Sound Radiation of a Submerged Plate With Four Square Piezoelectric Actuators

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Active noise control by means of the smart structure is one of effective method in this subject. It can be used to decrease the flow of energy into the water medium. This study is a continuation of earlier works by the author and explores the possible effects of reducing the amplitude of plate vibrations and of structural sound reduction through the use of piezoelectric actuators in an anti-symmetric configuration. The vibrating element is a steel plate with glued-on actuators, supported on all edges and loaded on one side by heavy fluid (sea water). The plate is excited by a harmonically variable, concentrated force with a constant amplitude value. The first order approximation method is used to obtain optimal voltage on each actuators. Sound radiation of the vibrating plate is calculated within an fluid volume surrounding the plate and in contact with it by a fluid–structure zone. The analysis uses the finite element modelling for structural vibrations and sound radiation.

1 Introduction

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.

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. Ship is a combined underwater source, it is a broadband source with well recognized discrete components[7]. One of the underwater noise sources is machinery noise originates as mechanical vibrations that are transmitted through the hull to the water, contributed to radiated noise. Machinery noise generates both broadband and narrowband noise [5].

One approach to control the noise radiated from vibrating structures is active noise control where amplifiers and loudspeakers are used to generate the signal out of phase with respect to the primary noise signal. Another approach involves the structural acoustic control where the motion of the structure is controlled so the radiated noise could be minimized [1],[3]. Among the smart materials [4],[12] piezoelectric materials are most frequently applied to structural vibration damping and suppression of noise generated by these vibrations[11][13] [15][16]

This study is a continuation of earlier works[8][9] by the author and explores the possible effects of reducing the amplitude of plate vibrations and of structural sound reduction through the use of piezoelectric actuators in an anti-symmetric configuration.

2 FEM model

The vibrating element is a steel plate 200x200x2 [mm] with glued-on actuators, supported on all edges and loaded on one side by liquid (sea water, density ρ=1005 [kg/m^3], sound velocity c=1475 [m/s]).

On its two external surfaces (see fig.1) were four pairs of piezoelectric actuators, integrated with the plate. The external excitation has the form of a point force harmonically variable in time, with an amplitude 1 [N]. It is applied at the point of position xyz = 90x90x0 [mm]. The coordinate system is chosen such that its origin is located in one fixed corned of the plate, the X and Y axes coincide with the plate edges and the Z-axis is perpendicular to the plate surface. The coordinates of all points on the plate are positive.

Actuators 20 x 20 x 1 [mm] made from PZT-4 material [14] are mounted symmetrically on the plate and driven 180 deg out of phase with respect to the input signal. The modal damping coefficient for the whole structure is assumed 0.0005. Material properties of the plate and actuators are summarised below in Table 1.

Table 1: Material Properties Piezoceramic and Steel.

Material	Piezoceramic PZT 4	Steel
Density	7500 kg/m^3	7820 kg/m^3
Poisson Ratio	0.29	0.29
Damping Ratio	0.0003	0.0003
Permittivity in X direction	7.124 E-9	Elasticity Modulus 2.07E11 Pa
Permittivity in Y direction	7.124 E-9	
Permittivity in Z direction	5.841 E-9	

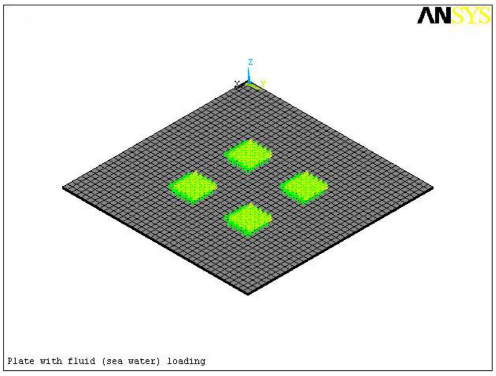

Figure 1: FEM model of a plate with four pairs of piezoelectric actuators

FEM analysis of plate vibrations was performed using the ANSYS package. Solid elements for coupled fields SOLID45 and SOLID5 were chosen. Piezoelectric layers and the steel plate were modelled by four layers of finite elements. The layer of adhesive agent was not considered in the analysis. The FEM structure and the points of application of the external force are shown in Fig 1.

3 Numerical analysis

The analysis covers the acoustic radiation due to steady-state plate vibrations for the first seven modes of natural vibrations. Each mode was examined individually.

Calculations of the pressure level at chosen control points are based on the acoustic field - structure interaction module available in the ANSYS package.

For the simplifying assumptions made[1][6] the liquid momentum equations yield the following relationships between the normal pressure gradient of the liquid and the normal acceleration of the structure at the liquid – structure interface.

