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Structural-acoustic behaviour of automotive-type panels with dome-shaped indentations

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ABSTRACT

This paper investigates the structural-acoustic characteristics of the automotive-type panels with dome-shaped indentations. These indentations are used to increase the stiffness of the panel whilst keeping its bulk weight constant. To investigate the effect of domes upon sound radiation, four panels with different arrangements of domes are investigated numerically and experimentally. In order to improve the effect of the indentations on the panels, a structural-acoustic optimisation technique is developed. The objective function of the optimisation is to reduce the sound radiating from the panels by selecting the optimal sizing and placement of the dome-shaped indentations. The objective function, in this case, the radiated sound power from the panel, is calculated in terms of the surface velocity of the panel predicted from a finite element model. The optimisation technique assumes equi-partition of modal energy. Thus, it optimises the panel without the necessity of knowing the specific force input characteristics. Numerical and experimental results are presented for the four different intuitively designed panels, the optimised panel and a reference flat panel. It is shown that if the panel design is chosen inappropriately then the structure may radiate more sound in the frequency range of interest. However, a correctly chosen design has the ability to reduce radiated sound power without increasing the weight of the structure.

Keywords: Panel sound radiation; structural-acoustic optimisation; finite element method; domed-shaped indentations.

1. Introduction

In the modern automotive product development, noise, vibration and harshness (NVH) characteristics are inextricably linked to the designing of associated sub-structures. The latest trend observed in different automotive brands is to reduce the bulk weight of the vehicle. The panels enclosing the vehicle body structure are designed to keep the body-in-white (BIW) mass low, which leads to efficient fuel economy. Vehicle body panels are made out of thin sheet metal and, thus, have a very low bending stiffness. Hence, it has become common practice to increase the stiffness of these thin body panels by introducing ribs, stiffeners or beads [1,2,3]. A recent 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 elliptical beads (domes) to criss-crossed swages, as illustrated in Fig. 1. Similar types of indentations have long been used in automobile body panels in order to ‘break up’ the first few modes of a large panel. With the advent of powerful computer capabilities in design, different configurations of indentations are used to modify the low- to mid-frequency vibration modes. Depending upon the configuration of indentations, the structural-acoustic characteristics of a body panel can be altered significantly [4]. For example, the resonant frequency of lower-order modes can be shifted out of the frequency range of interest, and, thus, the radiated sound power from the body panel decreases in the frequency range of interest [5].

Even with the application of damping pads to reduce body panel vibration levels, it still remains difficult to find intuitive and cost effective countermeasures that can reduce the sound radiation in the frequency range from 100 to 500 Hz, where powertrain structure-borne noise dominates. One of the few alternatives is to indent the panels with different geometrical shapes. The objective of this paper is to illustrate the dynamic behaviour of such panels when modified with dome-shaped indentations. A numerically predicted and experimentally measured dynamic response comparison is reported, comparing panels with a differing number of dome-shaped indentations and also by varying their respective placement. It is observed that the number of domes and their placement interfere with the modal characteristics of the lower and middle order modes, which thus, alters their radiation characteristics.

A structural-acoustic optimisation technique is also developed in order to optimise the design of similar rectangular panels. The optimisation procedure provides an optimum location for the coordinates of the centre of the domes and their respective dimensions. A selection of literature that is concerned with the minimisation of structural-acoustic responses by modifying different vehicle body parts is given in Refs. [6,7,8,9]. A detailed review of general structural-acoustic optimisation

was published by Marburg in Ref. [10]. In any optimisation problem, it 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 dimensions of the dome-shaped indentations. The number of domes required is a design constraint that is set prior to the optimisation procedure. The technique described in this paper calculates the optimal placement of one dome and then based on symmetry, calculates the positions of the remaining domes. This particular technique has a significant reduction in computation time compared to locating each dome separately. However, this symmetry-based approach is only suitable for geometrically symmetrical panels, for example rectangular panels.

In section 2, the structural and acoustic analysis is presented as well as a description of the optimisation strategy. In section 3, the experimental apparatus, measurement method and panel designs are described. Section 4 presents a comparison of structural and acoustic response of panels with one, two and four domes placed intuitively on the panel as well as a panel with four domes optimally located. Section 5 summarises the findings of the research.

