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Measurements and Simulations of Acoustical Performance of Plastic Air Intake Manifolds for Internal Combustion Engines

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Abstract Automotive engine manufacturers are focusing increasingly their attention on noise generated by air intake systems. In particular in recent years the increased usage of plastics for air intake manifold (AIM) production, in place of metallic materials, made the NVH optimization more complicated. In this framework, it is very important not only the minimization of the noise generated via fluid propagation (orifice-noise from inlets) but also of the noise radiated via the coupled fluid-structure interaction. In this work acoustical performance of an AIM prototype has been experimentally and numerically investigated. From the experimental point of view, the acoustic analysis was performed by means of acoustic intensity measurements around the body. In order to minimize the noise contribution from all the other sources of an operating engine, the AIM was assembled on an engine head which was mounted on a dynamic flow test bench, where the intake valve-train was driven by an electrical motor. The intake orifice noise was differently treated, both leaving the orifice open and employing a dissipative muffler. Simulations through a coupled fluid-structure approach were performed. The normal modes of the structure were previously calculated on a fully-detailed FEM model and checked through accelerometric data; material properties could then be properly chosen. Measured data of the pressure pulsations at the inlet valves was used as excitation and applied to a coarsened acoustic Indirect-BEM model of the AIM coupled with a structural FEM one. Tuning of experimental and numerical acoustic data was useful for defining a practical analysis procedure for NVH design of plastic AIMs.

1. INTRODUCTION

The use of plastic materials has become the state-of-the-art in intake system design, at least for SI engines, mainly because of the easier manufacturing process of those materials. However, it is well known that NVH properties of plastic air intake manifolds (AIMs) can be worse than those of the common aluminium ones, if a dedicated investigation of this aspect is not properly performed [1-2]. Many studies have been published in recent years concerning the vibro-acoustic analysis of intake systems for internal combustion engines (see for example [3]). Particular attention has been focused on the numerical simulation of plastic AIMs by means of 3D tools, as the finite-element and the boundary element methods [4-6]; on the other side, much less has been done for the experimental investigation of structure-borne sound [7].

Among the main noise sources of intake systems are flow noise radiated from the inlet (orifice noise) and the structure-borne one radiated from the manifold's surface (shell noise) [3]. While the former is obviously related to the engine breathing optimization [8], the latter is strongly affected by the material and the shape chosen for the AIM. These two phenomena are nonetheless undoubtedly coupled and they should be considered at the same time, at least if calculated acoustic data has to be compared with experimental one.

Aim of this work was then a vibro-acoustic study of a prototype plastic AIM, employing both experimental and numerical tools. Measurements and simulations were performed at several regimes and with different air intake conditions (e.g. open orifice, dissipative muffler). Particular attention was paid to the choice of the appropriate boundary conditions as a trade-off between accuracy and modelling effort. A practical methodology was developed using vibration, inner pressure and sound intensity measurements on the AIM mounted on a dynamic flow test bench, in concurrence with coupled fluid-structure FEM-IBEM analyses of a suitable model. This procedure should reliably reproduce the measured data and, at the same time, provide additional information on the effect of structural-borne noise, so that it could be employed for comparative studies on different virtual prototypes.

2. NUMERICAL AND EXPERIMENTAL METHODOLOGY

In recent years, several numerical and experimental research tools have become available for NVH investigation: among the former, FEM, direct and indirect BEM are widely used. An engineering approach duly needs a choice among those tools, taking into account their accuracy, their complexity and compatibility. In this framework, a numerical analysis procedure has been developed to provide reliable results, being able to be validated against experimental data.

