AIRBORNE SOUND TRANSMISSION ACROSS SOLID HOMOGENEOUS PLATES: A REVIEW OF THEORY AND PRACTICE FOR BUILDING ACOUSTICS APPLICATIONS

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1 INTRODUCTION

Predicting airborne sound transmission in buildings almost always requires consideration of the flanking paths. However, the most basic requirement by users of prediction models is a good estimate of the direct path for airborne sound transmission between two rooms separated by a solid homogeneous plate. The existence of a number of different theories and formulae to calculate resonant and non-resonant transmission can give the impression that this most basic of situations is not straightforward to model. In practice the choice of formulae is relatively straightforward if the required degree of accuracy is established beforehand. However, there are certain aspects that are difficult to predict accurately; the most notable of these is the dip in the vicinity of the critical frequency.

This paper reviews the application of various theories to plates that are used to form building structures. The focus is on qualitative descriptions rather than the underlying formulae and derivations; these can be found in a recent book on sound insulation by Hopkins[1].

2 SOUND TRANSMISSION MODEL

Calculation of the sound reduction index for a plate that separates two rooms can be carried out using well-established theory for diffuse sound and vibration fields without recourse to consideration of modal energy flow. However, the latter approach can be undertaken within the framework of Statistical Energy Analysis (SEA) and has certain advantages. The SEA framework simplifies consideration of the two sound transmission mechanisms; non-resonant and resonant transmission. The importance of this becomes apparent when starting to predict the combination of direct and flanking transmission. Flanking transmission between connected plates only concerns the resonant vibration; hence it is useful to have a framework that can isolate the different mechanisms. In addition, the prediction of direct and flanking sound transmission in buildings tends to be carried out nowadays using either first-order SEA path analysis (as used in EN 12354-1[2]), or with a full SEA model.

The SEA model is shown in Figure 1 and corresponds to the situation representing an idealized transmission suite. A sound power input $W_{\mathrm{in}(1)}$ is applied to the source room, and power W_{ij} is transmitted between subsystems i and j via the non-resonant path 1→3, and the resonant path 1→2→3. The plate subsystem only supports bending modes of vibration.

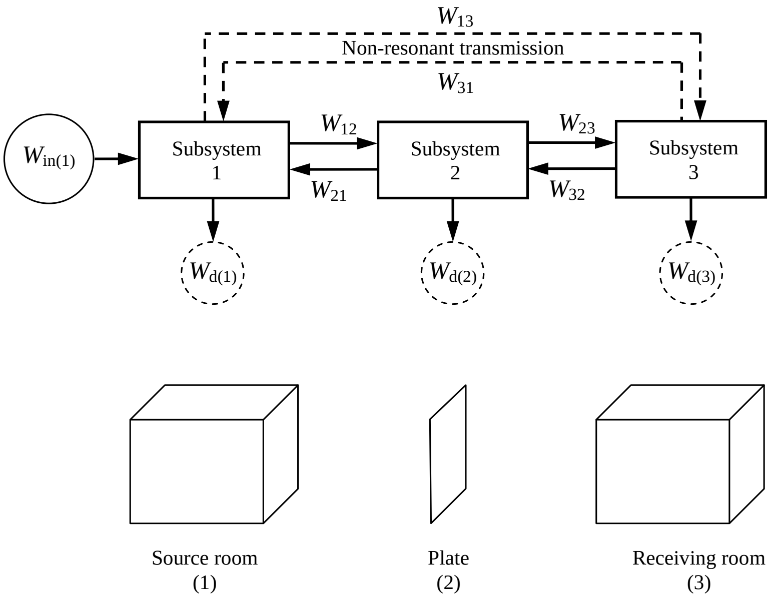

Figure 1 SEA model for airborne sound transmission between two rooms across a plate

3 NON-RESONANT TRANSMISSION

Non-resonant transmission can be quantified for plates of finite and infinite extent. It is sometimes referred to as mass law transmission, or forced transmission.

