Automotive cabin characterization by acoustic modal analysis

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ABSTRACT

The interior sound perceived in an automotive cabin is a very important attribute in vehicle engineering. Therefore, there is an industrial interest to be able to predict the interior acoustic behaviour by means of accurate simulation models. In order to understand the modelling challenges and improve the modelling know-how, experimental methods in which an acoustic characterization of the cabin is performed based on measurements play an important role.

When performing an interior acoustic study, it is important to relate the acoustics responses to the intrinsic system behaviour of cabin cavity. This can be done by means of acoustic modal analysis, i.e. modal parameter estimation methods decompose the system behaviour into a set of individual resonance phenomena, each characterised by a resonance frequency, damping ratio, and mode shape.

This paper will discuss the equipment requirements and advanced modal parameter estimation techniques. Specific acoustic modal analysis challenges are the high modal damping ratios resulting in highly overlapping modes with complex mode shapes and the use of a large number of references (sound sources) distributed around the cabin to get a sufficiently homogeneous sound field. The performance of Maximum Likelihood Estimation will be examined and compared to more traditional approaches.

Keywords: Acoustic Modal Analysis, Interior Car Sound, Sound Source

1. INTRODUCTION

When performing an interior acoustic study, it is important to relate the acoustics responses to the intrinsic system behaviour of cabin cavity. This can be done by means of acoustic modal analysis, i.e. modal parameter estimation methods decompose the system behaviour into a set of individual resonance phenomena, each characterised by a resonance frequency, damping ratio, participation factor and mode shape. The experimental data set to derive this model from consists of a set of Frequency Response Functions (FRFs) between a set of reference (i.e. acoustic source input) degrees of freedom and all response (i.e. microphone output) degrees of freedom.

This paper will discuss the equipment requirements (with emphasis on the sound sources) and advanced modal parameter estimation techniques. Specific acoustic modal analysis challenges are the high modal damping ratios resulting in highly overlapping modes with complex mode shapes and the use of a large number of references (sound sources) distributed around the cabin to get a sufficiently homogeneous sound field.

Recently, Yoshimura and co-workers published very interesting studies on the experimental challenges related to the application of acoustic modal analysis to an automotive cabin. They also proposed a non-linear least squares (NLS) method that is better suited to process FRF data with many references (1, 2).

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2. THEORY OF ACOUSTIC MODAL ANALYSIS

Consider a three-dimensional closed acoustic system with rigid or finite impedance but non-vibrating boundaries. The governing equation of this system, excited by a point monopole of volume velocity at $\vec{r}_0$ can be written in the form (3):

$$\nabla^2 p(\vec{r},t) - \frac{1}{c^2}\ddot{p}(\vec{r},t) = -\rho\dot{q}\delta(\vec{r}-\vec{r}_0)$$

where p is the sound pressure, which is a function of space $\vec{r}$ and time t; c is the speed of sound; ρ is the density of the medium; and $\dot{q}$ is the volume velocity.

Assuming now that a number of point monopoles of known volume velocity are placed in the cavity and the sound pressure across the volume is sampled at an appropriate number of points, it can be shown that the continuous wave equation can then be substituted by its discrete equivalent:

$$A\ddot{p} + B\dot{p} + Cp = \dot{q}$$

No direct physical meaning can be attributed to the matrices A, B, C, but the discrete governing equation above is equivalent to the discrete mechanical equations of motions, with A, B, C in the role of mass, damping, and stiffness matrices; p in the role of displacement; and $\dot{q}$ in the role of force. In view of this equivalence between acoustics and structural dynamics, it can be concluded that the classical modal parameter estimation approach can be followed also in the acoustic modal analysis case. An interesting expansion towards coupled vibro-acoustic modal analysis is provided in (4).

3. NEW SOUND SOURCE

Sources used for acoustic modal analysis have been developed to enable Transfer Path Analysis and Airborne Source Quantification. These sources do have either no effect on the acoustic field (LMS Qsources Miniature Source) or produce very high low-frequency noise levels but where the product dimensions are similar to a human torso (LMS Qsources Low Mid Frequency Source). The latter will result in high quality acoustic and vibro-acoustic FRFs with the assumption that occupants are present in the vehicle. This is an important feature for Transfer Path Analysis of interior noise. The Miniature Source has, due to its miniature size, no body diffraction and emits the noise as a monopole source up to several kHz. In high-end class vehicles where local damping is also very high, the noise level, necessary for acoustic FRFs in the cavity is at its limits. Therefore the need exists for a dedicated source that is compact, omnidirectional and capable of generating high noise levels in the low frequency range.

