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ACTIVE CONTROL OF VIBRATION AND RADIATED SOUND OF A SUBMERGED CIRCULAR PLATE

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Abstract

Complex analysis of characteristic of underwater noise allows to find several features typical for underwater noise produced by ships as well as correlations between mechanical activity of ship's mechanism and generated underwater sound. Their vibration activity might be reduced by applying active treatments utilising piezoelectric sensors and actuators. The paper explores the potentials of active control of vibrations and sound through the application of a thin round plate clamped on the edges in a rectangular enclosure. The plate is loaded on one side by heavy fluid (water) and on the other side has contact with an air. On the side of the gaseous medium, eight piezoelectric elements are bonded to the plate with a thin layer of glue.
Measurements of the acoustic pressure were taken over the whole length of the aquarium, for the fixed position of a hydrophone in the vertical and for variable height yet in a fixed distance from the radiation source. The finite element method analysis was performed, with the application of ANSYS computer package. The numerical solutions were confirmed experimentally.

1. INTRODUCTION

It is generally known that liquid motion is generated by the vibration of structures in contact with liquid and complex analysis of characteristic of underwater noise allows to find several features typical for underwater noise produced by ships as well as correlations between mechanical activity of ship's mechanism and generated underwater sound [7], [8].The contribution of total energy that is radiated by the moving ship is partially connected with the ships hull vibration due to mechanical activity of machinery inside of the hull. The mechanical vibrations that are transmitted through the hull to the water generates both broadband and narrowband noise [9].

One approach to control the noise radiated from vibrating structures is structural acoustic noise control where the motion of the structure is controlled so the radiated noise could be minimized [6], [17], [18], [20].

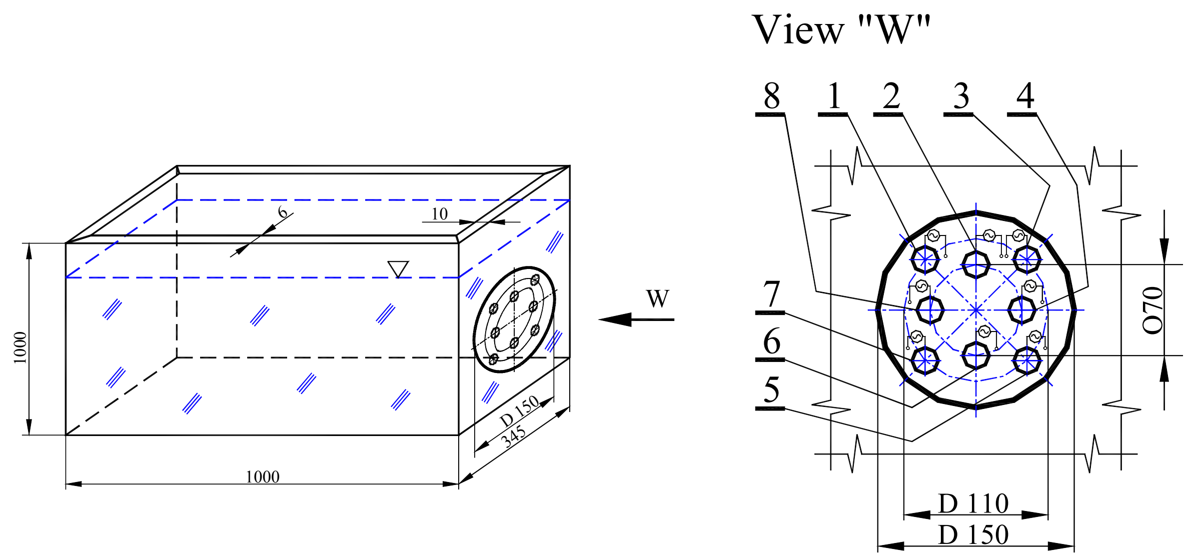
Using piezoelectric actuators, mounted on the external surface of a vibrating structure, it is possible to alter the dynamic characteristic of the whole system, thereby to control the system's response. Dimitriadis et al developed a detailed model that characterize the interaction between the piezoelectric material and the structure so that they could investigate the use of piezoelectric actuators to reduce the sound pressure radiated by thin rectangular plates [5] and circular plates [4]. They concluded that these types of actuators show great promise for controlling the vibration in distributed systems and subsequently the control of sound radiation. It was shown 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. [13], Tylikowski [16] and Sekuori et al. [15]. Niekerk et al. presented a comprehensive static model for a circular actuator and a coupled circular plate. Their static results were used to predict the dynamic behavior of the coupled system, particularly to reduce acoustic transmissions. Sekouri et al. presented an analytical approach for modelling of circular plate containing distributed piezoelectric actuators under static as well as dynamic mechanical and electrical loadings. Recently, Pan et al. [14] developed a control strategy for a large submerged cylinder by using a Tee-sectioned circumferential stiffener and pairs of PZT stack actuators driven out of phase to produce a control moment.

