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計畫主持人：金大仁 教授
計畫參與人員：王崧任、江宗鴻、陳致翔、施恩寶

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中文摘要

本報告研究奈米材料用來製作揚聲板的可行性，其中包括提出一製作奈米複合材料板的程序，然後利用理論和實驗方式研究奈米複合材料板的揚聲特性。在理論方面的研究中，建立一分析平板揚聲器聲壓曲線的方法，探討導致中音谷產生的原因及振形。實驗方面，製作奈米複合材料三明治及其他材料的揚聲板，並組裝平板揚聲器供聲壓量測用。比較理論與實驗結果，証明奈米複合材料有改進平板揚聲器聲壓曲線的功能，可使中音谷變小，提升聲音的品質。

關鍵字：奈米複合材料、複材板、有限元素分析、振動、聲傳

Abstract

The possibility of utilizing nano-composites in making sound radiating panels is studied. A procedure for fabricating rectangular nano-composites sound radiating panels is described. A number of rectangular flat-panel speakers comprising sound radiating panels made of different materials are fabricated for experimental investigation. In this study, the flat-panel speaker comprises a rigid frame, a flexible edge surround, a sound radiating panel, and an electro-magnetic type transducer. The transducer, which is located at the center of the sound radiating panel, is used to excite the panel to produce sound. A method is presented to study the sound radiation behavior of the rectangular composite flat-panel speaker. In the proposed method, the finite element method is first used to determine the vibration response of the sound radiating panel of the flat-panel speaker in which the edge surround is modeled as a continuously distributed spring system. The vibration responses of the panel excited at different frequencies together with the first Rayleigh integral are then used to construct the SPL curve of the speaker. Experimental SPL curves of the flat-panel speakers comprising different sound radiating panels were measured to validate the proposed method. The comparison between the theoretical and experimental results has shown that the proposed method is capable of producing reasonably good SPL curves for the flat-panel speakers. The effects of panel materials on the the SPL curves of the sound radiating panel are studied and discussed.

Keywords: Nano-composites, composite plate, finite element analysis, vibration,

sound radiation.

