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Topology optimization of composite material plate with respect to sound radiation

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ABSTRACT

In this paper, topology optimization of composite material plate with respect to minimization of the sound power radiation has been studied. A new low noise design method based on topology optimization is proposed, which provides great guidance for acoustic designers. The structural vibrations are excited by external harmonic mechanical load with prescribed frequency and amplitude. The sound power is calculated using boundary element method. An extended solid isotropic material with penalization (SIMP) model is introduced for acoustic design sensitivity analysis in topology optimization, where the same penalization is applied for the stiffness and mass of the structural volume elements. Volumetric densities of stiffer material are chosen as design variables. Finally, taking a simple supported thin plate as a simulation example, the sound power radiation from structures subjected to forced vibration can be considerably reduced, leading to a reduction of 20 dB. It is shown that the optimal topology is easy to manufacture at low frequency, while as the loading frequency increases, the optimal topology shows a more and more complicated periodicity which makes it difficult to manufacture.

1. Introduction

Noise control is becoming more and more important for engineering design [1,2] while the sound radiation of thin plate is always one of the key points. Although active vibration control could be used to decrease structural noise [3], Low noise design is the most reliable means of reducing radiated noise. It is noted that the reduction of low frequency noise is of particular interest, not only because of comfort-related issues, but also because this is usually the most difficult range to deal with. In the passive approach, sizes and shapes of the structural components and composite material tailoring are candidates for optimization [4–6].

"Designing for noise control" was presented at the 37th "INTER-NOISE 1978" about more than 30 years ago. Richard [1], the professor of Massachusetts Institute of Technology, made a brilliant exposition on the conference. Koopmann and Constans [7,8] performed further research on low noise design of structures. The optimization about acoustic radiation properties was considered with the objective of minimizing the total sound power radiated from the vibrating shell surface into a surrounding acoustic medium using simulated annealing algorithm. In addition, a wine glass is tuned optimally to move the first four eigenvalues into harmonic relationships by Koopmann and Belegundu [7]. The design variables were small masses that were added to the upper surface of the wine glass. The conversion of mode shapes of a vibrating shell into weak radiator was accomplished through the introduction of point masses and the calculation of their optimum distributions by Constans and Koopmann [8]. The IUTAM Symposium on "Designing for Quietness" [9] pointed out that "Designing for Quietness" was an important aspect in engineering acoustics and technology acoustics. The noise of the structure should be the basic design index in the technical design stage so as to predict the noise level and make the noise controlled.

Based on the acoustic radiation modes [10] and further study about acoustic radiation, acoustic design sensitivity (ADS) analysis [11,12] was presented to guide low noise design. Weak radiator could be acquired by ADS analysis which was important to engineering design.

ADS analysis presents structural changes in the sound radiation characteristics which can predict the structures with the minimization of sound radiation power. Weak radiator could be obtained by ADS analysis which was important to engineering design.

Up till now, when studying ADS analysis about acoustic radiation, sound pressure is often adopted for design objective [13], and the

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effect of design variable on sound pressure is considered. However, sound pressure is varying according to spatial position and the calculation is time consuming. The sound power reveals itself as the most adequate means for quantifying the radiation on the structure's surface. It is related to the characteristics of structure and does not change with spatial position. Therefore, sound power is more suitable than sound pressure for ADS analysis.

The method of topology optimization of continuum structures first appeared in the literature about twenty years ago by Bendsøe and Kikuchi [14]. Structural topology optimization has been an extremely active area of research since then [15]. The method of topology optimization was first applied to problems with structural–acoustical design objectives by Diaz and Kikuchi [16]. The IUTAM Symposium on "Topological Design Optimization of Structures, Machines and Materials" [17] proposed that topology optimization used for low noise design of structures could give weak radiators which meet the demand of low noise design. Through treating the problem as a material distribution problem, Luo and Gea [18] also found optimal stiffeners configuration which can reduce interior noise based on topology optimization. Olhoff and Jianbin [19], Mendonca et al. [20] dealt with topology optimization problems of a composite material plate-like structure with the design objective of minimizing the sound power radiation using the method of topology optimization, and finally they got satisfactory numerical results, but their model is over simplified. Maria [21] presented a method to control acoustic properties in a room with topology optimization, which made the squared sound pressure amplitude in a certain part of a room minimized by redistribution of material in a design domain.