A fully-detailed 3D model of the AIM under test was available as a basis for a mesh constituted by about 30 000 2D elements (see Figure 1). Several modal finite-element analyses have been performed by ANSYS® (Release 8.0) up to around 1.7 kHz: free-free and clamped-free boundary conditions were tested both including and not the mass of an intake flange. A thorough numerical and experimental investigation has been carried out to accurately estimate the materials properties: Young's modulus was roughly measured on a small beam sample of the material and its value (2500 MPa) was finally chosen tuning the first calculated modes with those experimentally estimated in free-free boundary condition. As an example, data reported in Table 1 show the results for the first eleven modes in the free-free condition with orifice flange mass. It has to be noticed that not all the calculated modes could be experimentally detected, as some of them regarded mainly small elements of the AIM. The material density (1300 kg/m^3) and the shell thicknesses of the various manifold's parts (e.g. duct walls, air plenum, ribs and flange) were accurately chosen to match the model mass with the measured one.

From the experimental point of view, the AIM's modal frequencies and shapes were estimated by means of an instrumented hammer and piezoelectric accelerometers (B&K Type 4502) on the manifold.

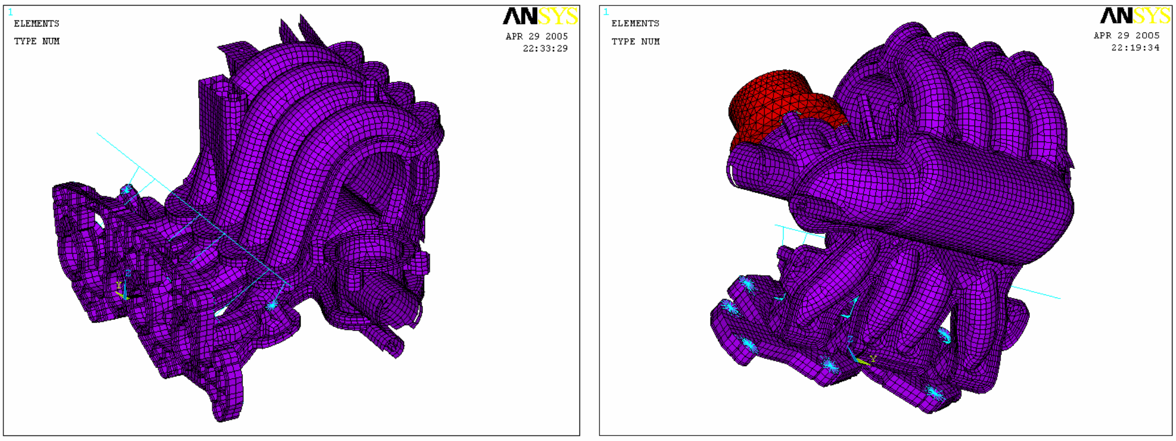


Figure 1: The fully-detailed 2D element mesh of the AIM under test. On the left, AIM without orifice flange mass; on the right, with flange mass.

Table 1: Modal analysis results for free-free AIM with flange mass. Comparison between experiments and simulations.

Mode #	Experimental frequency [Hz]	FEA results [Hz]
1	143	139.99
2	-	170.21
3	180	179.89
4	244	246.53
5	316	321.46
6	-	373.09
7	410	422.78
8	-	436.87
9	455	459.88
10	466	481.17
11	532	522.54
12 to 60	-	569.75 to 1518.3

Table 2: Modal analysis results for clamped-free AIM with flange mass. Comparison between experiments and simulations.

Mode #	Experimental frequency [Hz]	FEA results [Hz]
1	48	41.82
2	78	91.64
3	119	112.97
4	162	187.08
5	170	188.48
6	272	278.32
7	296	291.59
8	399	409.54
9	429	431.41
10	-	513.93
11	-	522.61
12 to 60	-	578.05 to 1663.4

Further experimental data were then collected for assessing the finite element model also in more realistic conditions, such as the clamped-free ones. Table 2 shows the comparison for the clamped situation (at the engine head coupling bolts) with flange mass applied to the orifice: a remarkable modification of the structural normal modes was observed, as compared to the data in Table 1, emphasising the importance of a careful modelling of the actual operating conditions.

