Vehicle Acoustic Synthesis Method 2nd Generation: an effective hybrid simulation tool to implement acoustic lightweight strategies

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Abstract
Compared to the 1st generation of the «Vehicle Acoustic Synthesis Method» [1] where the source strength characterization was performed using the classical p-p sound intensity technique, the 2nd generation validates the implementation of the microflown® pressure-particle velocity probes for both sound intensity and transfer functions measurements without moving the probes and without any specific measurement environment.
This new measurement technique allows to speed-up the measurement protocol by four: performing the «Vehicle Acoustic Synthesis Method» on a complete car with all powertrain operating sources then requires only two weeks. Moreover, intensity maps are more refined and unsteady operative conditions like run-ups becomes feasible.
The originality of this approach consists in the simulation of the acoustic package by introducing the insulators in the model as a modification of the power injected in the car compartment (simulated for example by simple Transfer Matrix codes or SEA models or even measured), and as a modification of the cavity transfer functions according to the modification of absorption properties. Using a Ray-Tracing simulation, this method becomes 100 \% numerical on the acoustic treatment for the middle and high frequency range. The «Vehicle Acoustic Synthesis Method» appears to be well fitted in order to demonstrate \textsc{faurecia} lightweight noise reduction strategy, that is based on an optimized trade-off between good insulation and broadband absorption.

Résumé
Comparée à la première génération de la «Vehicle Acoustic Synthesis Method» [1] (Méthode de Synthèse Acoustique Véhicule) où l'identification des sources était menée à l'aide de sondes intensimétriques p-p classiques, la deuxième génération valide l'utilisation de sondes pression-vitesse particulaire p- u microflown® à la fois pour les mesures d'intensité et de fonctions de transfert sans déplacer les capteurs et sans environnement de mesure spécifique.
Cette nouvelle technique de mesure permet de réduire la durée des essais par quatre : il est alors possible de réaliser la «Vehicle Acoustic Synthesis Method» (Méthode de Synthèse Acoustique Véhicule) sur un véhicule complet toutes sources opérationnelles en moins de deux semaines. De plus, les cartographies d'intensité sont plus précises et peuvent être réalisées pour des sources instationnaires comme les montées en régime.
L'originalité de la démarche réside dans la simulation des insonorisants, pour l'isolation par une modification de la puissance injectée dans l'habitacle (simulée par exemple par de simples codes matrices de transfert ou par des modèles SEA ou encore à l'aide de mesures de TL) et pour l'absorption par une modification des fonctions de transferts habitacle suivant les impédances de surface des panneaux. Combinée à la simulation Tirs de Rayons, cette méthode devient 100 % numérique pour les traitements insonorisants en moyennes et hautes fréquences. La «Vehicle Acoustic Synthesis Method» (Méthode de Synthèse Acoustique Véhicule) permet ainsi le déploiement de la stratégie de réduction de poids Faurecia qui est basée sur un compromis optimal entre bonne isolation et absorption large bande.

Optimizing the acoustic package of a vehicle requires the identification of the sources radiating in the interior cavity and the analysis of the way they propagate, taking into account the reflections (with or without absorption) and the diffractions that occur on the different surfaces.
The «Vehicle Acoustic Synthesis Method» (VASM) has been developed in order to separate insulation and absorption problems by calculating the Sound Pressure Level at ear points from the combination of sound
power measurements and acoustic transfer functions panel/ear measured or simulated -with Ray-Tracing methods- for the middle and high frequency range.

The first implementation of the «Vehicle Acoustic Synthesis Method» [1] has been carried out using the classical p-p sound intensity technique on a 4-wheels roller-testbench with all powertrain operating sources. In order to maintain a δp-I index below 10 [2], we have had to mock-up the complete interior with 100 mm thick foam blocks having an absorption and insulation role. The transfer functions had to be measured therefore in a second step with the vehicle in serial conditions, using a monopole source and microphones positioned on the panels.
We have developed the second generation of the "Vehicle Acoustic Synthesis Method" using microflown® intensity

p-**u** probes [3] in order to drastically speed-up the measurement protocol while increasing accuracy and addressing unsteady operating conditions like run-ups.

A first validation of the microflown® p-**u** probes has been realized in a coupled reverberation room/anechoïc room testbench where a complete vehicle has been mounted [4]. In fact, only the dash area has been excited by the airborne artificial source in order to avoid at first high reactive fields due to external sources. This work has proved that it was possible to measure both intensity and transfer functions without moving the probes by using the pressure channel of the p-**u** probe alone for the transfer function measurements.
The results compared to the p-p intensity technique were very good and promising both in term of quality and of efficiency of the protocol.

This paper deals with the second industrial validation of this new technique applied on a Peugeot 1007 but with all powertrain sources operating this time and without any absorbing mock-up, proving that it is now feasible to perform a «Vehicle Acoustic Synthesis Method» on a complete car within two weeks.

Goals and Basis of the Study

Besides the development of the new methodology, the goal of the study has been to design and optimize the acoustic package of a Peugeot 1007 for the floor and dash area reaching the challenging target of a weight reduction of 10% while maintaining the acoustic performance and a cost reduction of 5%.

