Weakly radiating structural modes of automotive-type panels

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


Additional Information:

- This conference paper was presented at the 16th International Congress on Sound and Vibration: www.icsv16.org/

Metadata Record: https://dspace.lboro.ac.uk/2134/5349

Version: Published

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/
In this paper, structural acoustic optimisation of automotive-type panels is considered. The aim of the optimisation method used is to impose on the panel structural modes that radiate acoustic energy weakly. In the proposed approach, the panel under consideration is isolated from the rest of the structure and hence reduce significantly the time required for its analysis and optimisation. The proposed method is applied specifically to the optimisation of a floor panel of a simplified car model. To achieve weakly radiating structural modes different geometric modifications on the panel are considered. The results show significant reduction in the radiated sound power of the optimised panels.

1. Introduction

A common engineering method for finding solutions to complex problems is to perform numerical optimisation. One such complex problem that numerical optimisation has been applied to is the minimisation of sound radiation from structures or structural components, such as vehicle structures. In the last two decades many researchers have been concerned with this problem. In some of the published work, structural-acoustic optimisation is considered from an geometrically idealised point of view where simplified structures, such as plates [1, 2] and beams [3], excited by simple excitation forces are studied. Moreover, structural-acoustic optimisation has a great application in vehicle NVH. Many publications are concerned with the minimisation of structural-acoustic quantities by modifying different parts of a vehicle body [4, 5, 6, 7]. An extended review of general structural-acoustic optimisation has been published by Marburg [8].

In an optimisation problem one needs to specify (at least) one objective function and a number of design variables. Common choices for the objective function is the sound pressure at a specific point or the sound power depending on the nature of the problem. For the design variables, the modification of the geometry of the structure has been used successfully. Marburg et al. has used geometry modification in references [11, 12, 13]. In reference [12] the concept of geometry modification based on modification functions, which changes the geometry of a panel in order to create beads, is discussed. Using this concept a variate of new structure designs can be created. A similar concept is used in this paper to create swages (rectangular beads) and domes (elliptical beads) on a panel.

In vehicle NVH it is often desirable to optimise a small number of panels of the whole vehicle structure that most contributes to unwanted acoustic effects. Even though only a small number of panels need to be optimised the response of the whole structure, of which they are part of, needs to be
calculated. This can be computationally very expensive and because this analysis needs to be carried out several times for every possible modification when it is fitted in an optimisation algorithm, this can increase the total time required dramatically. To avoid this, some reduction techniques that can be applied to Finite Element (FE) models have been used to reduce the total computation time. Two common techniques are the use of superelements [4], for all elements that are not modified during the optimisation, and substructuring techniques such as Wave-Based Substructuring (WBS)[5, 14].

In this paper an alternative to these methods is proposed. It is well known that the response of a structure can be given as a summation of its modes multiplied by unknown modal coefficients [15]. Given this, one needs to minimise the sound radiation from the modes of the structure that lie in a particular frequency range of interest in order to minimise the total sound radiation from the structure under any excitation conditions. Koopmann and his co-researchers [9, 10, 16] has suggested a method of creating structural modeshapes that have weak acoustic radiation characteristics by placing optimally sized masses on predetermined locations on a plate. These modeshapes, termed weak radiators, are designed in such a manner as to create strong cancellations of the radiating acoustic pressure in the vicinity of the surface of a plate and, hence, restrain acoustic energy from propagating in the far-field.

In this paper, the principle of weak radiators is used to optimise a floor panel of a simplified vehicle model. Using this approach, the extra computational cost of calculating the response of the whole vehicle structure is eliminated, since only modes of the panel under consideration need to be calculated. The design variables for creating weakly radiating modes are the location and properties of beads.

2. Description of the optimisation

For the purpose of the current research, a simplified automotive vehicle model has been created as shown in Fig. 1. The dimensions of the model vehicle are similar to a typical sedan type vehicle body. The material used is steel and all the panels have thickness of 0.8mm. In the model, the vehicle floor is separated by a frame of beams into four identical panels each with dimensions 0.75m × 0.55m. In the rest of the paper, optimisation of the front left floor panels is considered.

