STRUCTURAL ACOUSTIC CONTROL OF A ONE SIDE LOADED CIRCULAR PLATE WITH PIEZOELECTRIC ACTUATORS

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The experiment presented in this paper is a part of an ongoing research project connected with the structural acoustic control in one side fluid loading structures. The system considered in the study is built up of a thin, circular plate and eight square piezoelectric elements. The plate is clamped along its edge by finite rigid co-planer baffle. The baffle with other four rigid walls is formed aquarium filled with water. Numerical computations utilize the FEM approach supported by Ansys software. The natural frequencies were determined using modal analysis. The harmonic analysis covers the acoustic radiation due to steady-state plate vibrations for the first two modes of natural vibrations.

Each mode was examined individually. The test results indicate that actuators as flexural wave-source can decrease the displacement response as well as sound radiation for a pure tone input.

INTRODUCTION

Structural acoustic control of plate, shell and hull structure is very important for structural stability and has a wide range of applications in engineering science and technology. Plate structures, hull or cylindrical shell structures are widely used in many engineering structures such as submarines, aircraft, pressure vessels, chimney design, pipe flow, offshore technologies, etc. [18]. Submarines and ships have a number of broadband and narrowband underwater noise sources, such as: main engine, diesel generator, auxiliary machinery, etc [7, 8, 10]. Results of identification tests of acoustic field spectrum of underwater noise generated by ship in motion and connection with activity of ship mechanisms and devices in operation are presented in [9].

Structure vibrations and structural noise may be reduced by passive and active isolation, by passive and active vibration and sound absorbers or by active control [7, 20, 24].

Dimitriadis and Fuller demonstrated that piezoelectric elements bonded to a plate could be employed to reduce the harmonic sound transmitted or radiated from circular plates [4]. They concluded that the shape and position of the actuators markedly affects the distribution of the response among the different modes. The vibration of circular plates structures excited by piezoelectric actuators has been modelled by Van Niekerk et al. [16], Tylikowski [19] and Sekuori et al. [17]. Niekerk et al. [16] presented an active control approach in transient noise transmission through a plate in a circular duct. Sekouri et al. [17] presented the mathematical solution based on Kirchoff plate model for free vibration. The control of radiated noise from ship's cabin floor has been presented by Won-Ho at al. [24]. They concluded that the combined noise level of a cabin could be dominated by the radiated noise from the stiffened steel plate system in combination with deck fool.

Recently, Donoso et al. [5] presented a new way to systematically design distributed piezoelectric modal sensor/actuators for circular plates with polar symmetric boundary conditions. Sohn et al. presented active vibration control of the smart hull structure with macro-fiber composite [18]. Authors [18] demonstrated that structural vibration could be effectively suppressed by applying proper control input voltages to the MFC actuators determined by the control algorithm. Also prediction of natural frequency changes due to the presence of fluid is important for designing structures which are in contact with or immersed in fluid [2, 3, 11, 12, 14].

The experiment presented in this paper is a part of an ongoing research project connected with the structural acoustic control in one side fluid loading structures [21–23]. The system considered in the study is built up of a thin, circular plate and eight square piezoelectric elements. The plate is clamped along its edge by finite rigid co-planer baffle. The baffle with other four rigid walls is formed aquarium filled with water. The natural frequencies were determined using modal analysis. The harmonic analysis covers the acoustic radiation due to steady-state plate vibrations for the first seven modes of natural vibrations. Each mode was examined individually. The test results indicate that actuators as flexural wave-source can decrease the displacement response as well as sound radiation for a pure tone input.

1. FEM MODEL

The vibrating element is a circular aluminium plate of radius $\phi = 0.15$ m, with thickness $h_1 = 1$mm or with thickness $h_2 = 2$ mm. The plate is clamped along its edge by finite rigid co-planer baffle. The plate is loaded on one side by heavy fluid sea water and on the other side has contact with a gaseous medium (air). On the side of the gaseous medium, eight piezoelectric elements are bonded to the plate with a thin layer of glue.

