Prediction, improvement and experimental verification of low-frequency vibro-acoustic transmission through automotive door hinges

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Abstract

For automotive manufacturers, the simulation of door panel vibration is an important concern for various reasons. A mass of evidence has shown that the vibration behaviour of door panels that enclose the passenger cabin can affect low-frequency noise in the cabin. However, anecdotal evidence suggests that, until now, the characterisation of structure-borne door panel vibration and its subsequent noise emission characteristics in terms of its door hinge system has not yet been contemplated or studied within a vibro-acoustic setting.

Thus, the primary contribution of this original work is to develop flexible numerical models with full coupling between the passenger cabin and vehicle body, to predict and improve the vibro-acoustic performance of the door hinge system without the introduction of unique vehicle components.

The work undertaken entails the development of three-dimensional, predictive finite element models to address this unique application in terms of noise generation and mitigation. The improvement involves the study of different door hinge systems provoked by an external structural forcing function under steady-state conditions and subjecting them to sensitivity analysis. With reference to an existing door hinge system, frequency domain numerical models are developed and solved employing experimentally-determined damping ratios.

An experimental campaign is undertaken to validate the numerical models based on frequency response data. The measured data were compared with the predicted values, the results of which exhibited good correlation. Compared to the reference design, a significant reduction in the interior noise level was achieved by using the proposed sensitivity analysis procedure.

Keywords: fluid-structure interaction, NVH, automotive interior noise.

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1 Introduction

The vibrational assessment of door panels enclosing a fluidic cavity is an engrossing and practical field of research in vehicular cabin design, since it contributes towards the overall impression of the vehicle quality and therefore has a great influence on the buying decision of the customer. As a result, there has been considerable ongoing interest in investigating methodologies and techniques for reducing the vibration amplitudes of body panels. Since the passenger cabin of a vehicle forms a cavity, resonant conditions can develop, characterised by acoustic modes of vibration [1]. Such acoustic modes, which generally occur at low frequencies, are associated with specific pressure distributions and natural frequencies. When these modes are excited by the vibration of door panels, acoustic resonance can result, giving rise to objectionable low-frequency boom noise. According to many researchers such as Morrey and Barr [2], most internal noise is associated with the dynamic behaviour of the vehicle body structure at frequencies below 400 Hz.

Though there are extensive studies aimed at the estimation of sound radiation from vibrating plates with classical boundary conditions, no comparable efforts have been made to determine the contribution of accompanying structural components with non-classical boundary conditions, such as automotive door hinges. Furthermore, the morphology of complex, built-up, acoustic-mechanical structures possessing multi-body dynamics generally exhibit important degrees of uncertainty and variability due to the presence of damping, varying stiffness, joints and joint position.

According to Desmet and Sas [3], there is little documentation on the damping phenomena within fluid-structure interaction systems and an even smaller number of experimental verifications been made to confirm theoretical approaches. A possible reason could be that many researchers use hypothetical damping ratios for the sake of completeness, thereby making experimental validation impossible. In addition, according to Marburg [4], many of these studies have used classical support conditions or neglected the supports altogether, whilst a few used unrealistic rigid supports.

The simultaneous interaction of the above uncertainties, complexities and competing factors require computer-based tools, because such systems are too complex and time consuming to be analysed analytically. The development of predictive models is therefore an effective and sensible approach.

In the strategy developed and proposed in this work, the coupling of the passenger cabin and accompanying body structure is described in modal terms. A special improvement module, currently non-existent within the employed ABAQUS commercial finite element code, is developed to assist the sensitivity analysis process and facilitate the identification of feasible design regions that could influence the interior noise level. This paper outlines both a cost-effective numerical modelling and experimental verification procedures used to investigate and evaluate the vibro-acoustic behaviour of automotive door hinge systems for the prediction and reduction of interior noise levels in terms of its hinge system.

2 Physical description and geometry of the model

In order to minimise computational efforts and costs, a simplified car body of approximately one-third scale, stripped of its internal trimming but with its door hinges, door seal, latch and striker mechanism remaining as in production, is investigated. A stiff vehicle structure was developed using dimensional analysis and similitude theory in order to maintain approximate relative occurrence of acoustic and structural modes over the low-frequency domain on par with a typical full-size real vehicle structure. The cases under study consist essentially of a vehicle's front right-hand door panel with its respective upper and lower door hinge assemblies, a door seal and a vehicle body structure constituting an enclosed passenger cabin. The door is articulated by means of the hinge assemblies connected to the A-pillar and door, respectively. The general test structure under consideration is shown in fig. 1, the components of which are representative of systems used in domestic passenger vehicles. The test structure is constructed from both 25 mm square and 100 mm × 50 mm mild steel tubing (A-pillar); clad by 3 mm thick mild steel plates. The interior of the passenger cabin is sealed by means of a door seal at the front right-hand door.