For finite plates, non-resonant transmission describes transmission due to bending modes that have their resonance frequencies outside the frequency band of interest (Sewell[3]). Individual modes with resonance frequencies below the critical frequency will have a higher radiation efficiency at frequencies above their resonance frequency, than actually at their resonance frequency. So when the frequency band of interest is below the critical frequency, radiation from modes with resonance frequencies within the band can be lower than radiation from modes with resonance frequencies outside the band that have been excited ‘off-resonance’. For the former modes, the modal response is under damping-control; for the latter modes, it is predominantly under mass-control. Non-resonant transmission can therefore be considered as being unaffected by the plate damping. In contrast to infinite plate theory, finite plate theory considers the plate to have both mass and stiffness. Whilst the mass per unit area is still important in quantifying the non-resonant transmission, the role of bending modes means that it also depends on the plate dimensions and the critical frequency. For a finite plate, the non-resonant transmission coefficient below the critical frequency can be calculated according to Leppington[4].

For infinite plates, the theory is based upon a plate that acts as a limp mass. A minor advantage in using the approach for infinite plates compared to finite plates is that the equation is slightly more compact. Infinite plate theory requires integration of the component of the incident plane wave intensity that is normal to the surface. For a plate exposed to a diffuse field the angles of incidence considered in this integration are assumed to lie between 0 and 90°. However, diffuse fields are an ideal that are not realized in typical rooms over the entire building acoustics frequency range. There are also situations where the plate is positioned within a niche such that it is partly shielded from angles of incidence near 90°. To account for these issues, an empirical adjustment is used where the angle 90° in the integration is replaced by 75°, 78°, or 80°; although 78° is most common. This range of angles is referred to as field incidence. Its name unfortunately suggests that this assumption is always valid for field measurements; which it is not. The assumption simply gives fortuitous agreement with measurements for particular plate sizes with particular critical frequencies, usually with particular mounting conditions in a niche and usually where non-resonant transmission dominates in the low and mid-frequency ranges and the sound field in the rooms is far from being diffuse. Whilst the infinite plate formulae for diffuse and field incidence are very useful for quick calculations and illustrative purposes, they do not describe all the features of non-resonant transmission that relate to finite size plates.

Unlike finite plate theory, non-resonant transmission across infinite plates is defined above the critical frequency. This implies that if the plate damping is sufficiently high such that resonant transmission is negligible at frequencies well-above the critical frequency, then non-resonant transmission will dominate at these high frequencies. It is unusual to find homogeneous plates in buildings that are this highly damped in the building acoustics frequency range, and using infinite plate theory for non-resonant transmission above the critical frequency is not usually appropriate.

Taking 6 mm glass as an example, the non-resonant sound reduction index for finite plates can be compared with infinite plates in Figure 2. For a finite glass plate, a reasonable size to consider is 1.5 x 1.25 m, but to illustrate the effect of plate size, rather unrealistic dimensions of 4 x 2.5 m are also shown. For the finite plates, the smaller plate has higher values than the larger plate, and both finite plates have significantly higher values than an infinite plate assuming diffuse incidence. As the frequency approaches the critical frequency, the non-resonant sound reduction index for the finite plates starts to decrease. Above the critical frequency, the non-resonant transmission coefficient is undefined for the finite plates.