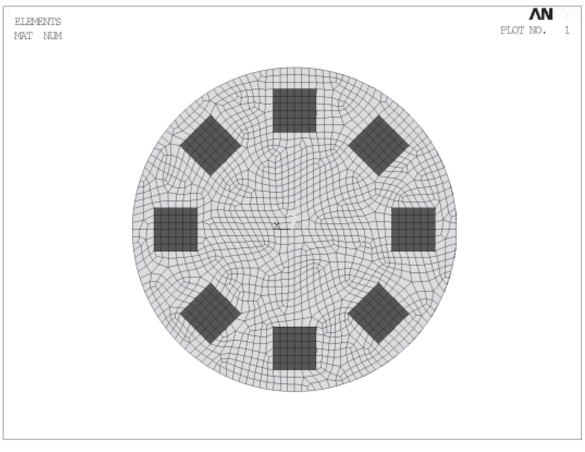

Fig. 1. FEM model of a plate with piezoelements PIC 140

Piezoelectric elements 20 x 20 x 1 [mm] made from PIC-140 material are arranged in a set located on concentric circle with radii ϕ = 0.55 m. The geometrical model of system circular plate-piezoceramics is presented on fig.1 and the properties of the plate and piezoceramics are summarized in table 1 and 2.

Table 1. Material properties of the piezoceramic PIC 140

Piezoceramic PIC 140				
Density [$kg \cdot m^{-3}$]	Poisson Ratio	Elastic constant [$10^{-12} m^2 N^{-1}$]	Charge constants [$10^{-12} m \cdot V^{-1}$]	Relative permitivity
ρ 7600	υ_{xy} =0.29 υ_{xz} = 0.34	S_{11}=11.7, S_{12}=-4,1, S_{33}= 14.7	d_{31} = 60 d_{33} = 200 d_{51} = 265	$\varepsilon_{11}/\varepsilon_0$ 680 $\varepsilon_{33}/\varepsilon_0$ 800

Table 2. Material properties of the experimental aluminium plate

aluminium		
Density [$kg \cdot m^{-3}$]	Elasticity modulus [Pa]	Poisson Ratio
ρ 2700	E $6.9 \cdot 10^{10}$	υ 0.33

Calculations of the pressure level and displacement at chosen control points are based on the acoustic field - piezoelectric - structure interaction module available in the ANSYS package[1]. For harmonic analyses, the following relationships are hold:

$$\begin{bmatrix} [M_e] & [0] & [0] \\ [M^{fs}] & [M_e^p] & [0] \\ [0] & [0] & [0] \end{bmatrix} \begin{Bmatrix} \{\ddot{w}_e\} \\ \{\ddot{p}_e\} \\ \{\ddot{V}\} \end{Bmatrix} + \begin{bmatrix} [C_e] & [0] & [0] \\ [0] & [C_e^p] & [0] \\ [0] & [0] & [0] \end{bmatrix} \begin{Bmatrix} \{\dot{w}_e\} \\ \{\dot{p}_e\} \\ \{\dot{V}\} \end{Bmatrix} + \begin{bmatrix} [K_e] & [K^{fs}] & [K^z] \\ [0] & [K_e^p] & [0] \\ [K^z]^T & [0] & [K^d] \end{bmatrix} \begin{Bmatrix} \{w_e\} \\ \{p_e\} \\ \{V\} \end{Bmatrix} = \begin{Bmatrix} \{F_e\} \\ \{0\} \\ \{L\} \end{Bmatrix} \quad (1)$$

where: $[M_e^p]$, $[M_e]$ – liquid and structural mass matrix, $[K_e^p]$, $[K_e]$ – liquid and structural stiffness matrix, $[C_e^p]$, $[C_e]$ – liquid and structural damping matrix, $[M^{fs}] = \rho_0[R_e]$ – coupling mass matrix, ρ_0 – mean liquid density, $[K^{fs}]$=- $[R_e]$, $[K^d]$ – dielectric conductivity matrix, $[K^Z]$-piezoelectric coupling matrix, $[F_e]$ – applied load vector, {L} - electrical load vector (nodal, surface, and body charge), $\{u_e\}$ – nodal displacement component vectors, $\{p_e\}$ – nodal pressure vectors, V - vector of nodal electrical potential.

The electromechanical constitutive equations for linear plate material behaviour can be written as:

$$\begin{Bmatrix} \{T\} \\ \hline \{D\} \end{Bmatrix} = \begin{bmatrix} [c] & [e] \\ \hline [e]^T & -[\varepsilon] \end{bmatrix} \begin{Bmatrix} \{S\} \\ \hline -\{E\} \end{Bmatrix} \quad (2)$$

where: {T} – stress vector, {D} – electric flux density vector, {S} – strain vector, {E} – electric field vector, [c] – elasticity matrix, [e] – piezoelectric stress matrix, [ε] – dielectric matrix (evaluated at constant mechanical strain).