Introduction

In recent years, nano-composites have become important materials in different industries. Many researchers are trying hard to find new applications for nano-composites [1-7]. For instance, Uddin and Sun [4] attempted to use nano-silica particles to increase the tensile strength of Gl/ep laminates. On the other hand, many people have presented different techniques to characterize the mechanical and/or material properties of nano-composites. For instance, Iizuka [8] and Wu [9] have studied the properties of W2C-nano-particle-reinforced Si3N4 matrix and PP/nano-SiO2 composite materials. Wang [10] has studied the material characterization of nano-SiC particles reinforced TiC/SiC composites. Regarding the sound radiation properties of nano-composites, so far not much attention has been drawn to this aspect. Conventional loudspeakers utilizing a cone-type diaphragm as a sound radiator have been in widespread use. The cone shape radiator which is mechanically driven at its smaller end and in a pistonic manner by a moving coil of electro-magnetic means can radiate sound waves from the front and rear of the radiator. In general, a cone-type speaker is not a full-range speaker and only able to produce desired sound in a portion of the audible frequency range depending on its size. For instance, a speaker of large diameter such as a subwoofer-type speaker can only produce base sound in the low frequency range. When the excitation frequency exceeds the upper limit of the designed frequency range, the speaker diaphragm will produce transverse deformation which can make the sound radiation destructive. Another ominous shortcoming of the cone-type speaker is its relatively large thickness. Since consumer electronics such as LCD TVs, notebook computers, DVD players etc. have the tendency to become thinner and thinner, it is not surprised that sooner or later cone-type speakers will have a great difficulty to find applications in these kinds of electronic products. The shortcomings of cone-type speakers have led to the intensive development of flat-panel speakers in recent years and many proposals have thus resulted. For instance, Watters et al [11] used the concept of coincidence frequency, where the speed of sound in panels subject to bending wave action matches the speed of sound in air, to design a light stiff strip element of composite structure that can sustain bending waves and produce a highly directional sound output over a specified frequency range. The vibration response of such speaker, however, was so low that the speaker could not radiate sound efficiently. Heron [12] proposed a flat-panel loudspeaker which had a resonant multi-mode radiator panel. The radiator panel which was a skinned composite with a honeycomb core was excited at frequencies above the fundamental and coincidence frequencies of the panel to provide, hopefully, high radiation efficiency through multi-modal motions within the panel. The operating frequency range of the radiating panel, however, is not wide enough for general purposes and thus only suitable for public address application. Matsushita [13] presented a method to design a long rectangular flat-panel speaker in which two transducers were used to excite the radiator panel for sound radiation. In this method, the locations of the transducer are chosen in such a way that the adverse effects on sound radiation of some resonant modes are suppressed or even eliminated. Azima et al [14] proposed a distributed mode method for the design of a flat-panel acoustic device which consisted of a panel radiator capable of sustaining bending waves associated with resonant modes in the panel radiator and used transducers to excite the resonant modes of the panel radiator. Their proposed distributed mode method includes analysis of distribution of flexural resonant modes and identification of dead/combined dead-spots of the panel radiator. The transducers are mounted at some particular points which, hopefully, will not be coincident with the dead/combined dead-spots, on the radiating panel so that all the resonant modes of the panel can be excited for sound radiation. The excitation locations determined in the distributed mode method, however, may create a number of problems for the panel-form speaker. For instance, some resonant modes may be overexcited so that undesirable sound intensities at the associated resonant frequencies may be generated and the resonant modes that have adverse effects on sound radiation may be excited to cause sound interference. Kam [15] presented a method to design laminated composite flat-panel speakers. In this method, excitation locations are determined in such a way that the resonant modes which have adverse effects on sound radiation are suppressed while those have beneficial effects on sound radiation are properly encouraged via an optimal design approach. In view of the previous design methods for flat-panel speakers, it is obvious that the properties of the resonant modes of radiating panels have direct effects on sound radiation of flat-panel speakers. Therefore, more in-depth research work on vibration and sound radiation behaviors of flat-panel speakers is needed if high performance flat-panel speakers that can radiate good quality sound are to be designed.

The sound radiating panel of a composite flat-panel speaker can be modeled as an elastically restrained composite plate. Regarding the vibration of composite plates, many researchers have proposed different techniques to determine the dynamic behavior of composite plates. Among them, the finite element method is one of the commonly used techniques to determine the modal characteristics and vibration responses of composite plates. Since a vibrating plate can radiate sound, to properly control the sound radiated from the plate, the vibro-acoustic behavior of the plate needs to be studied thoroughly. In recently years, a number of researchers have proposed different methods to study sound radiation of plates with various boundary conditions [16-38]. For instance, Rdzanek et al [16] presented a method to study the active and reactive sound power of the axisymmetric modes of free vibrations of elastically supported circular plates embedded in a rigid baffle. In their method, the integral formulations for the active and reactive sound power expressed in their Hankel representations were used to derive some elementary formulations in the form of some high-frequency asymptotes valid for frequencies higher than the successive coincidence frequencies of the plate. Qiao and Huang [17] used the Rayleigh integral, the Zwicker's loudness model and the plate dynamic response to study the effects of boundary conditions on sound loudness radiated from rectangular plates. The effects of boundary conditions on sound intensity level, sound intensity density and critical-band level were also studied, taking the frequency selectivity of human hearing into account. Anderson and Bratos-Anderson [18] studied the sound radiation efficiency of specially orthotropic, baffled rectangular plates via both theoretical and experimental approaches. The influence of the orthotropy on the distribution of different types of resonant modes of the plate was considered in the analysis. The average modal sound radiation efficiency of the resonant modes was determined and its dependence on the parameters associated with the material properties and geometry of the plate was discussed. Denli and Su [19] presented an optimization procedure to design optimal boundary supports for minimizing structural sound radiation of plates. In the previous studies, the plates under consideration were excited by surface, line or point loads. For a plate used as a sound radiator, besides restraining flexibly around the periphery, the plate is also supported by interior elastic restraints for attaining stable vertical motion and excited by a ring load. In general, the modal characteristics of this kind of plates play an important role in the sound radiation of the plates. An in-depth investigation of the effects of the modal characteristics on the sound radiating plates is thus needed if better understanding of the radiation behaviors of the plates is desired.

