Simulation of Vibro-acoustic Performance of an Automotive Door Hinge System in Relation to Its Hinge Separating Distance

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Abstract— **The reduction of automotive interior noise is an important concern in noise, vibration, and harshness (NVH) design for various reasons. Amongst them is the fact that the noise and vibration signatures of the passenger cabin contribute substantially towards the overall impression of the vehicle quality and therefore has a great influence on the buying decision of the customer. Because the time to market is constantly decreasing while the pressure to save costs is increasing, NVH engineers seek to define and optimize a vehicle's vibro-acoustic comfort characteristics as early as possible. This paper describes the development of a predictive finite element (FE) acoustic-structural analysis tool to investigate the vibro-acoustic behavior of the passenger cabin, as anecdotal evidence suggests that, until now, such characterization and its subsequent noise emission characteristics in terms of its door hinge system has not yet been contemplated or studied. Thus, the primary contribution of this original work entails the development of three-dimensional, numerical models to address this unique application in terms of noise generation and mitigation. The commercial code ABAQUS® is employed to simulate the complex door panel vibration behavior and acoustic wave propagation within the passenger cabin as a function of the door hinge separation distance when provoked by an external forcing function. With reference to an existing, typical door hinge system, frequency domain numerical models are developed and solved employing experimentally-determined damping ratios. The validity of the developed numerical models is experimentally verified based on frequency response functions, the results of which show good agreement. The results further revealed that the effect of relatively small, low-cost design modifications should not be underestimated.**

Index Terms—**Automotive door hinges, automotive interior noise, finite element analysis, fluid-structure interaction, NVH**

I. INTRODUCTION

STRICTER standards for noise emissions is forcing the automotive industry to go beyond what has been done in the past. Acoustic-structural interaction has been a dominant research area in aero-elasticity since the early sixties, with varying degrees of success. At present, a number of symposia have been devoted entirely to this class of problem, which has now gained considerable engineering interest. Though there have been extensive studies on sound radiation from vibrating plates with classical boundary conditions [1], [2], no comparable efforts have been made to determine the relative contribution of accompanying structural components with non-classical boundary conditions such as automotive door hinges. Therefore, in terms of its contribution to existing research, the investigation described in this work is relatively unique in synthesis, strategy, and application.

Furthermore, refinement of interior noise in today's vehicles is a very challenging process largely due to the fact that the modern passenger cabin is high refined in terms of its vibro-acoustic behavior. The acoustic signatures of primary noise sources such as engine, tire, and exhaust, together with their transmission paths and their related phenomena, have been well understood and controlled. Consequently, the relative contribution of secondary-level noise sources, such as vibrating structural surfaces like door panels, provoked by structure-borne components like wheel suspension inputs caused by undulations and road surface irregularities has become the limiting factor in the quest to improve passenger comfort. The resulting structural door panel vibrations with their subsequent deformations are hypothesized to be a secondary cause of passenger cabin noise.

In addition, these deformations, relative to the vehicle's structural frame, could facilitate air-borne acoustic paths into the passenger cabin, thereby increasing interior noise levels even further. Because of the high costs associated with tooling, it has become highly desirable to predict the dynamic behavior of components such as automotive door panels, during the initial design stage. This would allow the door panel to be "tuned"; in other words, its frequency is chosen, so that resonance can be shifted, either out of a troublesome range, or into a range of frequencies where palliative treatments might be more effective.

However, Jennequin [3] indicated that the nature of noise inside a vehicle is complex, because it is determined inter alia by the behavior and interaction of the vehicle structure, the vibrating door panels, and the air cavity within the vehicle itself. Since the passenger cabin of a vehicle forms a cavity, resonant conditions can develop, characterized by acoustic modes of vibration. 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

Manuscript received July 10, 2014; revised July 29, 2014. This work was supported by the Tshwane University of Technology, South Africa and Ford Motor Company, South Africa.

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panels, acoustic resonance can result, giving rise to objectionable low-frequency boom noise. According to Morrey and Barr [4], most internal noise is associated with the dynamic behavior of the vehicle body structure at frequencies below 400 Hz.