Underlying the FEM model is the assumption that there should be at least six grid elements per the considered wavelength. The layer of adhesive agent is not considered in the analysis. The plate and piezoceramic elements are modelled by solsh190 element and coupled fields element (structure – piezoelectric) solid 226 respectively [1].

Piezoelectric layers are modelled by four layers of finite elements. An absorbing material with the sound absorption ratio 0.01 was placed on the external surfaces bounding the water volume, the sound was totally reflected.

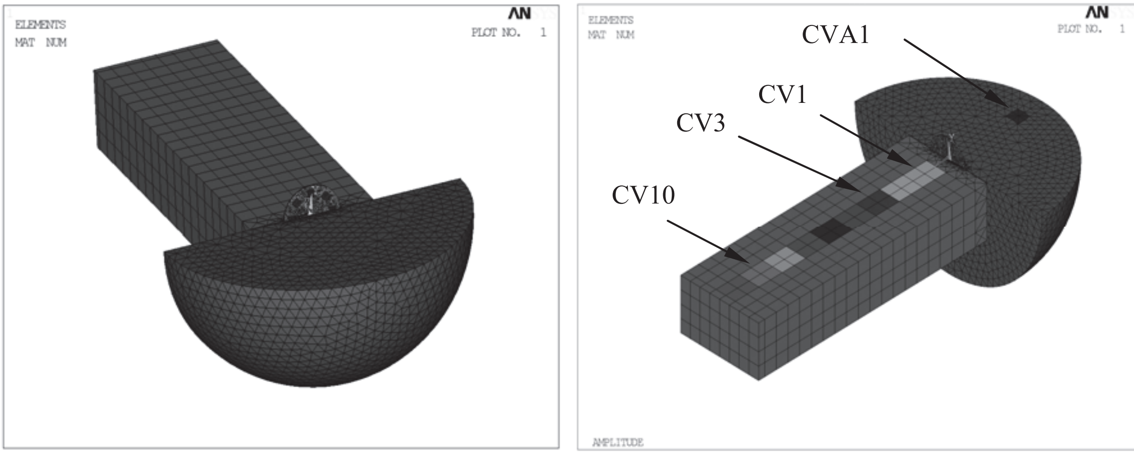

Fig. 2. A half of the water - air space and plate divided into finite elements

In the plate-acoustic space model structural sounds produced by the vibrating plate were radiated to the water space and to the semi-sphere (0.4 m) of air surrounding the plate (fig. 2).

A discretisation procedure was applied whereby the structure plate-piezoelectric should comprise nearly 17 thousand solid226 and solsh190 elements and the acoustic volume should comprise nearly 18 thousand fluid30 and infinite fluid130 elements. The mesh was finer in the plate's neighbourhood.

The parameters of the acoustic medium assumed for the numerical procedures were: air density – 1.225 kgm^{-3}, speed of sound in air – 343ms^{-1}, water density 1000 kgm^{-3}, speed of sound in water – 1490 ms^{-1}. The material damping ratio, independent of frequency is taken as 1 x 10^{-3} [–] for the whole system.

Values of sound pressure level in water is calculated at ten control volumes (Cv1-CV10) along the aquarium and at one in volume in air at 0.3 m distance from the plate surface (CVA) (fig. 2).

2. NUMERICAL RESULTS

Assuming: the fluid motion is small, incompressible, inviscid and irrotational, the dynamic loading of the fluid has an insignificant effect on the deflection curve (small changes in the kinetic and potential energies), the plate is thin-elastic plate, the small deflection theory of plates can be applied, the energies are not changed when evaluated in vacuum or in fluid [2, 3, 11, 15] the following relations between natural frequencies in vacuum and natural frequencies in fluids is obtained:

$$\omega_{Fmn} = \frac{\omega_{mn}}{\sqrt{1+\beta_{mn}}} = \frac{\omega_{mn}}{\sqrt{1+\Gamma_{mn}\dfrac{\rho_F}{\rho_P}\dfrac{r}{h}}} \tag{3}$$

where: Γ_{mn} – nondimensional added virtual mass incremental factor tabulated in tab. 3; ρ_p – density of plate material, kgm^{-3}; ρ_F – fluid density, kgm^{-3}; h - plate thickness, m, r – plate radius, m.

