NUMERICAL EVALUATION OF THE VIBROACOUSTIC BEHAVIOR OF A SIMPLIFIED AUTOMOTIVE CAVITY

Lucas Murelli Silveira, murelli@gmail.com
Thais Duarte Tavares, thais.dtavares@gmail.com
Alberto C. G. Diniz, adiniz@unb.br
Universidade de Brasília – Faculdade de Tecnologia – Depto de Eng. Mecânica (UnB/FT/ENM)
Campus Darcy Ribeiro – Asa Norte, 70.910-900, Brasília, Brazil

Marcus Vinicius Girão de Morais, mvmorais@unb.br
Universidade de Brasília – Faculdade do Gama (UnB/FGA) – Antigo Fórum do Gama – Setor Central, 72405-610 , Gama, Brazil

Abstract. *The automobile's emission of noise and vibration is an important industrial problem. NVH characteristics are responsible to car ride comfort and, at limit, several health issues. It is important to study the interaction between the car's structure and acoustic cavity, enabling the development of engineering tools. Therefore, automotive vibroacoustic control may be included since the design stage. The vibroacoustic behaviour of realistic automotive passenger cavities poses great challenges given by geometrical complexity and wide span of boundary conditions. In this context, the present work shows the design and characterization of a simplified vehicle cavity, allowing comprehension of vibroacoustic phenomena. FE numerical analysis was performed on computational platform's ANSYS (structural analysis) and LMS Virtual.Lab Sysnoise (acoustic and vibroacoustic analysis). Some analysis parameters, such as meshing, boundary conditions and response points, were tested in a preliminary geometry. Then, acoustic and vibroacoustic response analysis were compared in order to characterize the structure's influence.*

Keywords*: NVH, acoustic, vibration, modal analysis, numerical modelling*

1. INTRODUCTION

The automobiles produce noise and vibration because are constituted of several dynamic systems, which affect the ride comfort and also can cause health issues. Due to this, the reduction of the noise and vibration levels is an important problem to the industry to improve the quality of their product.

Nowadays, the industry can remedy the vibroacoustic problem with two different approaches: contemplating the vibroacoustic issue since the design stage of the vehicle or only after its conception by using palliative solutions. However, the first one requires advanced computational tools and the second, the introduction of more material inside the vehicle. Hence, the industry needs new engineering techniques to solve this problem without increasing the manufacturing costs.

Therefore, it is important study the interaction between the car's structure and the acoustic cavity and understanding the vibroacoustic phenomenon.

In this context, the present work shows the design and characterization of a simplified vehicle cavity, allowing comprehension of vibroacoustic phenomena. FE numerical analysis was performed on computational platform's ANSYS (structural analysis) and LMS Virtual.Lab Sysnoise (acoustic and vibroacoustic analysis). Some analysis parameters, such as meshing, boundary conditions and response points, were tested in a preliminary geometry. Then, acoustic and vibroacoustic response analysis were compared in order to characterize the structure's influence.

2. PRELIMINAR DESCRIPTION OF CONCRETE CAR MOCK-UP

There are in the literature two important studies of the vibroacoustc behavior of a simplified automobile design that were used as reference to develop the model of this study. The first one was developed by Nunes (2001) to perform numerical and experimental modal analyses with phono-absorbents materials. The second, known as the ConcretCar (Pluymers, 2006), is a real size vehicle workbench to investigate its acoustic behavior.

The model proposed by Nunes (2001) consists of a reduced size cavity with walls made of acrylic to permit better visualization of the microphones' position (Fig. 1). According to the author, the structure is not rigid enough for the problem, providing a vibroacoustic interaction from 35 Hz on, which reveals that this structure is not indicate to a model that intends to achieve a range of purely acoustic modes.

In the other hand, there is the Pluymers model (Fig. 2). In this one, the dimensions are similar to the ones of a station wagon vehicle and the walls are made of 100 mm of concrete in order to provide a rigid acoustic structure, then avoiding vibroacoustic interaction until 400 Hz. Nevertheless, the weight of the workbench is about 4000 kg, which poses great difficult on transporting it, and there is no possibility of testing other materials or elements as a solid contour/boundary due to the totally rigid concept.

The workbench for this study intends to achieve a range of purely acoustic modes. Then, the walls needed to be more rigid than the ones of the Nunes' model. However, it is also intended to have a lighter and more practical workbench than the Pluymers' one, enabling the tests of contour components. Therefore, the concept of the workbench for this study was a compromise between the models already presented. It was decided to use a modular real size model to enable the tests of cars' components, such as benches, doors and windshields and use walls made of a sandwich material of steel and wood.

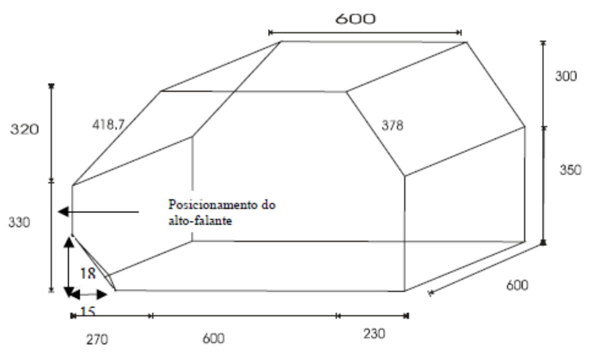


Figure 1. Dimensions of Nunes' model. [mm]

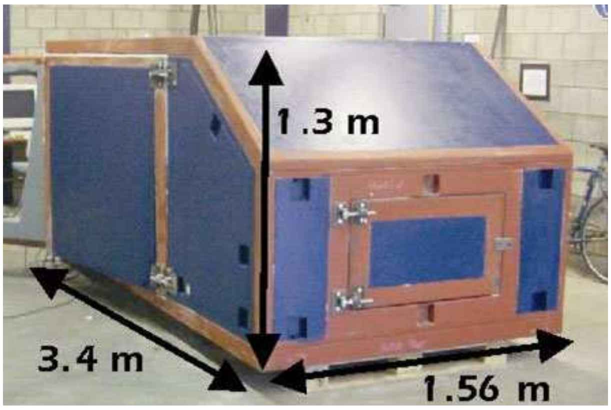


Figure 2. Internal dimensions of Pluymers' model. [m]

3. STRUCTRURAL-ACOUSTIC ANALYSIS

Problem solving structural-acoustic has always aroused the interest of several branches of engineering. Vibroacoustic (or structrual-acoustic) phenomena study the fluid-structure coupling between of acoustic cavities contained by flexible structures. The acoustic cavity is filled for a low density fluid, for example, the atmospheric air. Sometimes the vibration characteristics of the structure and acoustic cavity are considerably modified justifying a more detailed analysis.

