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TOPOLOGICAL OPTIMIZATION OF AN ACOUSTIC PANEL UNDER PERIODIC LOAD BY SIMULATION

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Abstract: *In the present paper a panel optimization example presents the potential of the topological optimization to support panel structural improvements for optimal dynamic response of acoustic panels under periodic load. An acoustic panel with some peculiar boundary constraints, has been considered for optimization. In order to limit the elastic deformation of the structure subjected to periodic load, ribs have to be added in the proper areas of the panel. After the optimization iterations, the maximal accepted elastic deformation is not exceeded for all the harmonic excitations performed at the natural damped frequencies in the frequency band of interest. At the predicted areas, the designer will propose a second optimization of size or shape type, that gives the ribs the final geometry.*
Keywords: *acoustic panel, frequency response function, topological optimization.*

1. INTRODUCTION

Acoustic panels have a number of properties in terms of noise attenuation. They can be used both indoors as in classrooms, studios, community centers, laboratories, corridors and so on, as well as outside. Multiple noise sources can be represented by simple speakers or human voice, industrial or laboratory equipment and even road, rail or air lines. The construction types of barriers encountered nowadays are made of various materials, including their matching to better fulfill the objective of attenuation or absorbing sound energy. A road barrier is formed from a concatenation of acoustic panels with fixed dimensions. The thickness of a panel or barrier may be very small for metal structures or combined with an air cavity and can reach up to several centimeters such as walls on the road. Noise barriers have been used extensively for several decades, along the highways for controlling traffic noise. The free space above the screen combined with the diffraction at the structural edges permit the sound to reach the receiver placed on the other side of the barrier. Lately, highly effective and aesthetical sound barrier systems for road and rail have been developed. As an example, individual aluminum elements with visible perforated sheets can be mentioned. Other materials they can be built of are wood, masonry, concrete, stucco etc. Because of structural and aesthetic reasons their height does not exceed eight meters.

Noise barriers are solid obstructions built between a traffic road and buildings along the highway. The role of a barrier is to reflect sound away from specific areas and its effectiveness can reduce noise levels by ten to fifteen decibels within finite regions [12], [14].

For enclosed spaces, acoustical panels can be placed on the ceiling or floor and on the walls of the room. Their size varies depending on the mounting space and the acoustic requirements.

Absorbent panels, unlike the porous one, are effective for low frequencies. By mounting them at some distance from a rigid wall will produce a resonant absorber that can be made of aluminum or plywood. Fundamental resonance frequency is determined by the mass per unit area of the panel and the thickness of the air layer. Higher order modes of vibration of a panel are not making such a good attenuation as the fundamental mode due to radiation resistance which is much smaller than the resonance frequency. It is difficult to predict the performance of a panel absorber due to its dependence on modal acoustic properties of the space in which they are located. Any flexible panel absorbs acoustic energy at low frequencies because it responds to incident sound and dissipated it by friction, mainly at the edges [1].

The structural optimization process in a simplified version can be schematized as in Figure 1 and presented as follows. The design variables are characteristic quantities, not predefined, that describe and define the sistem subjected to optimization. Constraints are

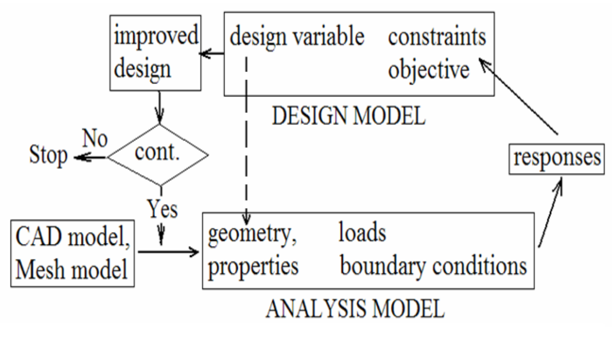

Fig. 1. FEA optimization flow chart

limiting the design variables variation. The design variable vector X and the associated constraints are defining the n-dimensional design space (1):

$$X = [x_1, x_2, ..., x_n]^T, \qquad x_i^L \le x_i \le_i^U \qquad (1)$$

The design model can be modified by the optimization engine in order to find an optimal design from the specified point of view. The design variables variation are determining the geometry and the properties p_i of the analysis model:

$$p_i = c_0 + \sum_j c_j x_j \qquad (2)$$

The analysis model is a concrete model that can be tested by runing the proper finite element solver (Nastran, Optistruct, Radioss etc). The sistem under analysis is loaded and constrained by boundary conditions. This run generates the responses under observation of the designer like the center of gravity, moments of inertia, volume, weight, eigenvalues, displacements, stresses, strains, forces, frequency response functions, acoustic pressure and so forth.

