STUDY OF CAVITY MODAL DAMPING: A NUMERICAL METHODOLOGY FOR ACOUSTIC EVALUATION USING THE FINITE ELEMENT METHOD IN VEHICLE BODIES BASED ON EXPERIMENTAL TESTS.

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Résumé

Ce travail se concentre sur la recherche d'une solution numérique pour les études acoustiques du véhicule et l'amélioration de l'utilité des paramètres numériques "expérimentaux" pour la phase de développement d'un nouveau projet automobile. Plus précisément, cette recherche porte sur l'importance de la cavité d'amortissement modal pour véhicule exerce pendant les études numériques. Cette recherche vise alors à suggérer les valeurs de paramètres normalisés de la cavité d'amortissement modal dans les études acoustiques des véhicules.

Cette valeur normalisée de modal cavité d'amortissement est d'une grande importance pour l'étude de l'acoustique des véhicules dans l'industrie automobile, car elle permettrait à l'industrie de commencer des études de la performance acoustique d'un véhicule neuf au début de la phase de conception avec une estimation fiable qui serait proche de la valeur finale mesurée dans la phase de conception. Il est commun pour l'industrie automobile à atteindre de bons niveaux de corrélation numérique-expérimentale dans les études acoustiques après la phase de prototypage parce que cette phase peut être étudiée par les commentaires de la simulation et les paramètres modaux expérimentaux.

Ainsi, cette recherche suggère des valeurs de cavité amortissement modal, qui sont divisés en deux parties en raison de leur comportement: celui qui va jusqu'à 100 Hz, et un autre au-dessus de cette valeur.

La séquence de cette étude montre comment nous sommes arrivés à ces valeurs.

Mots clefs : Méthode des éléments finis. Contrôle acoustique. véhicule entier. Corrélation expérimentale numérique. Amortissement modal.

Abstract

This work focuses on finding a numerical solution for vehicle acoustic studies and improving the usefulness of the “Numerical experimental” parameters for the development stage of a new automotive project. Specifically, this research addresses the importance of cavity modal damping for vehicle exerts during numerical studies. This research then seeks to suggest standardized parameter values of modal cavity damping in vehicular acoustic studies.

This standardized value of modal damping cavity is of great importance for the study of vehicular acoustics in the automotive industry because it would allow the industry to begin studies of the acoustic performance of a new vehicle early in the conception phase with a reliable estimation that would be close to the final value measured in the design phase. It is common for the automotive industry to achieve good levels of numerical-experimental correlation in acoustic studies after the prototyping phase because this phase can be studied with feedback from the simulation and experimental modal parameters.

Thus, this research suggests values for cavity modal damping, which are divided into two parts due to their behavior: one that goes up to 100Hz, and another above this value.

The sequence of this study shows how we arrived at these values.

Keywords: Finite Element Methods. Acoustic Control. Trimmed body. Numerical Experimental Correlation. Modal Damping.

1 Introduction

This study has been motivated by the conflict during the final stages of the development of a vehicle, as well as by the comparison between the results generated by the simulation team with those acquired from the experimental team.

For an appropriate correlation, it is always necessary to acquire cavity modal damping data originated from prototypes and subsequently assign them to the numerical model. In this manner, the phase of refinement of the numerical results requires a prototype, and this slows the progress of work and research.

With respect to the simulation methods used today, the finite element method (FEM), as described by Braess et al [1] proposes a quite different situation. Ever-increasing demands for greater comfort have elevated the dynamic design criteria as the primary elements of modern body engineering.

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Damping and sound-insulation measures are strictly applied to automotive body panels to prevent noise in the vehicle cabin [2]. Automotive body panels, which are made of steel sheets press-molded into a required form, are laminated with damping materials to reduce the vibration level. Furthermore, porous media, resin sheets (surface) and carpet are laminated and used to work as damping materials [3,4].
For this study 2 (two) different categories of vehicles were investigated: a pick-up truck and a popular compact vehicle. Hence, the study is expected to determine a range of values that cover the cavity damping behavior by analyzing these vehicles in the Trimmed-Body configuration [5].
Acoustic FEM analysis of the system was performed using standard MSC Nastran 2010 software. Following the FEM analysis, a modal analysis of the entire vehicle and cavity was performed; the data were treated as described by Moura et al. [6], in CRF VEIPROD 5.0® software for the evaluation of the SPL (Sound Pressure level).
With respect to the experimental data campaign, the bodyshell (TBIW) testing was performed in a laboratory at NVH in a semi-anechoic room, exploiting LMS Test Lab 11B [7].
Finally, this research work seeks to accomplish the following:
A) Determine the influence of this observed cavity modal damping variation in the physical testing on the simulation models, seeking to better identify the existence of the resonance modes between the cavity and the body-shell.
B) Propose a medium cavity modal damping (Cavity Damping Design) that can be used even in the early stages of vehicle development and provide results similar to those generated using the actual variable damping.
Figure 1 shows a schematic flowchart of the ideas presented previously.

