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INVESTIGATIONS OF A NOISE CONTROL OF A PLATE WITH FOUR PAIRS OF PIEZOELECTRIC ELEMENTS

J. WICIAK

AGH University of Science and Technology
Faculty of Mechanical Engineering and Robotics
Department of Mechanics and Vibroacoustics
Al. Mickiewicza 30, 30-059 Kraków, Poland
e-mail: wiciak@agh.edu.pl

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The paper explores the potentials of reducing the amplitude of plate vibrations and radiated noise through the use of piezoelectric actuators in an asymmetric configuration. Tests were run on a aluminium plate, bonded on one edge. On one of the surfaces there were four piezoelectric actuators, on the other surface there were four piezoelectric sensors. One of the actuators served as vibrations generator, while the remaining were used for vibration control. The influence of the actuators activation and shape form on the structure's response are investigated. The finite element method analysis was performed, with the application of the ANSYS computer package, and then obtained results were confirmed experimentally.

Key words: piezoelectric actuators, finite element analysis, sound pressure level.

1. Introduction

The unique properties of piezoelectric materials are utilised to convert electric energy into mechanical energy and piezoelectric transducers are widely used as actuators and sensors for vibration control of flexible structures [6, 8, 9].

The applications of PZTs elements with different control configurations for reducing vibrations and structure – borne noise have been studied extensively [3, 14]. Active control of a beam and plate like structures, and the models of coupled electromechanical behaviour within static, dynamic approach is presented by PIETRZAKOWSKI in [13]. The problem of the active vibration control of a simply supported circular plate was presented by L. LENIOWSKA [10, 11]. The axially symmetrical vibrations of the plate were measured by application of five pairs of strain sensors located along the plate radius for the considered system. The output error identification method was used and cascaded P-PID control algorithm for vibration cancellation [10]. In the article [11] the mathematical model of the clamped circular plate was obtained on a base of registration of the system response on fixed excitation and feed forward vibration algorithm is used for vibration cancellation.

In the article [1] the active control via piezoceramic patches of acoustic pressure in 2D cavity with a flexible boundary and approximation method in the concept of an LQR state space formulation are discussed. A new theoretical derivation for the dynamic response of a clamped, rectangular plate excited by steady-state harmonic point forces is presented in [15]. Authors obtained reduction 24 dB of the total power radiaded from the plate. The finite element modeling of actuation of an aluminum cantilever beam by two thin plates made of piezoelectric PZT – ceramic material and sound radiation of the vibrating beam is presented in the article [12].

A new volume displacement sensor made of PVDF film and the experimental implementation of this sensor in an active control system of sound radiation from a plate is presented in [4]. The modelling method for the active minimization of noise within a three-dimensional irregular enclosure using distributed lead zirconate titanate piezoelectric actuators and the control mechanism is presented in the article [7]. The irregular enclosure was modelled with four rigid panels and two simply supported flexible panel with PZT actuators. It was found that the control mechanism vary with disturbance frequencies and at the most disturbance frequencies, the SPL within enclosure was reduced by restructuring the modes of two panels simultaneously.

According to the theoretical principles, plate vibrations excited by actuators are affected by the position of piezoelectric actuators and the control voltage [3]. Changing the actuator positions on the plate surface and properly adjusting the control voltage, such values of these parameters are found that vibration damping be maximal for the given frequency [16].

This study is a continuation of earlier works of the author [2, 16] and explorations how the sequences of changing of activated actuators and the level of applied voltage affect on sound radiation of a aluminium plate with four pairs piezoelectric elements. The plate is actuated by one of the four actuators, while the remaining actuators are used to control plate vibrations. The value of the control voltage on control actuators depends on induced and measured voltage value on sensors.

