LOW FREQUENCY AIRBORNE SOUND TRANSMISSION IN BUILDINGS

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ABSTRACT

The aims of this paper are to provide further background description and present and implement a modal approach, the latter assisting in providing improved understanding of the noise transmission phenomenon in buildings. The derivation and numerical examples presented in this paper show how transmission efficiency is affected by room geometry. The transmission of sound between similar or dissimilar rooms, e.g. for the latter consider rooms attached to corridors, can equally be predicted using the Modal approach. The narrowband results of analyses were converted to one-third octave band spectra, to make comparisons with other data possible. Finally, a general discussion, based on the findings of the results obtained, is presented with some observations concerning potential improvements that can be considered.

1. INTRODUCTION

A detailed review of interior sound field problems was presented by Dowell [1], and others [2-3], who investigated acoustic-structural coupled systems. Recently Osipov [4] evaluated some numerical examples in order to verify the influence of the dimensions of rooms and partitions on sound transmission. The noise reduction of the system due to resonant and nonresonant coupling involving modal behaviour and spatial fluid-structural coupling is predicted. The model can include (i) different room sizes and absorption, including narrow corridors coupled to rectangular spaces; (ii)small partitions - a partition that does not cover the whole of the common wall and (iii)Non-diffuse excitation in the source room.

2. THEORETICAL MODEL FOR THE COUPLED SYSTEM

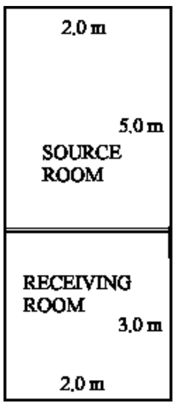

Figure 1a: Model 1: Two rooms (height=2.0m) separated by a common wall (2m x 2m).

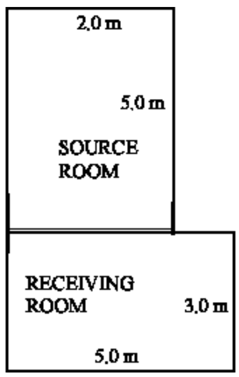

Figure 1b: Model 2: Two rooms (height = 2.0 m) separated by a common wall (2m x 2m)

Room-panel-room systems (Figures 2.1 and 2.2) are analysed by modal models based on a set of integro-differential equations for the interaction between a flexible plate and enclosed fluids [1]. The structural motion has been expressed as a summation of the response in the *in vacuo* modes öp driven by the fluid loading. The acoustic-field is approximated by the summation of the uncoupled acoustic modes ψn of the fluid volume enclosed by rigid walls. The acoustic modes were excited by a unit harmonic volume velocity placed inside the source room. The system response is obtained by solving the following coupled equations [7]

$$-\omega^2\Phi_{n_1} + j\omega\beta_{n_1}\Phi_{n_1} + \omega_{n_1}^2\Phi_{n_1} = \left(\frac{c_o^2 S}{\Lambda_{n_1}}\right)\sum_{p=1}^{P}\left(j\omega w_p C_{n_1 p}\right) - \left(\frac{c_o^2}{\Lambda_{n_1}}\right)Q_{n_1} \qquad \text{Eq. 1a}$$

$$-\omega^2 w_p + j\omega\beta_p w_p + \omega_p^2 w_p = -\left(\frac{\rho_o S}{\Lambda_p}\right)\sum_{n_1=1}^{N_1}\left(j\omega\Phi_{n_1} C_{n_1 p}\right) + \left(\frac{\rho_o S}{\Lambda_p}\right)\sum_{n_2=1}^{N_2}\left(j\omega\Phi_{n_2} C_{n_2 p}\right) \qquad \text{Eq. 1b}$$

$$-\omega^2\Phi_{n_2} + j\omega\beta_{n_2}\Phi_{n_2} + \omega_{n_2}^2\Phi_{n_2} = -\left(\frac{c_o^2 S}{\Lambda_{n_2}}\right)\sum_{p=1}^{P}\left(j\omega w_p C_{n_2 p}\right) \qquad \text{Eq. 1.c}$$

where the indices *n1*, *n2*, and *p* refer to source room, receiver room and panel mode numbers respectively and β is the generalized modal damping coefficient introduced for the acoustic volumes and structural modes. Φ and w represent generalized modal velocity potential and normal surface displacement of the partition respectively (see [2] for the complete definition of the other symbols). *w* The spatial structural-acoustic coupling coefficient *Cnp* is defined by

$$\frac{1}{S}\int_S \psi_n \phi_p dS \qquad \text{Eq. 2}$$

where simply-supported edges are assumed for the partition and *S* is its area.

