
A thesis submitted in fulfillment of the requirements for the Degree of Doctor of Philosophy at the University of Canterbury by Jeffrey P. Mahn

University of Canterbury
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# Table of Contents

## Acknowledgments

## Abstract

## Symbols

## 1. Introduction

1.1. Noise and Annoyance in Buildings

1.2. Current Regulation to Reduce Noise in Buildings

1.3. Controlling Flanking Transmission

1.4. Development of EN12354

1.5. Lightweight Building Constructions

1.6. Current Research

## 2. Limitations to the Application of the EN12354 Method

2.1. Introduction

2.2. Restriction to First Order Flanking Paths

2.3. Inclusion of Only Bending Waves

2.4. Elements as Ideal SEA Subsystems

2.5. Reciprocity

2.6. Gaussian Probability Density Functions

2.7. Uncertainty

## 3. Identification of Probability Density Functions

3.1. Motivation

3.2. Distributions Assessed in this Study

3.3. Goodness-of-Fit Tests

3.4. Analysis of Measured Data

3.5. Monte Carlo Simulations
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.6. Results</td>
<td>23</td>
</tr>
<tr>
<td>3.7. Discussion of PDF Results</td>
<td>25</td>
</tr>
<tr>
<td>3.8. Discussion of the PDF Study Results</td>
<td>38</td>
</tr>
<tr>
<td><strong>4. Uncertainty of the EN12354 Method</strong></td>
<td>39</td>
</tr>
<tr>
<td>4.1. Introduction</td>
<td>39</td>
</tr>
<tr>
<td>4.2. Method</td>
<td>39</td>
</tr>
<tr>
<td>4.3. Calculation of the Uncertainty</td>
<td>42</td>
</tr>
<tr>
<td>4.4. Discussion</td>
<td>57</td>
</tr>
<tr>
<td><strong>5. Proposed Changes to ISO10848</strong></td>
<td>60</td>
</tr>
<tr>
<td>5.1. Calculation of the Direction Averaged Velocity Level Difference</td>
<td>60</td>
</tr>
<tr>
<td>5.2. Proposed Estimates Based on Log-Normal PDF’s</td>
<td>62</td>
</tr>
<tr>
<td><strong>6. Reciprocity</strong></td>
<td>70</td>
</tr>
<tr>
<td>6.1. Introduction</td>
<td>70</td>
</tr>
<tr>
<td>6.2. Best Estimate of the Flanking Transmission Factor</td>
<td>71</td>
</tr>
<tr>
<td>6.3. Comparison Between Estimates</td>
<td>72</td>
</tr>
<tr>
<td>6.4. Proposed Correction Factor</td>
<td>74</td>
</tr>
<tr>
<td>6.5. Discussion</td>
<td>76</td>
</tr>
<tr>
<td><strong>7. Separation of the Resonant Component of the Sound Reduction Index</strong></td>
<td>78</td>
</tr>
<tr>
<td>7.1. Introduction</td>
<td>78</td>
</tr>
<tr>
<td>7.2. Separation Theory</td>
<td>79</td>
</tr>
<tr>
<td>7.3. Experimental Method</td>
<td>84</td>
</tr>
<tr>
<td>7.4. Comparison between Separation Methods</td>
<td>86</td>
</tr>
<tr>
<td>7.5. Comparison between Measured and Calculated Flanking Sound Reduction Indices</td>
<td>95</td>
</tr>
<tr>
<td>7.6. Discussion</td>
<td>102</td>
</tr>
</tbody>
</table>
8. Evaluation of the EN12354 Method - L-Panels................................. 105
  8.1. Introduction .................................................................................. 105
  8.2. Overview of the ESEA Models ...................................................... 105
  8.3. Element Properties ....................................................................... 108
  8.4. Comparison between EN12354 and ESEA Predictions .................. 110
  8.5. Discussion of L-Panel Results ...................................................... 124

9. Evaluation of the EN12354 Method - Field Testing ......................... 127
  9.1. Introduction .................................................................................. 127
  9.2. Summary of the Assessed Flanking Paths ..................................... 127
  9.3. Summary of the Measurements .................................................... 130
  9.4. Summary of the Calculations ....................................................... 134
  9.5. Measured Flanking Sound Reduction Index ................................... 137
  9.6. Evaluation of Each Flanking Path ............................................... 138
  9.7. Evaluation of the Apparent Flanking Sound Reduction Index .......... 147
  9.8. Walls Versus Smaller Elements .................................................... 148
  9.9. Discussion of Field Testing Results .............................................. 154

10. Observations Regarding Wood versus Metal Studs ....................... 156

11. Conclusions and Recommendations .............................................. 160
  11.1. Conclusions .............................................................................. 160
  11.2. Recommended Changes to EN12354 and ISO10848 ...................... 163
  11.3. Proposed Future Work ............................................................... 164

Appendix A: Derivation of the EN12354 Method ................................. 166
  A.1. Derivation of the Flanking Transmission Factor ............................ 166
  A.2. Estimate of the Flanking Transmission Factor ............................... 169
  A.3. Estimate of the Flanking Sound Reduction Index ......................... 170
  A.4. Modified Flanking Sound Reduction Index .................................... 171
  A.5. Vibration Reduction Index .......................................................... 173
Appendix B: Description of Panels for this Study ........................................184

B.1. Single Panels ..........................................................................................184
B.2. L-Shaped Panels ...................................................................................186
B.3. Damped Panels ......................................................................................191

Appendix C: Testing Procedure for L-Shaped Panels .................................194

C.1. Introduction ..............................................................................................194
C.2. Preparation for Testing ........................................................................194
C.3. Mounting of the Panels into the Test Rig ...........................................198
C.4. Installation of a Fame around Panel B ................................................199
C.5. Panel Notation .......................................................................................201
C.6. Double-Leaf Panel Testing Scenarios ...............................................201
C.7. Reverberation Time Measurements .....................................................202
C.8. Attachment of Electromagnetic Shaker .............................................203
C.9. Attachment of Accelerometers ..............................................................205
C.10. Measurement of the Intensity Sound Reduction Index of the Flanking Element .........................................................206

Appendix D: Comparison of Reverberant Room Data .................................207

Appendix E: Generation of Random Observations ......................................214

E.1. Introduction .............................................................................................214
E.2. Generation of Random Values from a Gaussian Distribution ..............214
E.4. Generation of Random Values from a Gamma Distribution ...............215
Appendix K : Sound Reduction Indices for the Field Testing .............276

K.1. Introduction ........................................................................................................... 276
K.2. Total Sound Reduction Index .............................................................................. 276
K.3. Resonant Sound Reduction Index ....................................................................... 278

Appendix L : Field Testing Procedure .................................................................280

L.1. Introduction ......................................................................................................... 280
L.2. Preparation of the Rooms .................................................................................. 280
L.3. Sound Pressure Measurements ........................................................................ 280
L.4. Sound Intensity Measurements ......................................................................... 281
L.5. Velocity Level Difference Measurements ......................................................... 281

Appendix M : Calculation of the Apparent Sound Reduction Index ......283

M.1. Predictions ......................................................................................................... 283
M.2. Measured Data .................................................................................................. 284

Appendix N : Equipment ..........................................................................................286

N.1. List of Equipment Used for this Study ............................................................... 286
N.2. Defective Brüel & Kjær Accelerometers ......................................................... 287

Appendix O : List of Publications from this Study .............................................289

References..................................................................................................................290
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Abstract

The standard, EN12354-1 describes a simplified statistical energy analysis (SEA) model to predict the apparent sound reduction index between two rooms inclusive of the contributions of the flanking paths. There is interest worldwide in applying the EN12354 model to lightweight building elements. However, lightweight elements typically do not meet the requirements of an SEA subsystem and therefore applying the EN12354 model to these elements may result in inaccurate predictions.

The purpose of this investigation was to systematically evaluate the application of the EN12354 model to lightweight building constructions. The evaluation included the determination of the probability density functions and the propagated uncertainty of the calculations. Knowledge of the probability density functions resulted in alternative calculations of the structure-borne sound transmitted through the constructions. The uncertainty analysis revealed that the uncertainty of the predictions is directly affected by the variance of the vibratory field measured on the elements. The vibratory fields of lightweight elements typically show large variances and therefore the propagated uncertainty of the EN12354 predictions for these elements can be significant.

The investigation included measurements both in the laboratory and in the field. The results of the laboratory measurements were compared to both predictions using the EN12354 methods and ESEA models which included higher order flanking paths and non-resonant transmission paths. The field measurements included in this investigation were unique because the flanking intensity sound reduction indices of the elements in the source room were measured. The measurements allowed for the EN12354 predictions for each flanking element to be assessed instead of just the apparent sound reduction index between the rooms.
The study resulted in proposed correction factors for when reciprocity does not hold and proposed changes to ISO10848 to improve the accuracy of the predictions when the EN12354 method was applied to lightweight building elements. However, neither the proposed correction factors nor the proposed changes to ISO10848-1 could correct for the potentially large differences between the predicted and the measured results.

Based on the findings of this study, the use of the EN12354 model for the calculation of the apparent sound reduction index of lightweight elements is not endorsed. Lightweight constructions may not be categorized as ideal SEA subsystems due to the lack of diffuseness of the vibratory field. Furthermore, in order for EN12354 to be applied to lightweight constructions, a reliable method of calculating the resonant component of the sound reduction index of double-leaf elements is required. Therefore, statistical methods including the EN12354 method may be unsuitable for use for the prediction of flanking noise for lightweight building constructions.
## Symbols

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<th>Physical Quantity</th>
<th>Unit</th>
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<tr>
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<tr>
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<tr>
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<tr>
<td>$R$</td>
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<tr>
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<td>dB</td>
</tr>
<tr>
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<td>dB</td>
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<td>$\Lambda$</td>
<td>shape function = $l_2/l_1$</td>
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<td>$\mu_y$</td>
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<td>$\tau$</td>
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<td>$\tau_T$</td>
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Poisson’s ratio

\( \chi^2 \)  
chi-squared

correction term for the average velocity level difference  
\( \Psi_D \)
dB

correction term for the vibration reduction index  
\( \Psi_K \)
dB

correction term for the in-situ sound reduction index  
\( \Psi_R \)
dB

\( \omega \)  
angular frequency

rad/s

Subscripts

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>( d )</td>
<td>dissipated</td>
</tr>
<tr>
<td>( EN12354 )</td>
<td>EN12354 estimate</td>
</tr>
<tr>
<td>( i )</td>
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<td>( inc )</td>
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</tr>
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<td>( j )</td>
<td>property of element ( j )</td>
</tr>
<tr>
<td>( ISO140 )</td>
<td>uncertainty study in ISO140-2 Annex A</td>
</tr>
<tr>
<td>( lab )</td>
<td>laboratory measured value</td>
</tr>
<tr>
<td>( LN )</td>
<td>log-normal estimate</td>
</tr>
<tr>
<td>( NR )</td>
<td>non-resonant component</td>
</tr>
<tr>
<td>( P )</td>
<td>proposed estimate</td>
</tr>
<tr>
<td>( PG )</td>
<td>proposed estimate based on a Gaussian distribution</td>
</tr>
<tr>
<td>( proposed )</td>
<td>proposed estimate</td>
</tr>
<tr>
<td>( R )</td>
<td>resonant component</td>
</tr>
<tr>
<td>( situ )</td>
<td>in-situ value</td>
</tr>
<tr>
<td>( trans )</td>
<td>transmitted</td>
</tr>
<tr>
<td>( T )</td>
<td>total, measured value inclusive of the non-resonant and resonant components</td>
</tr>
<tr>
<td>( Type A )</td>
<td>Type A uncertainty</td>
</tr>
<tr>
<td>( Type B )</td>
<td>Type B uncertainty</td>
</tr>
</tbody>
</table>
1. Introduction

1.1. Noise and Annoyance in Buildings

Noise and annoyance in buildings has become an important subject as multi-tenancy buildings have become more common [1]. The number and the power of noise sources such as home appliances, home theatres and heating units have increased at the same time that people are becoming more aware of noise problems [2]. For example, noise complaint statistics from the Christchurch City Council for the year ending 30 June 2008 show that a total of 10,566 noise complaints were received by the council [3]. A study in the Netherlands showed 1/3 of all households experienced annoyance due to noise from neighbors [4]. In one Swedish study 82% ranked good sound insulation as their top priority among desired improvements [5]. A study in Germany reported that noise was a particularly important criterion when people chose a new residence or when they decided to change residences [6].

Beyond being just an annoyance, noise in buildings can influence the quality of life of those it affects. In a study by Grimwood [7], people reported that they had modified their behavior and their social lives due to poor sound insulation. Thirty-five percent of people said that both they themselves and visitors had to behave quietly and 18% said that they didn’t have visitors because of the noise they might make or might hear. Adults who indicated chronically severe annoyance by neighbor noise were found to have an increased health risk in the cardio-vascular system, the movement apparatus, as well as increased risk of depression and migraine headaches [8]. The effect of moderate annoyance to neighbor noise on children (0-17 years) is associated with elevated risks for respiratory symptoms and skin disease as a result of emotional stress [9]. Neighbor noise induced annoyance is therefore a highly underestimated risk factor for healthy housing [8]. Conversely, those who cause the noise may not understand why others are annoyed and may see those who complain as being unreasonable [10].

Van Dongen [4] found that the percentage experiencing annoyance and/or the degree of annoyance experienced in many cases clearly diminished in proportion to the quality of the
sound insulation. Robert Frost’s quote, “good fences make good neighbors” could be rewritten in this context to “walls with excellent sound insulation make for better neighbors.”

1.2. Current Regulation to Reduce Noise in Buildings

To reduce the problem of noise in buildings, the New Zealand Building Code Clause G6: “Airborne and Impact Sound” states minimum requirements for the sound transmission class of building elements of habitable spaces of household units including walls, floors and ceilings for new constructions in New Zealand [11]. However, it is often the case that building elements which were chosen based on laboratory testing according to ISO140 to meet the minimum requirements are found to have a lower sound reduction index once they are installed in the building, sometimes much lower [12, 13]. For example, in 1972, Lang [5] found that on average the weighted apparent sound reduction index in finished dwellings was 8 dB less than that which the partition itself should have provided according to laboratory measurement. Hongisto [14] found that the weighted sound reduction indices of walls measured in situ were up to 9 dB less than those measured in the laboratory due to the effects of flanking transmission.

Flanking transmission can strongly affect the sound insulation between rooms in a building. In buildings with heavy constructions of concrete or masonry, the flanking transmission may account for approximately fifty percent of the sound transmission between rooms with a common dividing partition [15]. Usually, each flanking path will be less important than the direct path but, since there are many such paths, flanking will often be important in the overall transmission of sound between two rooms [16]. If the source room and the receiver room have no common wall the entire sound transmission is through flanking paths [17].

This was the case in a study by Constable in 1938 which was one of the first articles published about flanking transmission [5, 18]. Constable investigated noise in one room flats in a building made of reinforced concrete. The walls between adjacent flats were considered to have an acceptable level of insulation, yet Constable found that a loudspeaker located in a corner flat could be heard not only in the adjoining flat, but also in other flats on the same floor as well as in flats several stories above the source room. All of the flats shared a
concrete outer wall. Constable attributed the transmission of noise to flanking transmission and concluded that in the case of monolithic concrete structures, the vibration acquired by a wall or floor when air-borne sound falls upon it can be transmitted with little attenuation, along the length of the wall, to neighboring rooms.

In a study by Langdon [19] on noise in multi-tenant buildings, two-thirds of the people questioned heard noise from neighbors of one sort of another. What was striking about the response was that little noise was heard through the party walls but a great deal of noise was reported as coming from other parts of the building such as stairways, corridors and entrances and from outside. Langdon notes that the findings draw attention to the need to consider sound insulation not merely in terms of individual components such as party walls or floors between adjoining residences, but in terms of the building as a whole. Therefore, flanking transmission must be taken into consideration at the design stage when choices are being made concerning walls and floors and the junctions between them.

1.3. Controlling Flanking Transmission

Unfortunately, noise problems in buildings due to flanking transmission are often only discovered after the building has been constructed. At this point, efforts to meet the minimum requirements can be more costly and less efficient than if the building elements were properly specified in the design phase [20]. Very seldom has bad sound insulation been repaired. As a result, there are a large number of dwellings that do not comply with the minimum sound insulation requirements [5].

In conversations with acoustic consultants, many have confided that although they are aware of the problem of flanking transmission, the apparent sound reduction index is perceived as being too difficult to calculate and therefore flanking transmission is often ignored. This can lead to buildings that do not meet the building code or building elements which are overdesigned in an attempt to increase the apparent sound reduction index. Therefore, an accurate prediction method is needed that can be used to predict the apparent sound reduction index with confidence. Such a prediction method would not only be an aid in the assessment of new building designs, but it would also help the building industry to mitigate costs by
allowing less costly building elements which may not otherwise meet the minimum requirements for floor thickness, for example to be assessed.

1.4. Development of EN12354

In 1979 and 1986, Gerretsen of Nederlandse Organisatie voor Toegepast Natuurwetenschappelijk Onderzoek (TNO) in Delft published studies which described expressions for first order flanking paths in terms of the sound reduction of the flanking elements and the velocity attenuation at the junction between the building elements in the source and the receiver rooms [21, 22]. Gerretsen’s research became the basis for the European Committee for Standardization (CEN) standard, EN12354-1:2000: "Estimation of acoustic performance of buildings from the performance of products" [23, 24]. The EN12354 method offers a standard method of predicting the apparent sound reduction index between rooms inclusive of the flanking paths. Inputs for the method include the velocity attenuation at the junctions between the building elements which may be calculated according to the recently released ISO10848-1:2006 [25]. This standard is an integral part of the calculations according to EN12354-1 and is included in the description of the EN12354 method in this study.

In the derivation of the EN12354 method, a number of assumptions were made about the building elements and the junctions between them. The assumptions had the benefit of simplifying the calculations to result in a standard method of calculating the apparent sound reduction index. For the assumptions to be satisfied, both the source and receiving elements must be homogeneous, isotropic and moderately damped. Heavy cast-in-place as well as masonry constructions usually satisfy these conditions well [26]. For buildings with mainly heavy, monolithic structures, the EN12354 method has been described as quite useful [27].

1.5. Lightweight Building Constructions

There is great interest globally in also applying the EN12354 method to lightweight building constructions. These constructions constitute a large percentage of the single and double family houses built in North America, Australia, New Zealand, Japan and Europe [28].
However, lightweight constructions are considerably more difficult to model than the heavy monolithic structures and considerable effort will be required to adapt the EN12354 method to lightweight constructions [29].

One of the leaders in the investigation of flanking noise in lightweight constructions has been the team at the National Research Council of Canada Institute for Research in Construction (NRC-IRC). The team at the NRC which includes Nightingale and Quirt, has published a series of guides which are available on its website as well as journal articles regarding the application of the EN12354 method to lightweight constructions. Crispin of the Belgian Building Research Institute has published a number of studies in regard to the vibration reduction index which is an input for the EN12354 method. Rindel formerly of the Technical University of Denmark has published journal and congress articles regarding flanking noise in lightweight constructions. There are also a number of other studies being conducted worldwide which are published in journals such as Building Acoustics, the Journal of Sound and Vibration and Applied Acoustics as well as the proceedings of several international acoustics congresses.

For example, Pedersen [30] and Metzen [31] have compared the weighted apparent sound reduction index calculated according to EN12354-1 and ISO717-1 [32] to measured values. Both studies evaluated a large number of lightweight building constructions (the study by Pedersen evaluated over 200 constructions). Pederson reported that the EN12354 method overestimated the flanking sound reduction index in some cases and under predicted in others. The difference between the calculated and the theoretical values was as high as 10 dB in some cases, but the overall average was 0 to 0.5 dB with a standard deviation of 3.1 dB for walls and 3.2 dB for floors. The standard deviation for lightweight structures was higher than for monolithic structures which were also evaluated. Metzen reported that the EN12354 method under predicted the weighted apparent sound reduction index by 1.3 ± 1.1 dB.

However, neither study compared the calculated and the measured contributions from each flanking path. Furthermore, both of these studies only published the single number ratings. A separate study by Bradley [33] showed that comparing single number values for the
apparent sound reduction index can show good agreement even when the 1/3 octave band data shows significant disagreement. Furthermore, the study by Pedersen used theoretical values for the vibration reduction index from the annex of EN12354-1. Pedersen notes that applying the value from the annex to lightweight constructions was probably less accurate than for monolithic elements. Other studies have shown that the value of the vibration reduction index calculated from the annex does not match the behavior of the vibration reduction index measured in the laboratory [34, 35]. Pedersen also used the sound reduction index of the lightweight building elements which included the non-resonant component which EN12354-1 states should not be included in the calculations. Therefore, although both of these studies were extensive evaluations of the EN12354 method, the input data which was used and the use of single number ratings for the evaluation does not allow for a true assessment of the errors of applying the method to lightweight constructions. What is needed to assess the accuracy of the EN12354 method is a systematic evaluation of the contribution of each flanking path to the apparent sound reduction index.

1.6. Current Research

The goal of the current research was to systematically evaluate the application of the EN12354 method to lightweight building constructions. The evaluation included the calculation of the uncertainty of the estimate based on the uncertainty of the inputs such as the resonant sound reduction index and the vibration reduction index. Predictions using the EN12354 method were compared to predictions using Statistical Energy Analysis and laboratory and field measurements. Changes to the EN12354 method were proposed to improve the accuracy of the predictions.