This work focuses on the topology optimization of composite material structures with respect to sound radiation. The sound power radiated from the structure surface achieves the minimum in our design through topology optimization. We assume the air as the acoustic medium and a feedback coupling between the acoustic medium and the structure can be neglected. No material damping effects are considered in this paper. The topology optimization subjected to given material volume, admissible design domain, and boundary conditions is carried out by directly minimizing the sound power radiated from the structural surfaces. The structural vibrations are excited by a time harmonic mechanical loading with prescribed excitation frequency, amplitude, direction, and spatial distribution. A composite material SIMP (solid isotropic material with penalization) model is employed in the topology optimization.

It is shown that the optimum topology of the structure is easy to manufacture for low values of the loading frequency. But as the loading frequency is increased the optimum topology of the structure shows a more and more complicated periodicity.

The material of the paper is organized as follows. At the beginning of Section 2, the structural topology optimization subject to given volumes of the constituents of composite material structures is formulated for problems of minimizing the sound power radiated from the vibrating structural surface into the acoustic medium. In Sections 2.1 and 2.2, the finite element analysis of structural vibration and boundary element analysis of sound radiation are presented. The velocity distribution of structure is solved by the FEM method and the surface sound pressure is calculated by Rayleigh integral, respectively. The sound power can be expressed as a positive definite quadratic form of the Hermitian based on the sound radiation mode theory. In Section 2.3, the ADS analysis of thin plate with respect to sound radiation power is presented. It is transformed into the sensitivity of dynamic and impedance matrix with respect to design variables. An extended form of the SIMP model is introduced in Section 2.4 for the handling of composite material topology design of the structures. In Section 4, taking a simple supported thin plate as a simulation example, numerically obtained optimum topology designs for the two types of composite material structures are presented and discussed. The results show the validity of the presented method and give the optimum design of thin plate with low sound radiation power. Section 5 concludes the paper.

2. Minimization of sound power radiation using topology optimization of composite material structures

Topological design optimization is considered in this section with the goal of minimizing the total sound power of the structure. We consider a vibrating elastic composite material structure, in which an optimal distribution of two different materials is achieved for the minimal sound radiation. The structural vibrations are assumed to be excited by a time-harmonic mechanical surface loading vector $f(t)=F_a e^{-i\omega t}$ with prescribed frequency ω and amplitude vector F_a on S. A complete formulation of the problem of minimizing the sound power W is presented as follows:

$$\begin{aligned}
&\text{Find}: \rho=\{\rho_1,\rho_2,\ldots,\rho_x\}^T \in R^n, \quad (x=1,\ldots,N_e)\\
&\quad \min\left\{W=\int_S I_n dS=\int_S \frac{1}{2}\mathrm{Re}(p_f v_n^*)dS\right\}\\
&\quad \text{Subject to}:\\
&\qquad M\ddot{X}+C\dot{X}+KX=F_a e^{i\omega t}\\
&\qquad \nabla^2 p+k^2 p=0\\
&\qquad \sum_{x=1}^{N_e}\rho_x V_x \le \lambda V_0\\
&\qquad 0\le\rho_{\min}\le\rho_x\le\rho_{\max}\le 1, \quad (x=1,\ldots,N_E)
\end{aligned} \tag{1}$$

where ρ_e denotes the volumetric density of the stiffer material in element e and plays the role of the design variable in the sensitivity analysis. The symbols p_f and v_n in the expression for W represent the acoustic pressure and the normal velocity on the structural surface, and p denotes the corresponding vector of amplitudes of the acoustic pressure on the structural surface S. * is a symbol for the complex conjugate. The first constraint equation is the dynamic equation in which $X=X_a e^{-iwt}$ denotes the structural displacement response vector, K, C, and M represent the stiffness, damping, and mass matrices of the N dimensional structure, respectively. The second constraint equation is the governing differential equation in steady-state linear acoustics. λ is the maximum total fraction of stiffer material in the structure and N_e denotes the total number of elements. V_0 is the volume of the admissible design domain.

2.1. Finite element analysis of structural vibration

For an arbitrary shape structure, the motion equation with time harmonic mechanical load can be expressed as

$$M\ddot{X}+C\dot{X}+KX=F_a e^{i\omega t} \tag{2}$$

It is assumed that material damping can be regarded as the proportional damping and the fluid-structural coupling is weak coupling which can be neglected, especially for air and wide or open spaces.

It is assumed that displacement response on time harmonic mechanical loading can be expressed as

$$X=X_a e^{-i\omega t} \tag{3}$$

Therefore, the amplitude of displacement response can be stated as

$$X_a=\left(K+i(\alpha K+\beta M)-\omega^2 M\right)^{-1}F_a \tag{4}$$