3. RESULTS

The models adopted comprised three subsystems: a source room, a common wall and a receiving room. In 'model 1' (Figure 1a) both rooms have the same width and height whereas in 'model 2' (Figure 1b) the receiving room is wider than the source room. The source room was defined as an acoustic volume excited by a broadband acoustic point source placed at a corner position. Although the source position does not alter the spatial coupling coefficients between structural and acoustic modes, it has significant influence on exciting the source room modes. Thus, with the source located at one of the source room corners, all modes within a specific frequency range were excited. The results obtained from the numerical examples provide information about the sensitivity of the modal model to parameters, such as the spatial distribution of pressure and particle velocity in the acoustic volumes. Finally, some results for the modal model are compared to those obtained by different formulations.

The system properties are described as follows. For a partition made of plasterboard material, a value of υ = 0.24 and E = 2.12x109 N/m^2 were assumed for the Poisson's ratio and Young's modulus respectively. Also a density value of ρs = 806 kg/m^3 and a thickness of 0.01 m [21] were assumed for the material. On the other hand, for a partition made of steel, a value of υ = 0.24 and E = 210x10^9 N/m^2 were assumed for the Poisson's ratio and Young's modulus respectively. Also a density value of ρs = 7850 kg/m^3 and a thickness of 0.01 m [21] were assumed.

The assumption of only pure bending waves propagating in the panel remains valid as the panel thickness is much smaller than the wavelength at the highest frequency considered herein. When varying the other parameters, the receiving and source room surfaces were considered as being covered by a soft material with a constant modal frequency-average absorption coefficient. The loss factor for the rooms was chosen as a constant value η = 0.01 over the whole frequency range. The corresponding T_{60} that results using this constant loss factor is plotted in Figure 2.4.

In ref. [5] typical values for the absorption properties of a room are presented. If one used these absorption values the corresponding loss factor values η would vary from 0.001 to about 0.1 for some commonly used materials in buildings. An important approximation to note here is that the mode functions used have been chosen as the mode shapes of a volume bounded by rigid walls and that absorption has been introduced via a modal description, rather that involving a complex wall impedance in the model. The latter is much more complex and is unnecessary in the present case of rooms with low absorption; both models would produce similar results. Moreover, the acoustic source strength applied to the source room was a volume velocity equal to $3x10^{-5}$ m^3/s. The source was placed at the corner of the room for all of the simulations presented. The Noise Reduction (NR) parameters obtained from the modal and classical approaches [2, 6] were compared graphically as a function of frequency. It was verified in ref. [6] that Leppington's prediction approaches the values obtained from the infinite plate theory when the non-resonant transmission is modelled.

Figures 2 and 3 show the mean square sound pressure and particle velocity distribution (in the *x* direction normal to the panel) with respect to the horizontal plane y = 1m at 120 Hz. Figure 2a and 3a show the surface plot for the pressure and particle velocity respectively. It is observed that there is pressure discontinuity at the interface coordinate $x = 0$ (where there is a flexible partition in the whole common wall) as expected. On the other hand, the particle velocity just goes to zero at the interface. The results are also not constant across the crosssection or symmetric, due to the source location being positioned in one corner of the room (- 5,0,0) and the frequency being above the first acoustic mode with a half-wavelength across the section (85 Hz). Figures 2b and 3b show the corresponding contour levels.

