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이방성 압전 작동기를 이용한 복합재료 평판을 통한 공동(空洞) 내의 소음 억제

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Suppression of Sound Transmission through Composite Plate into Cavity with Anisotropic Piezoelectric Actuators.

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ABSTRACT

The anisotropy and shape of distributed piezopolymer actuator have advantages over isotropic piezo ceramic materials, since these features of PVDF can be utilized as another design variable in control application. This study is interested in the reduction of sound transmission through elastic plate into interior space by using the PVDF actuator. The plate-cavity system is adopted as a test problem. The vibration of composite plate and the sound fields through plate are analyzed by using the coupled finite element and boundary element method. Some numerical simulations are performed on sound transmission through elastic plates. To investigate the effects of anisotropy and shape of distributed piezopolymer actuator, various kinds of distributed PVDF actuators are applied in sound control simulation for isotropic and anisotropic plates. The PVDF actuators applied are different from each other in their shapes and laminate angles. The results of control simulation show that the control effectiveness of distributed PVDF actuator can be enhanced by using the coupling between shape of actuator and vibration modes of structure and the anisotropy of piezoelectric properties of PVDF.

1. Introduction

Sound from vibrating structures is an important problem in numerous engineering applications. The interior noise levels in aircraft cabin and the sound pressure levels within the payload compartment of launch vehicles are representative examples.[1] The major problem that faces aircraft or launch vehicle is the transmission of exhausted engine noise through fuselage sidewall structure or fairing structure. For the reduction of the interior sound levels, improved attenuation and absorption within vehicle structure are expected to be required. Such passive control techniques are likely to increase the weight of structures and not effective at lower frequencies. Therefore, considerable efforts have been devoted to active control techniques to reduce low-frequency sound transmission through elastic structures during last decades. For structurally radiated or transmitted noise, the noise field is directly coupled with structural motion. Therefore it is efficient to directly apply the control inputs to the vibrating structure for the minimization of radiated sound fields.[2]

As a result of rapid advances in smart structures, the research thrust today is toward utilizing the piezoelectric materials as distributed sensors or actuators in control applications. Polyvinylidene fluoride polymer (PVDF) has such good properties as flexibility, ruggedness, and light weight. But it is not powerful as much as piezoceramic material (PZT). Hence, compared to PZT, PVDF has usually been used as a sensor rather than an actuator. However, PVDF has anisotropic electro-mechanical properties, that is, the piezo strain constants d_{31} and d_{32} of PVDF differ by an order of magnitude. This anisotropic property of PVDF can be utilized in control application.

This paper is concerned with a numerical study on the active control of sound transmitted through a laminated composite plate into a rectangular cavity by adopting PVDF as distributed actuator. The PVDF actuator bonded to both surfaces of plate has its ply angle similar to each lamina in composite plate. The panel-cavity coupled system is a simplified model of a fluid-structure coupled phenomena such as noise transmission through aircraft fuselage into cabin.[3] Sound transmission through plate structure is highly influenced by the interaction of vibration modes of structure. The shape and directional characteristics of PVDF actuator will be used to design an efficient actuator by considering the relationship between vibration modes of composite plate and control effects from PVDF actuator.

The finite element method(FEM) is applied for the analysis of plate structure based on the first order shear deformation plate theory and the acoustic field is to be analyzed through the boundary element method (BEM) based on the Helmholtz integral equation.

In order to control global acoustic noise from the structure, the sound power is used as the objective function in control scheme. The control input is calculated so that the sound power from composite plate can be minimized, and then it is restricted with the constraint that input electric field into PVDF actuator should be less than the maximum operating field.

Some numerical simulations are carried out on the transmitted sound fields from composite plates. In order to investigate the potential use of anisotropy of composite material and piezopolymer actuator, various kinds of laminated composites and distributed PVDF actuators are applied in the control simulations.

2. Description of the Model

Figure 1 shows the plate-cavity system that is modeled in this analysis. It consists of a rectangular cavity(of width L_x, length L_y, and depth L_z) with the surface at z=0 being a rectangular composite plate.