The outcome of this research was to be an understanding of some of the possible errors of applying the EN12354 method to lightweight building constructions so that it can be either used with caution or with confidence.
2. Limitations to the Application of the EN12354 Method

2.1. Introduction

An advantage of the EN12354 method is that it is straightforward to apply as compared to a full Statistical Energy Analysis (SEA) model. Even a path by path analysis of a full SEA model requires the calculation of the loss and coupling factors whereas the EN12354 method always uses the same equations and input data from a library of test data be put into the model. However, the relative simplicity of the EN12354 method was achieved by making assumptions about the elements and junctions of the system being evaluated. The simplifying assumptions limit the wall and junction systems to which the EN12354 method can be accurately applied. A summary of a number of the limitations to the application of the EN12354 method is presented in this chapter.

2.2. Restriction to First Order Flanking Paths

To make the EN12354 method manageable, it was assumed that contributions from higher order paths (paths which involve more than one junction) were insignificant compared to the first order paths and of the direct paths. Therefore, the EN12354 method only considers the flanking transmission between two elements and the junction between them [26]. However, Craik [36] has written that while the individual higher order flanking paths may be insignificant compared to the direct transmission, the sum of the contributions of the higher order paths may be significant. Craik estimated that the errors could be on the order of 5 - 10 dB. In a separate study, Galbrun [37] also found errors due to the exclusion of higher order flanking paths, but notes that the estimate by Craik may not be as high for lightweight constructions.

For example, the exclusion of higher order paths requires that double-leaf elements such as those shown in Figure 2.1 be considered as homogeneous elements.
Limitations to the Application of the EN12354 Method

The EN12354 method can only include the transmission path: $1 \rightarrow 4 \rightarrow 6 \rightarrow 7$. Other paths such as $1 \rightarrow 2 \rightarrow 4 \rightarrow 6 \rightarrow 5$ cannot be included. Craik [38] found that modeling double-leaf walls as a single subsystem was possible at the low frequencies, but that the double-leaf wall should be modeled as a number of interconnected subsystems at the higher frequencies. The EN12354 method tries to reconcile this limitation by including the velocity level difference between leaf 4 and leaf 6 in the calculation of the flanking sound reduction index. However, the field testing presented in Chapter 9 showed that this method may be inadequate if leaf 4 is a heavier construction than leaf 2.

The exclusion of higher order flanking paths limits the application of the EN12354 method. For example, the EN13254 method can only be used to calculate the apparent sound reduction index between two adjacent rooms. Rooms that do not share a common element or junction would require the use of higher order flanking paths and a more complex SEA model would be required.

The exclusion of higher order paths also limits how problem elements, such as those with non-uniform energy density can be included in the analysis. It is possible to include a subsystem with non-uniform energy densities in a full SEA model by breaking up the subsystem into two or more subsystems with the appropriate coupling between them [26, 39]. However, the additional subsystems would require higher order flanking paths.
Nightingale investigated the possibility of including higher order flanking paths by using an effective vibration reduction index [40]. The study showed that it is not possible to simply sum the values of the vibration reduction indices along a longer flanking path. Furthermore, the use of an effective vibration reduction index was limited to elements which satisfy the requirements for a SEA plate subsystem.

### 2.3. Inclusion of Only Bending Waves

The transmission of structure-borne noise between building elements can include the contributions of bending, transverse and longitudinal waves [41]. A bending wave incident at a junction of wall surfaces will generate on the connecting surfaces quasi-longitudinal and transverse shear waves as well as other bending waves [26]. Furthermore, when a plane sound wave is incident on a panel at an oblique angle that is smaller than the limiting angle, the sound wave excites both longitudinal and shear waves in the panel [17]. For a full SEA model, the longitudinal and shear waves can be incorporated into the calculations by considering each wall as three subsystems, one for each wave type [42]. However, this method is not possible in the case of the EN12354 method since additional elements can not be added.

The EN12354 estimate of the flanking sound reduction index was derived under the assumption that only bending waves need to be considered in the description of structure-borne sound through the elements and the junctions. In-plane waves were assumed to be insignificant compared to bending waves and were ignored [26]. Gerretsen [26] notes that although due consideration must be made to the other wave types in calculating the power transmitted at a junction, the bending wave coupling itself is adequate to describe the resultant bending wave field on a connected plate for simple configurations. Furthermore, bending waves couple more effectively with room modes than other waves and therefore are primarily responsible for the radiation of sound [43].

Gibbs [43] notes that the bending wave coupling itself is adequate to describe the resultant bending wave field on a connected plate for simple configurations. Craik [42] writes that the additional waves make little difference close to the source. However, the in-plane waves are
most important at the higher frequencies where their wavelengths are more favorable in relation to the lateral dimensions of the walls [44]. Therefore, the omission of other waves can lead to large errors at the higher frequencies [36].

Craik showed that the addition of in-plane waves in a SEA model of a building made little difference close to the source, but large differences far from the source [42]. If an effective vibration reduction index can be applied to allow for higher order flanking paths, then ignoring the other wave types and the conversion of energy between them can lead to significant errors for flanking paths of many junctions [44].

2.4. Elements as Ideal SEA Subsystems

Several authors [36, 40] including Gerretsen [45] have equated the EN12354 method to a first order approximation of a SEA model. Therefore, the elements and transmission paths to which the EN12354 method is applied are subject to the same requirements as those of an SEA subsystem [24]. The requirements include [24, 46]:

- The vibration response is controlled by resonant modes of vibration.
- The frequency band under evaluation must contain resonant modes.
- The elements must support a uniform energy density (diffuse vibratory fields).
- The elements must be moderately damped.
- The elements must be weakly coupled.
- The forces which act on each element must be statistically independent.

These requirements have been shown to be acceptable for heavy, monolithic constructions such as concrete, but not necessarily applicable for lightweight constructions such as double-leaf timber framed walls [27, 30] as discussed in the following sections.
2.4.1. Resonant Modes of Vibration

The calculation of the EN12354 estimate of the flanking sound reduction index includes as inputs the sound reduction indices of the elements in the source and receiving room. In compliance with the requirement of a SEA subsystem that the vibration response is controlled only by resonant modes of vibration, the sound reduction index used in the calculations must only be the resonant component of the total sound reduction index. Although the non-resonant component of the sound reduction index may be larger than the resonant component below the critical frequency, it must not be included in the EN12354 calculations or else the EN12354 method may underestimate the value of the flanking sound reduction index [24].

The requirement that only the resonant component of the sound reduction index be used in the calculation of the flanking sound reduction index presents a significant difficulty in applying the EN12354 method to lightweight constructions which may have a critical frequency in or above the frequency range of interest. For these materials, the sound reduction index measured according to ISO140 [47] or ISO15816 [48] may include a significant contribution from the non-resonant component and therefore may not be used in the EN12354 calculations.

EN12354-1:2000 does not offer guidance for the calculation of the resonant component of the sound reduction index. Unless a consistently reliable method of determining the resonant component of the sound reduction index is possible, the EN12354 method should not be applied to elements with critical frequencies in or above the frequency range of interest. Several methods of determining the resonant component of the sound reduction index from measured data or from theoretical calculations have been proposed in the literature and these are compared in Chapter 7. The use of the calculated resonant component of the sound reduction index is also evaluated in the analysis of the laboratory and field measurements in Chapters 8 and 9.
2.4.2. Sufficient Modal Overlap

The fundamental element of the SEA model is a group of energy storage modes. These modes are usually of the same type (flexural, torsional, acoustical) that exist in the subsystem. The power transmitted between the subsystems depends on the difference in modal energy of the subsystems and the strength of the coupling between them. Energy storage in each subsystem is determined by the number of available modes for each subsystem in each frequency band \([49]\). If statistical energy analysis is to be used to predict structure-borne sound transmission, then it is necessary for the response of the structure to be controlled by resonant modes. This requirement can therefore put a limit on the frequency range over which SEA can be applied since there may be few modes present to store energy at the lower frequencies \([50]\).

The ratio of the number of modes \(N\) to the frequency band \(\Delta f\) is called the modal density \(n\) of the subsystem and is frequently used in SEA calculations \([49]\). Fahy \([51]\) also introduced the modal overlap factor \(M = \omega \eta - \eta(\omega)\) where \(\eta\) is the dissipation loss factor. Mace \([52]\) describes a coupling parameter which better describes the coupling strength between rectangular planes than \(M\). However, \(M\) remains a practical indicator of coupling strength due to the ease of calculating its value \([53]\). Hopkins \([53]\) notes that it is possible to estimate that SEA wave theory is appropriate for plate systems if \(M \geq 1\) and \(N \geq 5\). If these conditions are not met, Hopkins writes that it is still possible to apply SEA to the plate systems, but with errors of unknown magnitudes. Therefore, in order to apply the EN12354 method with confidence, the values of the modal overlap factor and the number of modes of the elements should be calculated in each 1/3 octave band to determine the validity of the calculations. However, EN12354-1 currently does not include limits on the frequency range of the estimations or guidance for calculating \(M\) or \(N\).
2.4.3. Uniform Energy Density and Moderate Damping

The ideal SEA subsystem and therefore the EN12354 element is required to support a uniform energy density. It is therefore also assumed that the surface velocity is uniform when the element is excited along the elements edge with the junction [26] as would be the case for flanking transmission in buildings. For these conditions to be satisfied, the elements are assumed to be homogeneous, isotropic and moderately damped [54]. Nightingale has reported that the vibration response of lightweight wood frame constructions is typical of a periodic plate/beam structure and therefore should not be considered as homogeneous and isotropic [54]. The vibration field in framed wooden structures can exhibit a strong gradient that will be different in directions parallel and perpendicular to the joints [55]. Therefore, these elements do not meet the basic requirements for simplified prediction models such as the EN12354 method [54]. Villot adds that since lightweight elements are often highly damped, the vibrational fields are no longer reverberant and existing standards often lose relevance [56].

It is possible to include a subsystem with non-uniform energy densities in a full SEA model by breaking up the subsystem into two or more subsystems with the appropriate coupling between them [26, 39]. However, the EN12354 method does not allow for an arbitrary number of subsystems due to the exclusion of higher order flanking paths.

The non-uniformity of the vibration response of lightweight panels is discussed in Chapter 4 and Chapter 5 where an alternative calculation of the velocity level difference is proposed.

2.4.4. Weak Coupling

Under the condition of weak coupling there is a power flow difference between two systems with the restriction that the external forces applied to the individual subsystems are statistically independent [57]. The assumption that the external forces are statistically independent is only ensured by “rain on the roof” excitation and not by a single point force or acoustic field excitation [58]. A related assumption which is central to SEA is that the modal vibrations of coupled sets are uncorrelated. This assumption is necessary to justify the linear dependence on modal energies of power flow expressions [57]. Craik [59] notes that
Limitations to the Application of the EN12354 Method

Fortunately in most real building structures the conditions of strong coupling will not often occur.

However, it is possible to have strong coupling in laboratory experiments where the subsystems are not connected to a number of other walls or floors as they would be in a real building. ISO10848-1, section 4.3.3 addresses the requirement of weak coupling during the measurement of the vibration reduction index in the laboratory by stating that the measurement of the velocity level difference between elements \( i \) and \( j \) is valid only if:

\[
\eta_{ji} \leq \frac{\eta_j}{2}
\]  

(2.1)

where \( \eta_{ji} \) is the coupling loss factor between element \( j \) and element \( i \) and \( \eta_j \) is the loss factor of element \( j \). If this condition is not met, then ISO10848 recommends adding damping material to the edges of the elements or connecting the elements to other structures. The additional damping affects the loss factor of the system under test, but in the case of heavy, monolithic structures, EN12354 makes use of the reverberation time measured in the laboratory and in the field to make the vibration reduction index invariant.

However, the requirement that the external forces are statistically independent by satisfying “rain on the roof” excitation is not addressed since ISO10848 states that using an electromagnetic shaker to excite the elements is the preferred excitation technique for the measurements.

2.5. Reciprocity

If the SEA and EN12354 equations are to hold for a given system then the coupling loss factors must satisfy the reciprocity relation \( \eta_{12} = \eta_{21} \) [52]. This relationship is also known as the consistency relationship. If reciprocity does not hold then the coupling loss factors can be negative and power can flow from a subsystem with low energy to one with high energy which is against the spirit of SEA [60]. Craik [61] found that many of the random errors due to low modal overlap or due to system simplifications cancel out when
reciprocity holds but that the standard deviation of the energy levels is higher when reciprocity does not hold.

In the derivation of the EN12354 method, Gerretsen assumed that the transmission of structure-borne sound through the system was independent of the transmission direction. The assumption had the benefit of canceling the often unknown radiation efficiency terms from the equations for the flanking transmission factor. However, Gerretsen [45] notes that in practice the flanking sound reduction index has been found to be different in each transmission direction.

If the system includes elements with critical frequencies in or above the frequency range of interest, then the response of the elements may include a significant non-resonant component. It is a SEA subsystem requirement that the response of the elements only be resonant. Since the non-resonant component should not be included in the calculation of the flanking sound reduction index its inclusion can cause the assumption of reciprocity to fail. Therefore, the assumption of reciprocity is only valid for heavy walls with sufficiently low critical frequency so that only resonant transmission is considered [15]. Furthermore, the assumption of reciprocity is only valid if the direct transmission path is the only transmission path. However, in an actual building there may be multiple flanking paths including higher order flanking paths and therefore assumptions based only on the direct transmission can be responsible for errors [37].

The requirement of reciprocity is discussed further in Chapter 6.

2.6. Gaussian Probability Density Functions

The EN12354 method appears to assume that all of the terms described by the method have Gaussian probability density functions (PDF’s). For example, the spatial average of the mean square velocity is calculated as the mean of the measurements. The best estimate for repeat observations from a Gaussian distribution is the mean of the measurements [62] which implies the mean square velocity was assumed to have a Gaussian PDF.
Limitations to the Application of the EN12354 Method

While the assumption that the time and spatially averaged mean square velocity has a Gaussian PDF may be acceptable for elements which have a low variance, observations of the response of lightweight elements may not be diffuse and therefore may not be described by a Gaussian distribution. The PDF’s of lightweight elements are discussed further in Chapter 3 and alternative calculations for the average velocity level based on the findings are described in Chapter 5.

2.7. Uncertainty

In general, no measurement is perfect and the imperfections give rise to errors in the results. Consequently, the result of a measurement or of a calculation based on measured data is only an estimate of the true value of the measurand and it is only complete when accompanied by a statement of the uncertainty of the estimate [63]. Currently, neither EN12354-1 [23] nor ISO10848-1 [25] give guidance for calculating the uncertainty of the terms the standards describe. Before the EN12354 method can be used with confidence, an estimate of the uncertainty of the prediction method is needed.

The uncertainty of the prediction method depends on several factors including the accuracy of applying the method to the structure in question, the type of elements and junctions involved in the construction, the effect of workmanship and the uncertainty of the input data [23]. The scope of this study is to determine the uncertainty of the predictions based on the uncertainty of the input data which has been propagated through the calculations. In addition to leading to an understanding of the accuracy of the predictions, an estimate of the propagated uncertainty will allow the sources of the propagated uncertainty to be determined. The uncertainty of the predictions is discussed further in Chapter 4.
3. Identification of Probability Density Functions

3.1. Motivation

The standards EN12354-1 [23] and ISO10848-1 [25] currently do not include estimates of the uncertainty of the prediction method they describe. If the uncertainty of the prediction method is to be understood, it is essential that the probability density functions (PDF’s) of the terms in the standards be determined. The method of calculating the uncertainty according to the ISO Guide to the Expression of Uncertainty in Measurement (GUM) [64] assumes that the PDF of the measurand is Gaussian (or in the case of finite effective degrees of freedom, by a scaled and shifted t distribution). If the PDF’s of the terms described by the EN12354 method can be described by a Gaussian distribution, GUM may be used to calculate the uncertainty. However, if the terms are described by PDF’s other than the Gaussian distribution, the method of propagating uncertainties according to GUM should not be used [65, 66]. The currently unreleased supplement to GUM [67] proposes the use of Monte Carlo simulations (MCS) in these cases [68]. MCS are able to provide much richer information about the measurand than the method of GUM by propagating the distributions of the inputs through the measurement model to calculate the distribution and the uncertainty of the output [69].

The PDF’s of the terms described by the EN12354 method were evaluated in this study using both experimental data and data calculated from MCS. The assessment of both the measured and the calculated terms was conducted using goodness-of-fit tests in each 1/3 octave bands between 100 Hz and 5000 Hz.

3.2. Distributions Assessed in this Study

The distributions considered in this study include the Gaussian, gamma, log-normal and exponential distributions. The choice of distributions was based on prior studies of the PDF’s of several of the terms of the EN12354 method.
Lyon [49] and Dejong [70] have described the PDF’s of squared terms such as the squared velocity or squared pressure as having a log-normal distribution. In the case of a term with a log-normal distribution, the logarithm of the term has a Gaussian distribution and therefore terms such as the velocity level or the sound pressure level are expected to have a Gaussian distribution. This agrees with a study by Ingalls [71] which described the PDF of the sound pressure level in a reverberation room as being Gaussian.

Alternatively, Bodlund [72] and Waterhouse [73] have described the PDF of the squared sound pressure in a reverberation room as a gamma distribution. In the same study, Bodlund described the reverberation time in a reverberation room also as a gamma distribution. The gamma distribution can have a wide variety of shapes based on the shape and scale parameters and therefore empirical data can often be fitted equally well by either the gamma or the log-normal distribution [74]. Lubman [75] and Ebeling [76] described the PDF of the mean sound pressure as being chi-squared, but this distribution is actually a special case of the gamma distribution [77] and therefore not considered separately in this study.

Huang and Radcliffe [78] predicted that the squared sound pressure would have a Gaussian distribution based on the Central Limit Theorem. Although this study did not include experimental data, the authors concluded that if enough data was measured, the distribution would tend towards a Gaussian distribution.

3.3. Goodness-of-Fit Tests

The PDF’s of the terms were assessed using goodness-of-fit tests which included the chi-square test ($\chi^2$), the Kolmogorov-Smirnov test [79] and the Anderson-Darling statistic [80, 81]. Although the $\chi^2$ test is a common means of assessing a PDF, the test has some disadvantages. The results of the $\chi^2$ test can be influenced by the bin size of the histograms used for the test, the signs of the deviations are not taken into account and each bin must include a minimum number of five samples [74]. The Anderson-Darling statistic does not have these disadvantages and was therefore used in this study to test for the Gaussian, exponential and log-normal distributions. However, the critical values required for the statistic are limited to a few distributions which do not include the gamma distribution.
Therefore, the Anderson-Darling statistic could not be used to assess the gamma distribution and either the $\chi^2$ test or the Kolmogorov-Smirnov test was used instead.

The Kolmogorov-Smirnov test is a better choice than the $\chi^2$ test for testing for the gamma distribution. The Kolmogorov-Smirnov test does not depend on the underlying cumulative distribution function being tested and it is an exact test that does not depend on an adequate sample size [79]. However, the Kolmogorov-Smirnov test does have the significant limitation that the scale and shape parameters must be fully specified and therefore can not be estimated from the data. Therefore, the use of the Kolmogorov-Smirnov test was limited to the verification of the distribution of data synthesized by the random number generator for the MCS since the shape and scale parameters were known in these cases. The $\chi^2$ test was used to evaluate the gamma distribution in all other cases, requiring a large number of observations of each measured term for the analysis.

To perform the $\chi^2$ test, a histogram with a set number of initial bins was generated for the observed values. The expected values for each bin were then calculated based on the distribution to be tested. To meet the requirement of a minimum of five expected values in each bin, bins with less than five expected values were combined with neighboring bins until all of the bins met the requirement. As part of the analysis, the initial number of bins was increased in steps prior to combining bins to increase the degrees of freedom of the $\chi^2$ test. The results of the $\chi^2$ test were accepted once the results of the $\chi^2$ test no longer changed when the number of initial bins was increased.

3.4. Analysis of Measured Data

The PDF’s of the measured terms for the EN12354 prediction method were assessed by making a large number of observations of the terms where a large number is defined as greater than twenty observations [66]. Histograms of the measured observations were generated and goodness-of-fit tests were performed in each 1/3 octave band based on the shape and the scale parameters which were estimated from the measured data. The PDF
which was assigned to the measured terms was that which best fit the data at the 95% confidence level.

The experiments used to obtain the measured data are described in the following sections.

3.4.1. Squared Velocity

The squared velocity was measured according to ISO10840-1 on a series of panels of different constructions and materials as described in Table 3.1 and in Appendix B.

<table>
<thead>
<tr>
<th>Construction</th>
<th>Material</th>
<th>Number of Panels Tested</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single</td>
<td>10mm gypsum board</td>
<td>2</td>
</tr>
<tr>
<td>Single</td>
<td>4mm MDF</td>
<td>1</td>
</tr>
<tr>
<td>Single</td>
<td>2mm steel</td>
<td>1</td>
</tr>
<tr>
<td>Single</td>
<td>2mm steel with a 3mm thick layer of Vibradamp</td>
<td>1</td>
</tr>
<tr>
<td>Double-Leaf</td>
<td>10mm gypsum board on metal studs</td>
<td>3</td>
</tr>
<tr>
<td>Double-Leaf</td>
<td>4mm MDF on wood studs</td>
<td>2</td>
</tr>
</tbody>
</table>

Table 3.1: Panels used to evaluate the PDF of the squared velocity.

The panels were clamped into a rigid frame for the testing and were excited with either a diffuse sound field or an electromagnetic shaker.