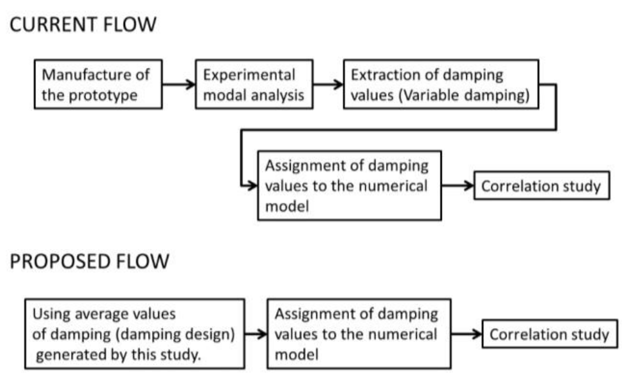

Figure 1: Flowchart of job steps (current and proposed).

2 Experimental methodology

As part of the validation process of the numerical experiment, the first step adopted in this study was to perform experimental modal analysis of the cavity, in which one expects a correlation regarding the global modes and to determine how the damping behavior of these cavities would vary in frequency when comparing various types of vehicles [8].
Consequently, all vehicles (two from different categories) underwent the same instrumentation as shown in figures 2 and 3. The experimental results were obtained by processing the data using LMS Test Lab 11B software. Table 1 shows the list of equipment used for the experimental measurements.

Table 1 : - List of Equipment

Equipment's	Sensitivity / Details
ASQ (Acoustic Source quantification)	41.15 mV/m³/s²
Microphone	50 mV/Pa
LMS Test Lab	Scadas Mobile - Modulo spectral Testing

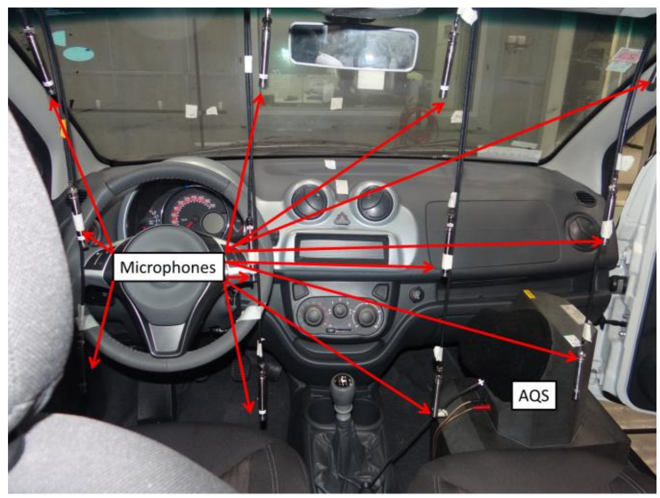

Figure 2: Instrumentation for cavity modal analysis.

The vehicle (TBIW) is placed in an acoustic camera (isolated) with the glass windows closed; a random noise source is placed inside the front and rear of the vehicle for reciprocal testing, and the vehicle has a cavity internally divided in planes defined by microphone chains. The excitation measured by the microphones (SPL) defines the modal behavior of the cavity of the vehicle.
Placement of accelerometers is presented in figure 3.

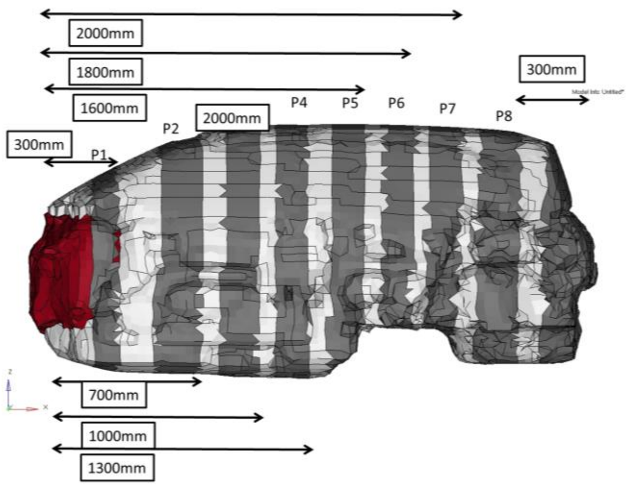

Figure 3: Measurement Planes - modal cavity.