The circular frequency, ω_{mn} of the "dry" plate with piezoceramic can be obtained from:

$$\omega = \omega_{mn} = \frac{\lambda_{mn}^2}{r_0^2}\sqrt{\frac{D_s}{\rho_p h}} \qquad m = 0,1,2,3,\ldots\ldots, \qquad n = 0,1,2,3, \tag{4}$$

where: D_s – plate (with piezoelements) bending stiffness, λ_{mn} – frequency parameter tabulated in tab. 3.

Table 3. λ_{mn} and Γ_{mn} factors for a clamped plates

	mode n	m = 0	m = 1	m = 2	m = 3	m = 4
λ^2_{mn} [13]	0	10.216	39.771	89.104	158.183	247.005
	1	21.26	60.84	199.96	297.005	416.20
Γ_{mn} [3]	0	0.65381	0.29883	0.20834	0.16321	0.13525
	1	0.27613	0.16914	0.13123	0.10968	0.09496

Table 4. Resonance frequencies, single plate h_1 = 1 mm

Resonance frequencies, Hz						
	mode n/m	0	1	2	3	4
plate – eq .4	0	446,9	929,5	1524	2234	3048
	1	1738	2659	3700	4858	6127
plate – ANSYS	0	447,3	932,8	1533	2246	x
	1	1750	2685	x	x	x
error %	0	0,1	0,4	0,6	0,5	x
	1	0,7	1,0	x	x	x

Table 5. Resonance frequencies, single plate h_1 = 1 mm, contact with fluid on one side

Resonance frequencies, Hz						
	mode n/m	0	1	2	3	4
plate – contact with fluid on one side (eq. 3and 4)	0	102,2	304,8	584,9	949,9	1 39,99
	1	590,5	1 113,9	1 716,8	2 414,8	3 212,3
plate contact with fluid on one side – ANSYS	0	108,1	313,10	608,3	943,9	x
	1	612,4	x	x	x	x
error %	0	5,8	2,7	3,99	0,6	x
	1	3,7	x	x	x	x

The harmonic analysis covers the acoustic radiation due to steady-state plate vibrations for the two modes of resonance vibrations. Each mode was examined individually. The plate was actuated by actuator P1 and piezoelements P3, P4P6 and P3P7 were used to control the plate vibration. The remaining piezoelements (e.g. P2, P5, P8) were used as sensors (fig. 3). The optimization of voltage values utilizes the tool sub problem method available in the package Ansys.

For the reduction of the acoustic pressure level in the control volumes it was assumed, that the parameter of minimization is the averaged value of the square powered normal velocity on the surface of the panel. It is algebraically convenient to define a cost function – a quadratic function of the response, to simplify the optimization problem. Accordingly, the cost function is written as:

$$J = \min \sum_{i=1}^{n} \frac{|V_i|^2}{n} = \frac{4 \cdot \pi^2 \cdot f^2}{n} \cdot \sum_{i=1}^{n} A_i^2, \qquad \left[\frac{m^2}{s^2}\right] \tag{5}$$

Some numerical results are presented on fig 4 - 6 and in table 6.

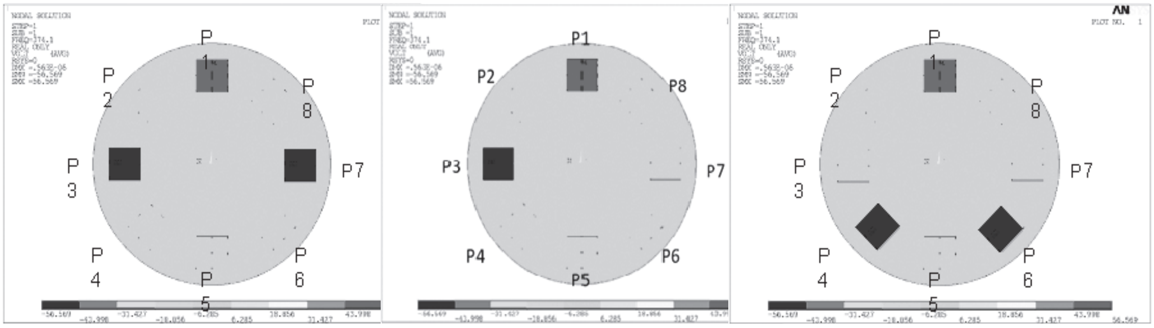

Fig. 3. Plate with piezoelements: P1 actuator, applied voltage VRMS = 40 V; P3 control elements: P3, P4, P6, P7; sensor: elements P2, P5, P8