The fully-detailed mesh could not be used for the acoustical analysis, mainly for limitations of the chosen acoustic simulation method, and by the way many small details (in comparison to the wavelength) were considered insignificant for the acoustic AIM's performance. A new 3D model was then especially created complying with the AIM's macroscopic geometry and dimensions. Then the mesh was coarsened until the number of shell elements could be reduced by an order of magnitude (up to about 3 000) and consequently could be managed by the simulation code; nonetheless, the upper frequency limit was 2.7 kHz using the 6-elements-per-wavelength assumption [9].

This simplified mesh was employed to build the structural and acoustical models and to run a structural FEM – acoustical indirect BEM coupled analysis (through SYSNOISE®, Release 5.6). The choice of an indirect BEM approach for the acoustic modelling is inevitably due to the presence of an open structure (the AIM) and hence to the need of simultaneously considering the fluid domains on both sides of the boundary elements. The structural modes calculated consistently with the experimental settings (i.e. clamped-free boundary conditions including the orifice flange mass) are shown in Table 2; they were projected, by means of an interpolation procedure, on the coarsened structural FEM model (Figure 2 on the left). The acoustic BEM model was created modifying the simplified mesh in order to properly define all the necessary boundary conditions: indeed, elements had to be added to model the duct portions which fall inside the engine head (from the valves to the AIM's outlet) and then to simulate the ducts in their full lengths (Figure 2 on the right). The coupled analysis was performed in the 20 - 1000 Hz range, with a proper step dependent on the chosen revolution speed in order to track all the harmonic frequencies of the pressure excitation. The indirect BEM demands for appropriate boundary conditions: at the common edges of composite structures constituted by more than two surfaces (T-junction lines) and at the free edges (zero jump of pressure) [9]. Acoustic quantities, and in particular sound intensity, were calculated on a field-point mesh purposely defined to match the measurement grid, as detailed below.

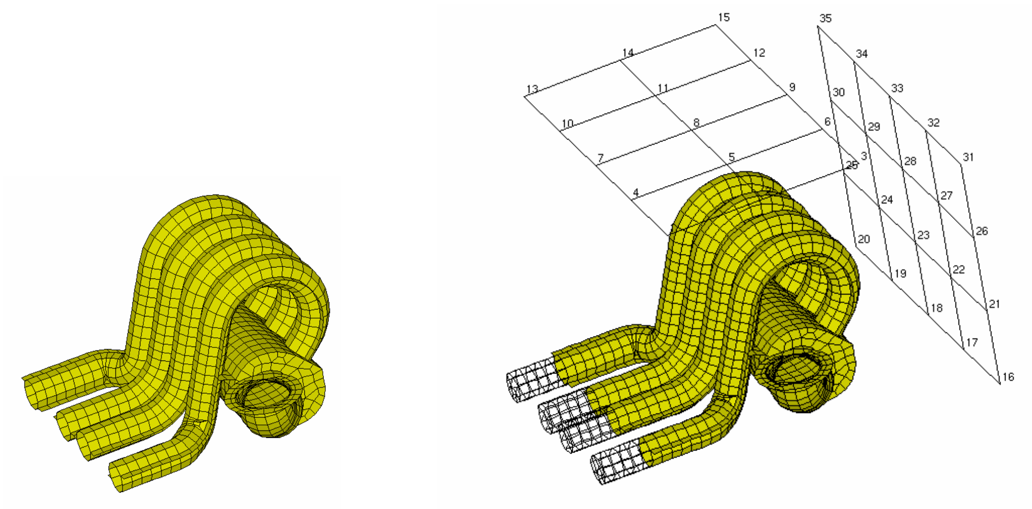


Figure 2: Simplified shell-element mesh of the AIM used for coupled analysis. On the left, FEM structural model; on the right, superimposed wire-frame BEM acoustic model and field points chosen for post-processing of the results.