The experiment reported on this paper is a part of research project connected with fluid-structure interaction problems [19]. The paper explores the potentials of active control of vibrations and sound through the application of a thin round plate supported on the edges in a rectangular enclosure. The plate is loaded on one side by heavy fluid (sweet 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. Piezoelectric elements are arranged in sets, each containing four elements located on two concentric circles with different radii.

2. EXPERIMENTAL AND FEM MODEL

The structure chosen for investigation is a thin, circular plate of radius ϕ = 0,15 m, thickness h= 0,21 mm. The plate is clamped along its edge by finite rigid co-planer baffle. The plate is loaded on one side by heavy fluid (sweet 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. Piezoelectric elements are arranged in sets (fig. 1), each containing four elements located on two concentric circles with different radii and thickness (h_1= 0.21mm and h_2= 0.28 mm). Only the same type of piezoelectric have been used during single investigation. The geometrical model of system circular plate-piezo-ceramics-aquarium is presented on fig. 1 and the properties of the plate and piezoceramics are summarized in table 1.

FEM analysis of plate vibrations was performed using the Ansys package [1]. Shell element shell93 and coupled fields element (structure – piezoelectric) solid226 were chosen. These elements are 8-node and 20-node elements, improving the calculation accuracy in relation to 4-node shell63 and 8-node solid5 elements, for the same grid density. Piezoelectric layers were modelled by four layers of finite elements. The distance between piezoelectric plane (solid element) and plate middle surface (shell element) were taken into consideration using rigid region and constrain equations. The layer of adhesive agent was not considered in the analysis. In the plate-acoustic space model structural sounds produced by the vibrating plate were radiated to the water space. 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.



1,3,5,7 piezoelements type 1, thickness h_1=0,28 mm, 2,4,6,8 piezoelements type 1, thickness h_2=0,21 mm, marked: A2, A4, A6, A8

Fig. 1. Geometry of aquarium and plate system

Table 1. Material properties of the experimental plate and piezoceramic

Piezoceramic P7		Steel	
Density [$kg \cdot m^{-3}$]	ρ 7800	Density [$kg \cdot m^{-3}$]	ρ 7820
Elastic constant [10^{-12} m^2 N^{-1}]	S_{11}=15.8, S_{12}= –5.7 S_{13}= –7.0, S_{33}= 18.1 S_{44}= 40.6, S_{66}= 43.0	Elasticity modulus [Pa] Poisson Ratio	E $2.1 \cdot 10^{11}$ υ 0.29
Charge constants [10^{-12} $m \cdot V^{-1}$]	d_{31}=–207·, d_{33}=410, d_{51} = 550		
Relative permitivity	$\varepsilon_{11}/\varepsilon_0$ 1930 $\varepsilon_{33}/\varepsilon_0$ 2100		

Sound radiated by the vibrating plate is determined in the 0.4 m semi-sphere (fig. 2) of air surrounding the plate. A discretisation procedure was applied whereby the acoustic volume should comprise nearly 44 thousand fluid30 elements (4-node tetrahedrons) and infinite fluid130 elements on the external surface of the sphere. The grid is denser in the vicinity of the plate. Underlying the model is the assumption that there should be at least six grid elements per the considered wavelength. 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 5 x 10^{-3} [-] for the whole system.