3.4.2. Structural Reverberation Time

Measurements of the structural reverberation time were made on the same panels and under the same conditions as the measurements of the squared velocity. The structural reverberation time of the panels was measured using an impact hammer and the integrated impulse response method with backward integration of the squared impulse response. The panel was impacted in fifteen locations and the surface velocity of the panel was simultaneously measured with four accelerometers. The locations of the impacts or of the accelerometers were changed for each measurement for a total of 60 different combinations of impact and measurement locations.
3.4.3. Squared Sound Pressure
Measurements of the squared pressure were made in fifty positions in the 217 m$^3$ reverberation room at the University of Canterbury. The reverberation room is equipped with randomly, hanging panels. The time of integration for the measurements was forty five seconds. The measurement positions were chosen at random so that the observations were mutually independent and avoided boundaries and the direct sound field of the dodecahedron loudspeaker.

In addition to the measurements made at the University of Canterbury, this study also included measurements of the squared pressure made in the reverberation rooms at other universities. John Davy provided data from the 199.9 m$^3$ reverberation room at the RMIT University in Melbourne, Australia and Colin Hansen provided data from the 179.9 m$^3$ reverberation room at the University of Adelaide in Adelaide, Australia. Data from the additional reverberation rooms was included to reduce the biases due to the room designs and to result in a more general description of the PDF’s.

3.4.4. Reverberation Time
The reverberation time in the reverberation room at the University of Canterbury was measured in thirty-two positions. The reverberation time was calculated using backward integration of the squared response.

3.5. Monte Carlo Simulations
3.5.1. Method
The design of the simulations begins with a mathematical model which describes the system to be analyzed. For this study, the mathematical models were the equations of EN12354-1 and ISO10840-1. For example, Figure 3.1 shows the propagation of PDF’s through the mathematical model in the calculation of the transmission factor of a panel.
Identification of Probability Density Functions

The PDF’s of the input terms for the mathematical model are shown on the left side of the figure. For each trial, a random value was drawn from the PDF’s of each of the input terms to calculate the output of the model. The process of drawing random values from the PDF’s of each of the inputs was repeated a number of times so that a distribution of possible output values is obtained from all of the trials.

Generally, the larger the number of trials synthesized for the MCS, the more precise will be the final answer. It is possible to estimate the number of trials required to achieve a desired level of accuracy for the output terms based on the goals of the analysis. If the goal is to synthesize values that emulate the mean or the variance of the measured data, the required number of tests can be estimated at a desired confidence level [82, 83]. Since the number of observations obtained from measurements will generally be insufficient for the MCS, additional observations must be synthesized by generating random numbers from the PDF of the measured inputs.

3.5.2. Generation of Synthesized Input Terms

Observations from the PDF of the terms were synthesized in MATLAB using a pseudo-random number generator as detailed in Appendix E. Once a set of observations was synthesized, statistical tests were performed to ensure that the synthesized distribution
adequately represented the PDF and the shape and scale parameters of the measured data. The tests included goodness-of-fit tests as well as comparisons of the shape and scale parameters of the synthesized and original data sets. If the goodness-of-fit test failed at the 95% confidence level or if the shape and scale parameters did not match the original data set within an allowable error of 0.5%, additional trials were synthesized and the data was again assessed. This procedure was repeated until the quality of the synthesized data was acceptable.

3.5.3. Output Terms
Once the MCS was complete, the PDF of the output term of the mathematical models was assessed in each 1/3 octave band using both the $\chi^2$ and the Anderson-Darling goodness-of-fit tests. The observations of the output term which had been generated by the MCS could then be used as an input for MCS for other terms.

The exceptions to this were the sound reduction index and the transmission factor which needed to have their shape and scale parameters adjusted to their use as an input for further simulations. The standard deviation of the sound reduction index and the transmission factor were adjusted to incorporate the standard deviation of reproducibility as specified in Annex A of ISO140-2 [84] as explained in Chapter 4. Therefore, once the PDF of the transmission factor and the sound reduction index were determined from MCS, the shape and scale parameters were changed to match the standard deviation of reproducibility as detailed in ISO140-2. The modified, synthesized data was then used for subsequent MCS which included the sound reduction index or transmission factor as inputs to the model.

3.6. Results
The PDF’s of the terms of the EN12354 prediction method are described in Table 3.2. The table includes the number of the standard and the equation where the term is defined, if applicable.
<table>
<thead>
<tr>
<th>Term</th>
<th>Symbol</th>
<th>Unit</th>
<th>PDF</th>
<th>Standard</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Squared velocity on a panel</td>
<td>$\nu^2$</td>
<td>(m/s)$^2$</td>
<td>Log-normal</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Velocity level on a panel</td>
<td>$L_v$</td>
<td>dB</td>
<td>Gaussian</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Structural reverberation time of a panel</td>
<td>$T_s$</td>
<td>s</td>
<td>Log-normal</td>
<td>EN12354-1:2000</td>
<td>-</td>
</tr>
<tr>
<td>Total loss factor of a panel</td>
<td>$\eta_{\text{panel}}$</td>
<td>-</td>
<td>Log-normal</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Squared sound pressure in a reverberation room</td>
<td>$p^2$</td>
<td>Pa$^2$</td>
<td>Gamma</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Sound pressure level in a reverberation room</td>
<td>$l_p$</td>
<td>dB</td>
<td>Log Gamma</td>
<td>(Gaussian)</td>
<td>-</td>
</tr>
<tr>
<td>Reverberation time in a reverberisation room</td>
<td>$T_R$</td>
<td>s</td>
<td>Gamma</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Loss factor in a reverberation room</td>
<td>$\eta_{\text{room}}$</td>
<td>-</td>
<td>Gamma</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Sound power in a reverberation room</td>
<td>$P_a$</td>
<td>W</td>
<td>Gamma</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Sound power level in a reverberation room</td>
<td>$l_{w}$</td>
<td>dB</td>
<td>Log Gamma</td>
<td>(Gaussian)</td>
<td>ISO3741:1988</td>
</tr>
<tr>
<td>Transmission factor</td>
<td>$\tau$</td>
<td>-</td>
<td>$F$</td>
<td>(Log-normal)</td>
<td>-</td>
</tr>
<tr>
<td>Sound reduction index</td>
<td>$R$</td>
<td>dB</td>
<td>Log F</td>
<td>(Gaussian)</td>
<td>ISO140-3:1995</td>
</tr>
<tr>
<td>Radiation efficiency</td>
<td>$\sigma$</td>
<td>-</td>
<td>Log-normal</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Vibration transmission factor</td>
<td>$d_{ij}$</td>
<td>-</td>
<td>Log-normal</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Velocity level difference</td>
<td>$D_{vij}$</td>
<td>dB</td>
<td>Gaussian</td>
<td>ISO10848:2006</td>
<td>8</td>
</tr>
<tr>
<td>Direction averaged velocity level difference</td>
<td>$\overline{D_{vij}}$</td>
<td>dB</td>
<td>Gaussian</td>
<td>EN12354-1:2000</td>
<td>12</td>
</tr>
<tr>
<td>Equivalent absorption length</td>
<td>$a$</td>
<td>m</td>
<td>Log-normal</td>
<td>EN12354-1:2000</td>
<td>11</td>
</tr>
<tr>
<td>Vibration reduction index</td>
<td>$K_{ij}$</td>
<td>dB</td>
<td>Gaussian</td>
<td>EN12354-1:2000</td>
<td>10</td>
</tr>
<tr>
<td>Flanking transmission factor</td>
<td>$\tau_{ij}$</td>
<td>-</td>
<td>Log-normal</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>EN12354 estimate of the flanking transmission factor</td>
<td>$\tau_{ij,\text{EN12354}}$</td>
<td>-</td>
<td>Log-normal</td>
<td>EN12354-1:2000</td>
<td>13</td>
</tr>
<tr>
<td>EN12354 estimate of the flanking sound reduction index</td>
<td>$\overline{R_{ij,\text{EN12354}}}$</td>
<td>Gaussian</td>
<td>EN12354-1:2000</td>
<td>13</td>
<td></td>
</tr>
<tr>
<td>Apparent sound reduction index</td>
<td>$R'$</td>
<td>dB</td>
<td>Gaussian</td>
<td>EN12354-1:2000</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 3.2: Summary of the PDF’s of the terms defined in EN12354 and ISO10848. Also noted are terms defined in ISO3741:1988 [85] and ISO140-3:1995. Distributions in parenthesis are approximations which are valid under certain conditions.
The flanking transmission factor $\tau_{ij}$ shown in Table 3.2 is defined as [22]:

$$\tau_{ij} = \frac{\tau_i d_{ij} \sigma_j S_j}{\sigma_i S_o}$$  \hspace{1cm} (3.1)

where $\tau_i$ is the in situ resonant transmission factor of element $i$, $d_{ij}$ is the in situ vibration transmission factor between elements $i$ and $j$, $S_j$ is the area of element $j$, $S_o$ is a reference area and $\sigma_i$ and $\sigma_j$ are the in situ resonant radiation efficiencies of elements $i$ and $j$, respectively.

The EN12354 estimate of the flanking transmission factor between elements $i$ and $j$ makes use of reciprocity between the calculated flanking transmission factor terms in each direction to exclude the radiation efficiency terms from the calculations such that [86]:

$$\tau_{ij,EN12354} = \sqrt{\frac{\tau_i d_{ij} \sigma_j S_j}{S_o^2}}$$  \hspace{1cm} (3.2)

where $\tau_{ij,EN12354}$ is the estimation of the measurand $\hat{\tau}_{ij}$.

The table shows that several of the PDF’s of terms based on measurements in the reverberation rooms may be approximated by other distributions. The approximations include the Gaussian distribution for the log gamma distribution, the log-normal distribution for the F-distribution and the Gaussian for the log F-distribution. The validity of these approximations is discussed in the following sections.

### 3.7. Discussion of PDF Results

#### 3.7.1. Squared Velocity

The squared velocity was found to have a log-normal PDF. For example, the squared velocity was measured at 30 positions on the surface of a 1.6 mm thick steel panel. A histogram of the measured data is compared to the expected values from the Gaussian, log-normal and gamma distributions in Figure 3.2.
The figure shows and the goodness-of-fit tests confirmed that the PDF of the data was log-normal at the 95% confidence level.

EN12354 defines the average velocity level $\bar{L}_v$ of an element is to be:

$$\bar{L}_v = 10 \log_{10} \left( \frac{\sum_{m=1}^{n} v_m^2}{nv_0^2} \right)$$  \hspace{1cm} (3.3)

where $v_m^2$ is the squared velocity at position $m$, $v_o$ is the reference velocity and $n$ is the number of measurement positions. Eqn (3.3) calculates the best estimate of the spatial average of the squared velocity based on a Gaussian PDF since the best estimate for repeat observations from a Gaussian distribution is the mean of the measurements [62].

The best estimate from a log-normal distribution may not be the same as that of a Gaussian distribution. However, the assumption of a Gaussian distribution may be valid if:

$$C = \frac{\sigma}{v} < \frac{1}{3}$$  \hspace{1cm} (3.4)
where $C$ is the coefficient of variation and $s$ and $\bar{v^2}$ are the sample standard deviation and the mean of the squared velocity, respectively [87].

The results of this PDF study have resulted in proposed estimates of the spatial average of the squared velocity, the average velocity level and the direction averaged velocity level difference. The proposed estimates are presented in Chapter 5 as part of a proposed revision of ISO10840-1:2006.

3.7.2. Sound Pressure and Sound Pressure Level in a Reverberation Room

A summary of the data measured in the reverberation rooms is shown in Table 3.3.

<table>
<thead>
<tr>
<th>Measurement Range: 1/3 Octave Bands between</th>
<th>Distribution</th>
<th>Number of Fits in the Measurement Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 Hz and 5k Hz (18 bands)</td>
<td>Gaussian</td>
<td>Room A: 15</td>
</tr>
<tr>
<td></td>
<td>Log-normal</td>
<td>Room A: 16</td>
</tr>
<tr>
<td></td>
<td>Gamma</td>
<td>Room A: 15</td>
</tr>
<tr>
<td></td>
<td>No Fit</td>
<td>Room A: 2</td>
</tr>
<tr>
<td>50 Hz and 100 Hz (3 bands)</td>
<td>Gaussian</td>
<td>Room A: 2</td>
</tr>
<tr>
<td></td>
<td>Log-normal</td>
<td>Room A: 2</td>
</tr>
<tr>
<td></td>
<td>Gamma</td>
<td>Room A: 2</td>
</tr>
<tr>
<td></td>
<td>No Fit</td>
<td>Room A: 1</td>
</tr>
<tr>
<td>6.3k Hz and 10k Hz (3 bands)</td>
<td>Gaussian</td>
<td>Room A: 1</td>
</tr>
<tr>
<td></td>
<td>Log-normal</td>
<td>Room A: 1</td>
</tr>
<tr>
<td></td>
<td>Gamma</td>
<td>Room A: 2</td>
</tr>
<tr>
<td></td>
<td>No Fit</td>
<td>Room A: 1</td>
</tr>
</tbody>
</table>

Table 3.3: Summary of the $\chi^2$ test results for the reverberation rooms. The table shows the number of 1/3 octave bands in the measurement range for which the hypothesis that the distribution fits the data was accepted. *The 1/3 octave bands between 6.3k Hz and 10k Hz were not measured in Room C.

The table shows that the PDF of the squared sound pressure measured in the reverberation rooms evaluated in this study was described by both the log-normal and the gamma distribution in the 1/3 octave bands between 100 Hz and 5000 Hz. This finding agrees with the prior studies by both Lyon and Dejong as well as the study by Bodlund and Waterhouse. The PDF could also be described as Gaussian in most of the 1/3 octave bands. Examples of
the histograms of the squared sound pressure as generated for the $\chi^2$ test are shown in Figure 3.3.

![Figure 3.3: Comparison between the histograms of fifty observations of the squared pressure in the 125 Hz 1/3 octave band and the expected values from the Gaussian, log-normal and gamma distributions.](image)

The figure shows the observed and the expected values for the Gaussian, the log-normal and the gamma distributions. The results of the $\chi^2$ test showed that the probability that the observed data came from the distributions was 68%, 30% and 42%, respectively in the 125 Hz 1/3 octave band.

It has been noted that the log-normal distribution may be approximated as a Gaussian distribution if the coefficient of variation is less than 1/3. Likewise, the gamma distribution may be approximated by the Gaussian distribution as the shape parameter of the gamma distribution $\alpha \to \infty$. The shape parameter may be estimated from the mean $\bar{p}^2$ and the sample standard deviation $s$ of the observations such that [79]:

$$\hat{\alpha} = \left(\frac{\bar{p}^2}{s}\right)^2 \tag{3.5}$$

where $\hat{\alpha}$ is an estimate of the shape parameter. When the shape parameter $\alpha = 1$ the Gaussian distribution is a poor approximation of the gamma distribution as shown in Figure 3.4.
Figure 3.4: Comparison between the distribution function of the Gaussian and the gamma distributions for increasing values of the shape function.

For small values of $\alpha$, the gamma distribution is not symmetric leading to errors in the approximation by the Gaussian distribution. However, due to the Central Limit Theorem, as the value of the shape parameter increases, the gamma distribution becomes more symmetric and is therefore better approximated by the Gaussian distribution so that the error in the approximation decreases as shown in Figure 3.5 [88].

Figure 3.5: Absolute error in approximating the gamma distribution by the Gaussian distribution.
The figure shows the absolute error of the approximation changing quickly for small values of $\alpha$, but then changing little as $\alpha \rightarrow \infty$. Das [89] estimated that the Gaussian approximation to the gamma distribution is considered to be satisfactory when $\alpha > 15$.

An example of the $\chi^2$ test results from reverberation room B between the 50 Hz and the 125 Hz 1/3 octave bands is shown in Table 3.4.

<table>
<thead>
<tr>
<th>1/3 Octave Band (Hz)</th>
<th>$\chi^2$ Goodness-of Fit Test Results - Reverberation Room B Squared Sound Pressure $p^2$ (Pa$^2$)</th>
<th>Coefficient of Variation $C$</th>
<th>Shape Parameter $\alpha$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Gaussian</td>
<td>Log-Normal</td>
<td>Gamma</td>
</tr>
<tr>
<td></td>
<td>Result</td>
<td>Probability</td>
<td>Result</td>
</tr>
<tr>
<td>50</td>
<td>Reject</td>
<td>0.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>63</td>
<td>Reject</td>
<td>0.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>80</td>
<td>Reject</td>
<td>0.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>100</td>
<td>Reject</td>
<td>1.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>125</td>
<td>Accept</td>
<td>64.3%</td>
<td>Accept</td>
</tr>
</tbody>
</table>

Table 3.4: Summary of the results of the $\chi^2$ goodness-of-fit test of the squared pressure in reverberation room B between the 50 Hz and the 125 Hz 1/3 octave bands.

The data in the table shows that between the 50 Hz and the 100 Hz 1/3 octave bands the hypothesis that the data came from a log-normal or from a gamma distribution was accepted but that the hypothesis that the data came from a Gaussian distribution was rejected. The coefficient of variation was less than 1/3 in most of the bands so the approximation of a Gaussian distribution would be expected to be accepted if the distribution was log-normal. The shape parameter had a value that was less than fifteen in the 50 Hz and 63 Hz 1/3 octave bands. In the 80 Hz 1/3 octave band, the shape parameter had a value of twenty five, but the probability that the distribution was gamma was only 5.9%. The data in other 1/3 octave bands and from other reverberation rooms showed similar results in cases where the shape parameter was larger than fifteen, but the probability of the distribution being gamma was low. It is hypothesized that the low probability that the distribution was gamma resulted in the low probability that the distribution could be approximated as Gaussian even though the shape parameter was greater than fifteen.
The results of the goodness-of-fit tests on the sound pressure level measured in the reverberation rooms showed that the measurement data could be described as Gaussian, log-normal, gamma and log-gamma in all of the 1/3 octave bands to which the distributions could be fit. An example of the results is shown in Table 3.5.

<table>
<thead>
<tr>
<th>1/3 Octave Band (Hz)</th>
<th>( \chi^2 ) Goodness-of-Fit Test Results - Reverberation Room B Sound Pressure Level ( L_p ) (dB)</th>
<th>Coefficient of Variation ( C )</th>
<th>Shape Parameter ( \alpha )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Gaussian</td>
<td>Log-Normal</td>
<td>Gamma</td>
</tr>
<tr>
<td>50</td>
<td>Accept</td>
<td>37.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>63</td>
<td>Accept</td>
<td>31.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>80</td>
<td>Accept</td>
<td>5.1%</td>
<td>Accept</td>
</tr>
<tr>
<td>100</td>
<td>Accept</td>
<td>50.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>125</td>
<td>Accept</td>
<td>74.3%</td>
<td>Accept</td>
</tr>
</tbody>
</table>

Table 3.5: Summary of the results of the \( \chi^2 \) goodness-of-fit test of the sound pressure level in reverberation room B between the 50 Hz and the 125 Hz 1/3 octave bands.

The data in the table shows that the coefficient of variation and the shape parameter were such that the log-normal or the gamma distribution could be approximated as a Gaussian distribution in every 1/3 octave band.

Following the study by DiCiccio [90], if \( T \) has a gamma distribution then \( Z = \ln T \) is described by the log gamma distribution. As the shape parameter \( \alpha \rightarrow \infty \), the log gamma distribution tends to the Gaussian distribution, whereas the distribution of \( T \) tends to the log-normal distribution. Prentice [91] also noted that the log-normal is a limiting case of the gamma distribution when \( \alpha \rightarrow \infty \). The tendencies of the gamma and log gamma distributions when \( \alpha \rightarrow \infty \) could explain the acceptance of both the log-normal and the gamma distributions for the PDF of the squared pressure and all three of the distributions for the PDF of the sound pressure level.

Therefore, based on these approximations, the results of the goodness-of-fit tests and theory [76], it is concluded that the PDF of the squared pressure is a gamma distribution and the PDF of the sound pressure level is a log gamma distribution.
Following from eqn (3.5), if the sample standard deviation of the squared pressure is such that:

\[ s_{p^2} \leq \frac{\bar{p}^2}{\sqrt{15}} \]  

(3.6)

then the sound pressure level can be approximated as a Gaussian distribution. The distribution of the squared pressure in the three reverberant chambers included in this study could be approximated as a Gaussian distribution in the 200 Hz 1/3 octave band and above. Therefore, in these 1/3 octave bands, the average sound pressure level in the reverberation room could be calculated according to IOS10848-1:2000 such that:

\[ \bar{L}_p = 10 \log_{10} \left( \frac{\sum_{m=1}^{n} p_{m}^2}{np_o^2} \right) \]  

(3.7)

where \( p_m^2 \) is the squared pressure at position \( m \) and \( p_o \) is the reference pressure and the best estimate of the mean is that for a Gaussian distribution. However, at the low frequencies where the sample standard deviation may be larger than the limit in eqn (3.6), the best estimate of the distribution may be different than that of a Gaussian distribution.

