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Interior structure borne noise prediction of the cabin of a high speed train using FE-SEA hybrid methods

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ABSTRACT

The interior noise levels in trains has been reduced considerably during the last decades. However, the acoustic behavior has nowadays an even more important role among the comfort parameters of high speed trains. The FE-SEA hybrid methodologies raised up during the last years have given a new approach to the problem of the structure borne noise, giving the opportunity to deal with the acoustic mid frequency range, where FE and SEA systems were unable to work. However this numerical models depend on several physical parameters and, although most of them are well known in fields like the automotive, they need to be further studied for the rolling stock applications. The present paper deals with the investigation process carried out by ALSTOM to set up an hybrid FE-SEA model of the cabin of a high speed train in order to predict the interior structure borne noise, from its first steps characterizing the most important transmission path, to the structural and acoustical responses predicted by given diverse model approaches.

1. INTRODUCTION

The noise levels in transportation systems like railways have been reduced considerably during the last decades. However, the same transportation systems have become more and more popular and, in consequence, their effects on environment and personal safety play a more important role in our society. In consequence, directives like the 96/48/EC-2008/32/EC[2] (as known as Technical Specifications of Interoperability, TSI) published by the European Union regulates the noise emission levels and interior noise levels for the driver's cab of high speed trains, concerning both the new train designs and refurbishment projects.

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On the other hand, Alstom Transport designs and produces high speed trains from several years ago, from the Orange TGV up to the last development with distributed traction, called *Automotrice Gran Vitesse*, AGV. Their products are designed not only to be compliant with TSI requirements, but also to achieve high quality comfort levels for their occupants.

During years the existent numerical methodologies have permitted to solve the problem of acoustic systems like trains for two different frequency ranges: on the one hand methods like Finite Elements (FE) or Boundary Elements (BE) usually worked well at the low frequency range, where the modal density is low and the system exhibits a global modal behavior. On the other hand energy methods like Statistical Energy Analysis (SEA) predicted with sufficient accuracy at high frequencies, where the modal densities of their components are high. However, the mid frequency range rested unpredictable, and given the diversity and complexity of the components existent on a train structure, this mid frequency range is fairly large and covers a very important part of the acoustic spectrum.

The hybrid methods appeared during the last years seems to open a door to the modeling of that mid frequency range. Nevertheless, even they showed their value in other sectors like the automotive, these methods have to be further investigated in the case of rolling stock. The project presented in this paper deals with the investigation process carried out to set up the hybrid FE-SEA methodology described by Langley et al.[8] to the case of the Duplex TGV driver's cab in order to finally predict the structure borne contribution to the internal noise. The project is still ongoing when the present paper is written. However some interesting results are included, concerning either the numerical simulations or the experimental and correlation works related to the project.

(Note: the vertical values of the presented charts have been omitted for confidentiality reasons. However, they are normally presented with scales of 5 dB per division)

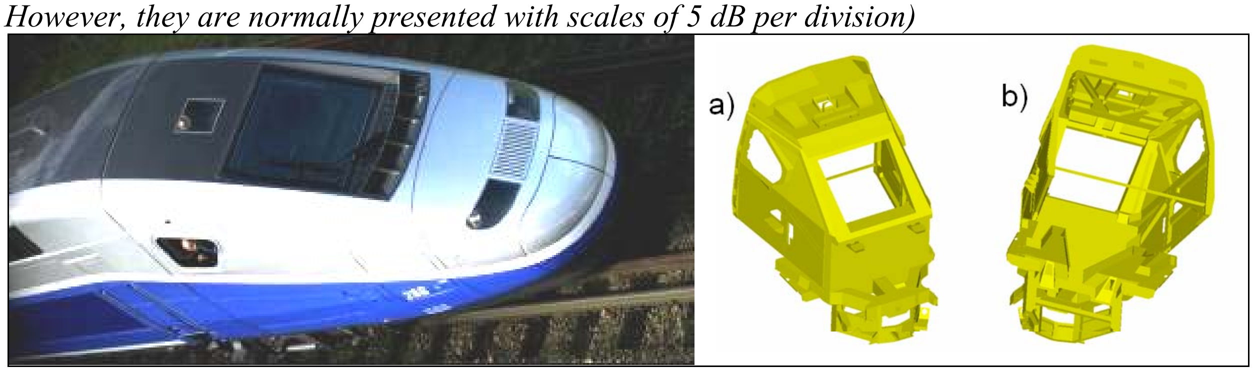


Figure 1: TGV Duplex Locomotive cabin and structure geometrical model –a) Exterior view, b) Interior view-

2. THE IMPORTANCE OF THE STRUCTURE BORNE NOISE IN THE DUPLEX TGV DRIVER'S CAB

The work here presented is part of an investigation process and its objective is not to the structure borne noise prediction but the study of the capabilities of the hybrid methodology, and the process and parameters necessary for future predictions. Indeed, the product on which the hybrid method is applied is an existent and well-known product. Therefore, previously to start the modeling process is important to determine the characteristics of the structure borne contribution to the total interior noise in the area of study. For this characterization Alstom uses an analysis tool called META-X. This tool, developed internally in collaboration with ICR (Engineering for the Noise Control) company and normally used for the experimental vibro-acoustic characterization of its products, is based on the Global Transfer – Direct Transfer (GTDT) techniques[3-5] and follows the "signal" approach instead of the "force" approach. This technique has been applied by several automotive manufacturers during the 90´s[6,7], and its main advantage is its applicability to a rolling stock projects, where normally there is no prototype available.