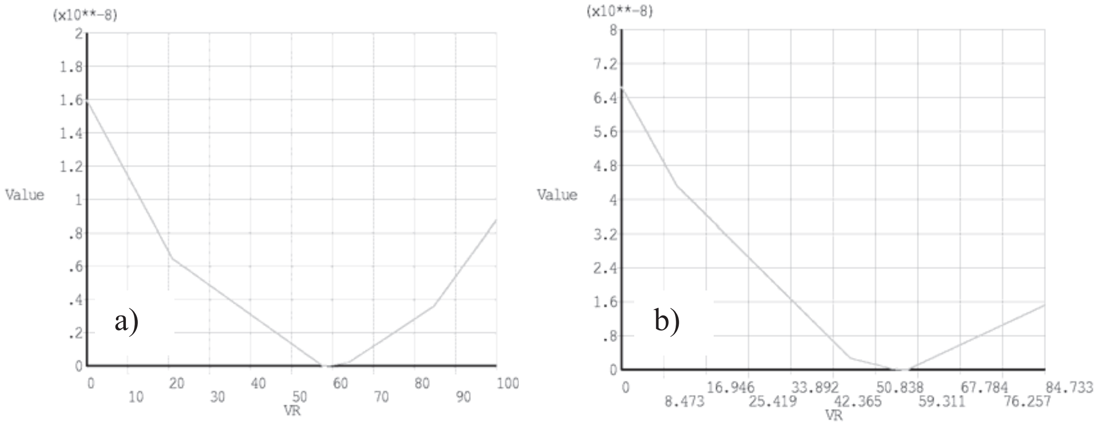

Fig. 4. Optimisation procedure: the cost function in each iteration (applied voltage to P3, a) h_1= 1mm, mode (0,0), b) h_2= 2mm, mode (0,0)

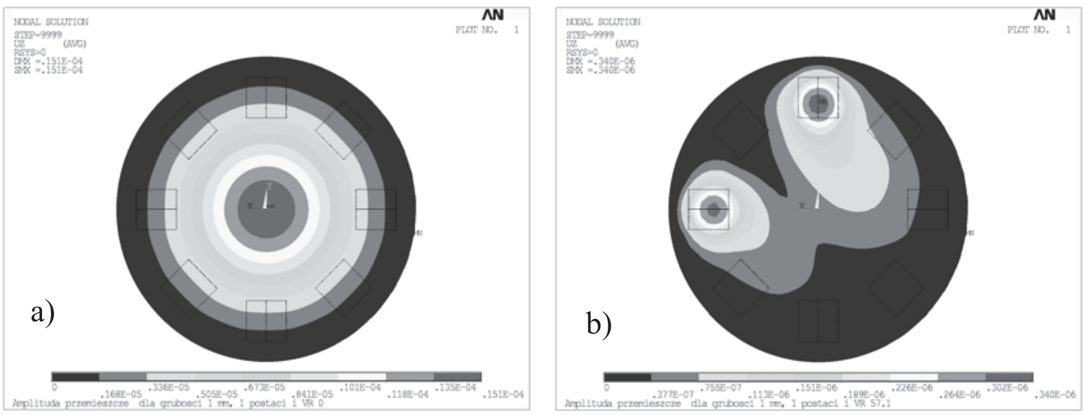

Fig. 5. Displacement h_1 = 1mm, mode (0,0): a) without control, b) with control element P3

