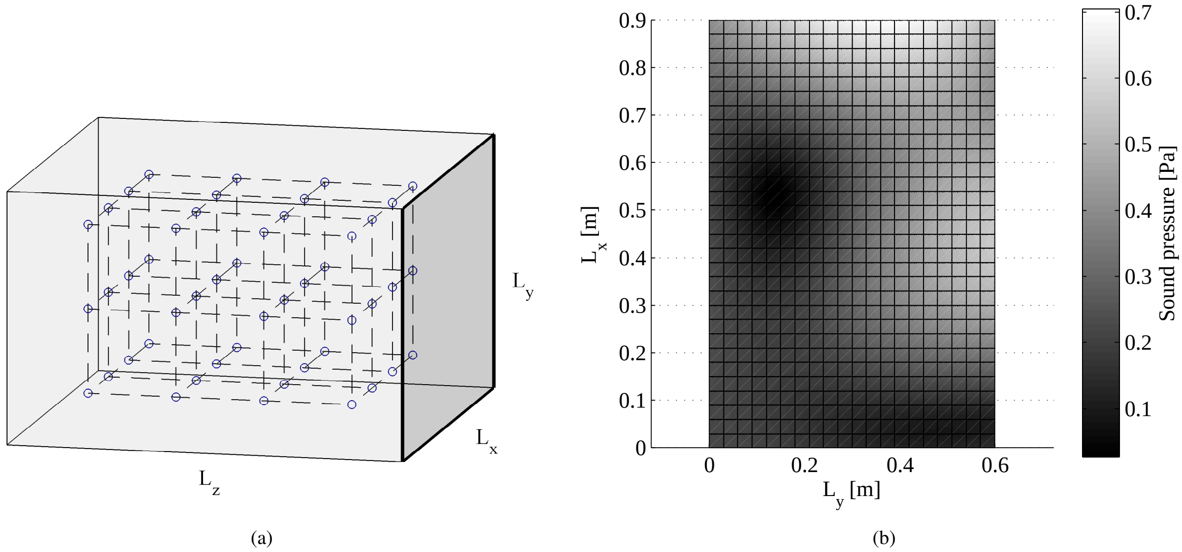


Figure 2 – (a) Schematic representation of the baffled aluminium plate (on t he right side) and the enclosure. The positions of the virtual microphones for the evaluation of the loudness inside the enclosure are also shown. (b) Sound pressure distribution on the aluminium plate due to the diffuse sound field excitation at 250 Hz.

The plate and fluid are modelled in the finite element software ANSYS® so that the meshes of plate and fluid have coincident nodes on the wet surface. 600 4-noded shell elements of type *shell63* are used for the structural mesh, while 18000 8-noded fluid elements of type *fluid30* are used for the acoustic fluid mesh. Both element types are modelled with 3 cm edge lengths.