2. Tested object and FEM model

The steady state transverse vibrations $w(x, y, \omega)$ of a thin plate are governed by the equation:

$$\frac{E_p h^3}{12(1-\nu_p^2)}\nabla^4 w + j\omega C w - \rho_p h_p \omega^2 w = p_e\,, \tag{1}$$

where E_p – Young modulus of plate material [Pa], h – plate thickness [m], ν_p – Poisson's ratio of plate material [–], C – damping coefficient [1/s], ρ_p – density of plate material [kg/m^3], p_e – pressure due to an external excitation.

For the plate excited by a single actuator and control due to the action of remaining three piezoelectric actuators, p_e has the following form:

$$p_e = -C_0 \frac{d_{31} V}{h_{pe}} \left\{ \begin{array}{l} -\left[\delta'(x-x_{1W}) - \delta'(x-x_{2W})\right] \cdot \left[H(y-y_{1w}) - H(y-y_{2W})\right] \\ -\left[H(x-x_{1W}) - H(x-x_{2W})\right] \cdot \left[\delta'(y-y_{1W}) - \delta'(y-y_{2W})\right] \\ +\sum\limits_{i=1}^{3} \left[\delta'(x-x_{1i}) - \delta'(x-x_{2i})\right] \cdot \left[H(y-y_{1i}) - H(y-y_{2i})\right] \\ +\sum\limits_{i=1}^{3} \left[H(x-x_{1i}) - H(x-x_{2i})\right] \cdot \left[\delta'(y-y_{1i}) - \delta'(y-y_{2i})\right] \end{array} \right\}, \tag{2}$$

where C_0 – material geometric constant for two dimensional asymmetric wafer actuator, V – applied voltage, h_p – thickness of plate, h_{pe} – thickness of piezoelectric material, d_{31} – piezoelectric material strain constant, (x_{1i}, y_{1i}) and (x_{2i}, y_{2i}) – coordinates of the piezoelectric actuators, E_p – Young modulus of plate material [Pa], h – plate thickness [m], H – Heaviside step function.

The plate vibration response $w(x, y, \omega)$ comprises mode shape functions ψ_{mn} and natural frequencies ω_{mn} (obtained from the solution of the eigenvalue problem)

$$w(x, y, \omega) = \sum_{m,n} W_{mn} \psi_{mn}(x, y), \tag{3}$$

where W_{mn} – modal amplitudes of the plate displacement, P_{mn} – modal force

$$W_{mn} = \frac{P_{mn}}{\Lambda_{mn}\left[\rho_p h_p (\omega_{mn}^2 - \omega^2) + jC\omega\right]}, \tag{4}$$

$$P_{mn} = \int\limits_S p_e(x, y) \psi_{mn}(x, y) \, \mathrm{d}S, \tag{5}$$

$$\Lambda_{mn} = \int\limits_S \psi_{mn}^2(x, y) \, \mathrm{d}S. \tag{6}$$

For the far field, the sound pressure in distance r is describe by the following equation:

$$p(r, \omega) = -\frac{\rho_0 \omega^2}{2\pi} \frac{e^{-jkr}}{r} \sum_{m,n} W_{mn} \int\limits_S \psi_{mn} e^{jkr_0 \cos\delta} \, \mathrm{d}S, \tag{7}$$

where k – wave number, ρ_0 – air density [kg/m^3], Θ – angle between observation vector r and z axis (perpendicular to plate).

3. Tested object and FEM model

The aluminium plate $100 \times 100 \times 2$ mm considered in the study was fixed along one edge. On its two external surfaces there were eight piezoelectric elements integrated with the plate. Actuators and sensors $20 \times 20 \times 1$ mm made from PIC-140 material were mounted symmetrically on the plate. Inner surfaces (electrodes) of piezoelectric actuators adhering to the plate were grounded and the potential was applied on the upper (external) electrode. The plate was actuated by single PIC – actuator (marked A1), while the remaining actuators (marked A2, A3, A4) were used to control plate vibrations. The piezoceramic sensors were marked S1, S2, S3, S4. The geometry of the model is shown in Fig. 1. The properties of the beam are summarised in Table 1.