3.7.3. Sound Power in a Reverberation Room

The sound power synthesized from MCS was found to fit both the gamma and the log-normal distributions. The sound power level was found to fit the Gaussian, log-normal and the gamma distributions. As with the squared sound pressure and the sound pressure level, the sound power was concluded to have a gamma PDF and the sound power level was concluded to have a log gamma PDF. In most cases, the PDF of the sound power level measured in reverberation room may be approximated as a Gaussian distribution.
### 3.7.4. Sound Reduction Index and Transmission Factor

The transmission factor as calculated according to ISO140-3 is written:

\[
\tau = \frac{\langle p_1^2 \rangle A_2}{\langle p_1^2 \rangle} \frac{A_2}{S}
\]  

(3.8)

where \( A_2 \) is the equivalent absorption area of the receiver room, \( S \) is the area of the partition and \( \langle p_1^2 \rangle \) and \( \langle p_2^2 \rangle \) have gamma PDF’s. If \( T_1 \) and \( T_2 \) are mutually independent and \( T_i \) has a gamma distribution with shape parameter \( \alpha_i \) and scale parameter \( \beta_i \), then the PDF of \( T_1/T_2 \) is described by Johnson [92] to be the F-distribution:

\[
\frac{\alpha_1 \beta_1}{\alpha_2 \beta_2} F_{2\alpha_1,2\alpha_2}
\]  

(3.9)

Therefore, it is concluded that the transmission factor which is calculated by dividing two gamma distributions as shown in eqn (3.8) has a F-distribution as predicted by Bodlund [72].

If \( W \) is F-distributed according to eqn (3.9) and if the shape parameters \( \alpha_1 \) and \( \alpha_2 \) are large, then \( Z = \ln W \) may be approximated by the Gaussian distribution [92]. Therefore, the PDF of the sound reduction index is concluded to be a log-F distribution which may be approximated as a Gaussian distribution if the shape parameters of the squared pressure, \( \alpha_1 \) and \( \alpha_2 \) are each greater than 7.5.

A random variable whose logarithm \( Y = \ln X \) follows the Gaussian distribution may be modeled as a log-normal distribution [74]. Therefore, the PDF of the transmission factor is a F-distribution which may be approximated as the log-normal distribution. Examples of the histograms of the transmission factor obtained from MCS are compared versus the expected values of the Gaussian, log-normal and gamma distributions in Figure 3.6.
Identification of Probability Density Functions

Figure 3.6: Comparison of histograms of 100,000 synthesized observations of the transmission factor versus the expected values from the Gaussian, log-normal and gamma distributions in the 100, 125, 1600 and 5000 Hz 1/3 octave bands.

The figures show that the results of the MCS agree with the approximation of the PDF of the transmission factor as a log-normal distribution.

3.7.5. Reverberation Time

The analysis of the measurements of the reverberation time measured in the reverberation room showed that the PDF could be described by both the gamma and the log-normal distributions in most of the 1/3 octave bands. The PDF could also be described by the Gaussian distribution in many of the 1/3 octave bands. For example, histograms of the measured reverberation time and the expected values are shown in Figure 3.7.
The results of the $\chi^2$ test showed that the probability that the observed data came from distributions was 32%, 45% and 32%, respectively. The Anderson Darling statistic also showed the PDF to be Gaussian or log-normal in the 125 Hz 1/3 octave band.

The gamma distribution may be approximated as the log-normal or the Gaussian distributions for large values of the shape parameter and the measured data agrees with this approximation. It is concluded that the PDF of the reverberation time has a gamma distribution.

The PDF of the structural reverberation time was described as a log-normal distribution. However, in agreement with the approximation of the log-normal distribution as a Gaussian distribution if the coefficient of variation is less than 1/3, the measured data shows that the PDF could also be described by the Gaussian distribution in many of the 1/3 octave bands.

### 3.7.6. Flanking Transmission Factor

The results of MCS showed that the PDF of both the flanking transmission factor $\tau_{ij}$ and the EN12354 estimate of the flanking transmission factor $\tau_{ij,EN12354}$ have a log-normal distribution as shown in Figure 3.8 and Figure 3.9, respectively.
Identification of Probability Density Functions

Figure 3.8: Comparison of histograms of 100,000 synthesized observations of $\tau_{ij}$ versus the expected values from the Gaussian, log-normal and gamma distributions in the 100 and 200 Hz 1/3 octave bands.

Figure 3.9: Comparison of histograms of 100,000 synthesized observations of $\tau_{ij,EN12354}$ versus the expected values from the Gaussian, log-normal and gamma distributions in the 100 and 200 Hz 1/3 octave bands.

An alternative to the EN12354 estimate of the flanking transmission factor is proposed in Chapter 6 that is based on the best estimate of a log-normal PDF. The proposed estimate differs from the EN12354 estimate in that it allows for the directional dependence of the flanking transmission factor which may be exhibited by lightweight building constructions with critical frequencies below the frequency range of interest [86]. However, the application of the proposed estimate is limited due to the inclusion of the radiation efficiency terms.
3.7.7. Uncertainty Analysis

The EN12354 estimate of the flanking sound reduction index is defined as [23]:

\[
R_{ij,\text{EN12354}} = \frac{R_{R,\text{i,situ}} R_{R,\text{j,situ}} }{2} + D_{v,ij,\text{situ}} + 10 \log \frac{S_i}{S_j} \tag{3.10}
\]

where \( R_{R,\text{i,situ}} \) is the \textit{in situ} resonant component of the sound reduction index of element \( i \).

Each of the terms in eqn (3.10) is shown in Table 3.2 and Figure 3.10 to be approximated by a Gaussian PDF.

![Figure 3.10](image_url)

\textbf{Figure 3.10:} Histograms of the EN12354 estimate of the flanking transmission factor \( R_{ij,\text{EN12354}} \), the sound reduction index \( R \) and the direction averaged velocity level difference \( D_{v,ij} \). A comparison of the histograms to the expected values of the Gaussian, log-normal and the gamma distributions showed that the PDF’s of the terms were best approximated by a Gaussian distribution.

Therefore, the method of GUM may be used to calculate the uncertainty of the flanking sound reduction index.
However, the last step of the EN12354 prediction method is to sum the contributions of all of the transmission paths to determine the apparent transmission factor $\tau'$ such that:

$$\tau' = \tau_d + \sum_{f=1}^{n} \tau_f$$  \hspace{1cm} (3.11)

where $\tau_d$ is the contribution from the separating element, $\sum_{f=1}^{n} \tau_f$ represents the contributions of $n$ first order flanking paths and it is assumed that there are no contributions from indirect or direct air-borne transmission. The apparent sound reduction index $R'$ is then calculated from:

$$R' = -10 \log \tau'$$  \hspace{1cm} (3.12)

Since the transmission factor and the flanking transmission factor were found to be approximated as a log-normal PDF, the method of GUM can not be applied to the calculation of the uncertainty of the apparent transmission factor or the apparent sound reduction index. Therefore it may be necessary to use MCS to calculate the uncertainty of these terms. MCS should also be used to calculate the uncertainty of the other linear terms such as the spatial average of the mean square velocity and the radiation efficiency since none of the linear terms had a Gaussian PDF.

3.8. Discussion of the PDF Study Results

Experimental data and the results of MCS have been used to describe the probability density functions of the terms for the prediction of the apparent sound reduction index according to EN12354-1. The knowledge of the PDF’s is an essential step in the determination of the uncertainty of the predictions as well as the expanded uncertainty. Based on the results of this study, the method of GUM may be used to determine the uncertainty of the flanking sound reduction index, but the use of MCS will be required to determine the uncertainty of the apparent sound reduction index.
4. Uncertainty of the EN12354 Method

4.1. Introduction

In general, no measurement is perfect and the imperfections give rise to errors in the results. Consequently, the result of a measurement or of a calculation based on measured data is only an estimate of the true value of the measurand and it is only complete when accompanied by a statement of the uncertainty of the estimate [63]. Currently, neither EN12354-1 [23] nor ISO10848-1 [25] give guidance for calculating the uncertainty of the terms the standards describe. Before the EN12354 method can be used with confidence, an estimate of the uncertainty of the prediction method is needed.

The uncertainty of the prediction method depends on several factors including the accuracy of applying the method to the structure in question, the type of elements and junctions involved in the construction, the effect of workmanship and the uncertainty of the input data [23]. The scope of this study is to determine the uncertainty of the predictions based on the uncertainty of the input data which has been propagated through the calculations. In addition to leading to an understanding of the accuracy of the predictions, an estimate of the propagated uncertainty will allow the sources of the propagated uncertainty to be determined.

4.2. Method

4.2.1. Method of GUM

The ISO Guide 98 Part 3, The Guide to the Expression of Uncertainty in Measurement (GUM) [64] has become the de facto standard for the evaluation of measurement uncertainty in metrology [93] and will serve as the guideline for calculating the propagated uncertainty in this study. The method of GUM assumes that the probability density function (PDF) of the measurand is Gaussian. If the PDF can not be approximated as Gaussian, the uncertainty is to be determined using Monte Carlo simulations (MCS) according to ISO Guide 98-3/Supplement 1 [65-67].
Although EN12354-1 and ISO10848-1 appear to assume that all of the terms have Gaussian PDF’s, it was shown in Chapter 3 that many of the terms described by the EN12354 method have PDF’s other than Gaussian and therefore the method of GUM should not be used to calculate the uncertainty of these terms. In this study, terms which have a log-normal PDF are noted and are treated accordingly in the derivation of the uncertainty equations. However, the uncertainty of these terms is treated generically so that it can be treated as either a log-normal PDF or as an approximately Gaussian PDF in the uncertainty analysis.

A summary of the GUM method and the assumptions that were made in the calculation of the uncertainty in this study are presented in Appendix F.

4.2.2. Monte Carlo Simulations

Monte Carlo simulations as described in ISO Guide 98-3/Supplement 1 are able to provide much richer information about the measurand than the method of GUM by propagating the PDF’s of the inputs through the measurement model to calculate the distribution and the uncertainty of the output [69]. The simulations require a mathematical model which describes the system to be analyzed. For this study, the mathematical models were the equations of EN12354-1 and ISO10840-1. Trials are generated by drawing a random value from the PDF’s of each of the input terms to calculate the output of the model. The process of drawing random values from the PDF’s of each of the inputs is repeated a number of times so that a distribution of possible output values is obtained from all of the trials. The supplement also describes the validation of uncertainty calculations according to the GUM method by comparing the results to those from MCS [67].

The simulations performed for this study incorporated in part a MATLAB code developed by the Group of Structural Integrity of the University of Burgos [94, 95] for adaptive Monte Carlo simulations in accordance with the guidelines of ISO Guide 98-3/Supplement 1.
4.2.3. Propagation of the Uncertainty of Terms with Log-Normal PDF’s
Since the GUM method can only be applied if the measurand has a Gaussian PDF, terms with log-normal PDF’s can not be included in the uncertainty analysis. However, the EN12354 method describes many of the terms with log-normal PDF’s such as the mean square velocity in terms of logarithmic quantities such as the average velocity level. The use of logarithmic terms is advantageous to the uncertainty analysis since the logarithmic terms have Gaussian PDF’s and therefore the uncertainty of these terms can be evaluated using the GUM method.

If $X$ has a two parameter log-normal distribution, then $Q = 10 \log(X)$ has a Gaussian distribution. The best estimate of $Q$ is:

$$ Q = \frac{10}{\ln(10)} \left[ \mu_Y + \frac{s_Y^2}{2} \right] $$  \hspace{1cm} (4.1)

where the mean $\mu_Y$ and sample variance $s_Y^2$ are [74]:

$$ \mu_Y = \frac{\sum_{i=1}^{n} \ln(X_i)}{n} $$  \hspace{1cm} (4.2)

$$ s_Y^2 = \frac{\sum_{i=1}^{n} [\ln(X_i) - \mu_Y]^2}{n-1} $$  \hspace{1cm} (4.3)

The uncertainty of $Q$ is:

$$ u^2(Q) = \left[ \frac{10}{\ln(10)} \right]^2 \left[ u^2(\mu_Y) + u^2(s_Y) s_Y^2 \right] $$  \hspace{1cm} (4.4)

The uncertainty of the standard deviation $s_Y$ is estimated to be [64, 96]:

$$ u^2(s_Y) \approx \frac{s_Y^2}{2(n-1)} $$  \hspace{1cm} (4.5)
where \( n - 1 \) are the degrees of freedom of \( s_y \). The term \( u^2(s_y) \) is the squared uncertainty of the sample standard deviation of the repeated observations of the measurand. In the derivation of the term, it was assumed that the unbiased estimate of the variance of the sample \( s^2_y \) is a satisfactory approximation of the variance of the population \( \sigma^2_y \). This approximation may be acceptable for very large sample sizes, but can lead to non-trivial uncertainty of the statistically estimated standard deviation for practical sample sizes [64]. For example, the uncertainty of the sample standard deviation of ten independent observations is 24%. Therefore, very small sample sizes will have a large uncertainty associated with the calculation of the sample standard deviation.

Substituting eqn (4.5) into eqn (4.4) yields:

\[
\begin{align*}
   u^2(Q) & \approx \left[ \frac{10}{\ln(10)} \right]^2 \left[ \frac{s^2_y}{n} + \frac{s^2_y}{2(n-1)} \right] \\
   & \quad (4.6)
\end{align*}
\]

which shows the uncertainty of \( Q \) is dependent on the standard deviation of the Gaussian distribution and the number of observations. The estimate shown in eqn (4.6) has been validated using MCS according to ISO Guide 98-3/Supplement 1.

### 4.3. Calculation of the Uncertainty

EN12354-1 estimates the flanking sound reduction index between elements \( i \) and \( j \) to be:

\[
R_{ij,EN12354} = \frac{R_{R,i,situ} + R_{R,j,situ}}{2} + \overline{D_{v,i,j,situ}} + 10 \log \left( \frac{S_o}{\sqrt{S_i S_j}} \right) \\
\quad (4.7)
\]

where \( R_{R,i,situ} \) and \( R_{R,j,situ} \) are the in-situ resonant sound reduction index of elements \( i \) and \( j \), respectively, \( \overline{D_{v,i,j,situ}} \) is the in-situ, direction averaged velocity level difference between the elements, \( S_o \) is a reference area (typically the common element between the rooms) and \( S_i \) and \( S_j \) are the areas of elements \( i \) and \( j \), respectively.
The propagated uncertainty of each of the terms in eqn (4.7) are derived in the following sections before deriving the total, propagated uncertainty of the flanking sound reduction index.

4.3.1. Uncertainty of the In-Situ Resonant Sound Reduction Index

Only the resonant component of the sound reduction index may be used in the EN12354 prediction method [97]. In Chapter 7, it is shown that methods of separating the resonant component from the sound reduction index measured according to ISO140-3:1995 [47] or ISO15186-1:2000 [48] are unreliable and therefore calculating the resonant component theoretically is preferable. However, the uncertainty of the theoretical predictions is dependent upon the uncertainty of the inputs values which may not be known and upon the bias from the true value of the measurand which can not be known.

Alternatively, GUM allows for the estimation of the Type B uncertainty from previous measurement data, experience or general knowledge. If it is assumed that the theoretical predictions of the resonant components of the sound reduction index have similar uncertainty to the measured values, then the uncertainty of the predictions may be estimated from experience or from standards such as Annex A of ISO140-2 [84] which provides a standard deviation of reproducibility for the measurement of the sound reduction index. The standard deviation of reproducibility in Annex A of ISO140-2 was based on measurements made in a number of laboratories and should be a reflection of the large variation between laboratories which has been reported [98, 99].

According to Note 17 of ISO140-2, a small value of the standard deviation of repeatability when compared to the standard deviation of reproducibility is common if the tests are carried out correctly. Therefore, the standard deviation of repeatability may be neglected and the uncertainty is given by:

$$u(R_{RI})_{ISO140} \approx s_R$$

(4.8)

with effective degrees of freedom:
The uncertainty in eqn (4.8) does not include the uncertainty of the theoretical calculation of the resonant sound reduction index.

EN12354-1 requires the use of the *in-situ* resonant sound reduction index in the calculations to make the laboratory results invariant [98]. The *in-situ* resonant sound reduction index is given by:

\[
R_{R,\text{situ}} = R_R - \psi_R
\]  

(4.10)

where \(\psi_R\) is a correction term. The value of \(\psi_R\) depends on the building elements such that:

\[
\psi_{R,\text{heavy}} = \frac{10}{\ln(10)} [\ln(T_{s,\text{situ}}) - \ln(T_{s,\text{lab}})] \quad \text{for heavyweight elements}
\]

(4.11)

\[
\psi_{R,\text{light}} = 0 \quad \text{for lightweight elements}
\]

where \(T_{s,\text{situ}}\) and \(T_{s,\text{lab}}\) are the structural reverberation times measured *in-situ* and in the laboratory, respectively.

For heavyweight elements:

\[
u^2(\psi_{R,\text{heavy}}) = \left[\frac{10}{\ln(10)}\right]^2 [u^2(\ln(T_{s,\text{situ}})) + u^2(\ln(T_{s,\text{lab}}))] \]

(4.12)

where the reverberation time has a log-normal PDF and therefore is described in terms of \(Y = \ln(T_s)\) with mean \(\mu_{Y,T_s}\) and variance \(s_{Y,T_s}^2\).

\[
u^2(\ln(T_s)) = \left[\frac{s_{Y,T_s}^2}{n_{r_s}} + \frac{s_{Y,T_s}^4}{2(n_{r_s}-1)}\right]
\]

(4.13)
and $n_{T_s}$ is the number of observations of $T_s$.

The effective degrees of freedom of $u^2(\psi_R)_{\text{heavy}}$ is given by:

$$v_{\text{eff}}(\psi_R, \text{heavy}) = \frac{u^4(\psi_R, \text{heavy})}{\ln(10)^2} \left[ \frac{u^4(\ln(T_s,\text{situ}))}{n_{T_s,\text{situ}}-1} \right] \left[ \frac{u^4(\ln(T_s,\text{lab}))}{n_{T_s,\text{lab}}-1} \right]$$  \hfill (4.14)

For lightweight elements:

$$u^2(\psi_R, \text{lightweight}) = 0$$

with effective degrees of freedom:

$$v_{\text{eff}}(\psi_R, \text{lightweight}) = \infty$$

Therefore, the uncertainty of the in-situ resonant sound reduction index is written:

$$u^2(R_{\text{situ}}) = u^2(R_R) + u^2(\psi_R)$$  \hfill (4.15)

with effective degrees of freedom:

$$v_{\text{eff}}(R_{\text{situ, heavy}}) = \frac{u^4(R_{\text{situ}})}{u^2(\psi_R) \cdot v_{\text{eff}}(\psi_R)}$$  \hfill (4.16)

$$v_{\text{eff}}(R_{\text{situ, lightweight}}) = \infty$$
4.3.2. Uncertainty of the *In-Situ* Direction Averaged Velocity Level Difference

**Uncertainty of the Average Velocity Level**

ISO10848-1 defines the average velocity level measured on an element as:

\[ \overline{L_v} = 10 \log \left( \frac{\mu_{v^2}}{v_o^2} \right) \]  \hspace{1cm} (4.17)

where \( \mu_{v^2} \) is the spatial average of the squared velocity and \( v_o \) is the reference velocity. The spatially averaged squared velocity has a log-normal PDF and therefore, following from eqn (4.1):

\[ \overline{L_{v,p}} = \frac{10}{\ln(10)} \left[ \left( \mu_{v^2} + \frac{s_{v^2}^2}{2} \right) - \ln(v_o^2) \right] \]  \hspace{1cm} (4.18)

where \( \overline{L_{v,p}} \) is a proposed estimate of the average velocity level. The uncertainty of the proposed estimate is written:

\[ u^2(\overline{L_{v,p}}) = \left[ \frac{10}{\ln(10)} \right]^2 \left[ \frac{s_{v^2}^2}{\nu_{v^2}} + \frac{s_{v^2}^2}{2(n_{v^2}-1)} \right] \]  \hspace{1cm} (4.19)

with effective degrees of freedom \( v_{eff}(\overline{L_{v,p}}) = n - 1 \). Alternatively, if:

\[ C = \frac{s_{v^2}}{\overline{v^2}} < \frac{1}{3} \]  \hspace{1cm} (4.20)

where \( C \) is the coefficient of variation and \( s_{v^2} \) and \( \overline{v^2} \) are the sample standard deviation and the mean of the squared velocity, respectively then the PDF of the squared velocity may be approximated as Gaussian [87]. In this case,

\[ u^2(\overline{L_v}) = \left[ \frac{10}{\overline{v^2} \ln(10)} \right]^2 \left[ \frac{s_{v^2}^2}{\nu_{v^2}} \right] \]  \hspace{1cm} (4.21)
Uncertainty of the EN12354 Method

with effective degrees of freedom $v_{eff}(L_v) = n - 1$.

**Uncertainty of the Velocity Level Difference**

The velocity level difference between elements $i$ and $j$ is defined in ISO10848 to be:

$$D_{v,ij} = \overline{L}_{v,ii} - \overline{L}_{v,ji} \quad (4.22)$$

where $\overline{L}_{v,ii}$ and $\overline{L}_{v,ji}$ are the average velocity level of element $i$ due the excitation of element $i$ and of element $j$ due the excitation of element $i$, respectively. The uncertainty of the velocity level difference is:

$$u^2(D_{v,ij}) = u^2(\overline{L}_{v,ii}) + u^2(\overline{L}_{v,ji}) \quad (4.23)$$

with effective degrees of freedom:

$$v_{eff}(D_{v,ij}) = \frac{u^4(D_{v,ij})}{u^4(\overline{L}_{v,ii}) - v_{eff}(\overline{L}_{v,ii})} \quad (4.24)$$

**Uncertainty of the Direction Averaged Velocity Level Difference**

The direction averaged velocity level difference is defined as an average of the velocity level difference in each transmission direction such that:

$$\overline{D}_{v,ij} = \frac{1}{2} (D_{v,ij} + D_{v,ji}) \quad (4.25)$$

Due to the direction averaging, the uncertainty of $\overline{D}_{v,ij}$ should be a combination of the Type A uncertainty calculated from the standard deviation of the observations and Type B uncertainty due to the uncertainty of the observations such that:

$$u^2(\overline{D}_{v,ij}) = \frac{1}{4} [D_{v,ij} - D_{v,ji}]^2 + \frac{1}{4} [u^2(D_{v,ij}) + u^2(D_{v,ji})] \quad (4.26)$$
Only two observations are included in the average and therefore the Type A uncertainty includes only one degree of freedom. In the derivation of the EN12354 method, it was assumed that there is reciprocity between the flanking sound reduction index in each transmission direction [21]. However, in practice, the flanking sound reduction index calculated according to the EN12354 method may not give the same result in each transmission direction [100]. As the difference between $D_{v,ij}$ and $D_{v,ji}$ is increased, the Type A uncertainty becomes much larger than the Type B uncertainty and the number of effective degrees of freedom of the combined uncertainty is reduced to one. Such a small number of effective degrees of freedom results in a large coverage factor for the calculation of the expanded uncertainty.

