SEA Modelling and Transfer Path Analysis of an Extensive RENAULT B segment SUV Finite Element Model

M. Brandstetter[1], C. Dutrion[2], P. D. Antoniadis[1], P. Mordillat[2], B. Van den Nieuwenhof[1]

[1] *MSC Software Belgium SA / Free Field Technologies, 1435 Mont-Saint-Guibert, Belgium, Email: markus.brandstetter@fft.be*

[2] *Renault, Guyancourt, France, Email: cecile.dutrion@renault.com*

Abstract

Simulation in automotive industry helps reducing vehicle development time and prototype costs and increasing the quality and robustness of the vehicles. Finite Element (FE) models are an established method to analyze Noise, Vibration and Harshness (NVH) problems. FE analysis is indeed a robust and versatile approach that can be used for a large number of applications, like noise prediction of different sources inside and outside the vehicle or pass-by noise simulation. Typically, results feature high quality correlations. Future challenges, such as electric motorized vehicles, with changes in terms of motor noise spectrum, will require an extension of the existing approaches.

In this article a Statistical Energy Analysis (SEA) approach is introduced, called Virtual SEA. This approach is based on FE models, from which all necessary information is extracted. It allows extending the frequency range of validity of these FE models. A Virtual SEA model of a RENAULT B segment SUV is presented and validated against a standard solution for the frequency response in modal coordinates. The advantages of the Virtual SEA model in terms of computational time and resources are shown. Finally, a Transfer Path Analysis (TPA) is performed to evaluate energy exchanges between subsystems, and thus efficiently determine vehicle modifications that could lead to noise reduction.

Introduction

Finite Element Analysis (FEA) is a well-established, robust, and versatile solution for various vibro-acoustic and aeroacoustic challenges in industry. For specific topics, high frequency calculations are of particular interest. They are difficult to achieve with FE methods, even if past years have shown that High Performance Computing (HPC) and mesh adaptivity allow to increase the FE high frequency limit drastically [1].

SEA method offers an interesting solution for such topics. Different approaches can be used to create SEA models,

- analytic SEA, with analytic modeling of subsystems and experience-based power exchanges between subsystems [2-4],
- experimental SEA, using the Power Injection Method (PIM) to compute the power exchange factors [5-6],
- simulation-based SEA, where the SEA model is built up to a certain degree from a vibro-acoustic FE simulation or combining FE and SEA [4,7-11].

Various topics can be tackled with such models in industrial applications, such as automotive, aerospace, ship building, machinery, train and building industry [2- 6, 8-13].

Depending on the method, SEA offers

- affordable solutions at high frequencies;
- additional capabilities for transfer path analysis;
- a model valid for mid and high frequency ranges if it is based on energy simulations.

Several options are available to create the Coupling Loss Factor (CLF) and Damping Loss Factor (DLF) for SEA. They can result from measurements, estimations or numerical simulations [14-15]. The latter option is used in the proposed Virtual SEA approach [9].

This article presents a method, which allows creating models based on FE modal extractions. The information is extracted uniquely from the FE models including the junctions / connections between the subsystems. The article presents an SEA energy-based model of an entire RENAULT B segment SUV. The model is validated between 80 and 250 Hz against a Modal Frequency Response (MFR), showing its low/mid frequency validity. Finally results and TPA capabilities of the SEA model are presented.

Methodology

This section describes the Statistical Energy Analysis (SEA) utilized for this study; in the second part the Virtual SEA approach is discussed.

Statistical Energy Analysis Modelling

SEA is an energy- and power-based method, initially introduced in the 60s, by the independent works of Lyon and Maidanik [16], and Smith [17]. Vibro-acoustic simulation can be based on SEA modelling. Unlike a finite element vibro-acoustic model, a SEA model relies on the computation of the energy level of different subsystems. Subsystems are a user-defined substructuring of the investigated object. In SEA, results are computed on subsystems and integrated over frequency bands. The number of degrees of freedom is equal to the number of subsystems. For each subsystem, the power dissipated inside the subsystem and the flux to the other subsystems are determined. A simple model containing 3 subsystems is displayed in Figure 1.