A unique monopole sound source (Figure 1) has been developed to acquire acoustic and vibro-acoustic FRFs in an accurate way without disturbing the acoustic behaviour of the passenger cavity. The main design drivers were high noise levels at low frequencies, omnidirectional behaviour and real-time sound source strength measurement. This has been accomplished by using two high performing magnetic drives with a patent pending voice coil stroke assembled in a rigid body.

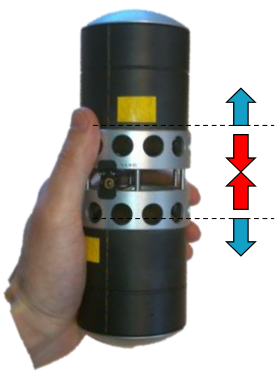

Figure 1 – Dedicated monopole low-frequency sound source.

The FRF database for modal analysis should be as accurate as possible. One of the elements is an omnidirectional sound source. This allows an accurate real-time sound source strength measurement. Figure 2 clearly shows that the emitted sound pressure does not vary more than 1.5 dB at 630 Hz. A comparison of acoustic FRFs measured in the passenger cavity of a compact class vehicle with an LMS Qsources Miniature Source is shown in Figure 3. The FRF are visibly identical although the size of both sources varies significantly. The LMS Qsources Miniature Source measures only 71 mm x Ø22 mm whereas the new source is 200 mm x Ø70 mm. The comparison shows that there are no relevant directional effects that deteriorate the FRF measured with the larger sized source.

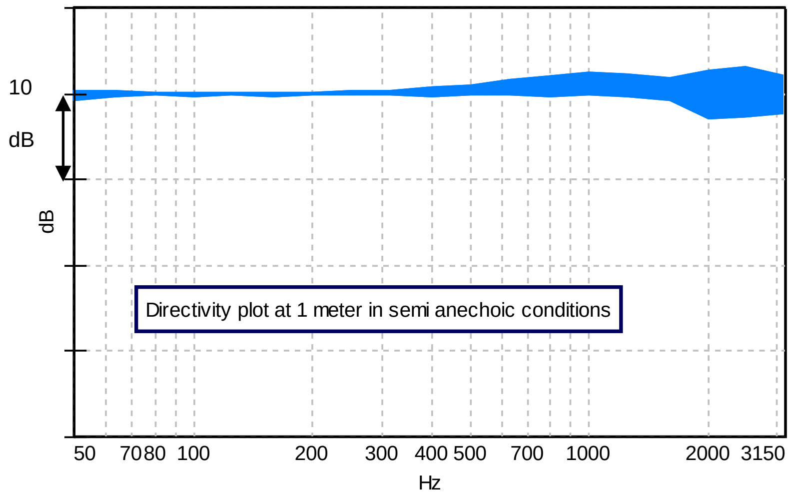

Figure 2 – Directivity plot. Y-ticks at 10 dB distance.

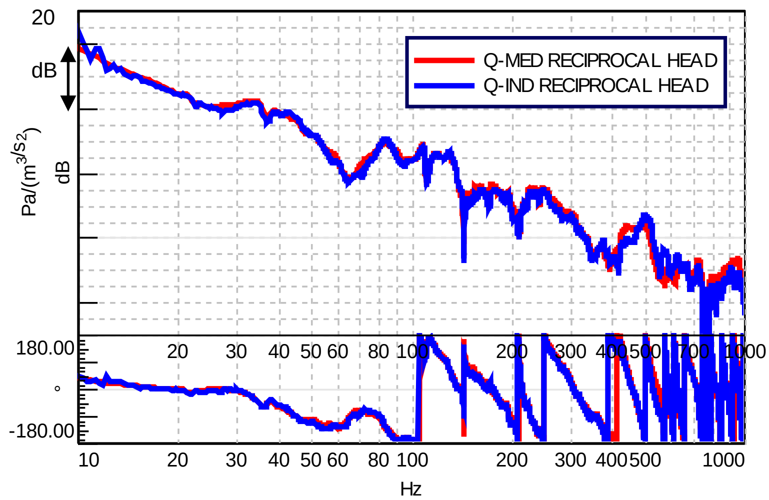

Figure 3 – In-vehicle acoustic FRF measured with 2 different sound sources.

The noise level of the acoustic source should make FRF measurements possible between all measurement points in the cavity even when the trunk is being included in the analysis.

