Numerical modelling techniques to optimise automotive-type panels for reduced sound radiation

This item was submitted to Loughborough University's Institutional Repository by the/an author.


Additional Information:

- This is a conference paper. Permission to deposit the paper has been granted by the University of Sussex, publishers of the proceedings of the International Symposium on the Computational Modelling and Analysis of Vehicle Body Noise and Vibration, University of Sussex, Brighton, UK, March 2012.

Metadata Record: https://dspace.lboro.ac.uk/2134/9685

Publisher: © University of Sussex

Please cite the published version.
This item was submitted to Loughborough’s Institutional Repository (https://dspace.lboro.ac.uk/) by the author and is made available under the following Creative Commons Licence conditions.

For the full text of this licence, please go to:
http://creativecommons.org/licenses/by-nc-nd/2.5/
NUMERICAL MODELLING TECHNIQUES TO OPTIMISE AUTOMOTIVE-TYPE PANELS FOR REDUCED SOUND RADIATION

G. Kumar, S.J. Walsh and V.V. Krylov
Department of Aeronautical and Automotive Engineering, Loughborough University, Leicestershire, United Kingdom, LE11 3TU.
G.Kumar@lboro.ac.uk

ABSTRACT
Numerical predictions are becoming ever more important in automotive development when analysing the Noise, Vibration and Harshness (NVH) performances of vehicles. In the low- to mid-frequency range, vibro-acoustic predictions are generally performed using the finite element method (FEM) and/or boundary element method (BEM). In this paper, a numerical optimisation technique is described that aims at reducing the sound radiation from automotive-type panels over the frequency range of interest. The objective function, i.e. the radiated acoustic power, is calculated with a quadratic equation in terms of surface velocities. The genetic algorithm (GA) based optimisation aims to minimise the value of the objective function by modifying the normal-direction (Z-direction in an X-Y plane) of a few nodes of the finite element model of the panel, thereby imposing a geometrical change in the panel. The equation of an ellipsoid is used to smooth out the discontinuity after the change in the nodal coordinates and, hence, dome-shaped indentations are obtained. The resulting panel designs are analysed, both numerically and experimentally to support the approach presented.

1. INTRODUCTION
For automobiles, one of the major issues in vehicle body Noise, Vibration and Harshness (NVH) is to reduce the panel sound radiation inside the vehicle’s passenger compartment. The panels enclosing the vehicle body structure are designed to keep the body-in-white (BIW) mass low, which accounts for the efficient fuel economy. So, generally, these panels are made out of thin sheet metal, which by itself has a very low bending stiffness. It is a common practice to look for ways to increase the stiffness of these thin quasi-flat panels by introducing ribs, stiffeners or beads. An audit of different brands of automobiles currently in production shows that the body panels are designed with a wide variety of beading configurations, ranging from circular or elliptical beads to criss-crossed swages as illustrated in Fig. 1. Depending on the configuration of indentations, the structural-acoustic characteristic of a panel can be altered significantly. For example, the frequency of lower-order modes can be shifted, the isotropic property may change to orthotropic, the modal density can be decreased by shifting the modes to a higher frequency zone and the radiated sound power can be decreased in the frequency range of interest [1,2,3].
Fig. 1: Examples of criss-crossed swages and elliptical domes on the body panels of a vehicle.

Except for the reinforcement of panels or the application of damping pads, it is difficult to find intuitive countermeasures that can reduce the structure-borne noise in the intermediate frequency range from 100 to 300 Hz where the booming and low frequency noise occur. Thus, a common engineering approach is to perform numerical optimisation on each panel separately, and observe its behaviour once it becomes the part of the whole vehicle body. The objective of this paper is to describe a numerical optimisation technique to design such automotive panels for reduced sound radiation. The reduction in sound radiation is achieved by stiffening the panel by indenting it with elliptical domes. In order to justify the increment in stiffness, a comparison is made with another plate in which the domes have been placed intuitively. It is demonstrated that the number of domes and their placement interfere with the low to mid frequency nodal lines and, thus, alter the isotropic nature of the flat panel.

Different optimisation methods can be selected based on the choice of the objective function. Many publications are available which are concerned with the minimisation of structural-acoustic responses by modifying different vehicle body [4,5,6]. A detailed review of general structural-acoustic optimisation has been published by Marburg [7]. In any optimisation problem, the analyst is required to specify at least one objective function, which needs maximising or minimising, and a number of design variables. In this paper, the objective function is to minimise the sound power radiated from the panel over a given frequency range of interest. There are six design variables, which account for the location and dimension of the dome-shaped indentations. The number of domes required is a design constraint and can be set as per the analyst’s decision. The technique described in this paper is based upon the optimal placement of one dome and then based upon symmetry, the placement of remaining domes. This particular technique is shown to have a significant reduction in computation time. This paper presents graphical results that illustrate the variation in acoustic response for panels with one, two or four domes.

