NUMERICAL ANALYSIS OF THE ACOUSTIC PERFORMANCE OF TREATMENTS USED FOR ENGINE ENCAPSULATIONS

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As the legislation for pass by noise (PBN) has recently become more stringent, car manufacturers face again a challenging task to reach the emitted noise targets (70dB(A)). It is clear that a good acoustic design of the engine bay is required to sufficiently attenuate the powertrain component in the radiated noise. For decades, engine bay treatments have been designed with the main purpose of “just absorbing” the noise radiated by the engine surfaces. The treatments were applied simply on the engine bay’s walls and their NVH targets were expressed in absorption coefficients. This indicates that these parts were meant to influence the transmission path between source and receiver just by dissipating as much as possible the noise impinging on them. Since a few years, though, thermo-acoustic powertrain encapsulation systems, often mounted also directly onto the powertrain, have appeared on the market. Their popularity can be understood by considering the synergy they provide in tackling both a noise issue and providing reduction of CO2 and fuel consumption, thanks to heat storage effects. The design of this type of innovative treatments, though, is somehow less obvious: next to pure acoustic absorption, also the panel transmission of the treatment will affect the noise finally radiated outside of the engine bay. It is therefore not so clear which metric should be used to assess the treatment’s performance. In this article, this important aspect of the design of engine bay treatments is analyzed in detail using as a test case an engine bay mock-up for which external Acoustical Transfer Functions (ATFs) are simulated and measured with and without acoustic treatments. The numerical methodology includes use of Finite Elements Method Adaptive Order (FEM AO) technology in combination with standard poroelastic FEM elements for the treatments. Both engine bay mounted and engine mounted treatments will be addressed.
Keywords: FEM AO, engine noise, sound absorption and insulation, pass-by noise, exterior noise.

1. Introduction

The motivation for writing this paper is driven by the need for simulation of engine bay trim designs to predict their acoustic performance in real life, upfront of actual production. The targets for the acoustic performance concerning exterior vehicle acoustics have recently become more severe, with for instance pass-by noise targets of less than 70 dB(A). In case of combustion engine driven vehicles, the powertrain noise remains an important contributor to such pass-by noise. Simulations are clearly a useful aid to reach the acoustic targets and they help to cut product development cost and time to market, on the premise that they provide accurate results sufficiently fast, allowing for evaluation of multiple design alternatives. This paper will not focus on the reduction of the powertrain structural vibrations, which are forming a main source of the radiated powertrain noise, but on the acoustic transfer paths, amplification and attenuation, of such noise source towards the outside locations of a vehicle where the noise can be heard.

A mock-up of an engine in engine-bay was manufactured in plywood in order to study the acoustic performance of different trim materials. Several configurations were tested before, yet this paper focuses on just 2 of them: one with a no trim applied at all and one with all trims applied. Different from previous setups [1], the trim this time was applied both at body panels and near the engine surface, and new materials [2] were used. A first section briefly illustrates the setup of this model.

In a second section of the paper, the ATFs for an equivalent monopole source model are obtained using a FEMAO (Finite Element Method Adaptive Order) simulation model. The idea here is to check on the accuracy of FEMAO. We can use ATF as a means to quantify the effectiveness of the engine bay trim. Such ATF describe the pressure/volume acceleration relations between a few microphones outside a simplified mock-up structure of an engine bay, and the engine surface vibrations. As obtaining such ATF for the whole engine surface might be tedious, especially in real-life physical experiments, the engine surface vibrations are typically replaced with an equivalent source model, consisting of a limited set of monopoles located close to a rigidly assumed engine surface. The ATF then relate SPL response outside the vehicle to monopole volume acceleration strength of a limited set of such sources. Whereas the sources are assumed independent of the trim design, the ATF clearly depend on it. The ATF were simulated for the two cases mentioned above and overlaid with the experimental results for comparison.

A third part of the paper elaborated a bit further on FEMAO and its performance showing results of different FEM and FEMAO models. Such exercise has been carried out several times over the past years by this paper's authors [3][4], comparing also with BEM (Boundary Element Method) techniques. From the past experiences, it was shown that the FEMAO is very suited for the analysis at hand. For a more in depth explanation on FEMAO, the reader is referred to [5]. Here, we will only repeat briefly a few of its properties, to help understand the differences in performance across several versions of the simulation model, each approaching the modelling of the trim in a different way.

