# Mid Frequency Vibroacoustic Modeling of an Innovative Lightweight Cab - Floor/Cavity Interaction

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Vibrations, Shocks and Noise

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**Highlights**

- Numerical vibroacoustic modeling, mid frequency domain, non-coincident mesh and projection method, application to an innovative lightweight cab.

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## 1. Introduction

The City Lightweight and Innovative Cab (CLIC) project is a scientific collaboration gathering different public and private organizations. The aim is to propose an innovative truck cab responding to new European legislations in security and CO2 emission. A very high strength steel will be used to lighten the structure. This could affects directly the acoustic environment of the cab. In order to control the noise requirements at the design stage, it is then necessary to be able to simulate the vibroacoustic behavior of a truck cab in the mid frequency range. In this context, a methodology is developed at INSA Lyon based on Statistical modal Energy distribution Analysis (SmEdA) [1, 2]. This method is considered as substructuring approach for vibroacoustic problems and post-process for finite element methods (FEM). It is based on the knowledge of the uncoupled subsystem modes that can be computed using the FEM [1, 2]. In this paper, one focuses on the fluid-structure coupling of the cab floor with the interior cavity when they are represented by non-coincident meshes.

## 2. Vibroacoustic modeling with non-coincident finite element mesh

FEM method remains the most common modeling method in industrial applications despite the huge number of degree of freedom (dof) (for information purposes, the structural FEM model contains 6Mdof). Moreover, in the mid frequency range, the vibroacoustic behavior of the global system is not so easy to analyze with the FEM results and SmEdA concept allows substructuring the global problem. To study the interaction between the floor and the interior cavity, one can perform modal FEM calculations on two independent uncoupled subsystems: the floor and the cavity. The energy equation of motions of the coupled subsystems can then be written and easily resolved. In these equations, the spatial couplings of the subsystem modes are expressed through the intermodal works which is deduced from the mode shapes. When dealing with non-coincident meshes of the two subsystems, the intermodal

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works cannot be calculated directly and an interpolation process is required as described below.

Interpolating the acoustic entities on the structural nodes or vice versa ensure the continuity of the problem. One presents here a projection method based on the barycentric coordinate system:

Let *A*, *B*, *C* and *G* be the geometrical points associated to four nodes in the coordinate system *R(O,x,y,z)*. *G* is the barycenter of the triangle *ABC* if and only if one attributes real weights *a*, *b* and *c* to the nodes *A*, *B* and *C* respectively, verifying the equation:

$$a\mathbf{GA} + b\mathbf{GB} + c\mathbf{GC} = \mathbf{0}$$

This equation verifies the eigenvalue problem associated to the matrix system below:

$$\begin{bmatrix} x_A - x_G & x_B - x_G & x_C - x_G \\ y_A - y_G & y_B - y_G & y_C - y_G \\ z_A - z_G & z_B - z_G & z_C - z_G \end{bmatrix} \begin{Bmatrix} a \\ b \\ c \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix}$$

*a*, *b* and *c* should be positive and normalized so that their sum is equal to 1:

$$a + b + c = 1 \qquad (a,b,c\,) \geq 0$$

Figure 1 illustrates the problem where the green mesh presents the fluid elements and the red mesh the structural elements. For each node *Gi* (of the red structural mesh) one aims to find the weights associated to *A*, *B* and *C* (of the green fluid mesh).

Finally the acoustic mode shapes on the node $G_i$ is obtained with:

$$\boldsymbol{\Psi}_{Gi} = a\boldsymbol{\Psi}_{A} + b\boldsymbol{\Psi}_{B} + c\boldsymbol{\Psi}_{C}$$

The example chosen to illustrate the method is the floor of the cab coupled to the cavity. Figure (2) shows the shape of the first acoustic pressure mode of the cavity on the floor interface and figure (3) shows the projection of this mode on the floor nodes.