Planes P2 and P7 were chosen to represent the performance of the instrumentation process of the vehicle. This example is shown in Figure 4.

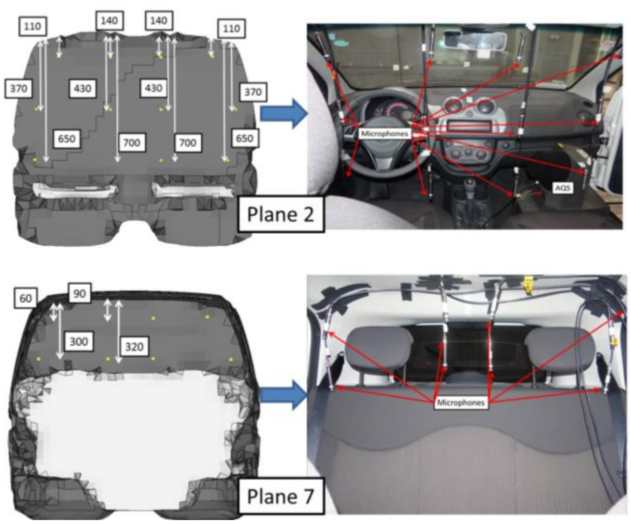

Figure 4: Measurement Plans definition.

For the initial assessment of the vehicle cavity modal behavior in the early stages of its development, the frequency response functions, known as “SPL”, are analyzed at the various points of the microphone positions in the Trimmed-Body configuration. The FRF “SPL” provides the ratio of (P/F), where “P” is the pressure [dB] and the “F” is the force [N] of an excitation point on the structure. The vehicle model is evaluated with the response measured at the points indicated in planes showed at figure 3.

In Figure 5, an example of the results of the experimental cavity model analysis is shown; all vehicles were subjected to the same test.

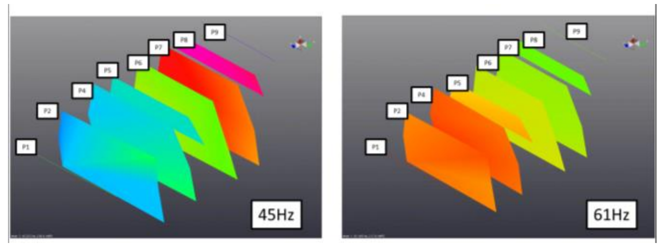

Figure 5: Experimental cavity modal analysis of the studied vehicles.

After plane-by-plane measurements were made, the results were compiled and processed by the LMS-PolyMax (Modulo spectral Testing) method. Figure 6 shows this processing; the left hand side is presented in addition to the overall mode of the cavity, the main modal frequencies of the cavity and its damping. The middle line of this measurement is highlighted at the center of the figure.

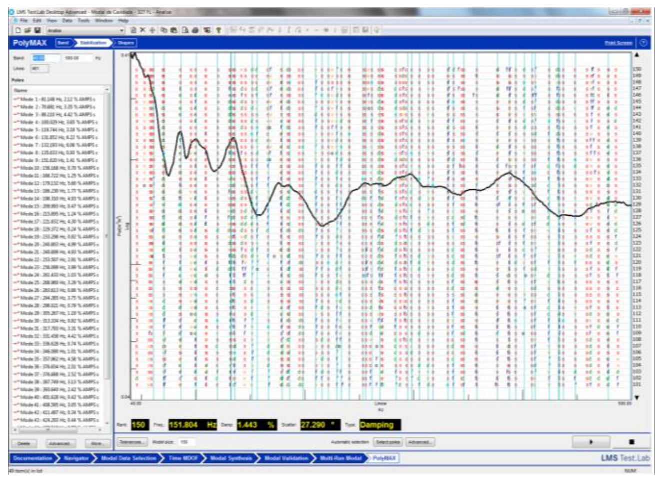

Figure 6: Cavity modal analysis processed by the LMS-PolyMax method.

Figure 7 presents the results of cavity modal analysis. When the measurements were complete, it was possible to extract the modal behavior (Cavity Damping factor) of all vehicles (the two different categories of vehicles) and analyze the results to obtain the value of each modal damping [5,10] of the cavity along the frequency (actual variable damping). This response is presented in figure 7; the damping factor was extracted from a frequency (Hz) sweep.