2. Theory and numerical implementation

2.1. Structural analysis

The optimisation technique described in this paper is based upon a real eigenvalue, or normal mode, analysis performed using a finite element (FE) model of the test structure. To perform the normal mode analysis, it is assumed that the structure is undamped and with no applied loading [11], so the equation of motion in matrix form becomes as

$$[M]\{\ddot{u}\} + [K]\{u\} = 0, \quad (1)$$

where $[M]$ is the mass matrix, $[K]$ is the stiffness matrix and the $\{u\}$ is the displacement vector. Assuming a harmonic solution there is an eigenvector $\{\phi_i\}$ that satisfies Eq. 1 corresponding to each eigenvalue ω_i . Therefore, Eq. 1 can be rewritten as

$$[K - \omega_i^2 M]\{\phi_i\} = 0, i = 1, 2, 3... \quad (2)$$

Each eigenvalue and eigenvector defines a free vibration mode of the structure. 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 X, Y and Z axes and the remaining three define the rotation about

X, Y and Z axes, for every eigenvalue (or natural frequency). The translational displacement along the three coordinates can be 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 calculate the radiated sound power. In this paper, the calculation of eigenvectors is restricted to a set frequency range of interest, for example, 0-1000 Hz, when targeting normal modes up to 750 Hz.

2.2. Acoustic analysis

The objective function used in the optimisation procedure is to minimise the sound radiation from the panel over the frequency range of interest. 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 sound power radiated by this set of noise sources is then calculated by using a quadratic equation expressed in terms of the surface velocities. The structure is assumed to be vibrating harmonically and the quadratic sound power expression is derived by using the boundary element method applied to the Helmholtz equation. The discretised structure requires interpolation functions to map the nodal values over the finite elements and facilitate the integral evaluations [12]. The numerical evaluation of the Helmholtz integral equation leads to an algebraic system of equations

$$p = D^{-1}Mv, \quad (3)$$

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 the surface and the integration of the Green's function over the surface, respectively. 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 [13].

The objective function to be minimised, P , is calculated from the summation of the sound power radiated by each individual element, P_j , on the surface of the radiating structure

$$P = \sum_{j=1}^{N_{el}} P_j = \frac{1}{2} \text{Re} \left\{ \sum_{j=1}^{N_{el}} \int_{S_j} p_j v_j^* dS \right\}, \quad (4)$$

where N_{el} is the total number of elements used in the FE model, S_j is the area of element j and S is the enclosed surface at a given distance from the structure. Using the impedance matrix, Z , and substituting for pressure, p , using Eq. (4) and using the same interpolation functions for the pressure and the velocity as used for the boundary element solution leads to

$$P_j = \frac{1}{2} \text{Re} \left\{ v^T Z_j^T \left(\int_{s_j} N N^T dS \right) v_j^* \right\}, \quad (5)$$

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 dimension of the matrix Z is equal to the total number of nodes multiplied by the number of nodes per element. The integral in Eq. (5) can be calculated separately as

$$Z_j^T \left(\int_{s_j} N N^T dS \right) = A_j, \quad (6)$$

such that A_j has same dimensions as the matrix Z . Substituting Eq. (6) into Eq. (5), and then summing for all the elements leads to the total radiated power

$$P = \frac{1}{2} \text{Re} (v^T A_1 v_1^* + v^T A_2 v_2^* + \dots). \quad (7)$$

Assembling all of the A_j submatrices into a single matrix, A , and all of the v_j vectors into a single column vector, v , then rationalising the real component operator yields a compact form for the total radiated sound power from the discretised structure as

$$P = v^T B v^*, \quad (8)$$

where B is Hermitian and is equal to

$$B = \frac{1}{4} (A + A^H). \quad (9)$$

2.3. Optimisation strategy

The structural and acoustic analysis represented by Eqs. (1)-(9) are now incorporated within an optimisation strategy in order to identify the optimum panel design for minimum sound radiation. The panels investigated in this paper are assumed to be an integral part of the vehicle body. However, following a sub-structuring approach [14] allows the test panel to be analysed in isolation from its neighbouring panels and, thus, limits the optimisation search space. The isolated body panel is then assumed to be an integral part of the vehicle body by applying translational and rotational restrictions to the boundary nodes of its finite element model.