Figure 4 shows a typical FRF where the acoustic source is placed in the trunk and a response has been measured in the front row foot area. The coherence is shown in the lower plot for two measurements. The coherence between input and output is close to 100% from 10 Hz on. Repetitive measurements also result in identical FRFs. At higher frequencies some coherence drops are caused by anti-resonances in the measured FRF.

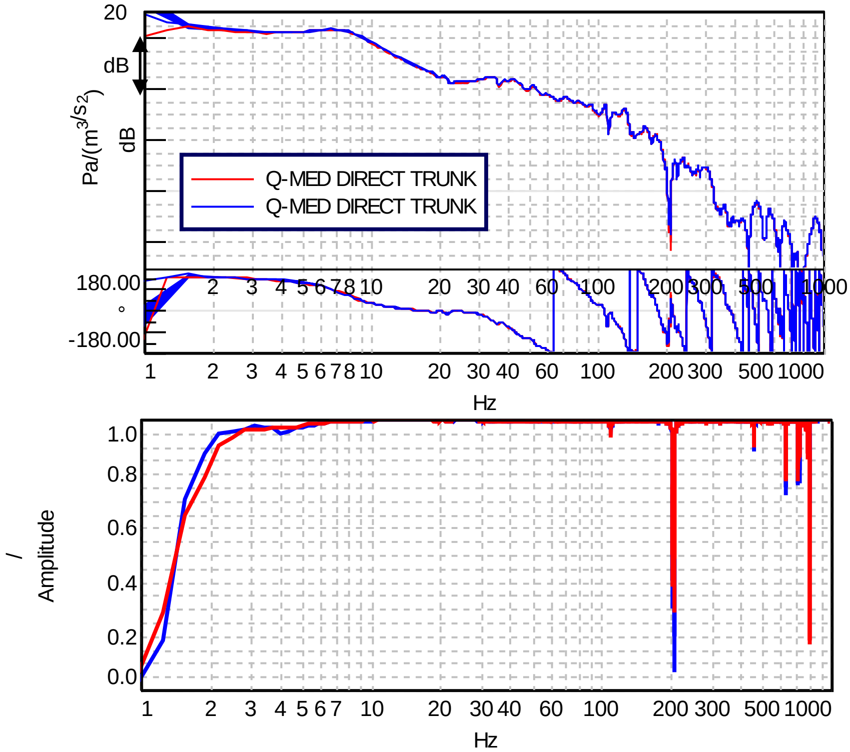

Figure 4 – Repeatability of the FRF measurements.

The measurements have been carried out in semi-anechoic test laboratories at the LMS Engineering Services facilities in Belgium. Figure 5 shows a comparison between a microphone response in the trunk when the source is active and when the source is switched off. Above 10 Hz, the artificial noise generation results in a response that is up to 50dB higher guaranteeing that FRFs can be measured in workshops where background levels are typically somewhat higher.

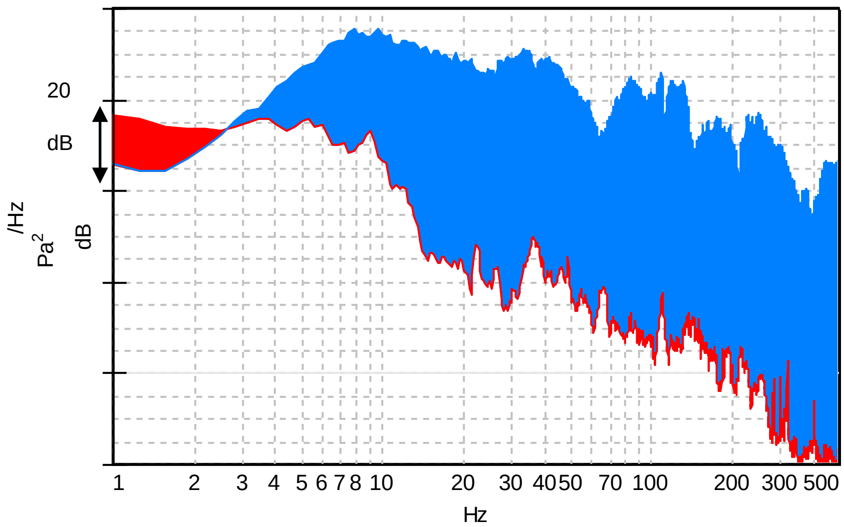

Figure 5 – Noise level versus background noise.