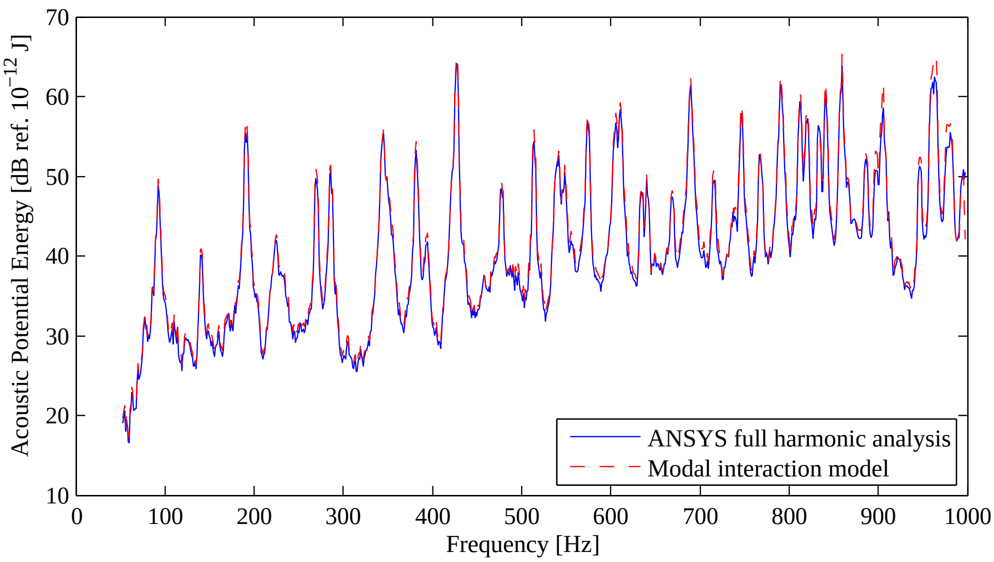


Figure 3 – Acoustic potential energy in the cavity in the case of a diffuse sound field excitation, as calculated with a coupled harmonic analysis in ANSYS® and using the modal interaction model.

The aluminium panel is excited by an external diffuse sound field. Diffuse sound fields are commonly used in sound transmission tests, where the excitation created in a reverberating room excites a test structure which is built into a separating wall. The transmitted sound is measured in a free field room on the opposite side of the wall. Such a diffuse excitation can be simulated by modelling point sources evenly distributed on a large hemisphere with random phase shifts, uncorrelated at different frequencies (10). The force applied by a diffuse sound field on a structure results in a pressure distribution on the structure's surface as the one depicted in figure 2b.

Due to the acoustic excitation of the aluminium panel, sound power is radiated into the cavity. In order to affect the sound transmission into the cavity, the concept of active structural acoustic control is chosen. In the optimal acausal scenario described here, the structural response of the plate can be obtained by combining the pressures produced by the sound field on the plate surface with the modal behaviour of the plate. This method is called modal superposition (11). The structural sensor net used in the ASAC system to calculate the excitation coincides with the nodes of the finite elements used to model the plate, as depicted in figure 2b.

The actuators used in this study are piezoelectric patches with a piezoceramic area of 50 mm $\times$30 mm. Five such piezoceramic actuators were used in this study. The integration of the piezoceramic patches into the aluminium plate is modelled as described in (11).

Apart from the uncoupled modal analysis of the plate and the fluid, a coupled harmonic analysis was also conducted in ANSYS® in order to examine the validity of the results provided by the modal interaction model, which is used within the ASAC pre-design tool. As can be seen in figure 3 there is a good compliance for the APE as calculated with the two methods for the case of a diffuse acoustic field excitation.

3.2 Simulation results

As described above, the ASAC pre-design tool was used in order to estimate the resulting APE in the cavity of the system described in section 3.1. The optimal placement of the five piezoceramic actuators was found using a genetic algorithm. Three different goals were used for the placement optimization: (a) minimal APE inside the cavity, (b) minimal loudness in the centre of the cavity, and (c) minimal mean loudness calculated over the loudness values in the positions shown in figure 2a.

The APE in the cavity for a diffuse acoustic field excitation is shown in figure 4 for the three optimization cases with ASAC as well as the system without ASAC. In all three cases a broadband attenuation, especially for lower frequencies, is achieved. Due to the use of only five actuators, some of the resonant peaks in the APE present at frequencies higher than 500 Hz cannot be effectively suppressed. In comparison to the optimization case of minimal APE, the optimization case of minimal mean loudness shows higher APE levels for low frequencies. This is due to the fact that low frequency components have a very small contribution in the calculation of psychoacoustic loudness. The same effect cannot be noticed in the case of minimal loudness in the cavity centre, as this measure for one single point in the cavity is less correlated to the more overall measure of APE than the mean loudness over more points in the cavity.