In this report, sound radiation of flat-panel speakers comprising sound radiating panels made of nano-composites as well as other materials is studied via both theoretical and experimental approaches. The forced vibration analysis of the flexibly supported sound radiating panel of the flat-panel speaker is accomplished using the finite element method. The vibration responses of the sound radiating panel excited at different frequencies are used in the Rayleigh first integral to construct the SPL curves for studying the sound radiation behavior of the flat-panel speaker. The capability of the proposed method in predicting accurate SPL curves is validated by experimental results. The mode shapes that have destructive effects on sound radiation of the flat-panel speakers are identified. The effects of panel materials on the SPL curves of the flat-panel speakers are investigated.

Finite Element Analysis of Flat-Panel Speaker

The rectangular flat-panel speaker under consideration is composed of a rigid frame and a sound radiator which are schematically shown in Fig. 1. The sound radiator, on the other hand, comprises a flexible surround, a sound radiating panel, two interior elastic restraints, and an electro-magnetic type transducer. The transducer is located at the panel center while the two interior elastic restraints are located symmetrically on the two sides of the panel center. Without loss of generality, the sound radiating panel is assumed to be a composite sandwich plate of track-and-field shape in which the core and face sheets are made of different kinds of orthotropic materials. For an orthotropic lamina, neglecting the normal stress in the thickness direction, the stress-strain relations can be expressed in matrix form as [39]

$$\begin{Bmatrix} \sigma_{x\,m}^{(k)} \\ \sigma_{y\,m}^{(k)} \\ \tau_{xy\,m}^{(k)} \\ \tau_{yz\,m}^{(k)} \\ \tau_{xz\,m}^{(k)} \end{Bmatrix} = \begin{bmatrix} \bar{Q}_{11\,m}^{(k)} & \bar{Q}_{12\,m}^{(k)} & \bar{Q}_{16\,m}^{(k)} & 0 & 0 \\ \bar{Q}_{12\,m}^{(k)} & \bar{Q}_{22\,m}^{(k)} & \bar{Q}_{23\,m}^{(k)} & 0 & 0 \\ \bar{Q}_{16\,m}^{(k)} & \bar{Q}_{26\,m}^{(k)} & \bar{Q}_{66\,m}^{(k)} & 0 & 0 \\ 0 & 0 & 0 & \bar{Q}_{44\,m}^{(k)} & \bar{Q}_{45\,m}^{(k)} \\ 0 & 0 & 0 & \bar{Q}_{45\,m}^{(k)} & \bar{Q}_{55\,m}^{(k)} \end{bmatrix} \begin{Bmatrix} \varepsilon_{x\,m}^{(k)} \\ \varepsilon_{y\,m}^{(k)} \\ \gamma_{xy\,m}^{(k)} \\ \gamma_{yz\,m}^{(k)} \\ \gamma_{xz\,m}^{(k)} \end{Bmatrix} \tag{1}$$

with

$$Q_{11} = \frac{E_1}{1-\upsilon_{12}\upsilon_{21}};\ Q_{12} = \frac{\upsilon_{12}E_2}{1-\upsilon_{12}\upsilon_{21}};\ Q_{22} = \frac{E_2}{1-\upsilon_{12}\upsilon_{21}}$$
$$Q_{44} = G_{23};\ Q_{55} = G_{13};\ Q_{66} = G_{12};\ C = cos\theta_i;\ S = sin\theta_i \tag{2}$$

where Q_{ij} are lamina stiffness coefficients; E_1 , E_2 are Young's moduli in the fiber and transverse directions, respectively; v_{ij} is Poisson's ratio for transverse strain in the *j*-direction when stressed in the *i*-direction ; G_{12} is in-plane shear modulus in the 1-2 plane; G_{13} and G_{23} are transverse shear moduli in the 1-3 and 2-3 planes, respectively.