Next to acoustic-acoustic FRFs, the Q-MED enables the acquisition of vibro-acoustic FRFs. These FRFs include information about the interaction between the acoustic cavity and surrounding panels at the boundary which can be used to correlate simulation models which include both acoustic and structural elements. Figure 6 shows three consecutive FRF measurements indicating highly repeatable measurements. The structural response due to the source excitation is significantly higher than the accelerometer self noise above 10 Hz. This is depicted in Figure 7.

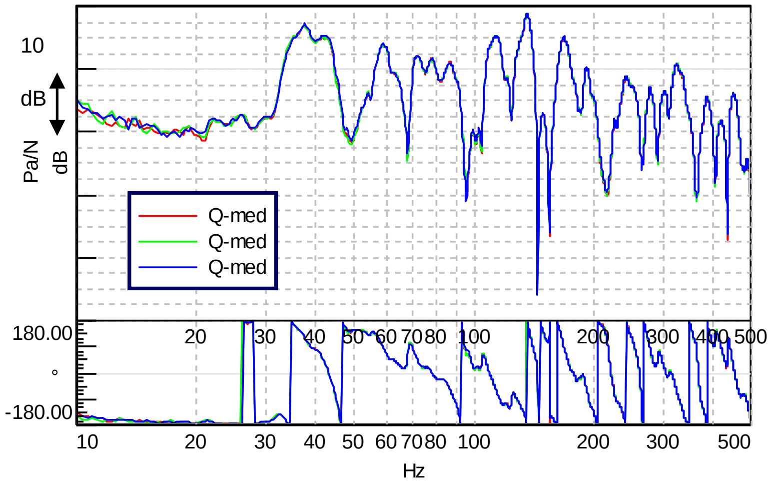

Figure 6 – Repeatability of vibro-acoustic FRF measurements.

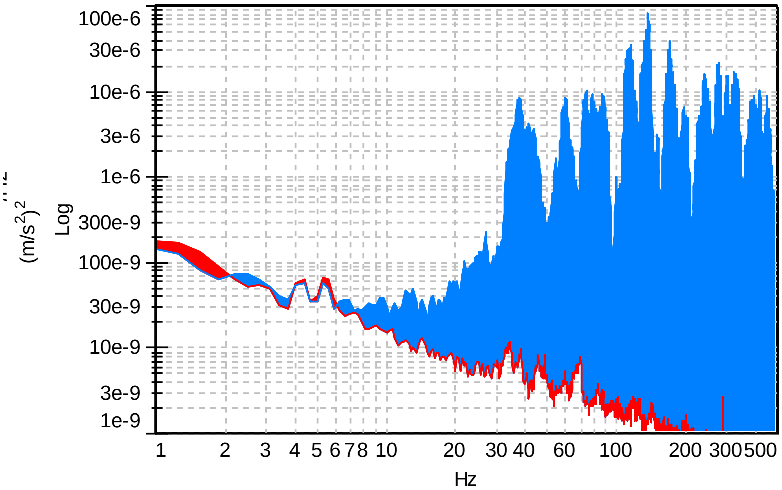

Figure 7 – Acceleration PSD and background noise.

The Q_MED even allows measuring vibro-acoustic FRFs to potential structure borne noise source interfaces such as powertrain mounts. This data contains also the sensitive frequencies because of the acoustic modes. Advanced analysis is now possible such as TPA analysis to understand structure borne noise generation.

The following plot contains three consecutive vibro-acoustic FRF where the source has been positioned at driver ear location and the structural acceleration response has been measured at one engine mount.

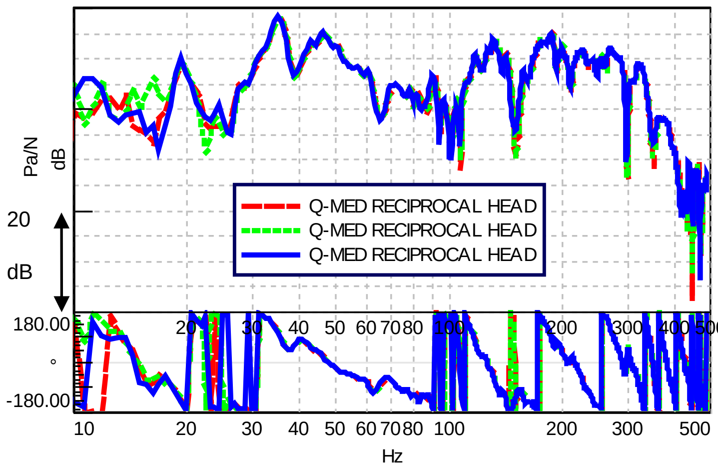

Figure 8 – Vibro-acoustic FRF.