For this, we define an elastic structure, domain Ω_S, filled by a no-viscous compressible fluid, domain Ω_F. In this case, the gravity effects are neglected. The structure's boundary condition is restricted in movement at contour Γ_u, and subject to external forces $\mathbf{F}_s$ at contour Γ_t. The fluid-structure interface is represented as Σ, and $\mathbf{n}^S$, $\mathbf{n}^F$ is respectively outside normal of structure and fluid.

3.1 Structural-Acoustic Model

As a starting point for the structrural-acoustic development, we take the equations of motion of the coupled system in the form given Everstine (1981), Zienkiewicz & Newton (1969) [conf. Morais (2000)].

$$\begin{bmatrix} \mathbf{M}_{ss} & \mathbf{0} \\ \mathbf{M}_{fs} & \mathbf{M}_{ff} \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{u}} \\ \ddot{\mathbf{p}} \end{Bmatrix} + \begin{bmatrix} \mathbf{K}_{ss} & \mathbf{K}_{sf} \\ \mathbf{0} & \mathbf{K}_{ff} \end{bmatrix} \begin{Bmatrix} \mathbf{u} \\ \mathbf{p} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F}_s \\ \mathbf{F}_p \end{Bmatrix} \tag{1}$$

where, **u** e **p** are the vector of *n* structural displacement and the vector of *m* nodal pressure for the enclosed fluid, respectively; $\mathbf{M}_{ss}$ and $\mathbf{K}_{ss}$ are the *n x n* structural mass and stiffness matrices; $\mathbf{M}_{ff}$ and $\mathbf{K}_{ff}$ are the *m x m* acoustic mass and stiffness matrices; $\mathbf{K}_{sf} = \mathbf{A}$ and $\mathbf{M}_{fs} = -(\rho c)^2\,\mathbf{A}$ with A is a sparse *n x m* coupling matrix. The determination of coupling elements (a_{ij}) is carried out with aid of surface area for the fluid-structure boundary corresponding to the acoustic pressure p_j and the associated structural displacement u_i. The term (ρc) is the characteristic impedance of the fluid. The terms $\mathbf{F}_s$ and $\mathbf{F}_p$ are the vectors of external forces applied to the structure and the acoustic domain.

The classical boundary conditions are the follow:

- acoustic pressure prescription, $p = p_o$, in Γ_p ;
- structural displacement prescription, $\mathbf{u} = \mathbf{u_o}$, in Γ_u ;
- structural external force prescription, $\mathbf{F}_s = \mathbf{F_o}$, in Γ_t ;
- fluid-structure interface, $\boldsymbol{\sigma}_s \cdot \mathbf{n} = p\mathbf{n}$, in Σ ,

For a simple monopole the pressure at a distance r is,

$$p(r,t) = j\rho_o c \frac{Qk}{4\pi r} e^{j(\omega t - kr)} \tag{2}$$

where, p(r,t) [Pa] is the acoustic pressure at a distance from source center r [m] and time t[s], ω is the frequency [rad/s], k [m^{-1}] is the wave-number, Q [m^3s^{-1}] is the source strength.

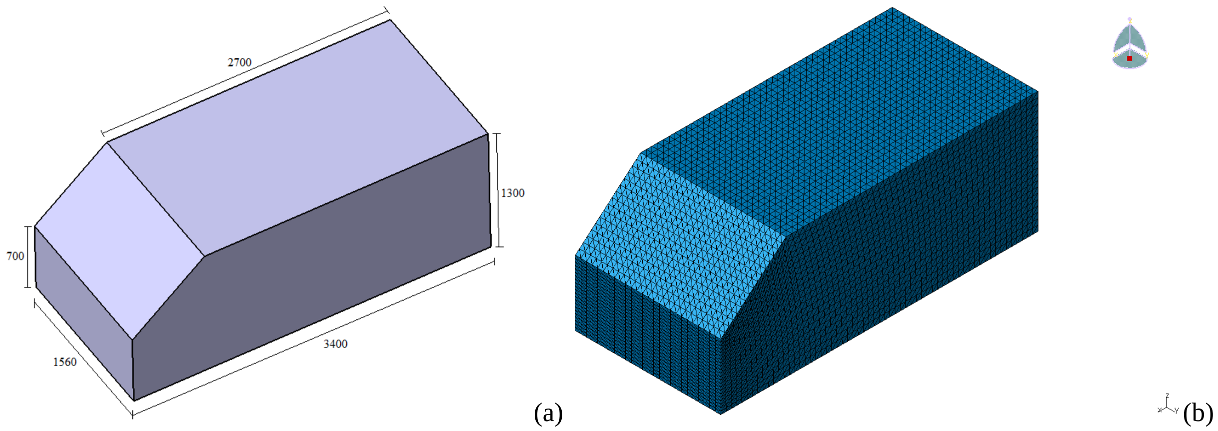
For the time-harmonic solution, fluid structure variable can be described as separated functions, i.e., acoustical pressure $p = p_o \exp(j\omega t)$, structural displacement $\mathbf{u} = \mathbf{u}_o \exp(j\omega t)$, and $\mathbf{F_i} = \mathbf{F_{i,o}} \exp(j\omega t)$ (i = s,f). In modal analysis, we consider structural and/or acoustical excitation null, $\mathbf{F_i} = 0$,

$$\left(\begin{bmatrix} \mathbf{K}_{ss} & \mathbf{K}_{sf} \\ \mathbf{0} & \mathbf{K}_{ff} \end{bmatrix} - \omega^2 \begin{bmatrix} \mathbf{M}_{ss} & \mathbf{0} \\ \mathbf{M}_{fs} & \mathbf{M}_{ff} \end{bmatrix}\right) \begin{Bmatrix} \mathbf{u}_o \\ \mathbf{p}_o \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \tag{3}$$

By the modal analysis, we can describe the present acoustic-structural model. The inference about uncoupled modes or a coupled mode can be possible.

4. DESCRIPTION OF CARACOUS FE-MODEL AND PRELIMINAR OBSERVATIONS

A CAD model (CATIA V5) was designed according to Plumeyr's cavity, Figure 2. Initially, we generated a mesh with 60159 linear tetrahedral elements and 13818 nodes, Figure 3. This geometrical model is exported to LMS Virtual.Lab to determine an acoustic modal shapes and natural frequencies by finite element method. The characteristic of acoustic cavity has sound velocity c = 340 m/s, $\rho = 1225\ g/cm^3$. And the numerical algorithm to modal extraction is Block Lanczos algorithm.



