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TRANSMISSION LOSS ASSESSMENT FOR A FILTER BOX BY BEM APPROACH

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ABSTRACT

This paper describes the numerical and experimental activities related to the characterization of the Transmission Loss (TL) of a filter box of a four-cylinder spark ignition engine. The overall aim is to improve the understanding of the component behaviour, in terms of acoustic performance. The numerical approach is based on the Boundary Element Method (BEM) as implemented in two different commercial codes: BEASY ® v.10 and VNOISE® v.2.3. The TL characterization is based on the four-pole methodology. Experiments in an acoustic chamber, based on the use of a sound intensity probe, are performed in all the investigated frequency range in order to verify the accuracy of numerical analyses. The results are finally summarized and compared. The results provided by BEASY code are compared with the experimental ones with reference to the TL, obtaining an acceptable correlation within the limits of the approximations made (no allowance for structural-acoustic coupling). Moreover a qualitative contour plot comparison between BEASY and VNOISE is provided.
The employment of different numerical codes informs about their relative efficiency in terms of pre/post-processing time and cpu time.

1 INTRODUCTION

The products of many industries can benefit from improved acoustic design. Automotive filter boxes give a major contribution to the sound quality of a vehicle and must be properly designed in order to produce acceptable acoustic performances. The structural born noise can be studied by numerical methodologies such as FEM, effective in the low frequency range, and BEM, effective in a larger frequency range (up to medium frequencies).

In this work, the noise attenuation characteristics of a filter box are investigated. Acoustic performances are specified at the component level as attenuation versus frequency: to this end numerical and experimental investigations are performed.

The numerical approach, also finalized to the prediction of the Transmission Loss (TL) factor, is a 3D boundary element acoustic analysis, realized using BEASY code (ver. 10) and VNOISE code (whose outcomes in terms of TL will be reported in a forthcoming paper). These codes are based on the employment of the Boundary Element Method (BEM) and in particular BEASY is based on a collocation approach and direct method. Such approach moves from the numerical solution of the Boundary Integral Equations (BIE) substituted to the Helmoltz equations.

This work is aimed to build up a reliable numerical procedure for the transmission loss prediction in many different acoustic problems, including acoustic characterization of mufflers, acoustic radiation of vibrating shells and so on.

The FEM code NASTRAN is used for the filter box structural dynamic assessment and the BEM codes BEASY and VNOISE for acoustic analysis.

From the experimental point of view, the TL is measured in an acoustically treated laboratory.

In both FEM and BEM methodologies acoustic models can be developed using general purpose modellers through the link to CAD provided, but with BEM the boundary mesh generation is very simple compared with volume mesh generation and can be completed automatically with little user interaction.

2 EXPERIMENTAL TL EVALUATION

A simple experimental apparatus is installed in a room with wall acoustic treatment to measure the attenuation curve of the tested system. An acoustic excitation is provided by a loudspeaker (fig. 1), that is insulated in a sound-proof box and radiates a random signal (generated by the analyzer) into the outflow exit section of the component.

A sound intensity probe is used in order to obtain the contribution coming only from the inflow section. At the same time, the loudspeaker signal is acquired and processed with the sound intensity measured at the inlet section.

The TL factor is computed as the ratio of the acoustic power associated with the incident and the transmitted waves.

In more details, the test facility consists of a sound measurement equipment and of a sound source equipment (a random noise generator, an amplifier and a loudspeaker unit), that excites a sound field with dominating plane wave mode. In particular, the sound source consists of a sound intensity probe (B&K model), a FFT analyzer (LMS Scadas III system), an anechoic box and the dedicated software for the TL calculation.