4. MAXIMUM LIKELIHOOD ESTIMATION BASED ON THE MODAL MODEL (ML-MM)

It has been observed that classical modal parameter estimation methods have some difficulties in fitting an FRF matrix that consists of many (i.e. 4 or more) columns, i.e. in cases where many input excitation locations have been used in the experiment. Due to the high damping, the many excitation locations are required to get sufficient excitation of the modes across the entire cavity. Therefore, a new iterative frequency-domain solver is proposed that has the potential to overcome the difficulties with many references.

The ML-MM method (5, 6) is a multiple-input multiple-output (MIMO) frequency-domain estimator providing global estimates of the modal model parameters. In the first step of the ML-MM estimator, initial values of all the modal parameters (i.e. poles, mode shapes, participation factors, and upper and lower residuals) have to be specified. In the next step, the error between the modal model equation and the measured data is minimized in a maximum likelihood sense. Assuming the different measured FRFs to be uncorrelated, the ML-MM cost function to be minimized can be formulated as:

$$K_{ML-MM}(\theta) = \sum_{o=1}^{N_o}\sum_{k=1}^{N_f} E_o(\theta,\omega_k)\, E_o^H(\theta,\omega_k)$$

where $()^H$ stands for the complex conjugate transpose of a matrix (Hermitian), N_o the number of outputs, N_f the number of frequency lines, ω_k the circular frequency, and $E_o(\theta,\omega_k)$ the weighted error equation corresponding to the o^{th} output degree of freedom (DOF) given as follows:

$$= \begin{bmatrix} \frac{H_{o1}(\omega_k) - \widehat{H}_{o1}(\theta,\omega_k)}{\sqrt{\mathrm{var}(H_{o1}(\omega_k))}} & \cdots & \frac{H_{oN_i}(\omega_k) - \widehat{H}_{oN_i}(\theta,\omega_k)}{\sqrt{\mathrm{var}(H_{oN_i}(\omega_k))}} \end{bmatrix}$$

Where $H_{oi}(\omega_k) \in \mathbb{C}$ the measured FRF, $\widehat{H}_{oi}(\theta,\omega_k) \in \mathbb{C}$ the modeled FRF, $\mathrm{var}(H_{oi}(\omega_k))$ the variance of the measured FRF for output o and input i and N_i is the number of inputs.
Assuming displacement FRFs, $\widehat{\mathrm{H}}(\theta,\omega_k) \in \mathbb{C}^{N_o \times N_i}$ can be represented using the modal model formulation as follows:

$$\widehat{\mathrm{H}}(\theta,\omega_k) = \sum_{r=1}^{N_m}\left(\frac{\psi_r L_r}{s_k - \lambda_r} + \frac{\psi_r^* L_r^*}{s_k - \lambda_r^*}\right) + \frac{\mathrm{LR}}{s_k^2} + \mathrm{UR}$$

with N_m the number of identified modes, $\psi_r \in \mathbb{C}^{N_o \times 1}$ the r^{th} mode shape, λ_r the r^{th} pole, $s_k = j\omega_k$, $()^*$ stands for the complex conjugate of a complex number, $L_r \in \mathbb{C}^{1 \times N_i}$ the r^{th} participation factor, $\mathrm{LR} \in \mathbb{C}^{N_o \times N_i}$ and $\mathrm{UR} \in \mathbb{C}^{N_o \times N_i}$ the lower and upper residual terms. The lower and upper residual terms are modeling the influence of the out-of-band modes in the considered frequency band. The maximum likelihood estimates of θ (i.e. ψ_r, L_r, λ_r, LR, and UR) will be obtained by minimizing the above-mentioned cost function $K_{ML-MM}(\theta)$. This will be done using the Gauss-Newton optimization algorithm. To ensure convergence, the Gauss-Newton optimization is implemented together with the Levenberg-Marquardt approach, which forces the cost function to decrease (7). To start the optimization algorithm, initial values for all the modal parameters are estimated by the well-known LMS Polymax method (8). More details about the ML-MM method (e.g. mathematical implementation, uncertainty derivation, …) are presented in (5, 6).

5. CASE STUDIES

In order to highlight the challenges related to acoustic modal analysis and to illustrate the benefits of the new modal parameter estimation approach, both a numerical study and an experimental case are presented.