(a) (b)

Figure 3. (a) Modelo CAD da cavidade acústica. Dimensões em mm
(b) 3D representation of quadratic tetrahedral FE-mesh

The analysis of mode shapes show unexpected asymmetries, raising a suspicion of spurious modes.

5. SOLUTION OF RECTANGULAR RIGIDITY CAVITY

The starting point was to investigate the origin of such asymmetries through analysis of analytical models (Kinsler *et al.*,2000), for example rectangular cavity. The analytical solution of rectangular cavity (Blevins, 1979; Kinsler *et al.*, 2000, Pedroso, 2001) is described below,

$$\left(\frac{\omega_{ijk}}{c}\right)^2 = \left(\frac{i\pi}{L_x}\right)^2 + \left(\frac{j\pi}{L_y}\right)^2 + \left(\frac{k\pi}{L_z}\right)^2, \qquad i, j, k = 1,2,3,\ldots \tag{4}$$

and,

$$p_o(x,y,z) = \sum_{k=1}^{\infty}\sum_{j=1}^{\infty}\sum_{i=1}^{\infty} P_{ijk,o} \cos\left(\frac{i\pi}{L_x}\right)\cos\left(\frac{j\pi}{L_y}\right)\cos\left(\frac{k\pi}{L_z}\right) \tag{5}$$

Initially we suspected that the source of asymmetries was geometrical symmetry of the cavity. For that we attempted to repeat the phenomenon of asymmetry now with a rectangular cavity with sides equal to 1m. The results of modal shape found did not correspond to the analytical solution, as observed in the sixth mode (Figure 4a).

Then the cube's dimensions were changed to 0.99 m *X* 1.01 m *X* 1 m in order to generate a slight asymmetry in geometry. Thus, the numerical method would not find the uniqueness of the occurrence of multiple frequencies. However, results remained disconnected from the analytical methods.

So we decided to modify the type of element for linear hexahedra to eight nodes, improving order interpolation of the sound pressure (Nunes, 2001). After these modifications the acoustic modes observed (Figure 4b) were in full agreement with the analytical methods. The problem of asymmetry was a numerical artefact due to the element type used.

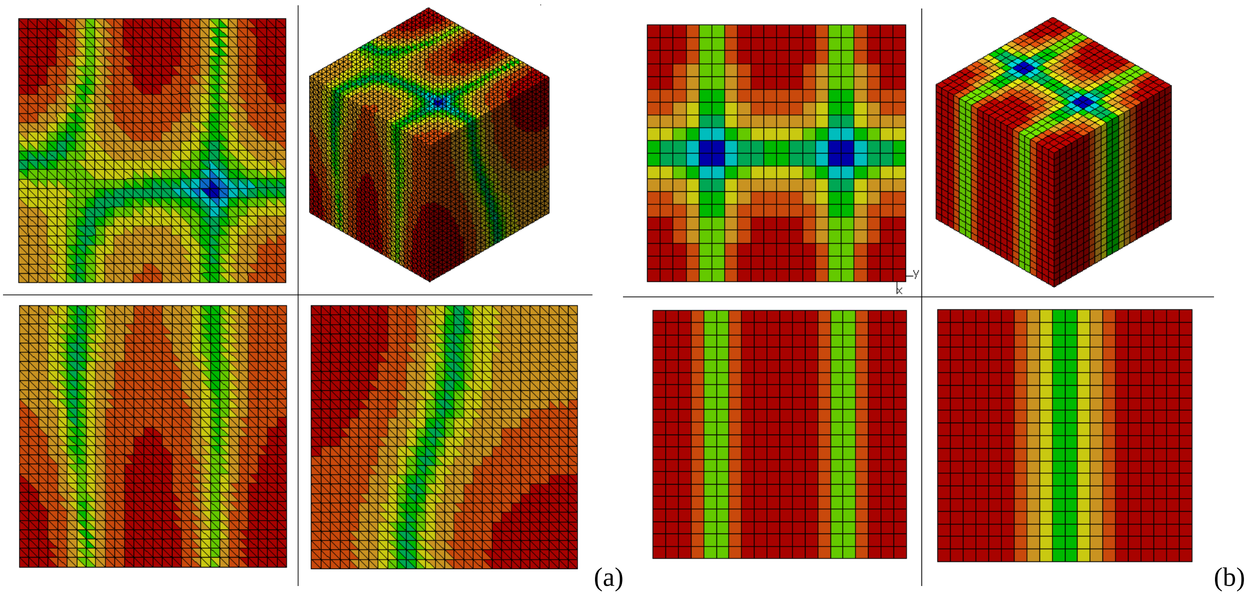


Figure 4. Isovalues of 6th acoustical pressure mode to a unitary cubic cavity ($L_x = L_y = L_z = 1$m): **(a)** tetrahedrical element $f_6 = 388$ Hz and **(b)** hexahedral element $f_6 = 378$ Hz. Representation of superior view, isometric, frontal view and left view.

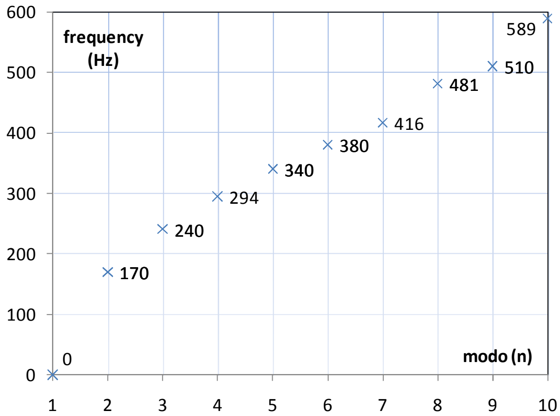


Figure 5. Analytical frequencies of a unitary cubic cavity ($L_x = L_y = L_z = 1$m) as described equation (5).

6. CORRECTED MODAL ANALYSIS

Now, the cavity was modeled with linear hexahedral elements to eight nodes, in a mesh of 17136 elements and 19404 nodes, Figure 6.