The objective function is defined to be one of the responses of the sistem which in general is maximized or minimized. The rest of the responses are limited in values by equality or inequality constraints.

Structural topology optimization is a high level optimization method that produces an optimized shape and material distribution for a finite element discredited structure within a given package space. The optimization algorithm calculates iteratively the material properties for each finite element, altering the material distribution in order to optimize the objective under given constraints [2], [15].

2. THE DYNAMIC LOAD AND THE PANEL RESPONSE

A periodic load coming from the exterior is acting on the panel at a specific location. The displacement of the panel is restricted by the proper functioning of the acoustic phenomena that take place on the panel perforations. The elastic deformation of the panel can be limited by adding ribs on the optimal areas of the panel and by adding extra displacement constrains on the panel margins.

Let us consider an acoustic panel of rectangular shape 2000 x 1000 mm. The mesh of the panel is based on shell elements.

Displacement constraints are imposed in nodes placed on the panel edges. Some of the constraints are blocking all six degrees of freedom (translations and rotations) and some of them are eliminating only the translation about the OZ direction (degree of freedom #3) normal to the panel.

The dynamic load is acting on the panel as follows. At the node number #4891 on the right side of the panel the dynamic load is applied normal to the panel (dof 3). The variation of the load is harmonically.

The force amplitude is decreasing with the increase of the frequency, starting with 4 N at one Hz and ending with 3 N at 1000 Hz.

First the normal mode analysis of the panel is done and the natural modes (natural frequencies and mode shapes) are obtained for the first 20 modes estimated to be in 0 to 1000 Hz frequency interval [5],[10]. Because the response amplitudes are critical at the structural resonances, the excitation frequencies f_i of the panel will be selected to be at the damped natural frequencies and around them.

$$f_i = c_i * f_{d\,j} \tag{3}$$

where $f_{d\,j}$ are the damped natural frequencies for $j=1,\ldots,20$ and c_i are the coefficients: 0.6, 0.8, 1, 1.2.

The frequency response function of

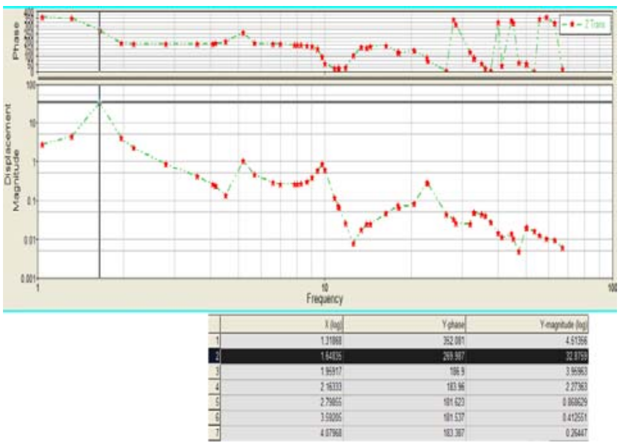

Fig. 2. Acoustic panel frequency response; phase (lin-log) and magnitude (log-log) versus frequency

compliance type is observed [3],[4],[7]. The excitations are by harmonic forces and the responses are of harmonic displacements. The modal response is observed, therefore the response is the sum of the participation of each mode of vibration to the total response instead of the direct integration.

The frequency response function for the initial (the baseline) structural design that gives the displacement of the node #3 as response to the excitation on the node #4891 on the frequency band 0 Hz to 1000 Hz, at the first 20 damped natural frequencies and around, is depicted in Figure 2.

The frequency axis for both magnitude and phase is in logarithmic scale. The magnitude of the response is in logarithmic scale in order to be able to observe both the small and the large magnitudes on the same graph. The phase is represented in linear scale.

The maximum displacement over the frequency band of interest is obtained for the first damped natural frequency at 1.65 Hz and the magnitude of the response is about 32.8 mm. One can observe that the phase of the response at the first mode frequency excitation is about 270 degree (-90).