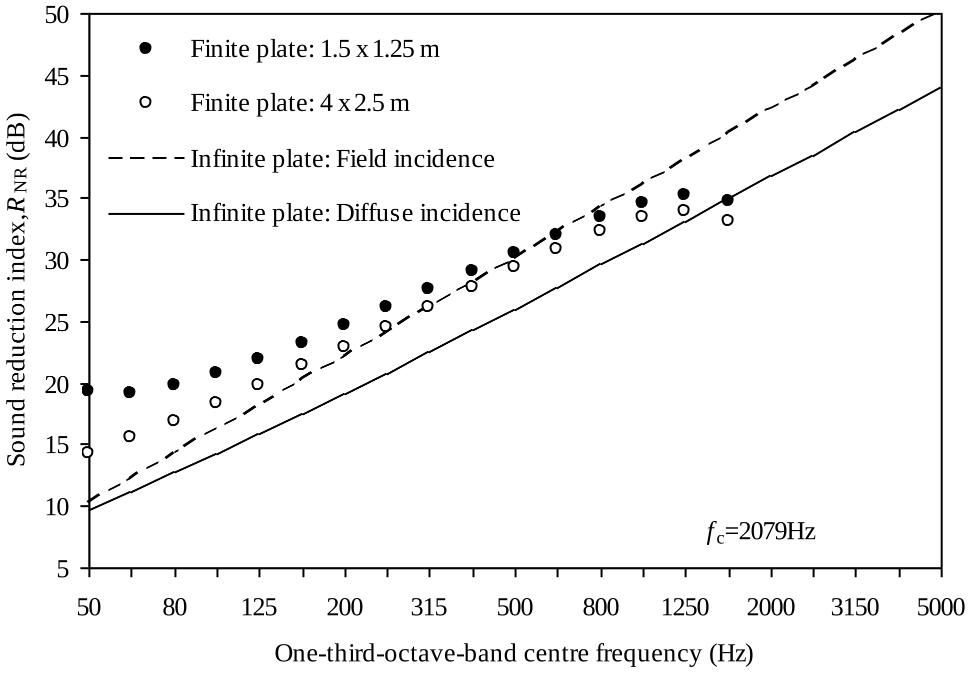

Figure 2 Non-resonant transmission across finite and infinite plates of 6 mm glass (ρ_s=15 kg/m^2, c_L=5200 m/s).

4 RESONANT TRANSMISSION

Resonant transmission occurs between the plate and source room, and the plate and receiving room. It describes the coupling between modal energy stored in these connected subsystems.

4.1 Critical frequency dip

Standard thickness sheets of plasterboard, wood and metal tend to have a pronounced dip at the critical frequency. However, in the vicinity of the critical frequency it is notoriously difficult to accurately predict the radiation efficiency, even when the statistical mode count and modal overlap factor are high. This can partly be overcome by calculating lower and upper limits for the radiation efficiency (Method No.2 in Hopkins[1]). The upper limit corresponds to the standard equation for radiation efficiency. The lower limit is calculated by setting the radiation efficiency to unity in and above the lowest frequency band where the calculated value exceeds unity. Measured data usually lie within the shaded area between the two limits. An example is shown for 12.5 mm plasterboard in Figure 3 where the critical frequency is 3483 Hz.

For walls and floors made from bricks, masonry or concrete the dip near the critical frequency is often difficult to discern and is rarely as pronounced as with standard thicknesses of plasterboard, wood and metal. By using only the lower limit for the radiation efficiency described above (see Method No.3 in Hopkins[1]) it is possible to get reasonable estimates of the sound reduction index in the vicinity of the critical frequency as shown by the example in Figure 4. Note that when the plate mode count is less than three in frequency bands below the critical frequency, a more accurate but time consuming approach is to calculate the radiation efficiency using regression analysis from the radiation efficiency of individual modes (Method No.4 in Hopkins[1]).