In the last part of the paper, the authors share some concerns/ideas concerning the right metric to use when dealing with truly engine-mounted trim. At least from the simulation side, some suggestions can be made on how to obtain a good metric to express the effectiveness of such engine mounted trim.

2. Test and simulation models setup

Stiff plywood material was used to build a mock-up of an engine bay with engine inside. This is evidently a simplification of a real life automotive case, yet the mock-up represents all relevant features to study the acoustic effectiveness of engine bay trim design: a geometry that is complex and large enough to represent the volume of a real engine bay, apertures that can be opened or closed to study their relative influence to the ATF; a set of available trim material patches and several locations to apply these. Figure 1 shows the construction of the mock-up in the simulation model, replicating the test setup. In the test, the ATF were measured by putting the sound sources outside the mock-up and measuring the responses near microphone locations close to the engine surface. This is the reciprocal result of ATFs representing the SPL response outside the engine bay for unit monopole type of sources near the engine surface, assuming the engine source can be captured with a limited set of such point sources. In the middle picture of Figure 1, we can see the trim panels: several materials, commercialised by Autoneum [2], were used: UltraSilent™ for an under engine shield, ThetaCell™ for the hoodliner and dash, and ThetaFiberCell™ as engine side panels. Except for the engine side panels, all trims were mounted on the engine bay (body) panels. The engine side panels, were loosely suspended in the test setup and freely 'hovering' in the simulation model.

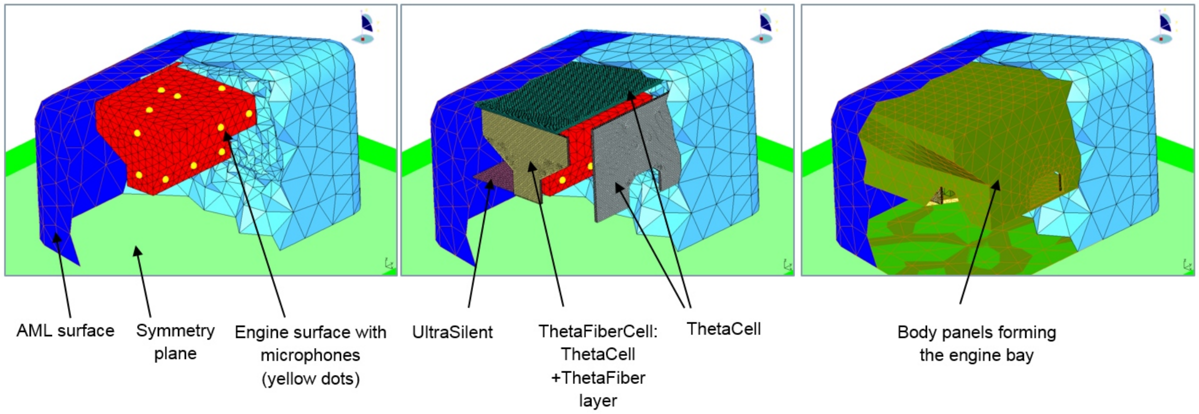

Figure 1. Simulation model construction using FEMAO AML
left: engine with microphones near the engine surface; middle: trim panels;
right: full model including engine bay surfaces

Figure 2 shows the actual experimental model and illustrates where the sources were positioned to measure the ATF. Evidently each ATF was obtained by applying only a single sound source in only one of the points outside the mock-up, to then measure the response in the (flat) microphones attached to the engine surface.