This can be summarized by the lightweight strategy called «**F**aurecia **A**coustic **C**oncept **T**riangle» (FACT) [1, 5]

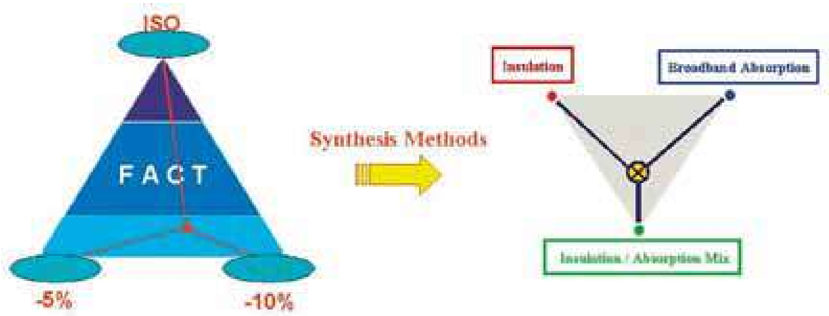

Fig. 1 : Faurecia Acoustic Concept Truangle (FACT)

This lightweight strategy is represented by two triangles (cf. figure 1), the first one represents the target imposed by the carmaker summarized in terms of acoustics / weight / price, the second one represents the related acoustic package summarized in terms of a barycenter between the acoustic concepts: classical insulation, broadband absorption and mixed absorption/insulation. The link between the triangles is performed by the «Synthesis Methods: Vehicle Acoustic Synthesis Method», showing the necessity of such hybrid simulation tools.

Synthesis Methods with Microflown

The first task of the «Vehicle Acoustic Synthesis Method» consists in determining and digitizing the 3D measurement mesh of the interior skin of the vehicle. This has been carried out with an ultrasonic digitizer leading to 654 elements (radiating facets measuring 23 cm2 in average) defined by 1331 geometric nodes.
The center points of the elements needed for the channel tables of Ideas-Test are computed with MATLAB to ensure that the measurements points are «inside» the car and that the numbering follows the elements numbers.

The originality of this way of working is that this mesh will live throughout the study being used to realize the measurements, to represent the 3D power, transfer functions and projected SPL maps, but also to build the Ray-Tracing model...

As requested by the carmaker, the measurement conditions are :

- 2nd gear, 4000 rpm
- 4th gear, 100 km/h
- 3rd gear, WOT run-up (1200 rpm - 5000 rpm)

The measurement protocol is the following: **a)** the p-**u** probes are positioned on the center points of the elements; **b)** vehicle running / engine «on», the intensity measurements are done for the two constant speed road conditions; **c)** vehicle stopped / engine «off» but volume velocity source «on», the transfer functions are measured; then **d)** vehicle running / engine «on», the intensity run-ups measurements are realized.

The calibration procedure of the p-**u** probes, which is rather delicate, can be found in reference [4].

Sound Power Measurements

The VASM is based on the hypothesis that each sub-area *S(j)*, having a global radiated power *W(j)*, can be modeled by *n(j)* fictitious uncorrelated substitution monopoles, the total mean squared volume velocity $Q^2_{eq}(j)$ being determined with $ka<<1$ by [6,7] :

$$W(j) = \frac{\rho\omega^2}{4\pi c} Q_{eq}^2(j) C(j) \qquad (1)$$

With *C(j)* weighting factor representing the influence of the surroundings on the power radiated by the equivalent monopoles (compared to that in free field):

- $C(j) = 2$ for a hard reflecting wall

- $C(j) \approx 2 - \alpha$ for an absorbing surface where α is a diffuse field absorption coefficient (plane wave approximation.

Figure 2 shows the measurement conditions and how the p-**u** probes are positioned.

Fig. 2 : Sound power measurement protocol

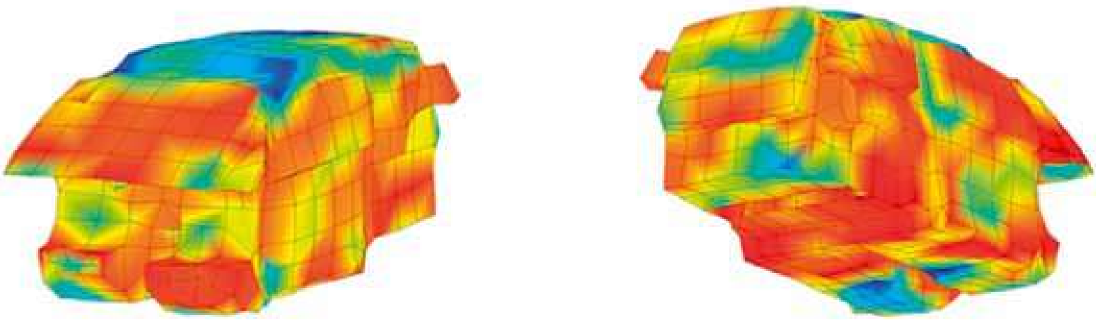

Fig. 3 : Sound power measurement 3D map at 1 000 Hz, 4th gear 100 km/h

Figure 3 shows an example of radiated sound power 3D map at 1 000 Hz, 4th gear 100 km/h.

Transfer Functions Measurements

The transfer functions have been measured using the reciprocity principle by positioning a constant volume velocity source at two receiver points: Driver's ear position and Rear right passenger's ear position. The reciprocity relation is given by equation 2 :

$$\left[\frac{p_r^2}{Q_i^2}\right] = \left[\frac{p_i'^2}{Q_r'^2}\right] \qquad (2)$$

Due to the point to point intensity measurement technique, transfer functions are not averaged any more compared to [1]. The discretization is homogeneous indeed in the new protocol between sound power and transfer function measurements and is actually quite refined.