The implementation of the optimisation strategy is based upon a method outlined in Refs. [4,5,7,15] and described in more detail in Ref. [16]. A flow diagram of the method is illustrated in Fig. 2. In stages 1 and 2 the optimisation procedure imparts a modification to the FE model of the domed panel. In stage 3 the eigenvectors or normal modes of the modified plate are calculated by solving the eigenequations, Eqs. (1)-(2), using NASTRAN[®]. In stage 4, retaining the generality of unspecified forcing, the radiated sound power, Eq. (4), of the modified plate is calculated using Matlab[®]. Stages 1-4 are repeated for each iteration of the optimisation procedure until the panel design is optimised for minimum sound radiation.

An effective way to parameterise the domed shaped modifications on the panel is by the use of modification functions [17-19]. The geometrical domain of the modification function for a dome can be defined by the equation of an ellipse. For each node on the mesh, a check is made as to whether or not the node falls inside the domain of the modification. This can be done using the following modification function

$$M(x,y,\theta) = \frac{\{x \cos(\theta) + y \sin(\theta)\}^2}{a^2} + \frac{\{y \cos(\theta) - x \sin(\theta)\}^2}{b^2} \quad (10)$$

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 (x_o, y_o) in the X- and Y-dimensions, θ is the angle of the ellipse with the X-axis and a and b are the two radii of the ellipse. The parameters are illustrated in Fig. 3. When the value of the Eq. (10) is negative, the node falls within the modification domain of the dome. The height or Z-coordinate of this node, z_d , can then be set according to its distance from the centre of the ellipse

$$z_d = (1 - M(x, y, \theta)) h_d \quad (11)$$

where h_d is the maximum height of the dome at (x_o, y_o) which also needs to be defined. Thus, in total, six design variables are used to define one dome of the panel: five design variables, x_o , y_o , θ , a , and b , are used to define the ellipse's geometry and one design variable, h_d , defines the maximum height of the dome. Quarter panel symmetry is then invoked to link the design variables of this dome to the three other domes on the panel. Thus, only the six design variables of the first dome are required for the optimisation. The advantage of this approach is that fewer design variables implies fewer dimensions in the optimisation search space and, thus, better convergence.

An optimisation algorithm is employed in order to find the best values of the six design variables of the dome, x_0 , y_0 , θ , a , b , and h_d , that minimise the radiated sound power of the panel as calculated using Eq. (4). A concept based on a genetic algorithm (GA) [20] is suitable for an optimisation problem like this, where traditional heuristic methods are not present or lead to unsatisfactory results. A genetic algorithm is a search technique used in numerical optimisation to find an exact or approximate solution. With the help of a GA, an exhaustive search over a relatively small search space can be performed in a reasonable amount of time. The genetic algorithm is implemented by having a population of randomly generated initial candidate solutions, called individuals, which evolve towards better solutions in a given number of iterations. For this research there are 40 individuals in the population. Each individual consists of a set of values of the six design variables (x_0 , y_0 , θ , a , b , h_d) for the domed panel. The initial solutions are selected at random. However, for the dome-shaped indentations to stay on the structure, the random placement of domes is restricted by excluding the boundary nodes in the search space. A further limitation is that the centre of each dome must be located on a node of the FE model.

In each iteration, the fitness of every individual in the population is evaluated and the individuals are then modified to form a new population. In this paper, the fitness function is the measure of the reduction in the total radiated sound power, Eq. (4), from the test panel calculated by implementing the procedure illustrated in Fig. 2. With each iteration a new generation of individuals is produced by combining the design variables of the parent population in a procedure termed crossover. In order for the algorithm to look outside the parents' population for a better solution, some of the parents' design variables will also be randomly altered in a procedure termed mutation. Since the number of individuals in the population should remain constant the number of individuals that die will equal the number of individuals that are born. A cumulative probability distribution is formed in order to make decisions for breeding of new offspring and allowing the less fit individuals to die. This new population is then used in the next iteration of the optimisation to further converge towards the optimum solution of a panel with minimum sound power radiation. The optimisation algorithm terminates when either the maximum number of iterations has been conducted, or a satisfactory fitness level has been reached for the population. If the optimisation has terminated due to the maximum number of iterations being reached, a satisfactory solution may or may not have been achieved. For the panels under test, the optimal solution search is limited to a maximum of 20 iterations, which is observed to produce satisfactory results. A flow diagram of the procedure implemented in the GA is illustrated in Fig. 4.