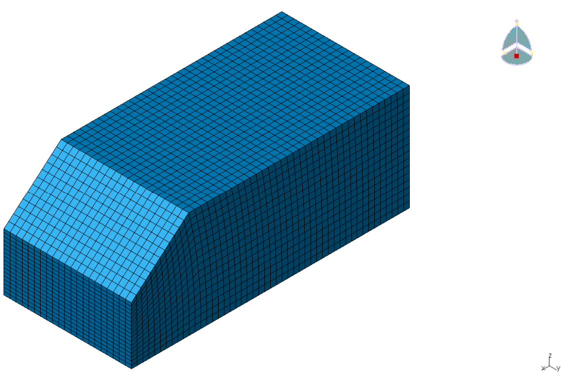
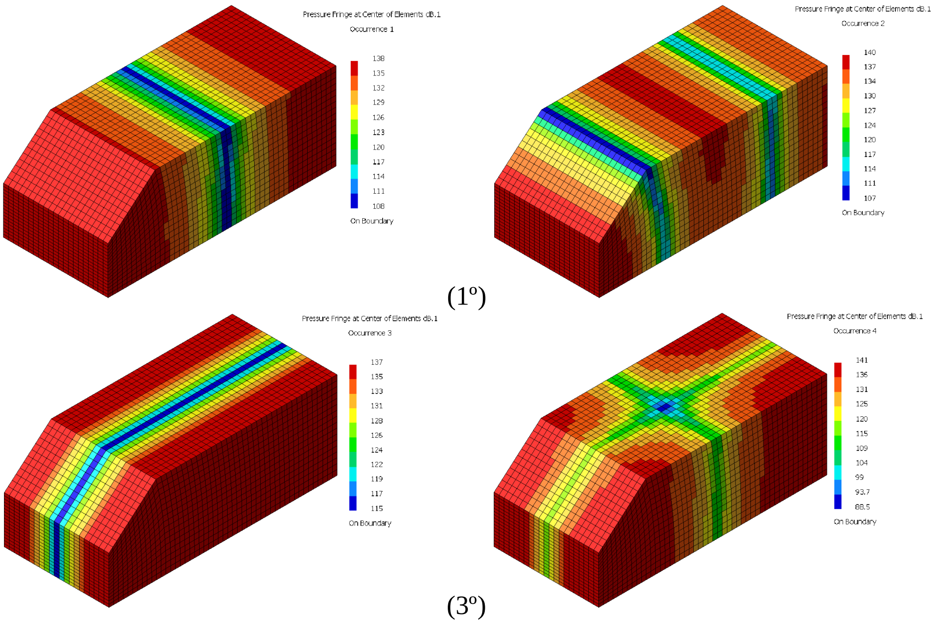


Figure 6. 3D representation of new quadratic hexahedral FE-mesh

The modal analysis the cavity use the numerical solution by Block Lanczos algorithm. Within the frequency range analyzed [0-500 Hz], there was no difference between results calculated using this algorithm or other methods -

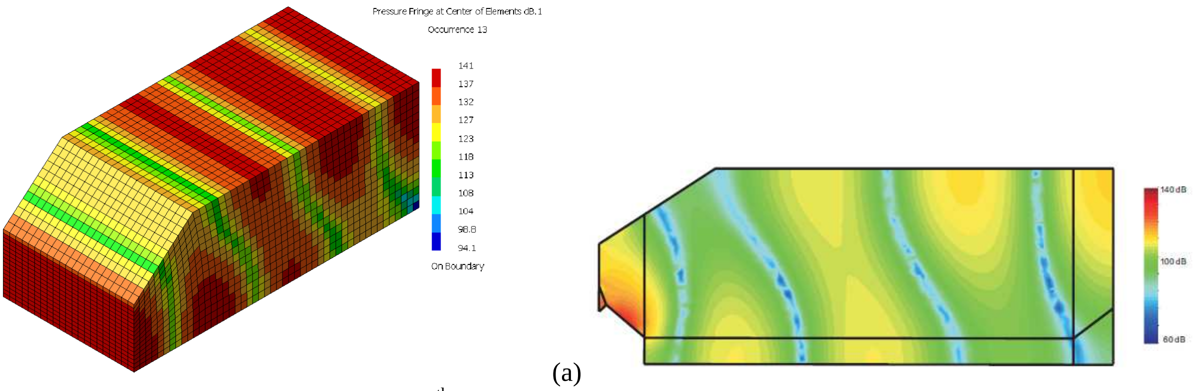
subspace, Block Arnoldi, Lanczos and Arnoldi - unless the increase in computational cost. This fact justifies the choice of Block Lanczos. The modal frequencies obtained are shown in Table 1 and the first four mode shapes are shown in Figure 7.



(1º) (2º)
(3º) (4º)

Figure 7. Isovalue representation of the first four acoustic modes

Below, we compare the first modal shape symmetric with respect to literature, Figure 8.



(a) (b)

Figure 8. Isovalue representation of 13th acoustic mode: (a) present work at 199Hz, (b) Desmet work at 198Hz

Table 1. Numerical frequencies for first fifteen acoustic modes of CarAcous cavity

Mode	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Frequency (Hz)	52	102	109	121	132	143	149	155	172	178	180	189	199	209	217

7. EXCITATION SOURCE AND FREQUENCY RESPONSE ANALYSIS

To characterize the acoustic cavity and allow comparison with experimental data is necessary to analyze the frequency response in some strategic points inside the cavity. It is then necessary to characterize the sound source, speaker on the form of monopole, and to determine the position of response points important of the noise problem in automotive passenger cavity.

In LMS Virtual.Lab Sysnoise, the field of acoustic response is obtained by modal syntheses. This is done from the calculation of previously computed modal shapes excited by sound source, treated as a boundary condition in finite element method. Thus, we determined the modal participation factors in the response field for each frequency.

7.1. Acoustic source: Speaker

Order to select the response position more representative of engine excitement, we perform several numerical simulations with a speaker in different locations. For this, new models of acoustic cavity were created in CATIA V5, considering a recess due to the presence the the speaker box, Figure 9. In all configurations tested, the box speaker is in contact with the floor in front of the cavity.

