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PREDICTED AND MEASURED LOW FREQUENCY RESPONSE OF SMALL ROOMS

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1. INTRODUCTION

Increasingly, it is recognised that noise from adjoining dwellings occurs at low frequencies, below 100Hz[1,2]. The recommended method of measurement of sound insulation, BS 2750 /Part 4 (1980), is limited to frequencies above 100 Hz and therefore may lead to an underestimate of likely complaints. Despite an Annex F in ISO 140/ part 3 (1990) for sound insulation measurements at low frequencies, Roland[3] shows that a problem remains of poor reproducibility between laboratories for sound insulation measurements from 50Hz and no method of measurements below 100Hz has been agreed. Some recent investigations[4], employing computer simulation, show that the sound insulation of party walls at low frequencies is strongly dependant upon the modal characteristics of the sound fields of the separated rooms. The utilisation of numerical techniques becomes therefore a very useful method in order to identify the influential room and partition characteristics with respect to low frequency sound transmission.

The work reported in this paper was to validate the simulation as a prelude to an investigation of sound transmission between dwellings at low frequencies. A Finite Element Method (FEM) was selected to model a room and the predicted frequency response was then compared with the measured frequency response of a 1:4 scale model.

2. MODELLING

In order to model the airborne sound transmission from one room to another room, two models had to be defined: an acoustic model representing the sound field of the two rooms using an Acoustic Finite Element method (AFE)[5] and a structural model, representing the radiation of the party wall, using a Structural Finite Element method (SFE)[5]. Each model is subdivided into connected finite elements to model the pressure field or the displacement of the surface of the party wall. The number of elements is defined from the upper frequency of interest on the assumption that 6 elements are required to properly represent the pressure/displacement over the governing wavelength. An incorrect selection of the number of elements can result in numerical errors and consequently affects the accuracy of the simulation.

The time taken for the simulation (CPU time) can also be very long, depending upon the number of finite elements and on the processing power of the computer. In our case, the simulation run was longer than CPU time, since the network system was shared with other users. It was also foreseen that the CPU time for simulating the transmission, obtained by coupling the acoustic finite element (AFE) and the structural finite element (SFE), would be greater than the CPU time required for the room modes only. Consequently, an optimum between accuracy of simulation and required CPU power had to be determined.

The dimensions selected for the transmission room represented typical adjacent dwellings, 4m x 4m x 2.5 and 3.5 x 4 x 2.5. The size of the mesh depends upon the maximum frequency of the interest. In this study, the upper frequency was 200 Hz corresponding to a wavelength of 1.7m. Compared with the longest room dimension, 4m, 14 elements ideally are required to represent the pressure variation over

a wavelength. In the case of a three dimensional model, 14^3 elements are required to model the sound field of one room. Such a model is called a 14 mesh model.
It was found that three days of processing were required for the simulation of the room modes for 12 mesh and greater models and it was necessary to reject the 6 elements per wavelength condition. Seven models were constructed from 5 meshes to 11 meshes in order to select a mesh model for which the CPU time is acceptable, and for which data are processed within an acceptable error.
To evaluate the error due to the finite element discretisation, a ratio ε was calculated by comparing the theoretical eigenfrequencies[6] with the finite element eigenfrequencies processed for each mesh model.

$$\varepsilon = \frac{\text{theoretical eigenfrequency - numerical eigenfrequency}}{\text{theoretical eigenfrequency}} \times 100$$

Figure 1 shows the error from a 5 mesh to a 11 mesh model plotted against the number of modes, within a frequency range of 0 to 220Hz.
The room mode calculation for a 5 mesh model has a maximum error of 17%, which occurs in the upper part of the frequency range of interest, whereas the calculated modes for a 11 mesh model are within an error of 6%. The errors for the 10, 9 and 8 mesh models were also acceptable as they were below 10%.