The validity of the combined uncertainty was evaluated by comparing the expanded uncertainty calculated from the GUM method ($U_{GUM}$) to that calculated from MCS ($U_{MCS}$) as detailed in the ISO Guide 98-3/Supplement 1. The uncertainty estimate is validated if the value of $d = |U_{GUM} - U_{MCS}|$ is less than the permissible numerical tolerance $\delta$ which is based on the number of significant figures of the measurand. For this study, the value of $\delta$ was 0.05 dB. The value of $d$ for the direction averaged velocity level difference of eqn (4.26) is shown in Figure 4.1 for increasing values of $\Delta D = D_{v,ij} - D_{v,ji}$ and increasing values of the standard deviation of the average velocity levels measured on the elements $S_L$. 
Figure 4.1: Evaluation of $d(D_{uij})$ as the magnitude of $\Delta D$ and $s_L$ were increased.

The increase in $s_L$ was identical for all of the elements. The figure shows that the value $d$ was significantly greater than the numerical tolerance $\delta$ over the majority of the range. The results indicate that treating $D_{uij}$ as an average when calculating the uncertainty does not accurately describe the uncertainty.

The averaging of the velocity level differences in each transmission direction is a consequence of reciprocity which was used in the derivation of the EN12354 estimate of the flanking sound reduction index such that [22]:

$$
\hat{\tau}_{ij} = \sqrt{\tau_{ij} \tau_{lj}} = \frac{1}{\sqrt{\frac{\bar{v}_{ii}^2}{s_i^2} \frac{\bar{v}_{jj}^2}{s_j^2}}}
$$

(4.27)

where $\tau_{ij}$ and $\tau_{lj}$ are the flanking sound reduction index in each transmission direction, respectively and $\bar{v}_{ji}^2$ is the best estimate of the mean squared velocity measured on element $j$ due to the excitation of element $i$. Rewriting eqn (4.27) as logarithmic quantities yields:
Therefore, a proposed definition of the direction averaged velocity level difference is:

$$\overline{D_{v_{ij,p}}} = \frac{1}{2} [\overline{L_{v_{ii}}} + \overline{L_{v_{jj}}} - \overline{L_{v_{ij}}} - \overline{L_{v_{ji}}} ]$$  (4.29)

which is a function of the average velocity levels measured on the elements. Therefore, the uncertainty $u(\overline{D_{v_{ij,p}}})$ is wholly Type B uncertainty such that:

$$u^2(\overline{D_{v_{ij,p}}}) = \frac{1}{4} \left[ u^2(\overline{L_{v_{ii}}}) + u^2(\overline{L_{v_{ji}}}) + u^2(\overline{L_{v_{ij}}}) + u^2(\overline{L_{v_{ji}}}) \right]$$  (4.30)

with effective degrees of freedom:

$$v_{eff}(\overline{D_{v_{ij,p}}}) = \frac{u^4(\overline{D_{v_{ij,p}}})}{\frac{1}{v_{eff}(\overline{L_{v_{ii}}})} + \frac{u^4(\overline{L_{v_{ii}}})}{v_{eff}(\overline{L_{v_{ji}}})} + \frac{u^4(\overline{L_{v_{ji}}})}{v_{eff}(\overline{L_{v_{ij}}})} + \frac{u^4(\overline{L_{v_{ij}}})}{v_{eff}(\overline{L_{v_{ji}}})}}$$  (4.31)

The value of $d(\overline{D_{v_{ij,p}}})$ was calculated for increasing values of $\Delta D = D_{v_{ij}} - D_{v_{ji}}$ and increasing values of the standard deviation of the average velocity levels measured on the elements $S_L$ as shown in Figure 4.2.
The figure shows excellent agreement between the expanded uncertainty calculated using the GUM method and MCS over the range of values of $\Delta D$ and $s_L$. Therefore, the eqns (4.30) and (4.31) for the calculation of the uncertainty of $\bar{D}_{v,ij,p}$ were validated for use in this study according to the ISO Guide 98-3/Supplement 1.

**Uncertainty of the Vibration Reduction Index**

The vibration reduction index is defined by ISO10848-1 to be:

$$K_{ij} = \bar{D}_{v,ij} + \psi_K$$  \hspace{1cm} (4.32)

where the use of the correction term $\psi_K$ results in an invariant quantity to describe the attenuation of the vibrational power flow through the junction [101]. The value of the correction term depends on the construction of the elements such that:

$$\psi_{K, heavy} = 10 \log \left( \frac{i_{ij,lab}}{\sqrt{\sigma_{i,lab}^2 \sigma_{j,lab}^2}} \right)$$  \hspace{1cm} (4.33)
Uncertainty of the EN12354 Method

\[ \psi_{K,\text{light}} = 10 \log \left( \frac{l_{ij,\text{lab}}}{\sqrt{s_{i,\text{lab}}^2 + s_{j,\text{lab}}^2}} \right) \]

where the equivalent absorption length of the element in the laboratory is defined to be:

\[ a_{i,\text{lab}} = \frac{2.2 \pi^2 s_{i,\text{lab}}}{T_s \alpha_{\text{lab}} \sqrt{f_{\text{ref}}}} \]  

(4.34)

The term, \( l_{ij,\text{lab}} \) is the junction length between elements \( i \) and \( j \) as measured in the laboratory, \( S \) is the area of the element, \( T_s \) is the structural reverberation time, \( f \) is the 1/3 octave band center frequency in Hz and \( f_{\text{ref}} = 1000 \) Hz. The structural reverberation time \( T_s \) and the equivalent absorption length \( a \) have log-normal PDF’s.

The uncertainty of the correction factor for heavyweight constructions is:

\[ u^2(\psi_{K,\text{heavy}}) = \left[ \frac{5}{\ln(10)} \right]^2 \left[ \frac{4u^2(l_{ij,\text{lab}})}{l_{ij,\text{lab}}^2} + \frac{u^2(s_{i,\text{lab}})}{s_{i,\text{lab}}^2} + \frac{u^2(s_{j,\text{lab}})}{s_{j,\text{lab}}^2} + \frac{u^2(\ln(T_{s,i,\text{lab}}))}{s_{i,\text{lab}}^2} + \frac{u^2(\ln(T_{s,j,\text{lab}}))}{s_{j,\text{lab}}^2} \right] \]  

(4.35)

where \( u^2(\ln(T_s)) \) is as defined in eqn (4.13). The number of effective degrees of freedom associated with \( u^2(\psi_{K,\text{heavy}}) \) is:

\[ v_{\text{eff}}(\psi_{K,\text{heavy}}) = \left[ \frac{5}{\ln(10)} \right]^4 \left[ \frac{16u^4(l_{ij,\text{lab}})}{l_{ij,\text{lab}}^4} + \frac{u^4(s_{i,\text{lab}})}{s_{i,\text{lab}}^4} + \frac{u^4(s_{j,\text{lab}})}{s_{j,\text{lab}}^4} + \frac{u^4(\ln(T_{s,i,\text{lab}}))}{s_{i,\text{lab}}^4} + \frac{u^4(\ln(T_{s,j,\text{lab}}))}{s_{j,\text{lab}}^4} \right] \]  

v_{\text{eff}}(T_{s,i,\text{lab}}) + v_{\text{eff}}(T_{s,j,\text{lab}}) \]  

(4.36)

The uncertainty of the correction factor for lightweight constructions is:

\[ u^2(\psi_{K,\text{light}}) = \left[ \frac{5}{\ln(10)} \right]^2 \left[ \frac{4u^2(l_{ij,\text{lab}})}{l_{ij,\text{lab}}^2} + \frac{u^2(s_{i,\text{lab}})}{s_{i,\text{lab}}^2} + \frac{u^2(s_{j,\text{lab}})}{s_{j,\text{lab}}^2} \right] \]  

(4.37)
with effective degrees of freedom:

\[
v_{\text{eff}}(\psi_{K,\text{i,j}}) = \frac{u^*(\psi_{K,\text{i,j}})}{\left[ \frac{1}{\ln(10)} \right]^2 \left[ \frac{4u^2(l_{ij,\text{situ}})}{l_{ij,\text{situ}}} + \frac{u^2(s_{i,\text{situ}})}{s_{i,\text{situ}}} + \frac{u^2(s_{j,\text{situ}})}{s_{j,\text{situ}}} + u^2(\ln(T_{\text{s,i,\text{situ}}})) + u^2(\ln(T_{\text{s,j,\text{situ}}})) \right]}
\]  

(4.38)

Therefore, the uncertainty of the vibration reduction index is written:

\[
u^2(K_{ij}) = u^2(D_{ij,p}) + u^2(\psi_{K})
\]  

(4.39)

with effective degrees of freedom:

\[
v_{\text{eff}}(K_{ij}) = \frac{u^*(K_{ij})}{v_{\text{eff}}(D_{ij,p}) + v_{\text{eff}}(\psi_{K})}
\]  

(4.40)

**Uncertainty of the In-Situ Direction Averaged Velocity Level Difference**

According to EN12354-1, the vibration reduction index must be changed to an *in-situ* value before it can be used to calculate the flanking sound reduction index such that:

\[
\overline{D_{ij,\text{situ}}} = K_{ij} - \psi_{D}
\]  

(4.41)

where the correction factor \(\psi_{D}\) depends on the construction of the *in-situ* elements such that:

\[
\psi_{D,\text{heavy}} = 10 \log \left( \frac{l_{ij,\text{situ}}}{\sqrt{a_{i,\text{situ}}a_{j,\text{situ}}} \alpha} \right)
\]  

(4.42)

\[
\psi_{D,\text{light}} = 10 \log \left( \frac{l_{ij,\text{situ}}}{\sqrt{s_{i,\text{situ}}}} \right)
\]

The uncertainty of the correction factor for heavyweight constructions is:

\[
u^2(\psi_{D,\text{heavy}}) =
\]  

\[
\left[ \frac{5}{\ln(10)} \left[ \frac{4u^2(l_{ij,\text{situ}})}{l_{ij,\text{situ}}} + \frac{u^2(s_{i,\text{situ}})}{s_{i,\text{situ}}} + \frac{u^2(s_{j,\text{situ}})}{s_{j,\text{situ}}} + u^2(\ln(T_{\text{s,i,\text{situ}}})) + u^2(\ln(T_{\text{s,j,\text{situ}}})) \right] \right]
\]  

(4.43)
Uncertainty of the EN12354 Method

with degrees of freedom:

\[ v_{\text{eff}}(\psi_{\text{D, heavy}}) = \left( \frac{u^4(\psi_{\text{D, heavy}})}{5 \ln(10)} \right)^{\frac{1}{2}} \left( \frac{u^4(t_{ij, \text{situ}})}{s_{ij, \text{situ}}} + \frac{u^4(s_{ij, \text{situ}})}{s_{ij, \text{situ}}} + \frac{u^4(\ln(T_{s,ij, \text{situ}}))}{s_{ij, \text{situ}}} + \frac{u^4(\ln(T_{s,j, \text{situ}}))}{s_{ij, \text{situ}}} \right) \]  

(4.44)

The uncertainty of the correction factor for lightweight constructions is:

\[ u^2(\psi_{\text{D, light}}) = \left[ \frac{5 \ln(10)}{10} \right]^2 \left( \frac{u^2(t_{ij, \text{situ}})}{s_{ij, \text{situ}}} + \frac{u^2(s_{ij, \text{situ}})}{s_{ij, \text{situ}}} \right) \]  

(4.45)

with effective degrees of freedom:

\[ v_{\text{eff}}(\psi_{\text{D, light}}) = \left( \frac{u^4(\psi_{\text{D, light}})}{5 \ln(10)} \right)^{\frac{1}{2}} \left( \frac{10u^4(t_{ij, \text{situ}})}{s_{ij, \text{situ}}} + \frac{u^4(s_{ij, \text{situ}})}{s_{ij, \text{situ}}} + \frac{u^4(\ln(T_{s,ij, \text{situ}}))}{s_{ij, \text{situ}}} + \frac{u^4(\ln(T_{s,j, \text{situ}}))}{s_{ij, \text{situ}}} \right) \]  

(4.46)

Therefore, the uncertainty of the vibration reduction index is written:

\[ u^2(D_{v,ij, \text{situ}}) = u^2(K_{ij}) + u^2(\psi_D) \]  

(4.47)

with effective degrees of freedom:

\[ v_{\text{eff}}(D_{v,ij, \text{situ}}) = \frac{u^4(D_{v,ij, \text{situ}})}{u^4(K_{ij}) + u^4(\psi_D)} \]  

(4.48)
Total Uncertainty of the *In-Situ* Direction Averaged Velocity Level Difference

Substituting eqns (4.26) and (4.39) into eqn (4.47) gives the total uncertainty of the *in-situ* direction averaged velocity level difference:

\[ u^2(D_{v,ij,situ}) = u^2(D_{v,ij,p}) + u^2(\psi_K) + u^2(\psi_D) \]  \hspace{1cm} (4.49)

with effective degrees of freedom:

\[ \nu_{\text{eff}}(D_{v,ij,situ}) = \frac{u^4(D_{v,ij,situ})}{\nu_{\text{eff}}(D_{v,ij,p}) + u^4(\psi_K) + u^4(\psi_D)} \]  \hspace{1cm} (4.50)

4.3.3. Total Uncertainty of the Flanking Sound Reduction Index

The squared uncertainty of \( R_{ij,EN12354} \) is:

\[ u^2(R_{ij,EN12354}) = \frac{u^2(R_{ij,situ})}{4} + u^2(D_{v,ij,situ}) + \left[ \frac{5}{\ln(10)} \right]^2 \left[ \frac{4u^2(s_0)}{s_0^2} + \frac{u^2(s_i)}{s_i^2} + \frac{u^2(s_j)}{s_j^2} \right] \]  \hspace{1cm} (4.51)

with effective degrees of freedom:

\[ \nu_{\text{eff}}(R_{ij,EN12354}) = \]  \hspace{1cm} (4.52)

Substituting eqns (4.15) and (4.49) into eqn (4.55) yields the total uncertainty of the flanking sound reduction index:

\[ u^2(R_{ij,EN12354}) = \frac{s_K^2}{2} + \frac{u^2(\psi_{R,i})}{4} + \frac{u^2(\psi_{R,j})}{4} + u^2(D_{v,ij,p}) + u^2(\psi_K) + u^2(\psi_D) + \left[ \frac{5}{\ln(10)} \right]^2 \left[ \frac{4u^2(s_0)}{s_0^2} + \frac{u^2(s_i)}{s_i^2} + \frac{u^2(s_j)}{s_j^2} \right] \]  \hspace{1cm} (4.53)
with effective degrees of freedom:

\[ v_{eff}(R_{ij,EN12354}) = \frac{u^4(R_{ij,EN12354})}{\sqrt{\frac{u^4(\psi_{ij})}{16\nu_{eff}(\psi_{ij})} + \frac{u^4(\psi_{Rij})}{16\nu_{eff}(\psi_{Rij})} + \frac{u^4(\rho_{ij})}{16\nu_{eff}(\rho_{ij})} + \frac{u^4(\rho_{ij})}{16\nu_{eff}(\rho_{ij})} + \frac{u^4(R_{ij})}{\nu_{eff}(R_{ij})} + \frac{u^4(S_i)}{\nu_{eff}(S_i)} + \frac{u^4(S_j)}{\nu_{eff}(S_j)}}} \]  \hspace{1cm} (4.54)

Therefore the uncertainty of the flanking sound reduction index depends on the standard deviation of reproducibility of the sound reduction index according to ISO140-2, the uncertainty of the average velocity level measured on the elements according to ISO10848, the uncertainty of the areas of the elements and the uncertainty of the correction terms.

4.3.4. Total Uncertainty of the Flanking Sound Reduction Index - Lightweight Elements

The uncertainty of the flanking sound reduction index may be simplified for lightweight elements since the uncertainty of the structural reverberation times are omitted from the correction factors for lightweight constructions. If it is assumed that the variance of the measured lengths and areas is negligible, then the uncertainty of the junction lengths and areas may be omitted from the uncertainty calculations. The assumption that the uncertainty of the measured lengths is negligible may not hold if the lengths and areas are only estimates and the standard uncertainty associated with the estimates is large.

The simplified uncertainty of the flanking sound reduction index is:

\[ u^2(R_{ij,EN12354})_{light} \approx \frac{s^2}{2} + \frac{1}{4} \left[ u^2(L_{ij,ii}) + u^2(L_{ij,ji}) + u^2(L_{ij,ii}) + u^2(L_{ij,ji}) \right] \]  \hspace{1cm} (4.55)

\[ v_{eff}(R_{ij,EN12354})_{light} \approx \frac{u^4(R_{ij,EN12354})_{light}}{\sqrt{\frac{u^4(T_{ij,ii})}{16\nu_{eff}(T_{ij,ii})} + \frac{u^4(T_{ij,ji})}{\nu_{eff}(T_{ij,ji})} + \frac{u^4(T_{ij,ji})}{\nu_{eff}(T_{ij,ji})} + \frac{u^4(T_{ij,ii})}{\nu_{eff}(T_{ij,ii})} + \frac{u^4(T_{ij,ji})}{\nu_{eff}(T_{ij,ji})} + \frac{u^4(T_{ij,ii})}{\nu_{eff}(T_{ij,ii})}}} \]  \hspace{1cm} (4.56)

The equations show that the magnitude of the propagated uncertainty of lightweight constructions depends on the standard deviation of reproducibility \( s_R \) which is a constant and the uncertainty of the average velocity levels measured on the elements.
4.3.5. Apparent Sound Reduction Index

The EN12354 estimate of the apparent sound reduction index \( R' \) is determined from the apparent transmission factor \( \tau' \) such that:

\[
\tau' = \tau_d + \sum_{f=1}^{n} \tau_{f,\text{EN12354}}
\]

(4.57)

Eqn (4.57) includes the contributions from the transmission factor of the separating element \( \tau_d \) and from the flanking transmission factor from \( n \) flanking paths where:

\[
\tau_{f,\text{EN12354}} = 10^{\frac{-R_{\text{f,EN12354}}}{10}}
\]

(4.58)

The transmission factor has been shown to have a log-normal PDF [102]. Therefore, although the propagated uncertainty of the flanking sound reduction index can be calculated using the equations shown in this study, the propagated uncertainty of the apparent sound reduction index shown in eqn (4.57) can not be calculated using the GUM method. MCS performed according to ISO Guide 98-3/Supplement 1 must be used to determine the propagated uncertainty of the apparent transmission factor and the apparent sound reduction index.

4.4. Discussion

4.4.1. Validity of the Uncertainty Calculations

The validity of the equations presented in this study was evaluated by comparing the expanded uncertainty calculated from the GUM method \( U_{\text{GUM}} \) to that calculated from MCS \( U_{\text{MCS}} \) as detailed in the ISO Guide 98-3/Supplement 1. The uncertainty calculated from the GUM method was validated if \( d = |U_{\text{GUM}} - U_{\text{MCS}}| < \delta \) which allows for an accuracy of one digit to the right of the decimal place. The results of the validation are shown Figure 4.3.
Figure 4.3: Evaluation of $d(R_{ij,EN12354})$ as $\Delta D$ and $\Delta s_L$ were varied.

The figure shows good agreement between the uncertainty calculations using the GUM method derived in this study and the uncertainty calculations using MCS. Therefore, the uncertainty equations presented in this study have been validated according to ISO Guide 98-3/Supplement 1.

4.4.2. Total Uncertainty of EN12354-1

The equations presented in this study may be used to determine the propagated uncertainty of the predictions made using the EN12354 method. However further measurements and calculations will be required to estimate the total uncertainty of the predictions. The further studies should include:

- The uncertainty due to the assumptions of the EN12354 method. The EN12354 method is a first order SEA approximation and statistical approaches are always subject to some uncertainty [49].
- The uncertainty due to the use of input data that does not correctly match the structures being evaluated. For example, the EN12354 method may be used to predict the
Uncertainty of the EN12354 Method

apparent sound reduction index between two rooms before they are built. Therefore, inputs such as the in-situ structural reverberation times of the elements may only be estimates of the true values based on prior knowledge or experience.

• The uncertainty due to workmanship. Craik [50] has estimated that the uncertainty due to workmanship may be as high as 5 dB. However, what is needed is a qualification based on a large amount of test data from different sites of different constructions.