Values of sound pressure level were calculated at six control volumes (diameter 0,04 m) in water along the aquarium and at one in volume (diameter 0.04 m) in air at 0.3 m distance from the plate surface. (fig. 2)

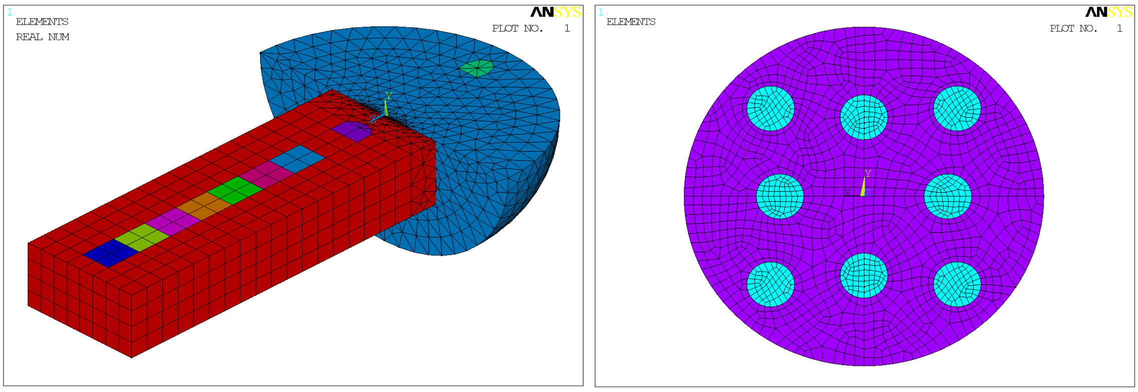


Fig. 2. A half of the aquarium and plate divided into finite elements

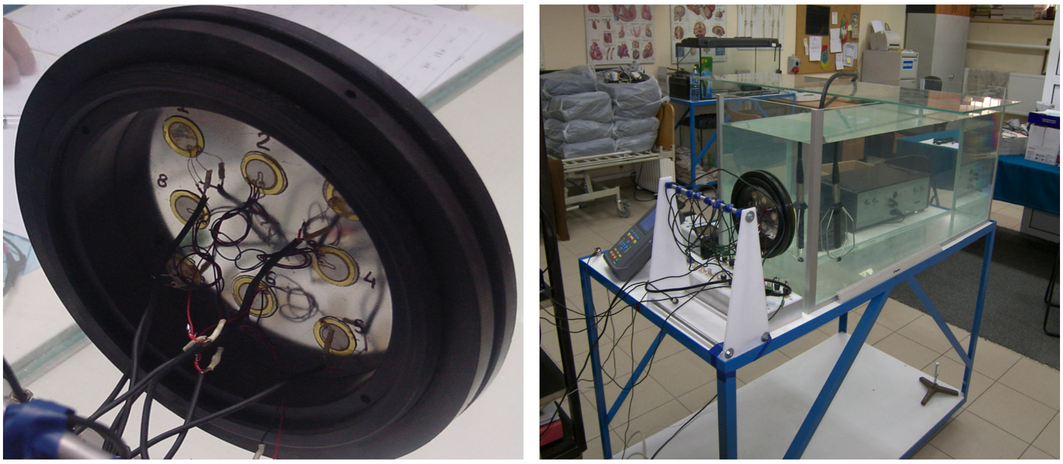


Fig. 3. The view of experimental aquarium and plate with piezoelements

3. NUMERICAL AND EXPERIMENTAL RESULTS

It is generally known, that the mode shapes of the plate vibrating in contact with fluid are assumed to be equal to those of plate vibrating in a vacuum, so frequencies of free vibration in fluid can be related to the natural frequencies in a vacuum [2], [10], [12]. When using Rayleigh quotient for coupled vibrations accounts for the square of the natural frequencies of the plate in vacuum ω^2_{mn} are proportional to the ratio between the maximum potential energy of plate U_{mn} and its reference kinetic energy T^*_{mn}, and also the for the square of the natural frequencies of the plate in fluid ω^2_{Fmn} is proportional to the ratio between the maximum potential energy of plate U_{mn} and the sum of the reference kinetic energies of both the plate T^*_{mn} and the fluid T^*_F. According to the assumption, the energies are not changed when evaluated in vacuum or in fluid, the following relations between natural frequencies in vacuum and natural frequencies in fluids is obtained:

$$\omega^2_{mn} \approx \left(\frac{U_{mn}}{T^*_{mn}}\right)_{air} \tag{1}$$

$$\omega^2_{Fmn} \approx \left(\frac{U_{mn}}{T^*_{mn} + T^*_F}\right)_{fluid} \tag{2}$$

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

where: Γ_{mn} – nondimensional added virtual mass incremental factor tabulated in [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:

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

where: D_s - plate (with piezoelements) bending stiffness, λ_{mn} - frequency parameter tabulated in [11]

Table 2. Resonance frequencies of the plate with piezoelements

Resonance frequencies, Hz					
mode	plate with piezo in air		plate with piezo contact with fluid on one side		
	Ansys	experiment	Ansys * eq (3)	Ansys + water	experiment
(0,0)	95,0	102,0	17,1	19,9	18,0
(1,0)	197,5	213,2	53,5	62,6	49,0
(2,0)	324,1	313,3	110,9	125,9	104,2
(0,1)	369,5	370,1	96,6	138,0	111,0
(3,0)	482,6	474,7	192,8	198,0	172,8
(1,1)	565,2	571,1	191,3	246,0	191,3
(4,0)	647,4	648,1	----	318,0	254,7
(2,1)	785,9	796,6	313,3	322,0	304,1
(0,2)	827,9	----	-----	414,0	255,3
(5,0)	843,2	851,5	-----	432,0	462,1

The harmonic analysis covers the acoustic radiation due to steady-state plate vibrations for the eight modes of resonance vibrations. Each mode was examined individually. The plate was actuated by single P9 – actuator (marked A2), while the remaining actuators (marked A4, A46, A48) were used to control plate vibrations.

Then the plate was actuated by next actuator and the remaining actuators (in similar configuration) were used to control plate vibrations. All measurements were realized seven times and than all data were averaged. Measurements of the acoustic pressure were taken over the whole length of the aquarium, for the fixed position of a hydrophone in the vertical and for variable height yet in a fixed distance from the radiation source. The effects of a vibrating plate on the acoustic pressure levels outside the aquarium were investigated, too.

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 (5).

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

In order to obtain a relatively minimal value of the cost function, the value of voltage amplitude for the first seven modes control was precisely controlled. The optimization of voltage values utilizes the tool available in the package Ansys. Some numerical results are presented on fig 4a -4c.

