Comparison of the Sound Transmission Variability with Published Results on Coupling Loss

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Résumé

Alors que la plupart des prédictions en matière d'acoustique et de conception des bâtiments utilisent invariablement des modèles publiés et facilement disponibles, une tentative de quantifier les limites de fiabilité qui couvrent la plupart des cas serait très précieuse. Par exemple, il est démontré que certains paramètres (i.e. les dimensions de la pièce, la position des panneaux, l'absorption de la pièce, etc) ont un effet substantiel sur la réduction du bruit et le facteur de perte de couplage, ce dernier étant un facteur très important pour prédire la transmission du son en utilisant l'analyse statistique de l'énergie (SEA). Un modèle SEA a été mis en œuvre et utilisé ici pour la prédiction des facteurs de perte de couplage entre deux pièces. Ainsi, l'objectif principal de cette recherche est d'effectuer une étude paramétrique initiale des facteurs de perte de couplage, puis de comparer leur variabilité avec les courbes théoriques des limites supérieures et inférieures, précédemment présentées dans la littérature pour le couplage des structures. L'utilité de l'EES comme cadre d'analyse peut être évaluée par l'estimation de la variance et des intervalles de confiance. En outre, la moyenne spatiale de la pression acoustique carrée pour chaque sous-système SEA a été estimée via un modèle de synthèse des modes de composantes développé dans un article précédent. En résumé, les pressions acoustiques de la pièce ont été obtenues par une procédure synthèse des modes de composantes et ensuite utilisées dans un modèle SEA où les facteurs de perte de couplage équivalents ont été évalués sur la base des hypothèses SEA. L'influence d'autres paramètres SEA, tels que la densité modale et le chevauchement modal, a également été prise en compte.

Keywords: Transmission du son, analyse statistique de l'énergie, facteur de perte de couplage, étude paramétrique

Abstract

Whilst most predictions in building acoustics and design invariably use published and readily available models, some attempt to quantify confidence limits that cover most cases would be invaluable. For instance, the parameters (e.g. room dimensions, panel position, room absorption, etc.) are shown to have a substantial effect on Noise Reduction (NR) and Coupling Loss Factor (CLF), the latter being a very important factor for predicting sound transmission using Statistical Energy Analysis (SEA). A Statistical Energy Analysis (SEA) model was implemented and used herein for the prediction of CLFs between two rooms. Thus, the main goal this research is to make an initial parametric investigation for the Coupling Loss Factors (CLFs) and then compare their variability with theoretical upper and lower bound curves previously presented in the literature for structure coupling. The usefulness of SEA as a framework of analysis can be assessed by the estimation of variance and confidence intervals. In addition, the spatial-average mean square sound pressure for each SEA subsystem was estimated via a Component Mode Synthesis (CMS) model developed in a previous paper. In summary, the room acoustic pressures were obtained via a CMS procedure and subsequently used in a SEA model where the equivalent CLFs were evaluated on basis of SEA assumptions. The influence of other SEA parameters, such as modal density and modal overlap was also considered.

Keywords: Sound transmission, Statistical Energy Analysis, Coupling Loss Factor, Parametric study

1 Introduction

Although the phenomenon of sound transmission through partitions has been investigated over many years, the problem of low frequency sound insulation in buildings is still an active research area. Modal methods are widely used for the low-frequency analysis of vibro-acoustic problems, including the problem of sound transmission between coupled rooms. On the other hand, Statistical Energy Analysis (SEA) is widely used for mid and high frequency analysis of vibro-acoustic problems. A general introduction to SEA is given in numerous references [1-3] which include discussion on the background theories.

The main advantages of SEA are: it can allow response predictions at mid and high frequencies, where other numerical methods cannot be used; the SEA method involves relatively few degrees of freedom in comparison to other determinist models. The main SEA disadvantages are: the accuracy of predicted average energy is not guaranteed and the model is not capable of modelling local behaviour. Since statistical approaches give statistical answers, they are always subjected to some uncertainties.

The potential errors in the SEA predictions at low frequencies were investigated by Craik *et al* [1, 2]. It was shown that the vibration level difference between two coupled building structures fluctuates with frequency significantly since building structures have few modes at low frequencies.

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The Coupling Loss Factor (CLF) is a statistical quantity defined in terms of the average behaviour of an ensemble of similar subsystems. It relates the power flow between connected subsystems to the stored energy in the transmitting subsystem. It is well-known that significant fluctuations with frequency are observed in the low frequency range. The 'modal overlap factor' is also an important parameter. It is a measure of the degree to which resonant behaviour dominates the response. At low modal overlap, which usually corresponds to low frequency, the actual energy transfer between subsystems can differ considerably from that predicted using the CLF estimates determined from the power transmission efficiencies for semi-infinite subsystems. These fluctuations are in part due to the particular realization of the subsystems within the notional ensemble.

This paper describes an initial parametric investigation into the variability of the effective CLF, in terms of the modal overlap factor and the number of modes in a frequency band. The reliability and accuracy of this empirical model was discussed in comparison with previously published models.