Figures 4 and 5 show the NR values for models 1 and 2 respectively. Partitions with mass per unit area equal to 8.1 kg/m^2 and 78.5 kg/m^2 were considered. Critical frequencies equal to 3815 Hz and 1196 Hz were obtained for the light and heavyweight partitions respectively. In Figure 4-a and 4-b it is seen that at very low frequencies (below 100 Hz), differences of up to about 20 dB occurred between the modal model and the diffuse incidence Mass Law. In this situation, the dimensions of the subsystems were small in comparison with the wavelength of the sound. Thus, for this condition the motion of the medium in the system is analogous to that of a mechanical system having lumped mechanical elements of mass, stiffness and damping. The lowest NR values shown in Figures 4-a and 4-b are in the one third octave frequency bands with centre frequencies at 8 Hz and 12.5 Hz respectively. For the lightweight partition, this value approximately corresponds to the coupled frequency 9.02 Hz. For the heavyweight partition (Figure 4-b), the lowest NR value corresponds to the coupled frequency equal to 12.53 Hz. It is seen that this frequency is the coupled version of the fundamental natural frequency of the heavyweight partition, which is equal to 12.08 Hz. It is well known that if a coupled system is excited acoustically and the acoustic volume responds predominantly as though the structure were infinitely rigid, this system is said to be weakly coupled. Therefore, the results confirmed the theory that 'weak coupling' effects occur in models with heavyweight partitions.

Moreover, at very low frequency the flexible partition behaves as a rigid-body and the resulting stiffness element is expressed by the acoustic bulk stiffness of the enclosed fluid in the room. The acoustic bulk stiffness is given by $k_A = \rho_o c_o S_A^2 / V_A$; where S_A is the room transverse area (height x width) and V_A is the volume of the acoustic room. In this case, the coupled frequency can be estimated by considering a one-degree-of-freedom mass-spring system. This simplified model consists of a structural mass connected to two 'springs' corresponding to both acoustic rooms. The natural frequency of free vibration of this simplified model was estimated and is approximately 15.5 Hz. For the 1/3 octave bands with centre frequencies above 100 Hz, the NR values shown in Figure 4-a tend to those obtained via Leppington's prediction. In other words, the trend of the curve for the modal model approximates the established values, which consider the resonant and non-resonant contributions at higher frequencies. This is justified by the fact that the 'Schroeder frequencies' [5] were approximately 298 Hz and 383 Hz for the source and receiving rooms respectively. Nevertheless, for the heavyweight partition Figure 4-b shows that the NR values are closer to the diffuse field Mass Law at higher frequencies.

Figure 5 shows the NR values obtained for model 2 (Figure 1-b). In Figure 5-a, the variation of the modal model from the Mass Law and Leppington's formulation at low frequencies is less pronounced than that for model 1 shown in Figure 4-a. This is due to the fact that model 1 (Figure 4) has exactly coincident resonance frequencies for the two rooms e.g. at 34, 68 and 85 Hz. In addition, there is geometric matching of modal distribution over the common partition. On the other hand, the NR results presented in Figure 5 show the effects of mismatch of modal properties of rooms having dissimilar geometrical characteristics. As the frequency increases, the results tend to the values calculated via the Leppington's approach. By comparing Figures 4 and 5, it is also evident that at higher frequencies the effect of room shape on NR is not so significant. For instance, in the frequency band with centre frequency at 400 Hz, a difference of less than 2 dB is found between models 1 and 2.

Finally, it is seen that in both configurations (Figures 4 and 5) the values obtained via Leppington's formulation approximated to those using the field incidence Mass Law when the frequency increases. These results may be explained by the fact that the resonant contribution, which is taken into account in Leppington's formulae, was not significant within the frequency range considered, where the forced non-resonant vibration contribution is the dominant factor. Furthermore, for the heaviest partition the diffuse field Mass Law is about 3-6 dB lower than Leppington's or the field incidence Mass Law values at frequencies greater than 100 Hz.