Remark : as we are working with power, it would not make much difference to consider energetic transfer functions $p_i'^2/W_r'$, *nevertheless it would be then compulsory to determine the injected power of the monopole source in the interior cavity which can be measured with a two or three microphones «intensity like» measurement technique or with what we are currently testing with a p-**u** probe in the source nozzle.*

Figure 4 shows the measurement conditions and how the constant volume velocity source is positioned.

Fig. 4 : Transfer function measurement protocol

Figure 5 shows an example of a transfer function 3D map not normalized to the mean squared volume velocity

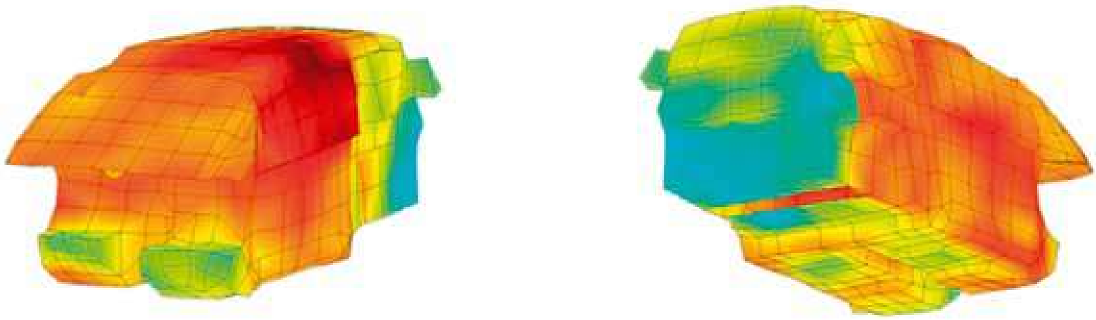

Fig. 5 : Transfer function 3DF map at 1 000 Hz, driver's position

$Q_r^{'2}$ of the source at 1 000 Hz (autospectrum), the volume velocity source being positioned at the driver's ear position.

SPL Calculation at Ear Receiver Points

The Sound Pressure Level resulting from the contributions of the *m* sub-areas is given by equation 3 :

$$p_r^2 = \sum_{j=1}^{m} H_{j,r}^2 Q_{eq}^2(j) \qquad (3)$$

As we can see on figures 6 and 7, the correlation between the simulation (or better said the recomposition obtained with equation 3 and the experiments is quite satisfactory for a full vehicle model, the level being predicted within 3 dB(A) on the third octave spectrums between 500 Hz and 3 150 Hz and below 2 dB(A) on the overall Sound Pressure Level calculated between 500 Hz and 5 000 Hz for both positions in the interior cavity and for both road conditions.

We begin to observe discrepancies upper than 4 kHz due to what has been identified by the probes manufacturer as a «string resonance» of the hot wires of the **u** channel,