3. MESH MODEL SELECTION

The intention in this study was to link the AFE model of the transmission room with the SFE model of the partition. In order to do so, both mesh sizes have to be the same[5]. By selecting the 10 AFE mesh model, the simulation ran within an error of 7%. When the panel was modelled with 10^3 elements, the simulation ran within an error of 8% except at 144Hz where the error was 10% The error was expressed with respect to calculated eigenfrequencies of a simply supported panel[6].
For validation, the simulation was compared with measurements of a 1/ 4 scale model, room with dimensions 1.2 x 1.2 x 0.6m. The maximum frequency range was [100- 800 Hz] corresponding to [25-200 Hz] full scale.
The enclosure mesh was designed using Patran3[7], then transferred into SYSNOISE 5.3 where the values of mass density and sound velocity were assigned to the sound field of the enclosure. 90 room modes, a number of eigenfrequencies above the frequency range of interest, were then processed assuming that the enclosure had 6 hard surfaces. The frequency response was obtained with a resolution of 1 Hz. A field point mesh was processed to produce the sound pressure level at position 1 (0.4, 0.5, 0.6m) and at position 2 (0.8, 0.8, 0.2m) to allow a comparison with sound pressure level measurements.

4. EXPERIMENT

An enclosure with the same dimensions as given above was made of 24mm thick blockwood, with one side of 10mm perspex. This physical scale model was in a small sound transmission suite where the background noise level was low and was placed on resilient foam to reduce vibration from the floor.
A loudspeaker radiated through a 10mm hole in one corner opposite to the perspex sheet. Two microphones were placed at the same positions as selected in the simulation.
MLSSA was used to obtain the spectrum of the sound level at two microphone positions from 100 Hz to 800 Hz with a resolution of 0.5Hz in order to record all dips and peaks in the response.

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5. VALIDATION

Figure 2 and 3 show the numerical and measured sound pressure levels at positions 1 and 2, from 100 Hz to 800 Hz. The trend and the characteristics of the curves for each position are the same. Due to two different chosen resolutions for prediction and measurements, sound level was calculated with a 12th octave band resolution in order to inspect the discrepancy between simulation and measurement for positions 1 and 2. Results are presented as a level difference (simulation-measurements) in Figure 4. The large differences are the results of often quite small shifts in observed with respect to expected resonant frequencies. In addition, the numerical model does not include surface or air absorption and therefore the simulation emphasises peaks and dips. Figures 5 and 6 show the level difference calculated with a 1/6 and 1/3 octave band resolution, respectively. In real measurements, the data are commonly presented in a 1/3 octave bands. The two figures, compared with Figure 4, show that the discrepancy between simulation and measurement decreases when calculated within a larger bandwidth. A peak at 141Hz (1st mode) is evident in all curves. The maximum discrepancy between simulation and measurements is 18dB when calculated within a 1/6 octave band and 15dB within a 1/3 octave band. The discrepancy is less for the microphone position 1 than for position 2. This can be explained by the fact that microphone position 1 was closer to the loudspeaker than microphone position 2, and was therefore less affected by the absorption of the room. The simulation overestimates the overall sound level by approximately 5-10dB. This could be for two reasons. The first is that the sound power of the point source may have been incorrectly assigned. Secondly, as stated earlier, damping was not included in the FEM model, as it is difficult to quantify at low frequencies.

6. CONCLUDING REMARKS

Although the size of the element is less than a 1/6 of the wavelength, the comparison between simulation and measurements for one room is promising and validates the choice of the mesh number to model the rooms and party wall. Three sources of error were identified: incorrect assignment of sound power to the source, non inclusion of absorption in the FEM model and shifts in the predicted resonance frequencies with respect to measured, which produce larger discrepancies than is indicated by visual inspection of the frequency response curves. However, in the prediction and measurements of the sound level difference between two rooms, yet to be reported[8], the effect of the first two sources of error tends to cancel.

