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NUMERICAL SIMULATION OF LABORATORY SOUND INSULATION DETERMINATION EXPERIMENTS

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1. INTRODUCTION

During the last years low frequency noise has become increasingly important. In most sound sources affecting dwellings (e.g. traffic noise or amplified music) lower frequencies are largely present. Therefore several national standardisation committees are planning to include the low frequency range (50 to 100 Hz) in guide-lines for measuring the transmission loss (TL) of partitions. At such low frequencies the sound field in a typical room is not diffuse, and the statistical approach can not be applied to its description. This has several consequences. On one hand, there has been some confusion caused by the fact, that the results of the sound insulation evaluation obtained for the same partition at different laboratories, differ significantly. On the second hand, other methods, referring to wave theoretical approach have to be used for simulation of experiments. To study the phenomena at low frequencies tools are necessary too describe the sound field and its interaction with the structure under test properly. Although the problem is not new, in the recent paper Atalla & Bernhard [1] complained about "relatively few realistic verification exercises" of numerical solutions for low-frequency structural-acoustic problems. The current paper can be treated as such verification. Two rooms coupled by a flexible partition constitute a classical task in building acoustics. Attempts of modelling it at low frequencies has been reported recently by Gagliardini *et al.* [2]. Problems concerning the meaning of sound reduction index of partitions at low frequencies has recently been addressed by Kropp *et al.* [3]. In the current paper, simulations with several numerical methods will be compared to the measurement results for two different laboratory configurations: one with equal rooms and one with sending room and receiving room significantly different. Attention will be paid to the details of the placing of the partition under test (possible niche and supporting frame), as well as, to its construction. The accuracy of the simulations will be evaluated, and the computational effort will be discussed.

2. TEST CONFIGURATIONS

The simulations concerned two Sound Transmission Laboratories. The first one consisted of two equal rooms and the other one of two rooms that were significantly different from each other. The geometry is shown in Fig.1. for the first case and in Fig.2. for the second case. In both cases, the measurements were carried out with a single leaf partition consisting of two layers of gypsum boards (total thickness 0.026 m) supported by steel studs. In both cases the partition under test did not cover the whole opening between rooms. In both cases the partition was placed in a niche in the supporting frame (i.e. at least at one side not flush with the frame's surface).

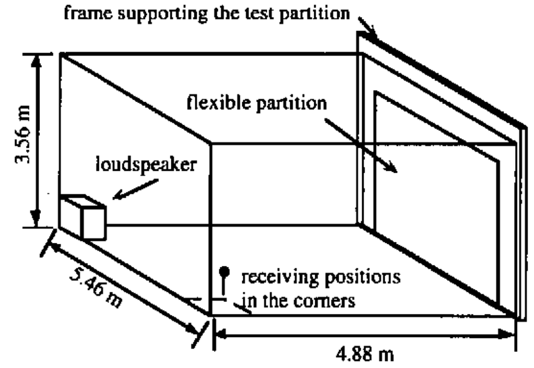

Fig. 1. The sending room of the sound transmission laboratory - case 1.

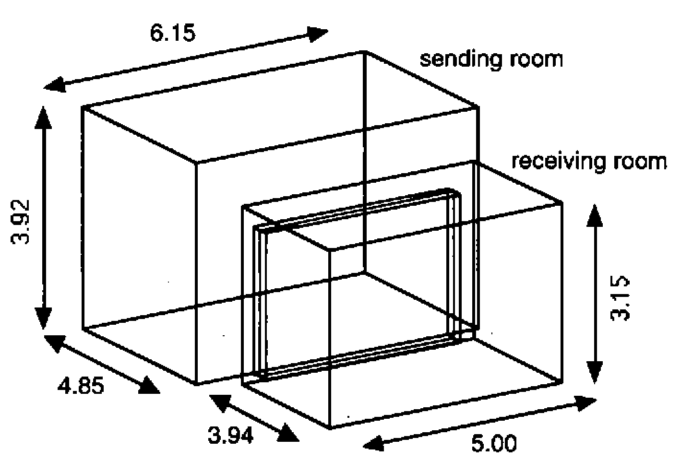

Fig. 2. The geometry of the sound transmission laboratory - case 2.

In case 1 the loudspeaker was placed in the corner of the sending room and the SPL was measured in the specified corner positions (0.30 m from each wall). In case 2 a loudspeaker moving along a line was used and the SPL was measured by means of two rotating microphones in each room.