The pressure fluctuations of the inner fluid domain caused by the valves' motion could be experimentally measured using four piezoresistive pressure transducers (Endevco®type

8530C-50) close to the AIM's outlets and compared with the results of a 1D fluid-dynamics engine model (AVL Boost®, Release 4.0.4), achieving a satisfying match. As unique loads for the coupled acoustical simulation, the normalised pressure FFTs were applied on the inner side of the AIM's outlet boundary elements. At the same time, a zero-admittance condition was imposed on the outer side of all the non-plastic duct portions, which pertain to components other than the manifold.

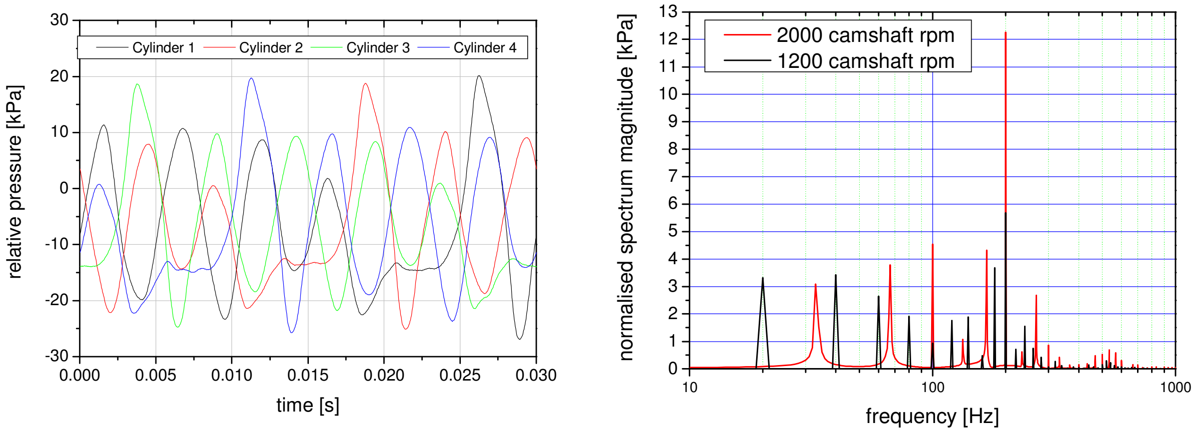
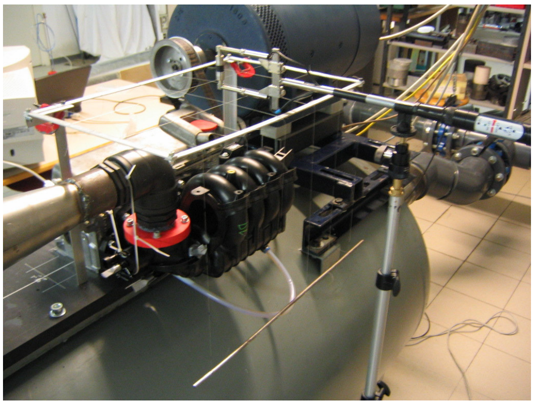


Figure 3: On the left, an example of measured pressure at the AIM's outlets (2000 rpm of the camshaft); on the right, examples of normalized pressure spectra (at 1200 rpm and 2000 rpm of the camshaft).

As far as the acoustic experiments are concerned, a 2-microphone intensity probe G.R.A.S. Type 50AI was chosen to get information on the noise emission by the manifold. The latter was then mounted on a dynamic flow test bench (commonly used for tuning engine intake systems), constituted by a blower, a vacuum-vessel (to simulate the in-cylinder pressure conditions) and by a DC motor which drives the valve-train camshaft at the desired revolution speed. In Figure 4 a picture and a schematic of the dynamic flow test bench is given, showing the main constituting elements. The engine head is directly faced to the stilling chamber, the lower side of the engine is removed and the piston-crankshaft absence is replaced by the electric driving. The desired constant pressure level beneath the head is imposed by means of a blower.