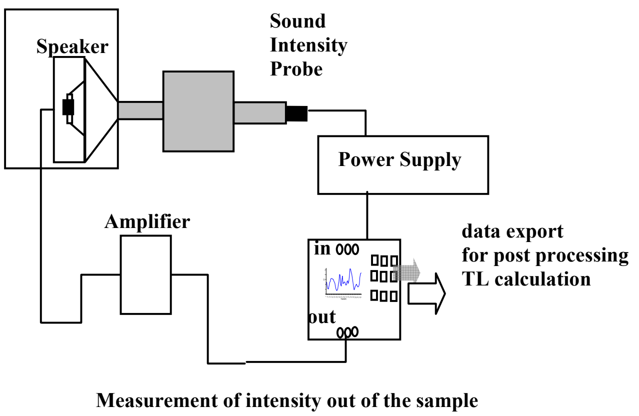

Figure 1 - Measurement test equipment.

3 3D BEM ACOUSTIC ANALYSIS

An alternative or complementary methodology to FEM for the numerical evaluation of the TL parameter is based on 3D boundary element (BEM) acoustic analysis [1-3]. The BEM approach is a powerful tool for a more accurate performance prediction and system design, but, in particular for internal problems, requires high computational and memory resources. Only a surface mesh is required in this case, which is constructed by means of the commercial software FEMAP 9.0 ® and imported in the used codes. In particular in BEASY the .nas file was imported, obtaining directly the mesh and having to only impose boundary conditions and material properties. The mesh adopted is the same for both codes (fig. 2): the model consists of 3660 nodes and 3454 linear elements; the two internal plates, which form acoustic resonators, are modeled, whereas the flexible part of the inflow duct and the filter elements are neglected.

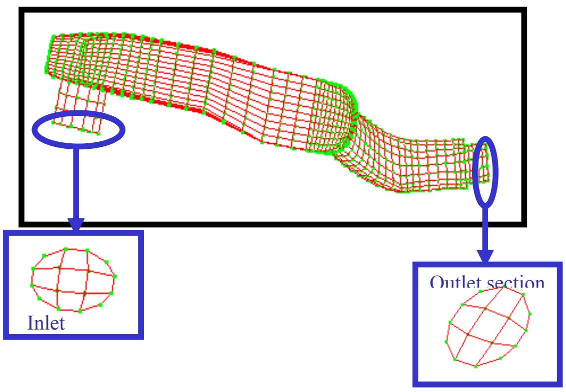

Figure 2 - Meshed model (inlet and outlet sketch).

Figure 3 shows the meshed model with its internal geometrical details, like the extended tube, whose length is 12 cm, placed within the expansion chamber.

Considering a generic system or a duct, as shown in fig. 4, the acoustic pressure p_1 at the inlet section is composed by two waves, namely one traveling from left to right (entering the air box), called p_1^+, and the other travelling in the opposite direction, called p_1^-. In correspondence to the outlet section the situation is analogous, and the total acoustic pressure p_2 is composed by two waves travelling in opposite directions. The velocity at the inlet (v_1) and at the outlet (v_2) sections can be also expressed in terms of the two wave components. The overall relations are therefore:

$$p_1 = p_1^+ + p_1^- \qquad p_2 = p_2^+ + p_2^-$$
$$v_1 = \frac{p_1^+ - p_1^-}{\rho c} \qquad v_2 = \frac{p_2^+ - p_2^-}{\rho c} \tag{1}$$

The TL factor is defined as the ratio between the sound power actually entering the system and the transmitted sound power.

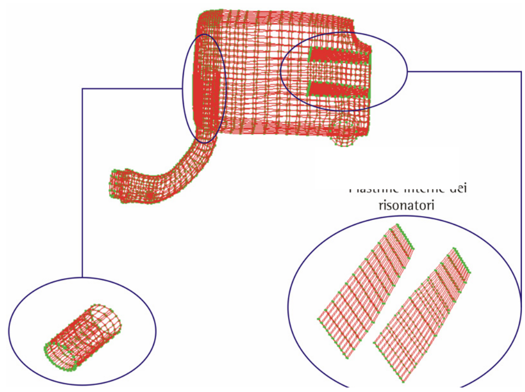

Figure 3 – BEM model.