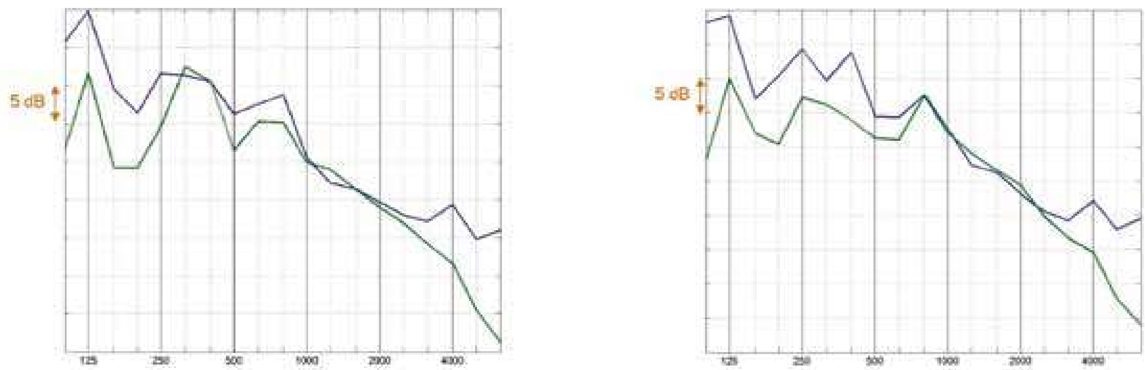

Fig.6 : SPL[dB(A)] recomposition VASM versus measurement, 4th gear 100km/h

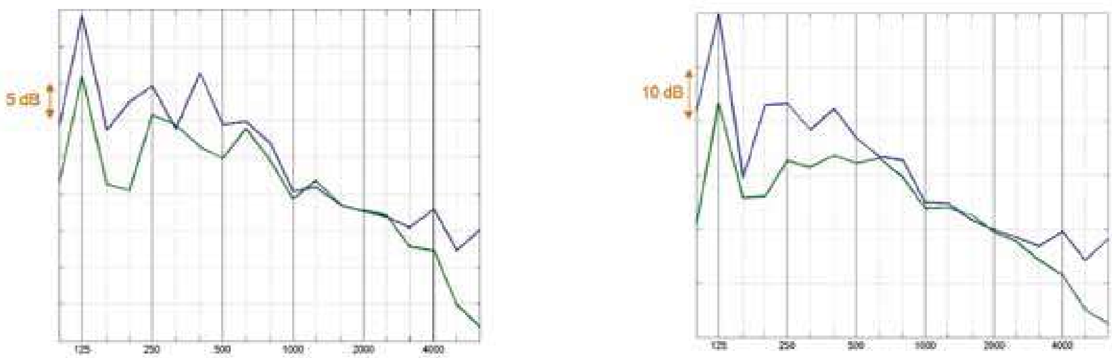

Fig. 7 : SPL[dB(A)] recomposition VASM versus measurement, 2nd gear 100km/h

linked to an insufficient tension during the mounting on some of our probes (a dip in the **u** calibration curve may be observed). This problem is already fixed on our new replacement probes.

Nevertheless, this new «one shot» protocol begins to give reasonable results upper than 250 Hz, which is logical regarding the uncorrelated monopoles hypothesis, the measurements being carried out without any absorbing foam blocks inside the car. *This i a very important result validating that the p-**u** probes are not sensitive to the δp-I index (which can easily exceed 15 dB in a vehicle with all powertrain operating sources), the **u** channel being directly measured.*
In fact, on classical p-p probes, the velocity is indirectly obtained following the Euler equation from the pressure gradient between the microphones and thus the intensity measurements are very sensitive to phase mismatch, especially in high reactive fields where the δp-I index is quite high [2].

Moreover, the p-**u** probes can then be positioned directly on the surface (with a thin closed foam decoupler), meaning that we do not need to retro-propagate anymore (in the holography sense) in order to precisely localize the sources... One has nevertheless to time average enough in order to filter the reactive «circulating» energy of the near field evanescent waves.

Remark: further investigation has to be carried out considering the complete power balance in the car in order to determine if the net power resulting from the radiated and absorbed energy of one facet is correctly determined without additional absorption blocks in the cavity.

We obtain this way a full vehicle synthesis model allowing to analyse the contribution of each panel both in terms of sound power radiated but also in terms of Sound Pressure Level at various ear points in the interior cavity. Moreover, it is then possible to «project» these SPL contributions on the panels (detailed or averaged) and get an original and powerful 3D representation of what the driver (or the rear right passenger in our case) actually hears (cf. figure 8)...

Acoustic Package Optimization

The acoustic package optimization phase aims at determining the barycenter of the acoustic concepts -balancing insulation and absorption- that better fits the target defined in terms of acoustic, weight and price (cf. section 2). This optimization procedure relies on the poro-elastic simulation of the acoustic package using simple transfer matrix codes like [8,9] or SEA models or even relies on measurements in coupled reverberating rooms for example, which is the case for the cockpit area here.
The resulting Transmission Loss and diffuse field absorption coefficient or impedance are introduced as a modification of the power injected in the passenger compartment (as ΔTL) and as a modification of the cavity transfer functions according to the modification of the absorption properties in the Ray-Tracing model [1]. The resulting optimized SPL is then calculated at various ear points and compared to the target.
An improvement of this optimization procedure has been introduced in this study by computing all poro-elastic data following the thickness cartographies of the insulators (instead of local average values), leading to a much more precise simulation (cf. figure 9)

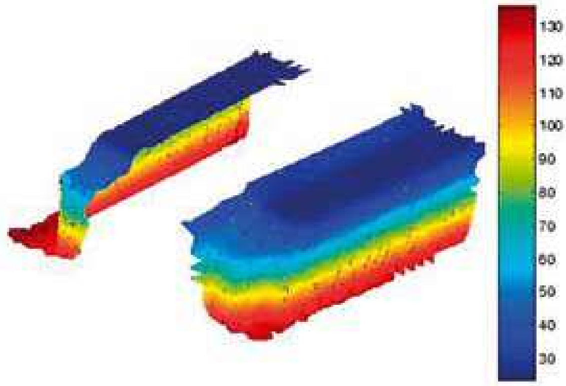

Fig. 9 : Tunnel insulator thickness cartography example

We will now analyse the quality of the poro-elastic simulation tools, we are using in the optimization phase, compared to measurements.

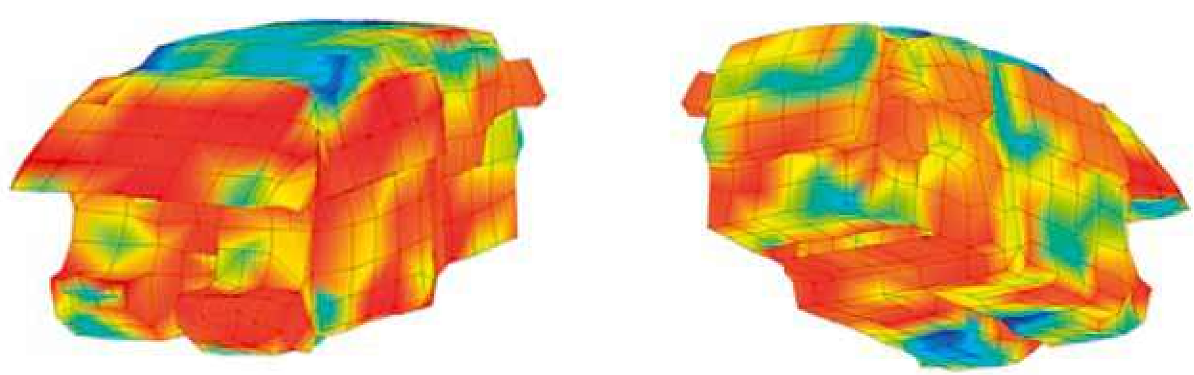

Fig. 8 : SPL recomposition 3D ma pat 1 000 Hz, 4th gear 100 km/h ; driver's position