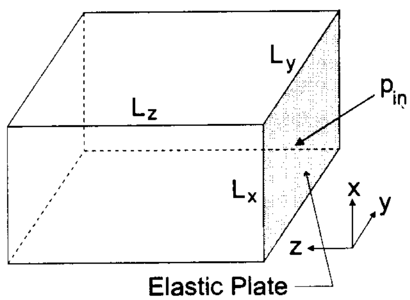


Figure 1. Configuration of plate-cavity model

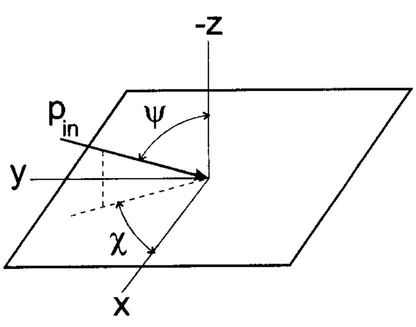


Figure 2. External plane wave incident on plate

An external sound field excites the plate, which in turn radiates sound power into the cavity. The external sound field is a plane wave whose incident angles are shown in Figure 2.

The plate structure (thickness h) contains fiber reinforced composite laminae and distributed piezoelectric actuators that are assumed to be perfectly bonded on the surface. Let the x-y plane coincide with mid-plane of the plate, with the z-axis being normal to the mid-plane. For a lamina, which is fiber reinforced composite or piezoelectric material, its material axes is denoted 1-2-3 axes. The lamination angle of ith lamina, θ_i, is defined as the angle from x-axis to 1-axis in counterclockwise direction along z-axis in xy-plane.

3. Finite Element Model for Structure

Finite element model of composite plate is based on the first order shear deformation theory. The classical plate theory has usually been applied in the analysis of sound fields from isotropic plates. However, it is well established that in the analysis of composite plates a theory that includes shear deformation is required.[4]

The displacement field $\{u_1, u_2, u_3\}$ based on a first-order shear deformation theory[5], is given by

$$\begin{aligned} u_1(x,y,z,t) &= u(x,y,t) + z\varphi_x(x,y,t) \\ u_2(x,y,z,t) &= v(x,y,t) + z\varphi_y(x,y,t) \quad (1) \\ u_3(x,y,z,t) &= w(x,y,t) \end{aligned}$$

where u, v, and w are the displacements of a point (x,y) on the mid-plane, respectively, t is time, φ_x and φ_y are the rotations of the line element, initially normal to mid-plane, about the y and x axes, respectively.

The infinitesimal strain relations give the strain written as

$$\boldsymbol{\varepsilon} = \left[\varepsilon_{xx}^o + z\kappa_{xx} \quad \varepsilon_{yy}^o + z\kappa_{yy} \quad \varepsilon_{xy}^o + z\kappa_{xy} \quad \gamma_{xz} \quad \gamma_{yz}\right]^T$$

where membrane strain $\boldsymbol{\varepsilon}^o$, bending strain $\boldsymbol{\kappa}$, and shear strain $\boldsymbol{\gamma}$.

Through coordinate transformation, the electro-mechanical constitutive relations for ith lamina can be written in x-y-z axes as

$$\boldsymbol{\sigma} = \mathbf{C}^i\boldsymbol{\varepsilon} - \mathbf{d}^i E^i \quad (2)$$

where $\boldsymbol{\sigma}$ is the stress, $\mathbf{C}^i$ is the elastic stiffness matrix, E^i is the applied electric field intensity in z-direction, and $\mathbf{d}^i$ is expressed in terms of piezoelectric strain/charge coefficients d_{31} and d_{32}.[6]

$$\boldsymbol{\sigma} = \left[\sigma_{xx} \quad \sigma_{yy} \quad \sigma_{xy} \quad \tau_{xz} \quad \tau_{yz}\right]^T$$

$$\mathbf{d}^i = \left[d_{xx}^i \quad d_{yy}^i \quad d_{xy}^i \quad d_{xz}^i \quad d_{yz}^i\right]^T$$

$$= \beta \begin{Bmatrix} \left(m_i^2 C_{11}^i + n_i^2 C_{12}^i\right)d_{31}^i + \left(n_i^2 C_{11}^i + m_i^2 C_{12}^i\right)d_{32}^i \\ \left(n_i^2 C_{11}^i + m_i^2 C_{12}^i\right)d_{31}^i + \left(m_i^2 C_{11}^i + n_i^2 C_{12}^i\right)d_{32}^i \\ m_i n_i\left(C_{11}^i - C_{12}^i\right)d_{31}^i - m_i n_i\left(C_{11}^i - C_{12}^i\right)d_{32}^i \\ 0 \\ 0 \end{Bmatrix}$$