5.1 Numerical simulation of an automotive cabin

An acoustical Finite Element (FE) model was created of an interior car cavity with rigid boundaries. Realistic modal damping ratios (ranging from 5 to 20%) have been introduced in the simulation model. Some FE mode shapes are shown in Figure 9. In addition, FRFs have been simulated by selecting 8 virtual inputs and 611 virtual outputs from the acoustic mesh (Figure 10). The idea is now to apply modal parameter estimation to these simulated FRFs and compare the outcome with the true FE results (like the ones in Figure 9). The Polymax method yielded almost perfect results (Figure 11). However, when using only 1 input in the analysis, it was observed that the identified mode shapes show distortions in the neighbourhood of the virtual input location (Figure 12). This phenomenon is well-known from experimental work (1, 2), but it was rather surprising to observe it also in noise-free simulations, indicating that it is not due to experimental error. Further investigations are needed, but it is probably due to the high damping that implies the need for distributed excitation of the modes. This is also clearly shown in Figure 13 where the identified mode shapes converge to the true values as more inputs are added.

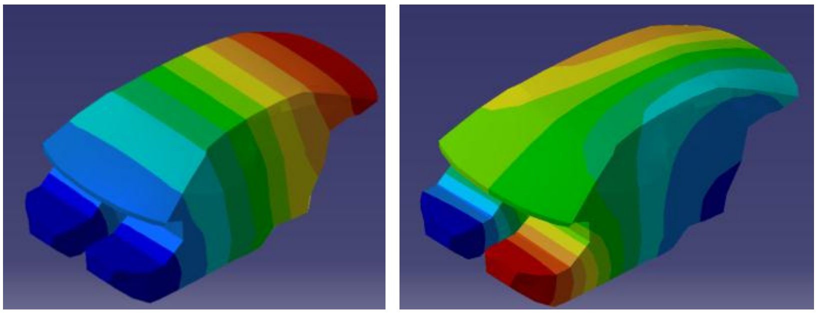

Figure 9 – Some typical FE mode shapes represented in LMS Virtual.Lab Acoustics.

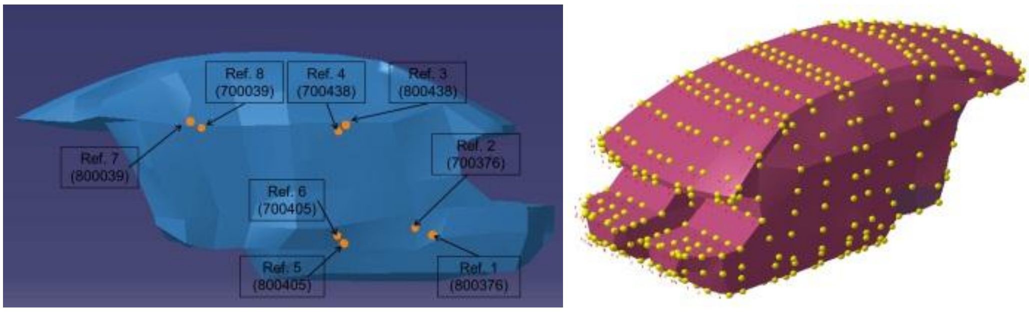

Figure 10 – Virtual loudspeaker and microphone locations.

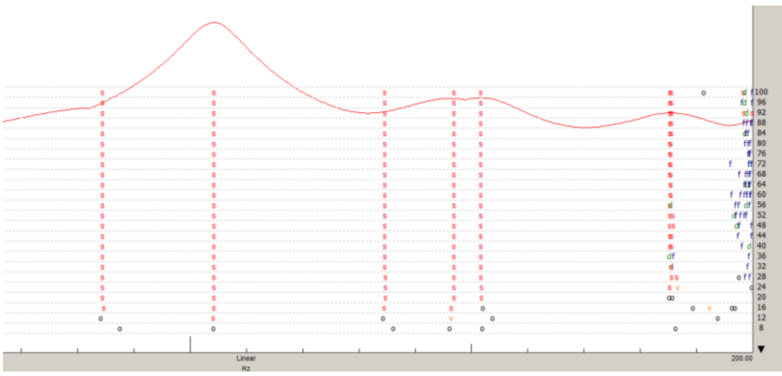

Figure 11 – LMS Polymax stabilization diagram, with correct identification of 7 modes.