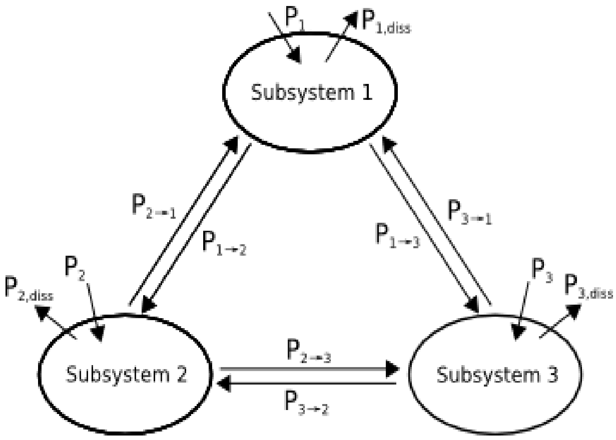

Figure 1: A 3 subsystem-SEA model.

Based on SEA assumptions, the coupling between subsystems needs to be weak. Weak coupling means for example that the power dissipated inside a subsystem is higher than the powers exchanged with the other subsystems. In SEA, results are averaged over subsystems, which should be taken into account during subsystems definition.

SEA solves the power balance between the subsystems, $P_i = \omega\eta_{ii}E_i + \sum_{j\neq i}^{N} \omega\left(\eta_{ij}E_i - \eta_{ji}E_j \right)$ with η_{ii} the damping loss factor (DLF) of subsystem$i$, and η_{ij} the coupling loss factor (CLF) between subsystems i and$j$. N is the number of subsystems, E_i the energy level of subsystem i, P_i the power excitation of subsystem i, and ω the angular frequency.

The entire SEA approach is illustrated in eq. (1), with n_i the modal density of the subsystem i.

$$
\begin{Bmatrix} P_1 \\ P_2 \\ \vdots \\ P_N \end{Bmatrix} = \omega \cdot \begin{bmatrix} (\eta_{11} + \sum_{j \neq 1}^{N} \eta_{1j}) \cdot n_1 & -\eta_{21} \cdot n_2 & \cdots & -\eta_{N1} \cdot n_N \\ -\eta_{12} \cdot n_1 & (\eta_{22} + \sum_{j \neq 2}^{N} \eta_{2j}) \cdot n_2 & \ddots & \vdots \\ & & & \vdots \\ \vdots & \ddots & \ddots & \\ -\eta_{1N} \cdot n_1 & \cdots & \cdots & (\eta_{NN} + \sum_{j \neq N}^{N-1} \eta_{Nj}) \cdot n_N \end{bmatrix} \cdot \begin{Bmatrix} E_1/n_1 \\ E_2/n_2 \\ \vdots \\ E_N/n_N \end{Bmatrix} \; [\mathrm{W}] \qquad (1)
$$

SEA approach has advantages, such as

- a dramatic reduction of the number of degrees of freedom compared to FE;
- a well-conditioned, symmetric and diagonal dominated matrix (see eq. (1));
- an affordable solution at high frequencies;
- meaningful response quantities in the so-called mid-frequency (MF) / high-frequency (HF) range (frequency-averaged energy levels on sub-systems).

Limitations of SEA modeling, which typically restrict its usage to the so-called HF range, are due to its requiring

- a sufficient level of modal density;
- a homogenous vibrational energy density;
- a weak coupling between the subsystems.

Virtual SEA Approach

The presented Virtual SEA method is based on FE-simulations and Actran / Nastran commercial codes [7]. It uses the structural modes, modal stiffness matrix and mass matrices obtained by a modal extraction. Besides a user-defined partitioning, no other information is needed to build the SEA model as the global modal representation includes junctions between subsystems.

Virtual SEA does the Power Injection Method (PIM) virtually; the PIM needs to excite all modes of a subsystem, which is best done with an uncorrelated random excitation. Three different types of energy can be computed: potential, kinetic or total energy. Substructuring should be chosen depending on the desired results. It can be finer or coarser; but in any case the weak coupling condition should be respected. Substructuring can be done in the user-friendly graphical interface Actran VI.