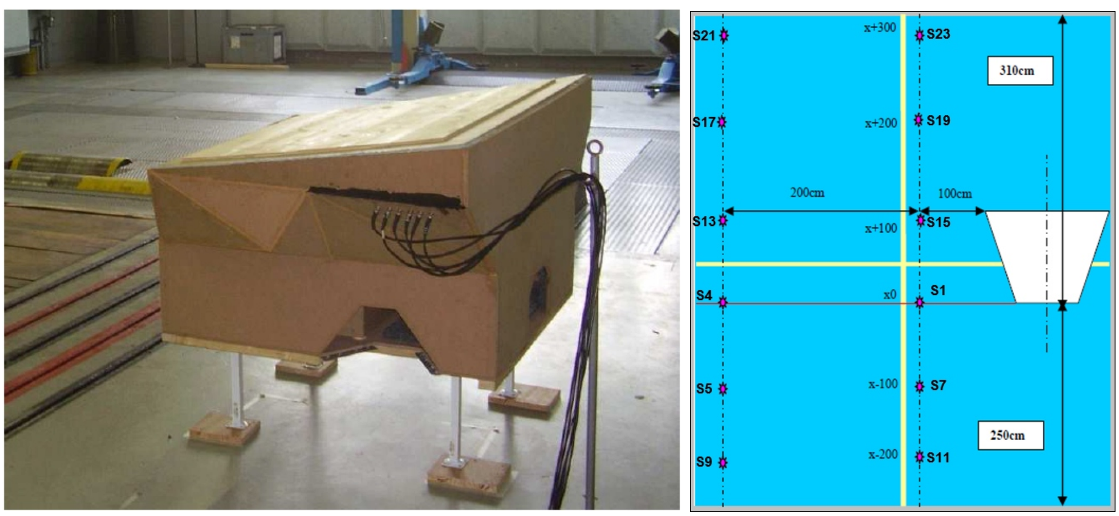

Figure 2. left: the plywood mock-up as simplified representation of an automotive engine / engine bay configuration, including apertures to represent wheel house, tunnel, radiator etc.
right: positions of source locations. All were located 1200 mm above the ground. 4 load cases, S11, S13, S15, S17, were compared with simulated results

3. Accuracy of the FEMAO model

Figure 3 provides a comparison for the microphone results from the tests and simulation model. Clearly the simulation model is able to capture the effect of applying the trim materials. The comparison here is between no trim at all, and all trim applied at once. The results show a match within on average 5dB between test and simulation.

The simulation model used for this case consisted of FEMAO elements for the air volume and for the trim panels applied to the engine bay panels. As we can assume that the vibrations of the panels are relatively small in a real car, the effect of such body mounted trim panels is mainly one of acoustic absorption: sound arriving at the trim panel is absorbed through visco-thermal dissipation mechanisms inside the material, yielding sound absorption. In the simulation model, these trim panels were therefore characterised by a porous Johnson-Champoux-Allard (JCA) material model, with a rigid frame, excluding frame elasticity. The trim panels next to the engine bay were in this test loosely suspended and in the simulation model they were hovering next to the engine and this at 6 cm from the engine side surfaces. In such setup, it is still OK to neglect the elasticity of the panel. However, in actual current trim designs, the trim is often also put very close or truly on top of the engine surface, and attached to it via a few point connections. This in view of enhancing the thermal efficiency of the engine. Such engine mounted trim panel then indeed is a thermo-acoustic shield. In case the side panels would have been attached to the engine side panels, and as in real life the true source of the engine are not monopoles at the surface yet real surface vibrations, it is best to model such engine-mounted panels as fully poro-elastic, capturing both the porous effects and the elasticity of the frame materials. A JCA model with elastic frame was therefore used for the engine side panels, the FEMAO solver can deal with such poro-elastic elements but a fixed lower order shape function basis is used in this case.