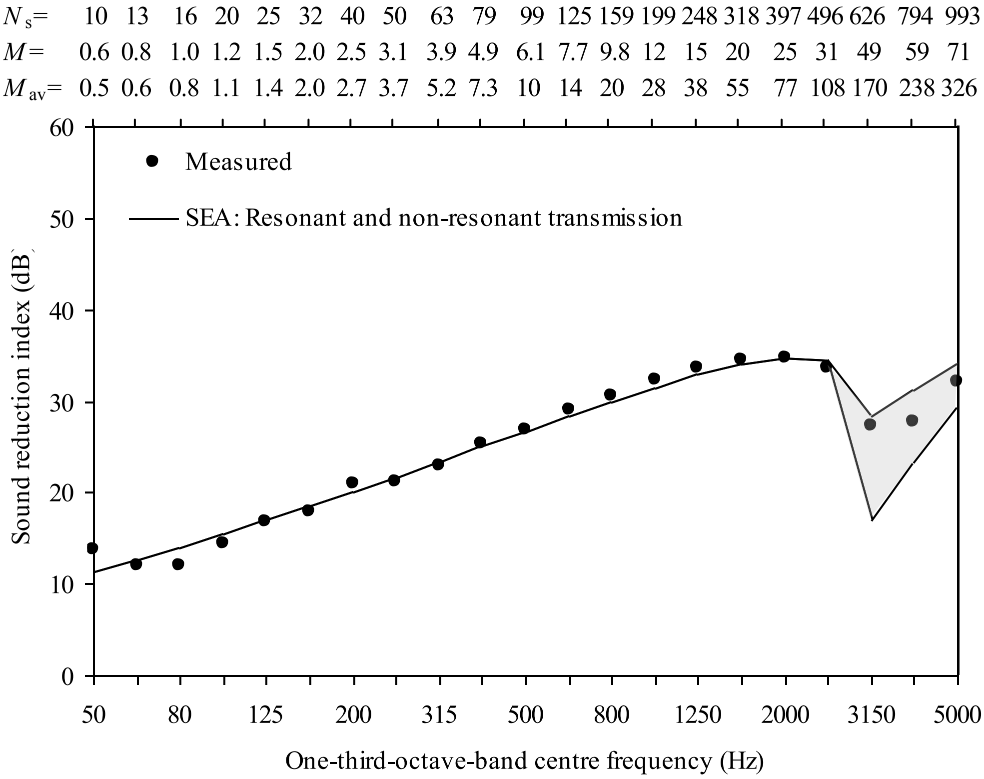

Figure 3 Measured and predicted airborne sound insulation of 12.5 mm plasterboard. Upper *x*-axis labels show the predicted statistical mode count (N_s) and modal overlap factor (M) for the plate, and the geometric mean of the modal overlap factors for the plate and room (M_{av}).

Plate properties: L_x=3.53 m, L_y=2.63 m, h=0.0125 m, ρ_s=10.8 kg/m^2, c_L=1490 m/s, ν=0.3, η_{int}=0.0141.