Furthermore, market research reveals that automotive door hinges are designed on the basis of strength only. Sometimes, selection procedures for door hinges are based on simple assumptions which are frequently of an anecdotal or historical nature. It is felt that these approaches may lead to conservative designs. Because of the challenges faced by automotive manufacturers in terms of passenger comfort, it has now become necessary to select door hinges that satisfy both strength and noise generation criteria. In order to effectively and economically achieve this, a scientific approach needed to be followed. For this reason, a predictive simulation model is developed in this work to study the low-frequency vibro-acoustic performance of an automotive door hinge system as a function of its separating distance. The flexibility of such a model can provide a powerful tool for in depth strength and noise investigations without the problems often correlated with trial and error methods. Interestingly, preliminary studies conducted by the author of this work showed considerable promise and indicated that the finite element method could be used to simulate acoustic-structural behavior as a function of a door hinge system in the low-frequency range. However, the inclusion of complex phenomena, such as multi-body contact, hyper-elasticity, interactions, and damping, could present different scenarios, the considerations of which are included in this study.

II. PROBLEM DESCRIPTION

The cases under investigation consist of a simplified, reduced-scale automotive vehicle, with its front right-hand door panel and respective upper and lower door hinge assemblies, a hyper-elastic 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. A stiff vehicle structure was constructed using 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. Actual in-service conditions in terms of loading (scaled accordingly), contact, friction, and damping phenomena have been considered.

In order to investigate the influence of hinge separating distance on the cabin vibro-acoustic behavior, two configurations are studied namely: a bolted door hinge configuration with normally spaced hinges (150 mm apart, hereby named as the Reference case) and another configuration with bolted door hinges closely spaced (120 mm apart, referred to as Case 1) with both cases having a centrally position door latch mechanism.

III. MODELLING APPROACH

The complexity of the cases under investigation demands a multi-step approach. The modelling method employed consists of a mixed, fully-coupled noise-vibration analysis in order to represent the dominant physics involved. The complete finite element simulation progressed in seven steps, namely:

Step 1: Nonlinear static analysis.
Purpose: To establish contact and the desired door seal compression state.
Step 2: Nonlinear static analysis.
Purpose: To establish bolt contact and apply bolt pre-tensioning loads.
Step 3: Nonlinear static analysis.
Purpose: To fix the bolts at their current lengths.
Step 4: Nonlinear 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 vibro-acoustic response of the coupled system subjected to a multi-frequency harmonic excitation.

The final analysis, which employs modal techniques, was based on FRF simulations of the fluid-structure interaction between the passenger cabin and its body structure with respect to the structural excitation. The vehicle structure and its respective components are modelled as deformable bodies comprising flexible wall panels enclosing an acoustic cavity. Since testing revealed that the natural frequencies for both the structure and acoustic medium span the same range, a coupled approach is adopted as these "vibro-acoustic" modes provide the information necessary to understand the physical phenomena studied.

According to Zienkiewicz and Taylor [5], the fully-coupled fluid-structure interaction system may be represented in a finite element type discretisation matrix form by

$$\begin{bmatrix} M_s & 0 \\ A & M_f \end{bmatrix}\begin{Bmatrix} \ddot{u}_s \\ \ddot{p}_f \end{Bmatrix} + \begin{bmatrix} C_s & 0 \\ 0 & C_f \end{bmatrix}\begin{Bmatrix} \dot{u}_s \\ \dot{p}_f \end{Bmatrix} + \begin{bmatrix} K_s & -A^T \\ 0 & K_f \end{bmatrix}\begin{Bmatrix} u_s \\ p_f \end{Bmatrix} = \begin{Bmatrix} f_s \\ f_f \end{Bmatrix} \quad (1)$$

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. $\mathbf{p}_f$ is the acoustic pressure with $\ddot{\mathbf{p}}_f$ and $\dot{\mathbf{p}}_f$ its second and first derivatives, respectively, at the grid points or nodes of the finite element model, which discretizes the air volume of the passenger cabin. The non-symmetric matrix **A** represents the coupling between the structure and the cavity and f_s is the external impact force vector. These governing equations are solved in ABAQUS using the modal superposition procedure.

In this work, the popular EPDM sponge-dense, elastomeric door seal is used. The door seal is modelled with the hyper-elastic material model, which is a nonlinear, 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 [6] and applied by Wagner *et al.* [7].

The Marlow strain energy potential may be expressed in the form

$$U = U_{dev}(\bar{I}_1) + U_{vol}(J_{el}) \quad (2)$$

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, Moens and Desmet [8], which represents the door seal compression level of a typical sedan vehicle, was used in all cases under investigation.