The electro-magnetic transducer uses the vertical motion of the voice coil of diameter d_v to excite the sound radiating panel. The interior elastic restraints are used to position the voice coil in the magnetic field existing in the transducer and stabilize the motion of the voice coil. The mathematical model for sound radiation analysis of the flat-panel speaker is shown in Fig. 2 in which the flexible surround is modeled as a spring system with translational and rotational spring constant intensities k_S and k_R, respectively, It is noted that in the adopted mathematical model, the stiffness of the voice coil are too small to be considered. Half of the mass of the surround is distributed uniformly around the periphery of the panel. The excitation force and the voice coil mass are distributed uniformly on the periphery of a circle with diameter d_v on the bottom surface of the sound radiating panel. The magnitude of the total excitation force F is calculated from the following equation.

$$F = BLI \tag{3}$$

where *B* is magnetic flux; *L* is wire length of voice coil submerged in magnetic flux. The electric current is obtained as

$$I = V / R(\varpi) \tag{4}$$

where *V* is voltage; *R* is impedance which depends on excitation frequency. The relation between impedance and excitation frequency for a transducer can be constructed experimentally using the commercial sound measurement code MLSSA [40]. A typical impedance-frequency curve is shown in Fig. 3. It is noted that the first peak of the impedance curve occurs at the first resonant frequency f_0 of the sound radiator at which the flexibly supported sound radiating panel undergoes pistonic motion. After the first peak, impedance increases as excitation gets higher. The commercial finite element code ANSYS is then used to perform the vibration analysis of the sound radiator of the flat-panel speaker. In the finite element model of the sound radiator, Spring-damper 21, Shell 91 elements, and Mass 21 are used to model the springs, panel, and distributed masses, respectively. If the sound radiating panel is an isotropic or orthotropic plate, Shell 99 rather than Shell 91 elements are used to model the panel. The finite element model of the sound radiator is shown in Fig. 4. The modal characteristics (natural frequencies and mode shapes) of the sound radiator can be determined by solving the following eigenvalue problem.

$$\left[\underline{\mathbf{K}} - \omega^2 \underline{\mathbf{M}}\right] \underline{\mathbf{C}} = \underline{\mathbf{0}} \tag{5}$$

where $\underline{\mathbf{K}}$, $\underline{\mathbf{M}}$ are structural nodal stiffness and mass matrices of the flat-panel speaker, respectively; C is the vector of structural nodal displacements; ω is natural frequency.

The equations of motion of the sound radiator can be expressed in matrix form as

$$\underline{\mathbf{M}}\ddot{\underline{\mathbf{C}}} + \underline{\mathbf{D}}\dot{\underline{\mathbf{C}}} + \underline{\mathbf{K}}\underline{\mathbf{C}} = \underline{\mathbf{F}} \tag{6}$$

where $\underline{\mathbf{D}}$ is the structural damping matrix; $\underline{\mathbf{F}}$ is the vector of structural nodal force.

In the modal analysis, the above equations of motion are uncoupled to give the following modal equations.

$$\ddot{u}_i + 2\xi_i^* \omega_i \dot{u}_i + \omega_i^2 u_i = f_i^* / M_i^*, \quad i = 1,2,\cdots \tag{7}$$

where f_i^*, M_i^*, and ξ_i^* are the modal generalized force, mass, damping ratio. According to the definition of Rayleigh damping, the modal generalized damping ratio can be determined using the following equation.

$$\xi_i^* = \alpha M_i^* + \beta K_i^* \tag{8}$$

where α and β are parameters which can be estimated using two modal damping ratios determined from the measured frequency response spectrum of the sound radiator.

The deflections of the sound radiating panel subjected to the driving force with different excitation frequencies can then be determined by solving eqn (6).