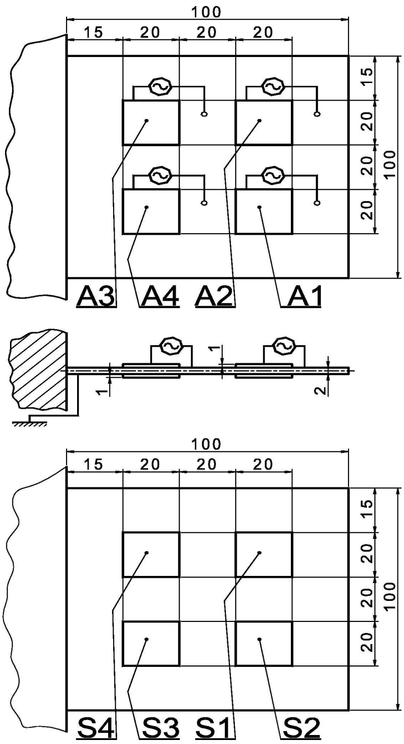

Fig. 1. Model geometry.

Table 1. Material properties of the experimental beam and piezoceramic.

Material	Piezoceramic PIC140				Aluminum	
Density [kg·m^{-3}]	ρ	7600			ρ	2704
Elasticity modulus [Pa]	E_{11}	$8.547 \cdot 10^{10}$	E_{33}	$6.803 \cdot 10^{10}$	E	$7.2 \cdot 10^{10}$
Poisson ratio	υ_{xy}	0.29	υ_{xz}	0.34	υ	0.33
Charge constants [m·V^{-1}]	d_{31} d_{33} d_{51}	$-60 \cdot 10^{-12}$ $200 \cdot 10^{-12}$ $265 \cdot 10^{-12}$				
Relative permittivity	$\varepsilon_{11}/\varepsilon_0$	680	$\varepsilon_{33}/\varepsilon_0$	800		

FEM analysis of plate vibrations was performed using the Ansys package. Solid element SOLID45 and coupled fields element SOLID5 were chosen. Piezoelectric layers and the steel plate by four layers of finite elements. The layer of adhesive agent was not considered in the analysis. The analysis was performed for the first four natural frequencies of vibrations of the system with piezoelectric actuators. The material damping ratio was taken from measurements (see Sec. 3).

In the plate-acoustic model, the structural sound produced by the vibrating plate were radiated into the space $2 \times 2 \times 2$ m. The parameters of the acoustic medium assumed for the numerical procedures were: density – 1.225 kgm^{-3}, speed of sound – 343 ms^{-1}. An absorbed material with the sound absorption ratio 0.95 was placed on six external surfaces bounding the acoustic volume.

The acoustic volume is modelled by solid elements of the type FLUID30. Two kinds of acoustic elements are generated: those in contact with walls and vibrating surface (with UX, UY, UZ, and PRESS as the degrees of freedom) and those in the interior volume (not in contact with walls, with PRESS as the degrees of freedom).

4. Results and conclusions

The damping loss factor was measured according to the decay time method. The reverberation time T was been measured and the loss factor has been determined from the relation [5]:

$$\eta = \frac{\ln(10^6)}{2\Pi f T_{60}}, \tag{8}$$

where f – frequency, T_{60} – reverberation time (within which the energy of the vibration is reduced to one millionth of its initial value).

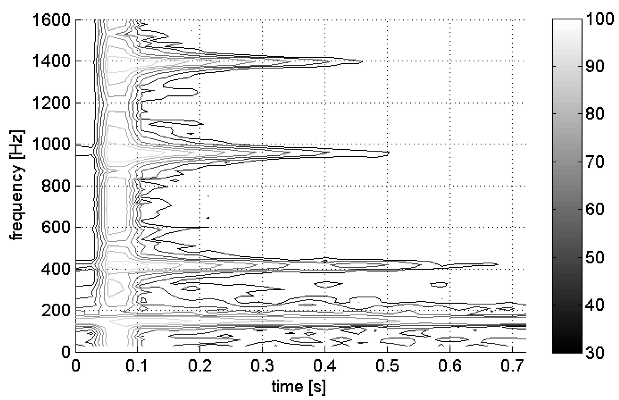

Fig. 2. Waterfall spectrogram.

