Transmission loss assessment by integrated FEM-BEM methodology

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Abstract

Nowadays, one of the most valuable criteria for vehicle quality assessment is based on acoustic emission levels: a car is judged comfortable also depending on the noise level transmitted inside. Consequently there is a general attention to design criteria aimed to improve the structural-acoustic behaviour in such a way to withstand the increasingly restrictive ergonomic standard. Such design approach, based on experimental and numerical procedures, enables the prediction of noise emissions and the correlation with the structural vibration source. The following step is the redesign of those components responsible for intolerable emissions, without the need for an extensive and expensive prototyping effort. This work, realised in collaboration with ELASIS S.C.p.A. (which provided the experimental data), is aimed to set up and validate a numerical procedure for a vehicle component transmission loss assessment, by a synergetic and integrated use of Boundary Element (BE) and Finite Element (FE) methods. In particular such procedure is applied to predict the transmission loss of a steel panel whose properties were previously assessed by experimental measurements in an anechoic-reverberant room. The panel structural behaviour is assessed by an FEM modal analysis and the calculated normal velocities are then imposed as boundary conditions for a BEM analysis of the transmitted noise field.

1 Introduction

The products of many industries can benefit from improved acoustic design. 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). Such procedures can be applied to provide the acoustic engineer with the information necessary to ensure that the design satisfies performance specifications and regulations imposed by governments and standard bodies. The use of computer simulation offers major advantages over trial and error prototyping in the quest for improved products (e.g. quieter products)[1]. While FEM has been used with success for interior problems (e.g., computing the sound field in the vehicle finite domain) [2], BEM performs better for exterior problems (e.g., radiation from a component in an infinite domain) [3]. When the analysis, instead, is aimed to solve a combination of interior/exterior problems the best approach probably resides in a combined use of the two methodologies [4]. In both FEM/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.

This work is aimed to build up a reliable numerical procedure for the transmission loss prediction of a simple structural component such as a panel. Experimental results are available from ELASIS S.C.p.A. laboratories and will be compared with the numerical ones for procedure validation. The FEM code ANSYS is used in combination with the BEM code BEASY in a coupled FEM-BEM approach. In particular the panel structural behaviour analysis is demanded to FEM, while the panel noise radiation in the infinite surrounding domain is better faced by BEM. BEASY code also provides powerful diagnostic facilities to enable users to identify the main sources contributing to the sound intensity at any point of interest [5].

2 Acoustic analysis

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

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

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 (1) can be transferred to the frequency domain so as to obtain the Helmhotz equation:

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

where $k=\omega/c$ 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 [6]:

$$c(P)u(P) + \int_S v^* u dS = \int_S u^* v dS \qquad (3)$$

Eqn. 3 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):

$$c(P)u(P) + \sum_{N_{elements}} \int_{S_{element}} v^* u dS = \sum_{N_{elements}} \int_{S_{element}} u^* v dS \tag{4}$$

where *c(P)* is dependent on the domain geometry, *v* is the fluid particle velocity and *S* the boundary surface. By substituting in eqn. (4)

$$u(x) = -p(x)/(i\omega\rho) \tag{5}$$

where ρ is the mass density of the acoustic media, it is possible to write eqn. (4) in matrix form:

$$HP=GV+B \tag{6}$$

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]