Sound Radiation of Laminated Composite Plates

A speaker is a device used to recover and magnify the original sound stored in the input electrical signals. For a high fidelity speaker, it is important to have a uniformly distributed SPL spectrum in the operating frequency range so that the sounds at different frequencies in this range are magnified equally. Therefore, the design of a relatively flat SPL curve for a speaker is an important step to achieve high fidelity. In the sound radiation analysis of a flat-panel speaker, it is assumed that air loading has negligible effects on plate vibration. Then referring to the baffled speaker with area S shown in Fig. 5, the sound pressure p(r, t) resulting from the vibration of the plate can be determined using the Rayleigh first integral.

$$p(r_i, t) = \frac{-\omega^2 \rho_0}{2\pi} \sum_i A_i e^{j(2\varpi t + \theta_i - kr_i)} \frac{\Delta S}{r_i} \tag{9}$$

where A_i is the amplitude of the nodal response; θ_i is nodal phase angle; ρ_0 is air density, k is wave number $(=\omega/c)$; c is speed of sound; r_i is the distance between the sound measurement point and the differential area on the panel. For air at $20\,°C$ and standard atmospheric pressure, $\rho_0 = 1.2\ kg/m^3$ and $c = 344\ m/s$. The sound pressure in the above equation is then obtained numerically. The SPL produced by the panel can be calculated as

$$SPL \equiv 20\log_{10}\left(\frac{p_{rms}}{2\times 10^{-5}}\right) dB \tag{10}$$

with

$$p_{rms} = \left[\frac{1}{T} \int_{-T/2}^{T/2} \left| p(r,t) \right|^2 \mathrm{dt} \right]^{1/2} \tag{11}$$

Fabrication of Nano-composite Sandwich Plate

The symmetric nano-composite sandwich plate under consideration is composed of two nano-composite face sheets and an isotropic core. The material of the face sheet is a mixture of epoxy and SiO2 nano-particles (4%wt.). The process for fabricating the nano-composite sandwich plate is described as the following. The epoxy (Shell epoxy resin Epikote 828) is first mixed with 4% wt. SiO2 nano-particles with diameter of 12nm (Degussa Taiwan Ltd. AE200). The resin and nano-particles are then blended mechanically at the speed of 800 rpm for 4 hours at 80℃ to disperse the nano-particles in the resin. The mixing is continued using the sonicator (Misonix sonicator 3000) with power around 90-120 Walt for 30 minutes to disperse the nano-SiO2 particles in the epoxy resin. The nano-SiO2 /epoxy is further blended with the hardener (D230) using the weight ratio of nano-SiO2 /epoxy to hardener of 1:0.32 for 15 minutes. The fully blended nano-composites/resin is then imprinted on the top and bottom surfaces of a light-weight foam core. The core coated with uncured nano-composites/resin is then vacuumed to reduce the bubbles in the resin for 30 minutes. Finally, the sandwiched core are covered by Teflon sheets and cured under room temperature for 24 hours to produce the nano-composite sandwich pane.

Experimental Investigation and Results

The sound radiation of a number of flat-panel speakers comprising sound radiating panels made of different materials and excited by a circular electro-magnetic type transducer at the center of each plate was studied experimentally. Silica nano-particles and foam panel were used to fabricate two types of sound radiating panels, namely, foam and symmetric composite sandwich panels with dimensions a = 128 mm and b = 35 mm. The sandwich panel was composed of two nano-composites sheets and a foam core. The foam panel was 2mm thick while the thicknesses of the face sheet and core of the sandwich panel were 0.125 and 2mm, respectively. The properties of the materials determined from experiments were given as follows.