3 Mathematical models

3.1 Fluid behaviour

The governing equilibrium equation for small motions of a compressible, inviscid fluid with a finite wave speed is taken to be

$$\frac{\partial p}{\partial \mathbf{x}} + r\,\dot{\mathbf{u}}^f + \rho^f\,\ddot{\mathbf{u}}^f = 0 \qquad (1)$$

where p is the acoustic pressure in the fluid, $\dot{\mathbf{u}}^f$ is the fluid particle velocity, $\ddot{\mathbf{u}}^f$ is the fluid particle acceleration, ρ^f is the density of the fluid and r is the volumetric drag. This equation is modelled using a pressure-based finite element (FE) formulation with nodal pressures on the finite element mesh.

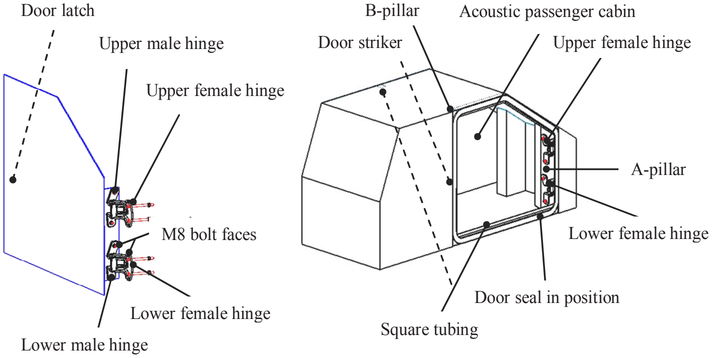

Figure 1: Simplified vehicle with actual door hinge system.

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Furthermore, the constitutive behaviour of the fluid is assumed to be inviscid and compressible so that

$$p = -K_f \frac{\partial}{\partial \mathbf{x}} . \mathbf{u}^f \tag{2}$$

where K_f is the bulk modulus of the fluid, $\mathbf{x}$ the spatial position, and $\mathbf{u}^f$ represents the fluid particle displacement. Hence, the fluid, which serves as an additional mechanical load on the inner body panels, is treated as an elastic solid with a finite bulk modulus and a negligible shear modulus. The A-weighted sound pressure level (SPL) at a particular location p_i may be defined as

$$p_i = 20 \log_{10} \left(\frac{P/\sqrt{2}}{P_{ref}} \right) + A\ weighting \tag{3}$$

where P is the pressure and P_{ref} is the standard reference pressure of 2×10^{-5} Pa.

3.2 Hyperelastic door seal model

In this work, the popular EPDM sponge-dense, elastomeric door seal is used. The door seal is modelled with the hyperelastic material model, which is a non-linear, continuum model. This model incorporates the highly compressible nature of EPDM sponge door seals and is characterised by the Marlow strain energy potential described by Morman [5] and applied by Wagner *et al.* [6]. The Marlow strain energy potential may be expressed in the form

$$U = U_{dev}(\bar{I}_1) + U_{vol}(J_{el}) \tag{4}$$

where U is the strain energy per unit of reference volume with U_{dev} as its deviatoric part, U_{vol} as its volumetric part and J_{el} represents the elastic volume ratio. The deviatoric part of the strain energy potential $\bar{I}_1$ is defined by providing biaxial test data. A 91% seal compression, similar to that used by Stenti *et al.* [7], which represents the door seal compression level of a typical sedan vehicle, was used in all cases under investigation.

3.3 Energy dissipation model

In general, three modes of damping are dominant in this study, namely hysteretic, viscoelastic and Coulomb damping; the latter is due to slippage in the bolted hinge connections. However, determining these damping characteristics in complex structures is a very difficult process. Therefore, the approximate spectral modal damping scheme is employed, which introduces an energy dissipation term δW_{diss} of the form

$$\delta W_{diss} = 2 \zeta_i\, \omega_i\, \dot{u}_i \tag{5}$$

where ζ_i is the *i*th modal damping ratio, ω_i is the natural frequency and $\dot{u}_i$ is the velocity. Acquisition of the damping factors are described in paragraph 6.2.

