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Tamara NESTOROVIĆ*, Jean LEFÈVRE**, Ulrich GABBERT*

MODEL-BASED ACTIVE NOISE CONTROL OF A PIEZOELECTRIC STRUCTURE

SUMMARY

Methods of active noise and vibration control involve different techniques which combine electro-acoustics with integration of active materials and controller into a complex active structural system in order to enable radiation of an anti-phased field for cancellation or attenuation of an original noise in a specific domain. This paper presents a model-based controller design procedure for the active noise attenuation of a piezoelectric structure surrounded by the acoustic fluid. Model development is based on a finite element method approach, which takes into consideration the electro-mechanical-acoustic effects. A FEM-based state-space model obtained after appropriate transformations and modal reduction is used for the controller design. The aim of the control, suppression of the acoustic fluid pressure in a prescribed point or field, is achieved using an optimal LQ controller designed based on a developed state-space model. The controller design involves also a novel approach, with included additional dynamics for the noise control in the presence of periodic excitations. The effects of the suggested method are tested on a smart acoustic box consisting of an aluminium plate with attached piezoelectric patches, surrounded by the acoustic fluid (air) inside the wooden box. The air pressure reduction at a selected point inside the box is observed in the controlled case.

Keywords: *active noise control, piezoelectric structure, optimal LQ controller, acoustic box*

AKTYWNY UKŁAD REDUKCJI HAŁASU NA BAZIE MODELU STRUKTURY PIEZOELEKTRYCZNEJ

Aktywne metody redukcji drgań i hałasu wymagają zróżnicowanych technik, które łączą elementy elektroakustyczne zintegrowane z elementami czynnymi oraz sterownikami wewnątrz złożonej aktywnej struktury, żeby umożliwić promieniowanie pola w przeciwfazie w celu eliminacji lub redukcji pierwotnego hałasu w wybranej strefie. Artykuł prezentuje projekt regulatora na bazie modelu do aktywnej redukcji hałasu struktury piezoelektrycznej w otoczeniu płynu akustycznego. Opracowany model bazuje na metodzie elementów skończonych, która uwzględnia zjawiska elektro-mechaniczno-akustyczne. Model w przestrzeni stanu powstały na podstawie MES, otrzymany w wyniku stosownych przekształceń i uproszczeń modelu, użyto do syntezy regulatora. Celem sterowania było wytłumienie ciśnienia płynu akustycznego w danym punkcie lub polu, co osiągnięto, wykorzystując regulator optymalny LQ zaprojektowany na podstawie opracowanego modelu w przestrzeni stanu. Projektowanie regulatora wymaga nowatorskiego podejścia uwzględniającego dodatkową dynamikę podczas redukcji hałasu w obecności wymuszeń harmonicznych. Efekty zaprezentowanej metody zostały przetestowane na inteligentnej skrzynce akustycznej zbudowanej z aluminiowej płyty z naklejonymi na niej płytkami piezoelektrycznymi, otoczonej przez płyn akustyczny (powietrze), umieszczonej w drewnianym pudle. Zaobserwowano redukcję ciśnienia powietrza w wybranym punkcie wewnątrz skrzynki dla układu z regulatorem.

Słowa kluczowe: *aktywna redukcja hałasu, struktura piezoelektryczna, optymalne sterowanie LQ, skrzynka akustyczna*

1. INTRODUCTION

Methods of active structural and noise control involve different techniques, which combine electro-acoustics with integration of active materials and controller into a complex active structural system in order to enable radiation of an anti-phased field for cancellation or attenuation of an original noise in a specific domain. Techniques of the active noise control (ANC) enable noise reduction or cancellation in specific areas of the affected acoustic space using in most cases the secondary source of noise for generation of the anti-phased acoustic field. Besides the reduction of the noise level in specified desired zones, a partially increasing noise level in other areas can occur [1]. Methods of active vibration control (AVC) with the aim of the noise reduction are based on the possibility to reduce the noise level as a secondary effect of the structural vibration suppression or isolation. Vibration suppression is often applied in order to achieve this goal, e.g. [2, 3]. Another approach to the noise reduction and cancellation is referred to as the active structural acoustic control (ASAC). The reduction of the noise generated by the oscillating structure represents the focus of this concept. The form or the intensity of the structural oscillations are less relevant for the realisation of this goal, since the noise reduction plays a decisive role. Consequently it is possible to reduce the overall vibration of the structure (analogue to the AVC concept), or even to change or excite the vibrations in order to reduce the noise [4].This paper presents a model-based controller design procedure for the active noise attenuation of a piezoelectric structure