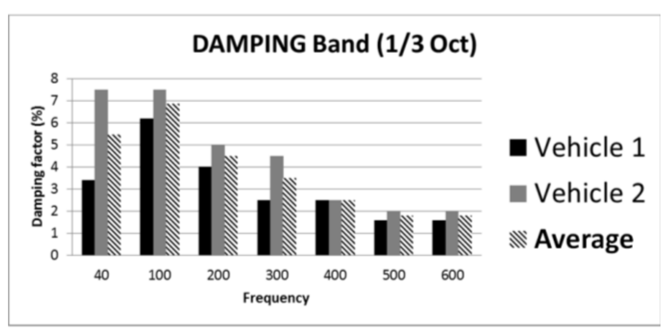

Figure 7: Damping factor response in frequency for different vehicles.

The half-power method was used to find the damping value. Half-power bandwidth is defined as the ratio of the frequency range between the two half power points to the natural frequency at this mode. Thus, although the analysis presented in figure 7 covers the frequency range of 0-500 Hz, the results of the damping factor (%) appear only starting at the 1/3 octave band of 40 Hz where the first natural modes of the cavity begins.

Next, the modal test vehicles were characterized to determine the dynamic "SPL" type. This test is performed with microphones positioned at the height of the right ear of the driver that collected the data as acoustic pressure was generated.

Figures 8 and 9 illustrate the excitation points used by the team during the experimental tests. The impact generated by the impact hammer occurred directly adjacent to the accelerometer presented in this figure.

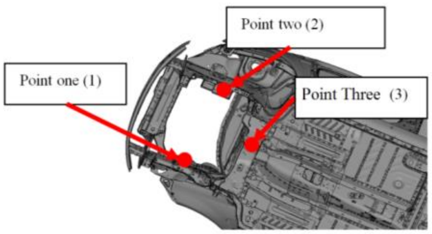

Figure 8 : Layout of experimental test, under view (point 1,2 e 3).

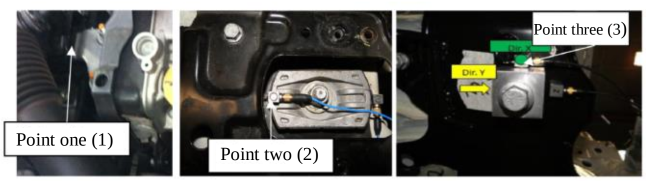

Figure 9 : Layout of experimental test, engine mount (point 1, 2 and 3).

After defining the excitation points, the measured acoustic point detailed above is presented in figure 10 in a Standard Fiat (2004). The figure presents the fixation point of the microphone at the height of the driver's right ear (left side of figure) and details the microphone positioning (right side of figure).

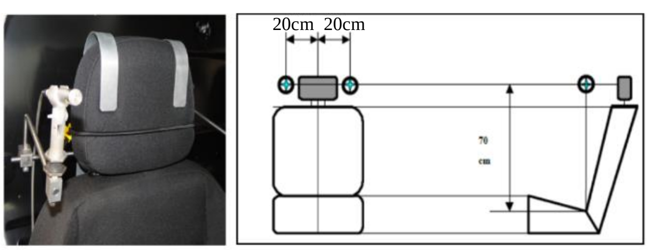

Figure 10 : Experimental configuration of the microphone position (the point of measurement).

With respect to the experimental data campaign, full vehicle testing was performed in a laboratory at NVH in a semi-anechoic room, using LMS Test Lab 11B software. Table 2 shows the list of equipment used for the experimental measurements.

Table 2 : List of equipment

Equipment's	Sensitivity / Details
Impact Hammer	2 mV/N
Microphone	50 mV/Pa
LMS Test Lab	Scadas Mobile - Modulo Impact Testing

The following results allow a comparison with numerical results; therefore, the frequency range used in this study is 0-500 Hz. The new frequency range was selected to concentrate on the influence of damping and avoid any influences caused by numerical errors in high frequency [11].

The results of the "SPL" of the settings shown previously are displayed in Figures 11 (Pickup) and 12 (popular compact vehicle). All of the results bellow were collected at the point illustrated in Figure 9 and are on a logarithmic scale.