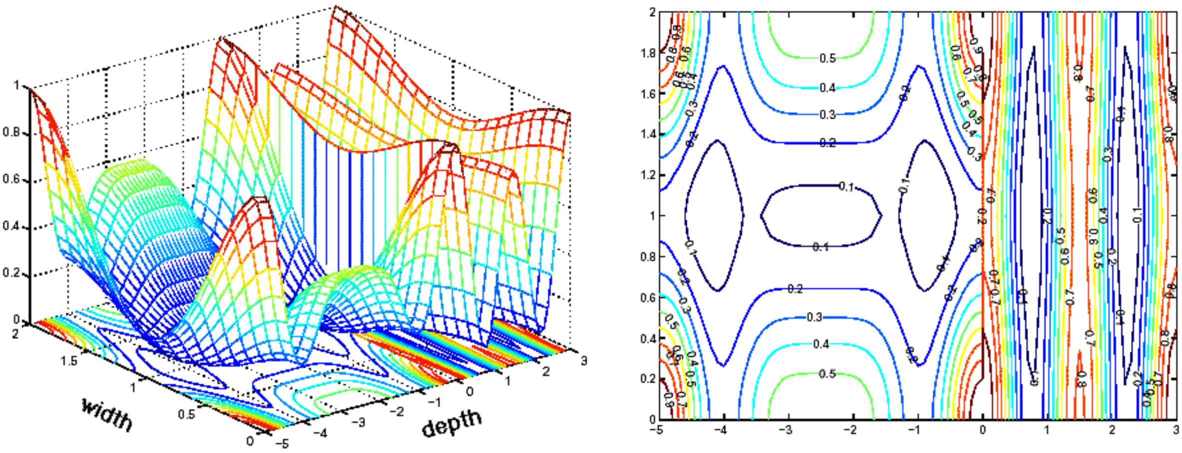

Figure 2: Normalized mean square pressure distribution (model 1) with respect to the horizontal plane y = 1 m at 120 Hz. The partition dimensions and mass per unit area are 2m x2m and 8.1 kg/m^2 respectively. a) surface plot; b) Contour levels in (Pa^2/Pa^2);

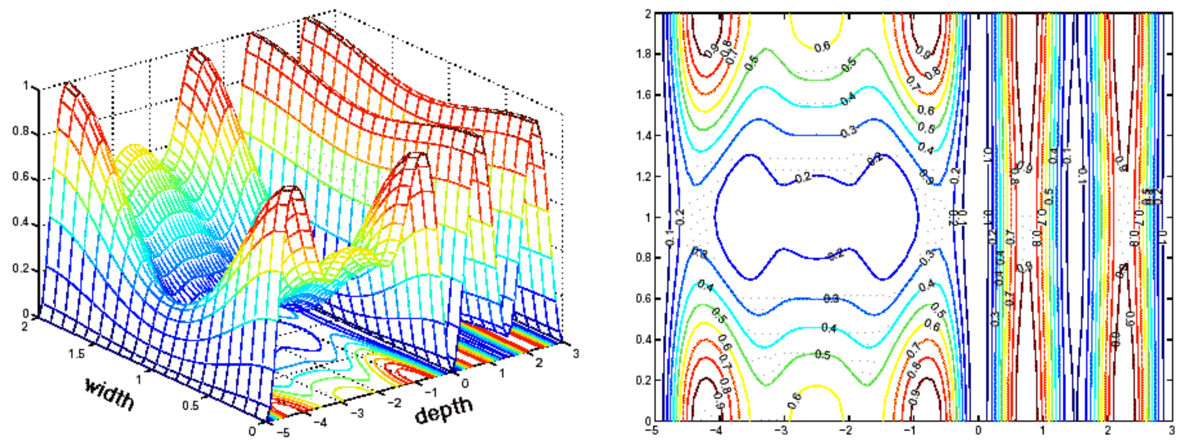

Figure 3: Normalized mean square particle velocity distribution (model 1) in the x-direction with respect to the horizontal plane y = 1 m at 120 Hz. The nominal partition dimensions and mass per unit area are 2m x 2m and 8.1 kg/m^2 respectively. a) surface plot; b) Contour levels in (m/s)^2/(m/s)^2;