Silica nano-paricle/ep

$E = 3.05$ GPa, $\nu = 0.28$,

$\rho = 1970\ \mathrm{Kg/m^3}$

Foam

$E = 2.03$ GPa, $\nu = 0.3$,

$\rho = 122\ \mathrm{Kg/m^3}$

The coefficients of variation of the

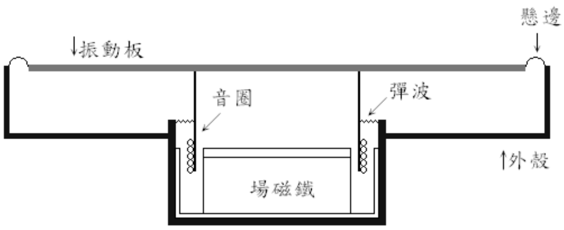
above material constants were less than 3%. A study of the mechanical behavior of the surround had shown that the rotational stiffness of the surround was negligible. The spring constant intensities of the surround and interior elastic restraints determined from experiments were $K_S = 7.5 \times 10^4$ N/m^2 and K_R =0.025 N, respectively. The mass and radius of the voice coil were 0.2 g and 25mm, respectively. The experimental setup for sound pressure level measurement of the flat-panel speakers are schematically shown in Fig. 6 in which the baffled speaker was tested in an anechoic chamber. The sound pressure level of the speaker subjected to excitation with one Watt power was measured by a microphone placed at the location one meter away from the center of the plate. The sound measurement instrument LMS [41] provided electrical signals sweeping from 20 to 20K Hz to the transducer which in term excited the panel of the speaker to radiate sounds. The sound pressure signals picked up by the microphone were processed by LMS to produce the SPL curve of the speaker. A number of flate-panel speakers with dimensions of 38x30mm, 60x30mm, and 42x20mm comprising different panels were also fabricated for investigating the effects of different materials on the SPL curves of the speakers.

Results and Discussions

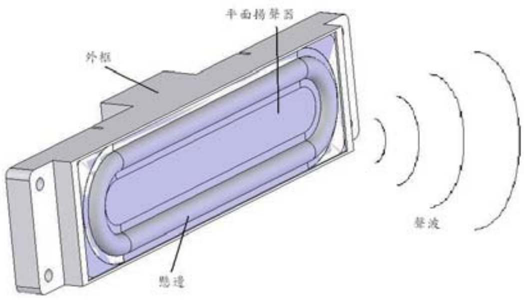
The present sound radiation method is used to study the sound radiation behaviors of the flat-panel speakers which have been tested. To demonstrate the capability of the present method, the theoretical SPL curves of the speakers are constructed and compared with the experimental ones. The theoretical and experimental SPL curves of the speakers composed of different sound radiating panel are plotted in Fig. 7 and 8 for comparisons. From the experimental SPL curves, it is noted that the average SPLs produced by the foam and composite sandwich panels were 78 and 75 dB, respectively. The foam panel could produce the highest SPL because its weight was lighter. The mid-range dips of the foam and sandwich panels occurred at around 3KHz and the associated depths of the mid-range dips were 12 and 2 dB, respectively. It is also noted that the depth of the mid-range dip became smaller as the panel stiffness increased. The above results have thus revealed the fact that a panel has lighter weight and higher stiffness can produce higher sensitivity and flatter SPL curve. Comparing the experimental and theoretical results, it is noted that the percentage differences between the theoretical and experimental average SPLs as well as the depths of the mid-range SPL dips are less than 1 %. Thus for each of the speakers, the theoretical and experimental SPL curves are in fairly good agreement. The mode shapes associated with the mid-range SPL dips of the sound radiating panels are symmetric with respect to the centers of the panels as shown in Fig. 9 in which the two nodal lines are mm apart from each other. The nodal lines of the mode shape divide the panel into three zones along the long axis of the panel in which the phase angle of the two end zones (zones 1 and 3) are 180^0 different from that of the middle zone (zone 2). The phase difference between the middle and the two end zones has caused sound interference among the sounds radiating from the zones. The sound interference has in turn led to the development of the mid-range SPL dip for each of the speakers. A detailed sound radiation analysis has shown that the locations of the nodal lines and the deflections in different zones have significant effects on the mid-range SPL dips of the sound radiating panels. Theoretical and experimental investigations of flat-panel speakers with different dimensions comprising nano-composite materials sound radiating panels have also been performed. Again the results obtained have revealed the fact that nano-composites can increase the stiffness of the sound radiating panels and thus reduce the depths of the mid-range SPL dips of the speakers.

Conclusions

Nano-materials have been applied to fabricate sound radiation panels. A procedure for fabricating the nano-materials coated sound radiating panels has been proposed. A method has been presented to study the sound radiation behavior of flat-panel speakers. The method has used the finite element method to determine the vibration responses of the sound radiating panel and Rayleigh's first integral to evaluate the SPL curve of the panel. A number of sound radiating panels have been fabricated for experimental and theoretical investigations. The comparisons between the experimental and theoretical SPL curves for different panels have shown that the present sound radiation method can obtain fairly good results. The mode shapes that have important effects on the mid-range SPL dips of the panels have been identified. These modes are symmetrical with respect to the panel center and have two nodal line running in the short axis direction of the panel. It has been shown that the coating of nano-composites on the panel can change this type of mode shape and modify the characteristics of the mid-range SPL dip. Therefore, the panel coated with nano-composites can have better sound radiation performance.