The estimated structural damping introduced in the finite element simulation is G=0.05.

For the Optistruct simulation a load case has been defined and the following cards have been used: RLAOD2 for the excitation or the dynamic load definition, DAREA for the location of the load, FREQ5 for the excitation frequencies selection, EIGRL for the modes extraction.

The magnitude of the response being unacceptably large an optimization is imposed in order to lower the displacement on the spot of interest.

3. TOPOLOGICAL OPTIMIZATION ON THE PANEL STRUCTURE

The target solution is to place optimal ribs over the panel surface in order to decrease the response magnitude to the dynamic load under a specified value. The ribs will be added over the initial thickness.

The design variables used for the topological optimization are the densities of the shell elements all over the panel. This variation can be seen as a thickness variation, as well. Some additional parameters are imposed to the shape of the proposed areas with altered density or thickness in order to help optimization engine to propose technological shapes. The density of the elements will vary between 0 and 1, while the thickness of the elements can be changed in parallel between the initial or baseline thickness of 3 mm and the maximal of 12 mm.

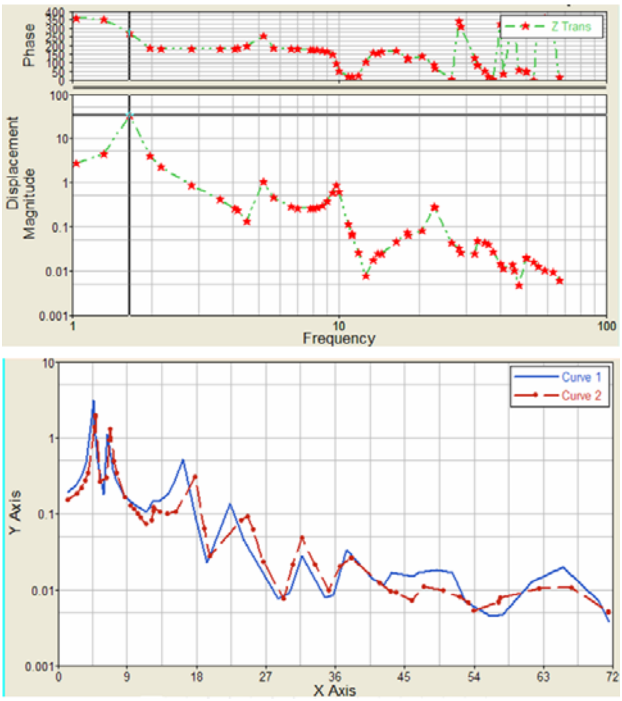

Fig. 3. First two optimization runs

Two responses out of each iteration are proposed. One is the displacement along the

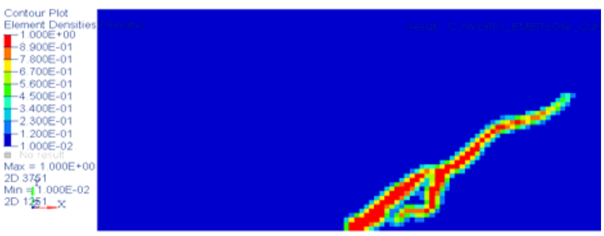

Fig. 4. Density variation over the panel; a unique area with increased thickness

OZ axis at the node of interest and the second is

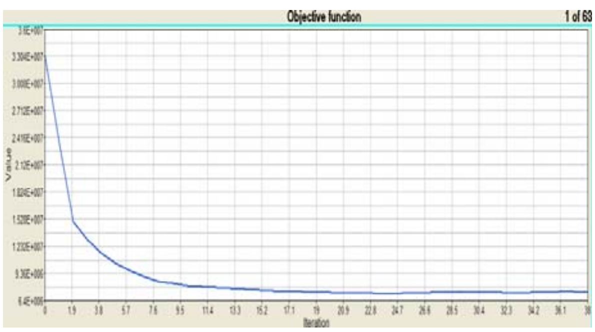

Fig. 5. Volume change over 38 iterations

the total mass of the panel. The magnitude of the response displacement is observed for all excitation frequencies.

The only one constrain is imposed to the first response mentioned before, to be less than 2 mm for each optimization iteration.

The objective function is the second response and is intended to be minimized over the optimization iterations.