* Institute of Mechanics Otto-von-Guericke University, tamara.nestorovic@mb.uni-magdeburg.de; ulrich.gabbert@mb.uni-magdeburg.de
** FEMCOS - Ingenieurbüro mbH, jean.lefevre@femcos.de

surrounded by the acoustic fluid. Model development is based on a finite element method approach, which takes into consideration the electro-mechanical-acoustic effects. An FEM-based state-space model obtained after appropriate transformations and modal reduction is used for the controller design. The aim of the control, suppression of the acoustic fluid pressure in a prescribed point or field, is achieved using an optimal LQ controller designed based on a developed state-space model. The controller design involves also a novel approach, with included additional dynamics for the noise control in the presence of periodic excitations. The effects of the suggested method are tested on a smart acoustic box consisting of an aluminium plate with attached piezoelectric patches, surrounded by the acoustic fluid (air) inside the wooden box. The air pressure reduction at a selected point inside the box is observed in the controlled case.

2. FEM BASED MODEL DEVELOPMENT

For the modelling and simulation of coupled electro-mechanical-acoustical problems regarding interior noise, the authors have proposed an FEM-based approach [5 8] which can be implemented within the general purpose FEM software package COSAR [9] for the FEM model development. For further simulation studies, controller design and implementation, the FEM-based model can be exported to Matlab/Simulink through a specially developed interface. The FEM software COSAR contains an extensive library of multi-field finite elements for 1D, 2D and 3D continua as well as for shell-type thin walled structures and acoustic brick-type elements in order to simulate piezoelectrically controlled vibro-acoustic systems. Considering small displacements and regarding acoustic responses as small perturbations to an ambient reference state, the derivation of the finite element model is based on the mechanical equilibrium, the electric equilibrium, the linear coupled electromechanical constitutive equations and the linear acoustic wave equation [7]. Using the standard FEM procedure [5] the semi discrete system of coupled equations of the electromechanical field and the acoustic field is obtained in the form of the following matrix equation

$$\begin{bmatrix} \mathbf{M}_{ww} & \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & -\rho_0 \mathbf{M}_a \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{w}} \\ \ddot{\boldsymbol{\varphi}} \\ \ddot{\boldsymbol{\Phi}} \end{bmatrix} + \begin{bmatrix} \mathbf{C}_{ww} & \mathbf{0} & -\mathbf{C}_{wc} \\ \mathbf{0} & \mathbf{0} & \mathbf{0} \\ -\mathbf{C}_{wc}^T & \mathbf{0} & -\rho_0 \mathbf{C}_a \end{bmatrix} \begin{bmatrix} \dot{\mathbf{w}} \\ \dot{\boldsymbol{\varphi}} \\ \dot{\boldsymbol{\Phi}} \end{bmatrix} + \begin{bmatrix} \mathbf{K}_{ww} & \mathbf{K}_{w\varphi} & \mathbf{0} \\ \mathbf{K}_{w\varphi}^T & -\mathbf{K}_{\varphi\varphi} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} & -\rho_0 \mathbf{K}_a \end{bmatrix} \begin{bmatrix} \mathbf{w} \\ \boldsymbol{\varphi} \\ \boldsymbol{\Phi} \end{bmatrix} = \begin{bmatrix} \mathbf{f}_{ww} \\ \mathbf{f}_{\varphi} \\ -\rho_0 \mathbf{f}_a \end{bmatrix} \tag{1}$$

with mass matrix $\mathbf{M}_{ww}$, proportional damping matrix $\mathbf{C}_{ww}$, stiffness matrix $\mathbf{K}_{ww}$, electric matrix $\mathbf{K}_{\varphi\varphi}$, piezoelectric coupling matrix $\mathbf{K}_{w\varphi}$, mechanical load vector $\mathbf{f}_{ww}$ and electric load vector $\mathbf{f}_{\varphi}$. The quantities regarding the acoustic field are: constant fluid density ρ_0, acoustic mass matrix $\mathbf{M}_a$, acoustic damping matrix $\mathbf{C}_a$, acoustic stiffness matrix $\mathbf{K}_a$ and the acoustic load vector due to prescribed normal velocities $\mathbf{f}_a$. Vibro-acoustic coupling is performed in terms of the coupling matrix $\mathbf{C}_{wc}$ arising from an additional load, which acts on the fluid-structure interface and originates from the sound pressure for the structure and from the normal velocity of the structure for the acoustic fluid. Vector **w** contains all nodal mechanical degrees of freedom, φ is the vector of nodal electric potentials and $\mathbf{\Phi}$ the fluid velocity potential vector.