2.4. Finite element implementation

The geometry of the test panels are designed in CATIA[®] and meshed into their respective FE models in HYPERMESH[®]. The number of grid points and mesh-elements in each FE model varies with each panel design, in order to accommodate the specific shape and number of the dome-shaped indentations. In general, the test panels are comprised of approximately 2666 grid points accounting for 2562 quadrilateral shell elements which sufficiently satisfies the theory of at least six elements per wavelength for the maximum frequency value of interest [21].

The optimisation code is generated in Matlab[®] with the objective function being to reduce the radiated sound power from the test panel over the given number of iterations. To initiate the optimisation, an FE model of the test panel is required that contains grid points (nodal coordinates), element information and the type of solution required (SOL103) [11]. The finite element data of the test panels are exported into the Matlab[®] environment using a text-type ‘.bdf’ file. In the ‘.bdf’ file, the material property (steel with Young’s modulus 210×10^9 N/m² and density 7800 kg/m³) for the test panels and the mesh-element (CQUAD4) type are defined. However, the boundary conditions need to be specified additionally by defining the translational and rotational stiffness values along the boundary nodes. Since the test panel is assumed to be isolated from the remaining vehicle, the optimisation does not involve any specific excitation point on the rest of the vehicle structure.

3. Experimental setup and test panels

3.1. Experimental apparatus

A total of six panels were investigated; four panels with intuitively placed one, two and four domes, and one optimised panel with four domes, along with the reference flat panel. Each panel was placed, in turn, inside a metal frame in order to achieve a clamped boundary condition along its edges. The block diagram of the experimental apparatus is illustrated in the Fig. 5. The panel was excited over the frequency range 0 to 1.25 kHz using an electro-dynamic exciter located at the coordinate 110 mm in x-direction and 80 mm in y-direction assuming the origin at the left-hand corner of the panel. The applied force and response acceleration signals were acquired using an LDS FOCUS II multi-channel real-time dynamic signal analyser together with a PCB Piezotronics integrated circuit piezoelectric (ICP) shear accelerometer type 352C33, a Brüel & Kjær force transducer type 8230-C-003 and a Brüel & Kjær conditioning amplifier type 7749.

A similar experimental set up was used for the sound power level measurements. However, each panel in its metal frame was now mounted in a concrete baffle as illustrated in Fig. 6. Each test panel was excited using an electro-dynamic exciter. To measure the response sound pressure, a set of GRAS prepolarised microphones, type 40AE, along with CCP preamplifiers type 26CA were used. The acoustic investigations were conducted in an anechoic chamber and the measurement procedure adhered to the ISO 3744 standard for the calculation of sound power [22]. Thus, a reference hemisphere is defined, centred on the middle of the test panel with the radius of the hemisphere being equal to 1 m. The coordinates of the microphone measurement positions are illustrated in Fig. 7.

3.2. Test Panels

The initial, trial, optimisation results suggested the placement of one dome towards each at the four corners of a flat rectangular panel. To reduce computation time, the optimisation process makes use of the symmetry in placing the dome-shaped indentations on the panel. Thus, the panel is divided into four quarters and the ideal dome location identified by optimising one quarter of the panel and then replicating this location symmetrically on the remaining quarters of the panel. Hence, in order to compare the effect of increasing the number of domes on the panel, a set of four test panels was constructed with intuitively placed dome-shaped indentations as illustrated in Fig. 8. The dimensions of the intuitively placed dome-shaped indentations, in all the panels, were kept the same in order to identify the pattern of the changed dynamic response when varying the number of indentations. The dimensions of the panel designs investigated are illustrated in Fig. 8. All the panels have a thickness of 1.2 mm.

