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A REFINED BE TECHNIQUE FOR MODELLING RADIATION AND SCATTERING FROM VIBRATING STRUCTURES: EXPERIMENTAL VALIDATION OF BEM RESULTS OBTAINED USING LASER DOPPLER VIBROMETRY

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ABSTRACT
In this paper, a refined Boundary Element technique, developed to model radiation and scattering from vibrating structures in the frequency domain, is validated in the case of a small reverberating room by comparison with acoustic measurements. An experimental set-up was built consisting of a box-shaped structure, with a wall excited into vibration by an electro-dynamic shaker. A fairly good agreement was found between numerical and experimental results. In order to ease the definition of boundary conditions in the case of experimental vibration data, a special interface with a Scanning Laser Doppler Vibrometer was also developed.

1 - BE TECHNIQUE
A variational form of the integral equation of fluid-structure interaction in the frequency domain was adopted in a previous work [1] to model the acoustic radiation from vibrating structures. The starting point was the acoustic equation written in the classical Helmholtz form in the frequency domain, which reads:

$$\nabla^2 P + k^2 P = 0 \qquad (1)$$

where P is the Fourier transform of the pressure and k is the wavenumber.
The function $P(x_1, x_2, x_3)$, solution of equation (1), can be expressed by specific representation formulae [2, 3]:

$$\begin{array}{ll} p_e(R) = -\dfrac{1}{2} d(R) - \displaystyle\int_s \frac{\partial G(R,r)}{\partial n_r} \cdot d(r) \cdot dS_r + \int_S G(R,r) \cdot f(r) \cdot dS_r & r \in S \\ \dfrac{\partial p_e(R)}{\partial n_R} = -\dfrac{1}{2} f(R) - \displaystyle\int_s \frac{\partial^2 G(R,r)}{\partial n_r \partial n_R} \cdot d(r) \cdot dS_r + \int_S \frac{\partial G(R,r)}{\partial n_R} \cdot f(r) \cdot dS_r & r \in S \end{array} \qquad (2)$$

where S is the surface enclosing the solution domain, $\vec{n}$ is the surface outward normal, $d(R)$ and $f(R)$ are the acoustic dipoles and sources distribution respectively, $p_e(R)$ and $\partial p_e(R)/\partial n_R$ are the pressure and its normal derivative on the positive side of the surface and $G(R,\ r)$ is the Green function of the Helmholtz equation. The representation formulae can be written in a variational form [4, 5] and solved in conjunction with appropriate boundary conditions using an iterative method for the numerical integration.

In the work mentioned [1], the Richardson extrapolation formula was used to calculate the surface integral, allowing to obtain satisfactory results also in the case of zero order elements and to overcome the element size-frequency ratio limit.

2 - EXPERIMENTAL VALIDATION

It was decided to consider the propagation in a reverberant field for the validation. In order to have in practice this fluid propagating condition and also a mechanical noise source to excite the fluid, a box-shaped structure with a vibrating side was built (see Figure 1).

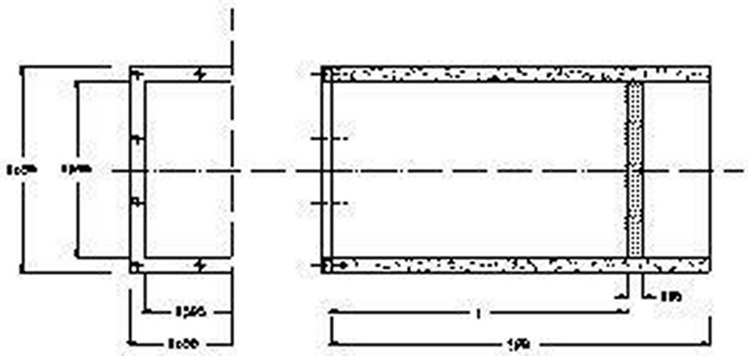


Figure 1: Box-shaped structure.

The two bases of the parallelepiped-shaped room were built using a 1 mm thick zincate plate for the vibrating wall and a plate of medium density fibre board (Mdf) for the opposite wall. The remaining four sides were built of concrete and internally treated with three layers of cement mortar, while the Mdf wall was coated with lac. The zincate plate was clamped to a metallic frame and then to the structure by means of screws.
The experimental set-up used for vibration measurements is illustrated in Figure 2. A shaker excited into vibration the metallic plate. In order to excite a lower number of modes, it was positioned at the plate centre.
The test was conducted using a random input signal in the 2-2000 Hz frequency range. A piezoelectric accelerometer was fixed to the plate and used as reference.
In order to gather vibration velocity data and use them as input for the acoustic field computation [6, 7], a series of vibration measurement was performed on the metallic plate. In particular, a grid of 36 points equally spaced on the plate was defined via computer through the use of the CCD camera placed inside the Vibrometer head. The vibration velocity spectrum relative to the plate is reported in Figure 2, together with some peak velocity distributions.
A B&K Falcon type 4188 microphone was introduced into the cavity through a small metallic tube inserted in the Mdf panel and positioned in correspondence of the cavity axis. It allowed to perform acoustic pressure measurements along the axis while the plate was vibrating. In particular, the different measurement points were positioned 10 cm spaced along the axis. A 4 channels PULSE system from B&K allowed to acquire the pressure signal coming from the microphone and to suitably post-process it. The fundamental acoustic modes in the longitudinal direction were clearly visible in the spectra, as well as the first two modes along the transversal direction of the cavity.
A Finite Element representation of the test structure was created through a commercially available dedicated software (ANSYS©). The mesh consisted of 250 nodes and 5 ×5 ×9 SHELL63 elements. The element length was coincident with the distance between the measurement points on the vibrating plate. This allowed to compute the acoustic pressure field accurately up to 510 Hz.