Various excitations for Virtual SEA are available, including

- injected power;
- turbulent boundary layer;
- diffuse sound field;
- uncorrelated random excitations, used for the PIM;
- etc.

Car Door Model - Frequency Extension

In a first step the extension of mesh validity is shown on a car door model, see as well [18]. The car door is being modeled with 2 mesh sizes, one being valid up to 2000 Hz (89336 elements, 8 mm), the other being valid up to 8000 Hz (272520 elements, 4 mm). Both meshes are shown in Figure 2, as well as the 7 subsystems.

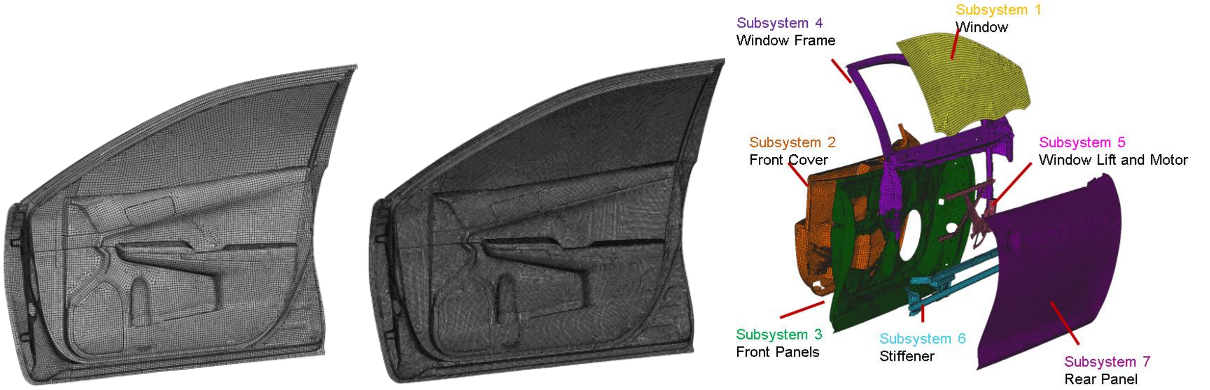

Figure 2: FE car door model with two different mesh sizes and substructuring into 7 parts. Left: mesh with 8 mm elements. Middle: mesh with 4 mm elements. Right: subsystems definition.

Results

First, the modal extraction is done for both models up to 8000 Hz. Then the Virtual SEA model is computed. The results are shown in Figure 3.