Therefore, the total uncertainty of the EN12354 estimate of the flanking sound reduction index is:

\[ u^2(R_{ij,EN12354})_{\text{total}} \approx u^2(\text{Propagated}) + u^2(\text{Method}) + u^2(\text{Estimates}) + u^2(\text{Workmanship}) \]  (4.59)

If all of the sources of error except the propagated uncertainty of the in-situ resonant component of the sound reduction index were ignored, the uncertainty of the EN12354 estimate of the flanking sound reduction index in the 100 Hz 1/3 octave band would be 3.2 dB. If an estimate of the uncertainty due to workmanship of 2.5 dB which is less than that estimated by Craik is included in the calculation, the uncertainty of the EN12354 estimate in the 100 Hz 1/3 octave band would be 4.1 dB with a 95% confidence of ±8.0 dB. When other sources of uncertainty are quantified and included in the estimate, it is expected that the values of the uncertainty and the expanded uncertainty of the EN12354 estimate of the flanking sound reduction index will be larger.

It is suggested that future versions of the ISO10848 and EN12354 series include guidance for the calculation and the declaration of the uncertainty of the input terms and the calculated terms described in the standards.
5. Proposed Changes to ISO10848

5.1. Calculation of the Direction Averaged Velocity Level Difference

The uncertainty analysis of Chapter 4 showed that the direction averaged velocity level difference should be considered as a function of the average velocity levels measured on the elements rather than an average of the velocity level differences in each transmission direction. A new calculation of the term was proposed such that:

\[
\overline{D_{vij,p}} = \frac{1}{2} \left[ \overline{L_{v,ij}} + \overline{L_{v,jj}} - \overline{L_{v,ij}} - \overline{L_{v,jj}} \right]
\]  (5.1)

which allows for the accurate calculation of the propagated uncertainty.

The difference between the proposed estimate and the ISO10848 estimate of the direction averaged velocity level difference was demonstrated by calculating the flanking sound reduction index for a system made of two elements of double-leaf gypsum board on metal studs connected by a junction. The expanded uncertainty was calculated according to the method of GUM using both \(\overline{D_{vij}}\) and \(\overline{D_{vij,p}}\) as well as from the results of MCS. The results are compared in Figure 5.1.
Figure 5.1: Comparison between the expanded uncertainties calculated according to the GUM method using $u(D_{v,ij})$ and using $u(D_{v,ij,p})$ and from MCS for a double-leaf gypsum board elements.

The figure shows that the expanded uncertainty calculated using the GUM method and $D_{v,ij}$ was significantly higher than the other calculated values in several of the 1/3 octave bands. GUM does not give explicit instructions for the estimation of uncertainty when the uncertainty is large compared to the results. However, the basic form of the law of propagation of uncertainties may cease to apply accurately in this region [103].

It is suggested that future versions of ISO10848-1 use $D_{v,ij,p}$ rather than $D_{v,ij}$. Although the difference may seem trivial since the magnitude of the terms is the same, the uncertainty of the terms has been shown to be potentially very different.
5.2. Proposed Estimates Based on Log-Normal PDF’s

5.2.1. Validity of the Approximation of a Gaussian PDF

EN10848-1 defines the average velocity level on a building element to be:

\[ \bar{L}_v = 10 \log_{10} \left( \frac{\sum_{i=1}^{n} v_i^2}{nu_0^2} \right) \]  \hspace{1cm} (5.2)

where \( v_i^2 \) is the mean square velocity at position \( i \), \( v_o \) is the reference velocity and \( n \) is the number of measurement positions. This equation assumes that the mean square velocity has a Gaussian distribution since the best estimate for repeat observations from a Gaussian distribution is the mean of the measurements [62]. However, the PDF study in Chapter 3 showed that the squared velocity has a log-normal PDF which will have a different best estimate than a Gaussian distribution by nature of the shape of the distributions.

For example, the log-normal distribution of \( X \) and the Gaussian distribution of \( Y = \ln(X) \) are compared in Figure 5.2.

Figure 5.2: Comparison between a log-normal distribution and the corresponding Gaussian distribution. Also shown are the mode, median and the mean of the distributions.
The figures show that in the case of the Gaussian distribution, the mean, mode and median are all equal. Conversely, the mean, mode and median of the log-normal distribution can differ by large amounts \cite{104} with the magnitude of the difference depending on the standard deviation of the observations. However, if the standard deviation is small, the mean mode and the median approach each other and the best estimate of the log-normal distribution will have a magnitude similar to the best estimate of the Gaussian distribution. The log-normal distribution can be approximated as a Gaussian distribution if:

\[
C = \frac{s}{\bar{v}^2} < \frac{1}{3}
\]  

(5.3)

where \(C\) is the coefficient of variation and \(s\) and \(\bar{v}^2\) are the sample standard deviation and the mean of the squared velocity, respectively \cite{87}.

The EN12354 method assumes that each of the elements involved in the flanking transmission support diffuse bending wave vibratory fields \cite{21}. For a truly diffuse vibratory response, it would be reasonable to expect a small standard deviation between the observations of the squared velocity measured over the surface of an element. However, it may not be possible to approximate lightweight building elements such as wood frame constructions as homogeneous and isotropic \cite{54}. Studies of wood framed elements \cite{24, 105, 106} have demonstrated that lightweight elements can show a strong gradient in the vibration levels. ISO10848-1 also notes that a strong decrease in the velocity level can occur in lightweight elements. An element with a non-diffuse vibratory response may exhibit a greater variance in the observations of the mean square velocity on the surface of the element than an element with an approximately diffuse vibratory response. Therefore, the approximation of the PDF of the mean square velocity as a Gaussian distribution may not hold for lightweight constructions.

If \(X\) has a two parameter log-normal distribution, then \(Y = \ln(X)\) has a Gaussian distribution with mean \(\mu_Y\) and sample standard deviation \(s_Y\). The best estimate of \(X\) is \cite{74}:
\[ \mu_x = \exp \left( \mu_Y + \frac{s_Y^2}{2} \right) \]  

(5.4)

where the mean \( \mu_Y \) and sample variance \( s_Y^2 \) of \( Y \) are:

\[ \mu_Y = \frac{\sum_{i=1}^{n} \ln(x_i)}{n} \]  

(5.5)

\[ s_Y^2 = \frac{\sum_{i=1}^{n} (\ln(x_i) - \mu_Y)^2}{n-1} \]  

(5.6)

and \( n \) is the number of observations. The variance of \( X \) is:

\[ s_X^2 = \exp \left( 2\mu_Y + s_Y^2 \right) \left( \exp(s_Y^2) - 1 \right) \]  

(5.7)

As \( s_Y^2 \to 0 \), the mean and the variance of the log-normal distribution approaches that of the Gaussian distribution. However, if \( s_Y^2 \) is large, the distributions will have different best estimates.

**5.2.2. Proposed Estimate of the Average Velocity Level**

The best estimate of the log-normal distribution shown in eqn (5.4) was used to derive an alternative calculation of the average velocity level which can be applied to all materials, but especially to lightweight materials. The proposed estimate is described as:

\[ L_{u, proposed} = \frac{10}{\ln(10)} \left[ \left( \mu_Y + \frac{s_Y^2}{2} \right) - \ln(v_c^2) \right] \]  

(5.8)

The difference between the ISO10848 estimate shown in eqn (5.2) and eqn (5.8) may be small when the coefficient of variation is less than one-third. However, in cases where the assumption of a diffuse field may not hold, the proposed estimate may result in a more accurate estimate of the expected value than the ISO10848 estimate.
For example, the average velocity level was calculated according to ISO10848 and according to the proposed estimate for velocity measurements made on a 7 m² double-leaf wall. The estimates of the average velocity level are compared in Figure 5.3.

![Graph showing comparison of velocity level estimates](image)

**Figure 5.3**: Comparison of the velocity level calculated for a 7 m² double-leaf wall excited by an electromagnetic shaker.

The figure shows up to a 6dB difference between the estimates at the lower frequencies. The expected values of the Gaussian and the log-normal distributions are compared to a histogram of the mean square velocity in Figure 5.4.
Proposed Changes to ISO10848

The histogram shows that the mean square velocity had a log-normal PDF which results in a different best estimate of the spatial average of the mean square velocity than when the PDF is approximated by a Gaussian distribution.

5.2.3. Proposed Estimate of the Direction Averaged Velocity Level Difference
The best estimate of a log-normal distribution was also used to derive an alternative estimate of the direction averaged velocity level difference such that:

\[
\overline{D_{v,ij,\text{proposed}}} = \frac{5}{\ln(10)} \left[ (\mu_{ii} + \frac{s_{ii}^2}{2}) - (\mu_{ji} + \frac{s_{ji}^2}{2}) + (\mu_{jj} + \frac{s_{jj}^2}{2}) - (\mu_{ij} + \frac{s_{ij}^2}{2}) \right] \quad (5.9)
\]

where \( \mu_{ii} \) and \( s_{ii}^2 \) are the mean and the variance, respectively of element \( i \) due to the excitation of element \( i \) and \( \mu_{ji} \) and \( s_{ji}^2 \) are the mean and the variance, respectively of element \( j \) due to the excitation of element \( i \). The proposed estimate is similar to the proposed estimate based on a Gaussian distribution shown in eqn (5.2).
5.2.4. Evaluation of the Proposed Estimates

If the error due to the approximation of the PDF of the squared velocity as a Gaussian distribution is defined as:

$$\varepsilon L_v = \bar{L}_v - \bar{L}_{v,\text{proposed}}$$  \hspace{1cm} (5.10)

then the approximation error in one transmission direction is:

$$\varepsilon L_{ij} = \varepsilon L_{v,ii} - \varepsilon L_{v,jj}$$  \hspace{1cm} (5.11)

where $\varepsilon L_{v,ii}$ and $\varepsilon L_{v,jj}$ are calculated according to eqn (5.10). Likewise, the approximation error in the opposite transmission direction is:

$$\varepsilon L_{ji} = \varepsilon L_{v,jj} - \varepsilon L_{v,ii}$$  \hspace{1cm} (5.12)

If $\varepsilon L_{v,ii} = \varepsilon L_{v,jj}$, then $\varepsilon L_{ij} = 0$ as shown in Figure 5.5.

![Figure 5.5: Plot of $\varepsilon L_{ij}$ as the values of $\varepsilon L_{v,ii}$ and $\varepsilon L_{v,jj}$ were varied between -5 and 5 dB.]

However, as the difference between the values of $\varepsilon L_{v,ii}$ and $\varepsilon L_{v,jj}$ increases, the error $\varepsilon L_{ij}$ increases and has a maximum value when $\varepsilon L_{vij} = -\varepsilon L_{vji}$.
The approximation error in the average velocity levels is propagated into the EN12354 estimate of the flanking sound reduction index according to:

$$\varepsilon R_{ij,\text{EN12354}} = \frac{1}{2}(\varepsilon L_{ij} + \varepsilon L_{ji})$$

(5.13)

If the absolute values of the approximation errors of all of the $\varepsilon L_v$ terms are equal or if the approximation errors are negligible, then $\varepsilon R_{ij,\text{EN12354}} = 0$ as shown in Figure 5.6.

When $\varepsilon L_{ij} \neq \varepsilon L_{ji}$, the approximation errors are propagated into the error of the estimation of the flanking sound reduction index.

For example, the approximation errors were calculated for a MDF L-shaped panel as described in Appendix B and tested according to ISO10848 as described in Appendix C. The approximation errors are compared in Figure 5.7.
The figure shows that the approximation errors can be positive or negative. The approximation errors of the average velocity levels are shown to vary between 1.8 dB and -0.7 dB. The highest approximation error in the EN12354 estimate of the flanking sound reduction index was 0.9 dB in the 5000 Hz 1/3 octave band. Periodic lightweight elements can show significant variances between the observations of the squared velocity. For example, in one study of a wood joist floor excited by an ISO tapping machine [105], the velocity level was localized around the point of excitation and over 15 dB lower elsewhere on the surface. This study has shown that such structures can exhibit significant approximation errors which are addressed by the proposed estimates.
6. Reciprocity

6.1. Introduction

In the derivation of the EN12354 estimate of the flanking sound reduction index, reciprocity was used to remove the radiation efficiencies of the elements from the calculations. The exclusion of the radiation efficiencies was beneficial since the values are often not readily available [45] and can be determined correctly only if the velocity amplitudes and the radiation efficiencies of all participating modes are known [41]. However, the assumption of reciprocity may only be applicable to the heavy monolithic constructions for which EN12354-1 was intended [15]. If EN12354-1 is applied to lightweight constructions which may exhibit direction dependent sound transmission, the assumption of reciprocity may result in an over prediction of the apparent sound reduction index.

The validity of point to point reciprocity is not being questioned in this study. Nor is the validity of the assumption of reciprocity to systems which are described using spatial averages of the velocity level when the velocity level when the subsystem supports a uniform energy density. A system can be assumed to support a uniform energy density if it is homogeneous, isotropic and moderately damped [54]. In such a case, the vibratory field will be approximately diffuse so that excitation on the corner of the element, for example will cause an approximately uniform velocity response. What is being questioned in this study is the validity of the assumption of reciprocity for systems such as double-leaf lightweight elements which have been shown to exhibit strong gradients in the vibratory field in directions perpendicular and parallel to the studs [55]. There can be significant spatial variations in the vibratory field where at higher frequencies, the response in the plates is typically much higher than the response on the ribs [107]. Gerretsen and Nightingale note that the plates and studs may actually behave as multiple subsystems in which case reciprocity may not hold [26].

A correction factor is proposed to correct for the over prediction due to the assumption of reciprocity. However, due to the inclusion of the radiation efficiencies in the correction factor, its use is limited. Alternatively, the correction may be used to estimate the error due
to the assumption of reciprocity so that the accuracy of the EN12354-1 predictions can be assessed for lightweight constructions.

### 6.2. Best Estimate of the Flanking Transmission Factor

The EN12354 estimate of the flanking transmission factor is:

\[
\tau_{ij, \text{EN12354}} = \sqrt{\tau_{ij} \tau_{ji}} = \sqrt{\frac{\tau_i \tau_j d_{ij} d_{ji} S_i S_j}{s_\tau^2}} \qquad (6.1)
\]

where the subscripts \(ij\) and \(ji\) indicate transmission between elements \(i\) and \(j\) and \(j\) and \(i\), respectively and it has been assumed that \(\tau_{ij} = \tau_{ji}\) due to reciprocity. Alternatively, \(\tau_{ij}\) and \(\tau_{ji}\) may be considered to be two independent observations of the possible values of the measurand, \(\hat{\tau}_{ij}\). It was shown in Chapter 3 that the probability density function (PDF) of the flanking transmission factor may be approximated by a log-normal distribution. If a variable, \(X\) has a log-normal PDF, then \(Y = \ln(X)\) has a Gaussian PDF with mean \(\mu_Y\) and sample variance \(s_Y^2\). The best estimate of a Gaussian distribution is the mean of the observations such that:

\[
\mu_{Y,\tau_{ij,\text{proposed}}} = \frac{\mu_Y \tau_{ij} + \mu_Y \tau_{ji}}{2} \quad (6.2)
\]

The sample variance of the proposed estimate is:

\[
s_{Y,\tau_{ij,\text{proposed}}}^2 = \left(\frac{\mu_Y \tau_{ij} - \mu_Y \tau_{ji}}{2}\right)^2 \quad (6.3)
\]

Therefore, the best estimate of the proposed transmission factor is:

\[
\tau_{ij,\text{proposed}} = \exp\left(\mu_{Y,\tau_{ij,\text{proposed}}} + \frac{s_{Y,\tau_{ij,\text{proposed}}}^2}{2}\right) \quad (6.4)
\]
Reciprocity

Substituting eqns (6.2) and (6.3) into eqn (6.4) and taking $-10\log$ yields:

$$R_{ij,Proposed,LN} = \left[\frac{R_{ij} + R_{ji}}{2} - \left(\frac{\ln(10)}{40}\right)(R_{ji} - R_{ij})^2\right]$$  \hspace{1cm} (6.5)

where $R_{ij} = -10\log(\tau_{ij})$.

### 6.3. Comparison Between Estimates

The estimations of the flanking sound reduction index are compared for increasing values of

![Comparison Between Precision Methods of $R_{ij}$](image)

$\Delta R$ in Figure 6.1 where $\Delta R = R_{ij} - R_{ji}$.

**Figure 6.1:** Comparison of the estimates of the flanking sound reduction index. The initial value of the calculations was set to $R_{ij} = R_{ji} = 30$ dB and $\Delta R$ was increased.

The figure shows that $R_{ij,EN12354}$ and $R_{ij,Proposed}$ are in agreement when $\Delta R$ is small. However, as $\Delta R$ is increased and the assumption of reciprocity looses validity, the magnitude of the difference between the estimates grows larger. The error due to the assumption of reciprocity is defined as:
The figure shows the value of $\varepsilon_{R_{ij}} = 0.06 \text{ dB}$ when $\Delta R = 1 \text{ dB}$, but increases to $\varepsilon_{R_{ij}} = 1.44 \text{ dB}$ when $\Delta R = 5 \text{ dB}$.

For example, the values of $R_{ij}$, $R_{ji}$, $R_{ij,\text{Proposed, LN}}$ and $R_{ij,\text{EN12354}}$ were calculated for two double-leaf elements made of gypsum board which were screwed to steel studs. The elements were connected by a corner junction and the vibration transmission factors were measured according to the standard, EN10848-1 [25]. The resonant sound reduction index of the elements was calculated according to the theory of Leppington [108]. Since the elements were identical, it was assumed that $\sigma_i = \sigma_j$ and therefore the radiation efficiency terms were excluded from the calculations. The values of $R_{ij}$, $R_{ji}$, $R_{ij,\text{Proposed, LN}}$, and $R_{ij,\text{EN12354}}$ are compared in Figure 6.2.
The figure shows that the values of $R_{ij}$ and $R_{ji}$ differed by as much as 5.4 dB in the 250 Hz 1/3 octave band resulting in a difference between $R_{ij,Proposed, LN}$ and $R_{ij, EN12354}$ of 1.7 dB.

The assumption according to EN12354-1 that the values of $\tau_{ij}$ and $\tau_{ji}$ for the L-shaped panel represent observations of the same distribution was evaluated using a one-way analysis of variance (ANOVA) on the data [87]. The one-way ANOVA requires sets of data for each observation, but in practice, $\tau_{ij}$ and $\tau_{ji}$ each represent single, calculated values which are insufficient for the analysis. However, if the uncertainty and the PDF of $\tau_{ij}$ and $\tau_{ji}$ are known, repeat observations of each value may be produced by sampling from the PDF describing the calculated values using a pseudo-random number generator as discussed in Appendix E.

A one-way ANOVA performed on synthesized values of $\tau_{ij}$ and $\tau_{ji}$ that had been log shifted [109] rejected the null hypothesis in seventeen of the eighteen 1/3 octave bands. The one-way ANOVA results indicate that the assumption of reciprocity did not hold over most of the frequency range. Therefore, the proposed estimate is a more accurate estimate of the measurand than the EN12354 estimate.

### 6.4. Proposed Correction Factor

The proposed log-normal estimate may be rewritten in terms of the EN12354 estimate and a correction factor such that:

$$R_{ij, Proposed} = R_{ij, EN12354} - C_{LN} \quad (6.7)$$

The correction factor, $C_{LN}$ is determined from:

$$C_{LN} = \frac{\ln(10)}{40} \left[ R_j - R_i + D_{v,ij} - D_{v,ji} - 20 \log \left( \frac{s_i}{\sigma_i} \right) - 10 \log \left( \frac{s_j}{\sigma_j} \right) \right]^2 \quad (6.8)$$
Reciprocity

where $R_i$ and $R_j$ are the resonant, \textit{in situ} sound reduction indices of elements $i$ and $j$, respectively and $D_{v,ij}$ and $D_{v,j}^i$ are the \textit{in situ} velocity level difference between elements $i$ and $j$ and elements $j$ and $i$, respectively.

The proposed correction factor includes the radiation efficiencies and therefore may be difficult to apply in practice. If the radiation efficiencies of the elements are predicted theoretically using the equations in Annex B of EN12354-1, for example, the corresponding uncertainty of the predictions may exceed the uncertainty due to the assumption of reciprocity. Therefore, the correction factor is best applied in cases where the radiation efficiencies are known, in cases where the radiation efficiencies of the elements along the flanking path are believed to be similar (as could be the case if the walls of the rooms were made of identical constructions), or in cases where the frequency is above the critical frequency and therefore the radiation efficiencies tend to a value of one \cite{41}. The correction factor may also be used to estimate the error, $\varepsilon_{R_{ij}}$ in cases where the difference between $D_{v,ij}$ and $D_{v,j}^i$ is large compared to the difference between the predicted radiation efficiencies and transmission factors of the elements as may be the case with lightweight constructions.

For example, the proposed correction factor was calculated for measurements made on the four L-panels described in Appendix B. The elements of each of the L-panels were identical and therefore the radiation efficiencies of the elements were assumed to be identical for the calculation of the proposed correction factors. The proposed correction factors are compared in Figure 6.3.
The proposed correction factors were as high as 2.3 dB for the double-leaf MDF on wood studs L-panel. Even the single-leaf L-panels showed differences between the flanking sound reduction indices in each direction. Although the proposed correction factors were less than 0.5 dB over most of the frequency range, the corrections shown are for simple systems tested in the laboratory. Elements in buildings are expected to show a larger difference in the values of the flanking sound reduction index in each transmission direction due to the number and the complexity of transmission paths [37].