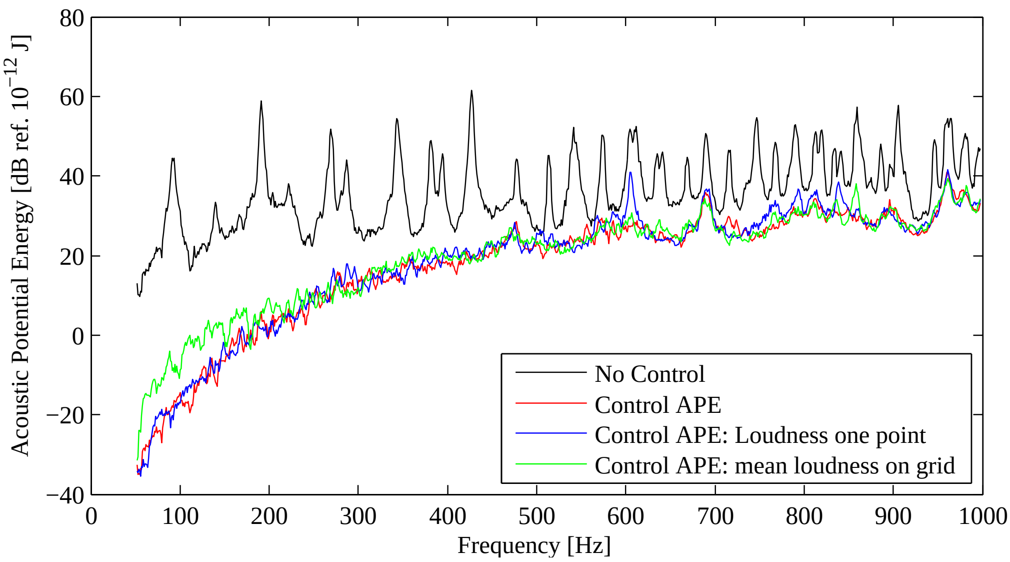
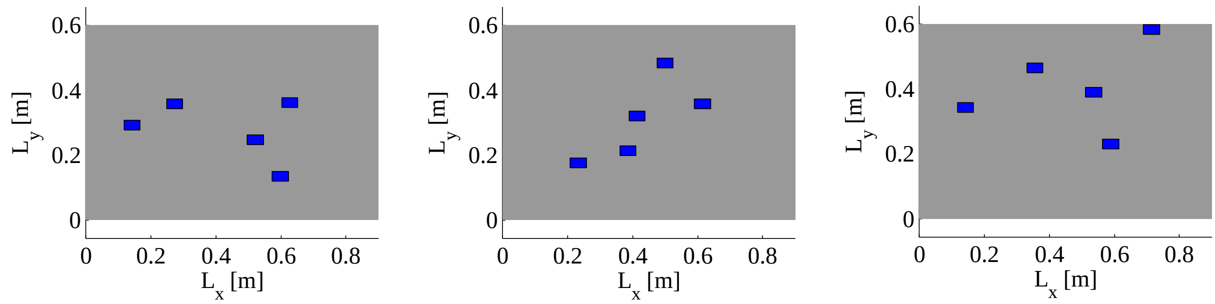


Figure 4 – Acoustic potential energy in the cavity in the case of a diffuse sound field excitation, before and after ASAC and for the three actuator placement optimization scenarios.

As can be seen in figure 5 the three optimization scenarios result in different actuator placements. Different results can also be observed regarding the APE and loudness values (see table 1). When the goal of the optimal actuator placement is minimal loudness in the cavity centre (case (b)), the resulting APE is slightly increased compared to the optimization goal of minimal APE (case (a)). The same is the case for the mean loudness on the grid points, whereas the loudness in the cavity centre is significantly decreased, as expected. In the case that minimal mean loudness on the grid positions is desired (case (c)), the APE is almost equal to that in case (a), whereas the loudness in the cavity centre as well as the mean loudness are decreased. Compared to case (b) the resulting loudness in the cavity centre in case (c) is higher. The mean loudness is however reduced, as was expected due to the placement criterion.



(a) (b) (c)

Figure 5 – Optimal placement of the five piezoceramic actuators (blue rectangles) on the aluminium plate using the ASAC pre-design tool for the three optimization scenarios: (a) minimal APE inside the cavity, (b) minimal loudness in the centre of the cavity, (c) minimal mean loudness on grid points.