4 Mathematical formulation

4.1 Definition of external loading conditions

Measurements performed by Nel [8] on a sedan driven on a representative highway road surface suggested a typical wheel suspension input force of 100 N. In order to preserve dynamic similarity to an actual vehicle, application of the relevant scaling laws suggests that the model under investigation be excited by an 11 N point force. This vertical harmonic force was applied sinusoidally at the front right-hand suspension point as a frequency sweep in the frequency range of 20 to 200 Hz in 0.25 Hz increments and is represented in classical form as

$$F = 11 \sin \omega t \tag{6}$$

4.2 Governing equations

The vehicle structure and its respective components are modelled as deformable bodies comprising flexible wall panels enclosing an acoustic cavity. Testing revealed that the natural frequencies for both the structure and acoustic medium span the same range, with only one acoustic mode being identified at 184 Hz. Thus, a coupled approach is adopted as these “vibro-acoustic” modes provide the information necessary to understand the physical phenomena studied.

Thus, the fully-coupled fluid-structure interaction system may be represented in a finite element type discretisation matrix form by

$$\begin{bmatrix} M_s & 0 \\ A & M_f \end{bmatrix} \begin{Bmatrix} \ddot{u}_s \\ \ddot{p}_f \end{Bmatrix} + \begin{bmatrix} C_s & 0 \\ 0 & C_f \end{bmatrix} \begin{Bmatrix} \dot{u}_s \\ \dot{p}_f \end{Bmatrix} + \begin{bmatrix} K_s & -A^T \\ 0 & K_f \end{bmatrix} \begin{Bmatrix} u_s \\ p_f \end{Bmatrix} = \begin{Bmatrix} f_s \\ f_f \end{Bmatrix} \tag{7}$$

where **M**, **C** and **K** are the mass, damping and stiffness matrices with the subscripts s and f representing the structure and fluid medium, respectively. The non-symmetric matrix **A** represents the coupling between the structure and the cavity and f is the external force vector. These equations are solved in ABAQUS using the modal superposition procedures.

4.3 Boundary conditions

At the acoustic-structural boundary where the motion of the acoustic medium is directly coupled to the motion of the solid structure, the acoustic and structural media have the same acceleration normal to the boundary, so that

$$\mathbf{n}.\ddot{\mathbf{u}}^{f} = \mathbf{n}.\ddot{\mathbf{u}}^{s} \tag{8}$$

where the vector **n** represents the inward normal to the acoustic medium at the boundary. In this study, the dynamics of the door hinge systems are analysed under free-free boundary conditions. Furthermore, welded hinge connections are specified as zero-value boundary conditions and are modelled as a fixed connection.

5 Numerical modelling

5.1 Contact interactions

Deformable-to-deformable, small-sliding contact conditions are imposed on the model where applicable employing the classical Coulomb friction model. The bottom of the bolt heads form contact bearing surfaces with the top surfaces of the hinge flanges lying directly beneath them. All other respective contact surfaces are fully defined. Lateral slip of these mating components occurs if the critical frictional shear stress limit is surpassed by lateral forces developed in the system.

5.2 Mesh design

Since geometry-based modelling was used to facilitate the improvement process, detailed, finite element models were created using a wide variety of appropriate finite elements and material properties in order to represent the behaviour of the various interacting bodies, adequately. The model essentially contains second-order structural and acoustic elements to model the vehicle structure and passenger cabin, respectively.

Since material properties affect mesh parameters for wave problems and hence affect the accuracy of the solution, the discretisation protocol of the FE method require at least six nodes per wavelength, as cited by Lim [9]. All meshes were check in terms of standard mesh quality protocols.

5.3 Step definitions and analysis methodology

The complexity of the cases under investigation demands a multi-step approach. In general, the complete finite element simulation progressed in seven steps. Steps 2 and 3 were omitted for welded connections. The general methodology entails the following:

Step 1: Non-linear static analysis.
Purpose: To establish contact and the desired door seal compression state.
Step 2: Non-linear static analysis.
Purpose: To establish bolt contact and apply bolt pre-tensioning loads.
Step 3: Non-linear static analysis.
Purpose: To fix the bolts at their current lengths.
Step 4: Non-linear static analysis.
Purpose: To include the effects of gravity on the door hinge system.
Step 5: Linear perturbation static analysis.
Purpose: To include the computation of residual modes.
Step 6: Linear perturbation frequency analysis.
Purpose: To extract the coupled eigenmodes of the system.
Step 7: Steady-state dynamic, mode-based analysis.
Purpose: To determine the vibratory response of the coupled system subjected to a multi-frequency harmonic excitation.

