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VIBROACOUSTIC MODELING OF A TRIMMED TRUCK CAB IN THE MID FREQUENCY RANGE

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The City Lightweight and Innovative Cab (CLIC) project is a scientific collaboration gathering public and private organizations. The aim is to propose an innovative lighten truck cab, where a high strength steel is used. As long as it could affect directly the acoustic environment of the cab, it is necessary to be able to simulate the vibroacoustic behavior of a truck cab in the mid frequency range. SmEdA (Statistical modal Energy distribution Analysis) method allows us substructuring the global problem, to study the interaction between the floor and the interior cavity. The process consists in building finite element models (FEM) of each subsystem (floor, internal cavity), including the dissipative material (damping layer, poroelastic material). Standard modal FEM calculations are then performed for each uncoupled subsystem. From the spatial mode shapes, and the modal strain and kinetic energies, the modal loss factors of both subsystems are estimated. Finally, the pressure levels inside the cavity are deduced from the resolution of the SmEdA equations. This process allows us taking into account the dissipative energy effects. To validate it, the cab is excited at the truck frame links, and the pressure levels on the driver's ears are compared between numerical and experimental results.

1 INTRODUCTION

With the new security and pollution standards, the industrials have to make more effort on their engines design. The CLIC (City Lightweight and Innovative Cab) project, that is gathering different private and public partners, aims to suggest a new design for lighten cabin truck, where a high strength steel is used. This lightening process decreases the NVH performances, and the mid frequency vibroacoustic behavior should be taken into account to ensure the driver comfort. The finite element method is the most common modeling method, despite the drawbacks

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in time consuming and computer memory requirements. The SEA method (Statistical Energy Analysis) [1] is well used for the high frequency studies, but it requires the validation of many hypotheses. The SmEdA method (Statistical modal Energy distribution Analysis) [2, 3] is a well compromise between those two methods. It is considered as a model reduction technique of a FEM model, and contrary to the SEA method, it does not assume the modal energies equipartition hypothesis: the energy distribution is described between the modes of each subsystem. Hence, the rain on the roof excitation is not an assumption, and a single point excitation can be studied. SmEdA concept allows to describe the coupling between two subsystems (cabin truck, and acoustic cavity), by postprocessing the modal analysis of each uncoupled subsystem. The coupling is described through spatial and spectral coupling loss factors. The spatial coupling loss factor is deduced from the intermodal work between the mode shapes of the subsystems, on the coupled area, issued from the Dual Modal Formulation (DMF) [2]. The spectral loss factor takes into account the resonance frequencies, and the modal loss factors of the subsystems.

In this work, one focuses on the modeling of the vibroacoustic behavior of a cabin truck in the mid frequency range. The NVH performances of the cabin truck should be studied, by taking into account the trimmed materials effects. These materials can be viscoelastic layers having a damping effect on the structural vibration or porouselastic materials having an acoustic absorbing effects. A methodology based on SmEdA, to take into account these dissipative treatment, has already been developed and experimentally validated on a plate-cavity system [4–6]. In this paper, one focuses on the application and the validation of this process on an industrial case; the cabin truck is excited on a single point of the chassis frame link, and viscoelastic layers are used to damp the vibration. Due to the impedance break between the different faces of the cab, the developed model describes only the interaction between the floor and the cavity cab (i.e. the transmission through the lateral faces are supposed negligible). The structural and acoustic FEM models contain 1Mdof and 300kdof respectively, to describe the vibroacoustic behavior in the frequency band [100Hz, 2000Hz]. The trims are introduced in the FEM model through a methodology resumed in in section 4. The modal coupling loss factor can be deduced, by projecting the mode shapes of each subsystem on their FEM matrices [4–6]. Results are presented in section 5. After solving the SmEdA equations, one can deduce the pressure level on the driver ears. These numerical results are compared to experimental tests to validate the method.

2 STATISTICAL MODAL ENERGY DISTRIBUTION ANALYSIS

The finite element model FEM method remains the most common modeling method in industrial applications, despite the huge number of degree of freedom (dof) (floor model 1Mdof and cavity model 3kdof c.f. figure 1).