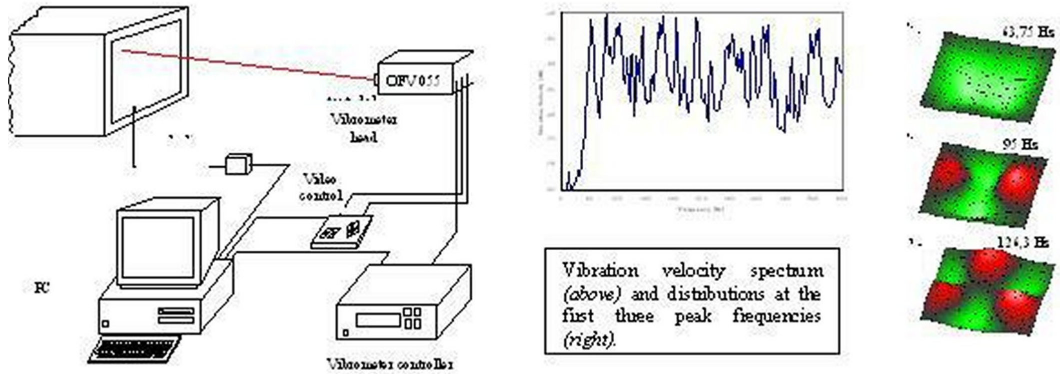
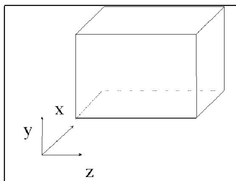


Figure 2: Experimental set-up for the vibration measurements.



Mode	Theoretical [Hz]	Experimental [Hz]	Numerical [Hz]
0;0;1	171.5	174	172.5
0;1;0	288.23	296	291.25
0;0;2	343	346	356.25
0;0;3	514.5	519	512.5
0;2;0	576.47	590	591.25
0;2;1	601.47	614	605

Table 1: Comparison between mode frequencies.

The numerical results were finally compared to the experimental ones. In Figure 3 the numerical and experimental pressure diagrams are superimposed. It can be seen that there is a substantial agreement between the two sets of data, either in frequency (see Table 1) or in amplitude.
The difference reported in correspondence of the first transversal mode frequency at 70 cm from the plate can be justified as follows. Due to the small dimension, compared to the acoustic wavelength at the frequency under consideration, of the test structure along the transversal direction, the measurement is largely sensitive to any small error committed in positioning the microphone and then the pressure amplitude is greatly affected by this error. It can happen therefore that the pressure value is not null along the axis, as it should be in the case of the first transversal mode.

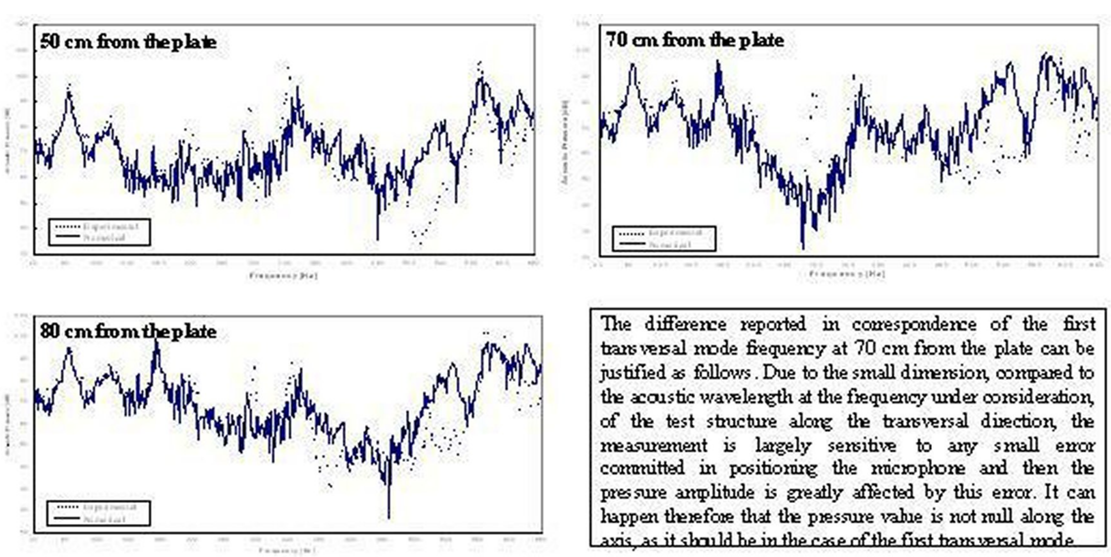


Figure 3: Comparison between numerical and experimental pressure spectra.

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