Through a preliminary study, the best representative values for the vessel pressure (200 mbar under atmospheric pressure) and the rotating regime (1200 rpm and 2000 rpm for the camshaft) were chosen. One-third octave band sound intensity levels were determined on a 35-point rectangular grid placed on the top and on the plenum side of the manifold (see Figure 5 on the left). The grid was not entirely embracing the source because of its mounting conditions; by the way, experimental results were devoted to the validation of numerical simulations and not to the estimation of the overall sound power emitted. Appropriated solutions were adopted to minimize measurement errors in the low-medium frequency range, e.g. 50 mm and 12 mm microphone spacers, long average periods, reduction of the influence of external sources and reverberating sound field through increase of the room walls' absorption.



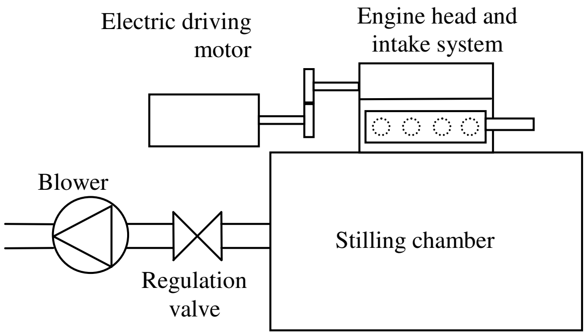


Figure 4: Picture of the dynamic flow test bench, on the left, and schematic of its layout, on the right.

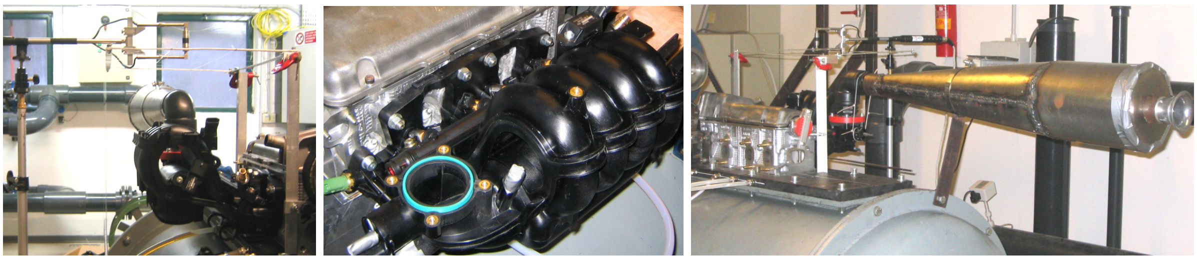


Figure 5: On the left, example of intensity probe location; on the middle, open inlet orifice condition; on the right, silenced inlet condition.

3. RESULTS

Measurements were initially carried out leaving open the intake orifice (see Figure 5 on the middle): however, sound intensity levels due to the orifice were so high (well above 100 dB, in particular at the lower frequencies) to make the measurement conditions very hard for the operators. An effective dissipative muffler was then designed to reduce the orifice noise and properly investigate the shell-noise (see Figure 5 on the right).

Figure 6 shows the effect of employing the muffler on the measured sound intensity levels at 2000 rpm camshaft revolution speed. The average sound attenuation is about 20 dB up to 1 kHz, while it is 10 dB at the higher frequencies. It is evident that the attenuation mainly affects the harmonics related to the orifice noise, which are due to the inner pressure fluctuations and are mainly limited to 1 kHz, as shown previously in Figure 3. To this extent, it must be noticed that the first engine order, in the present case, is equal to 67 Hz.

A coupled fluid-structure simulation for the open orifice conditions at 2000 rpm camshaft speed was then performed according to the methodology described in the previous Section 2, and the calculated active sound intensity results (merged into third-octave bands) are shown in Figure 7, for comparison with the measured ones (both evaluated as area-averaged levels on the top surface grid). In particular the excitation pressures were evaluated at the actual valve seats by means of the above mentioned 1D simulations. It can be seen that the calculated sound intensity matches quite well the measured levels below 400 Hz; for the higher bands the simulations results seem to be overestimated: this could be due to some difficulties to capture properly the actual inner pressure magnitude in that range. It is worth noting that the fluid-structure coupling demonstrated its main effect just in the same range, as detailed at the end of the Section.