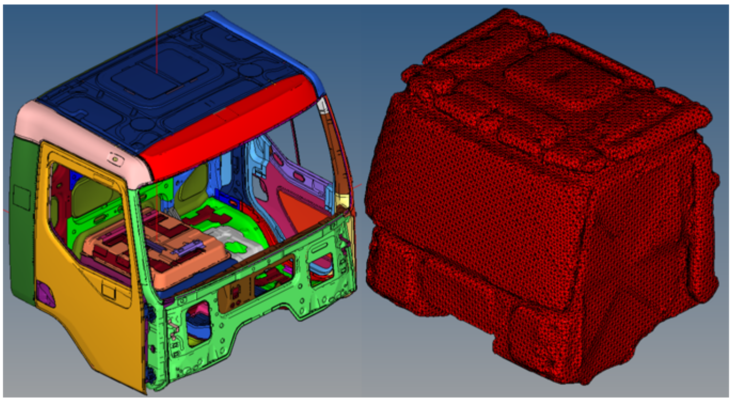


Figure 1: FEM model of the cabin and the cavity.

SmEdA method [2, 3] is based on a modal formulation of the subsystems. The coupling is described through the exchanged power between the modes of the coupled subsystems. The energy equipartition among subsystems modes is not assumed, and one deals with the energy level of each individual mode of each subsystem in a frequency range. Let one consider two subsystems denoted 1 and 2, composed of N_1 and N_2 modes respectively, in a frequency band Δf. Let p $(p = 1 : N_1)$ and q $(q = 1 : N_2)$ be the modes orders of subsystems 1 and 2 respectively, the SmEdA equation is written as [2, 3]

$$\begin{bmatrix} \Pi_{inj}^p \\ \Pi_{inj}^q \end{bmatrix} = \begin{bmatrix} \omega_p \eta_p + \sum_{q=1}^{N_2} \beta_{pq} & -\beta_{pq} \\ -\beta_{pq} & \omega_q \eta_q + \sum_{p=1}^{N_1} \beta_{pq} \end{bmatrix} \begin{bmatrix} E_p \\ E_q \end{bmatrix}. \tag{1}$$

Π_{inj} is the modal injected power of each subsystem, E is the modal energy, ω is the resonance frequency, η is the modal loss factor, and β_{pq} is the coupling loss factor between the mode p and the mode q of the subsystems 1 and 2 respectively:

$$\beta_{pq} = \beta_W \beta_\omega, \tag{2}$$

where,

$$\beta_W = \frac{\left(\int \mathbf{\Phi_p \Phi_q} dS\right)^2}{M_p M_q}, \tag{3}$$

and,

$$\beta_\omega = \frac{\eta_p \omega_p \omega_q^2 \eta_q \omega_q \omega_p^2}{\left(\omega_p^2 - \omega_q^2\right)^2 + \left(\eta_p \omega_p + \eta_q \omega_q\right)\left(\eta_p \omega_p \omega_q^2 + \eta_q \omega_q \omega_p^2\right)}. \tag{4}$$

β_W is the spatial coupling loss factor, and β_ω is the spectral coupling loss factor, M_p and M_q are the modal mass of the modes p and q respectively.

The energy distribution is described between the modes of each subsystem, and a single point excitation can be studied. One considers a noise excitation, where the injected power of a mode p on the node M is written as [1]

$$\Pi_{inj}^p = \frac{S_{ff} \mathbf{\Phi_p}^2(M) \pi}{q} \tag{5}$$

S_{ff} is the power spectrum density that is considered constant. Knowing the injected power and the modal characteristics of each subsystem, one can compute the modal energies.

3 FINITE ELEMENT MODEL WITH NON-COINCIDENT MESH

To describe the coupling between the cabin floor and the cavity, the first step is to perform an uncoupled FEM model for each subsystem. The interaction is described through the spatial and spectral coupling loss factors introduced in equations (3) and (4). The spatial loss factor depends on the intermodal work between the subsystems. It is a function of the mode shapes on the coupling surface. When dealing with non-coincident meshes between the cavity and the structure, the acoustic mode shapes on the coupling surface should be interpolated to the structure nodes or vice versa. One presents here a projection method based on the barycentric coordinate system [7]:

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

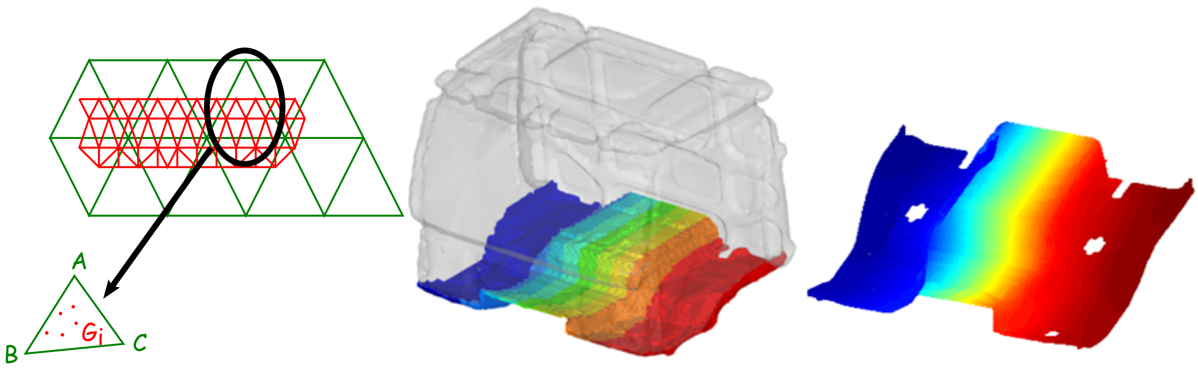
$$a\mathbf{GA} + b\mathbf{GB} + c\mathbf{GC} = \mathbf{0}. \tag{6}$$