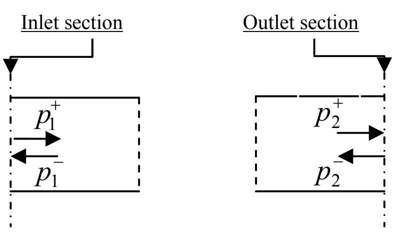

Figure 4 - Generic duct (inlet and outlet section).

The sound power exciting the system is associated to the right travelling wave at the inlet (p_1^+), while the transmitted sound power is associated to the right travelling wave at the outlet (p_2^+), so that it results:

$$\mathrm{TL} = 20\log_{10}\left(p_1^+ / p_2^+\right). \tag{2}$$

The standard procedure for the TL characterization is based upon the evaluation of the so-called four pole parameters (A, B, C, D) that characterize the component [4].

The BEM analysis is used to execute two sets of calculations, the first with a zero velocity at the outlet section, the second one with a zero pressure signal in the same section. The inlet velocity is set equal to the unity in both the cases. The four parameters (complex numbers depending on the frequency) can then be computed as:

A= $\left(p_1/p_2\right)$ and C= $\left(v_1/p_2\right)$ from set 1 (3)

B= $\left(p_1/v_2\right)$ and D= $\left(v_1/v_2\right)$ from set 2 (4)

An interesting property of the above parameters is that they theoretically satisfy the relation AD-BC=1. This can be used as an useful check in order to ensure a certain accuracy of the performed calculations. In this case, the above check is verified with an acceptable level of accuracy in almost all the frequency range analysed. Using the above definitions of A, B, C, D, it is possible to obtain an expression for the TL index. The transmitted pressure p_2^+ can be more easily determined if the outlet is non-reflecting (anechoic termination), that is if $p_2^- = 0$. Using the above definitions, the ratio p_1^+ / p_2^+ can be easily obtained together with the transmission loss:

$$\mathrm{TL} = 20\log_{10}[|\,\mathrm{A} + \mathrm{B}/\rho c + \mathrm{C}\rho c + \mathrm{D}\,|/2] \qquad (5)$$

The calculation is carried out in the frequency range 40-400 Hz, with a step of 40 Hz, for both the sets of analysis. The frequency range of interest is limited by the chosen mesh size (at least 4 quadratic or 8 linear elements per wavelength). Actually, this cut-off frequency of 400 Hz is lower than the maximum working frequency of the tested system (in fact, the harmonic components of the engine excitation lies in a range to reach 1000 Hz), so that a mesh refinement is needed in order to reach higher frequencies in the acoustic analysis.

4 BEM ACOUSTIC THEORETICAL FORMULATION

The basic equation for acoustic wave propagation through an elastic medium is the linear wave equation:

$$\nabla^2 u = \frac{1}{c^2}\frac{\partial^2 u}{\partial t^2} + b \qquad (6)$$

where u(x,t) is the velocity potential, c is the speed of sound, b(x,t) is the sound source, x and t are the position and time variables. Assuming that the problem is time harmonic, eqn (6) can be transferred to the frequency domain so as to obtain the Helmhotz equation:

$$\nabla^2 u + k^2 u = b \qquad (7)$$

where k=ω/c is the wave number and ω the angular frequency. Using the concept of a free field Green's function (v*, u*), the Helmhotz equation can be converted in the following integral equation, defined on the boundary [5-6]:

$$c(P)u(P) + \int_S v^* u dS = \int_S u^* v dS \qquad (8)$$

Eqn. 8 can be expressed in a boundary element formulation, in order to apply a numerical resolution method (in most cases the analytical treatment is overwhelmingly difficult):

$$c(P)u(P) + \sum_{N_{elements}} \int_{S_{element}} v^* u dS = \sum_{N_{elements}} \int_{S_{element}} u^* v dS \qquad (9)$$

where c(P) is dependent on the domain geometry, v is the fluid particle velocity and S the boundary surface. By substituting in eqn. (9)

$$u(x) = -p(x)\,/(i\omega\rho) \qquad (10)$$

where ρ is the mass density of the acoustic media, it is possible to write eqn. (9) in matrix form:

HP=GV+B (11)

where P and V are the vectors of nodal pressures and velocities on the BEM surface, while B is a body source vector. For a given velocity field on the panel, an acoustic BEM direct frequency response analysis calculates and stores the following data in the model database: pressure and normal velocity values in nodes on the BEM surface and at field points. It is worth to point out that the matrices H, G are fully populated, involving long run times for the system resolution. The pressure at an arbitrary field point is obtained by postprocessing surface pressure and normal velocity values: in this case only numerical integration is needed. There is one row and column for each boundary element node in the model and the matrices H and G are frequency dependent so as to require a full acoustic analysis for each frequency of interest.

If the fluid is not supposed to be conservative its physical properties are complex and consequently the solution is complex, existing phase relationships between the physical quantities like pressure and velocity [7], but this is not the case for our problem where an ideal fluid is considered.

An internal BEM acoustic analysis, in which it is sufficient to mesh the filter box by two dimensional surface elements (Fig. 3) provides the transmitted acoustic field. By a batch process, based on a BEASY routine (Beasyq.exe) and on in house made FORTRAN codes, it is possible to automatically run and postprocess an acoustic analysis for each frequency, reducing the user interaction in the pre- and post-processing phase. It is worthwhile to point out that for the acoustic analysis, the BEM mesh has to be sufficiently refined and, if necessary, such mesh could be optimised by a variable refinement depending on the frequency band considered.

For the acoustic analysis, the modelled fluid properties are shown in table 1.

density:	1.22E-6 kg/cm^3;
sound speed:	34000 cm/s;
reference pressure:	2E-9 N/cm^2.

Tab. 1. Fluid properties

5 FEM DYNAMIC ANALYSIS

If we consider an enclosed cavity, the boundary structural vibrations generate pressure variations in the fluid mass, causing the noise phenomena whose reduction and optimization is of capital importance in all the vehicle design steps. The dynamic response of a fluid mass inside a cavity is activated by the cavity structural modes but at the same time is influenced by its own dynamic behavior (fluid resonance), so as to involve a coupled dynamic analysis [8]. As a matter of fact, in general the dynamic behavior of the filter box should be studied by an FEM analysis in which the fluid is also modeled in a coupled fluid-structural analysis, but, when the fluid is air and its acoustic modes are not coincident with structural modes, it is possible to neglect such interactions (the structural dynamic behavior is not significantly influenced by the fluid inertia). With such an assumption it is possible to calculate separately the structural resonance by an FEM modal analysis (the NASTRAN code is adopted) and the results in the range 0-450 Hz are shown in Table 2 and Figs. 5-8. This results are useful to better explain the experimental TL behavior and the discrepancies with numerical results, as we will see in the following.

37 Hz	47 Hz	215 Hz	228 Hz	406 Hz	429 Hz

Tab. 2. Eigen-frequencies by FEM modal analysis.

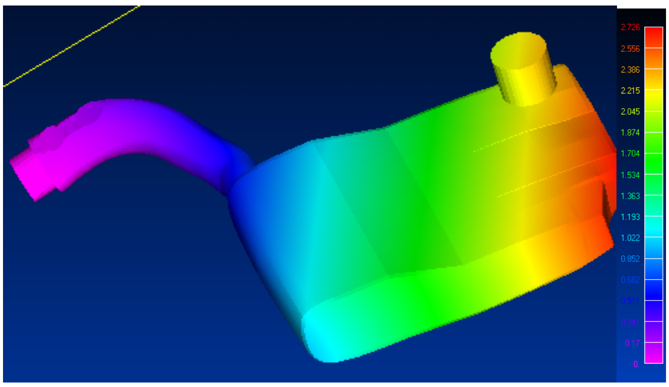

Figure 5. Mode 2 at 47.9 Hz.