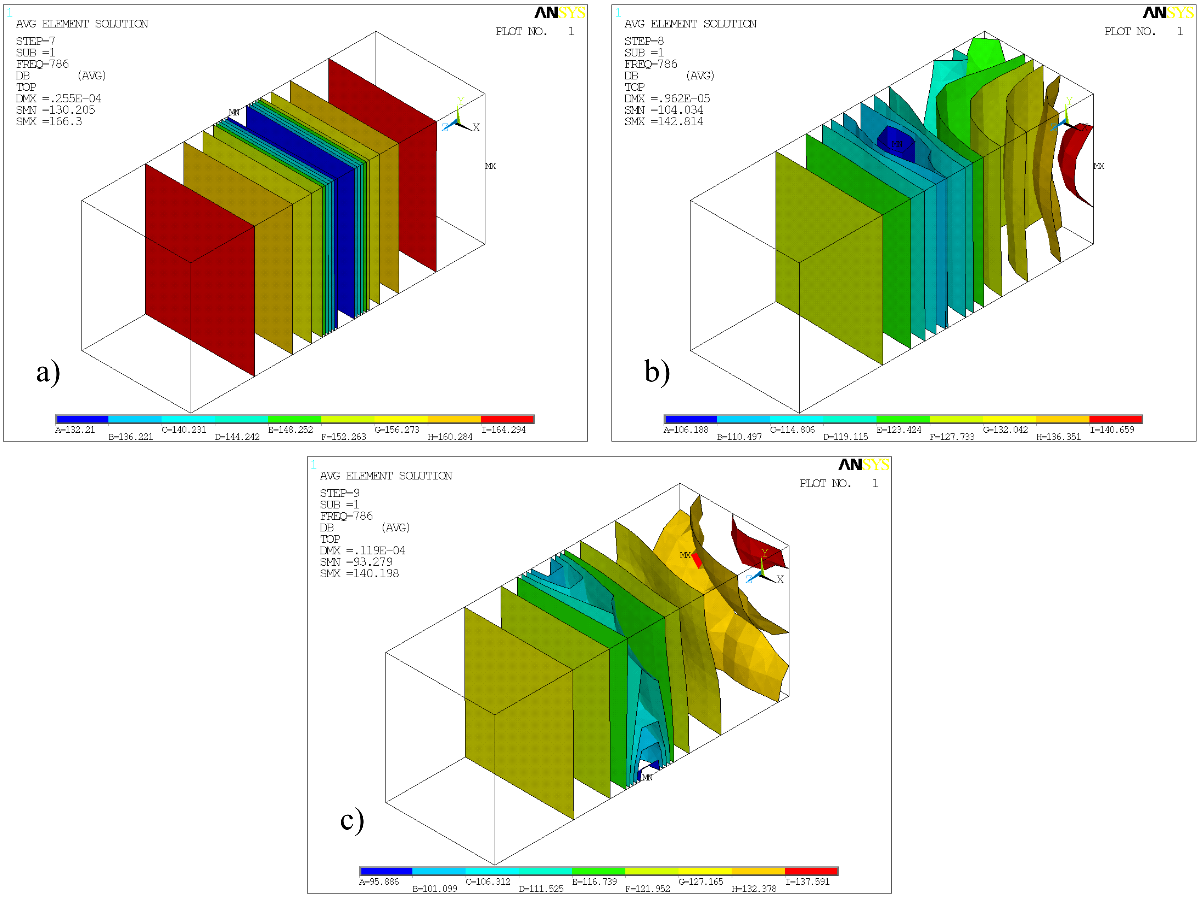


Fig. 4. Isosurfaces, SPL in water, frequency 786 Hz: a) without reduction, b) actuated A4, c) actuated A46

Measurements of the acoustic pressure were taken over the whole length of the aquarium, for the fixed position of a hydrophone in the vertical and for variable height yet in a fixed distance from the radiation source. The effects of a vibrating plate on the acoustic pressure levels outside the aquarium were investigated, too.

Table 3. Reduction of vibration level (sensor A6, A8) and SPL reduction

Frequency	Reduction of vibration level, dB		SPL reduction, dB	
Hz	A4	A8	water	air
172,8	16,2	16,5	16,2	4,7
254,7	14,9	14,4	15,2	1,3
304,1	5,8	5,5	5,3	0,8
414,0	21,1	20,5	27,8	22,4
455,5	24,3	25,1	24,8	20,5

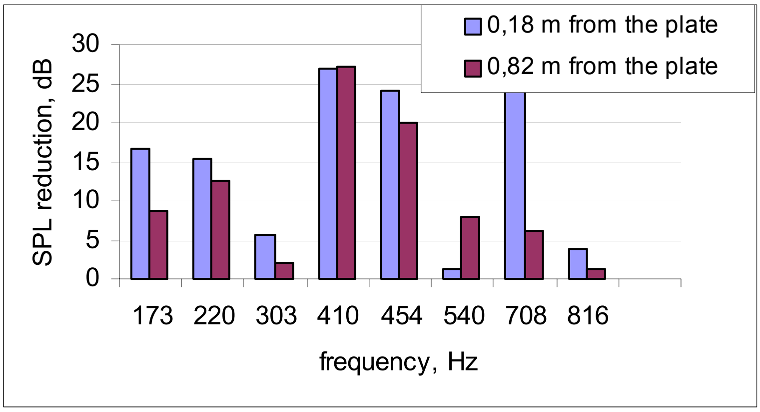


Fig. 5. Sound pressure level reduction at the distance: 0,18 m and 0,82 m from the plate