where β is +1 for positive poling and -1 for negative poling, $m_i = \cos\theta_i$, and $n_i = \sin\theta_i$. Equation (2) is the general expression of constitutive relation for the piezoelectric material, in case of composite lamina the piezoelectric strain/charge coefficients should be zero. For isotropic PZT, of which d_{31} and d_{32} are equal, d_{xx} is always equal to d_{yy} and d_{xy} is always zero irrespectively of its lamination angle. Thus isotropic PZT cannot generate shear stress component. While the anisotropic PVDF, of which d_{31} and d_{32} are different from each other, is able to generate the shear stress component. The shear stress component produced by directionally attached anisotropic PVDF actuator can be taken advantage of effectively in the control of sound fields from plate structure.

The laminate constitutive relations are expressed as

$$\mathbf{N} = \mathbf{A}\boldsymbol{\varepsilon}^{o} + \mathbf{B}\boldsymbol{\kappa} - \overline{\mathbf{N}}$$
$$\mathbf{M} = \mathbf{B}\boldsymbol{\varepsilon}^{o} + \mathbf{D}\boldsymbol{\kappa} - \overline{\mathbf{M}} \qquad (3)$$
$$\mathbf{Q} = \mathbf{A}_S\boldsymbol{\gamma}$$

where **N** and **Q** are force resultant vectors, **M** is moment resultant vector, which are expressed in terms of inplane stresses $\{\sigma_{xx}\ \sigma_{yy}\ \tau_{xy}\}$ and transverse shear stresses $\{\tau_{xz}\ \tau_{yz}\}$, and elastic coefficient matrices are obtained from elastic stiffness matrix.

External loads acting on the plate structure are the distributed external load induced by incident sound field, the acoustic loading by the acoustic fluid adjacent to the plate surface, and the control force by the PVDF actuators.

The weak form of equation is discretized using four-noded quadrilateral plate element. Finally finite element equation of motion for integrated structure is obtained as

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{f}_e + \mathbf{f}_a + \boldsymbol{\Gamma}\mathbf{E} \qquad (4)$$

where **M** is mass matrix, **K** stiffness matrix, **q** displacement vector, $\mathbf{f}_e$ external force vector, $\mathbf{f}_a$ force due to acoustic pressure, $\boldsymbol{\Gamma}$ is the matrix, whose element Γ_{ij} represent the ith froce component when unit voltage is applied to jth piezoelectric actuator, and E is the vector of electric voltage applied to piezoelectric actuators.

With the mass and stiffness matrices, the natural frequencies and mode shapes are found using the subspace iteration method. Equation (4) is transformed using the modal transformation

$$\mathbf{q} = \boldsymbol{\Phi}\boldsymbol{\eta} \qquad (5)$$

where $\boldsymbol{\Phi}$ is the modal matrix and $\boldsymbol{\eta}$ is the modal coordinate vector. Assuming the harmonic motion and modal damping, the transformed modal coordinate equation can be written as

$$\mathbf{H}\boldsymbol{\eta} = \boldsymbol{\Phi}^{T}(\mathbf{f}_e + \mathbf{f}_a + \boldsymbol{\Gamma}\mathbf{E}) \qquad (6)$$

where $\mathbf{H} = \boldsymbol{\Lambda} + \boldsymbol{\Xi} - \omega^2\mathbf{I}$,

$$\boldsymbol{\Lambda} = \boldsymbol{\Phi}^{T}\mathbf{K}\boldsymbol{\Phi},$$
$$\boldsymbol{\Xi} = \mathrm{Diag}\{2\zeta_i\omega_i\omega\},$$

I is the identity matrix, and ζ_i is the assumed modal damping coefficients.