Absorption Coefficients: normal and random incidence

The problem of the introduction of absorbents in SEA or Ray-Tracing models is not recent and the solution known for a long time: one has to use large reverberation rooms (about 200 m^3) and large surfaces of materials (12 m^2) to determine the diffuse field absorption coefficient of flat materials or parts using the reverberation time technique (cf. Norm ISO 354).

For practical reasons, small reverberant rooms have been developed, the most famous one being the Alpha Cabin (cf. figure 10) widespread in the automotive industry (a common measurement procedure has been written by Renault and PSA-Peugeot Citroën. : D 49-1977). We can find many articles in the literature

dealing with the now well-known overestimation of the Alpha Cabin (or small reverberant rooms), particularly in the low and middle frequency range compared to large reverberant rooms [10,11,12,13].

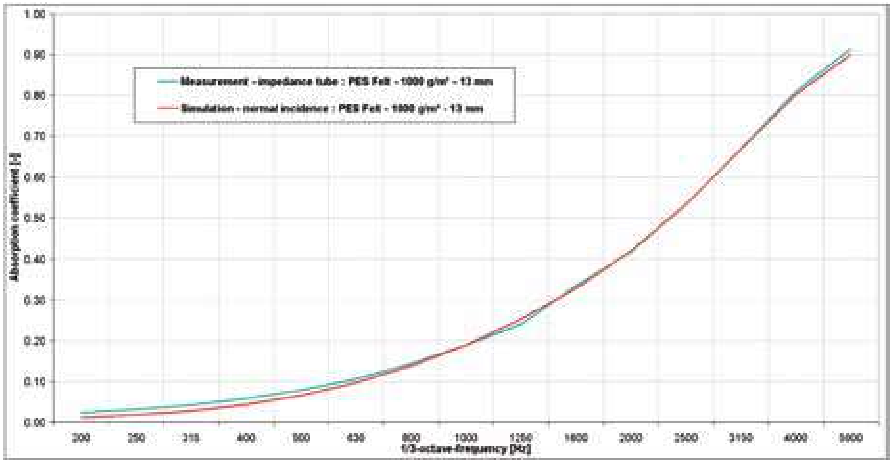

Fig. 10 : small reverberant room : Alpha Cabin

In fact, these small reverberant rooms are used below the «Schroeder frequency» defining the frequency f_c under which the field is not diffuse anymore [6] :

$$f_c = 2000\sqrt{\frac{T}{V}} \tag{4}$$

For the Alpha Cabin, with the maximum reverberation time being $T=2.5\ s$ and the volume being $V=6,44\ m^3$, equation 4 yields f_c = 1 246 Hz. The Alpha Cabin absorption coefficient measurements are nevertheless usually considered as valid between 400 Hz and 10 000 Hz…

As mentioned in [11,12] another important factor is the diffraction effect due to the finite sizes of the samples. The perimeter to surface ratio E of the sample has to be low in order to minimize the diffraction. It is the case for large reverberant rooms (E=1, 2 m^{-1} for a 12 m^2 sample) but not for the Alpha Cabin (E=3 ,7 m^{-1}} for a 1,2 m^2 sample).

Normal Incidence Absorption Coefficient

The Biot parameters of the absorbents and insulators of the car are determined using an inverse technique based on impedance tube measurements, direct airflow resistivity measurements and direct Complex Young modulus measurements.

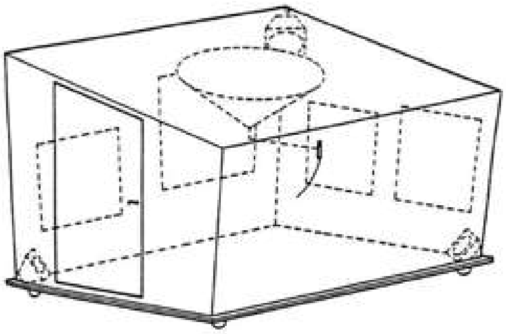

Fig. 11 : Normal incidence absorption coefficient : measurement versus simulation

Figure 11 shows the very good correlation between the impedance tube measurement of a 13 mm thick PES felt and the simulation using Maine 3A V1.3 [9].

The same work has been done for a 4 mm PES carpet and a 20 mm Cotton felt leading to very good correlation for normal incidence as well.

Random Incidence Absorption Coefficient

The three materials mentioned above have been successively measured in Alpha Cabin with a 1,2 m^2 flat sample (cf. figure 10) and in the large reverberant emitting room 2 of the Faurecia's Center of Acoustic Technology with a 12 m^2 flat sample (cf. figure 12).