The result of the optimisation process is a list of the numerical values of the design variables, along with an FE model of the optimised plate. The only change in the FE model after the optimisation process is in the grid point coordinates used to accommodate the dome-shaped indentations. Therefore, the FE model of the optimised panel still has the same number of nodes and elements as the non-optimised panel. The design of the optimised panel is illustrated in Fig. 9. The thickness of the panel is 1.2 mm. The panel has the material property of steel and has clamped boundary conditions. These boundary conditions are imposed in the normal mode analysis by constraining the edge nodes of the FE model by applying translational and rotational restrictions.

4. Results

4.1. Intuitively placed domes

Fig. 10 shows a comparison of the modulus of the measured point mobility of the flat panel with the finite element model predicted point mobility of the equivalent panel with clamped boundary conditions over the frequency range 0-1000 Hz. It can be seen in Fig. 10 that there is a good agreement between the measured and predicted values, thus, indicating that the experiment clamped panel rig achieved the desired clamped boundary conditions. Table 1 lists the FE predicted natural frequencies of the panel together with their corresponding analytical [23] and measured resonant frequencies. Also shown in Fig. 10 as a horizontal line at 0.0071 (m/s)/N is the point mobility of the equivalent infinite plate [24]. As expected the equivalent infinite plate mobility lies between the peaks and troughs of the finite plate data.

Fig. 11 shows a comparison of the predicted point mobility of flat panel with the panel with four intuitively placed domes. It can be seen in Fig. 11 that above 350 Hz there is an increase in the resonant frequencies of the four-domed panel compared to the flat panel response. Thus, the domes act as stiffeners increasing the resonant frequencies of the panel. An illustration of the effect of the domes upon the modeshapes of the panel is shown in Fig. 12. It can be seen in Fig. 12 that the dome-shaped indentations significantly change the modeshape of the panel with areas of low displacement amplitude being introduced around the indentations. However, the peak displacement response at the centre of the domed-panel has now been increased relative to the flat panel. This increase in the peak displacement was not apparent in the point mobility data shown in Fig. 11 as the excitation location was off-centre.

Fig. 13 shows a comparison of the measured and predicted point mobility of the panel with four intuitively placed domes. Although in broad agreement over the frequency range of interest, some difference between the measured resonant frequencies and predicted natural frequencies can be observed in Fig. 13. This may be due to the fact that the horizontal plane of experimental panel was distorted slightly by the introduction of the dome-shaped indentations. This may have led to a non-evenly clamped boundary condition along the plate's edges.

An illustration of the effect of the dome shaped indentations upon the radiated sound can be seen in Fig. 14. This shows a comparison between measured sound power level of the panel with four intuitively placed domes and the reference flat panel over the frequency range 10-1000 Hz. The critical frequency of the flat panel is 10417 Hz. Thus, the frequency range of interest is well below

the critical frequency. It can be seen in Fig. 14 that there is a difference in resonant frequencies of the two panels. However, the average sound power radiated by both panels over the frequency range shown is approximately the same. Thus, the dome-shaped indentations do not appear to have achieved a reduction in the total sound power radiated by the panel.

For automotive applications, generally, the number of indentations is limited to the space available on the body panels. But, if given the freedom to choose any number of indentations for the best design, the decision still remains ambiguous. This is illustrated by considering the response of the panel with one intuitively placed dome to the panel with four intuitively placed domes. Fig. 15 shows a comparison of the numerically predicted point mobility of the one-domed panel compared to the reference flat panel. As evidenced by the shift in natural frequencies, it can be seen in Fig. 15 that the one-dome panel is only slightly stiffer than the flat panel and significantly less stiff than the four-dome panel shown in Fig. 11. A comparison of the measured sound power radiated from the one-dome panel and the four-dome panel is shown in Fig. 16. The panel with one intuitively placed dome has achieved a greater reduction in radiated sound power, over the frequency considered, than the panel with four intuitively placed domes. This is not surprising as it is known that the presence of constraints can increase the radiation efficiency of a thin panel [25]. Ideally the stiffening of the panel should decrease the vibration response by a sufficient level to offset the increase in radiation efficiency so that the resulting sound power will be reduced. However, for the four-domed panel this has not occurred.