(a) Cross-sectional view



(b) Side view

Fig. 1 Schematic description of flat-panel speaker

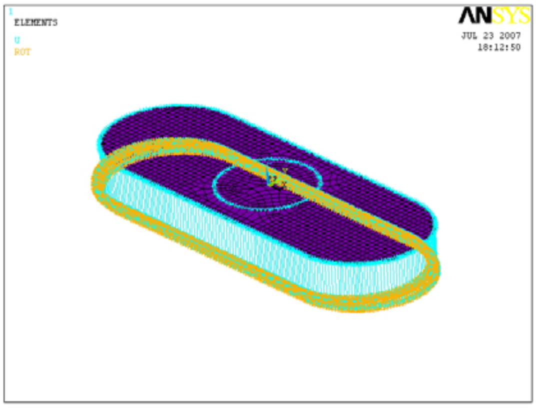


Fig. 4 Finite element model of sound radiator

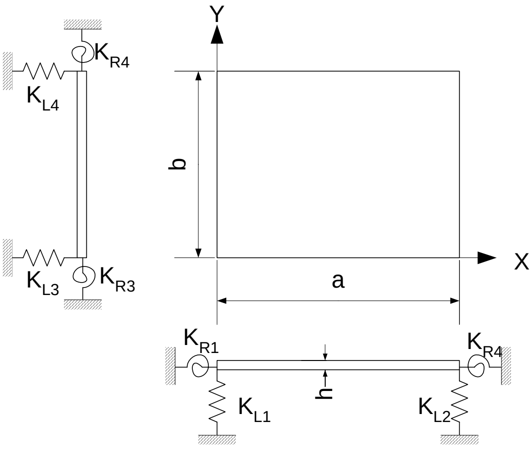


Fig. 2 Mathematical model of flat-panel speaker

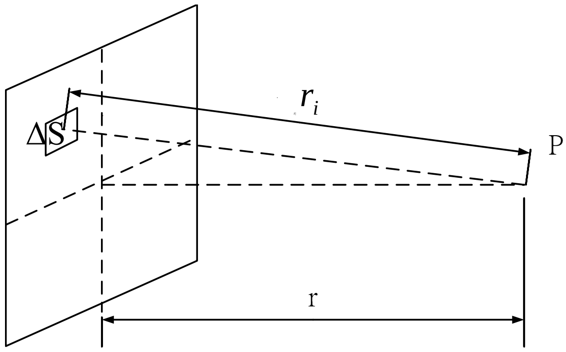


Fig. 5 Schematic description of sound measurement

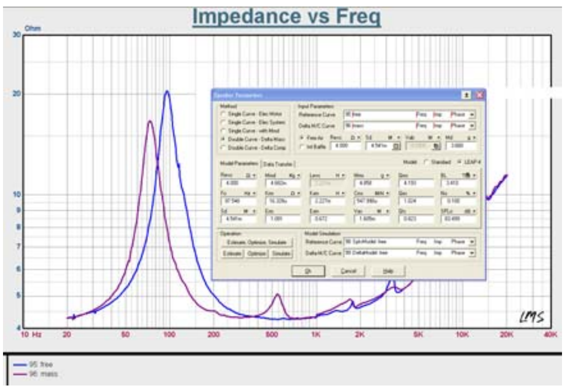
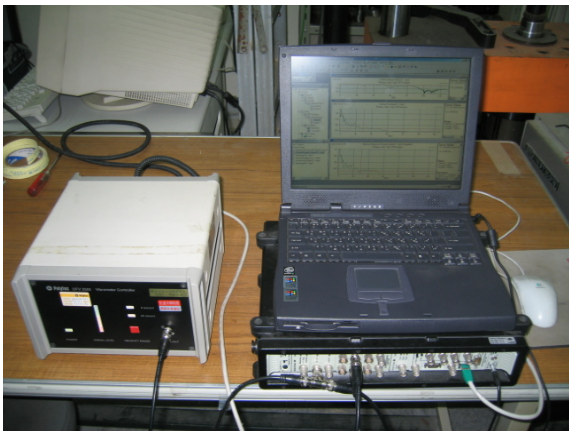


