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PUBLISHER

European Acoustics Association (EAA)

VERSION

VoR (Version of Record)

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Radiated sound power of automotive-type panels with dome-shaped indentations

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Summary

The automotive industry has witnessed a trend in recent years of reducing the bulk weight of a vehicle in order to achieve economical fuel consumption. Unfortunately, reduced bulk weight often compromises the noise, vibration and harshness (NVH) characteristics of the vehicle. This paper investigates the effect of using dome-shaped indentations on a flat panel in order to reduce the sound radiation whilst keeping the weight of the panel constant. The dimensions and placement of the dome-shaped indentations are numerically optimized using a genetic algorithm technique so that the panel radiates minimum sound power in the frequency range of interest. Both numerical and experimental tests are performed on the optimized panel which is then compared to the radiated sound power of an equivalent flat panel. Tests were also performed on intuitively placed domes on the panels. The results illustrate the fluctuation in radiated sound power by varying the number and location of the dome-shaped indentations. The paper concludes by demonstrating the advantages of a fast and user-friendly optimization approach to passive noise control problems.

PACS no. 43.40.Dx, 43.40.Rj

1. Introduction

In today's refined car market, Noise, Vibration & Harshness (NVH) is inextricably linked to the decision making ability of the customers [1]. This highlights the fact that one of the major issues in body NVH is to tackle the panel radiation inside the vehicle's passenger compartment. A common engineering solution to tackle this problem is to perform numerical optimization on the panels. Different optimization methods can be selected based on the choice of objective function. A detailed review of general structural-acoustic optimization has been published by Marburg [2]. Many publications are available which are concerned with the minimization of structural-acoustic responses by modifying different parts of a vehicle body [3,4,5]. Some publications, considered structural-acoustic optimization from a geometrically idealised standpoint where simple structures, such as plates [6] and beams [7], excited by a point force are studied.

Publications [8,9,10] demonstrate the use of geometrical modifications to minimize the resulting acoustical responses.

The present paper describes the results of the research into minimization of radiated sound power of automotive-type panels with dome-shaped indentations. In any optimization problem, the analyst needs to specify at least one objective function and a number of design variables. In this paper, the objective function used is selected so as to minimize the sound power radiated from the panel over a given frequency range. The design variables are the location and the properties of dome-shaped indentations in the panel. The number of design variables is directly proportional to the time taken for the optimization to complete. This paper demonstrates the use of 6 design variables used to get one domed indentation and then based on symmetry produces the remaining indentations. This technique is shown to have a significant reduction in computation time with

solidarity in optimization results. However, this symmetry based approach is only suitable for symmetrically identical panels, e.g. rectangular and circular panels, etc. An equivalent flat panel is taken as a reference panel in order to observe any improvement achieved by making the indentations (domes). A comparison is also made to demonstrate how the number of domes affect the panel's radiating characteristics by intuitively placing first, one and then, two domes. It was observed that increasing the number of domes helped shift the natural frequencies of the panels, which could be useful in the case when targeting a specific mode or two.

2. Optimization process

In this paper, the test panel is assumed to be a part of a vehicle floor. The boundary conditions at the four edges of the panel are, thus, chosen in a particular way so as to simulate how the panel alone will behave as if it is a part of the vehicle. This is simulated by applying translational and rotational restrictions to the boundary nodes of the finite element (FE) model of the panel [4]. Using NASTRAN[®], this can be achieved by using BUSH elements at the boundary nodes. The code is generated in Matlab[®] and different function files are called upon to perform the appropriate tasks. The analyst does not need to leave the Matlab interface as the process is implemented via a GUI.

A genetic algorithm based optimization code is developed so as to optimize the modeshapes that create acoustic pressure cancellation in near-field of the panel. The cancellation occurs when the acoustic pressure of the neighbouring areas are of the same magnitude but opposite in sign. For this reason, symmetry is used for the location of the dome-shaped indentations. It remains up to the judgement of the analyst if the symmetrical placement of multiple domes would be beneficial for the problem at hand.

The analyst needs to define the number of domes required. However, the design variables for only one dome is needed since the remaining domes are placed based on the symmetry of the panel. The main advantage of this optimization procedure, in comparison with other commercially available software, is that this procedure does not require any input force data to perform the structural-acoustic optimization. The panel is assumed to be isolated from the remaining vehicle, thus, the optimization does not depend on any specific excitation or the rest of the vehicle structure. Hence, for automotive applications, when other body parts are redesigned, the panel under

consideration will still be optimized for minimum sound radiation. This method is also effective at low frequencies where noise reduction using added damping treatment is usually less effective.