A further illustration of the effect of structural stiffness, vibration response level and radiated sound power is shown in Figs. 17 and 18. A comparison of the measured point mobility of the panel with two adjacent domes and the panel with two domes located diagonally opposite is shown in Fig. 17. It can be seen in Fig. 17 that the panel with two diagonally located domes has a higher peak vibration response, and an increase in natural frequency and hence, stiffness, for certain modes, compared to the panel with two adjacent domes. The resulting sound power level comparison shown in Fig. 18 shows a reduction of approximately 4.5 dB for the two adjacently placed domes compared with the diagonally placed domes over the frequency range considered.

The observed structural-acoustic response from the preceding analysis of all the intuitively designed panels demonstrated a complex relationship between the structural dynamics and sound radiation capabilities of the panels. For a given mode, it has been shown that the placement of domes can stiffen the panel and, hence, increase the resonant frequency, and they can change the

modeshape of the panel by ‘breaking’ its nodal lines. However, this does not necessarily lead to a reduction in radiated sound power over the entire frequency range.

4.2. Optimised panel design

The optimisation process is performed in order to design a panel with four optimally placed domes for clamped boundary conditions that minimises the total sound power radiated over a given frequency range. The optimisation is based on a normal mode analysis of the panel, performed for a frequency range from 0 Hz up to 1000 Hz. The final design of the optimised panel is illustrated in Fig. 9. Fig. 19 illustrates a comparison between the measured point mobility of the optimised panel and the measured point mobility of the reference flat panel. Below approximately 200 Hz both panels have a similar structural response. However, above 200 Hz the two panels show considerable differences in their resonant frequencies and peak amplitudes. Fig. 20 shows the corresponding comparison in the measured sound power levels between the two panels. It can be seen in Fig. 20 that there is very little similarity between the sound power spectrum of the optimised panel and the reference flat panel. The overall reduction in sound power level, across the frequency range of interest, achieved by the optimised panel is approximately 2 dB.

Fig. 21 shows a comparison between the point mobility of the panel with four intuitively placed domes and the panel with four optimised domes. Fig. 21 indicates that the point mobilities are relatively dissimilar. A comparison of Fig. 21 with Figs. 11 and 13 suggests that the point mobility of the panel with four intuitively placed domes has resonant frequencies closer to that of the flat panel than to those of the optimised panel. Fig. 22 shows a comparison of the radiated sound power level of the optimised panel compared to the panel with four intuitively placed domes. A comparison of the radiated sound power level spectra shown in Fig. 22 with the flat panel radiated sound power level shown in Fig. 20 indicates that the panels with domes are more similar to each other in radiation characteristics than they are to the reference flat panel. Over the entire frequency of interest, the optimised panel has an average sound power level 1.7 dB less than the intuitively designed panel.

5. Summary and conclusion

This paper has reported a numerical and experimental study into the vibrational and sound radiation characteristics of flat panels with dome shaped indentations. A total of six panel designs were investigated: (i) a flat reference panel; (ii) a panel with one intuitively placed dome; (iii) a panel with two intuitively placed domes located diagonally opposite each other; (iv) a panel with

two intuitively placed domes located adjacent to each other; (v) a panel with four intuitively placed domes; and (vi) a panel with four optimally sized and located domes. The optimisation procedure was based upon a normal mode analysis that minimised the radiated sound power over the frequency range of interest. As this technique was based upon quarter-panel symmetry, the optimised panel design contained one dome shaped indentation in each quarter section of the panel.

From a comparison of point mobility data it was discovered that the domes act as stiffeners increasing the resonant frequencies of the panel. In general, the panels with the greatest number of domes exhibit the largest shift in resonant frequencies. An illustration of how the dome shaped indentations change the resulting mode shape was shown for the panel with four intuitively placed domes. However, from a comparison of the measured sound power data it was also discovered that the panels with the greatest number of domes, and hence increase in stiffness, did not exhibit the greatest reduction in radiated sound power. Over the entire frequency range of interest, 0-1000 Hz, the panels with either one intuitively placed dome or two intuitively placed domes located adjacent to each other were shown to have achieved the greatest reduction in radiated sound power level. The panel with four intuitively placed domes did not achieve a reduction in radiated sound power. Whilst the optimised panel design was shown to radiate less sound power than the panel with four intuitively placed domes and the flat reference panel.