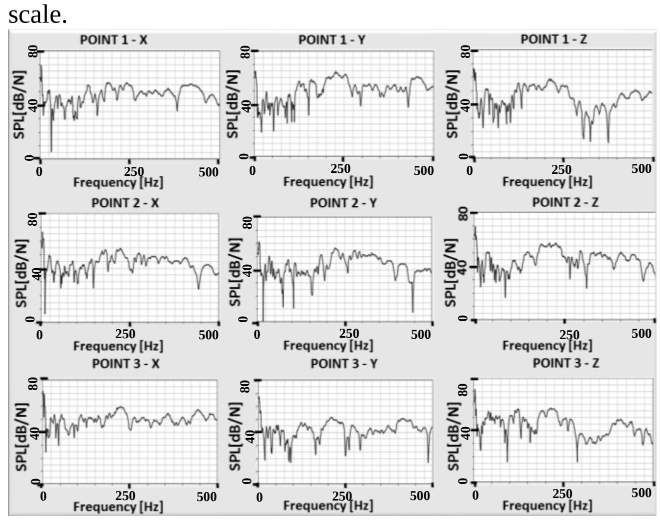

Figure 11: Pickup model; SPL experimental response of the attachment point of the engine. (The ordinate grid step is LOG scale.)

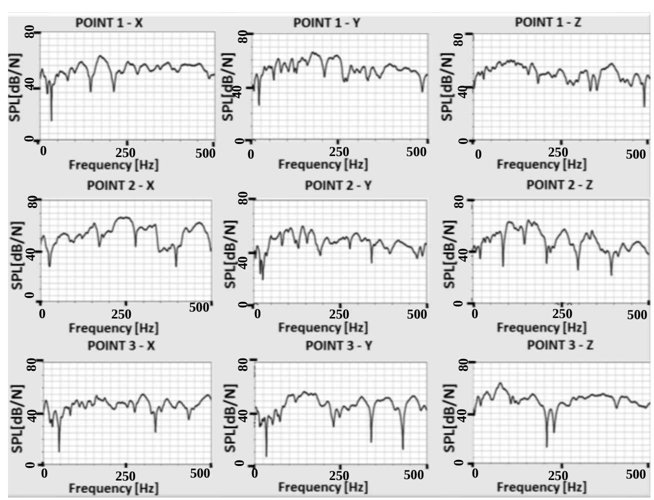

Figure 12: Popular compact vehicle model; SPL experimental response of the attachment point of the engine. (The ordinate grid step is LOG scale.)

3 Numerical formulation

This section describes the details of the FEM models of each vehicle. In addition to the modal behavior of the structure, it is also important to consider the residual vectors to compensate for the higher-order frequencies that are not directly extracted. The mesh size was tuned for the weaker part to have 8 elements for each wavelength at 500 Hz.
Table 3 describes the characteristics that make up each virtual model [12].

Table 3 - Virtual model characteristics.

FE structure – Vehicle 1	FE structure – Vehicle 2
Mass: 352 Kg	Mass: 251 Kg
WELD:5234	WELD:4336
RIGID:622	RIGID:172
RBE3:169	RBE3:0
SPRING:1	SPRING:0
Shell 3 nodes: 15349	Shell 3 nodes: 19122
Shell 4 nodes: 497450	Shell 4 nodes: 593584
Solid 6 nodes: 114	Solid 6 nodes: 120
Solid 8 nodes: 6545	Solid 8 nodes: 7079
Total elements: 536767	Total elements: 633389
Total nodes: 560135	Total nodes: 657227

The numerical models were implemented with damping data according to the experimental results to determine how the damping behavior of these structures would be effected by the frequency response of the various types of vehicles. Additionally, for this numerical study, we used the same two different categories of vehicles: a pick-up truck and a compact vehicle.
For this research, all vehicles were subjected to the same procedure in which HyperMesh 11.0 was used for all computational pre-processing, and NASTRAN software was used for the processing. Figure 13 illustrates the numerical models of vehicles in the Body-in-White configuration.

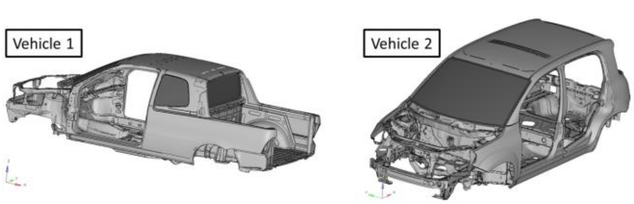

Figure 13: Numerical models in Body White version.

4 Numerical and experimental correlation

The responses of the numerical models were loaded with their respective cavity modal damping (actual variable damping, experimentally extracted from their respective prototype) presented in figure 7 and then generated using the SPL acoustic curves. These curves, Experimental (continuous line) x Numerical (dashed line), were extracted and compared and are shown in Figures 14 and 15. All results are on a logarithmic scale.