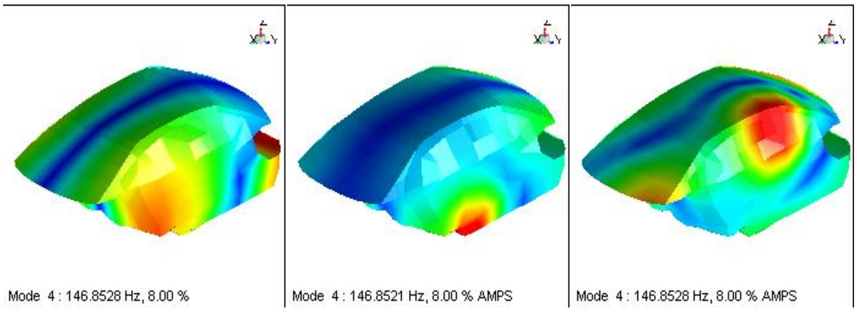

Figure 12 – (Left) true mode shape; (Middle) distorted mode shape – exciter at rear right foot; (Right) distorted mode shape – exciter at front right ear.

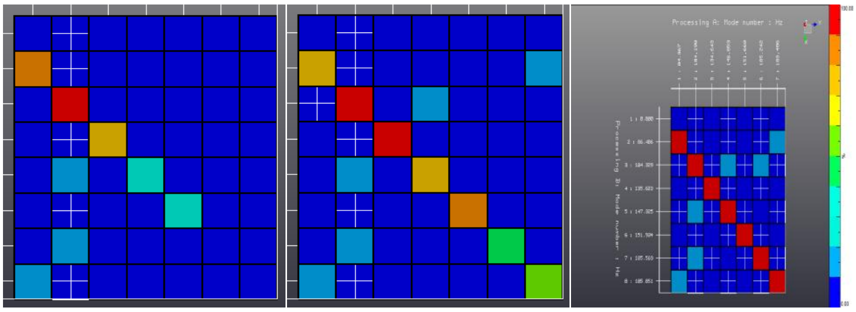

Figure 13 – MAC values between identified (bottom axis) and true modes (left axis). The MAC is improving as more references are included in the analysis: 1 – 2 – 4.

5.2 Experimental modal analysis case

The experimental case concerns an aircraft interior sound analysis. It has been selected because of the non-confidential nature of the data and yet it still represents some of the typical challenges that are also encountered in automotive applications: the use of many references and the presence of highly-damped complex modes.

Specific ground tests have been executed on the ATR42 aircraft to derive the intrinsic system information, which should render it possible to explain observed in-flight behaviour. In this case, simultaneous excitation was applied at four loudspeakers, two longitudinal and two lateral ones. Microphones at 20 positions captured the responses simultaneously. A total of 12 sections of the plane cavity were measured, resulting in 240 response locations (Figure 14).

Both Polymax and the new ML-MM method have been applied to the 960 FRFs. Figure 15 shows the decrease of the ML-MM cost function at each iteration. The analysis was stopped after 20 iterations. In Figure 16, the synthesized FRFs are compared with the measured ones. It is obvious that the ML-MM synthesis results are superior to the Polymax synthesis results. Finally, some typical mode shapes identified with ML-MM are represented in Figure 17.

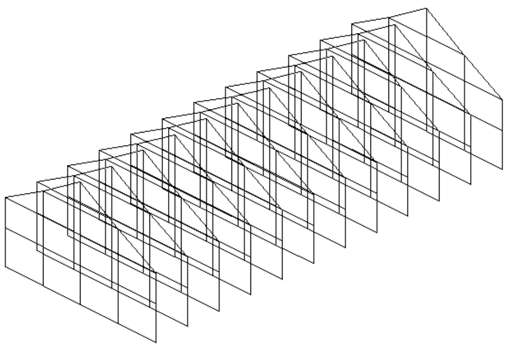

Figure 14 – Microphone locations inside the ATR42 propeller aircraft.