Table 1 – Results regarding the APE and loudness for the various optimization cases.

Optimization goal	APE [dB ref. 10^{-12} J]	Loudn. in centre [sone]	Mean loudn. on grid [sone]
(a) Minimal APE	57.70	56.92	59.14
(b) Min. loudn. in centre	59.07	47.36	61.96
(c) Min. loudn. on grid	57.72	50.28	57.48

For a better illustration of the results concerning the resulting loudness in the cavity for the three optimization cases, a distribution of the loudness values inside the cavity is shown in figure 6. As can be seen, the loudness in cases (a) and (c) is attenuated more homogeneously over the cavity volume compared to case (b), where there is a local minimum of the loudness value in the cavity centre. Case (c) leads to the best loudness results when regarded over the entire cavity volume.

4. CONCLUSIONS

This paper investigates the effect of a psychoacoustic criterion used for the optimization of the actuator placement in the case of a baffled plate enclosed by a rectangular cavity with an optimal ASAC system. The ASAC system minimizes the APE in the cavity as described in the modal interaction model of coupled structure-fluid systems.

The conducted simulations show that a different placement of actuators is reached when the objective of the genetic optimization is changed from the APE to the psychoacoustic loudness. Different results are also reached when the targeted loudness is that of one point in the cavity and when it is averaged over several points inside the cavity.

It should be noted that the ASAC pre-design tool used for the simulations may overestimate the possible performance of the controller as the causality constraints for the real-time implementation are not taken into account. Thus the results presented here should be compared to an experimental implementation.

ACKNOWLEDGEMENTS

This study was conducted within the project *Transferzentrum - MRO und Cabin Upgrade* (grant no. LAHH 144), which is funded by *Freie und Hansestadt Hamburg*.