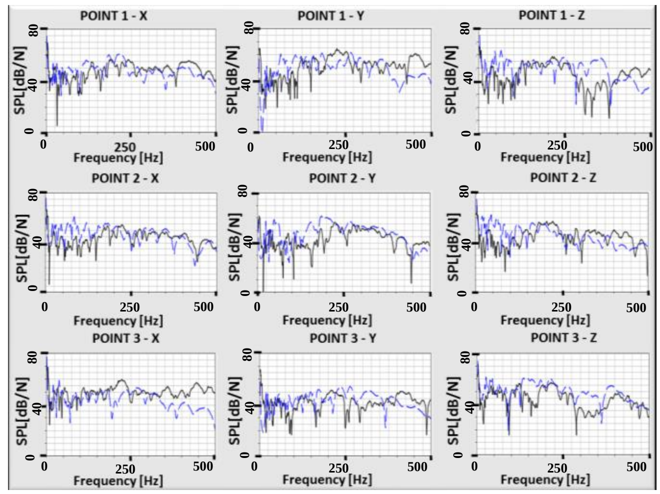

Figure 14: Pick-up model; Numerical and experimental SPL confrontation of the engine mount. (The ordinate grid step is LOG from 0.1.)

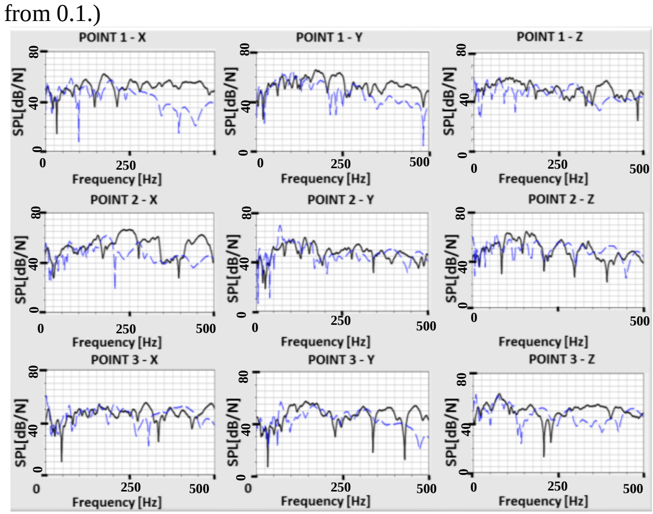

Figure 15: Popular compact vehicle model; Numerical and experimental SPL confrontation of the engine mount. (The ordinate grid step is LOG from 0.1.)

4.1 Data analysis

Based on the previous results, it is noticeable how the numerical results compare to the experimental results. Next, the challenge that faces NVH engineering simulation is to obtain a standard value of cavity modal damping for the TBIW vehicle model to present the same level of correlation when used with the actual experimentally measured cavity damping values.

Thus, based on the values in figure 7, a study was conducted to understand the variation of the average damping of these vehicles and the individual difference among them.
This result is shown in figure 16.

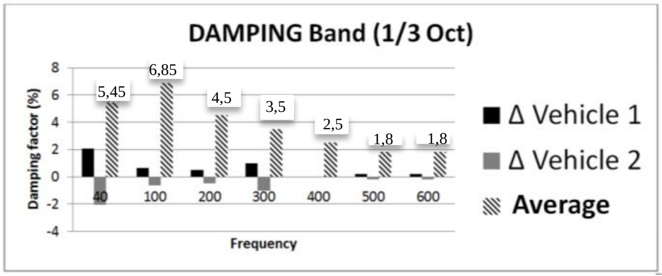

Figure 16: Overall average of the damping factor for all cavity vehicles and the individual differences.

Observing the behavior of the average variation of the cavity damping along the frequency, this graph can be separated into two regions: one up to 100 Hz and the other up to 600 Hz. For this reason, this study was remodeled by remaking an overall average up to 100 Hz and an average up to 600 Hz. With this methodology, we reached a modal damping that varies in its mean obeying a decreasing damping function.
In this first stage, we have a 6.15% cavity modal damping (100 Hz< x), and in the second step, we have a variable function: $\xi(x)=-0.0126(x-100)+6.15$ (to 100 Hz< x <600 Hz). Figure 17 illustrates this new average and the individual differences of these vehicles at this new average.