Modal truncation based on a reduced number of preselected uncoupled eigenmodes represents a common model reduction technique, which is used here for obtaining a reduced-order model convenient for the use in the Matlab/Simulink environment for the controller design, simulation studies and implementation. After a sequence of mathematical transformations (for more details see [5, 6]) the modal truncation results in a state equation (2) of the standard state-space model used in the control theory. The state-space model is completed with the output (measurement) equation (3).

$$\dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B}\mathbf{u}(t) + \mathbf{E}\mathbf{f}(t) \tag{2}$$

$$\mathbf{y}(t) = \mathbf{C}\mathbf{x}(t) + \mathbf{D}\mathbf{u}(t) + \mathbf{F}\mathbf{f}(t) \tag{3}$$

Vector **x** is the vector of modal coordinates (such that $\mathbf{z} = \mathbf{Q}\mathbf{x}$) obtained through the ortho-normalization, where **Q** represents the modal matrix obtained as a solution of the eigenvalue problem of the homogeneous part of the equation (1) with the introduced state-space vector

$$\mathbf{z} = \begin{bmatrix} \mathbf{w} & \boldsymbol{\varphi} & \boldsymbol{\Phi} & \dot{\mathbf{w}} & \dot{\boldsymbol{\varphi}} & \dot{\boldsymbol{\Phi}} \end{bmatrix} \tag{4}$$

In (2) and (3) the following nomenclature was used: **A** denotes the state matrix, **B** is the control input matrix, **E** is the disturbance coupling matrix, **C** output matrix, **D** input-to-output coupling matrix and **F** disturbance-to-output coupling matrix. Vector $\mathbf{f}(t)$ represents the vector of external disturbances, $\mathbf{u}(t)$ is the vector of the controller influence and $\mathbf{y}$(t) represents the output measurement vector.

3. OPTIMAL LQ CONTROLLER DESIGN

Starting from the state-space model (2), (3) developed using the FEM approach, which couples electro-mechanical-acoustic behaviour, the controller design can be performed in order to achieve the noise reduction with respect to the prescribed control objective, e.g. the reduction of the pressure level in the selected point of the acoustic fluid. As previously stated, the control of mechanical structures can affect mechanical vibrations, which is the main control objective in the case when the vibration suppression is a criterion for the controller performance. On the other hand, the main objective in the active structural acoustic control is the noise attenuation in the prescribed part or point of the surrounding acoustic field, regardless of the structural behaviour with respect to vibrations, which cause the acoustic effects. This control objective can be achieved using e.g. an optimal LQ controller.

The control technique suggested here is the optimal LQ controller with additional dynamics. The controller design includes available a priori knowledge about occurring disturbance type contained in the additional dynamics. Such an a priori knowledge is available in terms of the type of the disturbance function which has to be rejected or whose influence should be suppressed by the controller. Periodic disturbances with frequencies corresponding to the eigenfrequencies of the structure or the surrounding acoustic fluid can cause resonance states and their suppression is therefore important.

A discrete-time state-space equivalent (5), (6) of the continuous state-space model (2), (3) developed through the FEM procedure and modal reduction is used for the controller design.

$$\mathbf{x}[k+1] = \mathbf{\Phi x}[k] + \mathbf{\Gamma u}[k] + \mathbf{\varepsilon f}[k] \qquad (5)$$

$$\mathbf{y}[k] = \mathbf{Cx}[k] + \mathbf{Du}[k] + \mathbf{Ff}[k] \qquad (6)$$

Using the a priori knowledge about the disturbance type, which has to be suppressed, the model of the disturbance is represented in an appropriate state-space form, where the disturbance is assumed to be the output of the state-space representation. The poles λ_i of the disturbance transfer function are used to define the additional dynamics using the coefficients of the polynomial

$$\delta(z) = \prod_i (z - \mathrm{e}^{\lambda_i T})^{m_i} = z^s + \delta_1 z^{s-1} + \ldots + \delta_s \qquad (7)$$

where m_i represents the multiplicity of the pole λ_i. Additional dynamics is expressed in a state-space form

$$\mathbf{x}_a[k+1] = \mathbf{\Phi}_a \mathbf{x}_a[k] + \mathbf{\Gamma}_a \mathbf{e}[k] \qquad (8)$$

where $\mathbf{x}_a$ is the vector of the state variables for the additional dynamics, $\mathbf{e}$ is the error signal and the state-space matrices of the additional dynamics are:

$$\mathbf{\Phi}_a = \begin{bmatrix} -\delta_1 & 1 & 0 & \cdots & 0 \\ -\delta_2 & 0 & 1 & \cdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ -\delta_{s-1} & 0 & 0 & \cdots & 1 \\ -\delta_s & 0 & 0 & \cdots & 0 \end{bmatrix}, \quad \mathbf{\Gamma}_a = \begin{bmatrix} -\delta_1 \\ -\delta_2 \\ \vdots \\ -\delta_{s-1} \\ -\delta_s \end{bmatrix} \qquad (9)$$