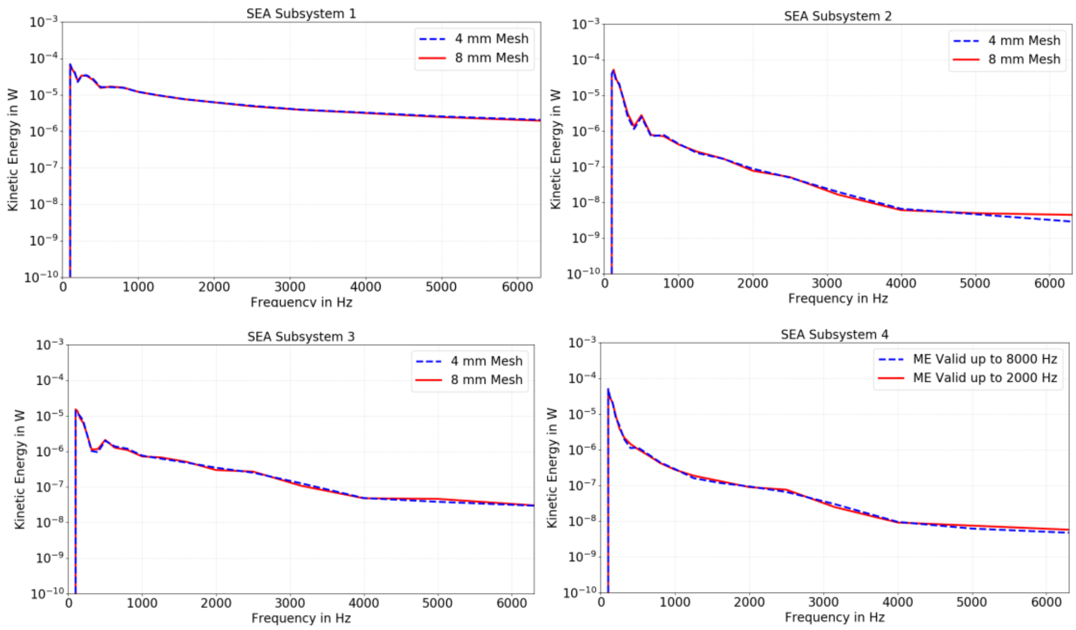

Figure 3: Energy levels for subsystems 1 to 4 for both mesh sizes (4 mm valid up to 8000 Hz, 8mm valid up to 2000 Hz). Results are output on third octave band mid frequencies.

Conclusions

FE mesh criterion does not need to be respected for Virtual SEA modeling. A reason is the smoothing through frequency and spatial band averaging. As a conclusion, one can use FE meshes valid up to 4 times their mesh size for Virtual SEA.

Energy levels of the 4 mm mesh are slightly below the 8 mm mesh, since the coarser mesh behaves slightly stiffer numerically. Using coarser meshes reduces the computational time in this case at least by a factor 2.

RENAULT B segment SUV Model

In a second step the Virtual SEA method is applied on an industrial case, a RENAULT B segment SUV. This section describes the FE element model and the substructuring of the SEA model. A global substructuring level is chosen for investigation.

The aim of the simulation model is to evaluate the behavior of the RENAULT B segment SUV under random road noise excitations of the structure.

General Description

A modal extraction is computed, outputting the Eigenmodes up to 710 Hz. The number of elements, nodes and modes up to 710 Hz are shown in Table 1.

RENAULT B segment SUV	Model parameters			
	Number of elements	Number of nodes	Number of degrees of freedom	Number of modes up to 710 Hz
FE model	2.9E6	2.3E6	12.3E6	6275

Table 1: RENAULT B segment SUV, FE-Model description

The model includes wheels, modeled as DMIG cards in the MSC Nastran input deck. Due to the wheel floor contact, the model has not 6 rigid body modes, but only one low frequency mode describing the car rolling forward. The model features a global structural damping of 2 %. Certain parts have been replaced by a mass node, such as the dashboard and the console, the front plastic cover and the fuel tank.

Substructuring – SEA Modeling

The substructuring needs to be chosen depending on the aim of the analysis. Two substructuring set-ups are chosen here: a coarse substructuring to investigate the global behavior of the car, see set-up 1 (Figure 6), and a fine substructuring, to investigating its local behavior, see set-up 2 (Figure 13). In this article only the global behavior of the car is investigated, based on set-up 1.

Validation

As described in the section Virtual SEA approach, the model is created on the basis of a MSC Nastran SOL 103 modal extraction. The model of the RENAULT B segment SUV is still in development phase; therefore measurements are not available yet for validation. In the scope of this paper, validation is proposed by comparing SEA results to Modal Frequency Response (MFR) calculation results.

The validation is performed up to 250 Hz. The models are excited with a random excitation on the windshield, one of the 24 subsystems, defined in set-up 1. The Virtual SEA model computes the kinetic energy in all the subsystems. The velocity is deduced from the kinetic energy E_{Kin} using the equation

$$E_{\mathrm{Kin}} = \frac{mv^2}{2}, \qquad [\mathrm{J}] \qquad (2)$$

with m the mass and v the normal velocity.

The validation is done in three steps. The results respect the same mass in both models.

1. First the global kinetic power of the entire model obtained with the MFR approach is compared to the summed kinetic powers of all subsystems obtained with the Virtual SEA approach (Figure 4 left side).