A. Description of the experimental tests

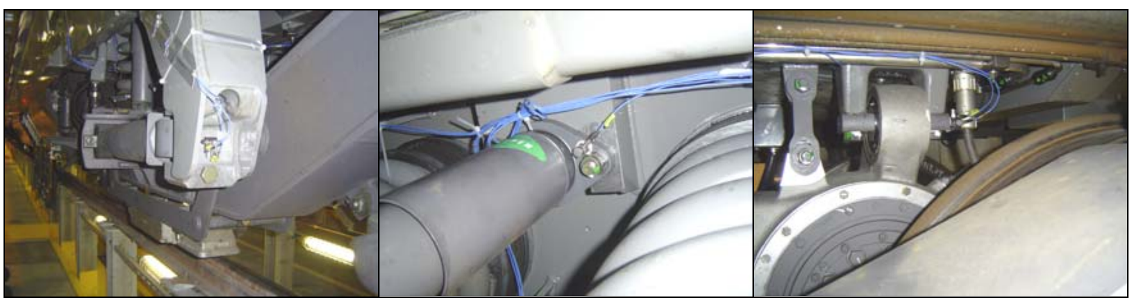
The experimental transfer path analysis was focused into two main objectives: the structure borne noise coming from the bogie area and the direct interior panel contribution to the driver's head position. In order to perform it, both areas were instrumented:

For the structure borne noise: attachment points of the bogie, traction motor and gearbox to the bodyshell were instrumented with accelerometers in order to control the vibration coming from these points and going through the structure to the interior of the cab. The main attachments measured were the following:

- Antiyaw dampers (left/right)
- Vertical dampers (left/right)
- Transversal damper
- Traction motor link
- Gearbox links

Some of these links can be seen in the figure 2.

For the interior panel contribution: all the main panels around the target microphone inside the cab were instrumented with accelerometers and control microphones.



a) Antiyaw damper link b) Vertical damper link c) Gear box link

Figure 2: Example of structure borne paths measured

The experimental characterization consisted of two different sets of measurements:

Static tests: the crossed Frequency Response Functions (FRF) and Autopower Spectra were measured in all the subsystems points as a response to a hammer impact on each one of the subsystems. The tests were performed with the train completely stopped and all the equipment off.

Dynamic tests: vibration and noise measurements simultaneously at all the subsystems positions with the train running along the track at different speeds.

Once both tests are completed, noise contributions from each path are computed. Details on the calculation process of these contributions can be found on the references[3,4].

B. Results of the tests

The results presented on the present paper are centered on the structure borne path analysis coming from the components of the bogie, which are the most concerned to the numerical study presented in the next sections. First of all, it is interesting to focus our attention to the overall noise levels (L_{Aeq}) measured inside the driver's cab at the height of the driver's ear (Figure 3). Measurements were done at 300 km/h with traction effort to keep a constant speed and without traction effort. The spectra shape is fairly similar and the main part of the acoustic energy is distributed between 63 and 630 Hz. An evident influence of the traction effort can be seen at the 1/3 octave of 2000 Hz due to gearbox contribution mainly.

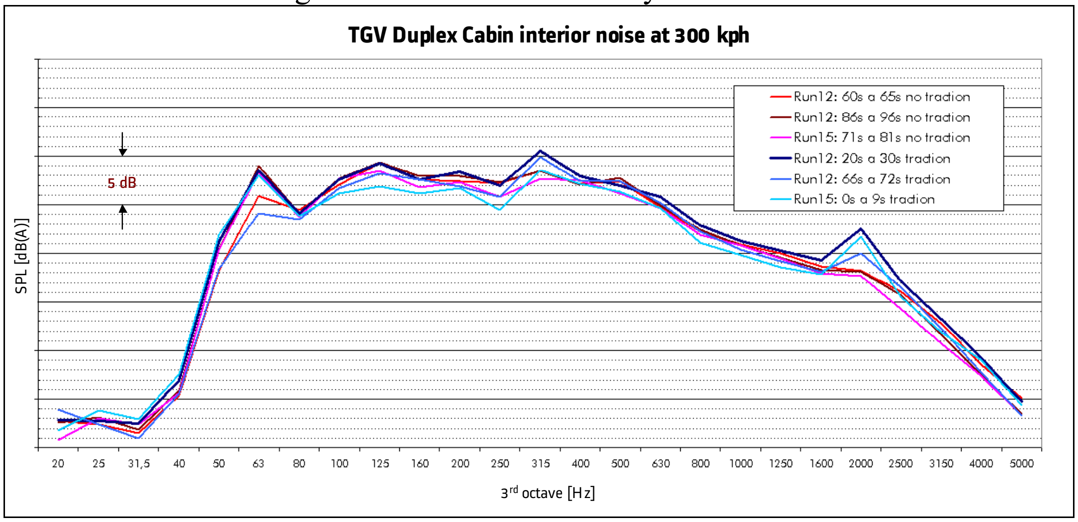


Figure 3: SPL measurements in driver's right ear position with traction effort and without it.