3. NUMERICAL MODELS

The Finite Element model

The application of the Finite Element method to the solution of the coupled structural-acoustic problem is discussed in the literature since the sixties (Gladwell [4], Craggs [5], Sandberg [6], Pietrzyk *et al.* [7]). Most of the commercially available FE packages, which allow acoustic analysis, use pressure as a nodal variable. Other formulations are possible, and may indeed be advantageous for coupled problems (e.g. Sandberg [6]), but they are not commercially available. Coupling between acoustic pressure field and structural displacement field is provided either by special options to structural elements or by specific interface elements. Sound source modelling can be done either by defining the particle acceleration's amplitude, or by forcing structural vibration on the part of structure adjacent to the acoustic fluid domain. For the description of structural damping the Rayleigh damping model is usually offered, enhanced with the discrete dashpot elements. For the description of damping phenomena in acoustic fluid either the surface impedance, or the flow resistivity can be used.

For many types of partitions, especially for lightweight ones, the wavelength of the bending waves at low frequencies is substantially smaller then the wavelength of the sound waves in air. While modelling both the fluid and the structure with FE (FEM/FEM) one is faced with the problem of 3D mesh generation. The mesh in the fluid domain has to match the mesh at the partition, but the element size can than increase with the distance from the partition in order to reduce the number of unknowns. Unfortunately, generation of unstructured 3D meshes is not a solved problem and is not offered in the FE pre-processors. One possible solution might be to model the partition as locally reacting with mass impedance (FEM/LRMI). This may seem to be a crude approximation, but it proves justified at low frequencies, and may be attractive while dealing with irregular rooms. The FEM will be demonstrated with ABAQUS [13], and SYSNOISE [14].

The Boundary Element model

Using the Boundary Element method a substantial reduction of the number of unknowns is achieved. Unfortunately, the resulting equation system, although smaller, is fully populated, and the solution for interior problems takes longer time than for the equivalent FE model. The BE model has, however, a practical advantage - mesh generation is restricted to 2D. There exists efficient and reliable mesh generators for automatic creation of complex, unstructured 2D meshes. Thus, the overall solution cost, including the human labour, can be significantly smaller for complex geometry or strongly irregular meshes. This method will be demonstrated with SYSNOISE [14].

The Modal Approach

The so called 'Modal Approach' follows the model used by Donner [8]. The sound field in both rooms is described by modal functions in the width (y-direction) and height (z-direction) of the rooms where the boundary conditions are given as rigid. This can be viewed as introducing waves with the wave fronts of the shape of modes in y- and z-directions, and unknown (complex) amplitudes, travelling along the x-direction. The velocity component in the x-direction is obtained from the momentum conservation equation. The damping is introduced by defining a flow resistivity. The unknown (complex) amplitudes are determined by the (velocity) boundary conditions. The boundary conditions in the length (x-direction) of the rooms are given by the conditions at the back walls (e.g. given velocity distribution) and by the conditions at the partition. There the sound fields in both rooms are coupled by the impedance of the partition.

The boundary conditions will be fulfilled in discrete points, the so called collocation points. The number of collocation points depends on the number of modes which one intends to consider. Although from the mathematical point of view the collocation method may not be satisfying (convergence is not guarantied by increasing the number of modes (Zurmuehl [9])), it is quite well tempered (see, e.g. Munjal [10]).

Partition: Finite Element model (MA/FEM). The FE calculation delivers a mobility matrix (i.e. the velocity response at all collocation points to excitation in all collocation points) for the partition. The boundary conditions at the partition are then formulated as

$$v_I(x=l_{Ix},y_e,z_e,\omega)=\sum_i\left(Y_{ie}^{I,I}p_I(x=l_{Ix},y_i,z_i,\omega)+Y_{ie}^{II,I}p_{II}(x=0,y_i,z_i,\omega)\right) \quad (1)$$

$$v_{II}(x=0,y_e,z_e,\omega)=\sum_i\left(Y_{ie}^{II,II}p_{II}(x=0,y_i,z_i,\omega)+Y_{ie}^{I,II}p_I(x=l_{Ix},y_i,z_i,\omega)\right) \quad (2)$$

where $Y_{ie}^{I,I}$, $Y_{ie}^{I,II}$, $Y_{ie}^{II,I}$, and $Y_{ie}^{II,II}$ are the transfer mobilities from the excitation point "i" to the receiving point "e". The indices I,I and II,II indicate that excitation and receiving points are on the same side (i.e. in the sending or the receiving room). The cross mobilities $Y_{ie}^{I,II}$ and $Y_{ie}^{II,I}$ (i.e. from the receiving room to the sending room and vice versa) have to be identical due to reciprocity. The method offers the opportunity of studying more complex partitions without the high computational effort which the FE model needs for describing the sound fields in both rooms. Since the mobilities are calculated for partitions *in vacuo* the computational effort can be reduced compared to the FEM/FEM model.