2. OPTIMISATION PROCESS

In the optimisation process described in this paper, it is assumed that the rectangular test panel is an integral part of the vehicle body. This is achieved by applying translational and rotational limitations to the boundary nodes of the finite element (FE) model of the panel [5]. Retaining the generality of unspecified forcing of the structure, the panel is optimised for a general response
condition. Thus, the optimisation is based upon the eigenvectors calculated during the structural analysis by solving the eigenequations using NASTRAN®. For the optimisation, an FE model of the structure is required that contains the grid points (nodal coordinates), element information and the type of solution required (SOL103 in this case) [8]. The optimisation code is generated in Matlab® with the objective function to reduce the sound power over a set series of iterations. The maximum number of iterations to be performed is fixed before initiating the optimisation.

2.1 Structural Analysis
The optimisation technique described in this paper is based on a real eigenvalue analysis or normal mode analysis. The first requirement for the optimisation is an FE model of the test structure. In our case, the FE model comprises of 2666 grid points accounting for 2562 quadrilateral plate elements which sufficiently satisfies the theory of at least six elements per wavelength for the maximum frequency value of interest [9]. The finite element data of the structure can be exported into a text-type file, generally with an extension .bdf, which is read into the Matlab® environment. In the ‘bdf file’, the material property (steel) for the test panel and the mesh-element (CQUAD4) type is defined, but the boundary conditions need to be specified additionally by defining the translational and rotational stiffness values along the boundary nodes.

In the FE model, each node has six degrees-of-freedom. So, for each node, there are a total of six eigenvectors, three of which define the translation in the X, Y and Z axes and the remaining three define the rotation about the X, Y and Z axes, for every eigenvalue (or natural frequency). The displacement along the three translating coordinate axes is differentiated with respect to time to obtain the velocity in their respective directions. The resultant velocity in the direction normal to the surface, when squared, is used to find the nodal acoustic power, later in the acoustic analysis. The calculation of eigenvectors is restricted for a set frequency range of interest, i.e. 10-1000 Hz, to target normal modes up to 750 Hz.

2.2 Acoustic Analysis
The objective function used in the optimisation code is to minimise the sound radiation from the panel. Since the panel has been discretised into finite elements, each node on the FE model is assumed to represent a monopole noise source. The total acoustic power radiated by a collection of noise sources is then expressed using a quadratic equation in terms of the source strengths or surface velocities. The structure is assumed to be vibrating harmonically and the quadratic acoustic power expression is derived by using the boundary element method applied to the Helmholtz equation. The discretised structure requires some interpolation functions to be used to map nodal values over the elements, and facilitate the integral evaluations [10]. The numerical evaluation of the Helmholtz integral equation then leads to an algebraic system of equations,

\[ Dp = Mv, \]

where \( p \) and \( v \) are the acoustic pressure and surface velocity, respectively. \( D \) and \( M \) are coefficient matrices derived from integration of the normal derivative of the Green’s function over a surface and the integration of the Green’s function over a surface, respectively. Pre-multiplying Eq. (1) by \( D^{-1} \) yields
where the matrix product on the right-hand side defines the impedance matrix, \( Z = D^{-1} M \). The individual elements in the matrix \( Z \) represent the contribution to the pressure at a given node due to a unit velocity at another node. For example, \( z_{ij} \) represents the contribution to the acoustic pressure at node \( i \) due to a unit velocity at node \( j \) [11].

The objective function to be minimised, \( P \), is calculated from the summation of the power radiated by each individual element, \( P_j \), on the surface of the radiator,

\[
P = \sum_{j=1}^{N_{el}} P_j = \frac{1}{2} \text{Re} \left\{ \sum_{j=1}^{N_{el}} \int_{S_j} p_j v_j^* dS \right\},
\]

where \( N_{el} \) is the total number of elements in the FE model, \( S_j \) is the area of an element and \( S \) is the enclosed surface in space. Using the impedance matrix relation, \( Z \), and substituting with pressure, \( p \), using Eq. (2) as well as the same interpolation functions for the pressure and velocity as used for the boundary element solution leads to [10]

\[
P_j = \frac{1}{2} \text{Re} \left\{ v^T Z_j^T \int_{S_j} N N^T dS \right\} v_j^*,
\]