a, b and c should be positive and normalized so that their sum is equal to 1 $(a + b + c = 1)$.

Figure 2a illustrates the problem, where the green mesh presents the fluid elements and the red mesh the structure elements. For each node G_i (of the red structural mesh) one aims to find the weights associated to A, B and C (of the green fluid mesh).

The acoustic mode shapes on the node Gi is obtained with

$$\mathbf{\Phi_{G_i}} = a\mathbf{\Phi_A} + b\mathbf{\Phi_B} + c\mathbf{\Phi_C} \tag{7}$$



(a) (b) (c)

Figure 2: FEM projection illustration: (a) Non-coincident mesh, (b) Acoustic mode shape at the floor interface, (c) Acoustic mode shape projected on the floor nodes.

Figure 2b shows the shape of the first acoustic pressure mode of the cavity on the floor interface, and figure 2c shows the projection of this mode on the floor nodes.

4 DAMPING MODELLING

Viscoelastic patches (IFF, "Insonorisant Fusible en Feuille" in french) are stuck on the floor of the cabin (figure 3) to damp the vibration. To take into account these layers in the numerical SmEdA model, one should find a modal damping loss factor of the multilayer system (floor and viscoelastic patches). The followed methodology is proposed in [4] and it can be divided in five steps:

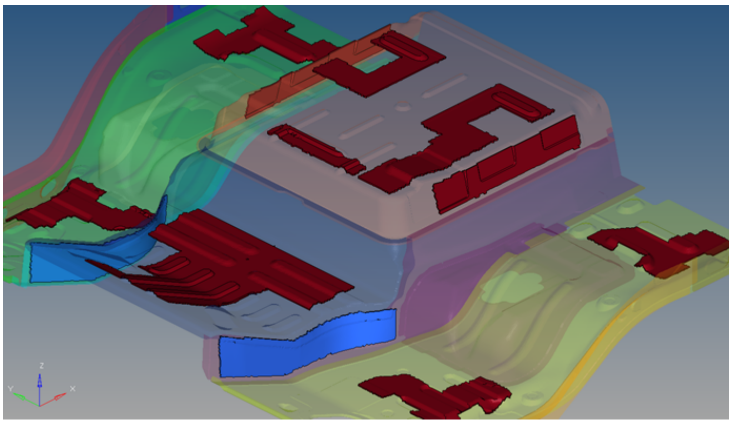


Figure 3: FEM floor model with viscoelastic patches.

1. Characterize the multilayer by computing the equivalent material properties (E, ρ, ν, η) using the method proposed in [8, 9]. Figures 4a and 4b compare the damping loss factors

and Young's Modulus of the steel and the equivalent multilayer. The equivalent properties are a function of the frequency; averaged values are considered for each frequency band.

2. Build the FE model using a shell element to model the multilayer, with the equivalent material properties computed previously that one assigns to the shell element properties.

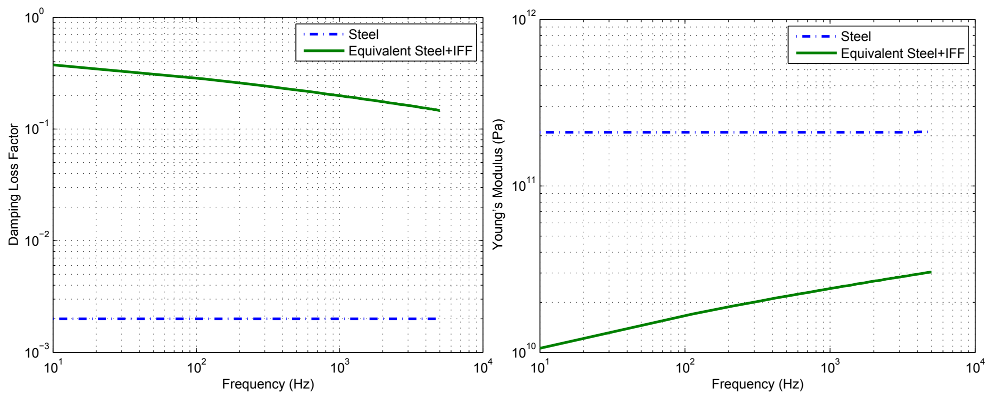
3. Compute the modal loss factors by assuming that the real mode shapes diagonalize the complex stiffness matrix K^* (Basil hypothesis). The modal loss factors are computed by