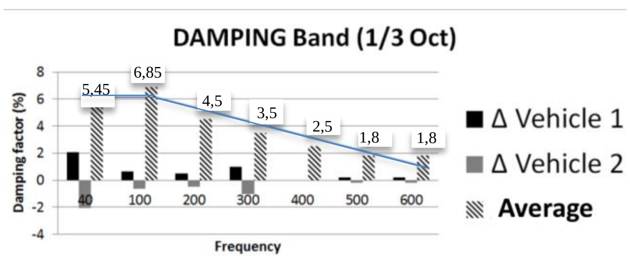

Figure 17: New general average damping factor (Damping_Design).

With a new average damping (here called Damping_Design), new results of Sound Pressure Level (SPL) were generated. Figures 18 and 19 compare the numerical results, one with damping measured experimentally and the other generated by this study (Damping_Design). The curves, Experimental (continuous black line), numerical results with damping experimental values (dashed blue line) and numerical results with "damping design" (continuous red line with triangle marks) from the average presented above, were extracted and compared and are shown in these figures. All results are shown on a logarithmic scale.

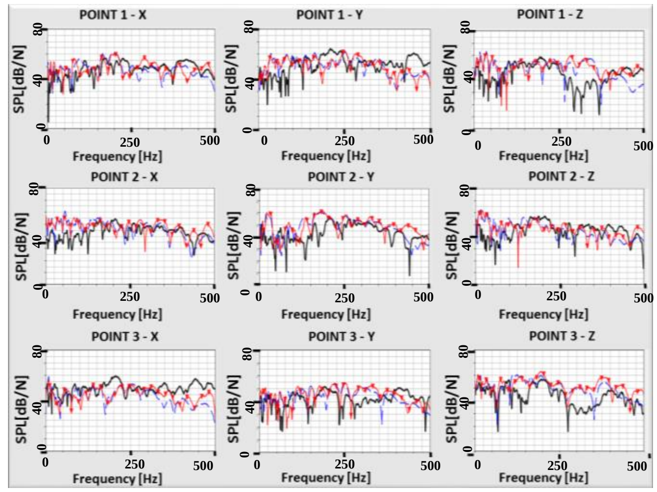

Figure 18: Pick-up model; Numerical confrontation of SPL models with experimental damping and Damping_Design of the engine mount point [1,2 & 3]. (the ordinate grid step is LOG from 0.1).

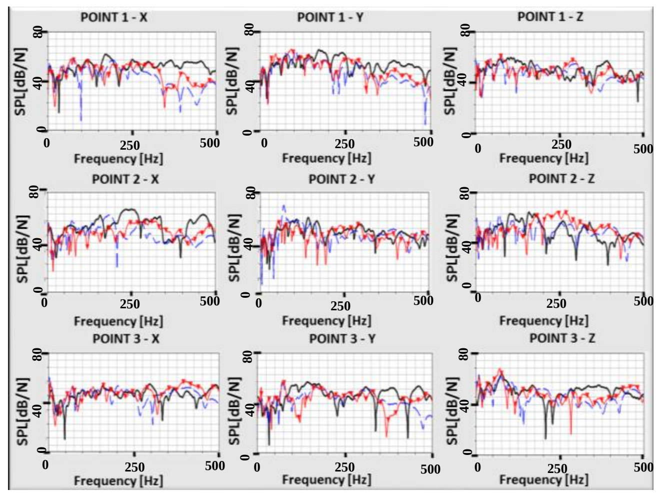

Figure 19: Popular compact vehicle model; Numerical confrontation of SPL models with experimental damping (Constant Cavity Damping – 6%) and Damping_Design of the engine mount point [1,2 & 3]. (the ordinate grid step is LOG from 0.1).

Analyzing the results illustrated in Figures 18 and 19, the numerical response loaded with their respective cavity modal damping (actual variable damping, experimentally extracted from their respective prototype) and the numerical response using the average of damping (Damping Design) developed in this paper show that the response sound pressure level (SPL) was very close to experimental response (black continuous curve).
This is observed in both vehicles used in this research and at the three points of the engine mount. The points with their respective directions of excitation that do not show good correlation responded best when the damping function was used. An example of this is point 3 in the X direction the pick-up and point 1 for the X direction.
It is also possible to see from the analysis of the results shown in Figures 18 and 19 that the use of the function $\xi(x)=-0.0126(x-100)+6.15$ (to 100 Hz< x <600 Hz) and 6.15% (to 30 Hz<x<100 Hz) to represent the modal

damping of the cavity of the vehicles used maintained a good numerical-experimental correlation of sound pressure results.

5 Conclusions

Based on the results, there was a good numerical-experimental correlation when using a modal cavity damping function extracted experimentally and including it in the numerical models. Thus, this research sought to ensure this same performance using a standard damping (Damping-Design) as input as could be used to identify the acoustic behavior of vehicles in the general TBIW configuration.