The discretized wave equation accounting for losses at the interface is given by (1):

$$[M_e^P]\{\ddot{P}_e\} + [C_e^P]\{\dot{P}_e\} + [K_e^P]\{P_e\} + \rho_0 [R_e]^T \{\ddot{u}_e\} = \{0\} \qquad (1)$$

The dynamic element equation of the structure with the liquid pressure load acting at the interface is described by (2)

$$[M_e]\{\ddot{u}_e\} + [C_e]\{\dot{u}_e\} + [K_e]\{u_e\} - [R_e]\{P_e\} = \{F_e\} \qquad (2)$$

$[M_e^P]$, $[M_e]$ – 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, $\rho_0[R_e]$ – coupling mass matrix, ρ_0 – mean liquid density, $[F_e]$ – applied load vector $\{u_e\}$ – nodal displacement component vectors, $\{p_e\}$ – nodal pressure vectors

For harmonic analyses, since both pressure and displacements are harmonic function of time, the following relationships are hold:

$$\dot{u} = j u_0 \omega e^{j\omega t}; \quad \dot{p} = j p_0 \omega e^{j\omega t}; \quad [M] = j\omega [C]$$
$$\ddot{u} = -u_0^2 \omega^2 e^{j\omega t}; \quad \ddot{p} = -p_0^2 \omega^2 e^{j\omega t}; \quad [K] = \frac{[C]}{j\omega} \qquad (3)$$

Based on the formula (1) and (2) the amplitude of pressure in the control distanced 1 m above the external surface of the panel with piezoelectric strips.

In this method structural sounds produced by the vibrating plate were radiated to the space 2 x 2 x 2[m]. Absorbing material with the sound absorption ratio 1.0 is placed on five external surfaces bounding the acoustic volume. In the plane XY, in which plate vibrations occur, the material with low absorption ratio 0.01 is placed.

The acoustic volume is modelled by solid elements of the type FLUID30. They are used for modelling liquid media and interfaces in liquid-structure interactions. The element has eight corner nodes and four dofs per node: translations in the nodal directions x,y,z, and pressure. The translations, however, are applicable only at the interface nodes. The system divided into FEM elements and the vibrating plate positions are shown in Fig. 2 and 3.

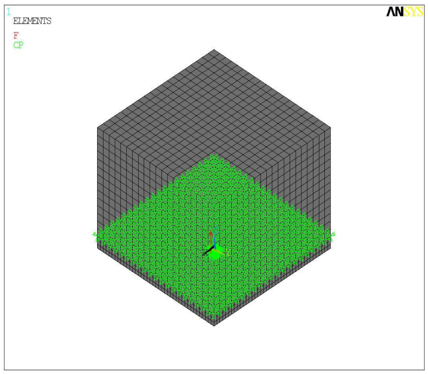

Figure 2: FEM model of the system

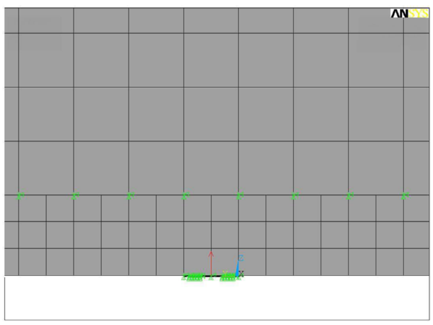

Figure 3: FEM elements of the attached volume

It is algebraically convenient to define a cost function- a quadratic function of the response, to simplify the optimisation problem. Accordingly, the cost function is written as (1).

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

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 optimisation of voltage values utilises the tool available in the package Ansys. The first order approximation method was applied, in which resistance was taken as the state variable and the objective function was written as (4).

4 Results

The calculation procedure was run for single node optimal values of harmonically varying voltages applied to piezoelectric actuator surfaces and for multi-mode optimal voltage value. Results expressed as displacement and sound pressures for the given vibration modes for single-mode optimisation are compiled in Table 2 and 3 and for multimode optimisation ate presented in Fig. 4 and 5.

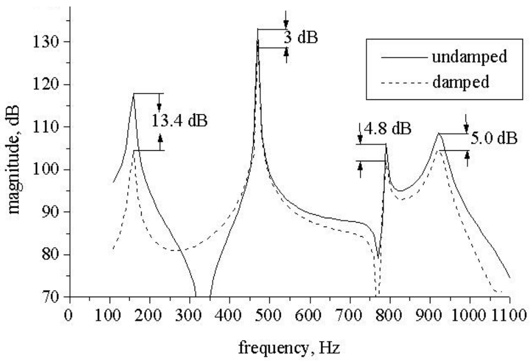

Figure 4: Displacement on the plate vs excitation frequency for the on-state and off-state controller

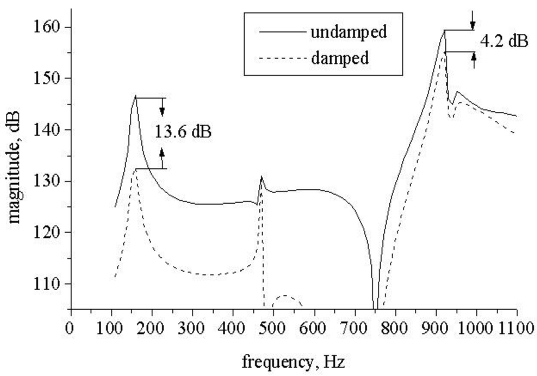

Figure 5: Sound pressure level vs excitation frequency for the on-state and off-state controller