3 Problem description and modelling

A rectangular panel is placed as a separation between a reverberant room, where a uniform acoustic field is artificially generated, and an anechoic room where it is possible to measure the transmitted (by the panel vibrations) acoustic field (Fig. 1). In the reverberant room a rotating microphone provides the pressure level frequency spectrum (Fig. 2) and allows to keep under control, in the frequency range considered, the pressure level and its uniform spatial distribution (with the exception of the first 2 bands the reverberant acoustic field hypothesis is approximated within a sound pressure level variation of 2 dB). This spectrum is necessary in order to define the loading boundary conditions for a dynamic FEM analysis of the panel (Fig. 3), whose output will be the normal structural velocities for each frequency and for each node in the FEM mesh. Such output nodal velocities are arranged (to this aim a FORTRAN code was written) in a previously generated “velocity file” that can be automatically read by the BEASY code, in order to impose the boundary conditions on the BEM panel model. An external BEM acoustic analysis, in which it is sufficient to mesh a void (Fig. 4) in an infinite domain (the panel is the largest surface of such void), provides the transmitted acoustic field. As a matter of fact, infinite fluid, simulated by the anechoic room when the frequency considered is higher than 100 Hz, does not need a BEM mesh process as when using FEM, getting a considerable saving in pre-processing and running times. In general it is not necessary to use the same panel mesh for the FEM structural analysis and for the BEM acoustic analysis that generally require a lower number of elements. To implement such option, there is a BEASY interface routine [8] capable to interpolate the velocity values calculated on the FEM mesh, in order to correctly attribute such velocities to different geometric points of the BEM mesh. Actually, such routine is effective when the FEM code is NASTRAN, but in this analysis we operate with ANSYS [9] and consequently, in order to simplify the FEM-BEM coupling (it is not strictly necessary), a mesh with the same number of nodes and placed in the same position, is adopted for the two acoustic and structural problems. The number of linear FEM elements is four times that of BEM quadratic elements (Figs. 3-5): such arrangement is necessary because in ANSYS a 9-node quadratic element was not available. For each frequency band, the recorded sound pressure level is imposed on the FEM panel model, and, for each frequency in the range 112-630 Hz, the velocity field spectrum is obtained and post-processed in order to generate the BEASY input file. The pressure values, calculated by BEASY on a predefined display patch (parallel to the panel), for each frequency in the spanned range, are recorded in an output file and postprocessed, in order to automatically generate the desired “transmission loss”. In such analysis the panel normal velocities are supposed coincident with the velocity of the fluid molecules in contact with the panel. The acoustic analysis is carried out for each frequency in the range 112–630 Hz and numerical results are grouped in 1/3 octave bands, in order to make them comparable with the 1/3 octave bands experimental results. By a batch process, based on a BEASY routine (Beasyq.exe) [8] 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.

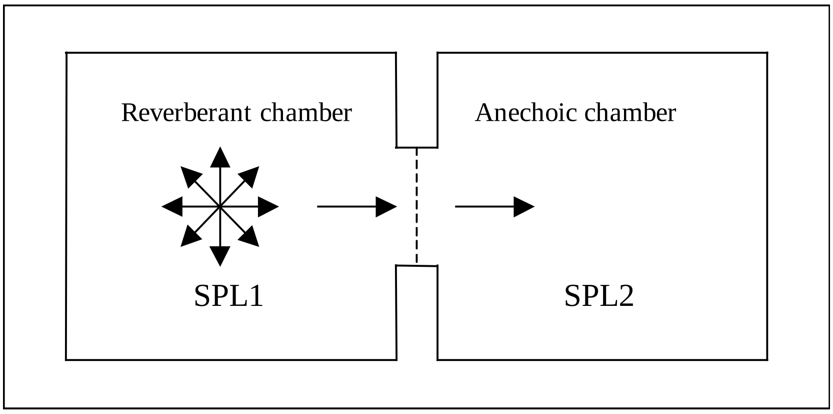


Figure 1: Reverberant-anechoic room with interposed panel

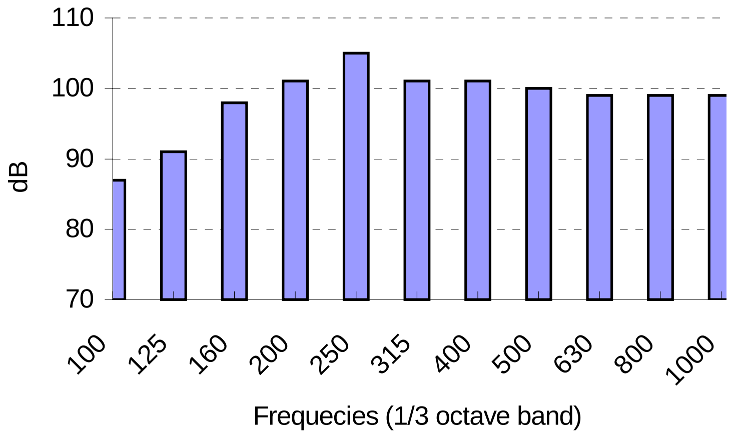


Figure 2. Sound pressure levels in reverberant room, from experimental measurements