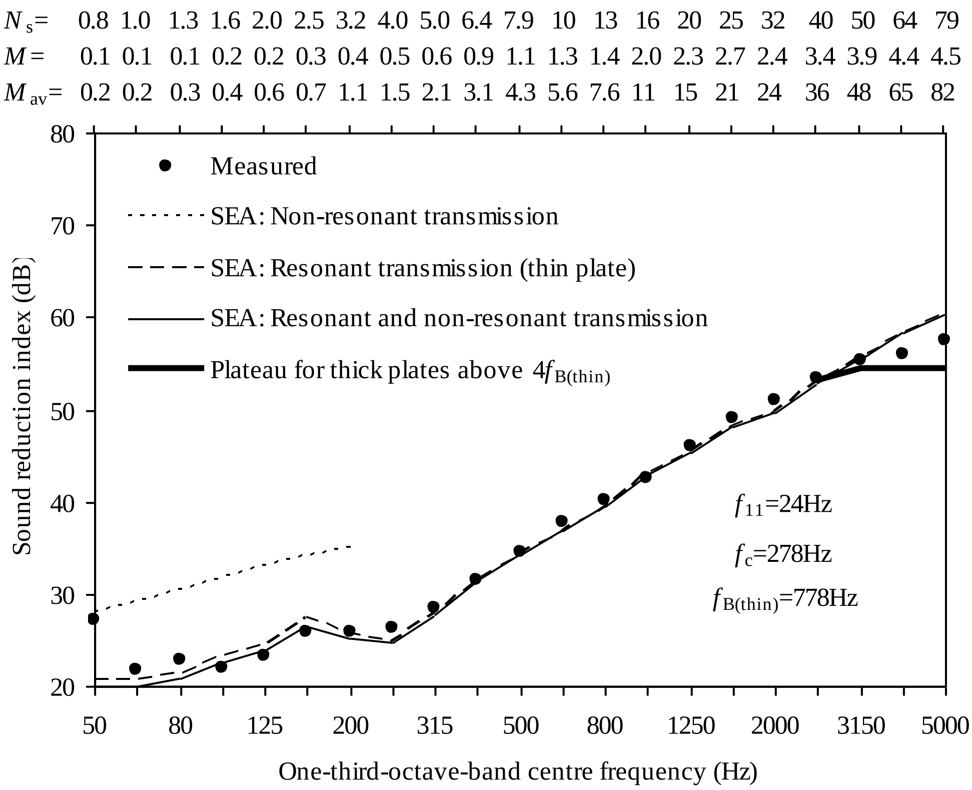

Figure 4 Measured and predicted airborne sound insulation of a 115 mm masonry wall (solid aircrete blocks) with a 13 mm lightweight plaster finish (one side). Upper *x*-axis labels show the predicted statistical mode count (N_s) and modal overlap factor (M) for the plate, and the geometric mean of the modal overlap factors for the plate and room (M_{av}).

Plate properties: L_x=3.53 m, L_y =2.63 m, h=0.128 m, ρ_s=71 kg/m^2, c_L=1820 m/s, ν=0.2, measured total loss factor. The plateau is calculated using material properties corresponding to the plate thickness: c_L=1920 m/s and an internal loss factor η_{int}=0.0125.

4.2 Low mode counts and low modal overlap

Over the building acoustics frequency range we find that walls, floors, and glazing can support vastly different numbers of bending modes in third-octave frequency bands. The absence of diffuse vibration fields on plates in buildings is sometimes quoted, albeit incorrectly, as the only reason why it is difficult to gain accurate predictions of resonant transmission for airborne sound. It is also useful to consider the degree of overlap in the modal response that is described by the modal overlap factor. Referring back to Figure 3, the upper *x* axis labels show that even when there is an abundance of modes and there are high modal overlap factors it can be difficult to accurately predict the sound insulation in the vicinity of the critical frequency. This difficulty in predicting sound transmission near the critical frequency is partly because the theory for radiation efficiencies does not adequately describe the phenomenon (particularly with wide frequency bands), and partly because of niche effects.

The example in Figure 5 shows that for a thick masonry wall with fractional mode counts and modal overlap factors less than unity it is still possible to get reasonable estimates for the sound reduction index over the building acoustics frequency range. The modal overlap factor for the plate is very low compared to each room to which it is coupled. Hence conditions relating to the adequacy of the theory can be linked to the geometric average of the modal overlap factors for the room (source or receiving) and the plate, M_{av}, These values are greater than unity over a larger part of the frequency range. Reasonable estimates for the sound insulation can usually be found when $N_s \geq 1$ for the plate system, and $M_{av} \geq 1$.