$$\eta_p = \frac{Im(\mathbf{\Phi_p}^T K^* \mathbf{\Phi_p})}{Re(\mathbf{\Phi_p}^T K^* \mathbf{\Phi_p})}. \tag{8}$$

4. Solve the SmEdA equation (1) to deduce the modal energies levels.

5. Compute the averaged energy level by summing the modal energies levels:

$$E_1 = \sum_{p=1}^{N_1} E_p \quad \& \quad E_2 = \sum_{q=1}^{N_2} E_q. \tag{9}$$



(a) (b)

Figure 4: Material properties: (a) Damping loss factor – (b) Young's modulus.

To show the efficiency of the IFF material, a numerical simulation of the floor-cavity model is illustrated, where the floor is excited on the truck frame link. Figure 5 shows the mean pressure in the cab, with and without the IFF patches. The mean pressure in the cavity is computed using the following equation:

$$< p^2 >= \frac{\rho_0 c_0^2}{V} E_2, \tag{10}$$

where ρ_0, c_0^2 and V are respectively the air density, sound speed and the volume of the cabin.

The IFF material improves directly the vibroacoustic comfort of the cabin, by decreasing of about 10dB the level from the first configuration without the IFF.

5 NUMERICAL AND EXPERIMENTAL COMPARISON

In order to be able to use the SmEdA method to study the vibroacoustic behavior of the concept lighten cabin, a numerical and experimental comparison needs to be validated in the case of a test cabin supplied by the industrial partners (figure 6). One focuses on predicting the pressure level on the driver ears for an excitation on the truck frame link. A hammer impact is used to

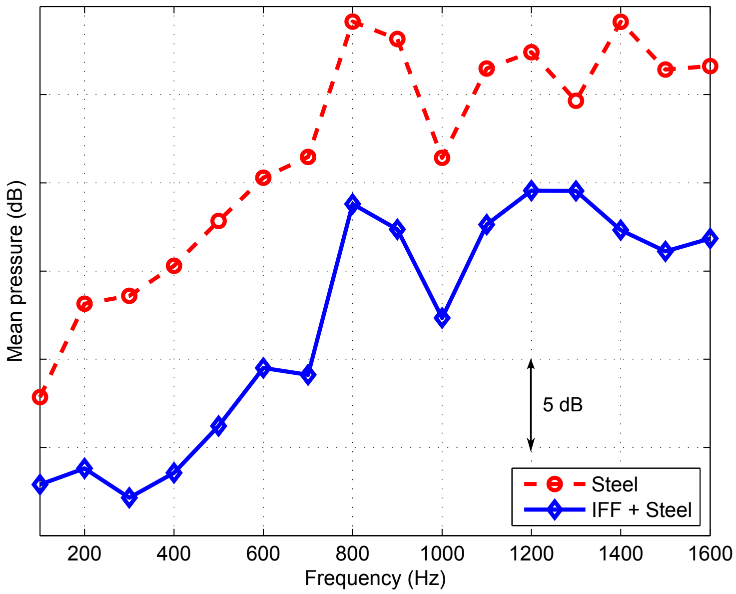


Figure 5: Mean pressure in the cabin with and without the viscoelastic (IFF) patches.

Figure 6: Test cabin measurements. Left: "body in white" test cabin; middle: truck frame link with the cabin suspension; right: impact hammer excitation.

excite the left back side frame link, an accelerometer is stuck near the excitation point, and two microphones are installed on the position of the driver ears.

To validate the impedance break hypothesis between the cabin floor and the other sides of the cavity, the acceleration frequency response function (Γ/F FRF) on the impact point is compared numerically and experimentally. Figure 7 shows the averaged FRF compared between the numerical floor model and the experimental test on the left back side of the floor. On the 16 frequency bands, only 4 bands are not well correlated.