This study used two (2) types of vehicles as a sample to ensure reliable coverage of the results.

It was observed that the results used for modal damping vary in their mean, obeying a decreasing function such as ξ(x=-0.0126(x-100)+6.15 up to 100 Hz and 6.15%; up to this and when applied to the numerical model, it was observed that the performance in which the variation of the initial result (with damping retrieved from experimental measurements) is very small, less than 0.5 dB. Thus, making these values an appropriate option for the standardization of values of “cavity modal damping” for acoustic analysis in the early stages of a new automotive project.

Acknowledgment

The author, and student, thank the generous support of the Pontificia Universidade Catolica de Minas Gerais – PUCMINAS; “Coordenação de Aperfeiçoamento de Pessoal de Nível Superior”- CAPES - "National Counsel of Technological and Scientific Development"; Conselho nacional de desenvolvimento cientifico e tecnológico-CNPq – “National Council of Scientific and Technological Development”; Fundação de Amparo à Pesquisa de Minas Gerais – FAPEMIG – “Foundation for Research Support of Minas Gerais”; also to FIAT Automóveis company and IFMG- Federal Institut of Minas Gerais.

References

[1] Braess H.H., Seiffert U., 2005, "Handbook of Automotive Engineering, " 1th ed. USA, Pennsylvania, pp. 67-69.

[2] Kurosawa Y., and Yamaguchi T., 2013, "Finite Element Analysis for Damped Vibration Properties of Panels Laminated Porous Media, " World Academy of Science, Engineering and Technology, Japan, Vol:7 2013-06-20, pp. 78.

[3]. Pockszevnicki C., Rodrigues E., Ferreira T.S., Barbosa R., Vieira A., Silveira M, 2011, "Finite element analysis considering material porosity, " SAE Technical Paper 2011-36-0136, DOI: 10.4271/2011-36-0136.

[4] Weiguo Z, Nickolas V and Kuangcheng W., 2005, "An energy finite element formulation for high-frequency vibration analysis of externally fluid-loaded cylindrical shells with periodic circumferential stiffeners subjected to axi-symmetric excitation, " Journal of Sound and Vibration, Vol: 282, Issues 3–5, 2005, Pages 679-700. DOI: 10.1016/j.jsv.2004.03.063

[5] Hörnlund M.; Papazoglu A., 2005, "Analysis and measurements of vehicle door structural dynamic response, " Master’s Dissertation, Lund University, Sweden.

[6] Moura F., Ferreira T.S., Danti M., Meneguzzo M., 2012, "Numerical and experimental comparison by NVH Finite Element Simulation in “Body in White” of a vehicle in the frequency range until 800Hz, " SAE Technical Paper 2012-36-0629, DOI:10.4271/2012-36-0629.

[7] LMS International. LMS PolyMAX, 2003, A Revolution in Modal Parameter Estimation. LMS International, Leuven, Belgium.

[8] Christopher J. Cameron, Per Wennhage, Peter Göransson, 2010, "Prediction of NVH behaviour of trimmed body components in the frequency, " Appl. Acoust., DOI: 10.1016/j.apacoust.2010.03.002.

[9] Performance Standard, 2004, Vehicle And Shell – Acoustic/vibration transfer functions analysis, Fiat Auto, N.7- R0151.

[10] Rao, Singiresu S., 2008, "Vibrações Mecânicas, " 1th ed., Brasil, São Paulo, p 82-85.

[11] Ferreira T.S., Moura F., Magalhões P., 2013, "Sensitivity analysis of numerical and experimental comparison by nvh finite element simulation in “trimmed body” to different excitation points of a vehicle in the frequency range until 500 hz. " 22nd International Congress of Mechanical Engineering (COBEM), ISSN 2176-5480.

[12] L. Komzsik. MSC.NASTRAN., 2001, Numerical User’s Guide. The MacNeal-Schwendler Corporation, Los Angeles, CA, USA.

[13] Gaurav Kumar, Stephen J. Walsh, Victor V. Krylov, Structural–acoustic behaviour of automotive-type panels with dome-shaped indentations. Applied Acoustics, 2013; 73: 897–908.

[14] John Laurence Davy , Timothy J. Phillips , John R. Pearse, The damping of gypsum plaster board wooden stud cavity walls. Appl. Acoust, 2014; 88: 52-56.

[15] Teik C. Lim, Automotive panel noise contribution modeling based on finite element and measured structural-acoustic spectra. Appl. Acoust, 2000; 60: 505-519.