The Figure 4 shows the contribution of every different structure borne path and the global synthesized structure borne level in front of the total level. It can be observed the range of frequencies where the main paths are contributing showing a meaningful contribution to overall noise at low and medium frequencies.

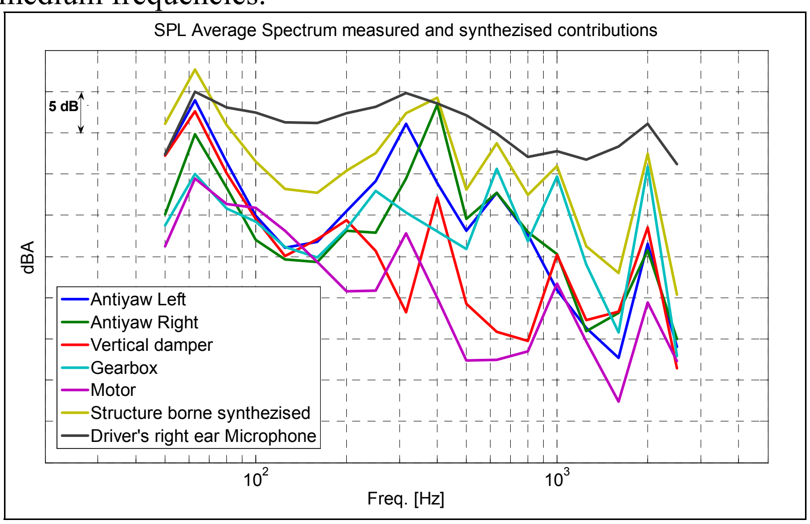


Figure 4: Structure borne noise paths contributions.

Analyzing the values measured in overall level the conclusions are that structure borne noise is close to be 50% of the overall noise level inside the cab, as can be seen in Table 1. Grouping all the different contributions give the results shown in the Figure 5, where it can be seen that the main contributor is the antiyaw damper link.

From the presented analysis it can be concluded the structure borne noise in the driver's cab is not negligible at all and it covers a wide frequency range. It is influence can be up to 50% of the overall noise level. This is the main reason why it is needed to apply whole frequency range prediction techniques in future developments to achieve noise targets at very high speeds. Furthermore, aeroacoustic excitation of the driver's cab is transmitted mainly through structure borne paths (panel excited by turbulent flow).

Table 1: SPL total measured level less structure borne synthesized level.

Test condition	$SPL_{measured} - SPL_{structure\ borne\ synthesized}$
Driver's cab in rear position, without traction	3.2 dB
Driver's cab in front position, with traction	2.2 dB
Driver's cab in front position, with traction	3.6 dB

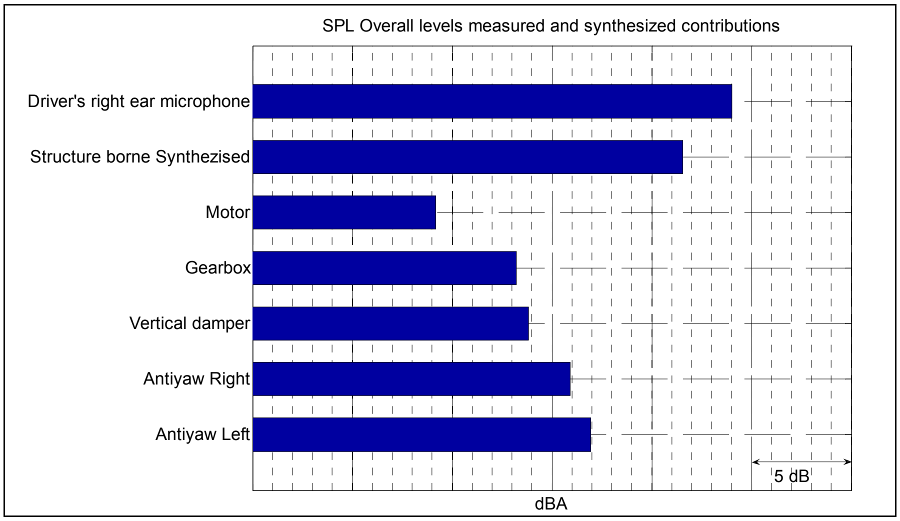


Figure 5: Structure borne paths contributions in overall levels

3. THE HYBRID FE-SEA METHODOLOGY APPLIED TO THE CABIN

C. General overview of the hybrid FE-SEA method

A detailed description of the hybrid FE-SEA methodology applied for the study is out of the scope of the present text. This is described in detail in the reference[10]. Nevertheless, a brief introduction on its general concepts and its basic equations is presented. The key idea for the development of an hybrid method is that in any complex mechanical system there would exist a frequency range in which some of their subsystems will exhibit a modal behavior while some others will have a modal density enough high to exhibit "diffuse" behavior. We can easily translate this image to the case of a train cabin, where a very stiff structure formed by beams and stiffeners is covered by thin and light plates and panels (see Figure 1). The hybrid method seeks to model the former with a FE description while the latter are described by a SEA formulation, and combine both descriptions into a single one. In order to do that, we should be able to model in someway the influence of the SEA subsystems on the FE ones and vice versa.