The experiments have been realised in the anechoic chamber (absorption coefficient – 0.95, see Fig. 4). The plate was driven by harmonic voltage with an amplitude $V_{\mathrm{RMS}} =$

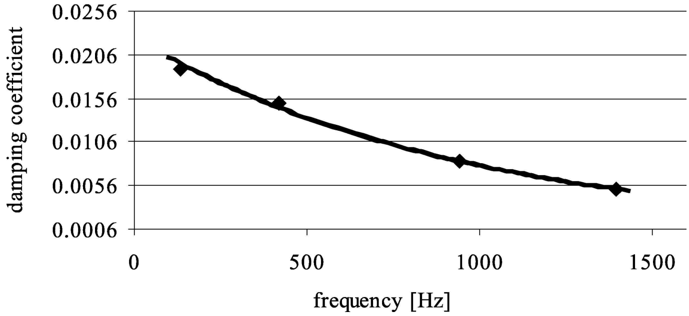

Fig. 3. Measured damping loss factor.

40 V. The set up was built up of generator NDN DF1642B or white noise generator BK 1405, preamplifier VTL MB 150 and 8 channel voltage amplifier. Sensors response was read on DAQ Prosig 2600 and oscilloscope Summit Scope Plus 760.

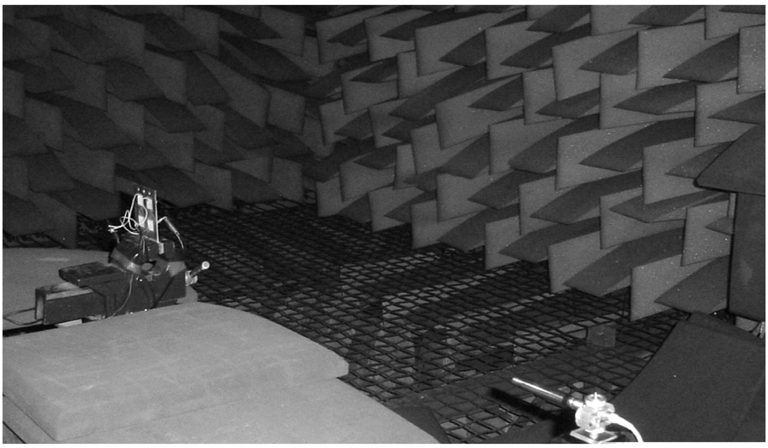

Fig. 4. Experimental model in the anechoic chamber.

Sound pressure level measurements have been realized in the plane perpendicular to the plate in five points (radius 1 m, angle $45°$), using following measurement set-up: 1/2″microphone, analyser Svan 912AE and PC (see Fig. 5). The frequency response of the plate system and sound pressure level in chosen points on white noise excitation and are shown in Fig. 6.

Numerical and measured results, without damping, expressed as sound pressure levels for the first two vibration modes are presented in Table 2. Since values of SPL for first and second resonance frequencies were small, further only third and fourth resonance frequencies were investigated.

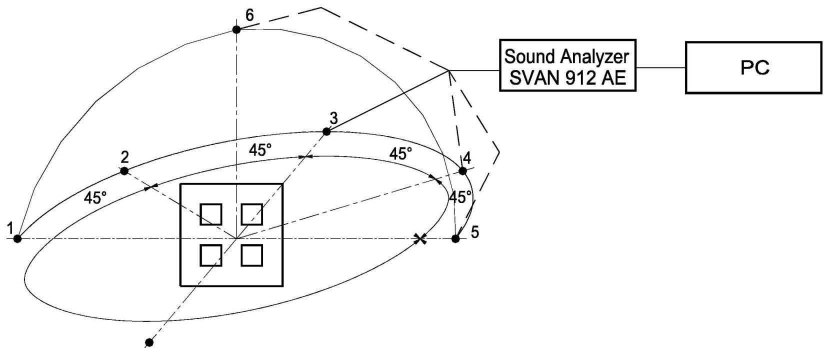

Fig. 5. Measurement points distribution.