6.5. Discussion

The EN12354-1 estimate of the flanking sound reduction index may overestimate the measurand when the assumption of reciprocity does not hold. The proposed estimate does not assume that the values of the flanking transmission factor are identical in each transmission direction and therefore can account for the differences between the terms. The differences may be small for massive, homogeneous structures but may become large for lightweight constructions. Therefore, the proposed estimate and correction factor may be a better estimate for these materials than the EN12354 estimate.
The proposed log-normal estimate and correction factor both include the radiation efficiency terms. The inclusion of these terms limits the usefulness of the proposed estimate. If the values of the radiation efficiencies are predicted theoretically, the uncertainty of the estimated values may be greater than the uncertainty due to the assumption of reciprocity. Therefore, the correction factor may be limited to cases where the radiation efficiencies are known, where the radiation efficiencies of the elements may be assumed to be identical or where the difference between the ratio of the radiation efficiencies and the transmission factor is small when compared to the ratio of the vibration level differences.

Even if the value of the radiation efficiencies are estimated, the difference between the proposed best estimate of the flanking sound reduction index and the estimate according to EN12354-1 may be used to estimate the error due to the assumption of reciprocity.

While the proposed correction can be used to correct for the error due to the assumption of reciprocity, the correction does not address the validity of the method when the assumption of reciprocity does not hold. It has been shown [36, 39, 40] that the equations of EN12354-1 may be equated to a first order Statistical Energy Analysis (SEA) model where there is only one junction and only bending waves are considered. Therefore, the method of EN12354-1 is subject to the same restrictions as SEA [26]. If reciprocity does not hold then the coupling loss factors can be negative and power can flow from a subsystem with low energy to one with high energy which is against the spirit of SEA [60]. Therefore, the validity of using EN12354-1 may be in question for systems where reciprocity does not hold.
7. Separation of the Resonant Component of the Sound Reduction Index

7.1. Introduction

The inputs for the EN12354 estimate of the flanking sound reduction index include the sound reduction index of each of the elements along the flanking path. EN12354 requires that only the resonant component of the sound reduction index be used in the calculations [97]. For monolithic structures with critical frequencies at the low end of the frequency range of interest, the inclusion of the non-resonant components may result in a conservative estimation of the apparent sound reduction index since the sound reduction index above the critical frequency is dominated by the resonant component [110]. However, in the case of lightweight structures which may have critical frequency above the frequency range of interest, the measured sound reduction index may include a significant contribution from the non-resonant component. The inclusion of the non-resonant component for these materials may lead to an underestimation of the apparent sound reduction index [26, 86].

In practice, it would be advantageous to be able to use the sound reduction index measured according to ISO140 or ISO15186 in the EN12354 calculations. However, the sound reduction index measured according to these standards includes contributions from both the resonant and the non-resonant components below the critical frequency. In its current form, EN12354-1 does not recommend a method for the separation of the resonant component of the sound reduction index for elements with critical frequencies in or above the frequency range of interest. However, several methods for the separation have been proposed in the published literature.

One method of separation which has been suggested [98] is to calculate the non-resonant component of the sound reduction index theoretically and then to subtract this from the total, measured sound reduction index. Alternatively, the use of a correction factor has been suggested [86] which can be applied to the total, measured sound reduction index. Lastly, the resonant sound reduction index may be predicted purely from theory. In this chapter, these
Separation of the Resonant Component of the Sound Reduction Index

various methods for calculating the resonant component of the sound reduction index will be examined for lightweight materials with a critical frequency above the frequency range of interest.

7.2. Separation Theory

7.2.1. Prediction of the Resonant Component by Subtraction

One method of calculating the resonant component of the sound reduction index which has been proposed is to predict the non-resonant component and then subtract it from the total measured sound reduction index such that:

$$ R_R = -10 \log \left[ 10^{\frac{-R_T}{10}} - 10^{\frac{-R_{NR}}{10}} \right] \quad (7.1) $$

where $R_R$ and $R_{NR}$ are the resonant and non-resonant components of the sound reduction index, respectively and $R_T$ is the total, measured sound reduction index. The predictions of the non-resonant component of the sound reduction index considered in this study will include the use of the mass law as proposed by Gerretsen [111]:

$$ R_{R, Gerretsen} = -10 \log \left[ 10^{\frac{-R_T}{10}} - 2\sigma_{NR} \left( \frac{2\rho_0 c_0}{2\pi f \rho_s} \right)^2 \right] \quad (7.2) $$

where $\sigma_{NR}$ is the radiation efficiency of non-resonant waves, $\rho_0$ is the density of air, $c_0$ is the speed of sound in air, $f$ is the frequency in Hz and $\rho_s$ is the mass per unit area of the element. Also included in the study are the mass law normal incidence, the mass law field incidence and predictions for finite panels in baffles below coincidence by Leppington [108] and by Sewell [112].

7.2.2. Correction Factors

The application of a correction factor to determine the resonant component of the sound reduction index is based on the premise that the total, measured transmission factor $\tau_T$ may be written in terms of its resonant component $\tau_R$ and non-resonant component $\tau_{NR}$ such that:
Separation of the Resonant Component of the Sound Reduction Index

\[ \tau_T = \tau_{NR} + \tau_R \]  

(7.3)

Dividing eqn (7.3) by \( \tau_R \) yields:

\[ \frac{\tau_T}{\tau_R} = 1 + \frac{\tau_{NR}}{\tau_R} \]  

(7.4)

which may be rewritten in terms of the resonant component as:

\[ \tau_R = \frac{\tau_T}{1 + \tau_{NR}/\tau_R} \]  

(7.5)

The correction factor \( C \) has been defined as:

\[ C = 1 + \frac{\tau_{NR}}{\tau_R} \]  

(7.6)

Substituting eqn (7.6) into eqn (7.5) gives the resonant component as:

\[ \tau_R = \frac{\tau_T}{C} \]  

(7.7)

Rewriting eqn (7.7) in terms of the sound reduction index yields:

\[ R_R = R_T + 10 \log C \]  

(7.8)

which defines the resonant component of the sound reduction index in terms of the total, measured sound reduction index and the correction factor.

There have been several correction factors proposed in the literature and these have been rearranged so that they may be applied according to eqn (7.8). The correction factors have been divided into two groups: those that depend on the separation of the components of the
time and spatially averaged mean square velocity (mean square velocity) of the element and those that depend on the properties of the element other than the mean square velocity.

### 7.2.3. Correction Factors That Depend on the Mean Square Velocity of the Element

The correction factors based on the components of the mean square velocity in theory may be more accurate than other correction factors since they are based on the response of the elements to excitation by an airborne noise source. However, the separation of the components of the mean square velocity may be difficult to achieve and may add additional uncertainty to the calculations. Two methods of separating the components of the mean square velocity are examined in Appendix H.

The correction factors that depend on the mean square velocity of the element include:

**Nightingale \( \xi \) [86]:**

\[
C_{\text{Nightingale}} \xi = \frac{2 \eta_{\text{tot}} \langle v_R^2 \rangle}{\pi f_c \langle v_{NR}^2 \rangle} \tag{7.9}
\]

where \( \eta_{\text{tot}} \) is the total loss factor of the element, \( f_c \) is the critical frequency and \( \langle v_R^2 \rangle \) and \( \langle v_{NR}^2 \rangle \) are the resonant and non-resonant components of the mean square velocity of the element, respectively.

**Proposed Correction Factor**

A correction factor that has been proposed as part of this study is based on the components of the mean square velocity and the radiation efficiencies such that:

\[
C_{\text{proposed}} = 1 + \frac{\sigma_{NR} \langle v_{NR}^2 \rangle}{\sigma_R \langle v_R^2 \rangle} \tag{7.10}
\]

where \( \sigma_R \) is the radiation efficiency of resonant waves. The derivation of eqn (7.10) is shown in Appendix G.
7.2.4. Correction Factors That Depend on Element Properties Other Than the Mean Square Velocity

These correction factors include:

**Method Metzen**

The correction factor from Metzen [113] is:

\[
C_{\text{Metzen}} = 1 + \frac{k_B \eta_{\text{tot}} l_1}{2 \sigma_R} \tag{7.11}
\]

where, \(l_1\) is the largest dimension of the element \(l_1 l_2\). The wave number \(k_B\) of the bending waves is [114]:

\[
k_B = \frac{2 \pi f}{c_o} \sqrt{\frac{f_c}{f}} \tag{7.12}
\]

Therefore, eqn (7.11) may be rewritten as:

\[
C_{\text{Metzen}} = 1 + \frac{\pi f \eta_{\text{tot}} l_1}{\sigma_R c_o} \sqrt{\frac{f_c}{f}} \tag{7.13}
\]

**Method Annex B**

A correction factor which Gerretsen [111] derived from the equations found in Annex B of EN12354-1 is:

\[
C_{\text{Annex B}} = 1 + \frac{2 \sigma_{NR}}{\sigma_B} \frac{(l_1^2 + l_2^2)}{(l_1 + l_2)^2} \eta_{\text{tot}} \sqrt{\frac{f}{f_c}} \tag{7.14}
\]

**Nightingale**

The correction factor defined by Nightingale [39] is:
Separation of the Resonant Component of the Sound Reduction Index

\[ C_{Nightingale} = 1 + \left[ \frac{1}{(1-\mu^2)} \right]^2 \ln \left( \frac{2\pi f_c \sqrt{l_1 l_2}}{c_0} \right) + 0.160 - \frac{U(\Lambda)}{2f \eta_{tot}} \]  

(7.15)

where \( \mu = \frac{\sqrt{l_2}}{f_c} \), \( U(\Lambda) \) is the shape function of the non-resonant vibration transmission factor defined by Sewell [112] which can be simplified if \( \frac{1}{\lambda} \leq 1 \) to:

\[ U(\Lambda) = -0.804 - \left( \frac{1}{2} + \frac{\Lambda}{\pi} \right) \ln \Lambda + \frac{5\Lambda}{2\pi} + \sum_{n=1}^{\infty} \frac{(-1)^{n-1} \Lambda^{2n+1}}{2\pi n (n+1)(2n+1)^2} \]

and \( \xi(\mu) \) is from the definition of the contribution of non-resonant modes to the transmission factor as defined by Leppington [108]:

\[
\begin{align*}
\xi(\mu) &= \frac{1}{4\mu^6} \left[ (2\mu^2 - 1) (\mu^2 + 1)^2 \ln(\mu^2 - 1) \\
&\quad + (2\mu^2 + 1) (\mu^2 - 1)^2 \ln(\mu^2 + 1) - 4\mu^2 - 8\mu \ln(\mu) \right]
\end{align*}
\]

**Correction Factor** \( C_{Appendix \ G} \)

A correction factor which is described in Appendix G is:

\[ C_{Appendix \ G} = 1 + \frac{4f \eta_{tot} \sigma_{NR}}{\pi f_c \sigma_R^2} \]  

(7.16)

This equation may be derived either from the resonant and non-resonant components of the transmission factor or from the resonant and non-resonant components of the mean square velocity of the element.
Villot
A correction factor based on the radiation efficiencies of the element is defined by Villot [115] to be:

\[ C_{\text{Villot}} = 1 + \frac{\sigma_{\text{NR}}}{\sigma_{\text{T}}} \left( \frac{\sigma_{\text{T}} - \sigma_{\text{R}}}{\sigma_{\text{NR}} - \sigma_{\text{T}}} \right) \]  \hspace{1cm} (7.17)

where \( \sigma_{\text{T}} \) is the total radiation efficiency of the element.

7.2.5. Theoretical Predictions
The theoretical predictions of the resonant component of the sound reduction index considered in this study include that by Leppington [108], Lee [116], calculations based on Annex B of EN12354 [23]:

\[ R_{R,\text{EN12354}} = 10 \log \left( \frac{\pi f \rho L}{\rho_o c_o} \right)^2 + 10 \log \left( \frac{l_1 + l_2}{(l_1 + l_2)^2} \frac{\eta_{\text{tot}}}{\sigma_{\text{R}}} \sqrt{\frac{f}{f_c}} \right) \]  \hspace{1cm} (7.18)

and as described by Rindel [15]:

\[ R_{R,\text{JHR}} = 10 \log \left( \frac{\pi f \rho L}{\rho_o c_o} \right)^2 + 10 \log \left( \frac{2f \eta_{\text{tot}}}{f_c \sigma_{\text{R}}} \right) \]  \hspace{1cm} (7.19)

These predictions depend on the accuracy of the estimates of the radiation efficiencies and the total loss factor.

7.3. Experimental Method
The methods of calculating the resonant component of the sound reduction index were evaluated for the single-panel L-panels made of steel and MDF. The values of the resonant and non-resonant radiation efficiencies used in the calculations were from the theoretical equations listed in Annex B of EN12354-1 and the total radiation efficiency was determined experimentally.
The sound reduction index of the panels was measured using the sound intensity technique according to ISO 15186-1:2000 as detailed in Appendix C. Only the Type A sources of uncertainty were considered in the calculation of the uncertainty of the sound reduction index. Jacobsen [117, 118] notes that the Type B errors due to the use of the intensity method should be treated as correlated sources of uncertainty and due to the complexity and unknowns of the analysis, these were not considered. The standard deviation of repeatability of the sound reduction index measurements was less than 0.3 dB through most of the frequency range with the exception of the 100 Hz 1/3 octave band where it was 0.6 dB. Also included in this study was the standard deviation of reproducibility. This standard deviation represents the standard deviation between measurements made on the same element at different test sites by different personnel using different measurement equipment. ISO 15186-1:2000 does not include an estimate of the standard deviation of reproducibility of the measurements, so instead the values listed in ISO140-2 [84] were used in this study. This standard deviation can be as high as 9 dB in the 100 Hz center frequency 1/3 octave band, resulting in a 95% confidence interval of ±6 dB at the low frequencies and ±3 dB over the rest of the frequency range.

The \textit{in-situ} direction averaged velocity level difference, $D_{v,ij,situ}$ was determined according to ISO10848-1. The L-panels used for this study passed the criterion of ISO10848 that the coupling between the elements be weak, but failed the criterion that the velocity field be diffuse. The differences between the minimum and maximum observations of the velocity level measured on the L-panels were found to exceed the 6dB limit stipulated by ISO10848 in almost every 1/3 octave band.

As part of the evaluation, a comparison was also made between the measured intensity flanking sound reduction index of the double-leaf MDF and gypsum board L-panels and the calculated values using the various methods of calculating the resonant component of the sound reduction index. The intensity flanking sound reduction index was measured by mounting the L-panel between a reverberant and a semi-anechoic chamber and exciting element $i$ with a diffuse sound field. The sound intensity from element $j$ due to the excitation of the L-panel with the diffuse sound field and the sound pressure level in the
 Separation of the Resonant Component of the Sound Reduction Index

reverberant chamber were measured to determine the flanking sound reduction index according to ISO15186-2 [119]. The measurement of the intensity flanking sound reduction index is discussed further in Appendix C.

7.4. Comparison between Separation Methods

7.4.1. Prediction of the Resonant Component by Subtraction

The non-resonant components predicted by the different theories are compared to the total, measured sound reduction index in Figure 7.1 and Figure 7.2 for the steel and MDF panels, respectively.

Figure 7.1: Comparison between the non-resonant sound reduction index predicted by the various theories and the total, measured sound reduction index with 95% confidence intervals for a 1.6mm thick steel panel.
Figure 7.2: Comparison between the non-resonant sound reduction index predicted by the various theories and the total, measured sound reduction index with 95% confidence intervals for a 4mm thick MDF panel.

The error bars in the figures represent the 95% confidence intervals of the total, measured sound reduction index which includes both the resonant and non-resonant components. The figures show that the predictions of the non-resonant component were generally less than the total, measured sound reduction index over much of the frequency range. A predicted non-resonant component which is less than the total, measured sound reduction index will result in a negative resonant component of the transmission factor and therefore a resonant component of the sound reduction index which cannot be calculated using eqn (7.1). The 1/3 octave bands where the resonant components of the sound reduction index could not be calculated are indicated discontinuities in the plots of the resonant sound reduction index shown in Figure 7.3 and Figure 7.4 for the steel and MDF panels, respectively.
Figure 7.3: Comparison between the resonant component of the sound reduction index calculated by the subtraction methods and predicted by theory for a 1.6 mm thick steel panel. Also shown is the total, measured sound reduction index with 95% confidence intervals.
Figure 7.4: Comparison between the resonant component of the sound reduction index calculated by the subtraction methods and predicted by theory for a 4 mm thick MDF panel. Also shown is the total, measured sound reduction index with 95% confidence intervals.

The figures show that only the mass law normal incidence could be calculated over the entire frequency range for the steel panel and none of the resonant components could be calculated over the entire frequency range for the MDF panel. Overall, the separation was not successful because the sum of the components was greater than the best estimate of the total, measured sound reduction index.

The success of the predictions of the resonant component by subtraction methods is influenced strongly by the uncertainty of the total, measured sound reduction index. Figure 7.1 and Figure 7.2 show that most of the predictions of the non-resonant component were within the 95% confidence intervals of the measured data. Since the uncertainty of the measured data is predominantly due to the standard deviation of reproducibility, another test site could theoretically have measured data at the lower end of the confidence interval. Therefore it would be interesting to examine the case where the best estimate of the total
Separation of the Resonant Component of the Sound Reduction Index

sound reduction index was considered at the lower end of the confidence interval as shown in Figure 7.5 and Figure 7.6, respectively.

![Steel: Predicted Resonant Components of the Shifted Sound Reduction Index](image)

**Figure 7.5:** Comparison between the resonant sound reduction index calculated from eqn (7.1) with the total, measured sound reduction index adjusted to the bottom of the 95% confidence interval and resonant component predicted from theory for a 1.6mm thick steel panel.
Figure 7.6: Comparison between the resonant sound reduction index calculated from eqn (7.1) with the total, measured sound reduction index adjusted to the bottom of the 95% confidence interval and resonant component predicted from theory for a 4mm thick MDF panel.

Figure 7.5 and Figure 7.6 show that if the total, measured sound reduction index is shifted to the bottom of the 95% confidence interval, the resonant component calculated by subtraction using the different predictions of the non-resonant component can now be predicted in more of the 1/3 octave bands for both the steel and MDF panels than in Figure 7.3 and Figure 7.4. However, only the mass law normal incidence could be predicted over the entire frequency range.

The total, measured sound reduction index has an equal likelihood of being at the top of the confidence interval as the bottom and in this case, none of the subtraction methods would have been successful. These results indicate that prediction of the resonant component by subtraction may be more successful when using the total, measured sound reduction index measured at some laboratories than others. Therefore, subtraction methods are not a reliable method of calculating the resonant component of the sound reduction index, predominantly due to the uncertainty in the measurements.
7.4.2. Prediction of the Resonant Component by Correction Factors

The correction factors that depend on the mean square velocity of the panel were calculated for the steel panel using a modified least square method and data from a series of damped, steel panels as described in Appendix H. The components of the mean square velocity of the MDF panels were not evaluated. The results of applying the correction factors which depend on the mean square velocity are compared to the theoretical resonant components in Figure 7.7.

![Steel: Predicted Resonant Components of the Sound Reduction Index from Correction Factors Based on the Mean Square Velocity](image)

Figure 7.7: The Nightingale correction factor $\xi$ and the proposed correction factor are both based on the resonant and non resonant components of the mean square velocity determined from a series of damped, steel panels. The resonant sound reduction index calculated from these correction factors are compared with the total, measured sound reduction index with 95% confidence intervals and theoretical predictions for a 1.6 mm thick steel panel.

Also shown in the figure is the correction factor, $C_{Appendix G}$ which is based on the theoretical resonant and non-resonant components of the mean square velocity.
The resonant sound reduction index calculated using the proposed correction factor falls within the range of theoretical values over most of the frequency range with the exception of the peak in the 5000 Hz 1/3 octave band and the dip around the 1600 Hz 1/3 octave band. As discussed in Appendix H, the difference between the calculated and the theoretical non-resonant component of the mean square velocity was greater than 8 dB in this frequency range. This may account for the difference between the resonant sound reduction index calculated using $C_{\text{Appendix C}}$ and $C_{\text{proposed}}$.

The results indicate that the use of the separated components of the mean square velocity may be a possible method of calculating the resonant component of the sound reduction index. However, the uncertainty of the calculation is dependant not only on the uncertainty of the total, measured sound reduction index, but also on the uncertainty of the separation components of the mean square velocity. Therefore, the benefit of using measured data may be outweighed by the combined uncertainty of the calculated resonant component of the sound reduction index.

An alternative is to use the correction factors based on element properties other than the mean square velocity as outlined in section 7.2.4. The resonant component of the sound reduction index calculated from these correction factors are compared to the values calculated from theory for the steel and MDF panels in Figure 7.8 and Figure 7.9, respectively.
Figure 7.8: Comparison between the resonant sound reduction index calculated from correction factors with the total, measured sound reduction index with 95% confidence intervals and theoretical predictions for a 1.6mm thick steel panel.
Figure 7.9: Comparison between the resonant sound reduction index calculated from correction factors with the total, measured sound reduction index with 95% confidence intervals and theoretical predictions for a 4mm thick MDF panel.

The correction factor $C_{\text{Annex B}}$ and $C_{\text{Appendix G}}$ are both based on the same theory by Sewell and Leppington for single, homogeneous panels, but $C_{\text{Annex B}}$ results in a larger magnitude resonant component of the sound reduction index. The figures show that most of the calculations and theoretical predictions of the resonant sound reduction index converge within a range of 11 dB at the higher frequencies with the exception of the calculation by $C_{\text{Annex B}}$ for both materials and the calculation by $C_{\text{Nightingale}}$ for the MDF. At the low frequencies, the values from the various calculations and predictions differ by up to 18 dB.