The link between the SEA and FE subsystems is modeled by means of some "hybrid" degrees of freedom (dof) defined on the FE-SEA subsystems' boundaries. The influence of the SEA subsystems on the FE description is divided into two different contributions: the "direct field" and the "reverberant field". When an excitation is applied on one of these "hybrid" dof a direct wave is generated and propagated directly through the SEA subsystem: this is the "direct field". This propagation can be analytically modeled for the diverse existent subsystems (plates, beams, shells, etc.) and therefore also the "direct" link between the hybrid dof. The second contribution, the "reverberant field" is related to the repeated diffuse reflections the direct field on the boundaries of the SEA subsystem. The "direct" field contribution is mathematically modeled as an additional term to the dynamic stiffness matrix of the FE system, while the reverberant field is modeled as a term added to the external forces exerted on its dof.

$$\mathbf{D}_{\mathrm{tot}}\mathbf{q} = \mathbf{f}_{\mathrm{ext}} + \sum_m \mathbf{f}_{\mathrm{rev}}^{(m)} \tag{1}$$

where $\mathbf{q}$ is the displacement vector on the hybrid nodes, $\mathbf{f}_{\mathrm{ext}}$ are the external forces applied to the model, $\mathbf{f}_{\mathrm{rev}}^{(m)}$ are the reverberant forces coming from the m-th SEA subsystem and

$$\mathbf{D}_{\mathrm{tot}} = \mathbf{D}_d + \sum_m \mathbf{D}_{\mathrm{dir}}^{(m)} \tag{2}$$

where $\mathbf{D}_d$ is the original FE stiffness matrix while $\mathbf{D}_{\mathrm{dir}}^{(m)}$ is the "direct" field stiffness matrix corresponding to the influence of the m-th SEA subsystem.

For example, consider a thin plate with four straight boundaries. If this plate is part of a larger FE model, then the degrees of freedom of the boundary can be described in terms of the nodal degrees of freedom of the FE mesh. To employ the hybrid method we need to find the direct field dynamic stiffness matrix associated with the edge degrees of freedom of the plate. This can be done by considering each straight boundary in turn, and taking it to form a segment of the edge of a semi-infinite plate. Motion of the boundary will generate waves into the semi-infinite plate, and for a given boundary motion the generated waves can be found by Fourier transform techniques. Calculation of the boundary forces associated with the generated waves then allows the dynamic stiffness matrix to be constructed: i.e., strictly the dynamic stiffness matrix of a segment of the edge of a semi-infinite plate, when the motion of the segment is described by FE nodal degrees of freedom.

This is the required "direct field" dynamic stiffness matrix for the plate edge, and repeating the process for each of the plate edges will give the total direct field dynamic stiffness matrix for the plate subsystem. The key fact is that a direct field dynamic stiffness matrix can be defined and computed for each subsystem. It is important to note that the direct field dynamic stiffness matrix can also be viewed as the ensemble average of the full dynamic stiffness matrix when averaged over random boundary reflections.

The next step is to do any kind of plausible assumption that allow to find an expression for the reverberant field[11]. On the one hand the reverberant forces $\mathbf{f}_{\mathrm{rev}}^{(m)}$ should lead to solutions fulfilling the boundary conditions of the deterministic (hybrid) boundaries and that of the diffuse boundaries. On the other hand, it is necessary to suppose an ensemble statistics. Instead of doing any particular physical assumption about the ensemble, we will suppose that one which provides the minimum amount of information: a maximum entropy ensemble. With these assumptions we obtain

$$\mathbf{S}_{ff}^{(m),\mathrm{rev}} = \alpha_m \,\mathrm{Im}\left\{\mathbf{D}_{\mathrm{dir}}^{(m)}\right\} \tag{3}$$

where the proportionality α_m is defined in terms of the energy density of the SEA subsystem

$$\alpha_m = \frac{4E_m}{\pi\omega n_m}\,\mathrm{Im}\left\{\mathbf{D}_{\mathrm{dir}}^{(m)}\right\} \tag{4}$$

where E_m is SEA subsystem reverberant energy and n_m its modal density.

Equation (3) implies that the cross-spectral matrix of the force exerted by the reverberant field is proportional to the resistive part of the direct field dynamic stiffness matrix, which is a form of diffuse field reciprocity statement.