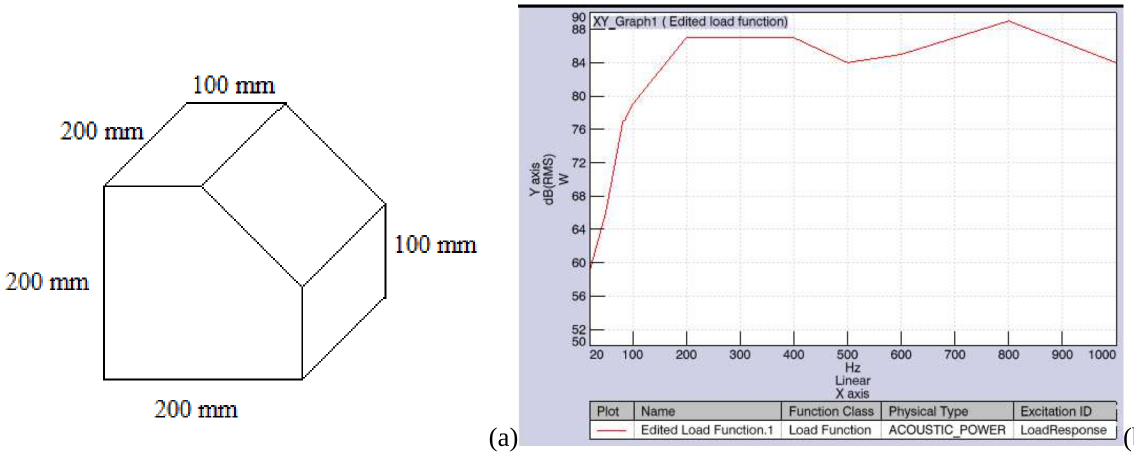


Figure 9. Schematic representation of loudspeaker system (a) and excitation function of Fostex speaker (b).

The presence of the speaker box was first studied by modal analysis, followed by a harmonic analysis of acoustic response of cavity. For these calculations was considered structural damping of 1%. The sound source used to model the speaker was a monopole, by their characteristic spherical wave propagation. The amplitude the sound power of the speaker FE103En Fostex 4 "Full Range, 20 Hz to 1 kHz, is show in Figure 9.

The speaker box was placed in contact with the side of the cavity and directed to the opposite side. Then, facing forward was positioned away with zero, one fourth and one-half of the cavity width from lateral side. The optimal position of the speaker box is at a one-half the cavity width. The Figure 10 shows the responses obtained of the harmonic analysis and modal analysis for the frequency of 178 Hz.

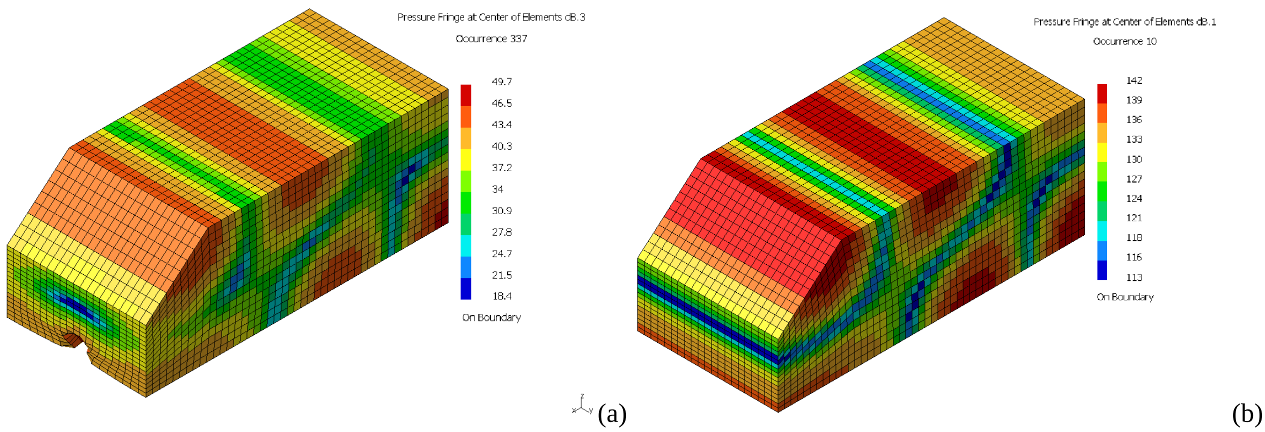


Figure 10. Isovalue of acoustic pressure field in CarAcous cavity due to a harmonic monopole source at 178Hz (a), and 10th acoustic shape mode (b). Position of loudspeaker – at middle frontal face turn to interior CarAcous cavity.

It analyzes the responses from of the box loudspeaker positioned at half-width. This configuration results in a mesh of 16065 elements EF and 18216 nodes. This mesh allows better characterization of the longitudinal modes and effects of symmetry.

7.2. Response point inside CarAcous cavity

To obtain the frequency response functions must be implemented response points within the cavity. In LMS Virtual.Lab this is done by creating a mesh field. The spherical grid was chosen due to its isotropy in capturing the acoustic pressure, and allows a measurement of average values in a finite space, such as the human ear or a microphone.

To evaluate FRF sensitivity due to the sphere radius, we select a point locate at the driver head, 1200 X 380 X 1000 mm from the bottom right corner of the cavity front face, to be the sphere center. Rays of 2, 20, 40, 60 and 100 mm were tested. We decided to select the sphere with 20mm of radius for present a well-defined FRF, and be close to the microphone size being used.

So, to better characterize, eight points were distributed in the cavity. Their placement simulates what would be the ears of four occupants, two in front and two in the rear of the vehicle. The approach is similar to that used for Pluymers (2006), for experimental validation of their numerical model using the Concrete Car, Figure 11.

The distance between the points of front line is as follow: 1-2 and 3-4 is 267 mm, and 2-3 is 266 mm. The distance between the points of back line follows the same distribution. The back line and front line are distance of 1000 mm. Thus, eight FRFs were obtained for comparison with the case vibroacoustic.

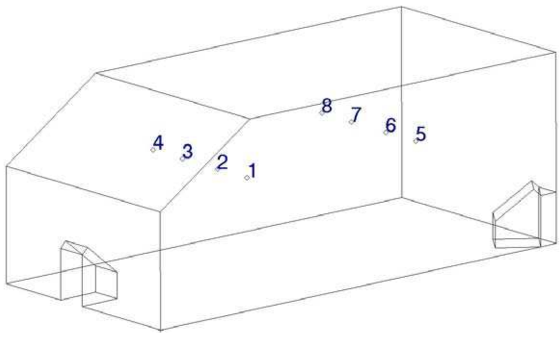


Figure 11. Position of response points (Pluymers, 2006).

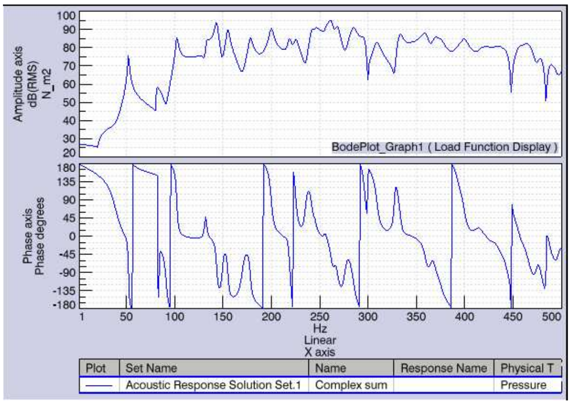


Figure 12. Function de response in frequency of acoustic pressure (FRF-p) at point 7.