First, the influence of the room dimensions on the CLF has been considered. Numerical experiments were made using sets of simulations, which follow a pre-established analysis pattern. In other words, this analysis was based on the variation of a particular geometrical parameter whilst keeping the others unaltered. Thus, the assessment of the variability and sensitivity of transmission efficiencies to a chosen parameter could be made. In general, there might be some interdependence, but this is outside of the scope of this initial investigation. A total number of 11 iterations were made in order to simulate the original and modified models in each case. The models were obtained by logarithmically varying one dimension at a time (height, width, or depth of receiving room) whilst keeping the others unaltered. For the baseline model, initially a total number of 48, 35 and 97 modes were used for room 1, room 2 and partition respectively. The frequency range and volume sizes considered dictated the choice of the number of modes used. Next, the effects of room absorption on transmission are considered and discussed. Finally, the influence of different panel positions in the common wall between rooms on CLF is considered.

Generally, the sound transmission mechanism in a real building involves a great number of different and complex transmission paths. In SEA these paths are classified as direct and flanking paths [4]. In this study, only the direct transmission was considered in the implemented SEA model, so that the problem was described as one room emitting noise and another room receiving it. The variation of NR with the ratio of the receiving room height to the source room height was considered.

The spatial averaged, time averaged energy for each acoustic subsystem was evaluated from this baseline model, which consisted of two rooms coupled by a limp partition. Later on this paper, one can see that it was necessary to use a limp panel model, so that some parameters (in terms of CLF variability) defined in the literature could be used herein for comparison.

The performance of a building can be predicted by a basic SEA technique, which is described in refs. [1, 2]. The power flow between SEA subsystems can be described by the coupling between them that takes places at their boundaries.

The results that are discussed herein were obtained via simulations using the CMS model developed previously [5]. The analysis was based on considering the influence of some variations in the 'input' parameters, which are required in the pre-processing stage of a numerical experiment, and on the subsequent sound transmission mechanisms of typical building configurations.

Therefore, the main goal of this paper is to examine the variability of CLF to some architectural parameters via a parametric study. This study is aimed at providing not only a better understanding of the sound transmission mechanism in itself but also to produce a useful set of data which for instance can be used by acousticians as input data for a SEA analysis. This data might also be useful for optimizing sound insulation in buildings at low frequencies, where the modal behaviour of rooms strongly influences the transmission. These considerations are discussed in detail in the section 3.

2 The SEA Model

The simplest method of estimating the CLFs is presented here for the sake of simplicity and in order to provide results that can be compared with published data [6]. Although this approach could be used to reduce the computing time required to obtain the CLFs, it is subjected to the common limitations of the Component Mode Synthesis (CMS) method.

The main assumption here is that there are only two subsystems in the SEA model, which correspond to the source and receiving rooms. It seems that this assumed condition is reasonable, as the non-resonant transmission or forced transmission is the most important contribution to the transmission mechanism. In SEA modelling, one of the most important parameters is the modal density. It is defined as the number of modes that lie in an increment of frequency. For instance, the modal density for a standard room is given by [2]

$$n(f) = \frac{4\pi f^2 V}{c_0^3} + \frac{\pi f S'}{2c_0^2} + \frac{L'}{8c_0} \qquad (1)$$

where V is the room volume, S' is the total surface area of the room and L' is the total perimeter of the room. Table 1 shows the variation of the modal density for room 2 in the one-third octave band with centre frequency at 250 Hz. The modal density for room 1 was equal to 0.419 in the same frequency band and L_{y1} = 1.8 m. According to Figure 1, the power balance equations for the two coupled rooms (which are represented by the subscripts *1* and *2* and excited one at a time are then given by [3]

$$P_{1,in}^1 = P_{1,diss}^1 + P_{12}^1 = \omega(\eta_1 E_1^1 + \eta_{12}^1 E_1^1 - \eta_{21}^1 E_2^1) \qquad (2)$$

$$0 = P_{2,diss}^1 + P_{21}^1 = \omega(\eta_2 E_2^1 + \eta_{21}^1 E_2^1 - \eta_{12}^1 E_1^1) \qquad (3)$$

$$P_{2,in}^2 = P_{2,diss}^2 + P_{21}^2 = \omega(\eta_2 E_2^2 + \eta_{21}^2 E_2^2 - \eta_{12}^2 E_1^2) \qquad (4)$$

$$0 = P_{1,diss}^2 + P_{12}^2 = \omega(\eta_1 E_2^2 + \eta_{12}^2 E_1^2 - \eta_{21}^2 E_2^2) \qquad (5)$$

where η_i is the internal loss factor for each subsystem, E_i is the spatial averaged, time averaged energy in subsystem i.

The CLF from subsystem i to subsystem j is denoted η_{ij}, ω is the angular frequency in radians per second, P_{diss} and P_{in} are the time averaged dissipated and input powers respectively, P_{ij} is the power transmitted from subsystem i to subsystem j. The superscripts 1 and 2 indicate in which subsystem the excitation is applied separately one at a time.

Therefore, by assuming that $\eta_{ij}^1 = \eta_{ij}^2$ and according to the concept of power injection method [2, 3], the 'effective' CLF η_{ij} for two conservatively coupled subsystems 1 and 2 can be obtained by rearranging the equations (3) and (5) as

$$\begin{Bmatrix} \eta_{12} \\ \eta_{21} \end{Bmatrix} = \frac{1}{\omega} \begin{bmatrix} E_1^1 & -E_2^1 \\ -E_1^2 & E_2^2 \end{bmatrix}^{-1} \begin{Bmatrix} \omega\eta_2 E_2^1 \\ \omega\eta_1 E_1^2 \end{Bmatrix} \qquad (6)$$

A limp panel model with nominal density equal to 8.1 kg/m^2 was considered. The thickness of the partition was 0.01 m. A Reverberation Time (RT) T_{60}=1 s was considered herein.