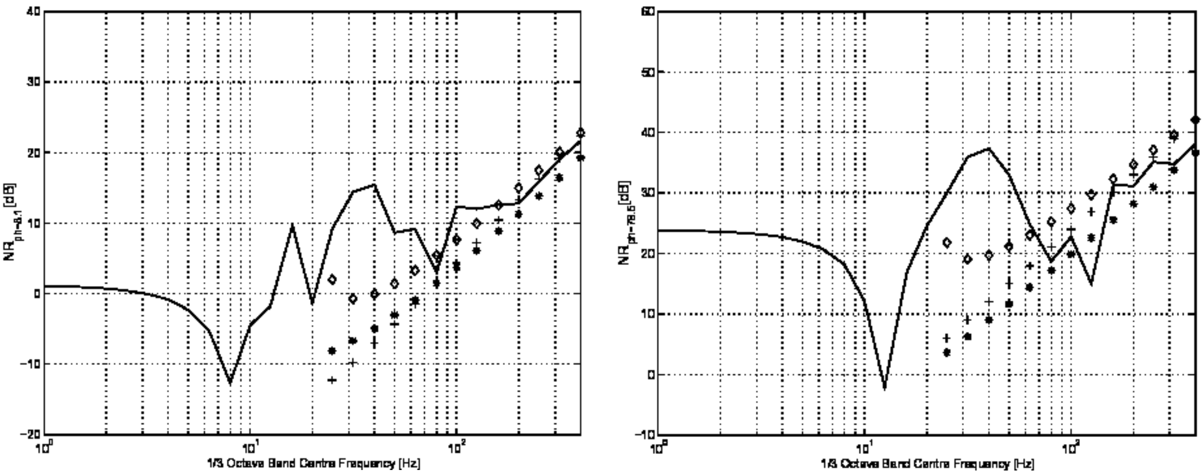

Figure 4: Comparison of the Noise Reduction (NR) levels between the modal model 1 (see Figure 2a) and the classical methods. a) ρh = 8.1 kg/m^2; b) ρh = 78.1 kg/m^2; _____ Modal model; * Diffuse incidence Mass Law; +++ Field incidence Mass Law; ◊◊◊◊◊ Leppington's prediction.**

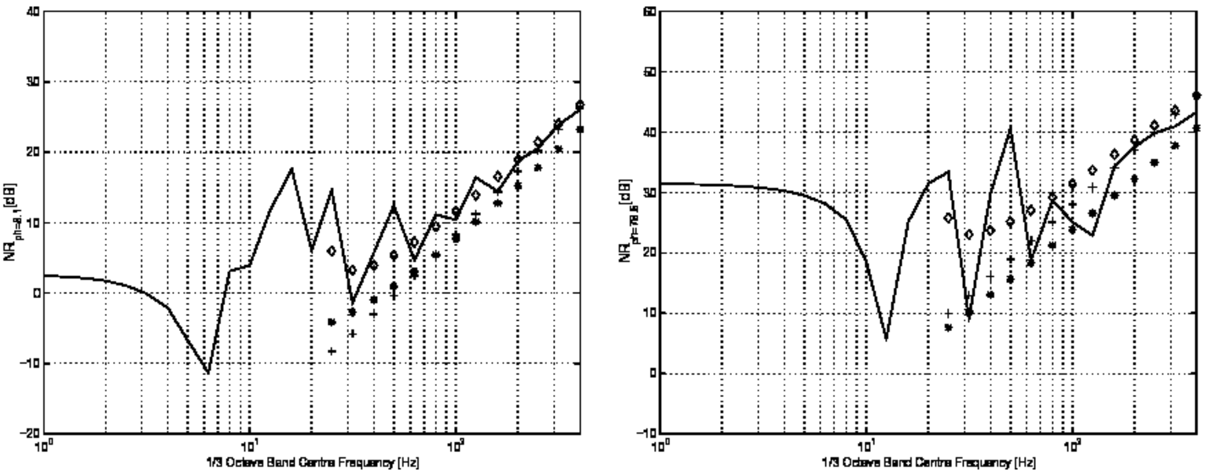

Figure 5: Comparison of the Noise Reduction (NR) levels between the modal model 2 (see Figure 2-b) and the classical methods. a) ρh = 8.1 kg/m^2; b) ρh = 78.1 kg/m^2; _____ Modal model; * Diffuse incidence Mass Law; +++ Field incidence Mass Law; ... Leppington's prediction.**