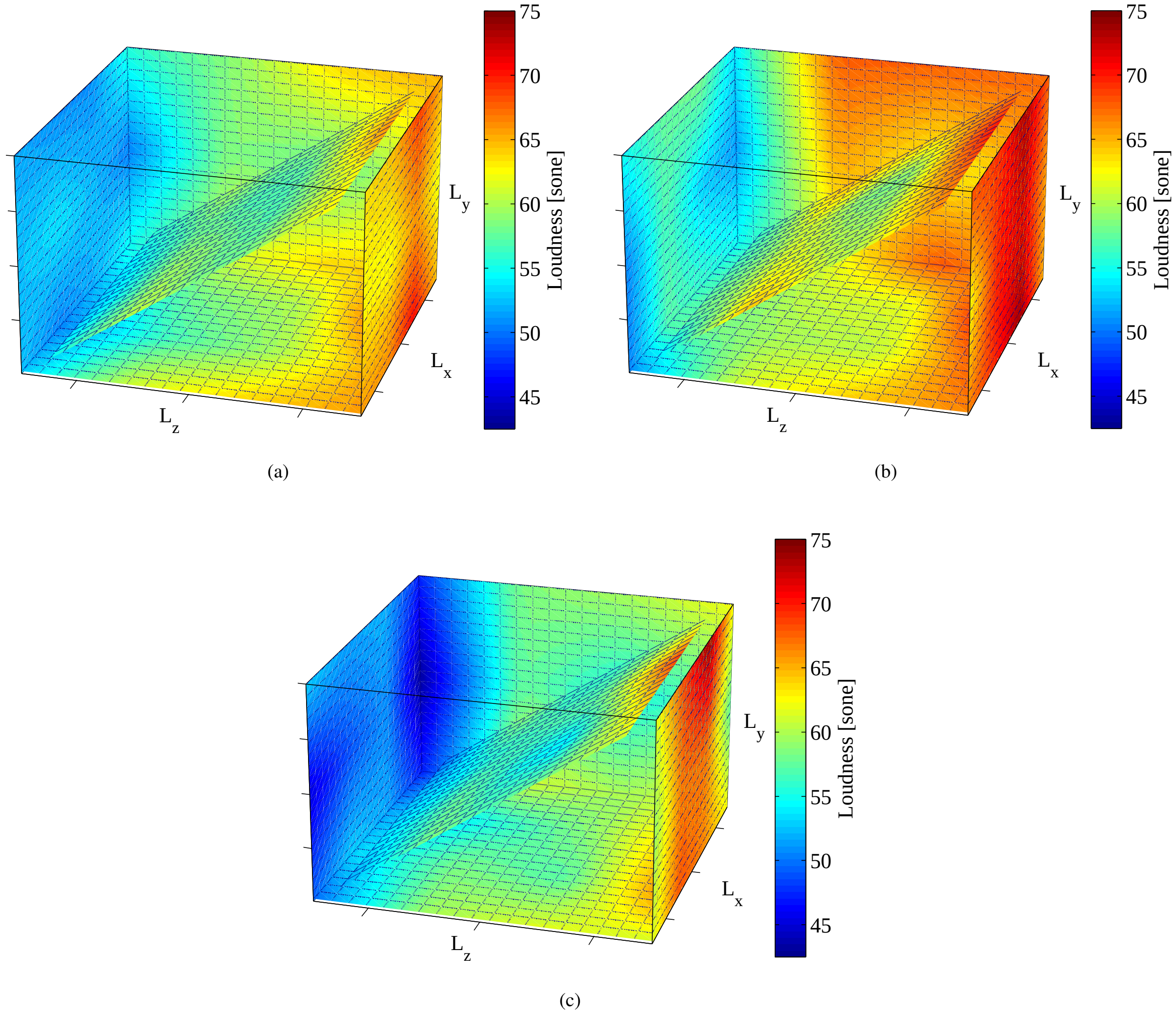


Figure 6 – Distribution of loudness inside the cavity for the three optimization scenarios: (a) minimal APE inside the cavity, (b) minimal loudness in the centre of the cavity, (c) minimal mean loudness on grid points.

REFERENCES

1. Gupta V, Sharma M, Thakur N. Optimization Criteria for Optimal Placement of Piezoelectric Sensors and Actuators on a Smart Structure: A Technical Review. Journal of Intelligent Material Systems and Structures. 2010;21(12):1227–1243.

2. Elliott SJ, Johnson ME. Radiation modes and the active control of sound power. The Journal of the Acoustical Society of America. 1993;94(4):2194–2204.

3. Snyder SD, Tanaka N. On feedforward active control of sound and vibration using vibration error signals. J Acoust Soc Am. 1993;94(4):2181–2193.

4. Cazzolato BS, Hansen CH. Active control of sound transmission using structural error sensing. J Acoust Soc Am. 1998;104(5):2878–2889.

5. Li DS, Cheng L. The design of synthesized structural acoustic sensors for active control of interior noise with experimental validation. Journal of Sound and Vibration. 2010;329(2):123 – 139.

6. de Oliveira LPR, Stallaert B, Janssens K, der Auweraer HV, Sas P, Desmet W. NEX-LMS: A novel adaptive control scheme for harmonic sound quality control. Mechanical Systems and Signal Processing. 2010;24(6):1727 – 1738.

7. Nelson PA, Elliott SJ. Active control of sound. Academic Press; 1993.

8. Fahy F, Gardonio P. Sound and Structural Vibration: Radiation, Transmission and Response. Academic Press; 2007.

9. Johnson WM, Cunefare KA. Use of principle velocity patterns in the analysis of structural acoustic optimization. J Acoust Soc Am. 2007;121(2):938–948.

10. Rose M, Unruh O, Haase T. Vibration Control of Stiffened Plates with Embedded Cavities Using Flat Piezoceramic Actuator Devices. Proc 18th International Congress on Sound and Vibration (ICSV 18); 10-14 July 2011, Rio de Janeiro, Brazil 2011.

11. Rose M. Modal Based Correction Methods for the Placement of Piezoceramic Modules, ASME International Mechanical Engineering Congress and Exposition; Orlando, Florida, November 5-11 (2005).

12. DIN 45631 (1991-03) / ISO 532B: Procedure for calculating loudness level and loudness; 1991.