For instance, the fraction of maximum stored energy of subsystem 1 transmitted to subsystem 2 per cycle is $2\pi\eta_{12}$, where η_{12} is the CLF. This is defined in the similar way to the definition of the loss factor η of a subsystem, namely $2\pi\eta$ is the fraction of the maximum stored energy which is lost or dissipated per cycle. This can be lost through mechanical and thermal means or can take into account losses due to other subsystems, which have not been explicitly defined.

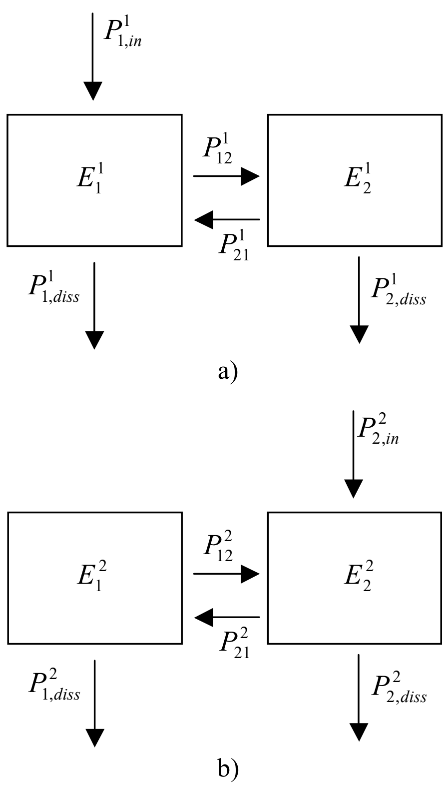


Figure 1: SEA models of two rooms separated by a single-leaf partition approximated by a two-subsystem model. Therefore, only the non-resonant transmission path is considered. a) Power is injected into subsystem 1; b) Power is injected into subsystem 2. The subscripts 'i j' denote the power flow from subsystem 'i' to subsystem 'j' and the superscript indicates which subsystem is under direct excitation.

The spatial average time averaged energy for an acoustic subsystem i can be obtained according to the general expression [1]

$$E_i = \left(\frac{\langle \overline{p_i^2} \rangle V_i}{\rho_0 c_0^2} \right) \qquad (7)$$

where V_i is the volume of subsystem i and $\langle \overline{p_i^2} \rangle V_i$ is the spatial averaged mean square pressure in subsystem i. This has been obtained by using the CMS model derived in [1], which was modified to calculate the coupling between the volumes by a limp panel. The calculations were run with no dissipation in the limp panel.

Likewise, the total loss factor of a particular acoustic subsystem i may be approximated by the expression [1]

$$\eta_i = \frac{13.8}{\omega T_{60,i}} \qquad (8)$$

where $T_{60,i}$ is the RT of the subsystem i.

For the SEA simulations $T_{60,i}$ was constant and equal to 1.0 s. Equation (8) is a general expression for the total loss factor which only gives the damping loss factor for weakly coupled systems (i.e. CLFs << internal loss factor) as measurements for the RT will normally include some effect of dissipation from other subsystems connected to the volume. Therefore, a value of T_{60} was set and then used to infer the damping loss factor.

Although the CLFs are only defined for finite systems, an expression for the CLF of 'semi-infinite' acoustic subsystems can be obtained by assuming diffuse field conditions in both rooms. In addition, it is assumed that there is direct transmission between rooms, where forced transmission is the most important contribution. Thus, the CLF η_{12} from subsystem 1 to subsystem 2, is given approximately by [1]

$$\eta_{ML} \approx \frac{c_0 \, S \, \tau_\infty}{4 \, \omega \, V_1} \qquad (9)$$

where τ_∞ is the diffuse transmission efficiency obtained via Mass Law theory described in ref. [5], V_1 is the volume of the source room and S is the partition area.

The CLF η_{21} can also be obtained from η_{12} by the consistency relationship [3]

$$n_1\eta_{12} = n_2\eta_2 \qquad (10)$$

Where n_1 and n_2 are the modal densities (see equation 1) for subsystems 1 and 2 respectively.

The variability of the CLFs with the subsystem properties in SEA models have been recently studied by Park et al [6]. A sensitivity analysis was performed using an analytical model for two coupled plates. The Dynamic Stiffness Method was used in the evaluation of their model. Thus, an 'empirical model' for the variability of CLF (σ^2) was derived for two coupled finite plates according to the expression [6]

$$\sigma^2 = \frac{6}{M_{comb} + N_{comb}^2/16} \qquad (11)$$

where

$$M_{comb} = \frac{2M_1 M_2}{M_1 + M_2} \qquad (12)$$

$$N_{comb} = \frac{2N_1 N_2}{N_1 + N_2} \tag{13}$$

where (σ^2 is the variance of the dB values; M_{comb} and N_{comb} are the combined modal overlap factor and number of modes respectively, M_1 and M_3 are the modal overlap factors for subsystems 1 and 2 respectively. They are defined as the ratio of the modal bandwidth to the average frequency spacing between modes [2]. Similarly, N_1 and N_2 are the mode counts for subsystem 1 and 2.

It has been established in ref. [6] that this variance represented a 95.7% confidence interval for all set of data considered for two coupled rectangular plates. Nevertheless, it is not known whether the acoustic system presented herein can be represented by the same value of confidence interval.