For multiple-input multiple-output (MIMO) systems additional dynamics is replicated q times (once per each output). In this case the replicated additional dynamics is defined as:

$$\bar{\mathbf{\Phi}} \overset{def}{=} diag(\underbrace{\mathbf{\Phi}_a, \ldots, \mathbf{\Phi}_a}_{q\,\text{times}}), \quad \bar{\mathbf{\Gamma}} \overset{def}{=} diag(\underbrace{\mathbf{\Gamma}_a, \ldots, \mathbf{\Gamma}_a}_{q\,\text{times}}) \qquad (10)$$

The discrete-time design model ($\mathbf{\Phi}_d$, $\mathbf{\Gamma}_d$) is formed as a cascade combination of the additional dynamics ($\mathbf{\Phi}_a$, $\mathbf{\Gamma}_a$) or ($\bar{\mathbf{\Phi}}$, $\bar{\mathbf{\Gamma}}$) and the discrete-time plant model ($\mathbf{\Phi}$, $\mathbf{\Gamma}$):

$$\mathbf{x}_d[k+1] = \mathbf{\Phi}_d \mathbf{x}_d[k] + \mathbf{\Gamma}_d \mathbf{u}[k] \qquad (11)$$

$$\mathbf{\Phi}_d = \begin{bmatrix} \mathbf{\Phi} & 0 \\ \mathbf{\Gamma}^* \mathbf{C} & \mathbf{\Phi}^* \end{bmatrix}, \quad \mathbf{\Gamma}_d = \begin{bmatrix} \mathbf{\Gamma} \\ \mathbf{0} \end{bmatrix}, \quad \mathbf{x}_d = \begin{bmatrix} \mathbf{x}[k] \\ \mathbf{x}_a[k] \end{bmatrix} \qquad (12)$$

where $\mathbf{\Phi}^*$ and $\mathbf{\Gamma}^*$ denote respectively $\mathbf{\Phi}_a$ and $\mathbf{\Gamma}_a$ in the case of single-input single-output systems or $\bar{\mathbf{\Phi}}$ and $\bar{\mathbf{\Gamma}}$ for MIMO systems. For the design model (11) the feedback gain matrix $\mathbf{L}$ of the optimal LQ controller is calculated in such a way that the feedback law $\mathbf{u}[k] = -\mathbf{L}\mathbf{x}_d[k]$ minimizes the performance index (13) subject to the constraint (11), where $\mathbf{Q}$ and $\mathbf{R}$ are symmetric, positive-definite matrices. For the solution of the optimal LQ control problem the Matlab functions can be used.

$$J = \frac{1}{2} \sum_{k=0}^{\infty} (\mathbf{x}_d[k]^T \mathbf{Q} \mathbf{x}_d[k] + \mathbf{u}[k]^T \mathbf{R} \mathbf{u}[k]) \qquad (13)$$

The feedback gain matrix $\mathbf{L}$ is partitioned into

$$\mathbf{L} = [\mathbf{L}_1 \quad \mathbf{L}_2] \qquad (14)$$

so that $\mathbf{L}_1$ corresponds to the state-space model of the controlled structure, and $\mathbf{L}_2$ to the modelled additional dynamics. Block diagram of the optimal LQ control system with additional dynamics is represented in Figure 1.