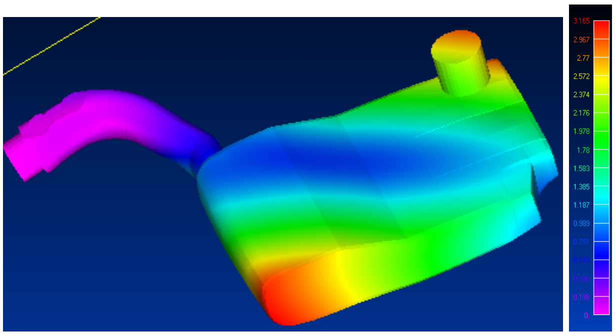

Figure 6. Mode 3 at 215.9 Hz.

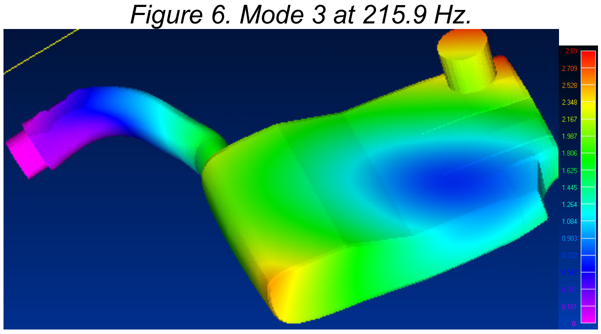

Figure 7. Mode 4 at 228.8 Hz.

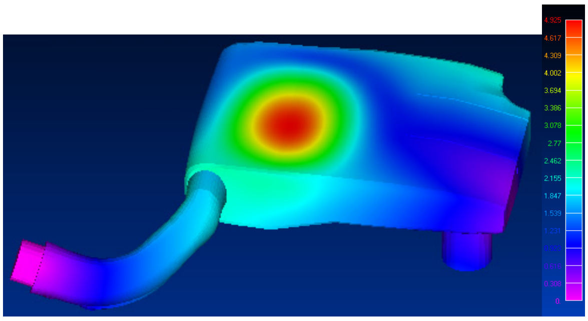

Figure 8. Mode 5 at 406 Hz.

6 BEASY vs VNOISE RESULTS

In figs. 9-15 the fluid resultant velocity (magnitude) is shown at different frequencies, as provided by the two BEM codes BEASY and VNOISE with reference to the boundary conditions v=1 at the inlet and p=0 at the outlet.

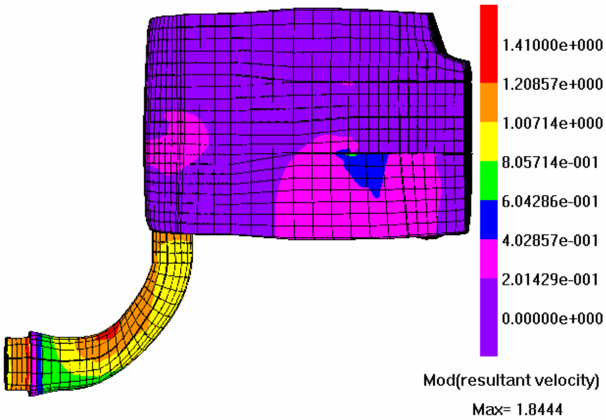

Figure 9. BEASY resultant velocity field (50 Hz).

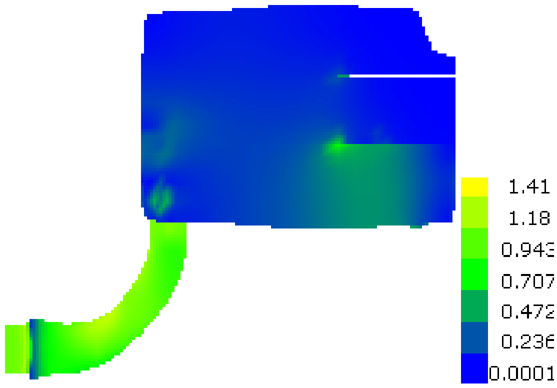

Figure 10. VNOISE resultant velocity field (50 Hz).