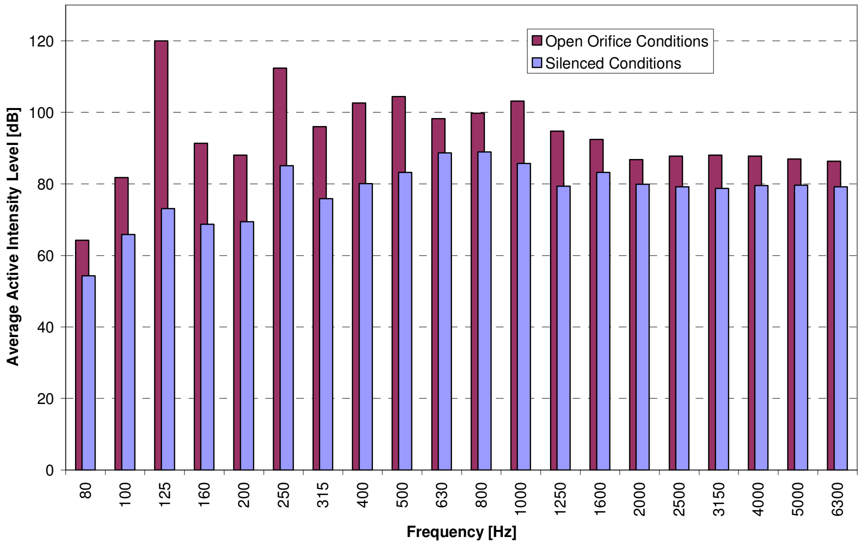


Figure 6: Effect of silenced orifice on the average sound intensity level recorded on a planar surface above the AIM, at 2000 camshaft rpm (using a 12 mm microphone spacer).

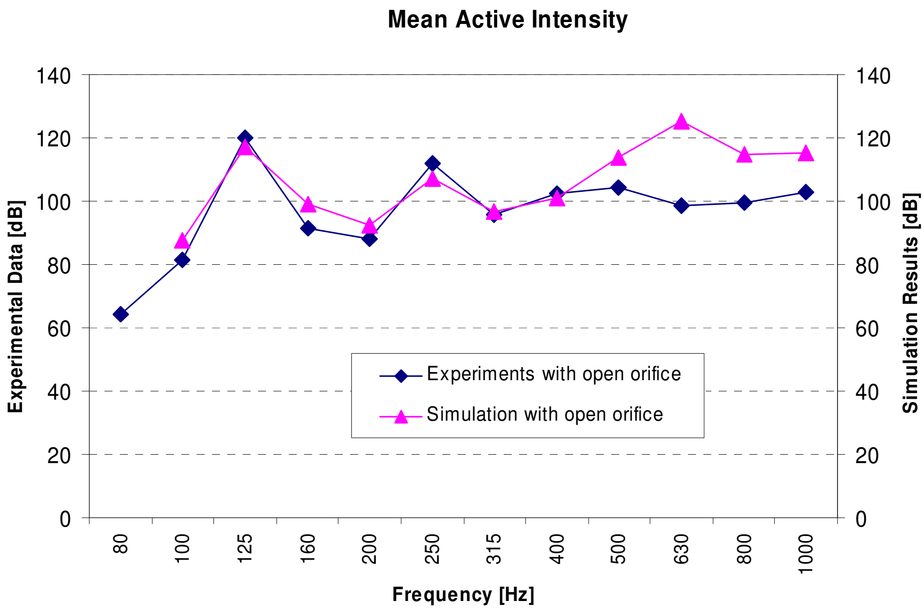


Figure 7: Average sound intensity level recorded and calculated on the planar surface above the AIM (see Figure 2, right), at 2000 camshaft rpm (measurement performed with a 12 mm microphone spacer).