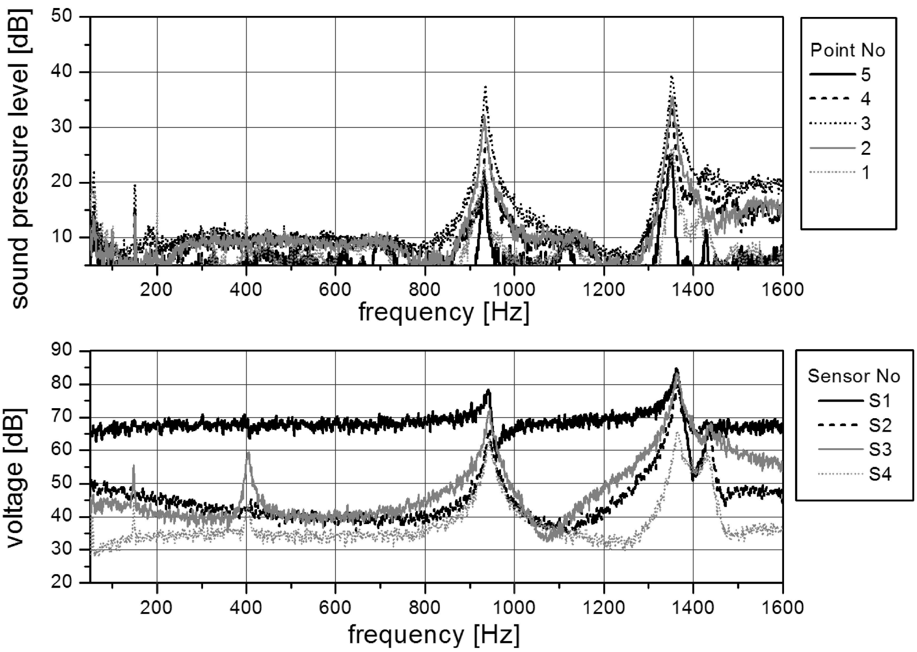

Fig. 6. The frequency response of the plate system on white noise excitation (measurements in the anechoic chamber).

Results from numerical calculations, without and with damping, expressed as sound pressure levels for the given vibration modes are presented in Tables 3 and 4.

Comparison between results obtained from the Ansys model and the measured values are presented in the Figs. 7 and 8.

The difference between the results obtained by means of the numerical models and the experimental data are approximately 3 dB but may be as high as 6 dB for in some points and single frequency bands. That means that the proposed FEM method and thus developed models can be used for analysis of such systems before investigation measurements.

Table 2. Estimated (Ansys calculations) and measured sound pressure levels for first two resonance frequencies.

	Sound pressure level [dB]					
	point/angle [°]					
	mode/Hz	0°	45°	90°	135°	180°
Exper.	151.0	–	11.2	15.3	11.4	–
Ansys	175.2	–	8.3	12.3	8.4	–
Exper.	421.0	–	10.0	12.2	11.0	–
Ansys	488.5	–	5.1	10.2	6.7	–

Table 3. Estimated sound pressure levels (Ansys) for the third resonance frequency.

point angle [°]	Sound pressure level [dB]							
	OFF	ON A2	ON A3	ON A4	ON A2, 3	ON A2, 4	ON A3, 4	ON A2, 3, 4
	$V =$	40 [V]	85 [V]	80 [V]	25 [V]	25 [V]	40 [V]	20 [V]
0°	48.0	11.0	14.0	12.1	28.6	28.0	19.1	16.3
45°	58.6	20.0	23.5	20.0	34.8	34.0	25.1	22.6
90°	60.4	23.5	28.0	25.1	38.6	37.5	28.9	26.4
135°	52.0	20.0	23.5	20.0	34.8	34.0	25.1	22.6
180°	37.5	13.0	18.6	14.0	25.8	26.0	16.3	13.5

OFF – without damping, ON A2 – activated actuator no. 2, ON A3 – activated actuator no. 3, ON A4 – activated actuator no. 4, ON A2, 3 – activated actuators no. 2 and no. 3, ON A2, 4 – activated actuators no. 2 and no. 4, ON A3, 4 – activated actuator no. 3 and no. 4, ON A2, 3, 4 – activated actuators no. 2, no. 3 and no. 4.