4. Boundary Element Model for Sound Fields

The acoustic sound pressure at **x** in acoustic domain is governed by the Helmholtz integral equation expressed as

$$\alpha p(\mathbf{x}) = \int_{S(\mathbf{x}_s)} \left\{ G\frac{\partial p}{\partial n_s} - p\frac{\partial G}{\partial n_s} \right\} dS \qquad (7)$$

where S is the surface of cavity, $\mathbf{x}_s$ the point on surface S, α is 1 when $\mathbf{x}_s$ is inside S, 1/2 when $\mathbf{x}_s$ on S, and 0 when $\mathbf{x}_s$ outside S, and $\mathbf{n}_s$ outward unit normal on S. The Green's function is

$$G(\mathbf{x}, \mathbf{x}_s) = -\frac{e^{-ik|\mathbf{x}-\mathbf{x}_s|}}{4\pi|\mathbf{x}-\mathbf{x}_s|}$$

where k is the wave number defined as $k = \omega/c$ with sound speed c and forcing frequency ω. At the interface of structure and fluid, the normal derivative of the pressure can be related to the outward normal component of velocity v_n on S as

$$\nabla p(\mathbf{x}_s) = -\rho_o \ddot{u}(\mathbf{x}_s) = -i\omega\rho_o \mathbf{v}(\mathbf{x}_s) \qquad (8)$$

When the equation (7) is discretized with equation (8) by using the boundary element technique, we obtain the matrix equation as

$$\mathbf{E}\mathbf{p} = \mathbf{G}\mathbf{v}_n \qquad (9)$$

where **p** is the vector containing the sound pressure at **x** and $\mathbf{v}_n$ is the vector of normal velocity at node on S.

Four-noded quadrilateral element with the same shape function applied in finite element is used for the discretization, which promises the compatibility of BEM with FEM through wet nodes.

4.1. Fluid-Structure Coupling

The vector of normal velocities $\mathbf{v}_n$ in equation (9) is related to the vector of modal coordinate by the modal transformation and the transformation matrix **T** which transforms the nodal velocities in finite element system into the normal velocity in boundary element system,

$$\mathbf{v}_n = \mathbf{T}\dot{\mathbf{q}} = i\omega\mathbf{T}\boldsymbol{\Phi}\boldsymbol{\eta} \qquad (10)$$

If velocity $\mathbf{v}_n$ is eliminated from equation (9), the resulting equation is

$$\mathbf{E}\mathbf{p} = i\omega\mathbf{G}\mathbf{T}\boldsymbol{\Phi}\boldsymbol{\eta} \qquad (11)$$

and the FE equation (4) becomes

$$\mathbf{H}\boldsymbol{\eta} = \boldsymbol{\Phi}^{T}(\mathbf{f}_e + \mathbf{T}\mathbf{A}\mathbf{p} + \boldsymbol{\Gamma}\mathbf{E}) \qquad (12)$$

where **A** is the matrix converting acoustic pressure into force vector.

In case of plate-cavity system, the fluid-structure coupling is not weak enough to be ignored even though fluid is the air which is a kind of light fluid. Therefore two equations (11) and (12) should be simultaneously solved. In this analysis, the structural variable method was applied. First, structural modal displacement is solved with taking into account the coupling effects of acoustic pressure. Then the acoustic pressure is calculated from equation (11)

5. Control of Transmitted Sound Fields

For the global minimization of sound fields induced by vibrating plates sound power is selected as the control objective. The control is based on a implementation of classical LMS (Least Mean Square) algorithm.[7] The anisotropic PVDF films bonded on the surface of plate are used as distributed actuators for sound suppression. Transmitted sound power is defined as the integration of the sound intensity over the surface of vibrating plate:

$$\Pi = \frac{1}{2}\int_S Re[p^*(\mathbf{x}_s)v_n(\mathbf{x}_s)]dS \qquad (13)$$

where superscript * means the conjugate value of complex quantity. We calculate the sound power by integrating the nodal values of pressure and normal velocity as

$$\Pi = \frac{1}{2}\mathrm{Re}\left[\mathbf{p}^{\mathrm{H}}\left\{\sum_{k=1}^{N^e}\int_{S^e}\mathbf{H}^{e\mathrm{T}}\mathbf{H}^e dS\right\}\mathbf{v}_n\right] = \frac{1}{2}\mathrm{Re}\left[\mathbf{p}^{\mathrm{H}}\mathbf{R}\mathbf{v}_n\right]$$

where $$\mathbf{R} = \sum_{k=1}^{N^e}\int_{S^e}\mathbf{H}^{e\mathrm{T}}\mathbf{H}^e dS$$

and superscript H means the conjugate transpose of matrix. Using equation (10) and (11), sound power expression can written as

$$\Pi = \frac{1}{2}\mathrm{Re}\left[\omega^2\boldsymbol{\eta}^{\mathrm{H}}\boldsymbol{\Phi}^{\mathrm{T}}\mathbf{T}^{\mathrm{T}}\mathbf{G}^{\mathrm{H}}\mathbf{E}^{-\mathrm{H}}\mathbf{R}\mathbf{T}\boldsymbol{\Phi}\boldsymbol{\eta}\right]$$