where \( N \) represents the vector of interpolation functions defined with respect to the element \( j \), \( v \) represents the vector of velocities on the entire structure, \( Z_j \) represents the submatrix of \( Z \) and \( v_j \) represents the vector of velocities on element \( j \) (the superscript * means complex conjugate). The dimension of the matrix \( Z \) is equal to the total number of nodes multiplied by the number of nodes per element. The integral in the Eq. (4) can be calculated separately as

\[
Z_j^T \int_{S_j} N N^T dS = A_j,
\]

such that \( A_j \) has same dimensions as the matrix \( Z \). Substituting Eq. (5) into Eq. (4), and then summing for all the elements, leads to the total radiated power,

\[
P = \frac{1}{2} \text{Re} \left\{ v^T A_1 v_1^* + v^T A_2 v_2^* + \ldots \right\}.
\]

Assembling all of the \( A_j \) submatrices into a single matrix, and all of the \( v_j \) vectors into a single column vector and rationalising the real component operator yields a compact form for the total radiated acoustic power from the discretised structure,

\[
P = v^T B v^* ,
\]

where \( B \) is Hermitian and is equal to [11]
2.3 Optimisation Strategy
A process based on genetic algorithm (GA) is suitable for the optimisation problem like this, where traditional solutions are not present or lead to unsatisfactory results. With the help of a GA, an exhaustive search over a relatively smaller search space can be performed within a reasonable amount of time. In this paper, the optimisation is limited to a maximum of 20 iterations. The optimisation starts with the six initial sets of random solutions provided, where each set represents the number of design variables to be optimised. This can be done by selecting random points in the space range of the structure, such as the centre of the domes. For the domes to stay on the structure, the random placement of domes is restricted by excluding the boundary nodes in the search space.

The origin of the genetic algorithm is based on the “survival of the fittest” concept. So, with every successive iteration, the design variables get closer to the optimised value. In other words, the fitter design variables replace the weak design variables from the previous iteration; this, in GA terms, is referred as breeding and killing [12]. A cumulative probability distribution is formed to make decisions for breeding new offspring and allowing less fit individuals to die. The fitness record is maintained based on the objective function, i.e. sound power calculations, for each set of individuals, which help in making decisions for breeding and dying.

2.4 Modification Function
The optimisation process will impart a modification to the meshed FE model, thereby, changing the element and nodal positions in the existing ‘bdf’ file. The geometrical domain of the modification function for domes can simply be defined by the equation of ellipse. For each node on the mesh, a check is made as to whether or not the node falls inside the domain of modification. This can be done using the following equation

\[ B = \frac{1}{4} \left( 4 + A^n \right). \]  

(8)

\[ \frac{(x \cos(\theta) + y \sin(\theta))^2}{a^2} + \frac{(y \cos(\theta) - x \sin(\theta))^2}{b^2}, \]  

(9)

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. The parameters are illustrated in Fig. 2. When the value 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.

![Fig. 2: Geometry of an ellipse at a given angle \( \theta \) with the x-axis.](image)
3. RESULTS

The results from the optimisation can be accumulated in a tabular form stating the optimised numerical values of all the design variables, along with an FE model of the optimised plate. The only change in the FE model after the optimisation is in the grid point coordinates to accommodate the dome-shaped indentations; therefore, the FE model still has 2666 nodes and 2562 elements. The panel has the material property of steel and is optimised for a clamped boundary condition, which is defined in the normal mode analysis by restricting the edge nodes of the FE model with applying the translational and rotational stiffness. Once, satisfied with the numerical predictions from the optimised FE model, the tabulated numerical values of the design variable are used to manufacture the actual test specimen. The details of the optimised plate are illustrated in the Fig. 3; thickness of the plate is 1.2 mm.

![Optimised Plate](image)

**Fig. 3:** The optimised plate with four domes. All the dimensions are in mm.

The test specimen for the optimised plate was fabricated in the mechanical workshop at Loughborough University, using the design details from the Fig. 3. The optimised plate is then clamped inside a metal frame to be tested for sound radiation in an anechoic environment. The measurement procedure followed for all the experimental testing inside the anechoic chamber is based on the ISO 3744 [13]. The sound power level comparison of the optimised four-dome panel with a reference flat panel is illustrated in the Fig. 4.

![Sound Power Level](image)

**Fig. 4:** Sound power level comparison of the four-dome optimised panel and its equivalent flat panel.
The optimised panel shows an improvement of around 2 dB over the entire test frequency range. This can be observed from the steady drop in sound power in the low frequency zone. It must be noted that this design is optimised for any general condition and could further be improved by the application of damping material layers.