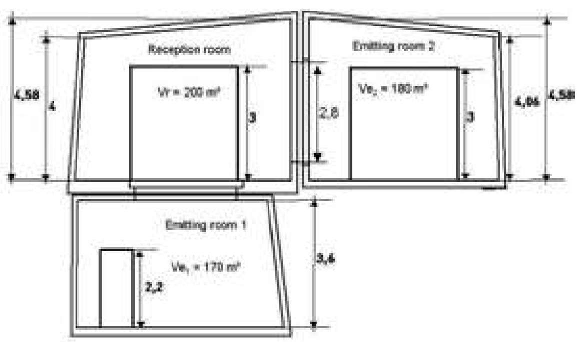

Fig. 12 : Large reverberant rooms : coupled reverberating rooms

The measurements are compared with the simulations obtained with Maine 3A V1.3 by integrating on the solid angle (the Biot parameters being unchanged compared to

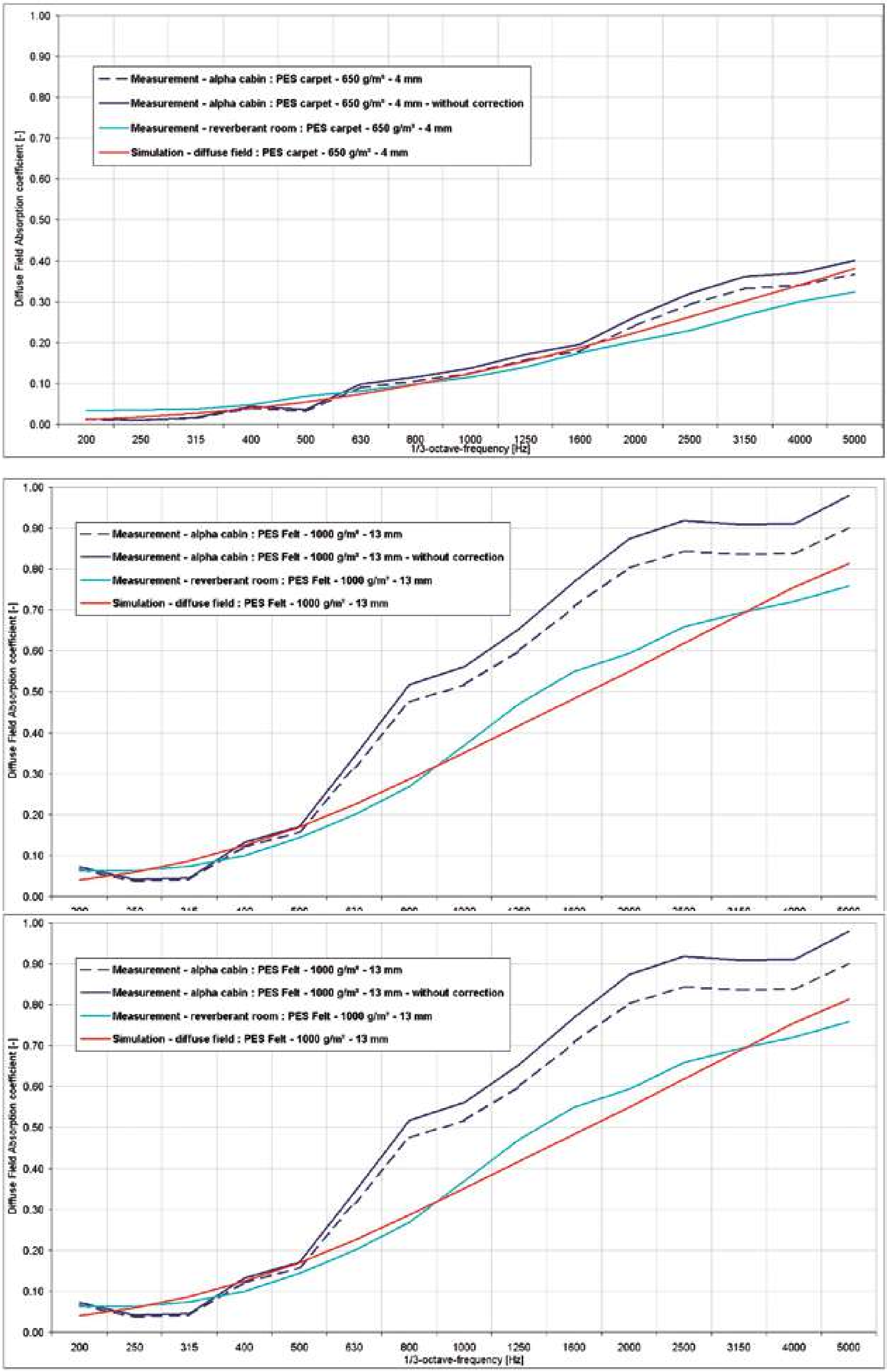

Fig. 13 à 15 : Random incidence absorption coefficient : measurement versus simulation

the normal incidence simulations). Figures 13, 14 et 15 show a discrepancy between the Alpha Cabin and the large reverberant room growing towards the low frequency with the performance of the materials (as already stated in [10]), *whereas the diffuse field simulation shows an excellent correlation with the large reverberant room measurements in all cases*.

This proves that the constant correction coefficient *C*=0,92 originally «designed» with the Alpha Cabin as a correction in order to fit the results with large reverberant rooms does not address the physics [12]… In fact, when absorption (or better said the equivalent absorption area) is growing, the field becomes less and less diffuse and therefore more and more sensitive to the boundary conditions (explaining why Alpha Cabins have to be compared with one another).