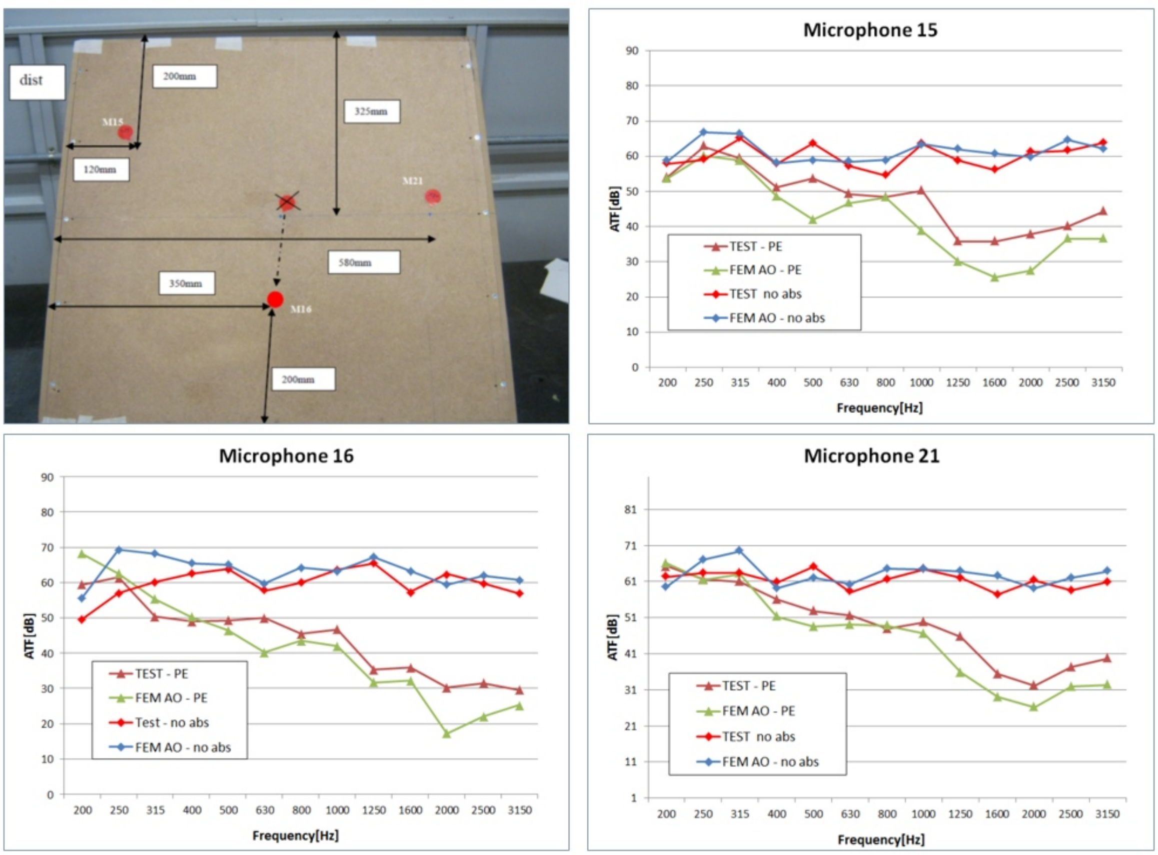

Figure 3. Measured and simulated SPL results in 1/3rd octave bands for mics 15, 16 and 21 at the top of the engine surface. The sound source for these results was put at location S13 (see Figure 2)

4. Effect of poro-elastic modelling and FEM approach on performance

To study the effect of poro-elastic modelling and FEM approach, the SPL results at the engine surface microphones were computed for 12 load cases between 800 and 2800 Hz. Figure 4 shows a good correspondence between the different models.

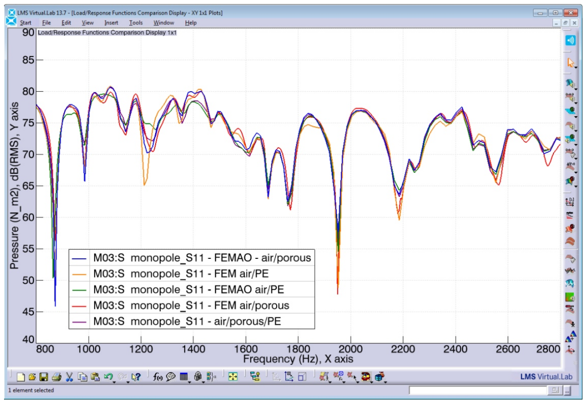

Figure 4. Simulated ATF results for different models
ATF for microphone 3, right engine panel side, at the bottom, for a source located at S11

Table 1. Performance results for different simulation models, differing in poro-elastic modelling approach and using either standard FEM or FEMAO.