Table 4. Estimated sound pressure levels (Ansys) for the fourth resonance frequency.

point angle [°]	Sound pressure level [dB]							
	OFF	ON A2	ON A3	ON A4	ON A2, 3	ON A2, 4	ON A3, 4	ON A2, 3, 4
	$V =$	45 [V]	90 [V]	110 [V]	30 [V]	35 [V]	50 [V]	25 [V]
0°	54.8	23.5	28.0	25.1	19.1	29.2	21.6	5.0
45°	53.6	21.9	25.6	22.6	20.8	32.0	19.5	3.0
90°	58.3	27.8	31.6	28.6	24.3	34.8	25.3	8.8
135°	53.0	21.2	26.2	23.2	20.8	32.0	20.0	3.0
180°	54.8	23.5	28.0	25.1	18.6	19.5	21.6	5.0

The position of the actuator no 2 seems to be more accurate and favourable. This position allows simultaneous noise reduction for both resonance frequencies. Depending on the frequency, measurement point position and the applied voltage sound pressure

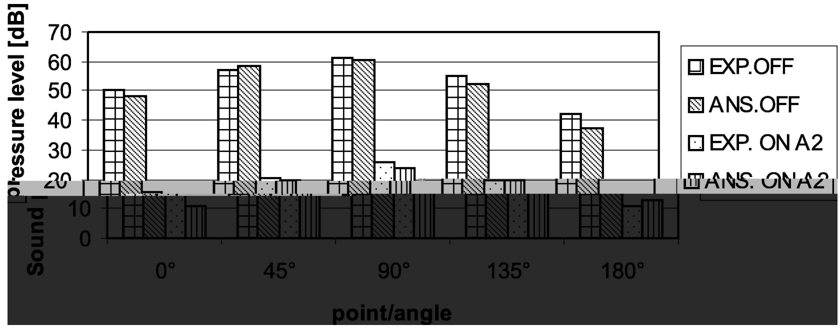

Fig. 7. Sound pressure level vs. excitation frequency, experimental and numerical data without and with damping – third resonance mode.

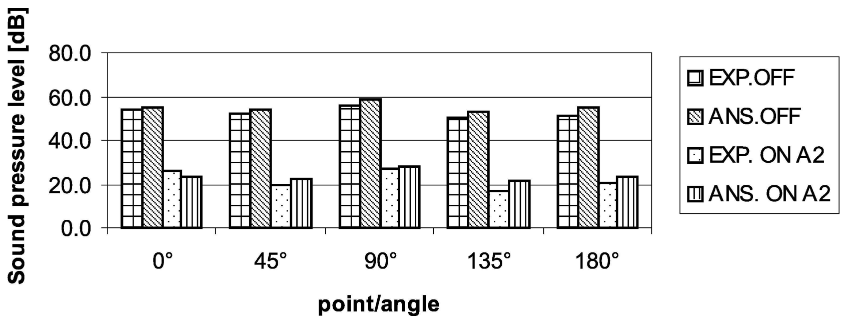

Fig. 8. Sound pressure level vs. excitation frequency, experimental and numerical data without and with damping – fourth resonance mode.

level is reduced from 30.8 dB to 36.7 dB for third resonance frequency and from 28.1 dB to 33.7 dB for the fourth resonance frequency in the anechoic chamber.

Similar reduction of the averaged squared normal velocity can be achieved for various actuators configurations and at various – bigger voltage levels. SPL is reduced from 26 to 31 dB.