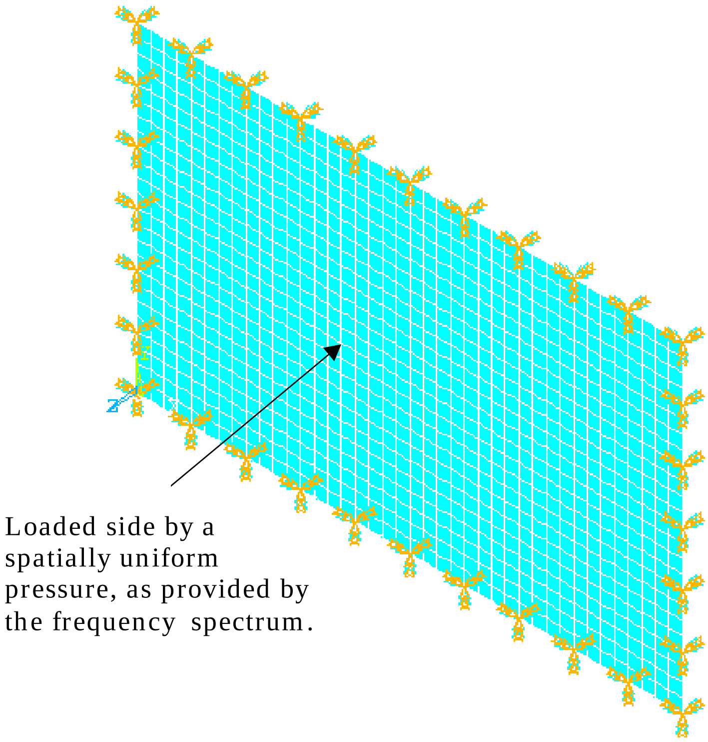


Figure 3: Panel FEM model and boundary constraints

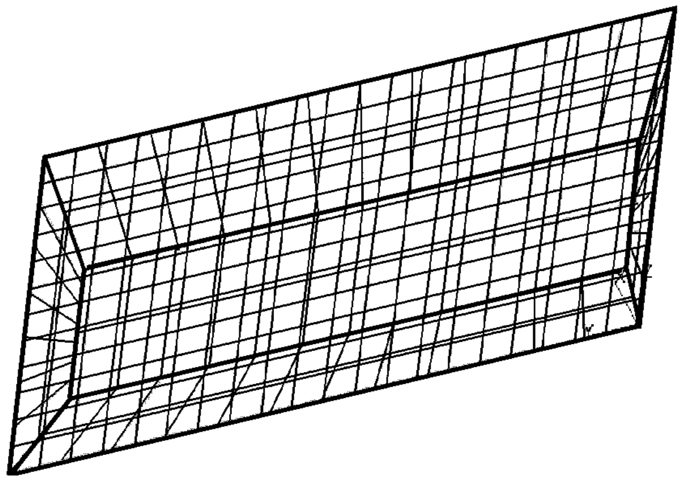


Figure 4: Cavity in an infinite domain with the largest surface simulating the vibrating panel.

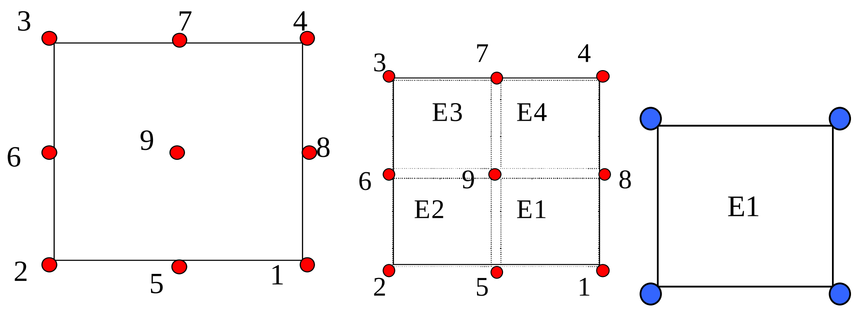


Figure 5: *left*: quadratic element used for the BEM mesh; *centre*: arrangement of linear shell FEM elements to reproduce the quadratic (Q3) BEM element; *right:* linear (shell 63) element used for the FEM mesh.