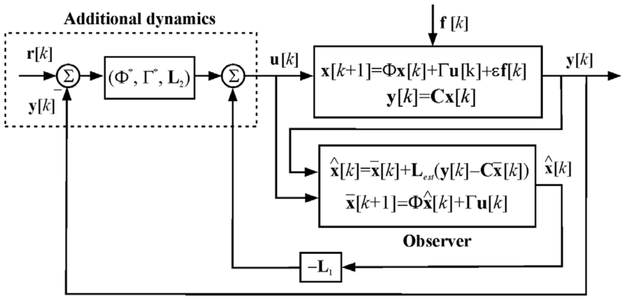

Fig. 1. Optimal LQ control system

The role of the observer is to estimate the model state variables, which cannot be directly measured, since they represent the modal states resulting from the ortho normalization procedure used in connection with the FEM modelling approach. For the state estimation the Kalman filter can be used. Equations for the Kalman filter design based on the current estimator assume the state-space equation of the plant in the form (5) and the measurements depending on the state variables and influenced by the measurement noise $\mathbf{y}[k] = \mathbf{Cx}[k] + v[k]$. The covariances of the process and measurement noise are denoted as $E(ww^T) = \mathbf{Q}_w$ and $E(vv^T) = \mathbf{R}_v$, respectively. Then the Kalman estimator is defined by the following equations:

$$\hat{\mathbf{x}}[k] = \bar{\mathbf{x}}[k] + \mathbf{L}_{est}[k](\mathbf{y}[k] - \mathbf{C}\,\bar{\mathbf{x}}[k])$$
$$\bar{\mathbf{x}}[k] = \mathbf{\Phi}\hat{x}[k-1] + \mathbf{\Gamma u}[k-1] \tag{15}$$

where the Kalman gain matrix is

$$\mathbf{L}_{est}[k] = \mathbf{P}[k]\mathbf{C}^{\mathrm{T}}\mathbf{R}_v^{-1} \tag{16}$$

and:

$$\mathbf{P}[k] = \mathbf{M}_k[k] - \mathbf{M}_k[k]\mathbf{C}^{\mathrm{T}}(\mathbf{CM}_k[k]\mathbf{C}^{\mathrm{T}} + \mathbf{R}_v)^{-1}\mathbf{CM}_k[k] \tag{17}$$

$$\mathbf{M}_k[k+1] = \mathbf{\Phi P}[k]\mathbf{\Phi}^T + \varepsilon \mathbf{Q}_w \varepsilon^T \tag{18}$$

Matrices $\mathbf{P}$ and $\mathbf{M}_k$ are determined by solving equations (17) and (18).

4. NOISE CONTROL OF A SMART ACOUSTIC BOX

Described procedure for the controller design is implemented for the noise control of a piezoelectric structure – smart acoustic box – which was modelled using the FEM approach described in section 2 taking into account the coupled electro-mechanical-acoustic behaviour. The structure consists of the clamped aluminium plate surrounded by the wooden envelope open at one side (opposite to the plate), which comprises the acoustic fluid – air (Fig. 2b).

The inner side of the aluminium plate is attached with fifteen piezoelectric patches denoted as 1–15 in Figure 2a, which can be used as actuators and sensors. Multifunctional piezoelectric material integrated with the plate enables actuation and sensing as well as the active control of the structure, when the control algorithm is implemented.

For the developed of the FEM-based state-space model, four piezo-patches (5, 6, 8, 9 in Fig. 2a) are designated as actuator-patches. Using an appropriate modelling procedure it is possible to obtain FEM-based models, which correspond to different actuator-sensor constellations. Modal reduction can also result in state-space models of different orders, determining in that way the number of inputs and outputs considered for the controller design. Excitation of the plate can cause its vibration and the consequent acoustic effects inside the box. Especially undesirable are the periodic excitations with frequencies corresponding to some of the structural or acoustic eigenmodes, since they can lead to resonant states. The behaviour of the acoustic field under excitation is expressed in terms of the air pressure change, which can be sensed by a microphone with accompanying supply Fig. 2b).

The control aim is the active noise reduction in a specified field of the acoustic box. It should be achieved using the supplied pizeo-patches, with the goal to cause the plate vibrations which will influence in turn the acoustic field inside the box in such a way that the air pressure at the selected point (where the microphone is placed) is reduced when the controller is active. For this purpose the optimal LQ controller was designed as suggested in section 3, based on the available information on the acoustic structure contained in FEM-based state-space model.

5. SIMULATION RESULTS OF THE CONTROLLER IMPLEMENTATION

Following the described modelling and controller design procedure an optimal LQ controller was designed to reduce the noise level in terms of the air pressure amplitude in a prescribed point of the acoustic box. Obtained state-space model is modally reduced and it comprises five structural (plate) and five acoustic fluid (air) eigenfrequencies. The state-space model is therefore of order 20. Furthermore the model has four inputs (actuator-patches signals) and the controlled output is the air pressure at a predefined point. The output in terms of the air pressure is converted in an appropriate way to yield the voltage [V], which can be acquired using the microphone supply. Depending on the definition of the system outputs, additional signals can be provided by the model, for example the responses of the selected sensor-patches. In this case the patches denoted as 2, 4, 7 and 11 in Figure 2a are used as additional sensors in order to return the information about the structural behavior of the plate. Since the primary aim of the controller design in this case is the noise level reduction regardless of the structural behavior, it is not required that the controller necessarily reduces the plate vibrations.