The Ray-Tracing method gives a good representation of those phenomena. If the beams meet high absorbing surfaces they will vanish after only a few reflections: there is then no diffusivity anymore.

These results validate this poro-elastic simulation chain which allows to get reliable diffuse field absorption coefficients from impedance tube measurements…

Transmission Loss

In the optimization phase, a MAINE 3A model is built for each material construction in order to get both diffuse field absorption coefficient and Transmission Loss.

Figure 16 presents the same Cotton felt 1 200 g/m^2 20 mm thick as above, complexed with a 3,5 kg/m^2 heavy layer on top of 0,7 m x 1 m steel plate 0,8 mm thick, the Transmission Loss being measured in our horizontal coupled reverberating rooms (cf. figure 12) first using sound pressure and afterwards with an absorbing environment in the reception room using intensity.

The Transmission Loss obtained from the pressure measurement is not reliable any more above 4 000 H zdue to a much too low signal to noise ratio in the reception room. One can also see above 1000 Hz, the first effects of flanking paths contributions, which will be even more critical for heavy layers weighting more than 5 kg/m^2.

The correlation between the measurement and the simulation is very good especially with MAINE 3A V1.3 [9], which implements the spatial windowing technique [14,15], allowing to compute with 1 or 2 spatial windows.

We observe experimentally in figure 13 that 2 spatial windows overestimate the Transmission Loss and that 1 spatial window gives excellent result (as stated in [15]) particularly below the respiration frequency where SEA underestimates systematically (compared to the intensity measurement considered as the reference).

This is not surprising because the spatial windowing is frequency dependent, which is not the case of SEA, where the windowing effect is taken into account in the non-resonant coupling loss factor η_{ij} *(dominant here) by the surface S (window area) to volume V_1 (emitting room) ratio}:*

$$\eta_{ij} = \frac{c_o S}{4\omega V_1}\tau_{12} \qquad (5)$$

In equation 5, τ_{12} is the Power Transmission Coefficient of the infinite poro-elastic multi-layer treatment. This correction is sufficient above 315 Hz but not below, giving the advantage to the spatial windowing technique.

Ray-Tracing Simulation

The ray-tracing codes that we have used to calculate the transfer functions of the VASM are EBINAUR and ICARE developed by the CSTB (Centre Scientifique et Technique du Bâtiment) [16,17].

EBINAUR is based on a weighted gaussian's beam formulation, takes into account reflection on plane

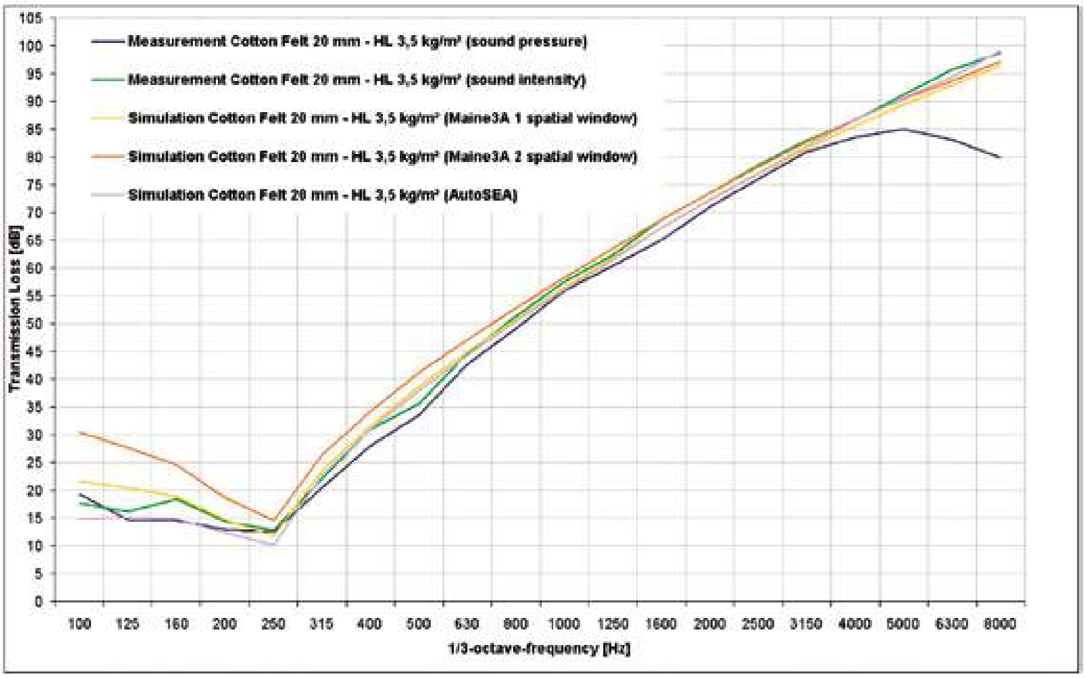

Fig. 16 : Transmission Loss : measurement versus simulation

surfaces only and, besides geometrical calculation, uses statistical prolongation for the higher order of reflections and also takes into account diffusion.
ICARE, a more recent code, is based on an adaptative beam-tracing, each beam being constituted by three rays, takes into account reflection on both plane and curved surfaces, and implements diffraction by edges using the Uniformed Theory of Diffraction (UTD). The solver of \textsc{icare} is purely deterministic (geometrical calculation), which implies to check the convergence and to determine if the absorption is sufficient to avoid statistical prolongations: it seems to be always the case in automotive applications.