One limitation of the proposed optimisation technique is the requirement for an initially flat rectangular panel. Further work to improve the method could include an extension to arbitrarily shaped three-dimensional FEM models. An initial consideration of this enhancement is reported in Ref. [16] where the dome modification function, Eq. (10), is defined in terms of a volume rather than an area. However, as noted in Ref. [16], this will require an indirect boundary element modelling of the sound radiation rather than the simplified sound power calculation of the current approach.

In conclusion, this paper has shown that dome-shaped indentations can be used to reduce the radiated sound power of a rectangular flat panel. However, the placement, dimensions and number of domes on the panel should be chosen with care in order to achieve maximum sound reduction. An optimisation procedure can assist the design process.

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(a)

(b)

Fig. 1. Photographic examples of indentations on the body panels of a vehicle: (a) criss-crossed swages; and (b) elliptical domes.

Modification function, Eqs. (10)-(11)

Design variables for
dome ($x_o, y_o, \theta, a, b, h_d$)

Generate mesh of panel
for FE model

Structural analysis, Eqs. (1)-(2)

Compute normal modes
of panel, Eq. (2), using
NASTRAN[®]

Acoustic analysis, Eqs. (3)-(9)

Compute radiated
sound power of panel,
Eq. (4) Matlab[®]

Fig. 2. Flow diagram of the structural and acoustic analyses executed during each iteration of the optimisation procedure.

Fig. 3. Geometry of an ellipse at a given angle θ with the x-axis used in the modification function M .

Fig. 4. Flow diagram of the procedure implemented in the genetic algorithm.

Fig. 5. Schematic representation of the experimental apparatus.

Fig. 6. Photograph of the steel test panel mounted inside the concrete baffle.

Fig. 7. Microphone positions on the hemispherical measurement surface.

Fig. 8. Dimensions of the test panels with intuitively placed domes: (a) four domes, (b) one dome, (c) two adjacently located domes, and (d) two diagonally opposite domes. All the dimensions are in mm.

Fig. 9. Dimensions of the optimised panel with four domes. All the dimensions are in mm.

Fig. 10. Measured and predicted modulus mobility comparison of the flat panel.

Fig. 11. Predicted modulus mobility comparison of the panel with four intuitively placed domes with the reference flat panel.

Fig. 12. Comparison of the (1,3) mode in: (a) the panel with four intuitively placed domes; and (b) the equivalent flat panel.

Fig. 13. Measured and predicted modulus mobility comparison of the panel with four intuitively placed domes.

Fig. 14. Comparison of the measured sound power level of the panel with four intuitively placed and the flat reference panel.

Fig. 15. Predicted modulus mobility comparison of the panel with one intuitively placed dome and the reference flat panel.

Fig. 16. Comparison of the sound power level of the panel with one intuitively placed dome and the panel with four intuitively placed domes.

Fig. 17. Comparison of measured modulus mobility of the panel with two adjacently located domes and the panel with two diagonally located domes.

Fig. 18. Comparison of sound power level of the panel with two adjacently located domes and the panel with two diagonally located domes.

Fig. 19. Measured modulus mobility comparison of the optimised panel and the reference flat panel.

Fig. 20. Comparison of sound power level of the optimised panel and the reference flat panel.

Fig. 21. Measured modulus mobility comparison of the optimised panel with the panel with four intuitively located domes.

Fig. 22. Comparison of sound power level of the optimised panel and the panel with four intuitively located domes.

Table 1

List of natural frequencies of the flat panel.

Mode order	Analytical (Hz)	FEM (Hz)	Measured (Hz)
(1,1)	189.3	188.5	175.8
(2,1)	295.9	294.3	283.4
(1,2)	461.4	459.2	433.6
(3,1)	475.2	472.3	460.0
(2,2)	560.9	557.3	564.0
(4,1)	722.3	717.8	693.6
(3,2)	729.0	724.4	756.6
(1,3)	872.4	868.7	913.3
(4,2)	967.1	960.2	945.6
(2,3)	970.3	963.9	1086