6 Experimental characterisation

6.1 General test set-up

The test structure was fabricated and assembled in a similar way to that of an actual sedan vehicle. Air gaps in the vehicle body were sealed with high-quality non-porous duct tape in order to acoustically seal the interior cavity, thus minimising acoustic losses. The test vehicle was supported using four soft elastic bands in an overhead fashion to simulate the free-free boundary conditions used in the dynamics calculations. The same set-up was used for all subsequent tests on the other door hinge configurations described in table 1.

6.2 Modal analysis

The experimental modal analysis was carried out using the so-called single reference testing method employing a miniature Dytran 3023M2 tri-axial accelerometer of mass 3 grams in conjunction with standard equipment comprising a PCB 086 C03 modal hammer linked to a four-channel DSP SigLab spectrum analyser via Piezotronics PCB 480E09 signal conditioning units. A PC was used for collecting and managing the acquired measurements.

A total of 113 equally distributed excitation points have been selected with a spatial resolution of 80 mm in the three global directions, resulting in a well defined experimental model for the frequency range of interest. Since closely spaced modes were expected, the required modal parameters including damping were estimated from 339 measured FRF accelerance spectra (113 points × 3 global directions) using the Global-M frequency domain curve-fitting algorithm found in the MODENT modal-analysis software package.

6.3 Forced response analysis

A steady-state, forced response, harmonic analysis was carried out in order to measure the interior SPLs. The interior acoustic response was measured using a precision TMS 130P10 ¼-inch condenser microphone. In order to minimise mounting resonances, the microphone was attached at the driver's head location in an overhead fashion using light elastic bands and duct tape.

A MB Dynamics M50A electromagnetic shaker representing the road suspension input was carefully aligned and connected to the front right-hand vehicle floor panel via a 2 mm diameter, 15 mm long steel threaded stinger coincident with that of the finite element model. The spectrum analyser and shaker were linked to a MB Dynamics SS250 amplifier. The shaker was driven with a swept sine signal, which was band limited to the frequency range of interest coincident with that of the finite element model. The data was sampled at 512 Hz with a 0.25 Hz resolution corresponding to the numerical model. The experimental set-up is depicted in fig. 2 below.

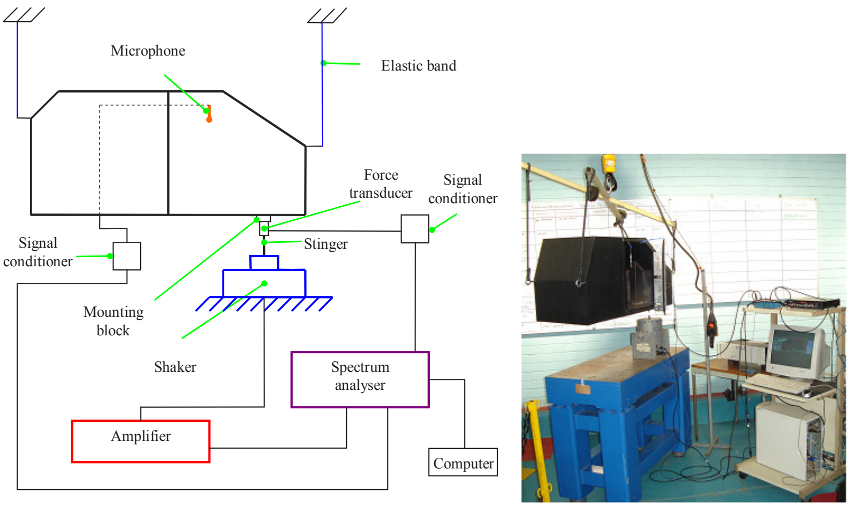

Figure 2: Experimental set-up for measurement of interior SPLs.

7 Sensitivity analysis

7.1 Criteria and objective function definition

In this work, acoustic-structural sensitivity analysis is defined as the mathematically controlled modification of the passive door hinge system, with regard to its geometrical properties, to facilitate feasible improvements to a certain objective function. In order to avoid focusing on the dominant peaks whilst considering the entire frequency range of interest, the average SPL at the driver's head location, is defined as the objective function to be minimised, with respect to the fluid's bounding structural geometry.