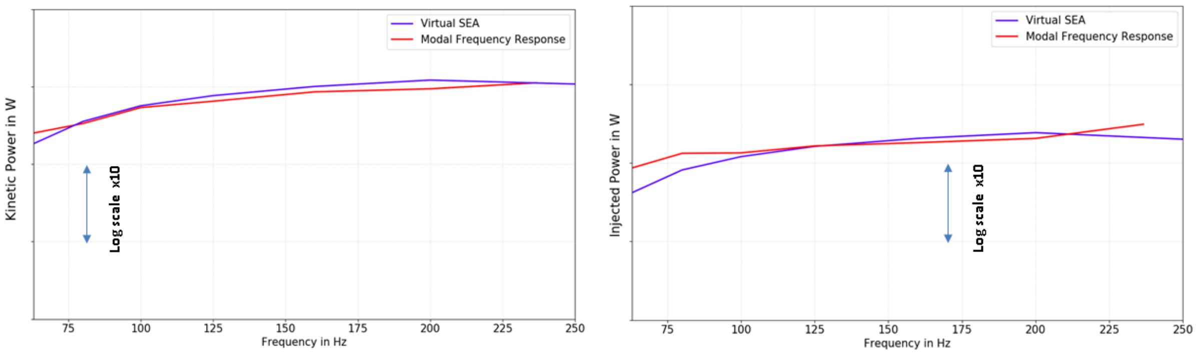

Figure 4: Comparison between the MFR approach and the Virtual SEA approach. Left: kinetic energy evaluated on the entire car model. Right: power injected as excitation on the windshield.

2. Second the injected power of the excitation is compared (Figure 4 right side).
3. Third the mean velocity of the SEA model computed with equation (2) is compared to the mean velocity of the MFR results. Figure 5 shows that the vibration of the windshield is mainly driven by the normal velocity, and that the SEA normal velocity level matches the MFR normal velocity level.

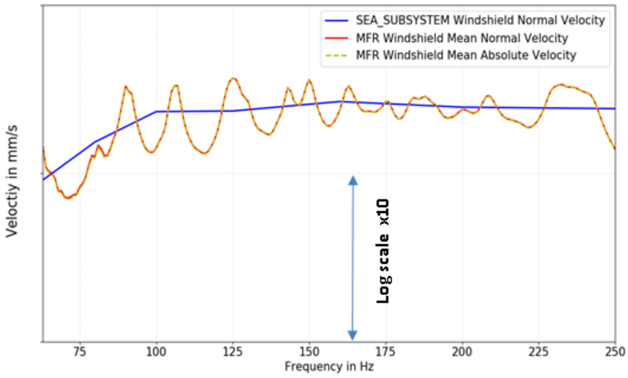

Figure 5: Comparison of the velocity between MFR and SEA model

A comparison between the SEA model and the conventional FE model is possible at low frequencies. Indeed, provided that the excitation and injected power level are meeting in both models, the weak-coupling assumption of SEA is not required to compare results.

MFR Calculation

In MFR kinetic energy for the global system can easily be computed. MFR also allows computing the kinetic energy on subsets of nodes. This can turn into expensive computational costs. Computing the kinetic energy on all subsystems cannot be done, as the results need to be output on all nodes over the entire calculated frequency range, using a small frequency step size. For all subsystems, RAM and disk memory, even of HPC computers, are not sufficient to compute the power level of each subsystem. However the MFR is affordable for one or two subsystems and it is done here for validation purposes.

Results

Subsystem set-up 1 is chosen to investigate the global behavior of the model, see Figure 6.

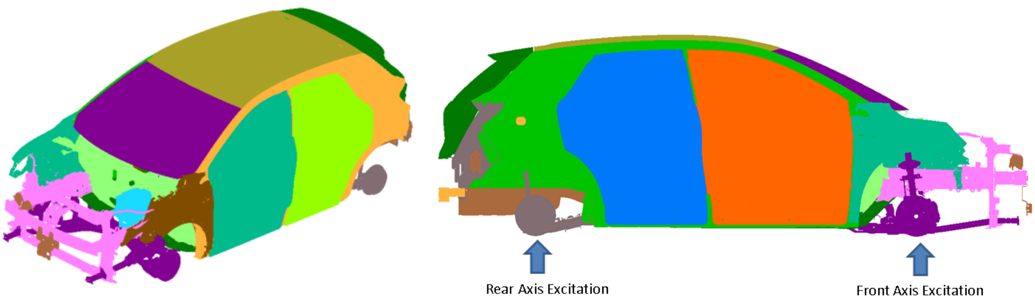

Figure 6: RENAULT B segment SUV subsystems

The different subsystems are given with their identification numbers in Table 2.