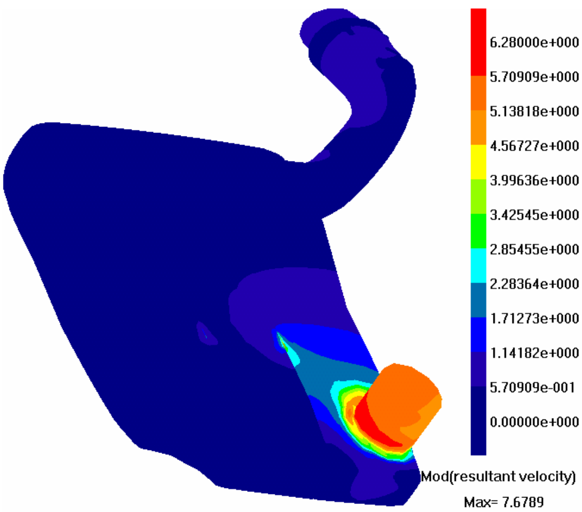

Figure 11. BEASY resultant velocity field (100 Hz).

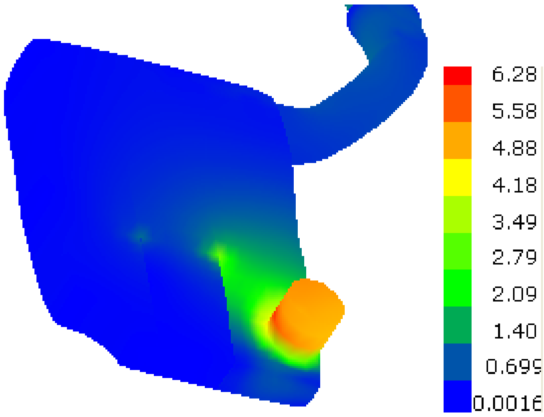

Figure 12. VNOISE resultant velocity field (100 Hz).

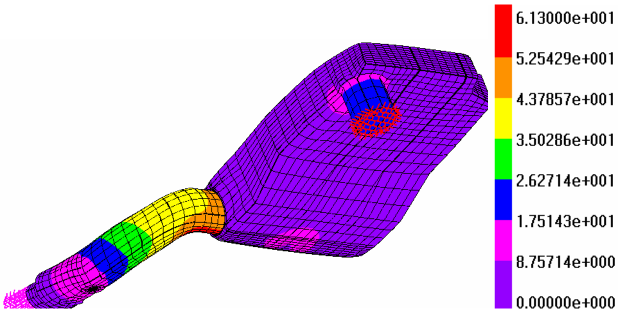

Figure 13. BEASY resultant velocity field (200 Hz).

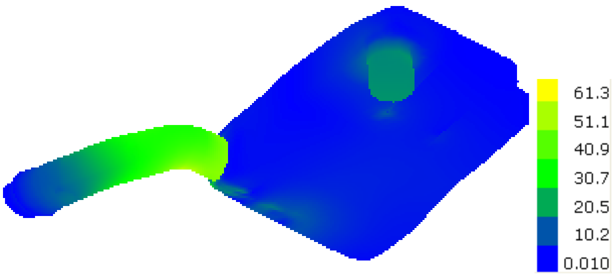

Figure 14. VNOISE resultant velocity field (200 Hz).

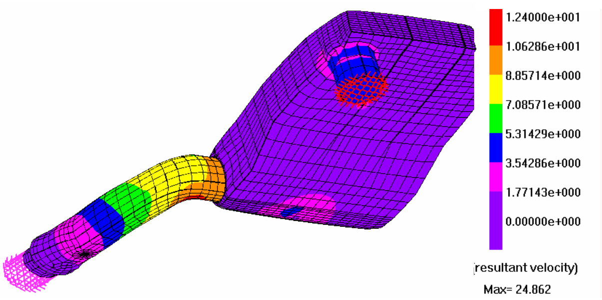

Figure 15. BEASY velocity field (200 Hz – Q38 elem).