Considering a certain set of design variables $\mathbf{X}$, chosen to comprise the method of hinge fixation (welded or bolted) and the position of the door latch and striker mechanism, the objective function $\varPhi$, may be formulated as

$$\varPhi(\mathbf{X}) = \frac{1}{\omega_{\max} - \omega_{\min}} \int_{\omega_{\min}}^{\omega_{\max}} p_{\mathrm{i}}(\omega, \mathbf{X})\, d\omega \quad \Rightarrow \quad \min. \tag{9}$$

subjected to the inequality constraint $p_{\mathrm{i}} > P_{ref}$, where p_{i} is defined by eqn (3). P_{ref} has been introduced into the objective function as testing revealed that problems might be encountered at certain points where the SPL approaches zero, thus causing infinite values in the sensitivity calculation to be generated.

7.2 Procedure

In this work, four door hinge system configurations, described in table 1, are modelled and investigated.

Table 1: Description of change parameter cases.

Case	Description	
	Fixation method at A-pillar: door panel	**Door latch position**
Reference	bolt: bolt	bottom
1	bolt: bolt	middle
2	bolt: bolt	top
3	weld: weld	middle

Since no improvement modules are available in the numerical code for vibro-acoustic analysis, a special improvement algorithm, in the form of a slave module that interacts directly with the output database file was created in PYTHON code and was used to interrogate the FE output database in order to compute the design change in terms of the objective function. In addition, since the acoustic mesh is different for each case due to substantial changes in the cabin's internal geometry, the improvement process had to be manually controlled within the FE code. The human effort involved with generating the various acoustic meshes should not be disregarded because of the complex geometry involved, as this makes the creation very labour intensive and tedious.

Modification domains with respect to their acoustic properties were identified for the door hinge systems in terms of the design variables. All geometric modifications (design variables) were parameterised and fully defined in order to minimise the number of design variables and allow the improvement procedure to become more manageable, whilst keeping the variety of new door hinge systems broad. Once the FE input files for the various cases were developed, a PYTHON code (Slave Module 1), was developed to run all model simulations consecutively in a single command. The performance profile of each case was evaluated in terms of the objective function, until an improved design solution was achieved as illustrated in fig. 3.

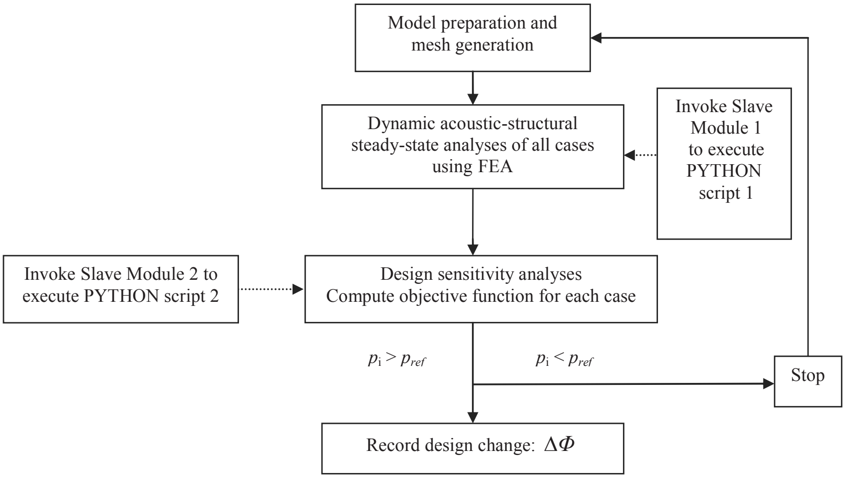

Figure 3: Design sensitivity improvement routine.

8 Results

8.1 Natural characteristics and acoustic signatures

The experimentally-determined damping ratios varied between 1.5% to 4.8% and were subsequently used as input into the respective FE models. The measured natural frequencies varied between 40.80 and 197.48 Hz. With a model size of approximately 1 million degrees-of-freedom (including contact elements and Lagrange multiplier variables), the reference case converged at 24.28 hours of CPU time in 732 increments. The analyses were executed on a 3 GHz dual Xeon Pentium Workstation with 4 GB of RAM, running on a Windows XP operating system. The left-hand plot in fig. 4 depicts the evolution of the SPL at the driver's head location for the various cases under investigation, used to compute the objective function defined by eqn (9). The correlation between the predicted and measured values for the reference case is shown in the right-hand plot.