8. MODAL ANALYSIS OF STRUCTURE

We perform a modal analysis of the structure. From this result we get the frequency from which the coupling occurs in fluid-structure vibration modes.

The numerical simulations in finite element package ANSYS 12.0, for its robustness and direct interaction with the LMS Virtual.Lab platform. All stages of numerical analysis were performed in ANSYS structural, from modeling the geometry of the extraction methods.

We chose models of plates and beams to model sandwich plates fixed on a metal frame with structural profiles. This approach allows good quality results, ease in modeling plates and reducing the computational cost, it results in a mesh with fewer elements and nodes in comparison to the use of solid elements.

The beam element used was BEAM188. This is a three-dimensional beam element with two nodes with six degrees of freedom at each node. He was chosen to take into account the effects of shear, according to the theory of Timoshenko beam (ANSYS, 2009). These effects are dominant due to the presence of torsion elements in the structure. Moreover, the theory of Timoshenko shows smaller deviations from the modal frequency compared to experimental results (Han et al., 1999), especially for little slender beams. According to Han et al. (1999), the beam models can be compared according to the effects included in the calculation, Table 2.

The plate element selected was SHELL181. This is a three-dimensional plate element to four nodes with six degrees of freedom in each. It is used by enabling the modeling of sandwich beams through the definition section and fit well to linear problems with low to moderately thick plates.

For the beams was considered a section of square tube. For the plates section was created with the superposition of three layers, the two sides of the same metal plates and the core of a material lighter and less rigid. The core function is to simply remove the sides, thus increasing the moment of inertia of the cross section of the plate.

The MDF plates chosen were in steel and wood agglomerate. The steel was used because of its high rigidity and high availability of structural profiles. The MDF plates were chosen because it is lighter than steel, have excellent availability and low cost.

As the results presented by Eleotério (2000), plates of wood agglomerate MDF exhibit elastic moduli very depending on density and resin content. Thus, average properties were chosen as representative of the material.

The adopted properties are presented in Table 2.

Table 2. Mechanical proprieties of materials in CarAcous cavity.

Material	Módulo de elasticidade (GPa)	Coeficiente de Poisson	Densidade (kg/m^3)
Aço	205	0,30	7830
MDF	2,2	0,25	720

The structural mesh used describes beams as lines and plates as surfaces. The complete mesh has 26556 elements and 25359 nodes. At first, the structure was idealized as beams with square section 50mm X 50mm and 5mm of thickness. The sandwich plates in steel and wood agglomerate has 5mm and 35mm, respectively. But, the first structural frequency found are inferior to 200Hz, a minimum established to obtain many pure acoustic modes (thirteen modes not coupled). The new dimensions of beams and plates are increasing progressively until attain a criterion minimum in frequency. Then, the final configuration has 200mm of width and 6.4mm of thickness. And the sandwich plates are composed by steel and wood aggregate plates with 12.7mm and 160mm of thickness, respectively.

For this final configuration, we obtain the sequence of natural frequencies, Table 3, and shape modal, Figure 13.

Table 3. Numerical frequencies for first fifteen structural modes of CarAcous cavity.

Mode	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Frequency (Hz)	207	218	241	274	274	282	286	293	310	320	323	330	355	373	376

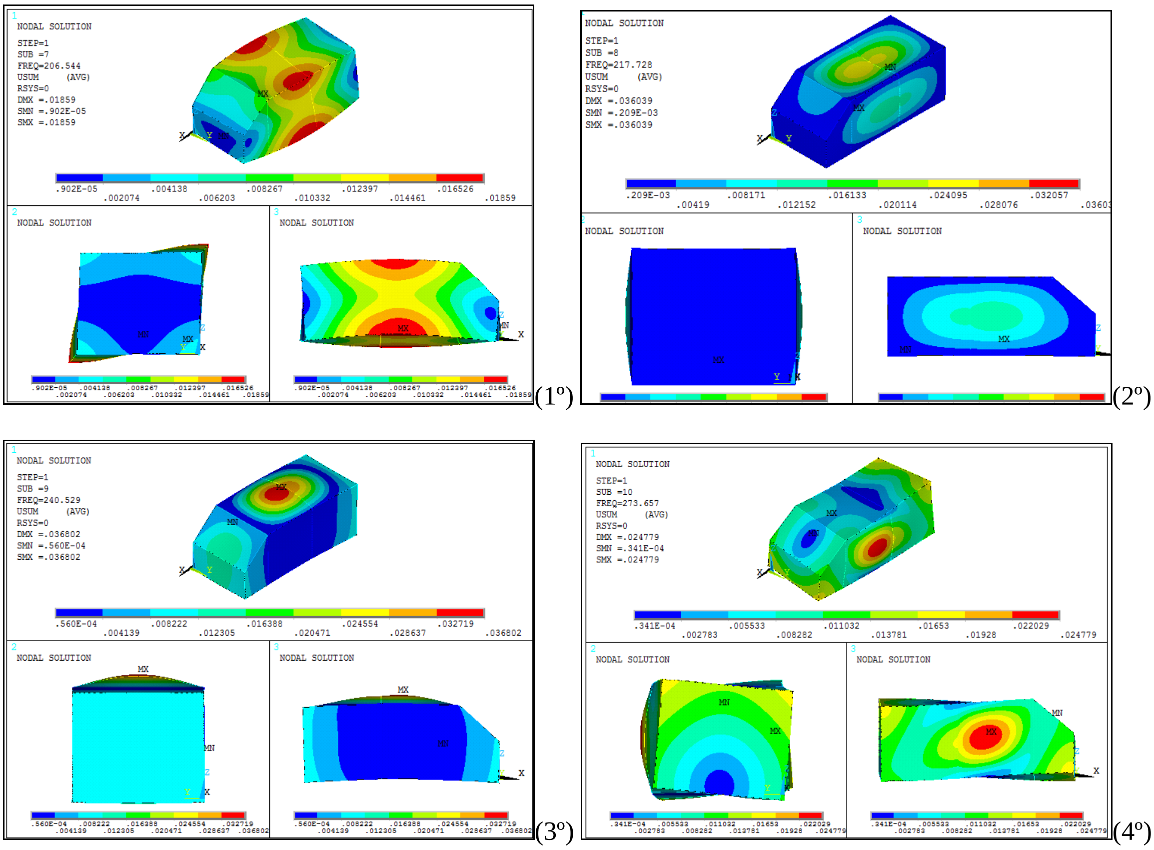


Figure 13. Isovalue representation of the first four structural modes

In general, it is observed that the modes are divided into two main groups: flexion-torsion of the structure and modes of the plate. The proximity between the stiffness of beam and plate partially explains this behavior.