4. CONCLUSIONS

A comparison between numerical modal analysis and theoretical predictions has been performed. A maximum frequency of 450 Hz was used for the frequency response of the systems to a volume velocity point excitation in the source room. Above this frequency limit the computational storage requirements for variables as well as the operational running time on a personal computer became extremely problematic. The effect of being selective in eliminating some modal contributions has not been reported here. This is because the results are highly sensitive to the non-resonant modes in the frequency range considered. For instance, the non-resonant mass modes of the partition significantly contributed to the energy transmission between rooms. This is evident from the results, which approximates to those for the Mass Law as frequency increases. Although there were many 'weak coupling coefficients', their summation was significant to the total coupling. In ref. [7], it is show that the contribution of certain modes to the fluid-structure interaction depends on the degree of spatial coupling between the modes at the common interface. Hence, all possible natural frequencies and their respective modes were included in this analysis. The results may also help in the understanding of the model, with the subsystems considered directly related to physical elements such as rooms and flexible partitions. They can also provide an initial discussion for the investigation of a SEA model, which can be useful for practical building acoustics.

Although this problem (the coupling between the panel and the acoustic fields) has been solved in previous work by several authors, the results obtained herein can also be used for guidance in real cases of architectural acoustic design. All the parameters, which affected the modal composition of the sound field in the subsystems, were fundamental in the determination of the Sound Reduction Index. The results may also be used to interpret measurements made in-situ at low frequencies, e.g. where the classical definition of SRI in ISO140 for diffuse sound fields may not be appropriate or reliable. Although the assumption of uncoupled 'rigid-walled' acoustic modes for the rooms [1, 2] has been assumed for many years, the actual boundary condition, which is due to the velocity of the partition, cannot be replicated. The convergence problem may be rather sensitive at low frequencies and may require a significant summation of modes to provide accurate velocity and pressure predictions at the panel location. This is necessary for accurate predictions of the acoustic intensity and hence Sound Reduction Index.

Existing methodologies, i.e. the Mass Law and Leppington's formulation, similarly have difficulty at low frequencies. For instance, the assumption of diffuse field, etc., is no longer valid at very low frequencies, as few acoustic modes exist in the volumes. However, it has been shown that the SRI values obtained using the Modal model converged reasonably well to Leppington's prediction as the frequency increases. If one is interested in the Noise Reduction and hence requires spatially averaged acoustic pressures, then the methodology of using the modal method with 'rigid-walls' is acceptable and provides good results. This statement can be confirmed by the fact that the results obtained converged to the established and accepted analytical models as frequency increases.

REFERENCES

Dowell, E. H., Gorman, G. F., III and Smith, D. A., "Acoustoelasticity: General Theory, Acoustic Natural Modes and Forced response to Sinusoidal Excitation, Including Comparisons with Experiment", Journal of Sound and Vibration, 1977, vol. 52(4),pp.519-542

Fahy, F. J., "Sound and Structural Vibration", Academic Press, U.K., 1985.

Leppington, F.G., Heron, K. H., Broadbent, F. R. S., & Mead, S. M., "Resonant and Non-Resonant Acoustic Properties of elastic Panels. II. The Transmission Problem", Proc. Roy. Soc. London, 1987, A 412, pp. 309- 337

Osipov, A., Mess, P. and Vermeir, G.; "Low-Frequency Airborne Sound Transmission Through Single Partitions in Buildings"; Applied Acoustics, 1997, vol. 52, no. 3-4, pp. 273-288.

D.A. Bies and C. H. Hansen; *Engineering Noise Control – Theory and practice.* E &FN SPON – Second Edition, 1996.

F. G. Leppington, K. H. Heron, F. R. S. Broadbent and S. M. Mead, "Resonant and Non-Resonant Acoustic Properties of elastic Panels. II. The transmission Problem", *Proceedings of the Royal Society of London*, 1987, **A 412,** pp. 309-337.

M. D. C. Magalhaes and N. S. Ferguson; "Low frequency airborne sound transmission"; *Proceedings of Inter Noise – The Hague* 4pp.