Since the acquisition of actual damping characteristics in complex structures is very difficult, an approximate spectral modal damping scheme is employed, which introduces an energy dissipation term δW_{diss} of the form

$$\delta W_{diss} = 2\zeta_i\,\omega_i\,\dot{u}_i \quad (3)$$

where ζ_i is the ith modal damping ratio, ω_i is the natural frequency and $\dot{u}_i$ is the velocity. The popular Coulomb friction model with limit on the shear stress is employed in this study. Finally, the sound pressure level (SPL) at a particular location p_i may be defined as

$$p_i = 20\log_{10}\left(\frac{P/\sqrt{2}}{P_{ref}}\right) \quad (4)$$

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

IV. LOADING AND BOUNDARY CONDITIONS

When driving on typical roads, modern vehicles are often sensitive to low-frequency interior rumble noise, occurring predominantly in the low-frequency range. According to Constant [9], this rumble noise has a structure-borne origin and is mainly excited by road roughness. Therefore, a structure-borne noise and vibration analysis in the frequency range of 20 Hz to 200 Hz due to wheel suspension forces (inputs) caused by undulations and road surface irregularities was considered in this study. In order to preserve dynamic similarity to an actual vehicle, application of the relevant scaling law resulted in a scaled input exciting force of 11 N as tests conducted by Nel [10] on a vehicle driven on a representative highway road surface suggested a typical wheel suspension input force of 100 N. This vertical harmonic force was thus used to generate structural vibrations associated with the effects of road irregularity transmitted via the front right-hand suspension and was applied sinusoidally as a frequency sweep in the frequency range of 20 Hz to 200 Hz in 0.25 Hz increments. The forcing function is thus described in classical form as

$$F = 11\sin\omega t \quad (5)$$

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

$$\mathbf{n}.\ddot{\mathbf{u}}^{f} = \mathbf{n}.\ddot{\mathbf{u}}^{s} \quad (6)$$

where the vector **n** represents the inward normal to the acoustic medium at the boundary.

Since the acquisition of actual damping characteristics in In terms of mesh design, although the meshes at the acoustic-structural tied boundary may be nodally nonconforming, mesh refinement; which depends on wave speeds in the two media, affects the accuracy of the solution. The mesh for the medium with the lower wave speed (air) should generally be more refined and therefore should be the slave surface. Since material properties affect mesh parameters for wave propagation problems and hence affect the accuracy of the solution, the discretization protocol of the finite element method require at least six nodes per wavelength [11]. The finite element model was created using a wide variety of appropriate finite elements and material properties in order to represent the behavior of the various interacting bodies, adequately. The model contains a mix of first and second-order structural elements to model the vehicle structure and first-order acoustic elements to model the passenger cabin. Adaptive meshing was defined at and around the impact zone. All meshes were check in terms of standard mesh quality protocols. The dynamics of the door hinge systems were analyzed under free-free boundary conditions.

V. EXPERIMENTAL SET-UP

The test structure was manufactured and assembled in a similar way to that of an actual sedan vehicle. All air gaps in the vehicle body were sealed with high-quality non-porous duct tape in order to acoustically seal the interior cavity, thus minimizing acoustic losses. The test vehicle was suspended using four soft elastic bands in an overhead fashion to simulate the free-free boundary conditions used in the dynamics calculations.

The experimental modal analysis was carried out using the 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 analyzer 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.

Subsequently, steady-state, forced response, harmonic analyses were carried out in order to measure the door panel vibration and interior sound pressure levels (SPLs). The door panel vibration and interior acoustic response were measured using a miniature Dytran lightweight tri-axial accelerometer model 3023M2 of mass 3 grams and a precision TMS 130P10 ¼-inch condenser microphone, respectively. In order to minimize mounting resonances, the

ISBN: 978-988-19253-7-4
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

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 analyzer 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-ups are depicted in Fig. 1 to Fig. 4 below.

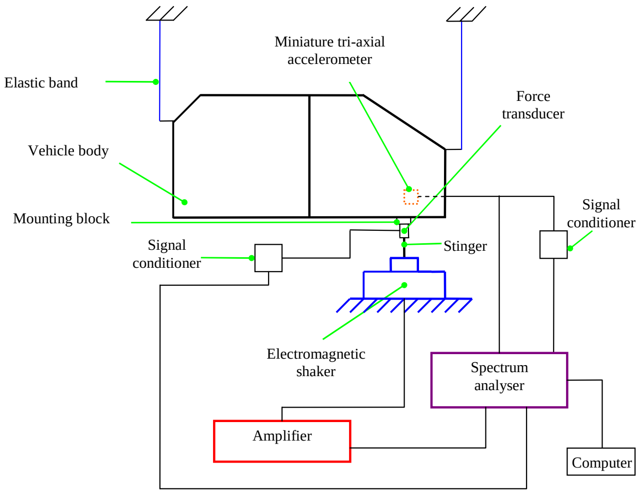

Fig. 1. Schematic illustration of experimental set-up for measurement of door panel vibration.