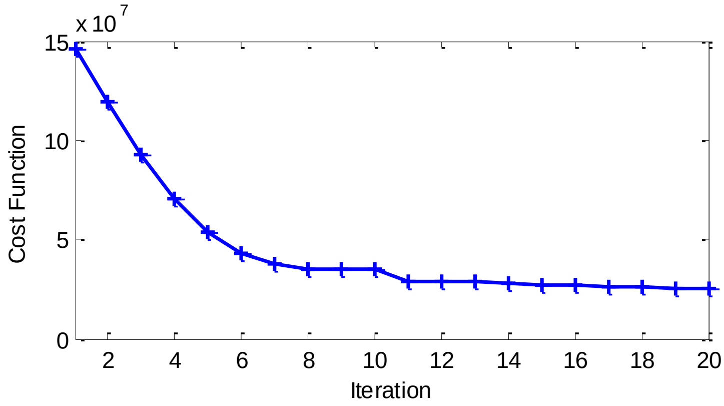

Figure 15 – Decrease of ML-MM cost function at each iteration

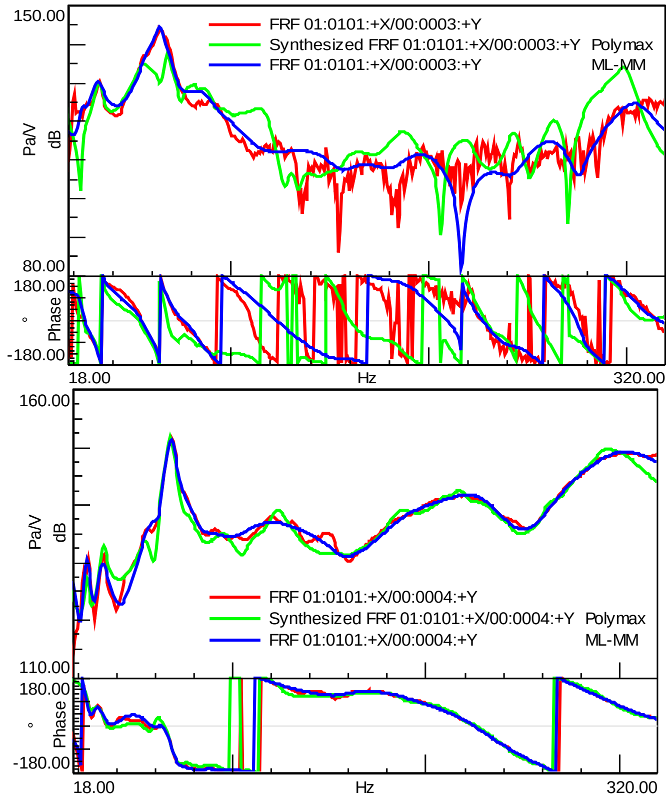

Figure 16 – Improved FRF curve fit when using ML-MM as compared to Polymax.

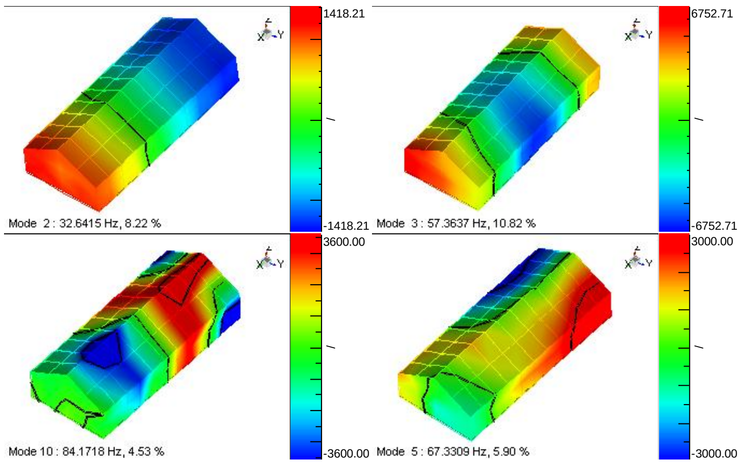

Figure 17 – Some typical acoustic mode shapes of the ATR42 interior cabin identified with ML-MM.

6. CONCLUSIONS

This paper introduced some new developments related to acoustic modal analysis. A new sound source was proposed that is compact, omnidirectional and capable of generating high noise levels in the low frequency range. A detailed experimental validation study was performed that confirmed that the new source is excellently suited for automotive cabin acoustic modal analysis.

Numerical simulations were performed that allowed to understand some of the issues with highly-damped acoustic data, e.g. mode shape distortion close to the loudspeaker location.

A new solver was introduced. Maximum Likelihood Estimation based on the Modal Model (ML-MM) deals properly with FRF matrices with many references and provides superior FRF synthesis results.

Future work will consists in additional numerical simulation (e.g. adding noise, considering vibro-acoustic coupling) and experimental validation of the new sources and the new ML-MM method by means of a detailed automotive acoustic modal analysis case study.

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