2.1 Structural Analysis

The flat panel model is meshed in Matlab [11] and a 'bdf' file is generated for the analysis. The modal behaviour of the panel is calculated using NASTRAN as an FEM solver. The mode shapes give the FE nodal displacements at the resonant frequencies which are then converted into nodal velocities in order to calculate nodal pressure in the acoustic analysis part of the optimization code.

A series of iterations are required to achieve an optimum solution. The maximum number of iterations is set by the analyst depending upon the time constraint associated with the project. For the panel under consideration, a maximum of 20 iterations was considered sufficient to obtain the optimum results. All of the parameters defined at the beginning of the optimization are saved automatically for future use.

2.2 Acoustic Analysis

The objective function used in the optimization code is to minimize the sound radiation from the panel. Thus,

$$P = \sum_n P_n \quad (1)$$

where P_n is the sound power radiated by the n^{th} structural mode. NASTRAN solves for the structural eigenvalue problem and returns modeshapes with arbitrary amplitudes. For this reason, before calculating the total sound power, all modes are normalised in order to have a unit mean squared velocity. The method described in references [12,13] is followed in order to calculate the sound power of a structure based upon the normal velocity at each node. Considering u being the normal velocity of the n^{th} mode, its sound power is calculated by:

$$P_n = u^T \mathbf{B} u^* \quad (2)$$

where superscripts T and * indicate vector transpose and complex conjugate respectively, and

$$\mathbf{B} = \frac{1}{4} (\mathbf{A} + \mathbf{A}^H) \quad (3)$$

where the matrix \mathbf{A} can be constructed from the submatrices A_j . In this case, the same interpolation functions are used for both the structural and acoustic elements. Thus, A_j is given by:

$$A_j = Z_j^T \left(\int_{S_j} NN^T dS_j \right) \quad (4)$$

where N is the vector of interpolation functions of the element j , S_j is the area of the same element and Z_j is the submatrix of the radiation impedance matrix that yields the pressure at the nodes of element j . The radiation impedance matrix can be calculated using the influence matrices after conducting a boundary element (BEM) analysis [12,13]. The panel for which the sound power needs to be calculated is assumed to be quasi-flat; this simplifies the calculation of the impedance matrix significantly. The assumption is valid as long as the out-of-plane modifications of the panel are small compared with the acoustic wavelength.

3. Modification function (domes)

The modification is made onto the meshed model, thereby, changing the element and nodal positions in the existing 'bdf' file.

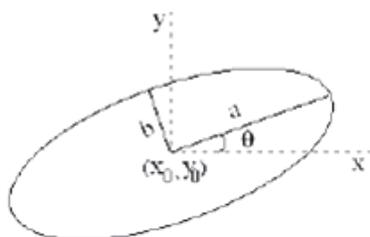


Figure 1. Geometry of an ellipse at a given angle θ with the X-axis.

For each node of the mesh, a check is made as to whether or not the node falls inside the domain of the modification. This is done using the equation of an ellipse with parameters defined in Fig. 1,

$$\frac{\{x \cos(\theta) + y \sin(\theta)\}^2}{a^2} + \frac{\{y \cos(\theta) - x \sin(\theta)\}^2}{b^2} \quad (5)$$

where $x = x_n - x_o$, $y = y_n - y_o$ are the distances of the n^{th} node to the centre of the ellipse in the X- and Y-dimensions. When this equation is negative, the node falls within the modification domain of the dome. The Z-coordinate of this node can then be set according to its distance from the centre of the ellipse. A maximum height of the dome at (x_o, y_o) can also be defined. In total, five design variables are used define the ellipse's geometry and one design variable defines the maximum height of the dome.

4. Results

In the course of the experimental investigations, the domes are intuitively placed on the flat panel, to investigate how the number of domes will affect

the radiating characteristics from different panels. A flat plate of equivalent dimensions (307mm x 208mm x 1.2mm) is taken as a reference plate. Each panel is mounted, in turn, into a large concrete baffle and excited using an electrodynamic exciter. All the experimental tests are performed within an anechoic chamber. An overall reduction of 5 dB was observed on the one-domed panel, compared to its equivalent flat panel over the frequency range 0 to 1000 Hz. The measured sound power against frequency of the panels is shown in Fig. 2.