7. REFERENCES

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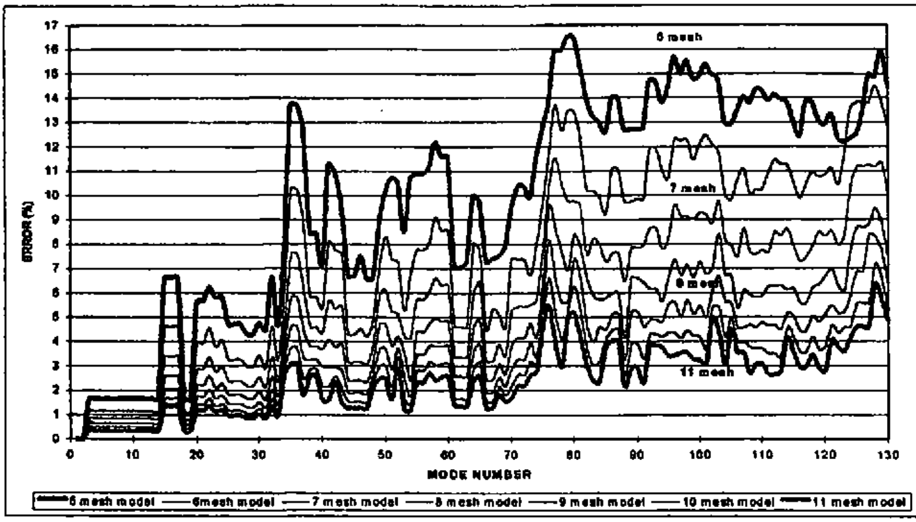