9. MODAL ANALYSIS OF VIBROACOUSTIC MODEL

The vibroacoustic analysis is needed to evaluate the influence of structural vibration on the acoustic response of the cavity. This investigation is important to determine the limits of the present mock-up. The solution of vibroacoustic problem takes place by simultaneous calculation of acoustic pressure fields and displacement contour. Therefore, it is necessary to carry out some computational tool that enables coupling and its numerical solution.

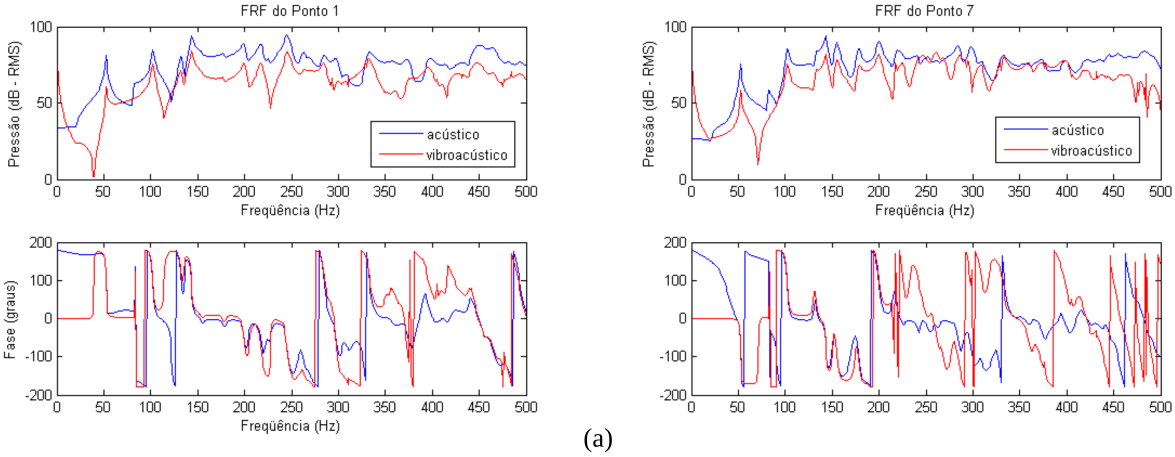
In the case of LMS Vitual.Lab Sysnoise the vibroacoustic response is obtained by the modal composition of the simultaneous structural and acoustic modes. Therefore, it is the transfer of the results obtained from ANSYS structural mesh to a mesh surface, consistent with the acoustic mesh by mapping node to node between them. This is necessary because the largest refining the structural mesh, compared to the acoustic surface of the mesh that has only 9132 elements and 5002 nodes, resulting in a discrepancy in spatial resolution and subsequent increase in the size of the problem (excessive number of nodes and elements).

As with the acoustic frequency response, is used a speaker box modeled as a monopole using the same previous excitation function, Figure 9.

Thus, it is possible to calculate the modal participation factors of the structure and couple them via a mapping between the mesh and the acoustic structure, using the results transferred of ANSYS. With this analysis we get the vibroacoustic response FRFs at response points defined in section 7.2. These results allow the comparison between pure acoustic and vibroacoustic solution in 1 and 7, Figure 14.

We observed a reasonable correlation between the results up to 250 Hz, which is corroborated by the comparison between the fields of purely acoustic and vibroacoustic response, Figure 15. However, the results lose their similarity above this threshold, an effect of greater influence of structural vibration, which can be seen in Figure 16 and Figure 17.

Thus, below 200 Hz allows the bench purely acoustic experiments, between 200 Hz and 250 Hz there is a considerable vibroacoustic coupling and above 250 Hz is purely vibroacoustic behavior.



(a) (b)

Figure 14 - Comparação das FRFs acústica e vibroacústica no ponto 1 (a) e ponto 7 (b) .

10. CONCLUSION AND PERSPECTIVES

The present work initiates a numerical characterization of simplified vehicle cavity, CarAcous cavity. This mock-up has a pretension to improve the comprehension of vibroacoustic phenomena in a simplified cavity. We use the platforms ANSYS (structural analysis) and LMS Virtual.Lab Sysnoise (acoustic and vibroacoustic analysis) to carry out a FE numerical model.

First of all, hexahedral finite element presents as better choice to discretize the acoustic domain. Hexahedral element could describe correctly analytical solution of a rigid rectangular cavity, by report to tetrahedral element. As consequence, numerical results of pure acoustic model presents a reasonable convergence as compare to numerical results in literature (Pluymers,2006).

Some analysis parameters, such as meshing, boundary conditions and response points, were tested in this preliminary geometry. Finally, we compare the purely acoustic and vibroacoustic response to study the influence of structural vibration.

We intend to perform experimental essays in a real mock-up. Others conceptions need to be tested to optimise the structure of CarAcoust cavity. But, a scaled model could be built before all. With this cheap mock-up, we could quickly compare experimental and numerical results and test our hypotheses about vibroacoustic behaviour of this cavity.

11. ACKNOWLEDGEMENTS

The authors would like to acknowledge the financial support of FAP-DF, referent to CarAcous projet "Vibroacoustic of automobile cavity – Experimental validation of a fluid-structure numerical model" nº 202/2008 (Processo nº 193.000.203/2008 from Edital PPP nº 02/2008). We are also very grateful to FINATEC, project "Vibroacoustic of Automobile Cavities – Numerical and Experimental Analysis" (Processo: 5641) Edital nº4/2009 "Auxílio Enxoval", for the financial support and partnership. Finaly we are reconnaissance by support of colleagues Profs Alessandro Borges (FGA/UnB faculty director), Profs. Fernando Neves, Mario Olavo and Alberto Diniz (lab GDS/FT/UnB).

12. REFERENCES

Blevins, R. D. (1979). Formulas for Natural Frequency and mode shapes. Ed. V.N.R. Ed.2, New York.

Everstine, G.C., 1981. "A symmetric potential formulation for fluid-structure interaction". Journal Sound and Vibration, Vol. 79, No. 1, pp. 157-160.

Morais, M.V.G., 2000. "Modelagens numéricas pelo método dos elementos finitos em problemas de interação fluido-estrutura". Master dissertation, Programa de Pós-graduação em Estruturas e Construção Civil (PECC/EnC/FT/UnB), Universidade de Brasilia, Brasilia, 157p.