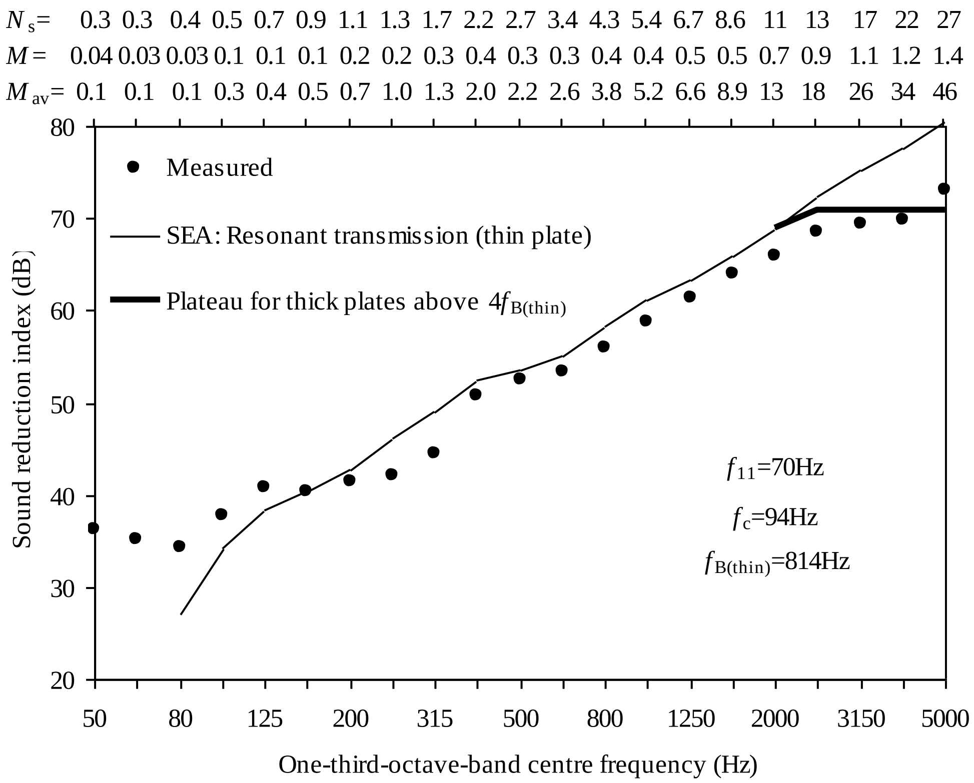

Figure 5 Measured and predicted airborne sound insulation of a 215 mm masonry wall (solid dense aggregate blocks) with a 13 mm lightweight plaster finish (each side). Upper *x*-axis labels show the predicted statistical mode count (N_s) and modal overlap factor (M) for the plate, and the geometric mean of the modal overlap factors for the plate and room (M_{av}).

Plate properties: L_x=3.53 m, L_y =2.63 m, h=0.215 m, ρ_s=430 kg/m^2, c_L=3200 m/s, ν=0.2, measured total loss factor. The plateau is calculated using material properties corresponding to the plate thickness: c_L=4000 m/s and an internal loss factor η_{int}=0.01.

4.3 Transition from thin to thick plate theory

For most solid masonry/concrete walls, the thin plate frequency limit for bending waves, $f_{B(thin)}$, falls in the mid or high-frequency range. Whilst it is referred to as a 'limit', thin plate theory does not instantly break down at a specific frequency. For direct airborne sound insulation across a solid homogeneous plate, thin plate theory can often be used up to a frequency of $4f_{B(thin)}$ (Ljunggren[5]). It is possible to treat $4f_{B(thin)}$ as a limit whilst acknowledging that it is not quite so clear-cut in practice. Errors from using thin plate theory in the range $f_{B(thin)} \leq f < 4f_{B(thin)}$ are usually less than 3 dB. This is often tolerable due to the uncertainty in predicting the total loss factor. Above $4f_{B(thin)}$ the airborne sound insulation effectively stops increasing with frequency and reaches a plateau with dips due to thickness resonances across the plate.

5 CONCLUSIONS

For homogeneous plates found in buildings, the examples in this paper indicate the close agreement that can be achieved between theory and measurement for airborne sound insulation. The measurements were taken under controlled laboratory conditions hence these represent 'best case' scenarios because additional measurements were needed to determine the material properties and damping of the plates. These examples show that difficulties in accurately predicting the airborne sound insulation are not restricted to frequency bands where the plates have a dearth of bending modes. In fact it is found that reasonable estimates can usually be found when the statistical mode count $N_s \geq 1$ for the plate system, and the geometric average of the modal overlap factors for the room (source or receiving) and the plate $M_{av} \geq 1$. Strategies were proposed to deal with difficulties in predicting sound transmission in the vicinity of the critical frequency. These were shown to be effective for masonry/concrete plates where the critical frequency typically occurs in frequency bands with low mode counts, and plasterboard, wood or metal plates where the critical frequency typically occurs in bands with high mode counts.

6 REFERENCES

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