By eliminating the modal coordinate vector with equation (6), the sound power is written in terms of external loads and control inputs to PVDF actuators as

$$\Pi = \frac{1}{2}\mathrm{Re}\left[(\mathbf{f}_e + \boldsymbol{\Gamma}\mathbf{E})^{\mathrm{H}}\mathbf{Z}(\mathbf{f}_e + \boldsymbol{\Gamma}\mathbf{E})\right] \qquad (14)$$

where

$$\mathbf{Z} = \omega^2\boldsymbol{\Phi}\mathbf{H}^{-\mathrm{H}}\boldsymbol{\Phi}^{\mathrm{T}}\mathbf{T}^{\mathrm{T}}\mathbf{G}^{\mathrm{H}}\mathbf{E}^{-\mathrm{H}}\mathbf{R}\mathbf{T}\boldsymbol{\Phi}\mathbf{H}^{-1}\boldsymbol{\Phi}^{\mathrm{T}}.$$

Since there exists the physical limit for electric field applied to piezoelectric material, the input electric field into piezoelectric actuator should be constrained. Therefore minimization problem of sound power expressed as equation (18) has the constraint that the control input should be less than the allowable limit of each PVDF actuator. The problem can be stated as

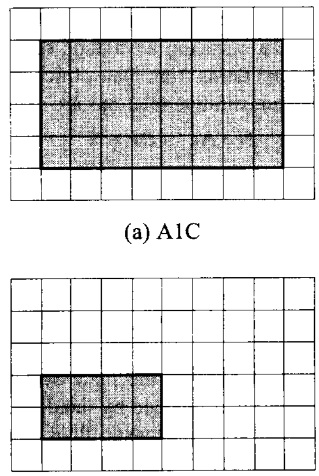
$$\text{Minimize} \quad \frac{1}{2}\mathrm{Re}\left[(\mathbf{f}_e + \boldsymbol{\Gamma}\mathbf{E})^{\mathrm{H}}\mathbf{Z}(\mathbf{f}_e + \boldsymbol{\Gamma}\mathbf{E})\right]$$

$$\text{Subject to} \quad \left\|E^k\right\| \le l_k, \quad k = 1,\cdots,n_A$$

where l_k is the allowable limit of input for kth actuator and n_A is total number of actuators. This minimization problem is approached by using the widely used program ADS(Automated Design Synthesis), so that the optimal control input can be obtained.

6. Numerical Examples

In this section the results of numerical simulations are presented. These analyses involve the active control of sound transmission through a rectangular integrated plate supported on the edges of rigid cavity. The acoustic medium is air whose density is 1.21 Kg/m^3 and wave velocity is 343 m/sec. The coupled FEM-BEM analyses are carried out for the cases of isotropic aluminum plate and anisotropic composite plate. PVDF actuators used for the control of sound transmission are varied in their shape and lamination angle to demonstrate some features of the actuators. In the analysis, the small amount of modal damping (ζ_i = *0.001*) is used for dynamic response of structure.



(a) A1C

(b) A1P

Figure 3. The shape of PVDF actuators used in the control simulation.

The actuators used in this analysis are PVDF pairs bonded with positive poling at the top surface and with negative poling at the bottom surface. The shapes of PVDF actuators considered here are depicted in Figure 3. The area filled with gray color is the region of the actuator: the first one (Figure 3(a)) is named A1C and the second one (Figure 3(b)) is named A1P. These PVDF pairs generate distributed bending moments in x and y direction when its ply angle is 0°, for the PVDF on top surface expands and that on bottom surface shrinks. The PVDF pairs directionally bonded with their own ply angle generate not only distributed bending moments in x and y directions but also distributed twisting moments due to the effects of ply angle.

Ply angles of PVDF layers on top and bottom surfaces are the same for an actuator. A1P$_\theta$ means the actuator A1P bonded at θ_{p+} on the top surface and bonded at θ_{p-} on the bottom surfaces, where the subscript p means the piezoelectric lamina and the +/- sign means the poling direction of each lamina.