The Ray-Tracing code ICARE has been preferred to EBINAUR in this study for numerical flexibility reasons.
The first reason is the necessity to be able to separate the geometrical computation which is highly CPU time demanding from the acoustical computation (changing the material properties for example) which is much quicker.
The second reason is the limitation of EBINAUR interfaced with OPTIMA (internal PSA- Peugeot Citroën pre-post) to compute more than 32 receivers at a time, the transfer functions simulation being done using the reciprocity principle (like the measurements).

Indeed, the 3D digitized mesh of the car has been imported to build the model and therefore requires to compute 654 transfer functions (as many as the number of facets). The convergence of the simulation has been reached at a depth of 5 reflections (the depth 0 meaning the direct field).
The Ray-Tracing model has been used in combination with the poro-elastic simulation in order to find the optimized trade-off between good insulation and broadband absorption, either implemented on the same part leading to use Light-Weight Concept four or three layers constructions (patented) or on other parts leading to use bi-permeable concepts.
By implementing these solutions, we have reached the weight target which was a weight reduction of 10\% (while maintaining the acoustic performance).

Vehicle Validation: Final Acoustic Results

The figure 18 and table 1 show that not only we have reduced the weight of the insulators by 10\% but also we have slightly improved the acoustic performance in the middle and high frequency range, particularly in terms of Articulation Index, which leads to a more pleasant sound.

4th gear front	-1,8dB(A)	+ 5,1%IA
4th gear rear	- 2,4dB(A)	+ 3,5%IA
2nd gear front	- 0,8dB(A)	+ 3,1%IA
2nd gear rear	+ 0,1dB(A)	- 0,1%IA

Table 1 : Final results: improvement of the Articulation Index and of the overall SPL dB(A)(500-5000Hz)

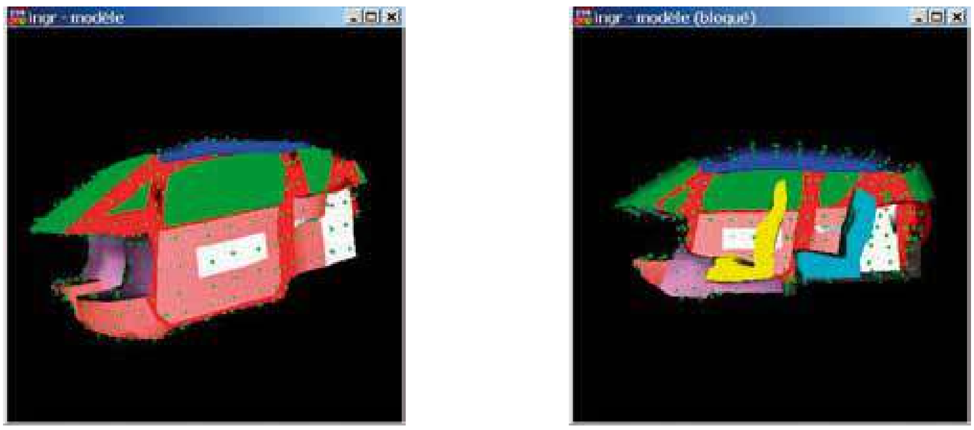

Fig. 17 : Icare Ray-Tracing Model

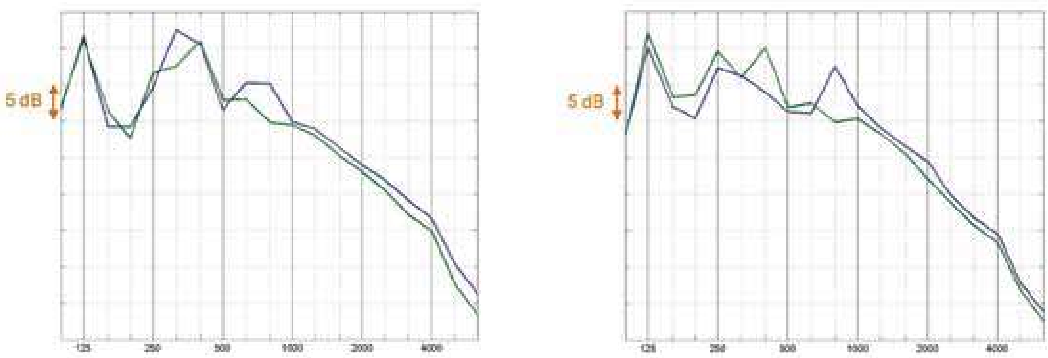

Fig. 18 : SPL [dB(A)] measurement comparison serial optimized, 4th gear 100 km/h

Conclusion

This study has been the first application of the microflown® probes in a complete «Vehicle Acoustic Synthesis Method» with all powertrain operating sources and without any specific measurement environment, allowing to speed-up our measurement protocol by four and making therefore feasible to perform a VASM on a complete car within two weeks.

The very good simulation-experiment results have led to the design of an optimized acoustic package reaching the challenging target of a weight reduction of 10\%, while maintaining the acoustic performance and the cost.
An important automatization effort has been done leading to a comprehensive MATLAB program supporting the modeling effort from the meshing, to the measurements and to the poro-elastic and Ray-Tracing simulation up to 3D colormaps post-treatments.

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