In order to better isolate the AIM properties from the global noise emission, strongly affected by the open orifice radiation, the silenced configuration was then conveniently analysed. In Figure 8, the spatial average of the active intensity, over the two planar surfaces around the AIM, is shown. The experimental data relate to the aforementioned silenced intake conditions, while the numerical results relate to the open intake orifice conditions, as a detailed numerical description of the absorbing characteristics of the actual muffler was beyond the scope of the present work. As can be noted, the levels obtained by the simulations overestimate in the whole range of frequencies the measured values due to the dissimilar intake conditions (results are shown on different scales). However the spectrum trend seems to be substantially captured. Indeed, all the main spectrum peaks can be detected (i.e. at 80, 160, 250 and 630 Hz). The main exception relates the 200 Hz band, where the calculated intensity is significantly overestimated, likely due to the relative high excitation content at this specific frequency, corresponding to the quarter wave of the intake runners (see Figure 3 on the right). To this extent, besides the differences caused by employing the muffler, it has to be noted that measured data can also include contributions belonging to surrounding sound sources unavoidably caused by driving the engine head (see Figure 4).

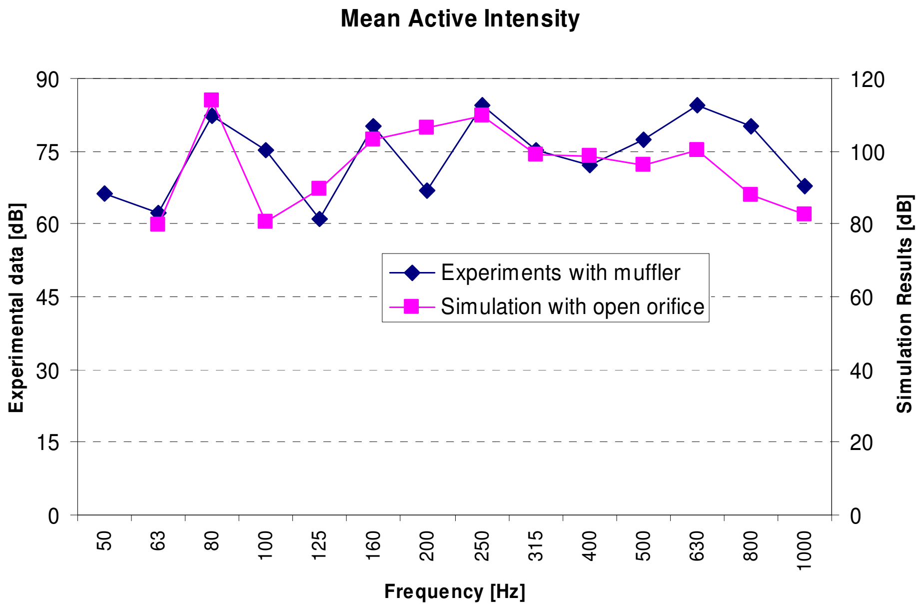


Figure 8: Average sound intensity level recorded and calculated on the two planar surfaces defined around the AIM (see Figure 2, right), at 1200 camshaft rpm (measurement performed with a 50 mm microphone spacer).

The last step of this work has dealt with the analysis of the shell-noise originated by particular areas of the AIM. Figure 9 shows an example of the noise contribution at 660 Hz correlated to the evident structural mode of the plenum surface. This phenomenon can be addressed as the main reason for the 630 Hz band peak, shown in the spectrum of Figure 8. The coupled fluid-structure analysis exhibits strong capability in localising these kind of effects, as shown by the peak of sound intensity level predicted on the plenum-side plane, even if the excitation pressure was not particularly high at this frequency (see Figure 3, on the right).