Partition: locally reacting with mass impedance (MA/LRMI). For a partition with pure mass impedance the boundary conditions are:

$$p_I(x=l_{Ix},y,z,\omega)-p_{II}(x=0,y,z,\omega)=Z(\omega)v_p(y,z,\omega) \quad (3)$$

where v_p is the velocity component of the partition in the normal direction (i.e. x direction) and $Z(\omega)=j\omega m''$ the mass impedance. m'' is the mass per square meter. Additionally, the x-component of the partition's velocity equals the x-component of the particle velocity in the sound fields on both sides of the partition. This is the fastest method of those considered in the current paper. The results are very good, at least for the examples presented here.

5. SIMULATION RESULTS

The results of the simulations are presented as a difference between averaged sound pressure levels in the sending and the receiving room. In Fig.3 the measurement results are compared to the results of simulations with both FEM/FEM and MA/LRMI techniques. The results are given as 3rd octave band spectra. For the first case the agreement is fairly good. The FEM/FEM approach gave better results, as expected. In this model the gypsum boards were treated as a flexible plate and interaction with supporting studs was taken into account. The MA/LRMI approach gave exactly the same results as FEM/FEM implementing the simplified model with no studs and very soft plate as a partition. The computational effort was, however, incomparable. It took 6 hours CPU to complete the MA/LRMI calculation with 11x11 modes, as compared to 20 CPU hours that the FEM/FEM approach (25000 dof) required. The memory and disk space requirements were also much more modest for the MA/LRMI (see [11] for the details concerning the simulations for this laboratory).

For the second case the results of the simulation were poorer. It can be due to the fact, that in the simulations only one loudspeaker position was used, while the loudspeaker was moved along a line in the experiment. Also, it was hard to use exactly the same positions for the evaluation of the averaged SPL in the rooms in simulation and in experiment, as rotating microphones were used during the measurements. The simulation should be repeated at least for several loudspeaker positions and the results should be appropriately averaged. Also, the influence of the studs was not analysed in the second case. Further simulations are being carried out, concerning e.g. the influence of the niche, the size of the test partition, and the treatment of the backwall in the receiving room [12]. The updated results will be given during the presentation at the conference.

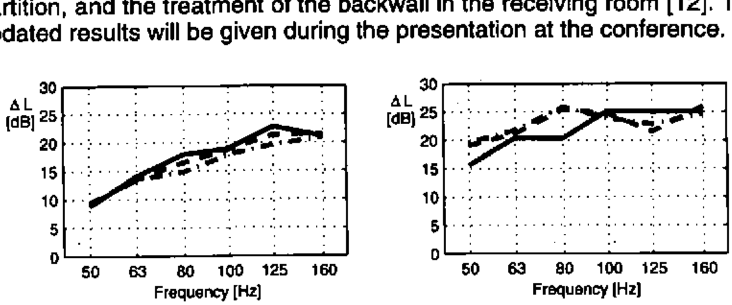

Fig. 3. Comparison between measurement data and simulation results for the two laboratories: case 1 - left, case 2 - right; measurement - solid line, simulation FEM - broken line, simulation MA - dotted line

6. CONCLUSIONS

The numerical methods based on wave theoretical acoustics proved successful in modelling the experiments. All methods tested gave the same results provided the same underlying model was used. The Modal Approach proved most efficient in case of the limp plate model for the partition. For plate models including the studs the FEM was the only solution for the plate, and the complication to the program and the amount of data concerning the full mobility matrix (see eq. (1), (2)) for each frequency step that were required for the Modal Approach reduced the benefits of faster calculation.

One interesting remark can be, that in principle the same partition was tested in both laboratories. Both the experimental results and the results of numerical simulations, however, differ substantially between the laboratories. It is yet another demonstration of the today already widely accepted fact, that at low frequencies in particular the rooms have significant influence on the results of sound reduction estimation (see [3] for the detailed discussion).

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