At 200 Hz the frequency is very close to a modal value for the fluid, as visible from the high velocity values obtained (Figs. 13-14). It is interesting to observe that even if the mesh is consistent with the requisites of 8 linear elements per wave length it is not converging as proven in fig. 15, where the same mesh but with 8 node quadrilateral elements (Q38) is adopted and different results are produced.
From the above it comes out that the mesh refinement is particularly critical at resonant frequencies. Anyway a satisfactory correspondence between the two code results is appearing.

7 NUMERICAL (BEASY) - EXPERIMENTAL RESULTS

In figs. 16-17 the results related to a boundary condition of zero pressure at the outlet (and unitary velocity at inlet) are shown with reference to respectively outlet velocity field and inlet pressure: two cavity modes can be approximately envisaged at about 80 Hz and 180 Hz (due to the coarse frequency step it is difficult to provide a precise assessment).
The cavity mode around 180 Hz is confirmed (fig. 18) by the inlet and outlet pressure results related to a zero velocity boundary condition at the outlet (and unitary velocity at inlet).
In every graph the results, obtained with the mesh refinement shown in Figs. 2-3, but with alternatively 8 node (Q38) or 4 node (Q2) elements, are compared.
Another important graph is shown in fig. 19 where the quantity *Module(AD-BC)* is plotted in order to countercheck how close to unity are these values and highlight possible mismatch (like at 100 Hz) were a further deepening is recommended.

The TL numerical and experimental results are expressed by the filter box transmission loss against frequency as reported in fig. 20: as expected, due to the lack of fluid–structure coupling allowance, strong differences appear around cavity or structural mode frequencies but especially when they turn out to be coincident (coincident acoustic and structural modes). In particular, based on the experimental data, the filter box acoustic insulation capabilities exhibit a broad attenuation in the 40-80 Hz frequency range where 2 structural modes (37 Hz and 47Hz) and at least one cavity mode (80 Hz) are present; clearly in this range the fluid-structural coupling allowance is mandatory. The same can be observed with reference to the frequency range 180-250 Hz where again two structural modes (215 Hz and 228 Hz) and at least one cavity mode (180 Hz) are present, causing a TL decrease starting from about 210 Hz. The latter experimental outcome is contradicted by the numerical results where the TL decrease starts later at about 280 Hz, showing a continuously increasing TL up to that frequency, probably because no allowance is made for such fluid-structure interaction.

From fig. 20 it is also possible to observe that the linear mesh can be considered sufficiently accurate up to 300 Hz (except around 180 Hz where a fluid resonance occur): with such mesh each frequency is resolved by BEASY in few minutes with a powerful PC, asking for few hours to span the whole frequency range considered. After 300 Hz the quadratic mesh is advisable and can be usefully applied up to about 550 Hz, where a mesh refinement becomes mandatory. With the latter the number of nodes increase to 10412 and the related run times of one order of magnitude.

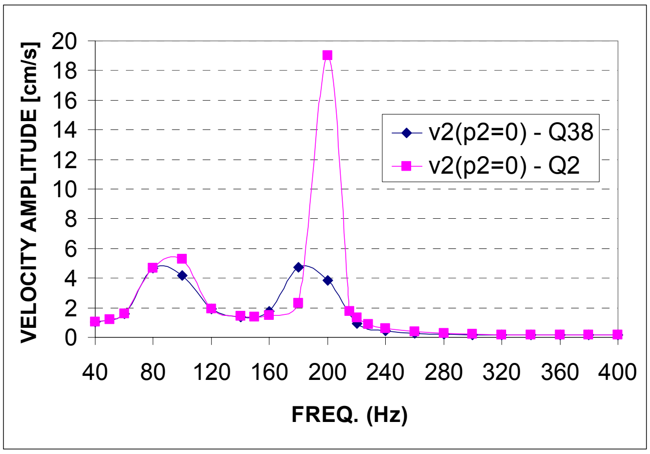

Figure 16 – Numerical outlet average velocity field.