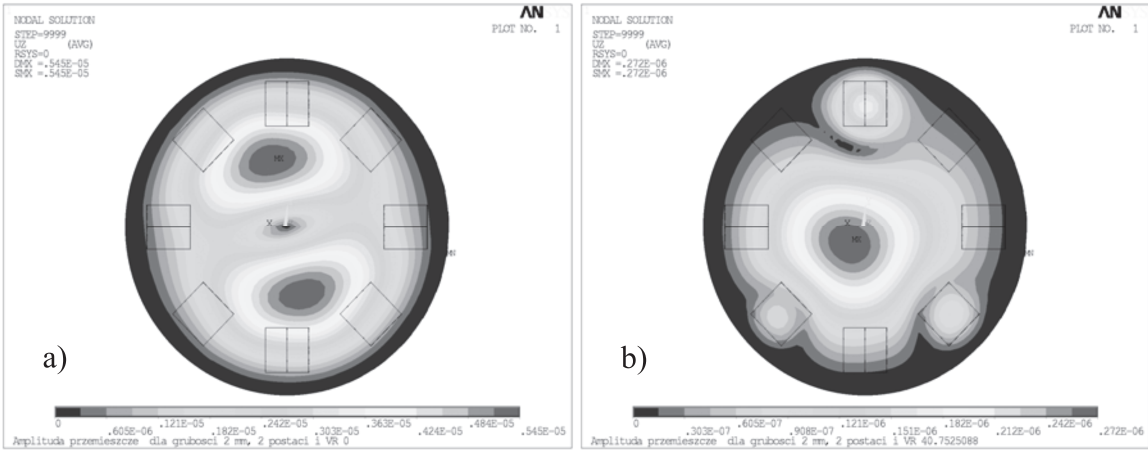

Fig. 6. Displacement h_1 = 1mm, mode (1,0): a) without control, b) with control element P4P7

Table 6. Displacement level and sound pressure level reduction in control volumes (control element P3)

mode	Sensors, dB		Control volumes, dB			Control volume - CVA dB
	P6	P7	CV3	CV6	CV9	
h_1 = 1mm, (0,0):	24.4	25.3	48.5	48.5	48.5	25.3
h_1 = 1mm, (1,0):	2.2	0.7	1.5	4.1	4.1	0.7
h_2 = 2 mm, (1,0):	28.3	25.4	56.8	56.8	56.8	28.2
h_2 = 2 mm, (1,0):	3.4	0.9	4.2	5.0	5.4	1.5

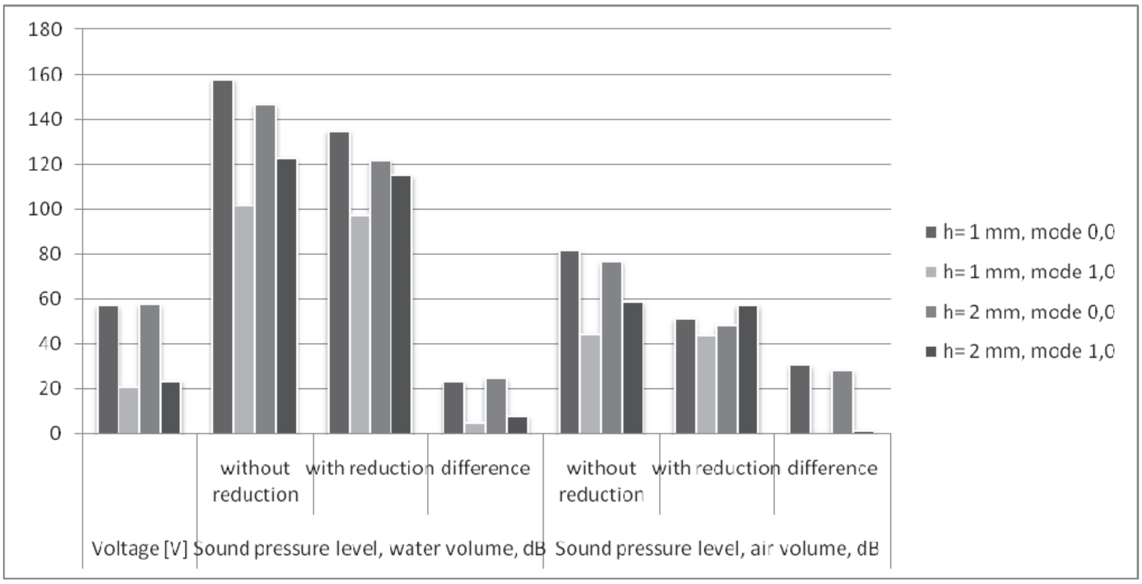

Fig. 7. Sound pressure level in whole water and air volumes (air – p_{ref} = 20 μPa, water – p_{ref} = 1 μPa, control element P3)

In the case of vibration damping for an individual resonant frequency, the displacement response reduction was observed from 0.6 dB up to 25.3 dB, depending on the resonance frequency. For considered resonance frequencies the active treatment resulted in control volumes: in 0.6–48.4 dB reduction of sound pressure level in water and in 1–56 dB reduction of sound pressure level in air.