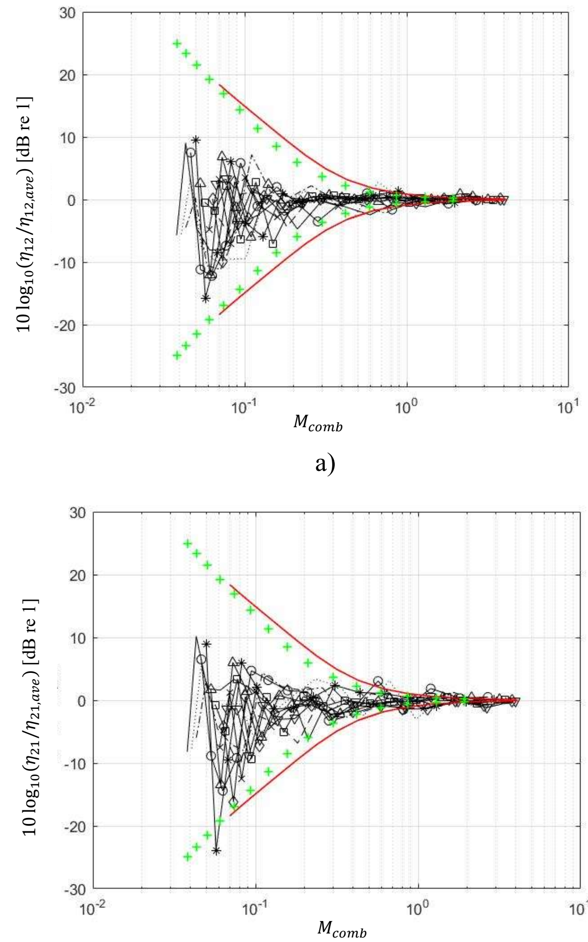
3 Results and discussions

Results were obtained in terms of the variation of the CLF ratio with the combined modal overlap factor M_{comb} for different room configurations. The numerical frequency range covered was 0 to 500 Hz, although the results are only plotted at values where at least one non bulk mode exists in either room. Firstly, the CLF ratio, in Figures 2-6, was defined as the ratio of the 'effective' CLF (equation 6), obtained for a particular system configuration, to the averaged 'effective' CLF, which was obtained by considering the mean value over all of the different configurations of a particular parameter, e.g. the height ratio of the rooms. The results were calculated in sets of one-third octave bands. Figures 2-4 show the variation of CLF ratio with M_{comb} whilst varying the height, width and depth ratio of the rooms. In Figure 2, the source room height was fixed and equal to 1.8 m. The receiver height varied from 1.8 to 18 m (see Table 1 below).

Table 1: Variation of room parameters with the height ratio L_{y2}/L_{y1}. L_x, L_y and L_z are room depth, height and width respectively. $n(f)$ is the modal density in the highest 1/3 octave band with centre frequency equal to 250 Hz and f_{Schr} is the *Schroeder* frequency (Hz) above which the acoustic field is assumed to be diffuse. The subscripts 1 and 2 represent the source and receiving rooms respectively.

L_{y2}/L_{y1}	L_{y2}(m)	$n_1(f)$	$n_2(f)$	$f_{1,Schr}$	$f_{2,Schr}$
1.000	1.800	0.419	0.290	430.3	527.0
1.259	2.266	0.419	0.356	430.3	469.7
1.585	2.853	0.419	0.438	430.3	418.6
1.995	3.591	0.419	0.542	430.3	373.1
2.512	4.522	0.419	0.673	430.3	332.5
3.162	5.692	0.419	0.837	430.3	296.4
3.981	7.166	0.419	1.045	430.3	264.1
5.012	9.022	0.419	1.305	430.3	235.4
6.309	11.356	0.419	1.634	430.3	209.8
7.943	14.297	0.419	2.047	430.3	187.0
10.000	18.000	0.419	2.567	430.3	166.6

It is seen that the results lay within the bounds for most of the M_{comb} range. At higher frequencies, the CLF ratio values vary within the range ± 1 dB. Likewise, Figures 3 and 4 also show that the convergence of the results rapidly increases with the combined modal overlap factor.



a)

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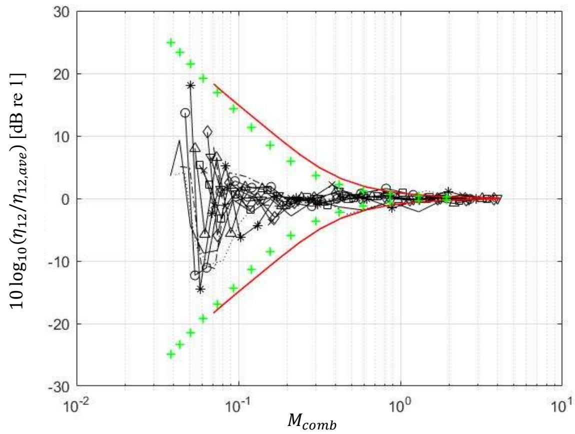
Figure 2: Variation of CLF ratio with the combined modal overlap factor M_{comb} for different values of height ratio (L_{y2}/L_{y1}) compared to the average over all of the height variations.
(a): $10\log_{10}(\eta_{12}/\eta_{12,ave})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{21,ave})$ [dB re 1].
The height of room 1 (L_{y1}) is 1.8 m. The height of room 2 (L_{y2}) varies from 1.8 to 18 m; — 1.8 m; ⋯⋯ 2.27 m; -·- 2.85 m; -o- 3.59 m; -*- 4.52 m; -Δ- 5.69 m; -□- 7.16 m; -x- 9.02 m; -◊- 11.36 m; -∇- 14.29 m; ---- 18 m; +++ bounds ($\pm 2\sigma$) for L_{y2} = 1.8 m; — bounds ($\pm 2\sigma$) for L_{y2} = 18 m.