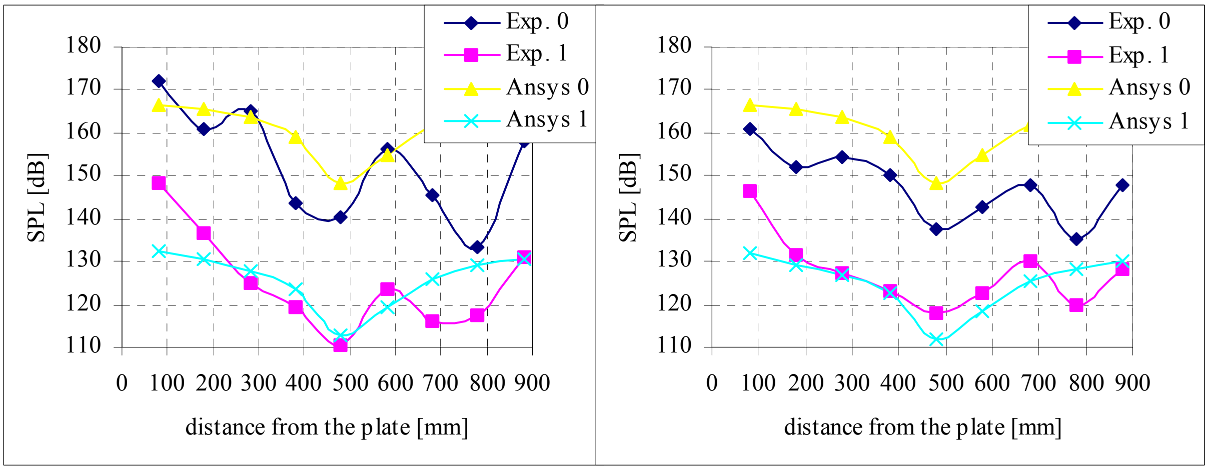


Fig. 6. Sound pressure level over the whole length of the aquarium, frequency 764 Hz, 0 – without reduction, 1 – with active reduction, a) actuated A4, b) actuated A4 and A6

In the case of vibration damping for an individual resonant frequency, the displacement response reduction was observed from 5.5 dB up to 25 dB, depending on the resonance frequency. For all considered resonance frequencies, the active treatment resulted in 3 - 28 dB reduction of sound pressure level in water. The responses at six control volumes appear to have the same response profile but differ in magnitude. Depending on the frequency application of two actuators instead of one increase the reduction of SPL in water by about 2 do 10 dB.

Activation of the piezoelectric elements changing the acoustic field in the water, in the water has arisen vortexes. During these activations it is also possible to find a local minimum. In the frequency 462 Hz this phenomena occur at the distance 180 mm from the plate.
In the frequency 764 Hz, in the aquarium has arisen the standing waves filed, and this the reason of changing of SPL values. The characteristic point of standing waves (local minimum) is located at the distance 480 mm from the vibrating plate.

5. CONCLUSIONS

The paper is concerned with the problem of activity attenuation of plate vibration in contact with fluid. The aim of this work was to investigate how effective a distribution of actuators could control the vibration and sound transmission through a flexible plate.

The geometry and placement of the actuators couple well with the plate's vibration modes with fixed point vibration generation. Accuracy of bonding of piezoceramic elements can greatly influence on electromechanical effectiveness of the piezoelements. Some resonance frequencies errors can be attributed to potential de-bonding

It was also shown that major factors affecting the vibration and sound pressure reduction performance include the shape and actual configuration of piezoelectric elements.

For all considered resonance frequencies and placement of the actuators, the average sound pressure level in the control volumes was significantly reduced in the water and in the air.

During measurements, it was not noticed, any changing of sound pressure level in water for variable height yet in a fixed distance from the radiation source. This is probably connected with small vertical dimension of the aquarium.

The experimental results show that the proposed model can be adopted to large surface elements for structural acoustic noise control in a fluid.

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