A common method for structural analysis of complex structures is the Finite Element Method (FEM). As mentioned earlier in the introductory section, even when optimisation of only one panel in a structure is considered, the response of the whole structure to which the panel is a member needs to be calculated. This will make the optimisation impractical because of the long time needed for the calculation of full FE models with a large number of elements (and possibly the subsequent calculation of acoustic radiation). In this paper, the panel under consideration is isolated from the rest.
of the structure. It is well known that the response of a structure to any excitation can be given as a superposition of its modes of vibration [15]. Making use of this, one can optimise the modes shapes of a panel without taking into account the rest of the structure in order to minimise the sound radiated by it. This will reduce significantly the time required for the optimisation. In this paper the modal behavior of the panel under consideration is calculated using NASTRAN as an FEM solver.

A problem that arises when the panel has been isolated is that boundary conditions need to be specified for the parts of the panel that were previously connected to the rest of the structure. Here this problem is overcome by applying translational and rotational restrictions to the boundary nodes of the FEM model of the floor panel. Using NASTRAN this can be achieved by using BUSH elements at the boundary nodes. BUSH elements are massless elements with translational and rotational stiffness. The values of the translational and rotational stiffness need to be set in such a manner as to simulate the behavior of the panel when it is attached to the whole structure.

A simple method was used to identify the correct stiffness values based on trial and error. The input mobility of the panel, when this is attached to the vehicle structure, is calculated and compared to that of the panel alone for different boundary conditions. When the boundary stiffness of the two panels are similar the resonant peaks in the input mobility curve should be close in frequency. In order to excite as many modes as possible the panel should be excited at one of the corner nodes close to (but not on) the boundaries.

It is easy to understand that in cases of panels attached on rigid frames or other stiff structural components, the translational stiffness will be very high since there is only a small translational displacement. To estimate the translational stiffness different values can be tried and, having in mind that more restricted boundary conditions will result in higher resonant frequencies and vice versa, the appropriate values can be found. For the case of this model the value for the translational stiffness chosen is $k = 10^{10}$ N/m and for the rotational stiffness $K = 5000$ N/m. These values give boundary conditions very close to a clamped boundary since modal calculations using the values mentioned above and theoretically clamped boundary conditions gave very similar results.

A comparison of the input mobilities of the two panels, that is the floor panel in-situ in the vehicle body structure and a panel with same properties and dimensions in isolation but with adjusted boundary conditions, is shown in Fig. 2. Note that the data of the floor panel contains some peaks caused by the resonances of the whole structure. To identify which of the peaks are because of the panel itself, one can look at the displacement (or velocity) profile of the plate at the frequencies of the peaks. The resonances of the floor panel have been identified and are shown with arrows in Fig. 2. The first mode, with a resonant frequency at around 57Hz, does not create a large peak in both curves.

After the panel has been isolated and its boundary conditions have been identified, the task of the optimisation procedure is to impose on the panel modes shapes that radiated acoustic energy weakly [9, 10, 16]. This is, as described in reference [16], to impose modes shapes that create strong

![Figure 2. Input mobilities of the floor panel and an equivalent panel with elastic boundary conditions.](image)
cancellations of the acoustic energy in the close vicinity of the structure (hydrodynamic short-circuit) and hence, prevent energy from radiating into the far field. This, of course, is possible only in the frequency range below the critical frequency which makes it appropriate to use for thin automotive panels with high critical frequencies. This method is not used to prevent vibrational energy from getting into the panel under consideration, as it is assumed that all possible measures have been used for this already (modification of the sources and their coupling to the structure). The proposed method is used only to prevent energy from radiating into the acoustic medium.

An advantage of this method is that when a panel has been designed to have weakly radiating structural modes it can be used in any structure, since the optimisations does not depend on the excitation or the rest of the structure (apart from the way it is connected to it via its boundary conditions). This means, for automotive applications, that when other parts of the vehicle body or the sources have been redesigned, the panel under consideration will still radiate acoustic energy weakly. Using conventional optimisation methods this is not the case since the optimisation depends on the way in which the panel is excited by the noise sources and/or neighbouring structural components. Hence, for every redesign of the rest of the structure the panel need to be re-optimised. This method also can be very effective at low frequencies where noise reduction using damping treatment is usually inefficient.