Recombining equations (1-4) we obtain the coupling between the SEA energy and the cross-spectral matrix of the displacements $\mathbf{q}$

$$\mathbf{S}_{qq} = \mathbf{D}_{\text{tot}}^{-1}\left[\mathbf{S}_{ff} + \sum_m \left(\frac{4E_m}{\pi\omega n_m}\right)\operatorname{Im}\left\{\mathbf{D}_{\text{dir}}^{(m)}\right\}\right]\mathbf{D}_{\text{tot}}^{-\text{H}} \tag{5}$$

Finally, we can state the SEA equations where injected power and the power leaving are calculated in terms of the cross-spectral matrix $\mathbf{S}_{qq}$, leading to

$$\omega\left(\eta_j + \eta_{d,j}\right)E_j + \sum_m \omega\eta_{jk}n_j\left(E_j/n_j - E_m/n_m\right) = P_{\text{in},j}^{\text{ext}} \tag{6}$$

Equations (5) and (6) form the two main equations of the FE-SEA coupled system: equation (6) has precisely the form of SEA, but the coupling loss factors η_{jk} and loss factors $\eta_{d,j}$ are calculated by using the FE model augmented by the direct field dynamic stiffness matrices; furthermore, equation (5) has the form of a standard deterministic FE analysis, but additional forces arise from the reverberant energies in the subsystems. If no SEA subsystems are included then the method becomes purely FE; on the other hand, if only the junctions between the SEA subsystems are modelled by FE, then the method becomes purely SEA, with a novel method of computing the coupling loss factors.

E. Modeling process and system partitioning

The modeled cabin is a very complex structure formed by several layers and components whether in the main structure itself or in elements attached to it (see Figure 1). Indeed, from the structure point of view, it is a very stiff structure made of beams with a heavy and rigid substructure on the front side designed for crash. In the laterals and the roof several light metallic panels are welded to the structure closing both sides of the cabin. Windscreen and lateral windows are multilayered glasses. The floor is made by a metallic panel with an aditional multilayered panel elastically suspended on it. Finally, the external side is covered by a polyester nose while the internal side is covered by several kinds of trimmings and components.

Given that complexity, the investigation process has been divided into several phases following in some way the production process, in order to acquire in each step the knowledge necessary to model correctly the diverse elements included. Thus, the modeling process has followed these phases:

Configuration 1. The Body-in-White (BIW) of the cabin is modeled alone. An experimental modal analysis of the cabin is performed in order to correlate the FE model.

Configuration 2. The BIW is modeled attached to BIW of the rest of the locomotive structure. Individual FRF tests are performed in order to obtain damping loss factors (DLF) and cross-inertances are measured to correlate the results of the models.

Configuration 3. The previous configuration is painted with damping materials and covered with fiber in the internal sides. Like in the previous, FRF tests are performed in order to obtain damping loss factors (DLF) and cross-inertances are measured to correlate the results of the models.

Configuration 4. The crash system, equipments, cables and the floor are installed to the previous configuration. An experimental modal analysis of the floor is performed and correlated with its own FE model. The models results are compared with cross-inertances measurements of the real system.

Configuration 5. The cabin is in its final conditions: windshield, lateral windows and doors fitted, as well as, the polyester nose and the interior trimmings; the cabin is finally sealed. At this stage the noise prediction is performed. Experimental modal analysis of the windshield and the lateral windows are performed individually and elastically suspended. Both results allow to correlate individual FE models of these components.

For each configuration, a complete FE model and an hybrid FE-SEA (by means of VA One software) are performed, and the results of both compared and correlated with experimental data. The frequency range of interest is [100, 1000] Hz. In order to apply correctly the SEA theory to any subsystem a minimum modal density is required. In our case two criteria were adapted in order to investigate their performances: 5 modes per 3rd octave band and 3 modes per 3rd octave band. In order to avoid uncertainties, the experimental tests were performed over the same cabin, following it through the different stages of the complete production process.