3 .1 FEM dynamic analysis

If we consider an enclosed cavity (e.g. inside a car body), the boundaries structural vibrations can generate pressure variations in the fluid mass, causing the internal noise phenomena whose reduction and optimisation 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 behaviour (fluid resonance), so as to involve a coupled dynamic analysis [2]. As a matter of fact, in general the dynamic behaviour of the panel should be studied by an FEM analysis in which the fluid is also modelled 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 panel dynamic behaviour is not significantly influenced by the fluid inertia). With such an assumption it is possible to model the acoustic load as uniform on the panel, with a pressure level variable for each frequency considered (as recorded by the moving microphone in the reverberant room). The procedure developed for an acoustic analysis of vibrating structural components starts with the dynamic assessment of the component behaviour by an FEM modal analysis, whose results are compared with the experimental measurements in order to validate the model.

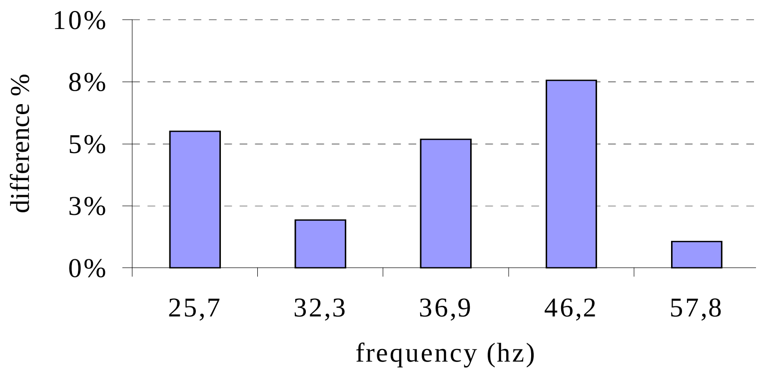
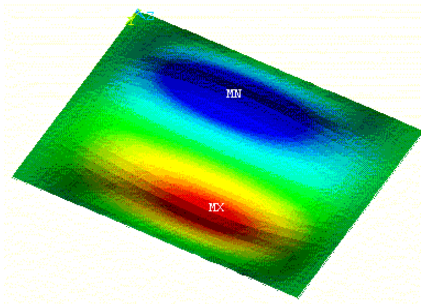


Figure 6: Differences between experimental and numerical (calculated with ANSYS) eigenvalues for modes one to five.



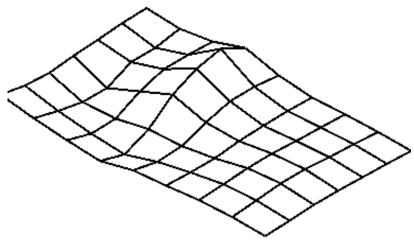


Figure 7: First vibration mode: numerical (on the left) and experimental (on the right).

The FEM panel model is fully constrained on 32 uniformly spaced points along the borders (Fig. 3), consistently with the welding points applied to the real panel. The following input data were used:

- panel dimensions : 1760x1020x2 mm;
- material : steel;
- density : 7850 kg/m^3;
- damping : 0.005

- Young modulus : $2.1E+05\ N/mm^2$;
- Poisson ratio : 0.3
- ANSYS element type : Shell 63

The panel damping value was not precisely known so that an average value, typical of that kind of steels, was chosen. Such approximation does not produce appreciable errors, but close to the "coincidence frequency" that is out of the frequency range considered. The reason is that the plate dynamic is damping controlled only at resonance, whilst for lower frequencies it is controlled by the plate inertia. This is also confirmed by a sensitivity analysis whose results are presented for just one band, the one centred at a frequency of 160 Hz (Table 1).