	ID	Subsystem	ID
battery	10001	steering column	10017
front axis	10002	windshield	10021
front bumper	10003	front part BIW	10022
front seats	10004	front left part	10023
left front door	10005	front right part	10024

left rear door	10007	firewall	10025
luggage door	10008	cross member	10026
package tray	10009	floor	10027
radar sensors	10010	roof	10028
rear axis	10011	rear floor	10029
rear bumper	10012	rear part BIW	10030
rear seat frame	10014	left structure BIW	10031
right front door	10015	right structure BIW	10032
right rear door	10016		

Table 2: Subsystem IDs of the set-up shown in Figure 6

The kinetic energy results are shown for two loadcases: for a front axis excitation (Figure 7), and for a rear axis excitation (Figure 8). Results are computed for all subsystems, but only the ones with the highest energy levels are displayed on the graphs of Figure 7 and Figure 8.

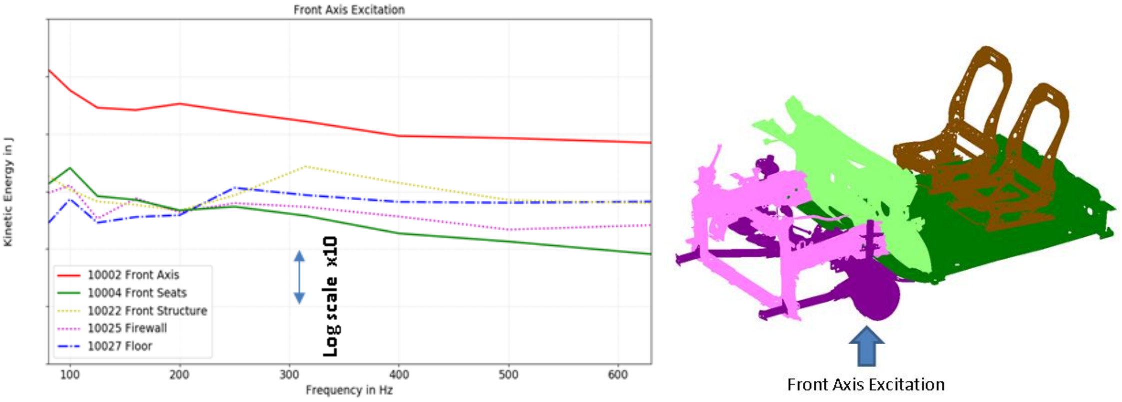

Figure 7: Kinetic energy levels for front axis excitation. Only the main contributing subsystems are shown.

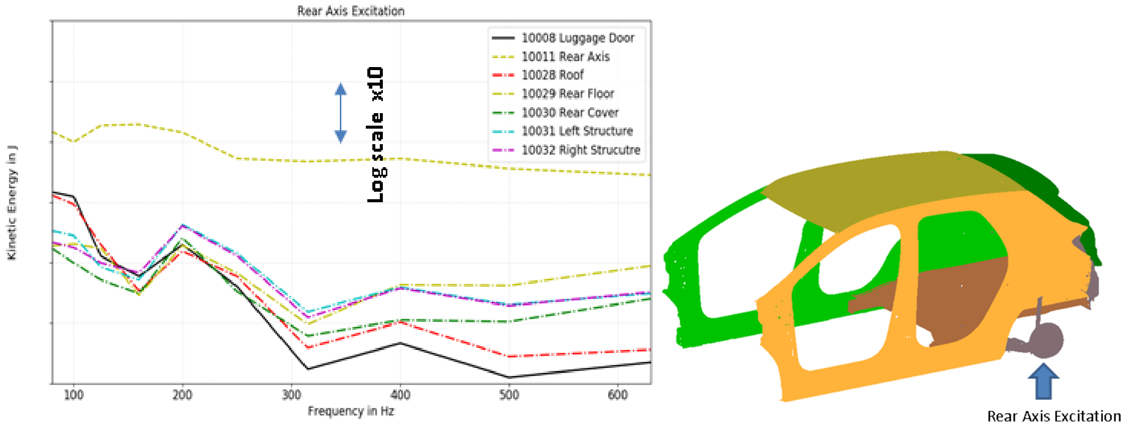

Figure 8: Kinetic energy levels for rear axis excitation. Only the main contributing subsystems are shown.