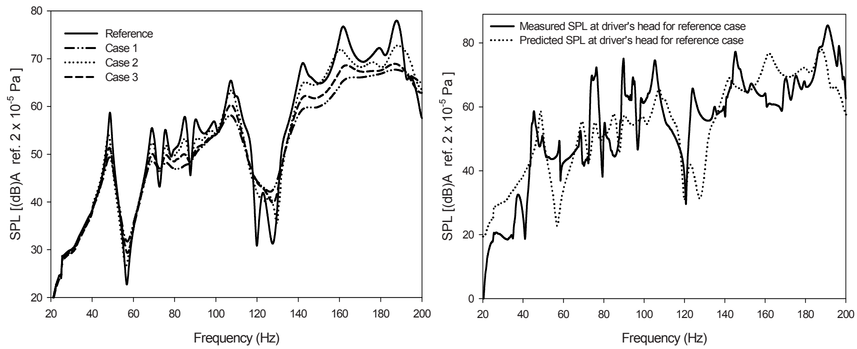

Figure 4: SPLs at driver's head location.

8.2 Objective functions

The sensitivities of the objective function with respect to the various design variables and the eigen characteristics are summarised in table 2 below.

Table 2: Comparison of eigen properties, objective functions and sensitivities.

Case	**Fundamental mode (Hz)**	**Number of modes in frequency range of interest**	**Objective function Φ (dB)A**	**Percentage change in objective function $\Delta\Phi$**
Reference	39.93	44	67.10	
1	39.92	42	60.95	− 9.17
2	39.88	43	64.75	− 3.50
3	39.92	42	62.12	− 7.42

8.3 Results of the improved design

Based on the results of the sensitivity process, it is clear from table 2 that case 1 yielded the lowest objective function value. Hence, this case is proposed as the improved door hinge system. Fig. 5 shows the correlation between the predicted and measured SPLs. The acoustic-structural coupling at around 187 Hz is also verified.

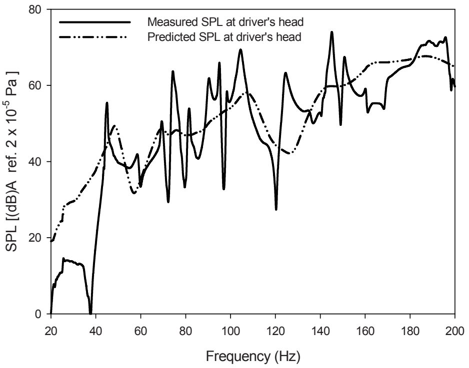

Figure 5: Comparison of predicted and measured SPLs for improved solution.

9 Discussion

By comparing the experimental and numerical results, it is evident that the overall characteristics are similar to each other, despite the complexity of the structure under investigation. The results illustrate that the model predicts the important resonant peaks as well as the general trend of the vibro-acoustic behaviour. Differences may be attributed to estimation of damping, noise within the measuring equipment, limitations in the numerical procedure and physical measurement process such as the phenomenon of “force drop-out” around resonances; neglecting stiffness residue, pre-strain; and panel curvature effects due to welding of the test structure that have not been explicitly modelled.

It can be inferred from the simulations and experimental measurements that the structural and acoustic modes couple with each other at around 187 Hz, where both mediums have similar wavelengths. Hence, the acoustic-structural mode is experimentally verified.

Compared to real vehicles, the SPLs reported may seem high at first glance. However, this difference is primarily due to the effect of the scaling factor applied to the vehicle model. It can also be deduced that the objective function is most sensitive to the position of the latch. The latch is therefore ideal to use as a design variable for the improvement of the door hinge system and must be accurately characterised. The objective function is not very sensitive to the hinge fixation method. However, it seems that bolted hinge connections are a slightly better alternative to welded hinge connections due to their inherent damping characteristics and thus should be exploited.

10 Conclusion

A computational procedure for predicting vibro-acoustic transmission in terms of the door hinge system has been proposed and demonstrated by FE analysis. Compared to the reference design, a reduction in the objective function value of 6.15 dB(A) representing a 9.17% decrease was achieved by using the proposed improvement procedure. The results further revealed that the effect of relatively small design modifications should not be underestimated. It is believed that the proposed modelling procedures and results presented would bring a greater understanding of the behaviour of door hinges in a coupled setting and would be of practical benefit in cabin structural design and NVH analysis.

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