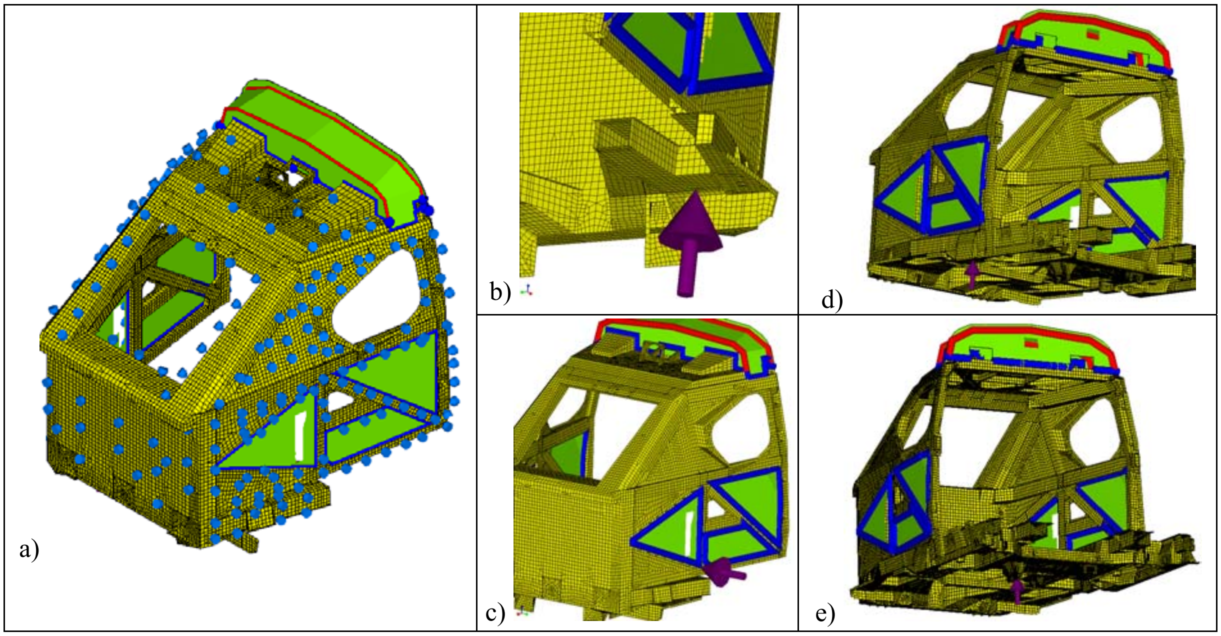


Figure 6: a) Hybrid model general view (the green areas are the SEA subsystems).
b) Input point on the lower side. c) Input point on the left lateral side.
d) Input point on the antiyaw position. e) Input point on the gearbox position.

4. RESULTS

At the moment of writing the present document, only the first three configurations were realized. Therefore, no acoustic prediction at these stage exists, and the results presented deals only with the mechanical dynamic response of the cabin. Nevertheless, some interesting results are presented for each one of the existent configurations.

F. Results for Configuration 1

The Figure 7 shows the response of several lateral panels to an input force on a lower side point. The results represent the averaged acceleration over the panel area for both the full FE model and the hybrid model compared to the same experimentally measured quantity. In general both models give responses with same order of magnitude of the experimental and, without exact agreement in any case, the hybrid model seems to be closer to the tests.

It is important to note that in that first configuration the structure studied was the BIW. Thus, the damping factors of the model were very low.

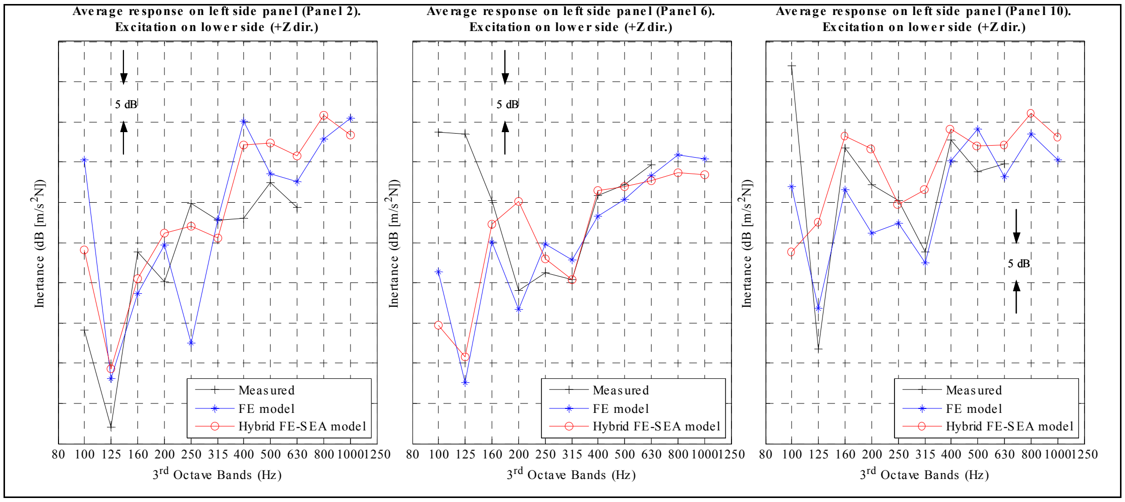


Figure 7: Results for Configuration 1. Comparison for the averaged cross-inertance (measured, calculate by FE and hybrid) at three lateral panels between 100 and 1000 Hz. Input on the lower side of the cabin in vertical direction.

G. Results for Configuration 2 and 3

The following figures show the comparison between the results for configuration 2 and 3. The left chart of every figure corresponds to the configuration 2, while that one in the center is for configuration 3. The vertical scales are the same in all the charts, with 5 dB per division. The curves, unlike the previous ones, do not correspond to an average over an area, but to a punctual response.

The Figure 8 shows that comparison for the point #15 in the left lateral side as a response to a unitary force applied in the antiyaw position in the vertical direction (see Figure 6.d). We can observe a good agreement in both configurations between the tested curve and the modeled ones. We can also observe the effect of the damping added in configuration 3, making the response significantly lower. Similar results are found in Figure 9 for another point in the left lateral side.