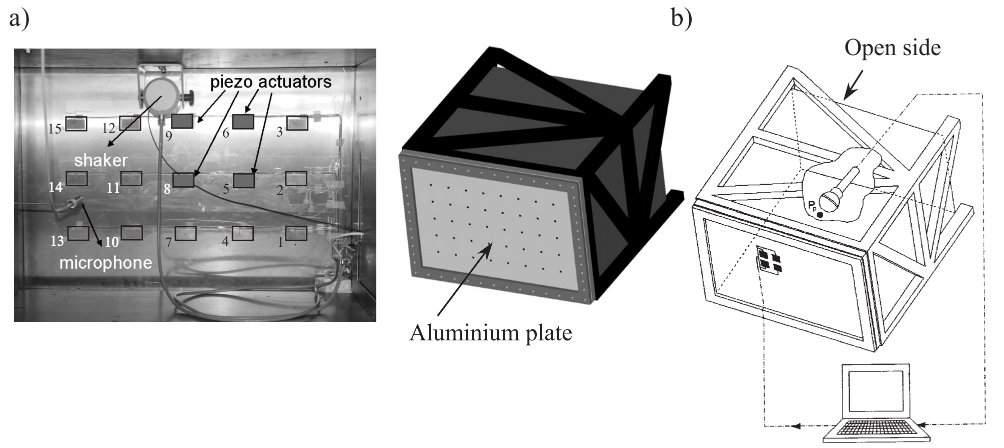

Fig. 2. Scheme of the acoustic box with the aluminium plate and the placement of the piezo-patches

Control inputs are represented in terms of voltage [V] at the actuator patches. The controller is designed and tested assuming the periodic sinusoidal excitation of the aluminium plate, where the excitation frequencies are equal to the first three selected structural eigenfrequencies of the acoustic box, calculated for the system modeled using the FEM approach. The excitation in a state-space model is force [N]. The eigenfrequencies of the structural (f_{wi}) and of the acoustic eigenmodes (f_{ai}) are listed in the Table 1.

Table 1

Calculated eigenfrequencies of the elastic plate and of the acoustic cavity

Number i	1	2	3	4	5
f_{wi} [Hz]	66.7	106.2	163.8	172.1	201.2
f_{ai} [Hz]	68.0	200.8	204.0	278.1	291.5

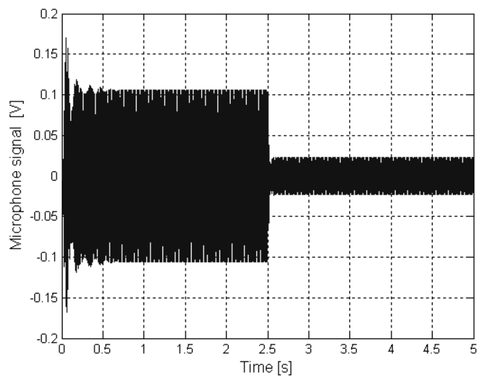

Fig. 3. Uncontrolled and controlled microphone output signal

The simulation results are represented in the following figures. Figure 3 represents the output microphone signal [V] when the excitation is sinusoidal with the frequency f_{w1} and amplitude 0.4. The controller is switched on after 2.5 s. The reduction of the air pressure amplitudes is obvious, which confirms the controller efficiency for the noise reduction. The four actuator signals are represented in Figure 4a. During the period without control (2.5 s) this signals are equal to zero, and after switching the controller on, they are within the range which allows the implementation of the controller for example using dSPACE system. The zoomed portion of the control signals is shown in Figure 4b.

For the periodic excitations with the frequencies equal to the second and third plate eigenfrequencies, the result of the uncontrolled and controlled case are shown in Figures 5a and 5b. In this case the controller is again switched on after 2.5 s. In the presence of the periodic excitation 0.4sin($2\pi f_{w3}$) the control system performs besides the air pressure reduction also the structural vibration suppression. The simulated signals of the four sensor patches (2, 4, 7 and 11, see Fig. 2a) are represented in Figure 6. In the presence of other excitations the vibration suppression is not observed by all sensor patches, but since the control aim is the reduction of the pressure level, the structural vibrations were not relevant with respect to the control objective defined in such a way.