3. Modification functions

In this section possible structural modifications of a panel are discussed in order to achieve the desired weakly radiating structural mode shapes.

In an optimisation procedure design variables need to be defined, that is the parameters that need to be optimised in order to minimise (or maximise) a given objective function. It is desirable to keep the number of design variables to a minimum in order to simplify the search space and, hence, improve the optimisation. For vibroacoustic optimisation design modification in the form of modification functions have been proposed by Marburg and his co-researchers [11, 12, 13]. Using this approach the geometry of the panel is modified to create beads. In this paper two such modification functions are used, one for swages (rectangular beads) and one for domes (elliptical beads).

3.1 Swages

The modification function needs to define the type of modification and the geometrical domain in which that will take place. In the case of swages, for a flat panel lying in the X-Y plane, a rectangular area needs to be defined and the Z-coordinate of the nodes lying in this area are modified. This will result in a panel flat in its whole area apart from rectangular areas that are pressed out.

The geometrical domain of modification is illustrated in Fig. 3. The parameters defining the domain are two sets of coordinates in the plane of the panel, \((x_1, y_1)\) and \((x_2, y_2)\), which define a line. Another parameter, \(w\), is used to define the width of the rectangular area and \(\theta\) defines the angle of the line to the X-axis. Using simple trigonometry the coordinates of the four corners of the rectangular area, needed for the specification of the modification domain, can be determined.

Once the modification areas have been defined for a number of swages, they can be meshed independently from the rest of the panel and then the common nodes between them and the panel mesh can be connected. In the code developed for this study a number of fixed nodes, depending on the length of the sides, are defined on the boundaries of the mesh of the rectangular areas and the same fixed nodes are defined for the panel mesh. This makes the connection of the nodes at the panel/swages interface easier. Once the whole plate has been meshed the Z-coordinate of the nodes in the rectangular areas of the swages are set to have a value that represents the height of the swages (whereas the Z-coordinate of the rest of the nodes is 0). Hence, in total the design variables for each swage are 6, 5 defining the geometry and 1 defining the height of the swage.
3.2 Domes

The geometrical domain of the modification function for domes (elliptical press-outs on a panel) can be simply defined by the equation of an ellipse. In the case of domes, a mesh is created for the whole area of the panel. Then for each node of 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 equation of an ellipse, with parameters as defined in Fig. 4

\[(\frac{(x \cos(\theta) + y \sin(\theta))^2}{a^2} + \frac{(y \cos(\theta) - x \sin(\theta))^2}{b^2}) < 1\]  

where \(x = x_n - x_0, \ y = y_n - y_0\) are the distances of the \(n^{th}\) node to the centre of the ellipse in the X- and Y-dimensions. When this equation, for a given node, is less than 1, the node falls in 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 (measured using Eq. (1)). A maximum height of the dome at \((x_0, y_0)\) can be defined. In total, the design variables needed for each dome are again 6, 5 defining its geometry and 1 defining the maximum height of the dome.

4. Acoustic analysis

The objective function of the optimisation procedure is the total sound power \(P\) radiated by certain modes of the structure [16].

\[P = \sum P_n\]  

where \(P_n\) is the sound power radiated by the \(n^{th}\) structural mode. The summation is carried out over all the modes falling within a specific frequency range.

The solution of the structural eigenvalue problem returns modes with arbitrary amplitudes. For this reason, before the calculation of the total sound power, all modes are normalised in order to have a unit mean squared velocity. In this case, the sound power calculations of different modes can be compared in order to find those modes that radiate weakly.