Figure 3 shows that at higher modal overlap factors, the CLF ratio values tend to be less than ± 0.5 dB. At low frequencies, variability of the effective CLFs is particularly large, while it generally reduces as frequency increases.

However, in Figure 4, the case of varying the depth shows large variation at high frequencies. It might be due to the influence of axially directed modal pattern of pressure that propagates above its cut-off frequency.

Figure 5 shows the variation of CLF ratio with M_{comb} for different values of the RT ratio ($T_{60,2}/T_{60,1}$). The RT of the source room was fixed and equal to 1.0 s. However, for the receiving room it was varied from 1.0 s to 0.2 s. It appears



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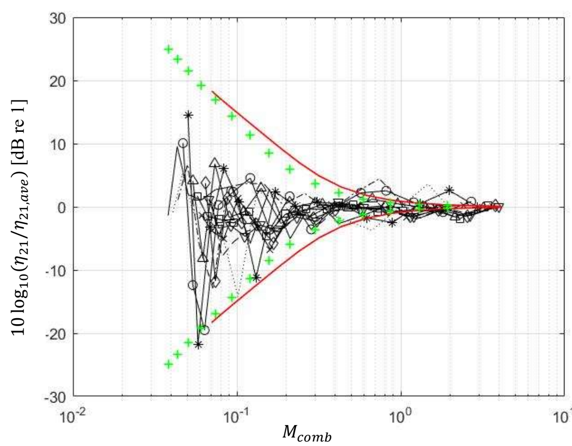

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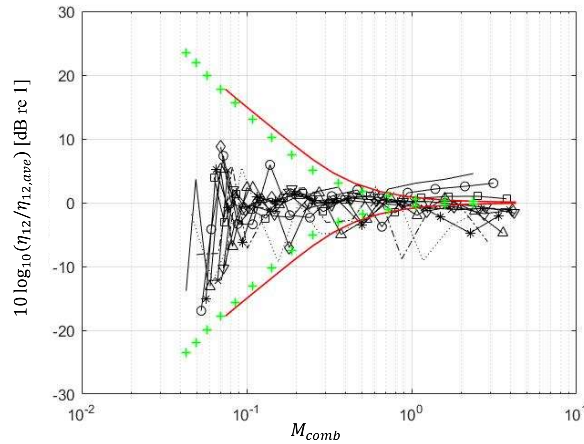
Figure 3: Variation of CLF ratio with combined modal overlap factor M_{comb} for different values of width ratio (L_{z2}/L_{z1}) compared to the average over all of the width variations.
(a): $10\log_{10}(\eta_{12}/\eta_{12,ave})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{21,ave})$ [dB re 1].
The width of room 1 (L_{z1}) is 2 m. The width of the room 2 (L_{z2}) varies from 2 to 20 m; — 2 m; ······ 2.52 m; -·- 3.17 m; -o- 3.99 m; -*- 5.02 m; -Δ- 6.32 m; -□- 7.96 m; -x- 10.02 m; -◊- 12.62 m; -∇- 15.89 m; ---- 20 m. **+++** bounds ($\pm 2\sigma$) for L_{z2} = 2 m; — bounds ($\pm 2\sigma$) for L_{z2} = 20 m.



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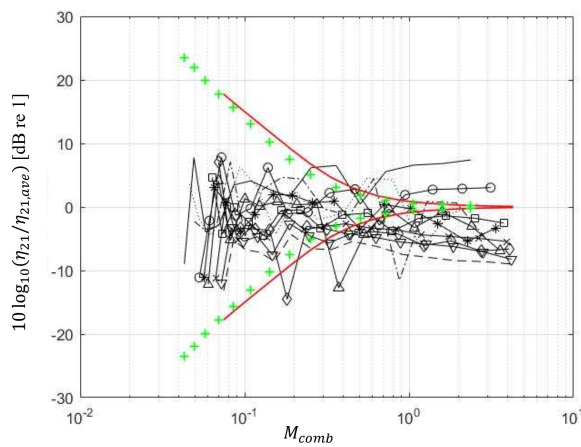

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Figure 4: Variation of CLF ratio with the combined modal overlap factor M_{comb} for different values of depth ratio (L_{x2}/L_{x1}) compared to the average over all depth variations.
(a): $10\log_{10}(\eta_{12}/\eta_{12,ave})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{21,ave})$ [dB re 1].
The depth of room 1 (L_{x1}) is 3 m. The depth of the room 2 (L_{x2}) varies from 3 to 30 m; — 3.00 m; ······ 3.77 m; -·- 4.76 m; -o- 5.99 m; -*- 7.54 m; -Δ- 9.49 m; -□- 11.94 m; -x- 15.04 m; -◊- 18.93 m; -∇- 23.83 m; ---- 30 m. **+++** bounds ($\pm 2\sigma$) for L_{x2} = 3 m; — bounds ($\pm 2\sigma$) for L_{x2} = 30 m.

that the most significant variations in terms of the CLF ratios occurred for the case of varying the RT of the source room whilst keeping the RT of the receiving room constant. As the RT of both rooms increase, the variation in the effective CLF becomes small. At high frequencies (above the Schroeder frequency [5]) when the RT is decreased, the modal overlap factor is increased and vice-versa. This results in a higher probability of better coupling between individual modes and therefore lower sound insulation.