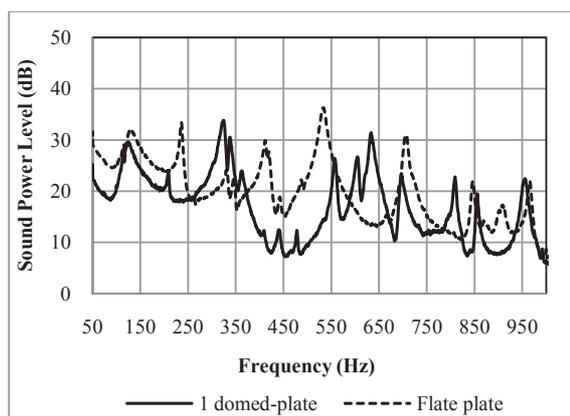


Figure 2. Sound power comparison between a flat plate versus an equivalent panel with 1 dome.

Another distinct observation was made with the two 2-domed panels; one with 2-domes placed adjacently to each other, the other with diagonally opposite domes as illustrated in Fig. 3.

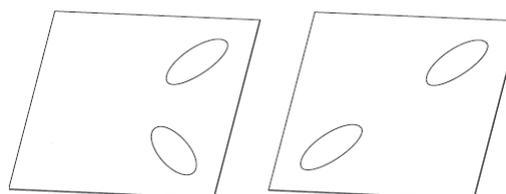


Figure 3. Two adjacent and diagonal domes, respectively.

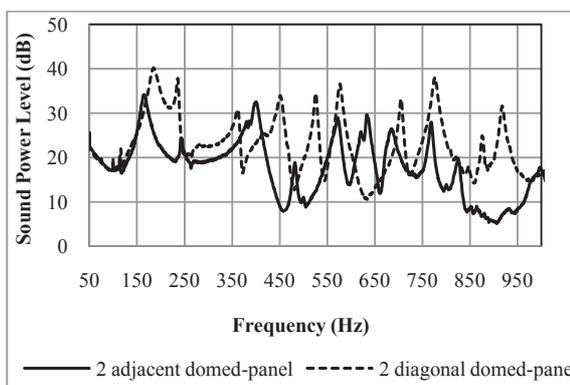


Figure 4. Sound power comparison between a panel with two domes but placed at different locations.

Fig. 4 depicts the difference in radiation characteristics of the two adjacently and diagonally placed domes. It is observed that the panel with two adjacent domes radiates less than the panel with two diagonal domes, in the degree of around 5 dB overall. This method is effectively used to shift modes beyond the frequency range of interest. This is demonstrated by the numerical analysis performed on an optimized plate, and compared with its equivalent flat plate, as shown in Fig. 5.

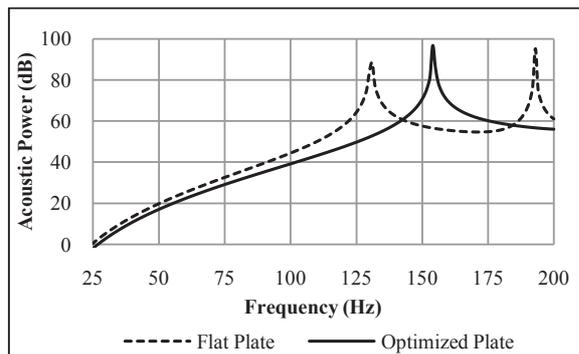


Figure 5. Acoustic harmonic BEM analysis of a flat plate and an optimized plate with similar boundary conditions.

The predicted results shown in Fig. 5 are from a panel optimized for 10-200 Hz to include 5 domes with all its edges clamped. The inclusion of domes has shifted the second acoustic mode out of the defined frequency range.

5. Conclusion

This paper has demonstrated the usefulness of geometrical modifications in the form of elliptical indentations made on a flat panel which help increase the stiffness. One needs to be cautious when increasing the stiffness of the panel because the stiffness, if inappropriately increased, would make the structure radiate even more sound. The general idea is to target only few specific modes, encompassing a defined frequency range the acoustic response of which can easily be controlled using the proposed optimization method.

In this paper a methodology is proposed for automotive-type panel optimization, by providing geometrical modifications, which are suitable for any load cases since the structural modes are independent of any specific point excitation. This approach is a computationally fast solution to passive noise control issues, where optimally designed panels are assumed not to be affected if the other neighbouring structures are re-designed.

The optimization code is programmed in a flexible manner and it can include different objective functions. One of the key factors to be kept in consideration whilst running optimization is to accurately define the boundary conditions. The results could be expected to improve if the appropriate boundary conditions are applied in terms of translational and rotational stiffness data. Overall, this optimization method could be a useful design tool in the early product development stage.

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