Application of the three actuators increase the damping affect by about 10 dB with relation to single actuator.

The manner of plate constraining and improper bonding of piezoceramic elements have great influence on the plate resonance frequencies. Also improper bonding of piezoceramic elements can greatly reduce electromechanical effectiveness of the piezoelements.

It is readily apparent that structural vibrations and hence the level of sound radiation, can be reduced through the application piezoelectric actuators.

It was shown that major factors affecting the vibration reduction performance include the shape and actual configuration of piezoelectric elements and also the amplitude of applied voltage. These parameters are associated with the mode shapes and might be optimised or controlled in active noise reduction and vibration reduction systems.

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References

[1] BANKS H. T., FANG W., SILCOX R. J., SMITH R. C., *Approximation methods for control of structural acoustics models with piezoceramic actuators*, J. of Intel. Mat. Systems and Structures, **4**, 98–116 (1993).

[2] FILIPEK R., WICIAK J., *Sound radiation of beam with shunted piezoceramics*, Archives of Acoustics, **30**, 4 (Supplement), 41–46 (2005).

[3] FULLER C. R., ELLIOT S. J., NELSON P. A., *Active control of vibration*, Academic Press, San Diego, USA 1997.

[4] CHARATTE F., BERRY A., GUIGOU C., *Active control of sound radiation from a plate using a polyvinylidene fluoride volume displacement sensor*, J. Acoust. Soc. Am., **103**, 3, 1493–1503 (1998).

[5] CREMER L., HECKL M., UNGAR E. E., *Structure-Borne Sound*, Springer Verlag, 1988.

[6] FLAMING A. J., MOHEIMANI S. O. R., *Adaptive piezoelectric shunt damping*, Smart Materials and Structures, **12**, 36–48 (2003).

[7] GENG H. C., RAO Z. S., HAN Z. S., *New modelling method and mechanism analyses for active control of interior noise in a irregular enclosure using piezoelectric actuators*, J. Acoust. Soc. Am., **113**, 3, 1439–1447 (2003).

[8] HAGOOD N. W., VON FLOTOW A., *Damping of structural vibrations with piezoelectric materials and passive electrical network*, J. of Sound and Vibr., **146**, 2, 243–268 (1991).

[9] KIM J., KIM J.-H., *Multimode shunt damping of piezoelectric smart panel for noise reduction*, J. Acoust. Soc. Am., **116**, 2, 942–948 (2004).

[10] LENIOWSKA L., *Active vibration control of simply supported circular plate*, Proc. Open Seminar on Acoustics, pp. 533–538, Warszawa – Stare Jabłonki 2002.

[11] LENIOWSKA L., SZELA S., *Vibration suppression of circular plate via piezoelectric actuators and feed forward controller*, Proc. Open Seminar on Acoustics, p. 187–190, Gdańsk – Sobieszewo 2004.

[12] NGUYEN C. H., PIETRZKO S. J., *Piezoelectric-mechanical-acoustic coupling from PZT-actuated vibrating beam and its sound radiation*, Mechanical Systems and Signal Processing, **18**, 929–945 (2004).

[13] PIETRZAKOWSKI M., *Active damping of transverse vibration using distributed piezoelectric elements*, Warsaw University of Technology, Warszawa, Poland 2004.

[14] REAVES M. C., HORTA L. G., *Test cases for modeling and validation of structures with piezoelectric actuators*, 42nd AIAA/ASME/ASCE/AHS/ASC Structures, Struc. Dyn., and Materials Con., Seattle, Washington, AIAA 2001–1466.

[15] SUNG C. C., JAN C. T., *Active control of structurally radiated sound from plates*, J. Acoust. Soc. Am., **102**, 1, 370–381 (1997).

[16] WICIAK J., *The influence of position of piezoelectric actuators on vibrations and acoustics radiation reduction of simply supported plate*, Polish Journal of Environmental Studies, **13** (Supplement III), 125–130 (2004).