Figure 1. Error versus mode number

Figure.2: Sound level at position 1

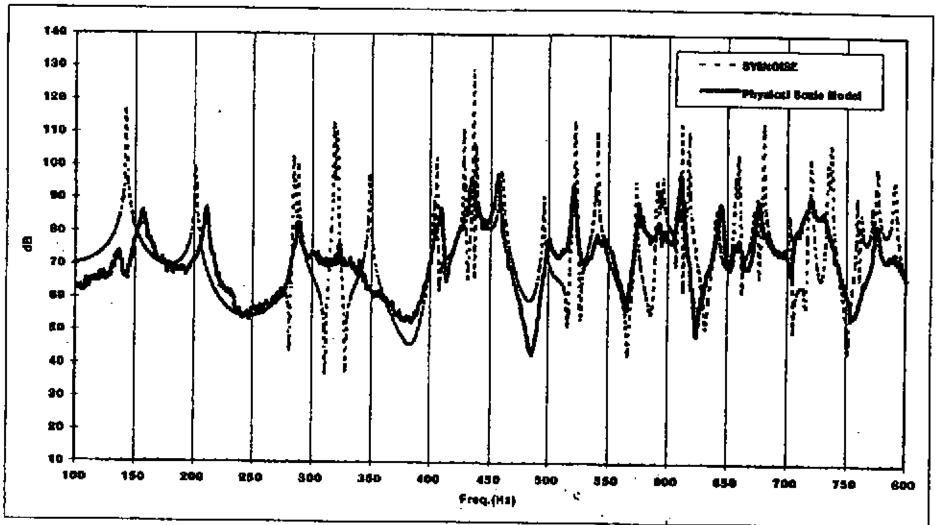

Figure 3: Sound level at position 2

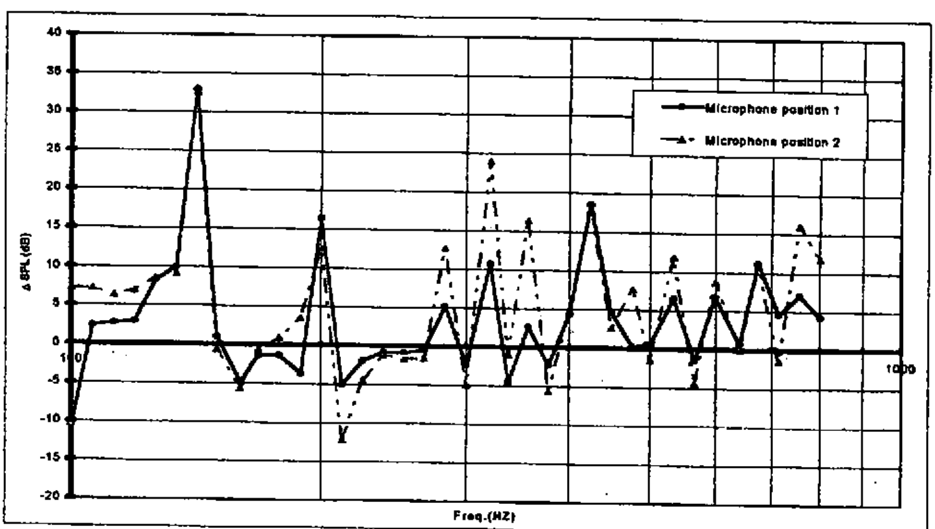

Figure 4: Simulation compared with measurements in a 12[th] octave band.

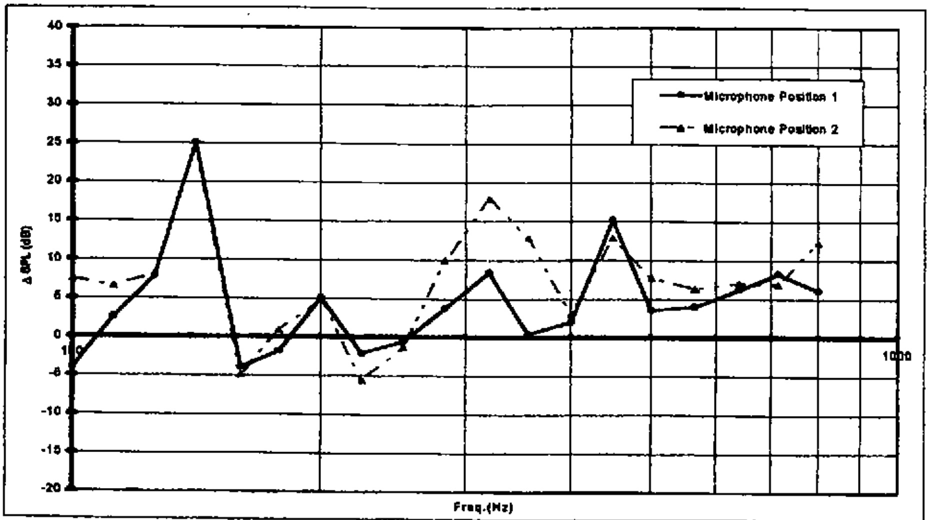

Figure 5: Simulation compared with measurements in a 6^{th} octave band.

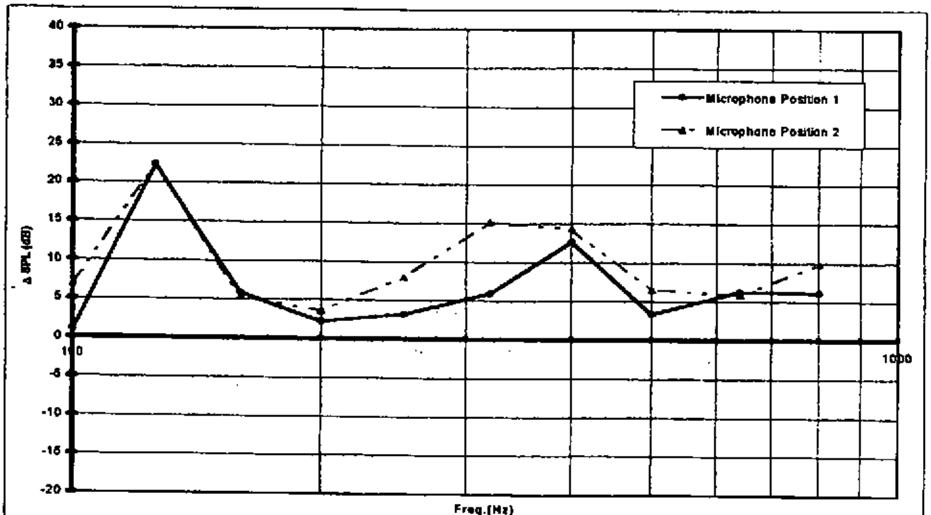

Figure 6: Simulation compared with measurements in a 3^{rd} octave band.