The next application, is the comparison of the pressure field on the driver ears. During the tests, the pressure field is measured through the microphones after the hammer impact on the frame link. Numerically, after computing the SmEdA system using the equation (1), one computes the local energy level on a single point M, using the following equation [10]:

$$e(M, \Delta f) = \sum_{q=1}^{N_2} \frac{E_q}{N_2} \mathbf{\Phi_q}^2(M). \tag{11}$$

The local pressure level on the point M is deduced using the following equation:

$$p(M, \Delta f) = 2\rho_0 c_0^2 e(M, \Delta f). \tag{12}$$

Figure 8 compares the numerical and experimental results of the local pressure on the left and right ears of the driver. The numerical and experimental results are well correlated: A gap

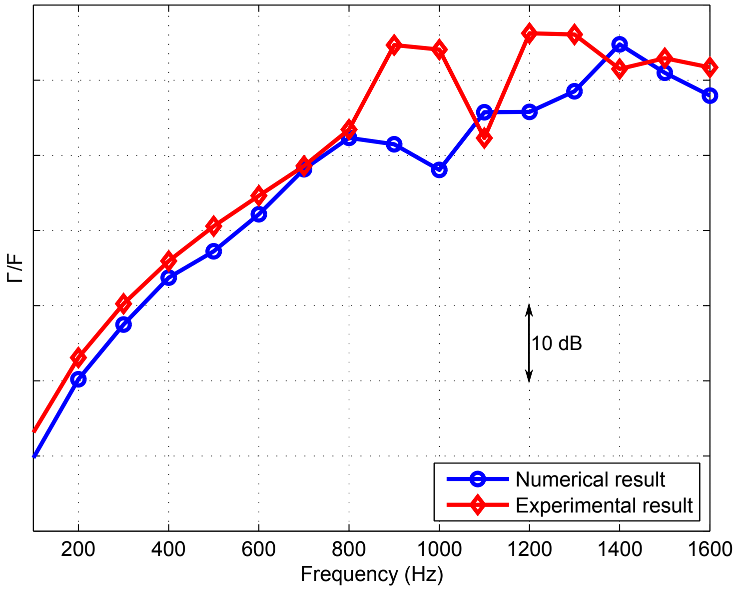


Figure 7: Acceleration level at the excitation point: numerical-experimental comparison.

average of 2.5dB is observed over 14 of the 16 frequency bands. One can also conclude that the numerical results are shifted to the low frequency starting 600Hz compared to the test results. Finally, one can mention the gap between the left and right ears: this gap is more important in the test results compared to the numerical ones.

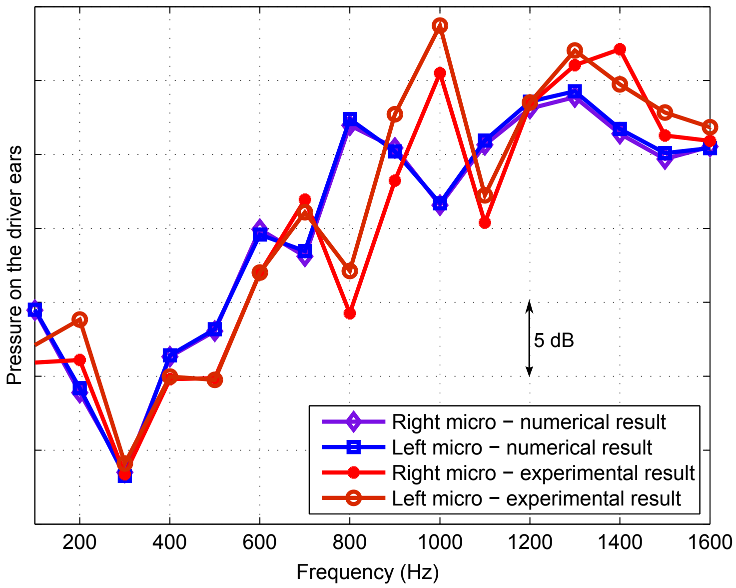


Figure 8: Pressure level on the driver ears: numerical-experimental comparison.

6 CONCLUSIONS

A numerical method is suggested to predict the vibroacoustic behavior of an industrial structure. The complexity lies in taking into account the non-coincident mesh between the subsystems, and the trim material to model the damping behavior. This method is based on SmEdA to predict the pressure level on the driver ears for a local excitation on the frame link. A mesh projection method is proposed to interpolate the fluid mode shapes to the structure nodes, on the coupling surface, using the barycentric coordinates. Trim behavior is considered through an equivalent multilayer modeling method. This method consists in identifying the equivalent mechanical parameters (density, young modulus, damping loss factor) of the viscoelastic multilayer model and to use a modal projection method to deduce the coupling loss factors required for SmEdA. The numerical-experimental comparison validate the methodology with a gap average of 2.5dB between the results. The next step is to take into account the acoustic absorbing trim in the numerical method.

7 ACKNOWLEDGMENTS

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