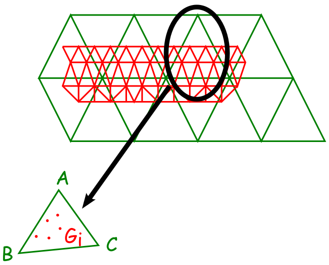


Fig. 1. Non-coincident mesh

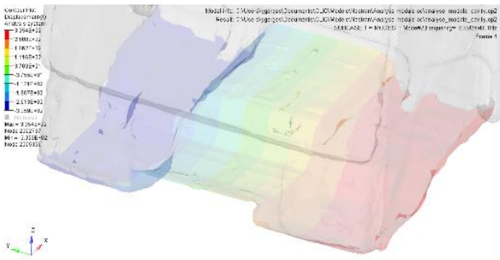

Fig. 2. Acoustic mode shape at the floor interfaces

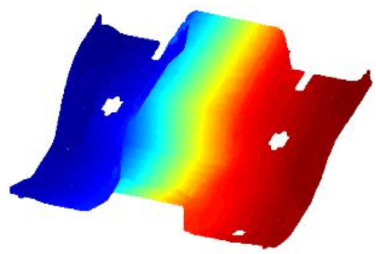

Fig. 3. Acoustic mode shape projected on structural nodes

This result shows the efficiency of the proposed method and allows extending this work to the vibroacoustic interaction.

## 3. Vibroacoustic interaction in mid frequency

SmEdA is based on the linear equation of power exchanged between coupled subsystems. Energy equipartition among subsystems modes is not assumed and one deals with energy levels of individual modes of subsystems in a frequency range [1, 2]. The modal coupling loss factor between the mode *p* of the floor and the mode *q* of the cavity is given by:

$$\beta_{pq} = \beta_{W*}\,\beta_{\omega}$$

$\beta_W$ is the modal spatial coupling factor and $\beta_\omega$ is the modal spectral coupling factor.

Figure (4) shows the spatial and spectral modal coupling factor in a frequency band, figure (5) shows the modal coupling factor and the floor/cavity shapes of the strongest coupled modes. As we can observe, this couple of modes is strongly coupled due to frequency coincidence.

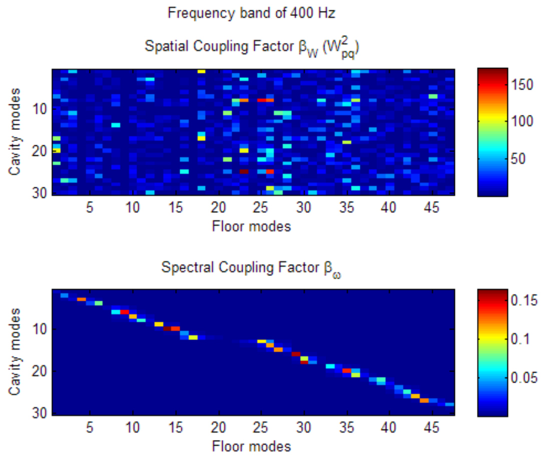


Fig. 4. Spatial (top) and spectral (bottom) coupling factors in a frequency range

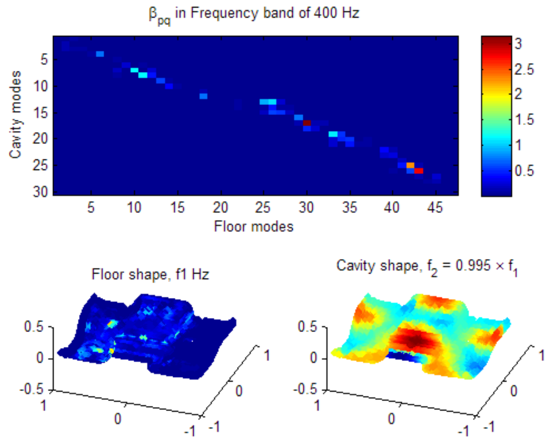


Fig. 5. Modal coupling loss factor and mode shapes of the strongest coupled modes

## 4. Conclusion

This work presents a vibroacoustic study of a truck cab in the mid frequency range. A projection method in non-coincident mesh case is illustrated and SmEdA method is used for the vibroacoustic study. The next step will be to introduce damping materials to improve the vibroacoustic comfort [3].

## Acknowledgements

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