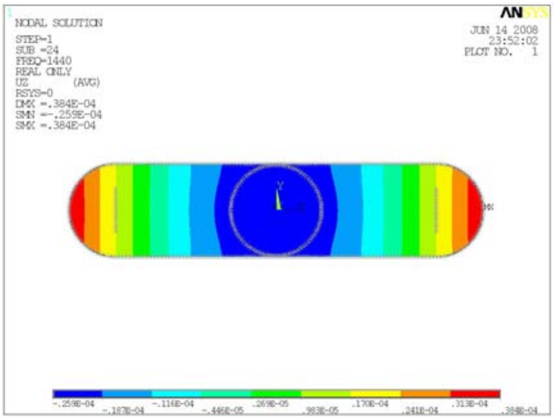
Fig. 3 Typical impedance curve

(a) Anechoic chamber



(b) Sound measurement instrument

Fig. 6 Experimental setup of sound measurement



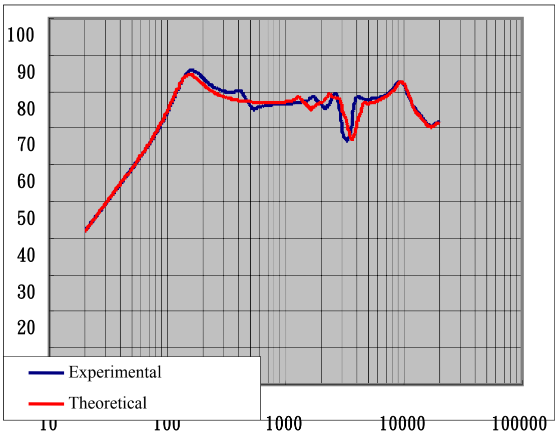


Fig. 7 Theoretical and experimental SPL curves of foam panel

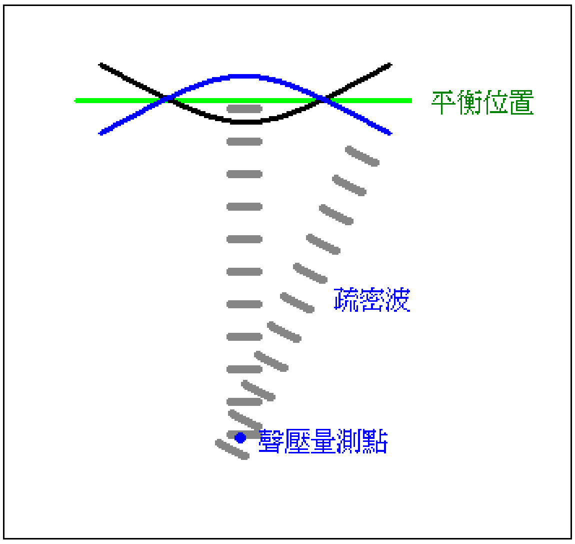


Fig. 9 Mode shape for mid-range SPL dip

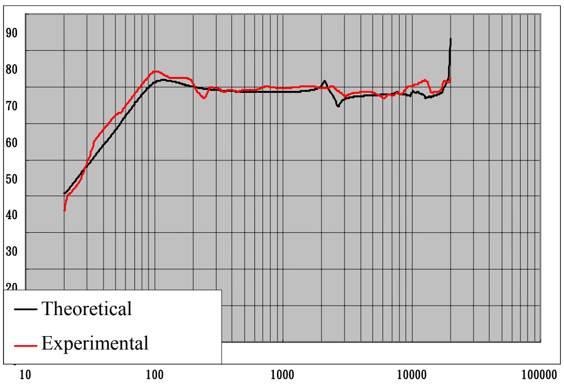


Fig. 8 Theoretical and experimental SPL curves of composite sandwich panel

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