Figure 6 shows the variation of CLF ratio with M_{comb} for different values of panel position on the common rigid wall. Very small variation is observed at the lower values of M_{comb}, i.e. at lower frequencies for the source and receiving rooms where there are few if any acoustic modes and transmission is low. On the other hand, significant variations occur in the range where acoustic modes exist. These variations indicate very high spatial coupling sensitivity. When the frequency increased, oblique modes tended to be dominant in the rooms and the difference between the panel positions became less important on the sound insulation.

The CLF ratio, in Figures 7 and 8, was calculated as the ratio of the 'effective' CLF to the one obtained using equation (9). Although an average result was used for reference, it did not converge to the diffuse incidence Mass Law. It is shown that the variation of CLF ratio, which is defined here as the ratio of the actual transmission to the diffuse incidence Mass Law transmission, with M_{comb} whilst varying the height and width of the receiving rooms.

In Figure 7, the source room height was fixed and equal to 1.8 m. The receiver height varied from 1.8 to 18 m. It is seen that the results approximately lay on the upper bound for

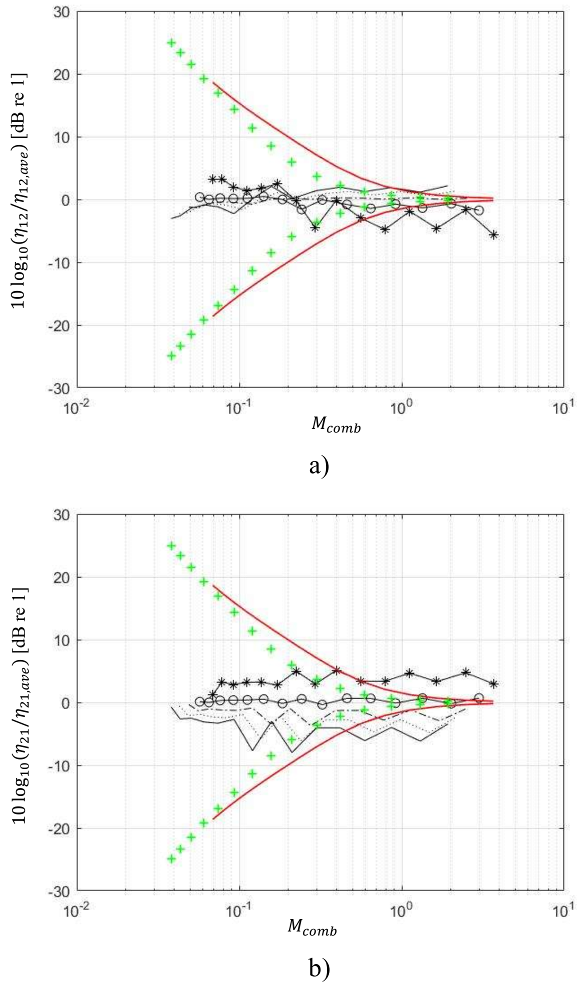


Figure 5: Variation of CLF ratio with the combined modal overlap factor M_{comb} for different values of RT ratio ($T_{60,2}/T_{60,1}$) compared to the average over all of the RT variations
(a): $10\log_{10}(\eta_{12}/\eta_{12,ave})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{21,ave})$ [dB re 1].
The RT of the room 1 ($T_{60,1}$) is 1.0 s. The RT of room 2 ($T_{60,2}$) varies from 1 s to 0.2 s; — 1 s; ⋯⋯ 0.8 s; -·- 0.6 s; -o- 0.4 s; -*- 0.2 s. +++ bounds ($\pm 2\sigma$) for $T_{60,2}$ = 1 s; — bounds ($\pm 2\sigma$) for $T_{60,2}$ = 0.2 s.