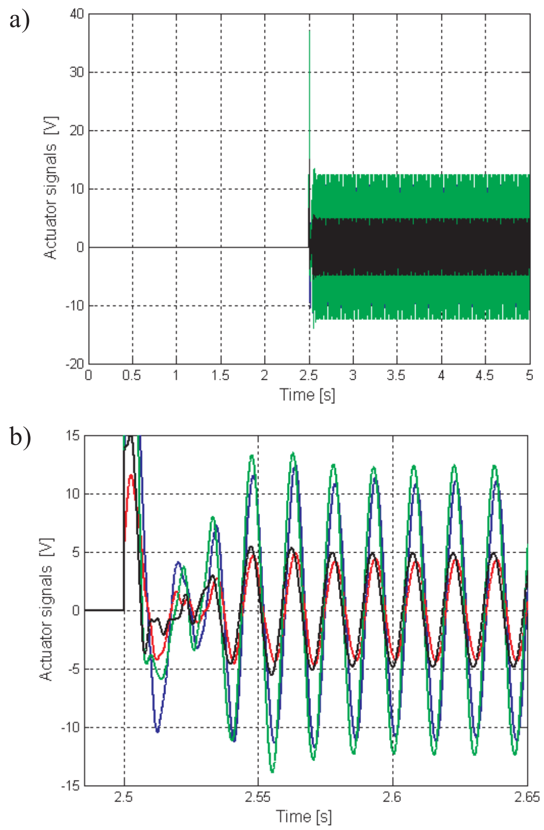

Fig. 4. Control inputs and the zoomed portion of the signals

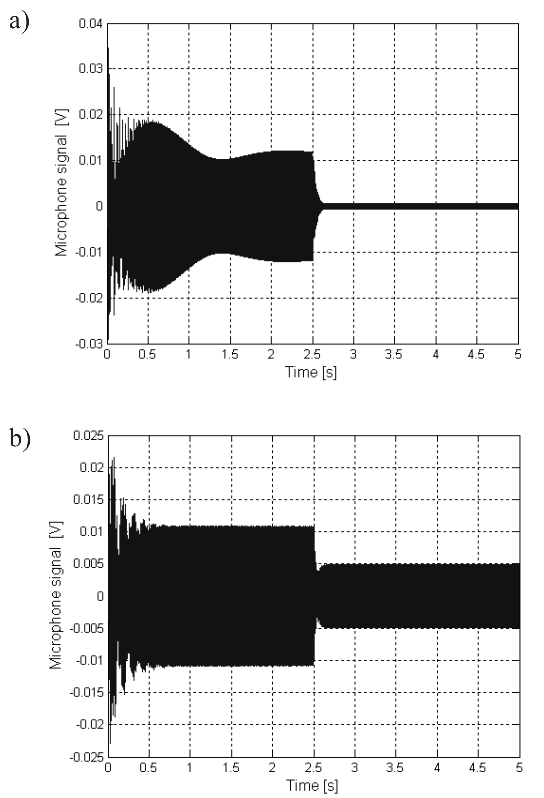

Fig. 5. Uncontrolled and controlled microphone output signal (the controller is switched on after 2.5 s): a) excitation 0.4sin($2\pi f_{w2}$); b) excitation 0.4sin($2\pi f_{w3}$)

