

Structural Optimization Method and Techniques to Reduce Radiation Noise

2011-01-1505

Published
05/17/2011

Iku Kosaka, Juan Pablo Leiva, Brian Watson and Phani Adduri
Vanderplaats R&D Inc.

Takanori Ide
Aisin AW Co Ltd

Copyright © 2011 SAE International

doi:[10.4271/2011-01-1505](https://doi.org/10.4271/2011-01-1505)

ABSTRACT

A methodology to optimize sound pressure responses of a structure, producing a radiation noise due to structural vibration, is presented. The method involves a finite element analysis module to calculate structural vibration, an optimization module to perform sensitivity analysis and structural optimization, and an acoustic module to compute acoustic transfer vectors. The proposed design system is successfully implemented and is demonstrated in the paper using several example problems.

INTRODUCTION

Radiation noise coming from vibrating objects is not only a nuisance but also can cause fatigue to human beings. As consumers are becoming more sophisticated and are paying more attention to unwanted sound, industries are trying to respond to make quieter products. There are many effective ways to control and reduce noise by introducing vibration isolation mechanisms, as well as active and passive control techniques [1]. However, such approaches require additional materials, parts, or equipment and can be costly. The most cost effective approach to reduce the radiating noise can be designing the structures to produce less structural vibrations that affect important frequency ranges for noise [2,3]. The challenge is finding such structures or structural modifications.

In this paper, we describe an efficient and effective design process to optimize structures using sound pressure responses. The proposed design process integrates finite element (FE) analysis and boundary element (BE) analysis,

computes sound pressure responses, and performs structural optimization in FE space. In recent years, the structural optimization has been improved quite dramatically to solve large-scale optimization problems [4,5,6]. It has become easy to use, more flexible, and powerful [5,6]. Structural optimization is one of the most reliable techniques to solve such challenging problems. This design process framework is implemented in Genesis® [7] and Virtual.Lab® [8].

The rest of the paper is organized as follows: first, the optimization process to perform sound pressure and sound power optimization is described. Then, analysis and optimization that are used in the design process, including frequency response analysis, acoustic analysis, adjoint sensitivity analysis, various optimization techniques, and beta method, are discussed. After the design considerations and the detailed model descriptions of examples are explained, the objective and constraint functions are defined in each example. For each example problem, a suitable optimization technique is selected and the results are shown. In conclusion, the proposed process and benefits are summarized.

ANALYSIS AND OPTIMIZATION

In order to perform the proposed design process, two separate but equivalent discretized models must be created first. These are: the boundary element (BE) model, used to compute the acoustic transfer vector (ATV) [9], and the finite element (FE) model, used to compute surface velocity from modal frequency response analysis. In the boundary element analysis, as model size increases, computational costs dramatically increase. Therefore, a relatively coarse model is commonly employed. On the other hand, a finite element

model must be constructed precisely to represent geometry since structural optimization will be performed using the FE model. It must also predict the desired frequency range and be well correlated to the physical model.

Since acoustic analysis is performed in BE space, computed ATV cannot be directly used in FE based structural optimization. ATV in BE space must be transferred to ATV in FE space before the structural analysis and optimization is performed. Although FE and BE meshes are separately created and described by different node and element numbers, they represent the identical physical model. Therefore, the mapping file to describe corresponding nodes between coarse BE model and fine FE model can be created. After the mapping file and ATV in BE space are obtained, the ATV in BE space can be converted into ATV in FE space.

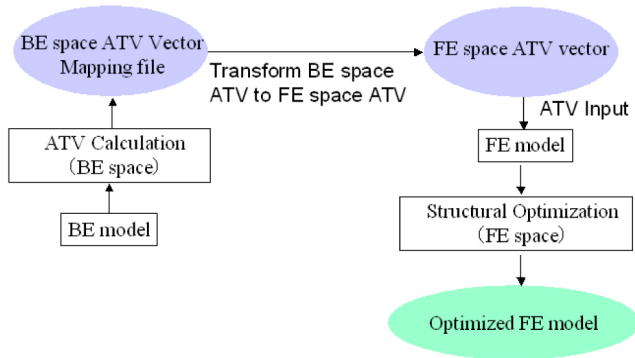


Figure 1. Schematic flow of sound pressure optimization process

Figure 1 shows schematic flow of the design process mentioned above. Once ATV in FE is constructed, it is treated as constant data entry to the FE analysis. Finally, structural optimization will be performed to improve sound pressure responses. Typical structural optimizations involve Topology, Sizing, and Shape optimization techniques as described in the following section in detail, but the focus in this paper is placed on the Sizing and Shape optimizations and small grid movements are assumed in Shape optimization. Therefore, BE mesh remains the same and the ATV can be assumed as constant.

MODAL FREQUENCY RESPONSE FORMULATION

In the discretized finite element space, the governing equation for the dynamic frequency response can be defined as follows:

$$[M]\{\ddot{u}\} + [B]\{\dot{u}\} + [K]\{u\} + i[K_s]\{u\} = [P] \quad (1)$$

where $[M]$ is mass matrix, $[B]$ is viscous damping matrix, $[K]$ is stiffness matrix, $[K_s]$ is structural damping matrix, $[P]$ is

load vector, and $\{u\}$ is dynamic displacement. Assuming the loading vector and dynamic displacement be periodic functions, we can rewrite the [equation \(1\)](#) as following

$$(-\omega^2[M] + i\omega[B] + [K] + i[K_s])\{u\} = [P] \quad (2)$$

Solving the eigenvalue problem in [equation \(3\)](#) yields natural frequencies and mode shapes $[\Phi]$ that can be used to reduce [equation \(2\)](#) to modal space:

$$([K] - \lambda[M])[\Phi] = [0] \quad (3)$$

The following is the governing equation of a modal dynamic frequency problem:

$$(-\omega^2[m] + i\omega[b] + [k] + i[k_s])\{z\} = [c]\{z\} = [p] \quad (4)$$

where,

$$[c] = -\omega^2[m] + i\omega[b] + [k] + i[k_s], \quad (5)$$

$$[m] = [\Phi]^T [M] [\Phi],$$

$$[b] = [\Phi]^T [B] [\Phi],$$

$$[k] = [\Phi]^T [K] [\Phi],$$

$$[k_s] = [\Phi]^T [K_s] [\Phi],$$

$$[p] = [\Phi]^T [P],$$

$$\{u\} = [\Phi]\{z\}$$

Dynamic velocity $\{v\}$ can be easily obtained by solving [equation \(4\)](#).

ACOUSTIC ANALYSIS FORMULATION

The governing equation of sound pressure is expressed as Helmholtz equation and in the direct boundary element formulation, Helmholtz integral equation is expressed as [1,10,11]:

$$p_F = - \int_S \left(i\rho\omega G v_n + p_s \frac{\partial G}{\partial n} \right) dS \quad (6)$$

$$G(r) = \frac{e^{-ikr}}{4\pi r} \quad (7)$$

where p_F is the complex sound pressure in the field of a source, G is the free space Green's function, n is the unit normal on the surface of the radiating body, S directed from the acoustic domain. v_n is surface normal velocity and p_S is sound pressure on the boundary surface S .

Once the [equation \(6\)](#) is discretized using boundary element space, it can be rewritten as:

$$p_{F,i} = [ATM]\{v_n\} \quad (8)$$

where $[ATM]$ is acoustic transfer matrix and $\{v_n\}$ is surface velocity vector which can be obtained by solving [equation \(4\)](#). For a single field point, [equation \(8\)](#) can be expressed as:

$$p_{F,i} = \{ATV\}_i \{v_n\} \quad (9)$$

where $\{ATV\}_i$ is acoustic transfer vector and represents the i -th row of the $[ATM]$.

SENSITIVITY ANALYSIS

The sensitivity of the modal dynamic displacement can be obtained by differentiating the [equation \(4\)](#) as follows:

$$\frac{d\{z\}}{dX} = [c]^{-1} \left[\frac{d[p]}{dX} - \frac{d[c]}{dX} \{z\} \right] \quad (10)$$

where, X is design variable vector and $[c]$ is defined in [equation \[5\]](#). The sensitivity of a response R with respect to design variable vector X can be expressed as the following equation:

$$\frac{dR}{dX} = \frac{\partial R}{\partial X} - \frac{\partial R}{\partial \{z\}} \frac{d\{z\}}{dX} \quad (11)$$

By substituting [equation \(6\)](#) into [equation \(12\)](#), it can be rewritten as:

$$\frac{dR}{dX} = \frac{\partial R}{\partial X} - \frac{\partial R}{\partial \{z\}} [c]^{-1} \left[\frac{d[p]}{dX} - \frac{d[c]}{dX} \{z\} \right] \quad (12)$$

If the operation $[c]^{-1} \left[\frac{d[p]}{dX} - \frac{d[c]}{dX} \{z\} \right]$ is done first, the sensitivity process is called the direct method. If the operation

$\frac{\partial R}{\partial \{z\}} [c]^{-1}$ is done first, on the other hand, the process is called adjoint method or adjoint sensitivity. As one can see, if the number of independent design variables (NDV) is bigger than the number of retained responses (NR), the adjoint method is more efficient than the direct method. Since the number of field points where the sound pressures are calculated is small for a typical sound pressure optimization, adjoint sensitivity is used in most cases.

OPTIMIZATION TECHNIQUES

The structural optimization techniques can broadly be categorized into the Topology, Size, and Shape optimizations. In this paper, however, Sizing, Shape, Topometry, Topography, and Freeform optimization techniques are extensively tested and used to solve problems. Therefore, we briefly summarize those techniques, as well as a useful and necessary technique called beta method.

Sizing and Topometry Optimization

Sizing optimization is a method to design the specific size of structural properties (e.g. thickness of a shell, cross sectional dimensions of bars, etc.). In the finite element code, collections of elements are commonly referencing to one property entry. In the Sizing optimization, therefore, one design variable controls the specific property value of all elements that are designed. On the other hand, Topometry optimization [5] is a technique to design property values by element bases rather than property entry, and properties and design variables for designed elements are internally created. Topometry can be used with manufacturing constraints (such as symmetry) and/or design element coarsening technique. Topometry is easy to use and is a very powerful technique that often yields innovative solutions for some classes of problems.

Shape, Topography, and Freeform Optimization

Shape optimization is a method to design the shape of a structural boundary (i.e., designing the locations of the grids) typically using user specified basis vectors. Conversely, Topography optimization [6] does not require any basis vector or direction of grid movements and it internally creates design variables. The basis vectors are also internally created for a normal to the surface direction as a default. Topography optimization can be applied only to the shell elements and was originally intended to design shell surface such as bead pattern, however, solid element surfaces can also be designed by applying a thin shell on top of a solid surface. On the other hand, Freeform can be applied to any arbitrary grid surface (e.g. directly to solid element surface). It automatically assigns independent design variables to basis vectors. If the desired shape of basis vectors as well as where to apply those shape changes are known, standard Shape optimization can be an efficient and cleaner approach. If the boundary surfaces