After the first two runs in which the displacement constrains have been limited to 10 mm (Curve 1) and 2 mm (Curve 2) respectively, the frequency response functions obtained in comparison to the baseline are depicted in Figure 3.

The peak abscissa is smaller than the natural frequency.

For the target constraint of 2 mm imposed to the FRF displacements, after 38 iterations the density variation for the whole panel is depicted in Figure 4. The associated objective function is minimized as it can be seen in Figure 5.

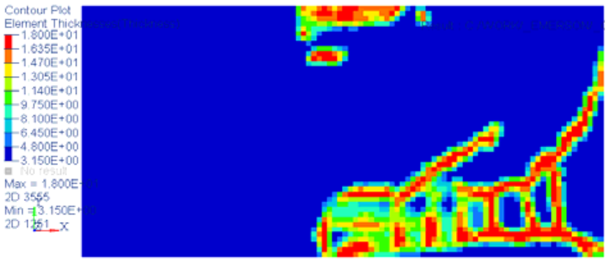

Fig. 6. Element thickness variation for maximum one mm displacement; two areas where ribs can be added

For the third run in which the target of one mm maximal displacement has been imposed, the thickness variation over the whole panel is depicted in Figure 6. The first area of increased thickness is larger and a second extra area on the upper edge can be noticed.

Comparatively one can observe the FRF for the initial panel model (Fig. 3/ upper, log-log

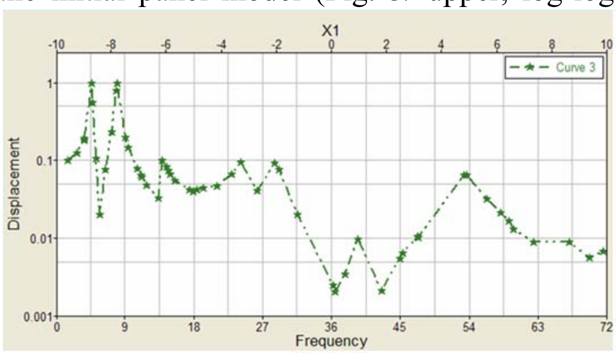

Fig. 7. Third run, max displacement = 1 mm for the first two resonant peaks

FRF magnitude versus frequency graph) and the FRF after the third optimization run (Figure 7). The FRF magnitude of deformation for the first two panel resonances under 9 Hz, are equal to unity. The rest of the peaks are lower in magnitude and are associated to the higher panel resonances.

4. CONCLUSIONS

The panel optimization examples prove the potential of topology optimization to support panel structural improvements for optimal dynamic response of acoustic panels under periodic load. An acoustic panel, with some peculiar boundary constrains, has been considered for optimization.

The interest is to add ribs in the proper locations in order to limit the elastic deformation of the structure, subjected to periodic loads in a frequency band, by minimizing the total mass. The topology optimization is proposed. The deformation limit is obtained for the excitation at the first two natural damped frequencies.

For the rest (the higher) of the observed frequencies the amplitude of the elastic deformation is lower. In the predicted areas it follows the designer to propose a second size or shape optimization that gives to the ribs the final geometry.

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Optimizarea topologică prin simulare a unui panou acustic supus unei forțe periodice

Rezumat: In prezenta lucrare este aplicată optimizarea topologică unui panou acustic dreptunghiular, prins în puncte pe contur, în vederea imbunătățirii răspunsului dinamic al panoului la excitație cu o forță periodică într-o bandă de frecvențe. La fiecare iterație din cadrul procesului de optimizare este modificată grosimea elementelor finite astfel încât deplasarea în punctul de control sub acțiunea forței periodice de excitație să fie limitată la mărimea țintă. Odată cu modificarea grosimii în zonele cele mai influente sunt afectate frecvențele de rezonanța ale panoului. Noile frecvențele de rezonanță ale structurii sunt prin urmare determinate la fiecare iterație deoarece poziționarea acestora afectează răspunsul la excitație. Zonele cu grosime crescută indică locațiile cele mai sensibile în care se vor adăuga nervuri de rigidizare. In etapa a doua se poate aplica o optimizare de mărime pentru găsirea grosimii potrivite a nervurilor. Deplasarea maximă permisă asigură conservarea parametrilor acustici ai panoului în condițiile de prindere adoptate.

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