The Virtual SEA approach also enables to highlight the vibration transfer paths of the vehicle. By visualizing the CLF between the subsystems, one has the information of the amount of power exchanged between the subsystems. This allows following the power flow through the subsystems. The CLF from the front axis to the front part in Figure 9 explains the frequency peak of the front part visible at 315 Hz in Figure 7. Moreover the fluxes can be computed, as shown in Figure 10.

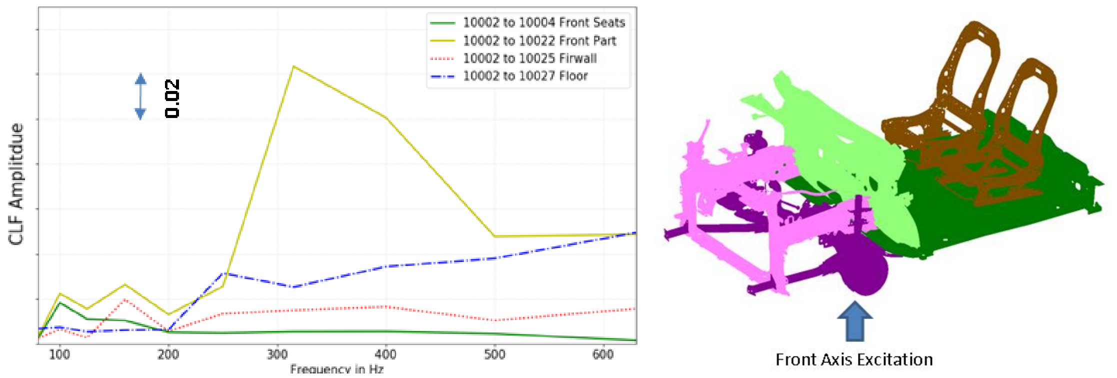

Figure 9: CLF from the excited subsystem (front axis) to front seats, front part, firewall and floor.

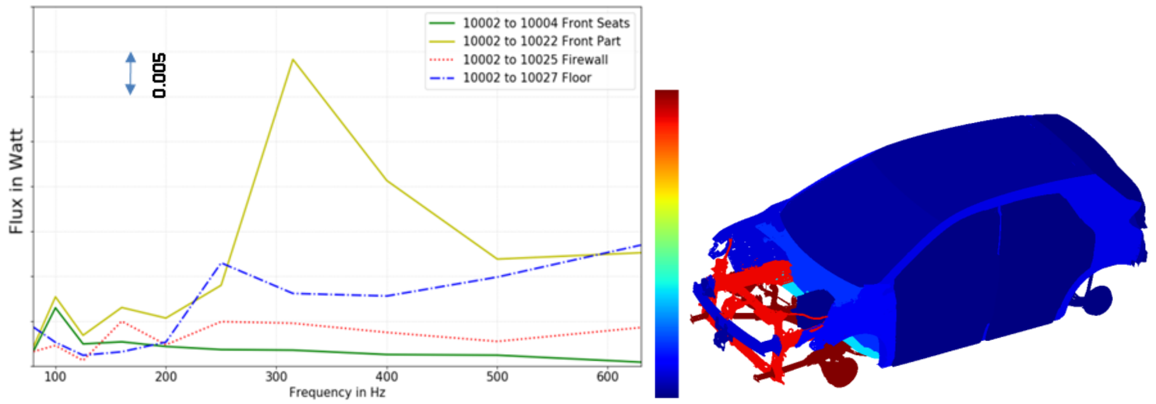

Figure 10: Flux from the excited subsystem (front axis) to the other subsystems. Left: fluxes to the most important subsystems over frequency. Right: fluxes to the different subsystems at 400 Hz (blue: low level, red: high level).