Along with the optimised plate, tests were conducted on few panel designs with intuitively placed domes. The intuitive placement of domes is inspired from the initial, trial, optimisation results [1] and is tested to investigate the structural-acoustic response of the plates under the effect of one, two and four domes. The dimensions of the intuitively placed domes, in all the panel designs, are kept the same in order to identify a pattern of changed dynamic response with varying the number of domes indented. All the panel designs are illustrated in Fig. 5; all the panels are 1.2 mm thick.

![Fig. 5: The plates with: (i) four intuitively placed domes, (ii) one intuitively placed dome, (iii) two adjacent domes, and (iv) two diagonally opposite domes. All the dimensions are in mm.](image)

All the panels under consideration demonstrated an increase in the resonant frequency for a given mode due to the application of the dome(s). This is illustrated in Fig. 6, which shows a comparison between the predicted point receptance of the four-domed (intuitively placed) panel and its equivalent flat panel. The shift in resonant frequency is the result of the increased stiffness in the respective panels, which doesn’t necessarily mean that the stiffest panel is the least sound radiating.
The aim still remains to target the lower order modes, up to the frequency limit of 600-700 Hz, and to improve the structural-acoustic response under this frequency range.

Fig. 6: Comparison of the point receptance of the 4 domes and the equivalent flat panel.

The change in structural vibration characteristics leads to a change in the panel’s sound radiation characteristics. This can be observed from sound power level plots for different domed panel designs, experimentally tested in an anechoic environment. Fig. 7 shows a comparison of measured sound power level for an intuitively positioned single-domed panel with its equivalent flat panel.

Fig. 7: Measured sound power level of a single-domed panel and its equivalent flat panel.

It is clearly observed in Fig. 7 that a drop in the sound radiation is achieved for the low (50-250 Hz) and mid frequency range (375-550 Hz). However, from approximately 575 Hz to 675 Hz, the dome appears to make the panel a better radiator of sound. This can be shown, again, when comparing the panel with four intuitively placed domes with its equivalent flat counterpart, as illustrated in Fig. 8.

Fig. 8: Experimental sound power level comparison of 4-domed panel with equivalent flat panel.
Judged by the shift in resonant frequencies, the panel with four domes is seen to be the stiffest of all the test panels and yet it is not the least radiating panel. The total sound power over the test frequency range is almost the same for both the four-domed panel and the reference flat panel. So, it is the stiffness distribution along the panel, which needs to be balanced so as to make it radiate less. These dome-shaped indentations are, in effect, breaking the nodal lines for the lower order modes and, hence, result in stiffening it across the nodal lines. For example, considering the (2,1) mode on a simply-supported rectangular panel as shown in Fig. 9, the specific arrangement of the dome will stiffen the panel across its nodal line, thereby, reducing the structural-acoustic response of that mode.

![Fig. 9: Representation of (2,1) mode on a simply-supported rectangular panel.](image)

The difference in stiffness distribution patterns, in different panels, will alter their respective sound radiation characteristics. To demonstrate the variation in stiffness distribution, Fig. 10 illustrates the comparison of measured sound power level of the two adjacent placed domes and the panel with two diagonally placed domes; refer to Fig. 5 (iii) & (iv). It is observed in Fig. 10 that the panel with two adjacent domes radiates less than the panel with two diagonal domes, in the degree of around 5 dB overall. Thus, it becomes important to optimise the panel design for a given number of domes.

![Fig. 10: Sound power level comparison of panels with two domes, placed adjacently and diagonally.](image)

When the panels are indented with domes, it loses its isotropic behaviour and imposes orthotropic attributes [3]. This phenomenon makes most of the linear numerical analysis lose confidence when comparing it with experimental test results. But a numerical comparison of different designs should demonstrate the same distinctive pattern as observed from experimental test comparisons. Since the design itself is a product of numerical analysis, the improvement in low frequency range (< 250 Hz)
is quite discernible. After this range, it is expected that the orthotropic nature of the panel makes it no longer possible to follow the linear assumptions.

4. CONCLUSION

This paper has illustrated the usefulness of geometrical modifications, in the form of elliptical indentations or domes, pressed into a flat panel that help reduce sound radiation. However, care needs to be taken when increasing the stiffness of a panel because the stiffness, if inappropriately altered, might make the structure radiate more sound in the frequency range of interest. The technique described in this paper targets lower frequency modes, encompassing a defined frequency range of interest. The methodology used is suitable for automotive-type panel optimisation, which is suitable for any load cases since the structural modes are independent of any specific form of excitation. This approach is a computationally fast solution to the structure-borne noise issues, where optimally designed panels are not affected if the neighbouring structures are re-designed.

REFERENCES