Table 2: Estimated displacement for the off-state and single-mode on-state controller

Mode No.	Freq. [Hz]	Displacement [dB]	
		un-damped	damped
1	157	122,0	82,0
2	471	129,4	92,6
3	787	114,0	113,2
4	923	108,3	91,5
5	1318,5	117,2	113,0
6	1778	81,2	69,9
7	1948	104,0	100,0

Table 3: Estimated sound pressure level for the point xyz = 0.0,0.0,1.072 [m] for the off-state and single-mode on-state controller

Mode No.	Freq. [Hz]	Sound pressure level [dB]	
		un-damped	damped
1	157	155,0	106,8
2	471	135,6	122,7
3	787	130,0	127,5
4	923	160,9	136,7
5	1318,5	145,9	137,6
6	1778	143,4	124,1
7	1948	151,3	140,6

In the case of vibration damping for an individual resonant frequency, the displacement response reduction was observed from 4 dB up to 40 dB dB and 36 dB for the first and second mode. In the case of multiple mode damping, the respective response reductions were from 3,0 to 13,4 dB for the first, second, third and fourth resonant frequency.

In the case of vibration damping for an individual resonant frequency, the sound pressure response reduction was observed from 10 dB up to 48 dB, for the first free odd modes. In the case of multiple mode damping, the respective response reductions were: 13,6 dB, 4,2 dB and 3,8 dB, for the first free odd modes.

5 CONCLUSIONS

1. The numerical experiment have shown the effectiveness of the proposed piezoelectric actuators both in single modal and multi modal control of smart structures. Structural vibrations

and hence the level of sound radiation to the sea water or to the air, can be reduced through the application.

2. It was shown that major factors affecting the vibration reduction performance include the shape and actual configuration of piezoelectric elements and 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.
3. For good efficiency the shape of piezoelectric elements should be dedicated to the primary interested mode.

REFERENCES

[1] Ansys user manual

[2] S. P. Elliott, P.A. Nelson, 'Active Control of Vibrations'. Academic Press. London, 1997.

[3] C.R. Fuller, 'Active Control of Sound Trasmission/ Radiation From Elastic Paltes by Vibration Inputs: I Analysis' J. of Sound and Vibration (1990), 136(1), 1-15.

[4] Giurgiutiu V., Craig A.R., 'Power and Energy Characteristics of Solid State Induced-Strain Actuators for Static and Dynamic Applications' J. of Intelligent Material Systems and Structures, Vol. 8. pp. 738-750 (1997)

[5] I. Gloza, G. Grelowska, 'Reduction of Ship's Vibration as an Element of Protection of the Sea Environment from the Noise Pollution' Polish Journal of Environmental Studies, Vol. 13. Supplement III. pp. 92-95 (2004)

[6] S. I. Hajek, 'Encyclopedia of Vibarion', Ed. S. Braun, D. Evins, S.S. Rao, Academic Press pp. 544 – 553 (2002)

[7] E. Kozaczka, G. Grelowska, 'Shipping noise' Archives of Acoustics, Vol.29 (2). 169–178 (2004)

[8] M. S. Kozień, J. Wiciak, 'Acoustics Radiation of a Plate With Line and Cross Type Piezoelectric Elements' Molecular and Quantum Acoustics, Vol.24. pp.97-108 (2003)

[9] M.S. Kozień, J. Wiciak, 'Influence of Configuration of Piezoelements on the level of reduction of

[10] plate Vibrations', Technical Bulletin, Cracow University of Technology No. 5-M. pp.149-156 (2004)

[11] L. Leniowska, R. Leniowski, 'Development of Active Vibration Control System for Circular Plate by Using the 2-Layer Piezo Disk System' Mechanics, Vol.22. (3) pp. 337-344, (2003)

[12] J. N. Reddy, 'On laminated composite plates with integrated sensors and actuators' Engineering Structures, Vol. 21 pp.568-593 (1999)

[13] A. Tylikowski, 'Active damping in continuous elements of construction and machine', Z. Osiński, Damping of vibrations, PWN, Warszawa, ISBN 83-01-12404-0 (1997).

[14] J. W. Waanders, 'Piezoelectric ceramics. Properties and applications', Philips Components, Eindhoven, 1991.

[15] J. Wiciak, The Influence of Position of Piezoelectric Actuators on Vibrations and Acoustics Radiation Reduction of Simply Supported Plate Polish Journal of Environmental Studies. Vol. 13. Supp. III. pp. 125-130 (2004)

[16] J. Wiciak, 'Acoustics Radiation Of A Plate Bonded On All Edges With Four Pairs Actuators', Molecular and Quantum Acoustics, Vol.25, pp.2812-290, (2004)

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