In this analysis the dimension of rigid cavity is L_x = 0.3 *m*, L_y = 0.2 *m*, and L_z = 0.4 *m*. The rigid cavity system is modeled with 4-node boundary elements. The number of finite elements for plate structure is 72 and that of boundary elements for rigid cavity is 432.

6.1. Isotropic Plate

An integrated structure consists of aluminum host structure and PVDF materials bonded on the top and bottom surfaces of plate. Plate has the lay-up of $[\theta_{p+}/I/\theta_{p-}]$, in which I represents the isotropic material, the subscript p represents the PVDF material. The plate is simply supported along the edges at $x = \pm L_x/2$ and $y = \pm L_y/2$. Material properties of PVDF film are shown in Table 1 and those of aluminum are: Young's modulus 70×10^9 N/m^2, Poisson's ratio 0.3, and density 2700 Kg/m^3.

Control simulations with distributed PVDF actuators are performed. Actuators $A1C_0$, $A1P_0$, and $A1P_{45}$ are used in the analysis. First, actuator $A1C_0$ is employed in sound transmission control of aluminum plate. The thickness of plate is $h = 0.002$ m. The magnitude of external sound wave is 3 N/m^2 and incident angles are $\psi = 45°$ and $\chi = 45°$.

The transmitted sound power (TSP) is shown in Figure 4 and average mean square velocity of plate structure is shown in Figure 5. The average mean square (AMS) velocity, defined as

$$\langle v_n^2 \rangle = \frac{1}{S}\int_S \left[\frac{1}{T}\int_0^T v_n^2 dt \right] dS,$$

a measure of the velocity of structural vibration.[8] Comparing TSP and AMS velocity of uncontrolled system, there are peaks at 440 Hz and 870 Hz in TSP while no peak at these frequency in AMS velocity. The acoustic field of plate-cavity in these frequency ranges is not involved with the resonance of structural system. When an actuator $A1C_0$ only used, TSP at the resonances of even modes of plate cannot be controlled. The actuating forces of actuator A1C is not coupled with even vibration modes of structure due to its symmetrical shape. Thus TSP peaks induced by these even vibration modes are not removed. From 440 Hz to 580 Hz, TSP is reduced by sound control but AMS velocity is increased.[3] At frequencies above 700 Hz, TSP is reduced while AMS velocity is not changed.

Actuators $A1P_0$ and $A1P_{45}$ are used in control simulation of aluminum plate. The thickness of plate is $h = 0.0012$ m. The magnitude of external sound wave is 2 N/m^2 and $\psi = 60°$ and $\chi = 30°$.

TSP of the plate-cavity system is shown in Figure 6. The shape actuator A1P is different from that of A1C. With these actuators, transmitted sound power is reduced at most frequencies. However, the area of actuator A1P is not large enough to control the first vibration mode. Actuator $A1P_0$ cannot control the *6th* and *7th* peaks of TSP due to its lack of control force for these vibration modes. Actuator $A1P_{45}$ which is attached with ply angle 45° can control these TSP peaks, for the lamination angle of distributed PVDF actuator invokes the coupling between the control force and the acoustic field in cavity.

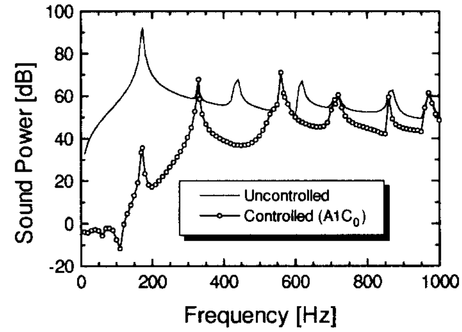


Figure 4. Transmitted sound power of isotropic plate in case of actuator $A1C_0$.

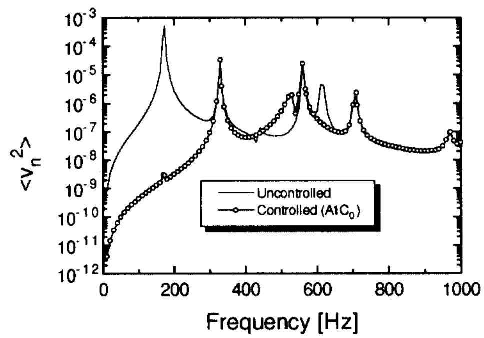


Figure 5. Average mean square velocity of isotropic plate in case of actuator $A1C_0$.