Simulations for Engine Mockup **22 microphone points** **Direct (MUMPS) solver** **12 load cases** **151 freqs, 800-2910 Hz** **FEM AO models, min order=2**	**FEM no damping**	**FEM equivalent fluid**	**FEM Poroelastic (Biot, UP)**	**FEM AO no damping**	**FEM AO equivalent fluid**	**FEM AO Poroelastic (Biot, UP)**	**FEM AO Equivalent Fluid + Poroelastic (Biot, UP) optimal mesh**
# nodes total (not incl AML)	**1,446,884**	**1,446,884**	**1,446,884**	**692,393**	**692,393**	**692,393**	**214,390**
# elements air (tetra10)	560,368	560,368	560,368	27,553	27,553	27,553	27,553
# elements treatment volume (hexa8) (air or treatment)	137,247	137,247	137,247	137,247	137,247	137,247	35,297
# nr DOFs total	1,785,753	1,785,753	3,731,220	max 1,604,589	max 1,921,665	max 3,338,013	max 1,266,418
Solver Performance							
# threads/process	8	8	8	8	8	8	8
# process (freqs parallel)	2	2	1	2	2	1	2
peak memory (Gb all process)	**62**	**63**	**32**	**48**	**63**	**29**	**75**
total time (h)	**14.61**	**14.78**	**194.80**	**9.01**	**10.05**	**72.32**	**12.90**

Parabolic elements were used for both air and trim volumes. For standard FEM, the number of elements is driven by the max frequency of interest for air, and by a mix between max frequency and frequency dependent sound speed in the porous trim. Indeed, the lower, frequency dependent sound speed in the trim material requires to model it with sufficiently small elements. In FEMAO, the elements shape function basis is auto-adapted to higher orders (hierarchical Lobatto shape functions), which allows to use less mesh elements in the model. Note however that there is no fixed relation between number of elements and number of DOFs (shape function coefficients) any more when using FEMAO. This is true when FEMAO is used for air or porous elements. For poro-elastic elements, FEMAO currently works as standard FEM, meaning that you need to put in sufficiently small elements to capture the reduced wavelengths in the material. The JCA model used is implemented with UP formulation, requiring 4 DOFs per node, which clearly has an impact on total model size. With exception of poro-elastic elements, FEMAO allows to use larger elements to create your model and let the solver automatically decide, per each element, and per frequency, which order is required depending on local speed of sound, frequency and element size. This obviously reduces the modelling effort: the solver injects the right amount of DOFs at each frequency, yielding a constant accuracy and optimal computation time. All this can be seen in Table 1: a lesser amount of elements is used to model the air in FEMAO. For all models except one, the same fine mesh has been used for the trim volumes. In the last column, the engine bay mounted panels were modelled with larger FEMAO elements, assuming these can be represented as porous only (no elasticity) elements. In this last model, only the engine mounted side panels were modelled with a fine mesh and as fully poro-elastic. This last FEMAO model was chosen as most performant and representative configuration for the model with trim: it is relatively fast, and captures the relevant physics: as mentioned also in section 3, this model includes engine bay mounted panels as porous elements and engine side panels as poro-elastic elements, which would have been required in case these panels would be truly attached to the engine surface.

5. How to quantify the effectiveness of engine mounted trim

For trim panels which are actually mounted on top of the engine surface, one idea could be to keep using a limited set of monopole sources near the engine surface as equivalent source model. But would that be accurate? If yes, then we can measure ATFs as before and use these as metric to quantify effectiveness of the trim configuration design. However, some considerations can be made leading to a slightly adapted metric:

- The real source is the vibrating engine surface. If we put trim directly on the engine and in case we use monopole sources on top of the trim, then the strength of such sources already reflects a sound attenuation achieved by the engine mounted trim. An ATF between such monopole sources and the exterior would mostly capture the effect of the body mounted trims.
- If we put monopole sources at the bare engine surface, where the trim touches the metal engine surface, then the ATF do capture the effect of this trim as well. This actually corresponds to acknowledging that the ATF comprises of a pure absorption and a panel insulation (transmission loss) part as well.
- Now, in case of a bare engine surface, the vibrations can be replaced with a rigid surface, plus a limited set of monopoles. For engine with surface mounted trim, such highly absorbent trim makes that the sound field of only a few monopoles remains very local, and different results are obtained for different choice of monopoles. This is less of a problem in case there is no engine mounted trim. Therefore, to obtain a source representation invariant to the applied trim, best would be to use a very dense distribution of monopole source all over the engine surface. Reformulating this for the reciprocal approach: we can ask for pressure results at all nodes of the engine surface underneath the engine mounted trim. These results can then be surface averaged to get an idea of the average ATF. Whereas this is rather straightforward in simulation, the authors realize this might be more difficult to execute in experiments, as the microphones need to be put at the engine surface under the engine mounted trim and a lot of points would need to be measured. When computing such ATF for a lot of points, the transfer functions are sometimes also referred to as Acoustic Transfer Vectors (ATV). Figure 5, left shows a schematic representation of a case with engine mounted and body mounted trim, plus the ATV derivation. We can refer to these ATF or ATV as acoustic paths.
- Next to the acoustic paths, there are also structural transfer paths. Forces, induced by enforced engine surface vibrations (assumed unaffected by the trim), act on the trim panel causing panel vibrations and sound radiation. Figure 5, on the right, shows how these paths add to the total SPL as well.

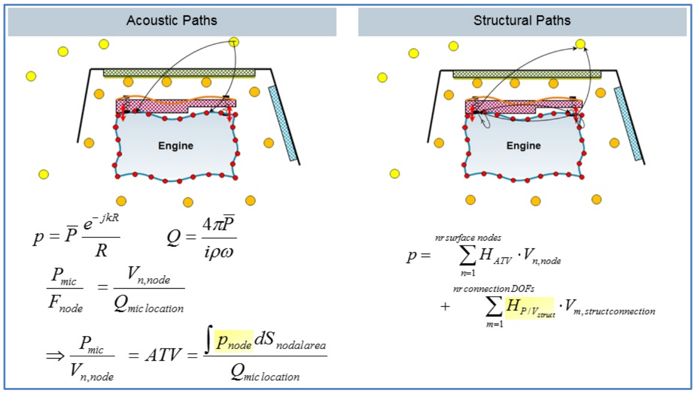

Figure 5. Acoustic and structural path contributions to outside SPL in case of engine mounted trim

Figure 6 shows the pressure results of at the engine surface. For this result, the same trims were used as in the previous sections, yet the engine side panels were moved to <2mm distance from the engine surface, and spring/damper connections were made at 10 points per panel between the outer layer of the trim panel and corresponding points at the engine surface (clamped). One can clearly see the effect of the trim panels in terms of pressure field amplitude and complexity. This spatial complexity underlines again the concern to use the full pressure field and average over the surface to get a metric for trim efficiency.

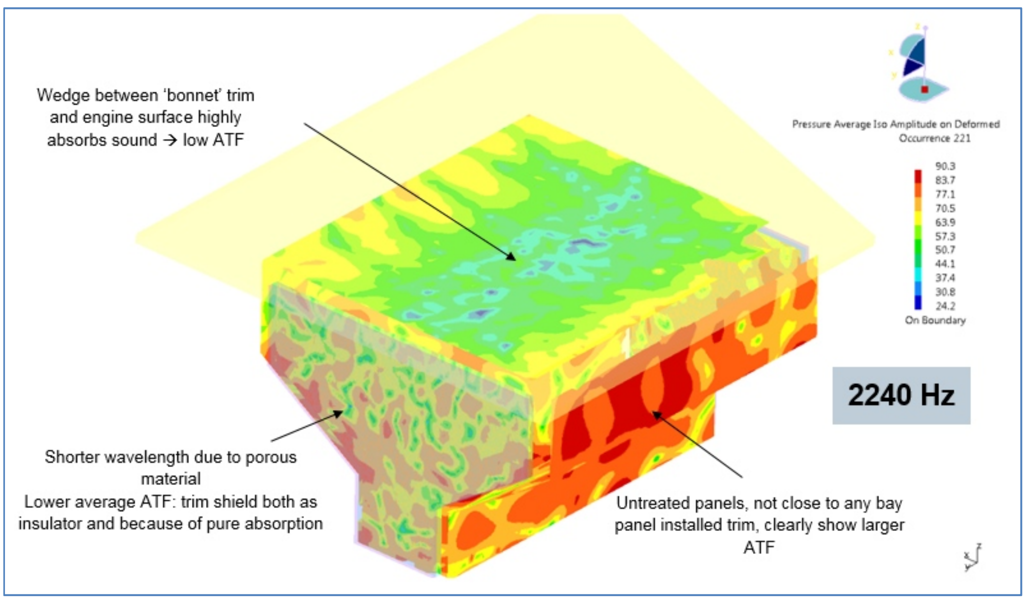

Figure 6. SPL at engine surface for source at S11, 2240 Hz.