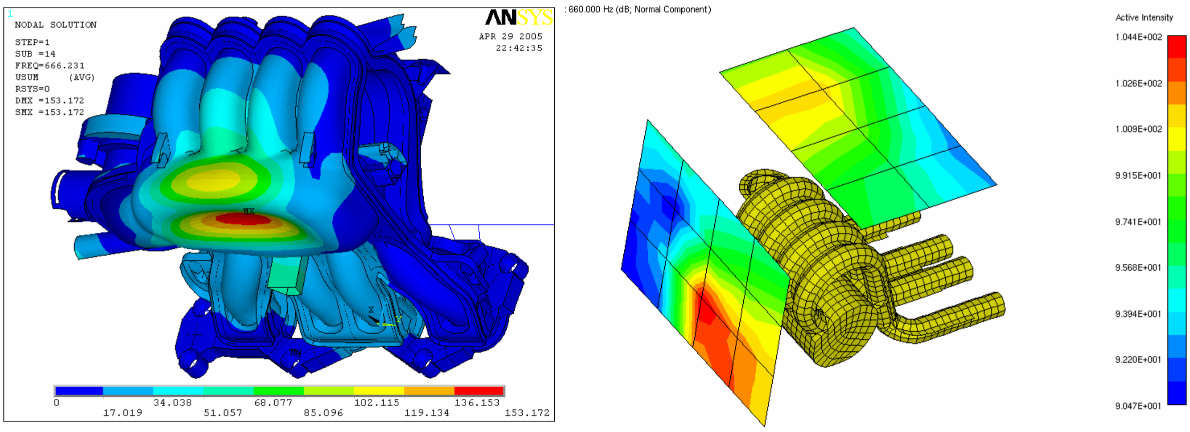


Figure 9: Detection of a particular shell-noise source. Evident structural mode of the plenum at 666.2 Hz, on the left; active intensity map [dB] on the field points mesh at 660 Hz, after the coupled FEM - IBEM analysis for the 1200 camshaft rpm case.

4. CONCLUSIONS

A practical methodology for the analysis of the acoustical performances of plastic air intake systems for SI engines has been discussed. The work focused on the combined usage of experimental and numerical characterisation of the noise field around the AIM body. Results from numerical simulations were used to assess the vibro-acoustic response of the plastic intake system and to achieve better predictive capabilities on the design optimisation. Satisfactory results were obtained both for predicting the global sound intensity level and for discerning the main shell noise contributions.

By means of a dynamic flow test bench, suitable sound intensity data were acquired, limiting to an acceptable level the inclusion of contaminating sound sources by the test rig. Moreover pressure pulsations on the intake runners and accelerometric characterisations of the AIM structure were collected and used for tuning a 1D gas-dynamic model of the intake system.

At the same time, the interpretation of some phenomena was carried out by means of the numerical procedure. First a proper tuning of the detailed finite element model of the intake system for the evaluation of the normal modes and frequencies was performed and checked through the experimental data, with particular emphasis on the modelling of the actual assembling layout and constraints. Subsequently, through a coarsened shell mesh, a straightforward numerical procedure was used for a coupled FEM-IBEM analysis of the whole system, using the runner pressure data as acoustical excitation loads.

Considering the complexity of the involved physical phenomena, it can be stated that a satisfactory comparison of experimental and numerical data can only be achieved once the following aspects have been thoroughly investigated:

- Structural behaviour of the manifold: it is strongly dependent on the mounting conditions and it mainly affects the medium frequencies (400 – 1000 Hz), as can be noticed carrying out simulations with an uncoupled approach;

- Inner pressure fluctuations: they must be accurately measured for operating conditions consistent with the calculations – if calculated data (i.e. with 1D tools), is employed care has to be take to avoid discrepancies with the experiments;

- Acoustical model boundary conditions: it is evidently hard to simulate all the acoustical effects occurring at the AIM's outlets and related to the valve motion – a procedure based on considering just the pressure at the valve locations seems to provide reliable results, at least in comparison with the available experimental data.

5. ACKNOWLEDGEMENTS

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