Fig. 2. Attached accelerometer at door reference point.

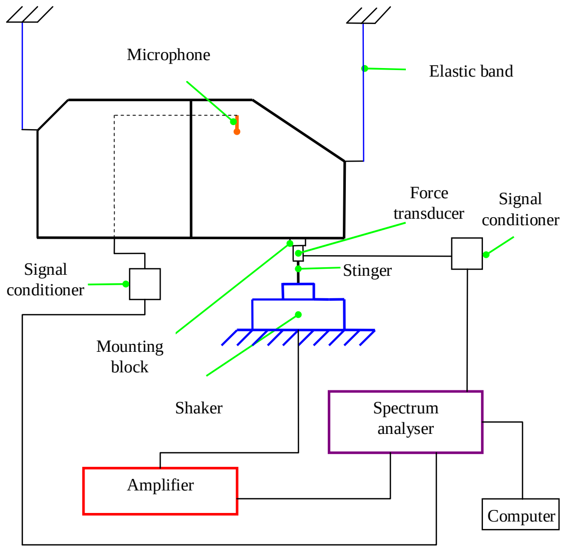

Fig. 3. Schematic illustration of experimental set-up for measurement of interior SPLs.

Fig. 4. Vehicle test set-up with attached microphone at driver's head location.

VI. SENSITIVITY ANALYSIS

Considering a certain set of design variables **X**, hereby chosen to comprise the position of the door hinges, the objective function Φ, may be formulated as

$$\Phi(\mathbf{X}) = \frac{1}{\omega_{max} - \omega_{min}} \int_{\omega_{min}}^{\omega_{max}} p_i(\omega, \mathbf{X})\, d\omega \;\Rightarrow\; \min. \qquad (7)$$

subjected to the inequality constraint $p_i > P_{ref}$, where p_i is defined by (4). 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. 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.

ISBN: 978-988-19253-7-4
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

Since no sensitivity modules are available in the numerical code for vibro-acoustic analysis, a special 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 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 acoustic meshes should not be disregarded because of the complex geometry involved, as this makes the creation very labour intensive and tedious.

The design variable (hinge separating distance) was parameterised and fully defined in order to allow the sensitivity procedure to become more manageable. Once the FE input files for the various cases were developed, a PYTHON code in the form of a Slave Module was developed to run both model simulations consecutively in a single command. The performance of each case was evaluated in terms of the objective function.

VII. RESULTS AND DISCUSSION

The measured natural frequencies varied between 39.92 Hz and 199.01 Hz while the experimentally-determined damping ratios varied between 0.9% and 4.8% and were subsequently used as input into the respective FE models. With a model size of approximately 1 million degrees-of-freedom (including contact elements and Lagrange multiplier variables), the reference case converged at 24.27 hours of CPU time in 699 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.

Fig. 5 below depicts the predicted and measured vibration profiles in the *z* global direction at the inner reference point of the front right-hand door panel for Case 1. The vibration amplitudes in the *x* and *y* global directions were negligible and were therefore omitted from this study. The FRF data are inclusive of residual and RBMs with the standard acceleration reference level of 10^{-6} m/s^2 been used as recommended by Smith, Peters and Owen [12].

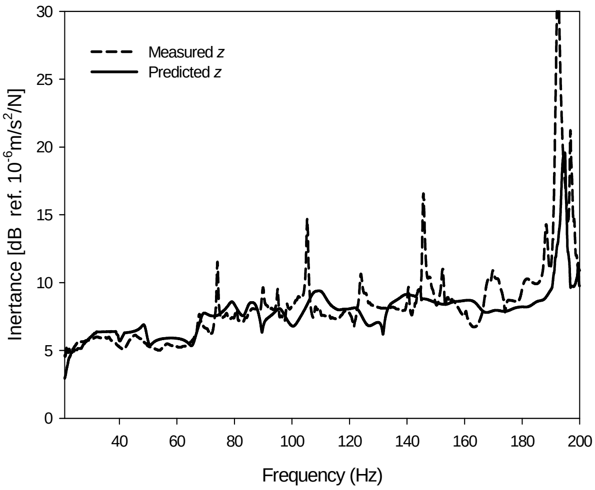

Fig. 5. Door panel vibration signatures for Case 1.