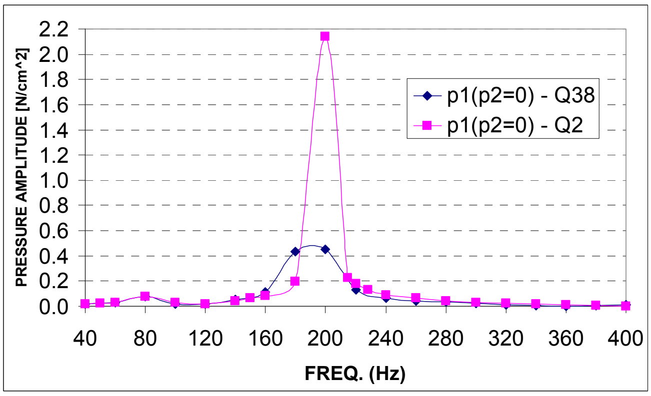

Figure 17 – Numerical inlet average pressure.

8 CONCLUSIONS

An acoustic analysis is performed by estimating the TL factor both numerically and experimentally. The system noise attenuation is measured and compared to the results of the application of two numerical methodologies.

At this intermediate stage of the procedure development the level of accuracy, provided by the BEASY code against the experimental evidence, can be considered acceptable, except around resonant frequencies where a coupled approach is mandatory.
There are many opportunities given by a BEM approach to easily solve vibration acoustic problems but most of them are related to a coupled usage of FEM too.
The required hardware resources do not prevent from extending the frequency range considered, provided that a more refined BEM mesh is adopted.

Future efforts will be devoted to explore a wider frequency range and to include the structural-fluid coupling; the latter being particularly critical around fluid-structure resonance frequencies.

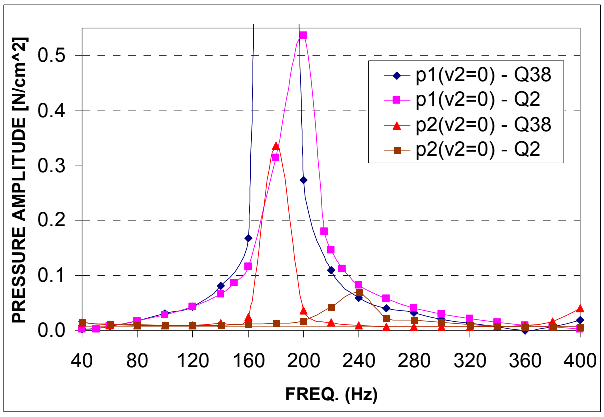

Figure 18 – Numerical inlet and outlet average pressure.

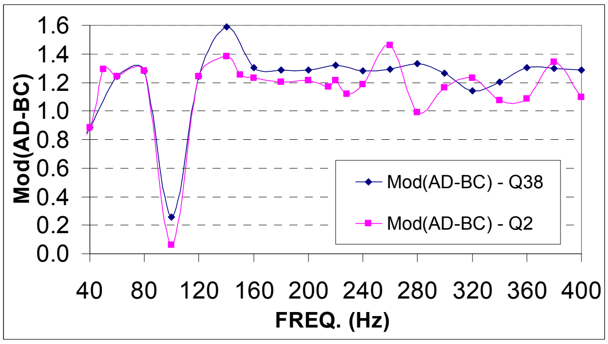

Figure 19 – Counter check parameter AD-BC.

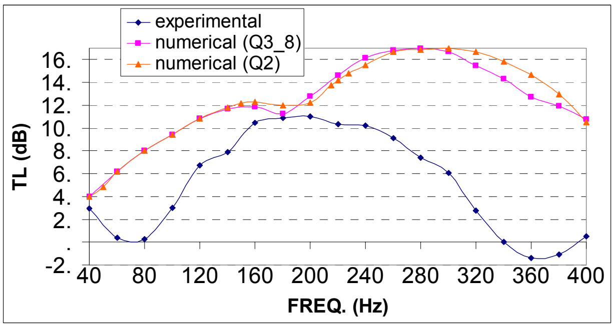

Figure 20 - TL factor (dB) versus frequency by the BEASY software vs experimental results.

9 REFERENCES

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