Table 1. Sensitivity analysis for transmission loss against variable damping ratios, at 160 Hz frequency.

Damping	0.3%	0.5%	0.7%
Transmission loss (dB)	25.1	25.2	25.3

3 .2 BEM acoustic analysis

It is worthwhile to point out that for the acoustic analysis, the BEM mesh has to be sufficiently refined depending on the higher frequency of interest (at least four quadratic elements per wavelength). If necessary such mesh could be optimised by a variable refinement depending on the frequency band considered. Moreover the void shape is such to minimise the reflecting border effects on the panel emission: the vibrating panel is the larger frontal surface (Fig. 4) and has a more refined mesh with respect to the other secondary boundaries which have a zero displacement conditions. For the acoustic analysis, the modelled fluid properties are the following:

density: $1.22\ kg/m^3$;
sound speed: 344 m/s;
reference pressure: $2E\text{-}0.5\ N/m^2$.

4 Numerical results

The differences between the mode number one to five (such modes are the most important in the overall panel behaviour) are highlighted in Fig. 6, while in Fig. 7 a comparison between the first numerical and experimental vibration mode is depicted.

The numerical and experimental results are expressed by the panel transmission loss against frequency as reported in Figs. 8-9: differences are within 10%.

5 Conclusions

It is possible to point out the satisfactory agreement provided by the implemented numerical procedure with the experimental evidence and the opportunities given by an FEM-BEM coupled approach to easily solve vibration acoustic problems. The required hardware resources do not prevent from extending the frequency range considered, provided that a more refined FEM and BEM mesh is adopted and an higher number of well positioned microphones installed in anechoic room, in order to capture all the acoustic energy radiated by the panel [10-11]. The following step will provide extension of such procedure to complex components such as car bodies.

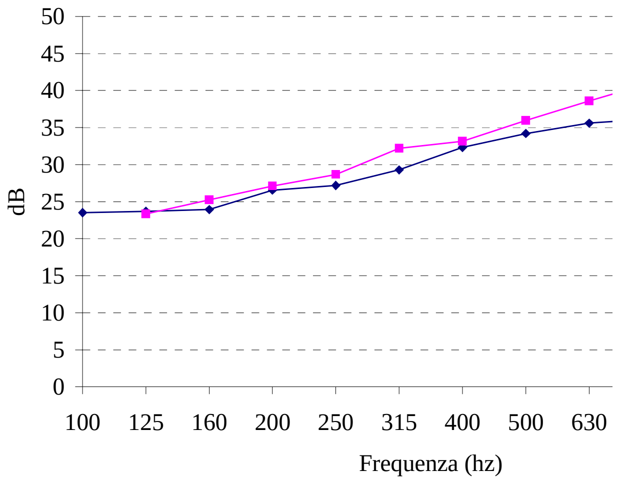


Figure 8: Experimental (lower curve) and numerical (higher curve) transmission loss.

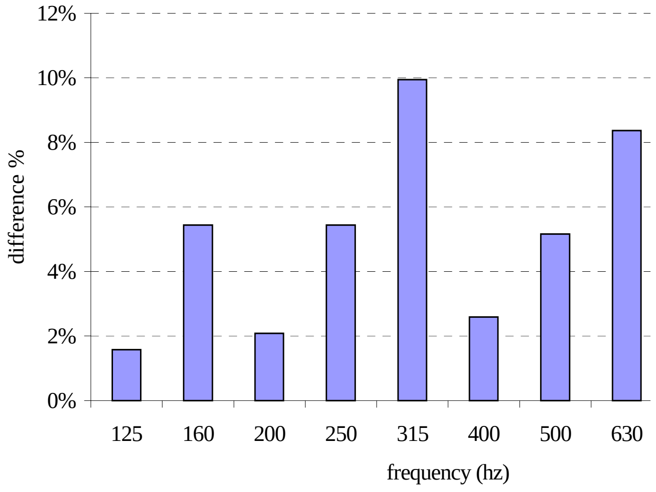


Figure 9: Differences between numerical and experimental transmission loss.

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