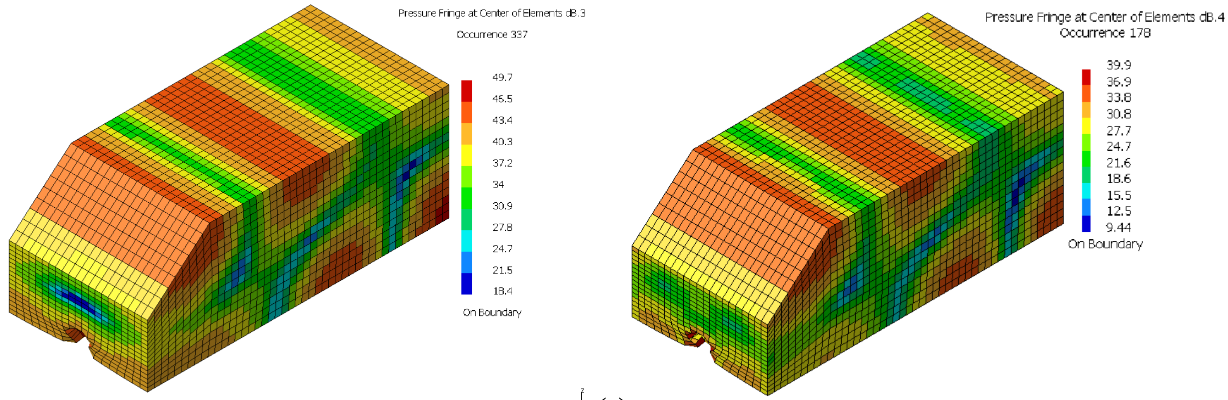
Nunes, A., 2001. "Análise Modal Teórica e Experimental Acústica de Cavidades com Absorção Sonora". Master dissertation, Programa de Pós Graduação da Faculdade de Engenharia Mecânica, Universidade de Campinas, SP.

Pedroso, L.J. (2001). Interação Fluido-Estrutura. Notas de aula do curso e apostila didática, UnB-FT/ENC, Brasília,DF.

Pluymers, B. (2006) "Wave based modelling methods for steady-state vibro-acoustics". Tese de doutorado. Katholieke Universiteit Leuven, Leuven, Bélgica, 400pp.

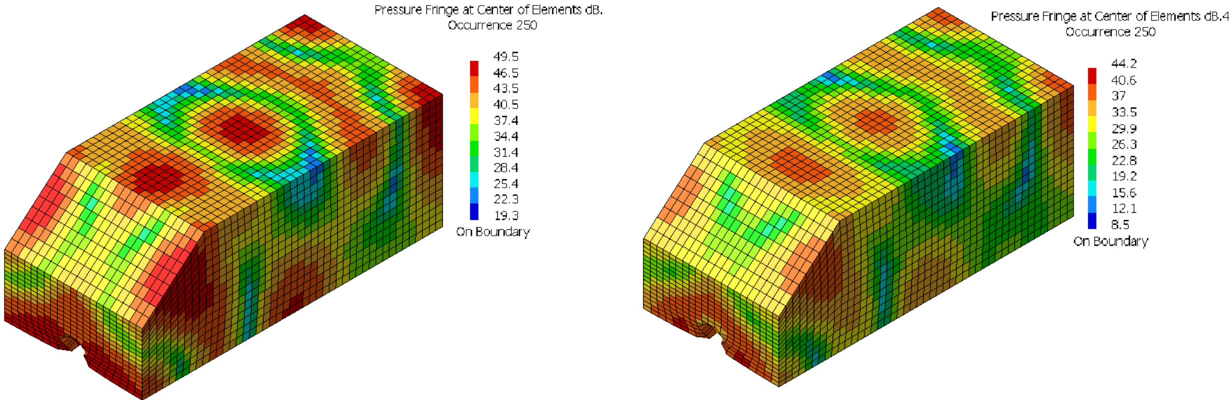
Kinsler, L.E. et al., 2000. "Fundamentals of Acoustic". John Wiley & Sons, 4ed., New York.

Zienkiewicz, O.C. and Newton, R.E., 1969. “Coupled vibrations of a structure submerged in a compressible fluid”. Symposium on Finite Element Techniques, Stuttgart.



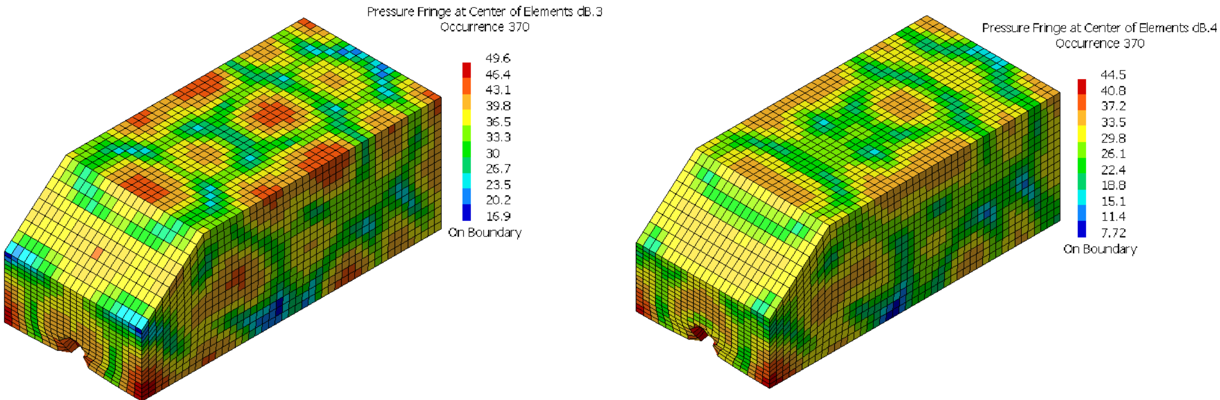
(a) (b)

Figure 15 – Isovalue acoustic pressure field of acoustic response (a) and vibroacoustic response for 178 Hz.



(a) (b)

Figure 16 Isovalue acoustic pressure field of acoustic response (a) and vibroacoustic response for 250 Hz.



(a) (b)

Figure 17 - Isovalue acoustic pressure field of acoustic response (a) and vibroacoustic response for 370 Hz.

10. RESPONSIBILITY NOTICE

The following text, properly adapted to the number of authors, must be included in the last section of the paper: The author(s) is (are) the only responsible for the printed material included in this paper.