The evolution and correlation between the predicted and measured A-weighted SPLs at the driver's head location for Case 1 is represented in Fig. 6 below.

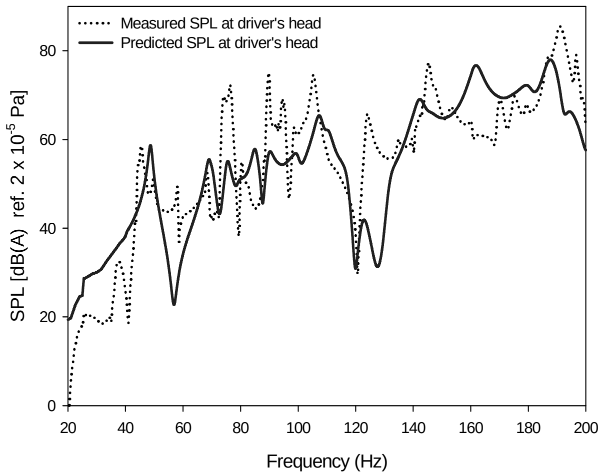

Fig. 6. Comparison between predicted and measured SPLs at driver's head location for Case 1.

For the purpose of comparison, the predicted SPLs at the driver's head location for the Reference case and Case 1, in relation to (7) is shown in Fig. 7.

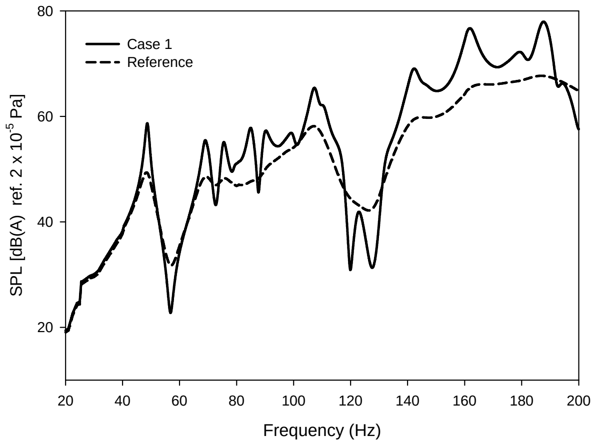

Fig. 7. Comparison of SPL signatures at driver's head location between Reference case and Case 1.

The sensitivities of the objective function with respect to eigen characteristics are summarized in Table I below.

TABLE I
COMPARISON OF EIGEN PROPERTIES, OBJECTIVE FUNCTIONS AND SENSITIVITIES

Case	Funda-mental mode (Hz)	Number of modes in frequency range of interest	Objective function Φ (dB)A	Percentage change in objective function $\Delta\Phi$
Reference	39.92	42	60.95	
1	39.93	44	67.10	+ 10.09

Based on Table 1, it is clear that Case 1 elicits a substantial increase in the objective function value. This implies that a reduction in the hinge separating distance has an unfavorable consequence and should be avoided where possible. 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 numerical 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 very sensitive to the position of the door hinges and therefore ideal to use as a design variable for the improvement of the door hinge system and must be accurately characterised.

VIII. CONCLUSION

In this work, a computational procedure with numerical models, based on the finite element approach, has been presented in order to predict vibro-acoustic performance as a function of the door hinge separating distance. Achieving a high degree of correlation for complex structures, such as the door mount systems under investigation, is difficult because simulation models generally represent an ideal discretized world, in contrast to experiments where variability is present.

Despite this, the developed numerical models were successfully validated by actual measurements and were amenable to furnishing reliable frequency response information at the points of interest. The predicted and measured vibro-acoustic signatures gave interesting and valuable insight into the coupled behaviour of the door hinge systems under investigation. The study demonstrated for the first time the relationship and quantification of panel vibration and interior SPLs as a function of the door hinge system. The insights gained into the noise generation and transmission behavior of the system under investigation may be materialized into realistic design modifications enabling NVH engineers to move closer towards cabin acoustic refinement in pursuit of their quest for quieter passenger cabins.

ACKNOWLEDGMENT

The author would like to express his sincere gratitude and appreciation to the University of Cape Town and Finite Element Analysis Services (Pty) Ltd, both of South Africa for assistance in creating the basic finite element model of the passenger cabin. Special thanks to the University of Pretoria, South Africa for assisting me with the experiments. Last but not least, the author is grateful to Prof. S Marburg of the Technical University of Mommsenstrasse (Germany), for providing useful information on the formulation of the objective function.

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