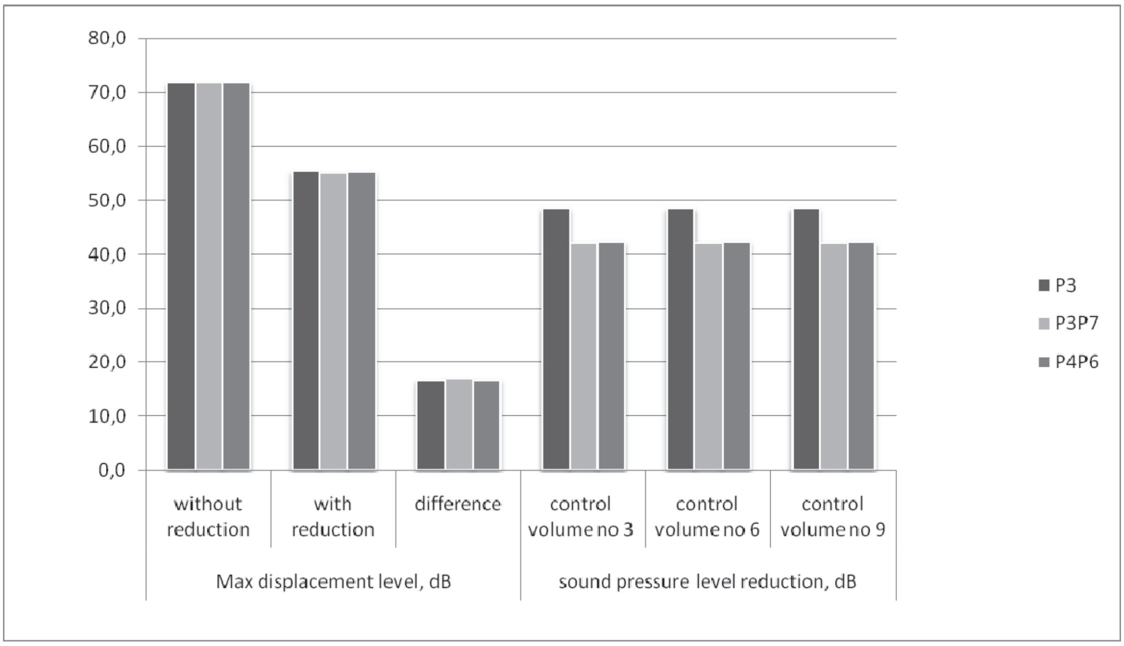

Fig. 8. Sound pressure level in whole water and air volumes (p_{ref} = 1 µPa, control elements: P3, P3P7, P4P6)

With regards to the influence of actuators' position on reduction of plate vibrations amplitude for the fixed position of the exciting force - piezoelement P1, the following conclusions can be drawn:

– the most significant amplitude reductions are reported for the first resonance frequency when the averaged squared normal velocity is reduced about 18 dB;
– the averaged sound pressure level reduction was reduced 48.5, 43.2 and 43.2 for activated P3, P3P7, P4P6 elements and for the precisely controlled voltage.

3. CONCLUSIONS

In this work, a smart circular plate with piezoelements was proposed and structural vibration control performance was evaluated. Modal characteristics, such as natural frequencies and mode shapes were obtained using the finite element method.

The finite element method and the non dimensionalized added virtual mass incremental factors for circular plates placed on an aperture of an finite rigid wall and in contact with a fluid on one side have been used to obtain theoretical natural frequencies.

The difference between the results obtained by means of the theoretical models and the numerical models are approximately 1% for single plate model and 5.8% for the plate with piezoelements and fluid contact on one side. That means that the proposed FEM method and thus developed models can be used for analysis of such systems before investigation measurements.

A suitably designed actuator could indeed perform very effectively in reducing acoustic transmissions through plate to the water and to the air. The geometry and placement of the actuators allowed it to couple well with the plate's vibration modes.

It has been shown that structural acoustic was effectively suppressed by applying proper control input voltages to the PIC actuator. A suitably designed actuator could indeed perform very effectively in reducing acoustic transmissions through plate to the water and to the air.

The geometry and placement of the actuators allowed it to couple well with the plate's first vibration mode.

The proposed method in this paper is readily applicable to the determinations of the natural frequencies of structural systems and can be adopted to large surface elements for structural acoustic noise control in a fluid.

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