Using FEM for the analysis of the panels described in the previous section, the normal velocity of all the nodes of the structure for a given modeshape can be calculated. In references [17, 18] a method for calculating the sound power of a structure based on the normal velocity at each node is described. For \(u\) being the normal velocity vector of the \(n^{th}\) mode, its sound power is given as:

\[P_n = u^T \mathbf{B} u^*\]  

where superscripts \(T\) and \(*\), indicate vector transpose and complex conjugate respectively, and
B = \frac{1}{4} \left( A + A^H \right) \tag{4}

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

$$A_j = Z_j^T \left( \int_{S_j} N N^T dS_j \right) \tag{5}$$

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 a BEM analysis [17, 18]. In this case, since this analysis will be a part of an optimisation procedure and calculation time is critical, the following assumption is also made; that the panel for which the sound power needs to be calculated is assumed to be flat. This simplifies the calculation of the impedance matrix significantly. The assumption is valid as long as the out-of-plane modifications of the plate are small compared with the acoustic wavelength.

5. Results

The procedure described above was used to optimise the floor panel of the simplified car model, presented in Section 2. The two aforementioned modification functions were used as design variables and a genetic algorithm have been used as optimisation algorithm. Since the objective of the optimisation is to design modeshapes that create cancellation of the acoustic pressure in the near-field of the structure, this requires some symmetry of the modeshapes since acoustic cancellation happens when the acoustic pressure of neighbouring areas are of the same or similar magnitude but opposite sign. For this reason, symmetry is used for the location of the modifications of the plate. Numerical experiments using optimisation and imposing, (i) no symmetry; (ii) symmetry in one dimension of the plate; and (iii) symmetry in both dimensions showed that imposing symmetry improves significantly the results of the optimisation procedure. For plates of arbitrary aspect ratio, symmetry in only one dimension gives better results than symmetry in both dimensions.

Using symmetry also reduces the number of design variables. For example in the optimisation of the vehicle floor panel using swages, two swages have been used but the properties and location of only one swage needs to be specified due to symmetry of the structure. For the case of domes, four domes have been used and again the properties of only one dome needs to be specified and the other three are determined from symmetry. In both examples the number of design variables is just 6.

Fig. 5 shows the panels optimised for the frequency range 10-100Hz, using swages and domes. In the optimisation the structural analysis has been performed using NASTRAN and the Lanczos algorithm, that allows for calculation of modes in a specific frequency range, and the sound power calculation procedure described in Section 4.

The optimised panels were then fitted into the floor of the simplified car model. The frequency response of the modified structure is calculated for two point forces acting on the front panel, which supports the engine, using NASTRAN/PATRAN. The acoustic power radiated only by the modified floor panel was calculated using LMS Virtual Lab. Fig. 6 shows a comparison of the radiated sound power between the modified and unmodified panels for the frequency range 10-100Hz over which the optimisation was performed. In the case of the domes the reduction in the average dB level over this frequency range is 14.3dB whereas for the case of swages the reduction is 16.8dB. It can be seen that in both cases significant reduction has been achieved for the specific frequencies which coincide with the frequencies of the optimised modes of the structure (for example around 45Hz and 75Hz).
6. Conclusions

In this paper, a methodology for automotive-type panel optimisation is proposed and it is applied for the optimisation of a floor panel of a simplified car model. In the proposed approach, the modeshapes of the panel under consideration are optimised in such a manner as to radiate acoustic energy weakly. In this procedure the panel is isolated from the rest of the structure and analysed independently. This speeds up the optimisation time considerably and, hence, makes it more practical. Another advantage of the proposed method is that once a panel has been optimally designed, it has weak acoustic radiation characteristics even when the rest of the structure has been redesigned. Also weakly radiating structural modes can be very efficient noise reduction technique at low frequencies where most damping treatments are inefficient.

One problem that needs to be considered when the panel has been isolated from the rest of structure is that its boundary conditions need to be specified in such a way as to simulate the boundary conditions of the panel when this is assembled in the vehicle structure. Here, elements with translational and rotational stiffness have been used at the boundary nodes of the isolated panel and their values have been determined by comparing the input mobilities of the isolated panel and the floor panel.

For design variables, two modification functions have been used for two different types of beads on the panel: rectangular and elliptical. Results have shown that when symmetry is imposed on the location and properties of the modifications on the panel, the optimisation performance is better. The results for the optimum designs of the floor panel studied here using first two swages and then four domes show significant reduction in the sound power radiated by the panels when these are re-attached to the vehicle structure.


7. Acknowledgment

The financial support of Jaguar and Land Rover Research is gratefully acknowledged.

REFERENCES