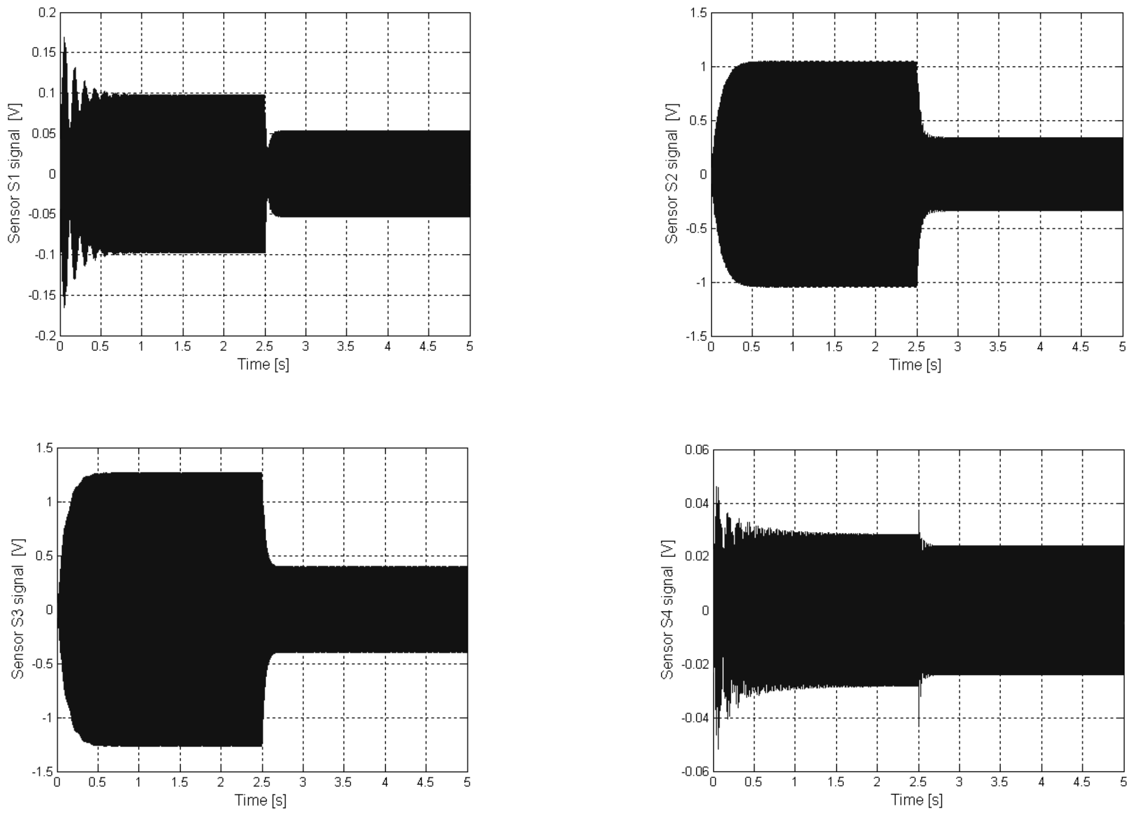

Fig. 6. Sensor-patch signals in the presence of the excitation $0.4\sin(2\pi f_{w3})$

a)

b)

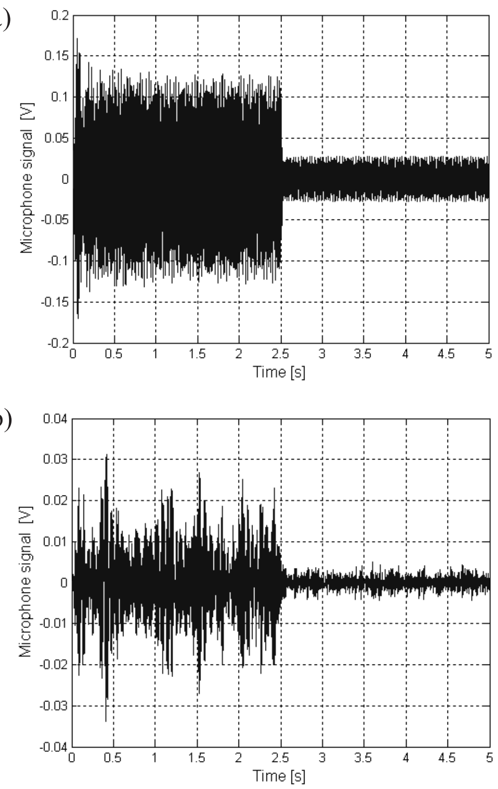

Fig. 7. Uncontrolled and controlled microphone output signal: a) excitation $\sum_{i=1}^{3}\sin(f_{wi})$; b) random excitation

Finally the controller was tested with the excitation obtained as a sum of three periodic sinusoidal signals with the frequencies f_{w1}, f_{w2}, f_{w3} as well as with the random excitation signal. The results for the output response are represented in Figures 7a and 7b, respectively. The pressure amplitude reduction is observed in these cases as well.

6. CONCLUSION

In this paper the FEM-based approach to modelling of the coupled electro-mechanical-acoustic behaviour of complex piezoelectric structures is suggested as a meaningful way to obtain the state-space model suitable for the controller design. The control objective in this case is the noise reduction in terms of the pressure suppression of the acoustic fluid at a predefined point. For this purpose the optimal LQ controller with additional dynamics was designed and tested using the FEM model of a smart acoustic box with attached piezoelectric patches used as actuators and sensors. For different types of excitations the controlled system performed reduced fluid pressure amplitudes in comparison with the uncontrolled case.

Acknowledgement

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