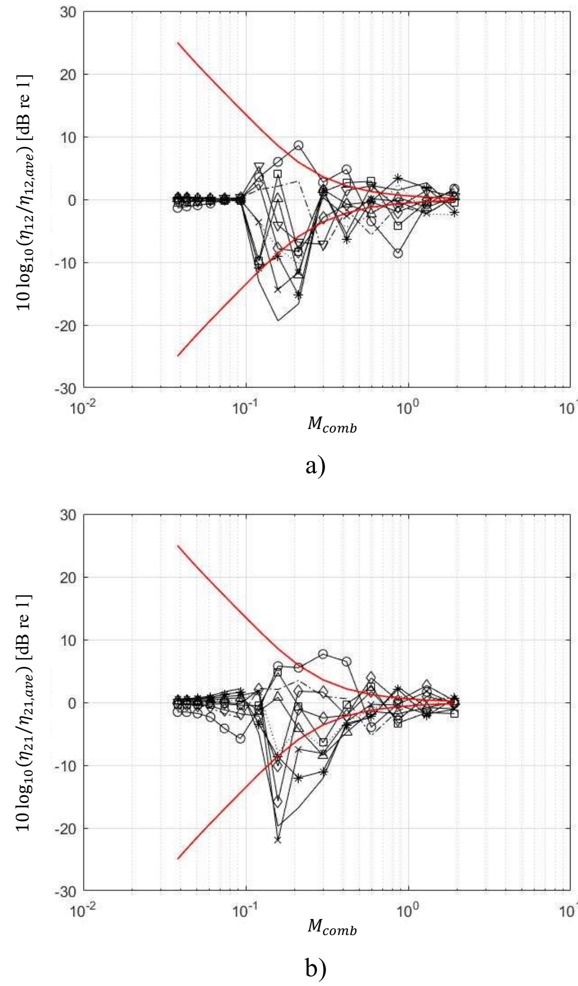


Figure 6: Variation of CLF ratio with the combined modal overlap factor M_{comb} for different values of panel position on the common wall compared to the average over all of the panel positions.
(a): $10\log_{10}(\eta_{12}/\eta_{12,ave})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{21,ave})$ [dB re 1].
— P_1; ⋯⋯ P_2; -·- P_3; -o- P_4; -*- P_5; -Δ- P_6; -□- P_7; -x- P_8; -◊- P_9; -∇- P_{10}. — upper and lower bounds ($\pm 2\sigma$) obtained from equation (11).

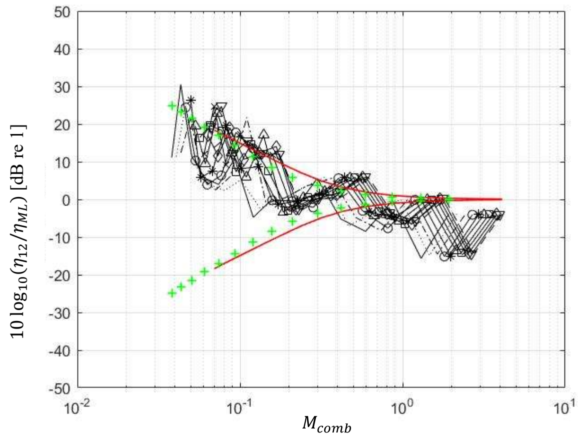
most of the M_{comb} range. However, they tend to diverge from the mass law results η_{ML} as the combined modal overlap increases.

Likewise, Figure 8 shows that the mass law results η_{ML} are lower than the 'effective' CLFs at low frequencies. These deviations at high frequencies might be due to effect of resonant modes in the source and receiving rooms included in the CMS model but not in the incident diffuse field mass law assumptions. In other words, this fact was predictable at low frequencies, where the diffuse incidence mass law overestimated the transmission efficiency due to the assumption of diffuse field behavior in the source room. To quantify the reliability of results from the SEA predictions, an investigation on the confidence interval of the coupling between the partition and the acoustic room is also required.

There are many uncertainties and potential errors in the low to mid frequency range that still need to be contemplated in the SEA models. At low modal overlap (M< 0.4) the results fluctuate considerably, and most are found to fall within the bounds described herein. The results below the first cut-on frequency of either room were discounted as SEA assumptions would not be valid. For multiple subsystems models the CLFs will not be independent and the SEA prediction requires more detailed investigation.

The 'effective' CLF tends to be lower than the η_{ML} when frequency increases. For a large bandwidth the number of modes in a frequency band is much more important than the modal overlap factor.

In summary, the results obtained shows the variability in the CLF using two coupled acoustic rooms as an example to quantify the uncertainties in the CLF. The CMS was used to quantify the sound pressure response in a wide frequency range. It is seen that a wide range of parameter investigations



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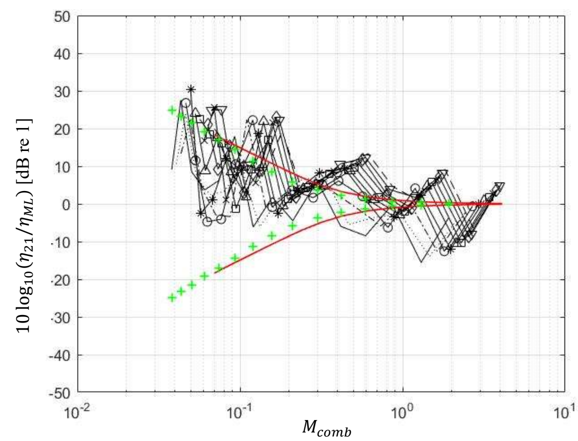