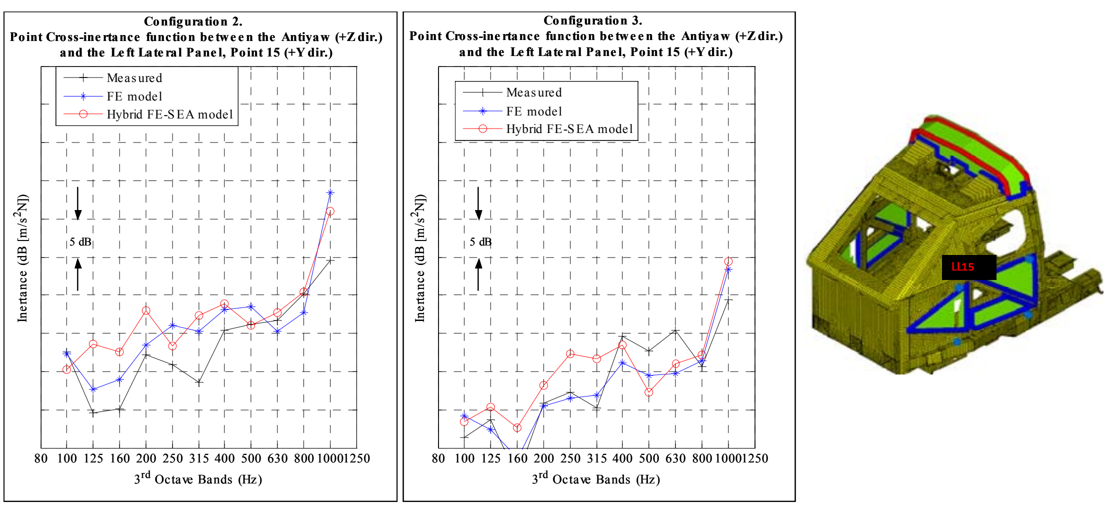


Figure 8: Comparison between the results for Configurations 2 and 3. Point cross-inertance (measured, calculate by FE and hybrid) between 100 and 1000 Hz. Input on the antiyaw position in vertical direction (Z). Response at left lateral panel (Point #15) in normal direction (Y).

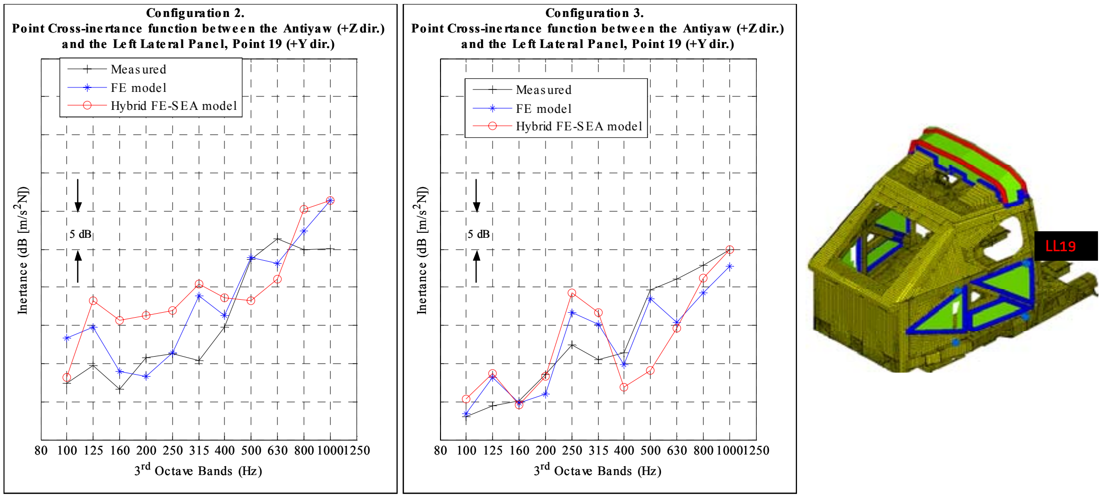


Figure 9: Comparison between the results for Configurations 2 and 3. Point cross-inertance (measured, calculate by FE and hybrid) between 100 and 1000 Hz. Input on the antiyaw position in vertical direction (Z). Response at left lateral panel (Point #19) in normal direction (Y).

Figures 10 and 11 show the same kind of result with the same previous response points (#15 and #19 in the left lateral side respectively), but with the input unitary force applied to the gearbox position in the vertical direction (see Figure 6.e)