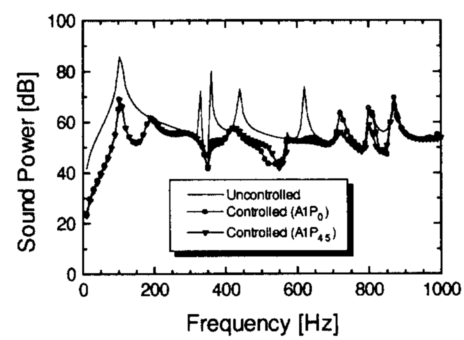


Figure 6. Transmitted sound power of isotropic plate in case of actuator $A1P_0$ and $A1P_{45}$.

6.2 Laminated Composite Plate

In this section, composite plate and PVDF actuators with their own shapes and laminate angles are considered to investigate the effects of shape and anisotropy of PVDF. The composite plate is made of Graphite/Epoxy fiber-reinforced composite whose lamination sequence is $[0_c/45_c/-45_c/90_c]_s$, in which the subscript c represents the composite material. The material properties of Graphite/ Epoxy composite and PVDF film are shown in Table 1. The allowable voltage input for this PVDF is 1100 volts. The dimension of host structure is 0.3 × 0.2 × 0.001 m. The harmonic incident plane wave is at $\psi = 30^\circ$ and $\chi = 60^\circ$.

Table 1. Material properties of Graphite/ Epoxy and PVDF film

Graphite/Epoxy	PVDF film
E_{11} = 181. Gpa	E = 2. GPa
E_{22} = 10.3 Gpa	ν = 0.3
G_{12} = 7.17 Gpa	ρ = 1780 Kg/m^3
G_{13} = 7.17 Gpa	d_{31} = 23.× 10^{-12} V/m
G_{23} = 2.87 Gpa	d_{32} = 3.× 10^{-12} V/m
ν_{12} = 0.28	t = 110 × 10^{-6} m
t = 0.125 mm	
ρ = 1520 Kg/m^3	

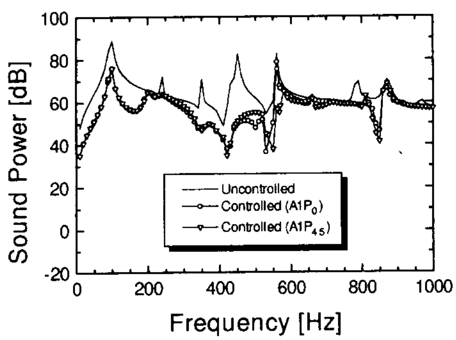


Figure 7. Transmitted sound power of composite plate in case of actuator $A1P_0$ and $A1P_{45}$.

Two actuators $A1P_0$ and $A1P_{45}$ are employed in control simulation. These actuators represent similarly good effectiveness for most frequencies as shown in Figure 7. However the actuator $A1P_{45}$ is more effective in the sense that it can control TSP from 550 Hz to 600 Hz. In this region, sound power is related to the 7th vibration mode of plate and $A1P_{45}$ can control this mode better than $A1P_0$. This enhancement of control effectiveness results from the anisotropic characteristics of PVDF.

7. Conclusion

By using the anisotropic features and shape of distributed polyvinylidene fluoride(PVDF) actuator, numerical investigations on the control of sound transmission of plate-cavity system was performed. Fluid-structure interaction and the interior sound fields are efficiently calculated with coupled FEM-BEM approach. For simply supported isotropic and laminated composite plate, numerical control simulation was performed with distributed PVDF actuators. Actuator $A1C_0$ has large control force. However it cannot control the sound transmission at the frequencies related to the anti-symmetric vibration modes of plate. Actuators $A1P_0$ and $A1P_{45}$ which have different shape from A1C can control most frequencies. Especially actuator $A1P_{45}$ shows the better performance from the viewpoint of control effectiveness. The coupling effects between interior sound field and actuating force by PVDF actuator is increased by the directionally attached PVDF on the surface of plate. The shape of distributed actuator is highly coupled with vibration modes of composite plates. Therefore it can be taken advantage of to improve the effectiveness in acoustic noise control. Furthermore the effects due to coupling between vibration modes and actuating forces by distributed PVDF actuator could be made better, without increasing the number of actuators, by changing the lamination angle of the actuator with the determined shape.

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