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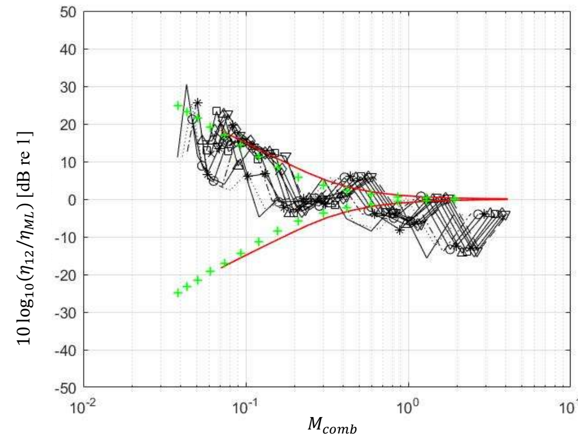
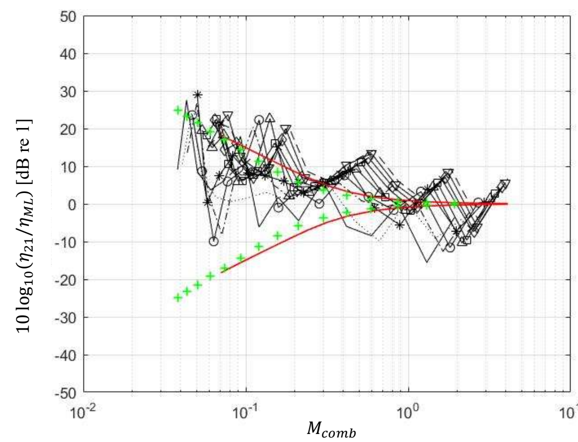
Figure 7: Variation of CLF ratio with the combined modal overlap factor M_{comb} for different values of height ratio (L_{y2}/L_{y1}) compared to the diffuse incidence Mass Law.
(a): $10\log_{10}(\eta_{12}/\eta_{ML})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{ML})$ [dB re 1].
The height of room 1 (L_{y1}) is 1.8 m. The height of room 2 (L_{y2}) varies from 1.8 to 18 m; — 1.80 m; ······ 2.27 m; -·- 2.85 m; -o- 3.59 m; -*- 4.52 m; -Δ- 5.69 m; -□- 7.16 m; -x- 9.02 m; -◊- 11.36 m; -∇- 14.29 m; ---- 18 m; **+++** bounds ($\pm 2\sigma$) for L_{y2} = 1.8 m; — bounds ($\pm 2\sigma$) for L_{y2} = 18 m.



a)



b)

Figure 8: Variation of CLF ratio with combined modal overlap factor M_{comb} for different values of width ratio (L_{z2}/L_{z1}) compared to the diffuse incidence Mass Law.
(a): $10\log_{10}(\eta_{12}/\eta_{ML})$ [dB re 1];
(b): $10\log_{10}(\eta_{21}/\eta_{Ml})$ [dB re 1].
The width of room 1 (L_{z1}) is 2 m. The width of the room 2 (L_{z2}) varies from 2 to 20 m; — 2 m; ······ 2.52 m; -·- 3.17 m; -o- 3.99 m; -*- 5.02 m; -Δ- 6.32 m; -□- 7.96 m; -x- 10.02 m; -◊- 12.62 m; -∇- 15.89 m; ---- 20 m. **+++** bounds ($\pm 2\sigma$) for L_{z2} = 2 m; — bounds ($\pm 2\sigma$) for L_{z2} = 20 m.

was performed using two acoustics volumes separated by a limp panel. At low modal overlap the CLFs fluctuated with frequency considerably, whereas the variability generally reduced as frequency increased. As the modal overlap factor increases, the bounds of the SEA simulation decrease slightly. It was shown that the SEA predictions are more reliable when the modal overlap factor and frequency bandwidth are large [6], as expected according to the fundamental SEA hypothesis.

4 Conclusion

Numerical simulations for the investigation of the variation of CLF ratio with the combined Modal Overlap Factor were obtained for a limp panel model. Hence, there was no resonance contribution of the panel on the frequency response of the system. Even though there was no stiffness term in the equation of motion of the panel, i.e. the panel was limp, its mass term was allowed to contribute.

The sound transmission results thus had no resonant panel behaviour, and the variation of results were mainly due to the panel position and also the matching or separation of the room natural frequencies (i.e. modal overlap).

The results were then compared to previously published envelope results given for structure-to-structure coupling limits (Park *et al* in reference [6]). It is seen that most of the results, which are presented in terms of CLF ratio, fit reasonably well within the published envelope results [6] for the frequency range investigated. Only the results due to variation of the panel position are not such a good comparison and it is suspected that this might be due to extreme sensitivity of the modal model to the spatial coupling terms. The actual fluid-structure interaction problem considered herein was evaluated at very low frequencies. In addition, small acoustic volumes were considered for the baseline models. Consequently, small values of Modal Overlap Factors were obtained. The envelope results presented by Park *et al* [6] were developed on the basis of only two coupled subsystems, namely two coupled rectangular plates. Hence, there was no 'intermediate' connection between them, such as a beam. In other words, the modal model formulated here was equivalent to the structure-to-structure coupling problem published in ref. [6], as the model herein considered the contribution of a limp partition with no modes on the transmission mechanism.

No attempt has been made here to produce alternative limits for the acoustic-structural problem, as it does not appear to be particular easy to solve or generalize.

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References

[1] R. J. M. Craik. The Prediction of Sound Transmission Through Buildings Using Statistical Energy Analysis 1981 *Journal of Sound and Vibration* **82(4),** 505-516.

[2] R. J. M. Craik. Sound Transmission through Buildings using Statistical Energy Analysis. *Gower Publishing Limited*, U.K., 1996

[3] R. H. Lyon and R. G. Dejong. Theory and Application of Statistical Energy Analysis. Boston: Butterworth-Heinemann, Second Edition, 1995.

[4] F.X. Magrans. Definition and Calculation of Transmission Paths Within An S.E.A. Framework. *Journal of Sound and Vibration* 165(2), 277-283. 1992.

[5] M. D. C. Magalhaes. A New Efficient Matrix Algorithm for a 3D Component Mode Synthesis (CMS) Model Used on Sound Transmission Problems. *Computer Modeling in Engineering and Science*, 115 (6), 441-465A, 2015.

[6] W. S. Park, D. J. Thompson and N. S. Ferguson. An Empirical Model for the Variability of the Coupling Loss Factor. *Institute of Sound and Vibration Research – ISVR. Technical Memorandum* 874, 2001.