Transfer Path Analysis

The transfer path analysis is done for two transfer paths:

1. from the front axis to the front seat (see Figure 11),
2. from the rear axis to the roof and luggage door (see Figure 12).

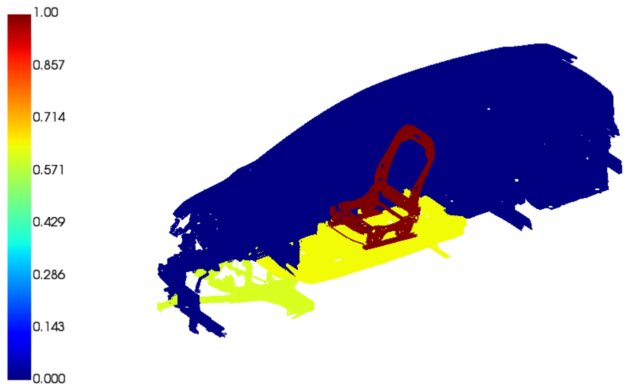

Figure 11: Transfer path “front axis to front seat” for the third octave band centered on 630 Hz. Main contributors are the front axis and the floor.

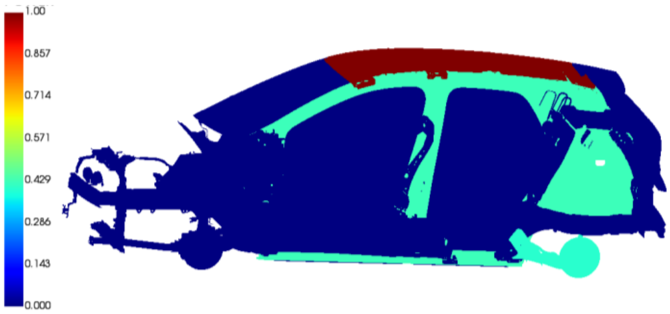

Figure 12: Transfer path “rear axis to roof” for the third octave band centered on 400 Hz. Main contributors are the rear axis and the left and right sides of the BIW structure.

Summary and Conclusions

The article shows two industrial application cases demonstrating the feasibility and value of the Virtual SEA approach. Both, the mid-size model of a car door and the Virtual SEA model of the RENAULT B segment SUV are presented.

It is shown on the car door model that Virtual SEA allows extending validity of low frequency finite element models. Besides, Virtual SEA can be correlated with a low frequency modal frequency response, where matching kinetic energy results were shown.

This article covers structure borne SEA Transfer Path analysis for large scale subsystems. Virtual SEA Transfer Path (TP) capabilities are shown through the CLFs and fluxes of the model with large subsystems.

An investigation of the RENAULT B segment SUV is performed in two steps. First the vibration level of the subsystems due to front and rear axis excitation is shown. This helps to identify critical / high vibration levels in car parts (see Figure 7 and Figure 8). Then, the transfer paths are shown (see Figure 11 and Figure 12)

For the first analyzed transfer path (front axis to front seats), vibrations are transmitted mainly through the floor, which is expected. The transfer path through the firewall is of less importance. Vibration reduction measures should tackle connections from to front axis to the floor and from the floor to the front seat.

The second transfer path being studied in the paper shows that most of the kinetic energy passes from the rear axis, through the left and right side of the BIW structure, to the roof. Connections between the three parts could be investigated for vibration reduction in the roof.

A more detailed substructuring could give further details in the exact path of the pattern, as suggested in the outlook section.

Outlook

Further topics will be treated in up-coming publications are:

- Investigation of the local behavior of the car, with a fine substructuring as shown in Figure 13;

Figure 13: Substructuring for transfer path analysis on a PID level.

- Fluid/Structure modelling and trimmed body simulation for the RENAULT B segment SUV;
- Frequency extrapolation of results without modal results;
- Comparisons with measurements.

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