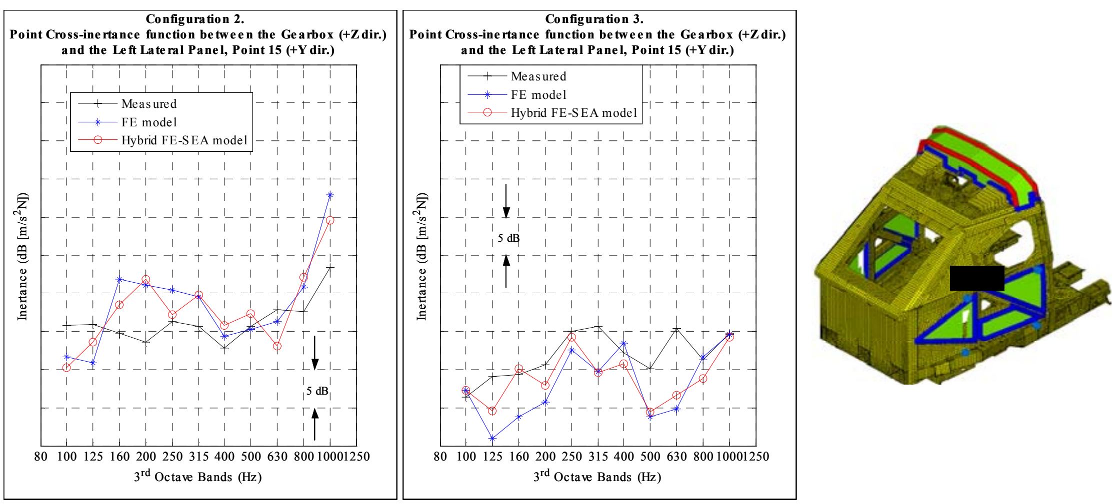


Figure 10: Comparison between the results for Configurations 2 and 3. Point cross-inertance (measured, calculate by FE model and hybrid model) between 100 and 1000 Hz. Input on the gearbox position in vertical direction (Z). Response at left lateral panel (Point #15) in normal direction (Y).

In general, the results show a good agreement between two models (full FE and hybrid FE-SEA ones), and normally they lie not far from the measured values (with a discrepancy lower than 5 dB in the most of the frequency bands). Furthermore, it is quite general (with some exceptions) that when the discrepancies between the modeled and the tested curves are higher, the hybrid model gives results closer to the experiment.

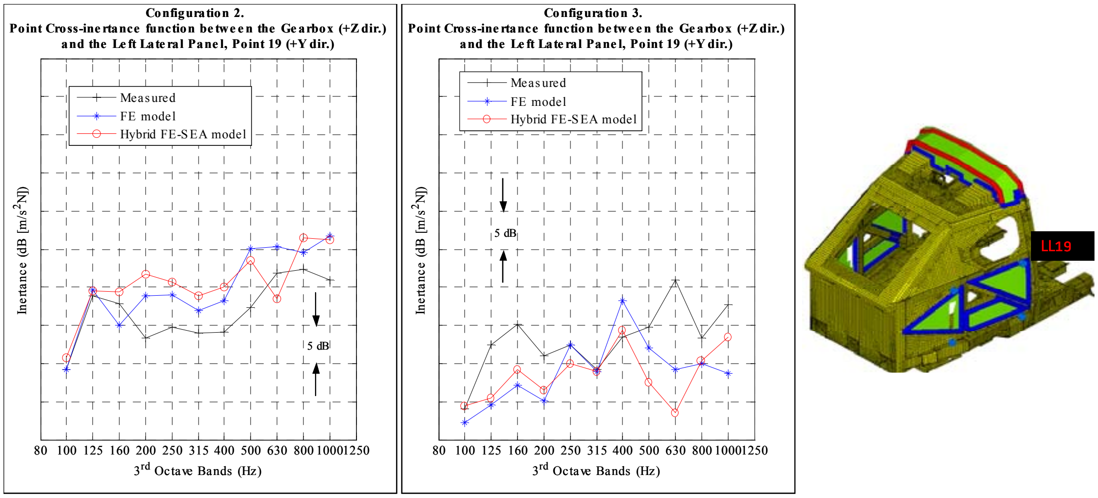


Figure 11: Comparison between the results for Configurations 2 and 3. Point cross-inertance (measured, calculate by FE model and hybrid model) between 100 and 1000 Hz. Input on the gearbox position in vertical direction (Z). Response at left lateral panel (Point #19) in normal direction (Y).

6. CONCLUSIONS AND FURTHER WORK

The feasibility of the hybrid FE-SEA methodology to the rolling stock is investigated by its application to the case of a high speed train cabin. The investigation process deals with the different production stages in order to obtain information about the correct parameters of every part of the system. That investigation is still in progress at the moment of the paper presentation and the results presented deals only with the dynamic mechanical response of the system. The results obtained are quite promising showing a good correlation between FE prediction and FE/SEA hybrid methods, allowing in the future to work with hybrid models to represent the vibroacoustic behavior of high speed trains in the whole frequency range. The use of FE-SEA hybrid models covering the medium-high frequency range allows to handle the type of problem described and it will be specially useful for large models (like the whole driver's cabin or passengers area) where classical FE models are not possible to work with.

The hybrid FE-SEA partitioning criteria have been set up to a minimum of three and five modes per 3rd octave band. The results obtained show normally a good accuracy for the vibration prediction at the frequency range of [100, 1000] Hz and the discrepancies between the models and the experimental curves are normally below 5 dB. Furthermore, in those cases where the differences between the models and the tests are higher, the hybrid methodology seems to give better a agreement (with some exceptions).

Finally the software tools associated to the hybrid FE-SEA used during this application have showed a good integration into the industrial process. The technology is being integrated into the Alstom design process to assure reliable structure borne noise predictions during early stages of the design of a new product.

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