Figure 7 shows the averaged pressure results for the different sides of the engine and compares between: all trims applied with engine side panels mounted to the engine surface, and a case with all trims except the engine side panels. This very clearly shows the effect of including the side panels: left and right averaged ATF results show huge difference at higher frequencies, as expected.

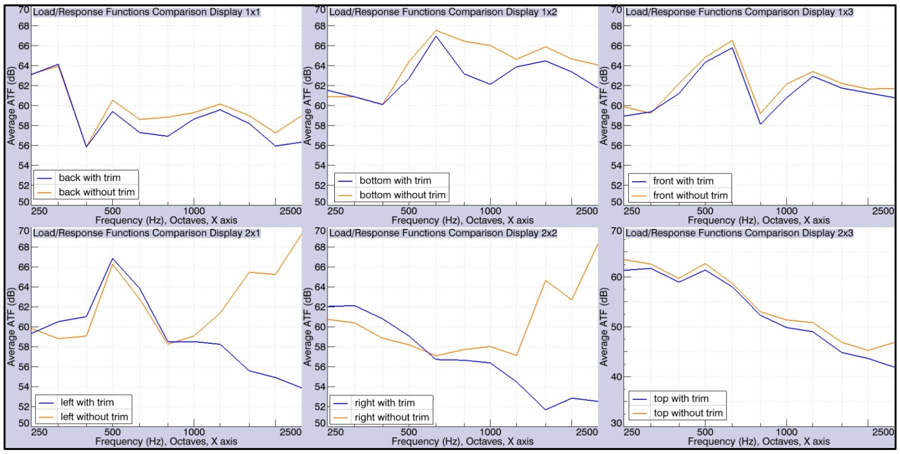

Figure 7. Panel averaged ATF (S11): with (blue) and without (orange) engine mounted trim side panels

Figure 8 illustrates the results for the structural paths (20 out of plane vibrations in the 20 connections were considered).

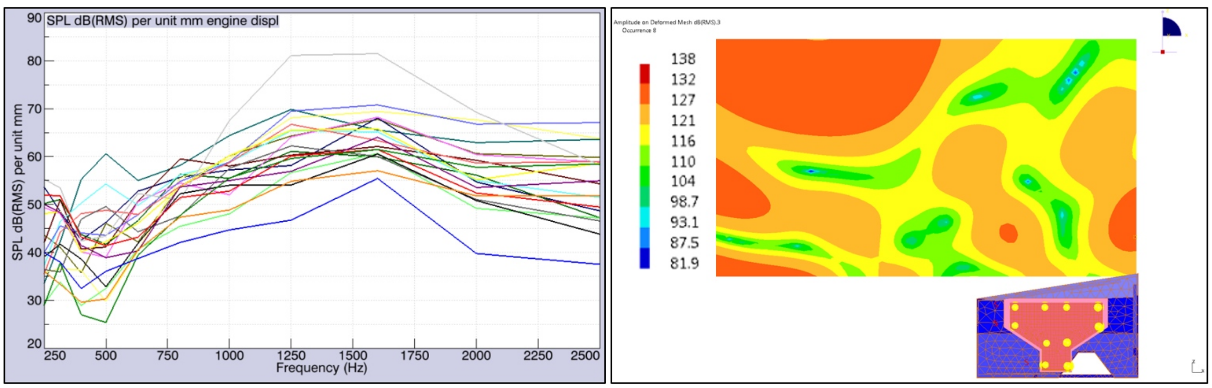

Figure 8. Left: Noise Transfer Functions: SPL in virtual microphone at location S13, per unit mm engine surface vibration in the trim panel connection points.
Right: SPL for a plane of microphones put one meter to the right of the mock-up.
SPL result per unit meter at one of the trim connection points at 1400 Hz

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