7.5. Comparison between Measured and Calculated Flanking Sound Reduction Indices

The EN12354 estimate of the flanking sound reduction index was calculated for each of the L-panels using the various calculations of the resonant component of the sound reduction index of the elements. The use of the L-panels to evaluate the calculated resonant components of the sound reduction index assumes that the vibration reduction index
measured according to EN10848-1 is valid. However, since the measurements for the calculation of the vibration reduction index failed the EN10848-1 criteria for a diffuse vibration field on the elements, the predictions of the flanking sound reduction index may include large measurement uncertainty. Therefore, although the comparison of the predictions of the flanking sound reduction index may be applicable to the L-panels used in this study, conclusions of the overall accuracy of the different methods of calculating the resonant component of the sound reduction index may not be possible.

7.5.1. Single-Leaf Steel L-Panel

The predictions of the flanking sound reduction index according to EN12354-1 are compared to the measured flanking sound reduction index of the steel L-panel in Figure 7.10. The vibration reduction index was the same for each prediction of the flanking sound reduction index and only the calculated resonant component of the sound reduction index of the elements was changed.
Figure 7.10: Comparison between the measured and predicted flanking sound reduction index of the steel L-panel. The different curves represent predictions using the calculated or theoretical resonant component of the sound reduction index. The total, measured sound reduction index is also included in the comparison. The errors bars are the 95% confidence interval of the measured data.

Also shown in the figure is the predicted flanking sound reduction index if the total, measured sound reduction index is used in the prediction. The total, measured sound reduction index includes contributions from the non-resonant component and can be seen to under predict the flanking sound reduction index by up to 9 dB when compared to the measured data.

Most of the predictions are shown to over predict the flanking sound reduction index between the 200 and the 2500 Hz 1/3 octave bands by up to 15 dB. The smallest deviation between the measured and the calculated values was the calculation using the theory by Lee. The next closest predictions were the theory of Leppington and the proposed MLS correction factor.
The wide range between the estimates of the flanking sound reduction index underlines the need for guidance by EN12354 in choosing which calculation of the resonant sound reduction index to use in the prediction. Otherwise, different organizations could use different methods of calculating the resonant component of the sound reduction index, resulting in predictions which should not be compared to those of an organization using another method of calculating the resonant component.

7.5.2. Single-Leaf MDF L-Panel
The flanking sound reduction indices predicted for the single-leaf MDF L-panel are compared in Figure 7.11.

![Graph showing comparison between measured and predicted flanking sound reduction index for MDF L-panel](image)

**Figure 7.11:** Comparison between the measured and predicted flanking sound reduction index of the MDF L-panel. The different curves represent predictions using the calculated or theoretical resonant component of the sound reduction index. The total, measured sound reduction index is also shown with 95% confidence intervals.
The figure shows that all of the predicted values of the flanking sound reduction index over predict the measured value by 10 to 15 dB over most of the frequency range. The theoretical resonant component by Lee, Leppington and JHR had the smallest deviation from the measured value. The prediction with no correction was overall the most accurate, but this prediction over predicted by over 10 dB in the 4000 Hz and 5000 Hz 1/3 octave bands.

7.5.3. Double-Leaf MDF L-Panel

The predictions of the flanking sound reduction indices of the double-leaf MDF L-panel are shown in Figure 7.12

![Figure 7.12: Comparison between the measured and predicted flanking sound reduction index of the double-leaf MDF on wood studs L-panel. The different curves represent predictions using the calculated or theoretical resonant component of the sound reduction index. The total, measured sound reduction index is also shown with 95% confidence intervals.](image)

Figure 7.12 shows that the prediction using the resonant component from Villot had a similar slope to that of measured flanking sound reduction index up to the 1000 Hz 1/3 octave band, but that it over predicted the measured value by up to 20 dB. The closest fits to the measured
value were those using the resonant component by Lee, Leppington and Villot but none of the predictions truly fit the measured data. The prediction using the total, measured sound reduction index was again the best prediction of the measured flanking sound reduction index.

Below the 500 Hz 1/3 octave band, the predicted flanking sound reduction index using the total sound reduction index is shown to be of the same order of magnitude or higher than the measured flanking sound reduction index. It was expected that the use of the total sound reduction index which includes the non-resonant component would underestimate the measured flanking sound reduction index. The results of the comparison indicate that either the magnitude of the true value of the resonant component of the sound reduction index was approximately the same magnitude as the total, measured sound reduction index or the EN12354 predictions were in error.

7.5.4. Double-Leaf Gypsum Board L-Panel
The predictions of the flanking sound reduction indices of the double-leaf gypsum board L-panel are shown in Figure 7.13.
Figure 7.13: Comparison between the measured and predicted flanking sound reduction index of the double-leaf gypsum board on metal studs L-panel. The different curves represent predictions using the calculated or theoretical resonant component of the sound reduction index. The total, measured sound reduction index is also shown with 95% confidence intervals.

The figure shows that all of the predictions over predict the flanking sound reduction index over the entire frequency range. The prediction using the theories of Lee, Leppington and JHR all have similar slopes to the measured values up to the 800 Hz 1/3 octave band, but overestimate the measured value by 10 dB. The shapes of the curves are again in agreement above the 2000 Hz 1/3 octave band. The smallest deviations from the measured values were the predictions using the resonant component calculated according to the theory of Lee, Leppington and Villot.

The prediction using the total, measured sound reduction index was again the best fit to the measured flanking sound reduction index. However, the over prediction by the calculation using the total sound reduction index suggests that there were other potential sources of error in the prediction in addition to possible errors in the calculated resonant component of the sound reduction index.
7.5.5. Summary

The results of the comparison between the predicted and the measured values of the flanking sound reduction indices are summarized in Table 7.1.

<table>
<thead>
<tr>
<th>Rank</th>
<th>Single-Leaf Steel Calculation</th>
<th>Average Deviation (dB)</th>
<th>Single-Leaf MDF Calculation</th>
<th>Average Deviation (dB)</th>
<th>Double-Leaf MDF Calculation</th>
<th>Average Deviation (dB)</th>
<th>Double-Leaf Gypsum Board Calculation</th>
<th>Average Deviation (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Lee</td>
<td>4.2</td>
<td>Lee</td>
<td>8.3</td>
<td>Lee</td>
<td>14.7</td>
<td>Lee</td>
<td>8.0</td>
</tr>
<tr>
<td>2</td>
<td>Leppington</td>
<td>4.2</td>
<td>Leppington</td>
<td>8.5</td>
<td>Leppington</td>
<td>14.8</td>
<td>Leppington</td>
<td>8.2</td>
</tr>
<tr>
<td>3</td>
<td>MLS</td>
<td>5.6</td>
<td>JHR</td>
<td>11.7</td>
<td>Villot</td>
<td>15.0</td>
<td>Villot</td>
<td>10.9</td>
</tr>
<tr>
<td></td>
<td>No Correction</td>
<td>7.1</td>
<td>No Correction</td>
<td>3.6</td>
<td>No Correction</td>
<td>3.3</td>
<td>No Correction</td>
<td>2.0</td>
</tr>
</tbody>
</table>

Table 7.1: Ranking of the best fits of the prediction of the flanking sound reduction index calculated with the different predictions of the resonant component of the sound reduction index. The average deviation is the average of the differences between the measured and the predicted values over the entire frequency range.

The data in the table shows that the predictions of the flanking sound reduction index which were calculated using the resonant components of the sound reduction index based on the theory of Lee and of Leppington consistently resulted in the best prediction of the measured data. However, with the exception of the steel panel, the deviation of the calculated flanking sound reduction index using the calculated resonant component was worse than that using the total measured sound reduction index which included both the resonant and the non-resonant components.

7.6. Discussion

The prediction of the resonant component of the sound reduction index by subtracting the theoretical non-resonant component from the total measured sound reduction index was shown to be an unreliable method of calculating the resonant component. One of the largest problems with this method is that the resonant component may be very small and therefore, the subtraction may result in a negative value for the resonant transmission factor due to measurement uncertainty. Nightingale [39] has suggested that more accurate estimates of the
in situ resonant sound reduction index can be obtained by direct calculation than by applying a non-resonant correction to measured data.

The correction factors based on the separation of the components of the mean square velocity in theory would be expected give an accurate indication of the response of the panel to excitation by the noise source. However, this method is susceptible to uncertainty in the measurement of the components of the mean square velocity. This uncertainty is propagated along with the uncertainty of the total, measured sound reduction index into the calculated resonant component of the sound reduction index.

The correction factors based on element properties could be successfully calculated over the entire frequency range. However, these correction factors are applied to the total, measured sound reduction index and the uncertainty of the measurements may result in a wide range of values for the resonant component. Both the correction factors based on element properties and the theoretical predictions of the resonant component of the sound reduction are affected by the uncertainty of the measurement of the element properties such as the loss factor and the radiation efficiency. Furthermore, both the correction factors evaluated in this study and the theoretical predictions are based on single, homogeneous panels which are not typical of most lightweight constructions. For example, Gerretsen [120] has noted that double partitions require that the calculation of the sound reduction index includes the effect of the cavity damping in addition to the sound reduction index of each of the leafs. Therefore, the use of predictions based on single panels may lead to errors in the predictions.

If EN12354 is to be applied to lightweight double-leaf structures, then a reliable theory for calculating the resonant component of the sound reduction index of double-leaf constructions is needed. Otherwise, the apparent lack of accuracy of the calculations for single-leaf constructions when applied to double-leaf constructions may limit the application of EN12354 to constructions with critical frequencies below the frequency range of interest.
The calculations of the resonant component of the sound reduction index of the steel and MDF panels by the different methods evaluated in this study varied by as much as 18 dB over the frequency range. As an international standard, EN12354 may be used by people with varying degrees of experience in measuring the sound reduction indices of different materials. It would be difficult for an inexperienced person to choose which separation method consistently results in an accurate prediction of the sound reduction index. Definitive guidance is needed in future versions of EN12354-1 to ensure that consistent estimates of the resonant component of the sound reduction index are used.
8. Evaluation of the EN12354 Method - L-Panels

8.1. Introduction

The accuracy of the EN12354 predictions was evaluated by comparing predictions of the flanking sound reduction index of simple systems of two elements separated by a junction to laboratory measurements. The EN12354 predictions were also compared to predictions obtained from full ESEA models. If the EN12354 method failed to accurately predict the flanking sound reduction index of the simple systems, the failure could be due to the limitations specific to the EN12354 method. However, if both the EN12354 method and the full ESEA model failed to accurately predict the flanking sound reduction index of the simple systems, the legitimacy of applying SEA to the systems or to more complex systems of lightweight elements in buildings would be questionable.

The simple L-panels are described in Appendix B. The ESEA models are detailed in Appendix I. The EN12354 and ESEA estimates and uncertainty calculations of the flanking sound reduction index for this study used the velocity level difference from ISO10848 and the calculated resonant sound reduction index from Leppington [108].

8.2. Overview of the ESEA Models

The ESEA models used for this study differed from the EN12354 model by including higher order flanking paths. Since only one junction was being evaluated, it was assumed that only bending waves needed to be considered in the ESEA model.
8.2.1. Single Leaf L- Panels

A schematic of the single leaf L-panel is shown in Figure 8.1.

![Figure 8.1: Subsystems of the single-leaf L-shaped panels.](image)

The two transmission paths considered in the analysis were:

Path 1: \(1 \rightarrow 4 \rightarrow 6 \rightarrow 7\)

Path 2: \(1 \rightarrow 4 \rightarrow 5 \rightarrow 6 \rightarrow 7\)

Path 1 is the first order path, similar to that used by the EN12354 method. Path 2 includes contributions from both the resonant and the non-resonant transmission through the panels (if considered separately, the non-resonant path would be \(1 \rightarrow 5 \rightarrow 7\)). The inclusion of the non-resonant component is an advantage of using the full SEA model where the non-resonant transmission can be incorporated as direct coupling between two resonant systems while bypassing the physical element through which the sound is passing non-resonantly [59].
8.2.2. Double Leaf L-Panels

A schematic of the double leaf L-panel is shown in Figure 8.2.

![Schematic of double leaf L-panel](image)

Figure 8.2: Subsystems of the double-leaf L-shaped panel.

The figure shows studs at the end of the cavities. Following from Schoenwald [121], a path by path analysis of what were expected to be the major transmission paths gives:

- Path 1: \(1 \rightarrow 2 \rightarrow 6 \rightarrow 7\)
- Path 2: \(1 \rightarrow 2 \rightarrow (3) \rightarrow 4 \rightarrow 6 \rightarrow 7\)
- Path 3: \(1 \rightarrow 3 \rightarrow 4 \rightarrow 6 \rightarrow 7\)
- Path 4: \(1 \rightarrow 2 \rightarrow 9 \rightarrow (8) \rightarrow 6 \rightarrow 7\)
- Path 5: \(1 \rightarrow 2 \rightarrow 9 \rightarrow 8 \rightarrow 7\)
- Path 6: \(1 \rightarrow 2 \rightarrow (3) \rightarrow 4 \rightarrow 5 \rightarrow 9 \rightarrow (8) \rightarrow 6 \rightarrow 7\)

The subsystems in parenthesis are the cavities of the panels when only resonant transmission is considered. Paths 3 and 5 include non-resonant components and are only valid up to the critical frequency. Of the paths listed, only path 1 is a first order flanking path. Path 1 differs from the first order path \(1 \rightarrow 4 \rightarrow 6 \rightarrow 7\) which the EN12354 method uses to describe the flanking sound reduction index.
8.3. Element Properties

8.3.1. Modal Overlap Factors

The modal overlap factors for the panels in each 1/3 octave band are compared in Table 8.1.

<table>
<thead>
<tr>
<th>1/3 Octave Band Center Frequency (Hz)</th>
<th>Single-Leaf Steel</th>
<th>Single-Leaf MDF</th>
<th>Double-Leaf MDF</th>
<th>Double-Leaf Gypsum Board</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Panel $i$</td>
<td>Panel $j$</td>
<td>Panel $i$</td>
<td>Panel $j$</td>
</tr>
<tr>
<td>100</td>
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<td>125</td>
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<td>1.49</td>
<td>1.49</td>
</tr>
<tr>
<td>160</td>
<td>0.83</td>
<td>0.77</td>
<td>1.67</td>
<td>1.67</td>
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<tr>
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<tr>
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<td>2.26</td>
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<tr>
<td>315</td>
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<td>2.74</td>
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<td>3.16</td>
<td>3.16</td>
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<tr>
<td>500</td>
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<td>1.71</td>
<td>3.37</td>
<td>3.37</td>
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<tr>
<td>2500</td>
<td>2.49</td>
<td>2.58</td>
<td>2.69</td>
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</tr>
<tr>
<td>3150</td>
<td>2.46</td>
<td>2.74</td>
<td>3.14</td>
<td>3.14</td>
</tr>
<tr>
<td>4000</td>
<td>2.68</td>
<td>2.72</td>
<td>2.91</td>
<td>2.91</td>
</tr>
<tr>
<td>5000</td>
<td>3.40</td>
<td>2.83</td>
<td>3.25</td>
<td>3.25</td>
</tr>
</tbody>
</table>

Table 8.1: Modal overlap factors $M$ of the L-shaped panels such that $M = 2\pi f n \eta(f)$ where $n$ is the modal density and $\eta$ is the loss factor.

The table shows that only the single-leaf MDF had a modal overlap factor greater than one over the entire frequency range. The results of the ESEA and EN12354 predictions for the rest of the L-panels were expected to have large uncertainty in the 1/3 octave bands with modal overlap factors less than one.

8.3.2. Mode Count

The estimated number of modes in each 1/3 octave band are shown in Table 8.2.
The table shows that the requirement that $M \geq 1$ is a more stringent requirement than $N \geq 5$ for the L-shaped panels since only the double-leaf gypsum board L-panel lacked the sufficient number of modes in the lower frequency bands.

### 8.3.3. Strong Decrease in the Velocity Level

The difference between the maximum and the minimum velocity levels measured on each element are shown in Table 8.3.
The table shows that each of the elements tested had a velocity level decrease in excess of the 6 dB limit set by ISO10848. Therefore, none of the vibratory responses measured on the panels met the ISO10848 definition of a diffuse vibratory field.

8.4. Comparison between EN12354 and ESEA Predictions

8.4.1. Single-Leaf Steel L-Panel

The calculated flanking sound reduction indices of the paths of the ESEA model are compared in Figure 8.3.
The figure shows that the first order path, path 1 was predicted as the dominant transmission path for the steel L-panel. Therefore, the prediction of the flanking sound reduction index by the EN12354 method and by ESEA could be expected to be similar.

The results of the EN12354 and ESEA predictions are compared to the measured intensity flanking sound reduction index in Figure 8.4.
Figure 8.4: Comparison between the values of the flanking sound reduction index of the steel single-leaf L-panel. The error bars are the 95% confidence. The shaded area indicates the 1/3 octave bands with modal overlap factors less than 1.

The EN12354 and the ESEA predictions differ by a maximum of 3 dB in the 630 Hz 1/3 octave band. Even though the EN12354 and the ESEA models evaluate the same transmission path, the values are different because the EN12354 method uses the direction averaged velocity level difference whereas the ESEA model used the velocity level difference in one transmission direction.

The predictions by both the EN12354 method and the ESEA model under predicted the flanking sound reduction index below the 200 Hz 1/3 octave band. In this frequency range, the modal overlap factor was less than one and therefore the EN12354 and ESEA predictions could include potentially large errors. Between the 250 Hz and the 2000 Hz 1/3 octave bands, the EN12354 method and the ESEA model over predicted the flanking sound reduction index by up to 5 dB. Above the 3150 Hz 1/3 octave band, the predictions under predicted the flanking sound reduction index. One possible reason that the EN12354 and ESEA predictions were not in better agreement with the measured data was the failure of the
elements to meet the requirements of an SEA subsystem due to the non-diffuse vibratory field.

The use of the total, measured sound reduction index in the EN12354 prediction underestimated the flanking sound reduction index by an average of 7.2 dB. The under prediction was in agreement with other studies when the non-resonant component was included in the calculations [26].

Even for the very simple system of two homogeneous steel panels the prediction of the flanking sound reduction index by the EN12354 method differed from the measured value by 8 dB.

8.4.2. Single-Leaf MDF L-Panel
The calculated flanking sound reduction index of each path of the ESEA model of the single-leaf MDF L-panel are compared in Figure 8.5.

![Figure 8.5: Comparison between the calculated flanking sound reduction indices of the paths of the ESEA model of the single-leaf MDF L-panel.](image)
The flanking sound reduction index of the first order path, path 1 was up to 20 dB higher than that of path 2. The transmission along the higher order path, path 2 was predicted to be the dominant path up to the 4000 Hz 1/3 octave band. From the 630 Hz 1/3 octave band to the 4000 Hz 1/3 octave band, path 2 and the measured flanking sound reduction index are shown to be in very good agreement. However, below the 630 Hz 1/3 octave band, the estimate of path 2 is shown to underestimate the flanking sound reduction index by up to 15 dB.

The results of the EN12354 and ESEA predictions are compared to the measured intensity flanking sound reduction index in Figure 8.6.

![Figure 8.6: Comparison between the values of the flanking sound reduction index of the single-leaf MDF L-panel. The error bars are the 95% confidence.](image)

The inclusion of the higher order flanking path resulted in good agreement between the prediction by the full ESEA model and the measured flanking sound reduction index between the 500 Hz and the 3150 Hz 1/3 octave bands. In this same frequency range, the EN12354 method over predicted the flanking sound reduction index by an average of 10 dB. Below the 500 Hz 1/3 octave band, neither the EN12354 method nor the full ESEA model accurately predicted the measured flanking sound reduction index.
A potential source of error in the predictions of the flanking sound reduction index below the critical frequency was error in the prediction of the resonant component of the flanking sound reduction index. Figure 8.7 shows predictions by the EN12354 method and the ESEA model using the total, measured sound reduction index of the MDF panel instead of the calculated resonant component.

![Figure 8.7: Comparison between the predictions using the calculated, resonant component of the sound reduction index and the predictions using the total, measured sound reduction index for the single-leaf MDF panel.](image)

The predictions by both the EN12354 method and the ESEA model using the total, measured sound reduction index underestimated the measured flanking sound reduction index over most of the frequency range as would be expected. However, the predictions using the first order flanking paths follow the trend of the measured flanking sound reduction index better than the first order paths using the calculated resonant component. These results suggest that the calculated resonant component may over predict the true value above the 500 Hz 1/3 octave band. Below the 500 Hz 1/3 octave band, the results of the EN12354 prediction suggest that the resonant component is being over predicted whereas the ESEA prediction suggests that the resonant component is being under predicted. Furthermore, the non-diffuse
vibratory field may also be adding to the differences between the predictions and the measured values.

Based on the results, it is concluded that the EN12354 method did not accurately predict the flanking sound reduction index of the simple, single-panel MDF L-panel. The ESEA method also failed to results in accurate predictions below the 500 Hz 1/3 octave band suggesting that the elements failed to meet the requirements of a SEA subsystem in this frequency range.

8.4.3. Double-Leaf MDF L-Panel
The ESEA estimate of the flanking sound reduction index included contributions from six flanking paths. The values of the flanking sound reduction indices of the paths are compared to each other and to the measured flanking sound reduction index in Figure 8.8.

![Figure 8.8: Comparison between the calculated flanking sound reduction indices of the paths of the ESEA model of the double-leaf MDF L-panel.](image)

The total flanking sound reduction index and the flanking sound reduction index of path 1 are shown in the figure to be nearly identical. The figure shows that the first order path, path 1 → 2 → 6 → 7 was the dominant path and that path 6 contributed the least to the flanking sound reduction index of the double-leaf MDF L-panel. The difference between the total
prediction of the flanking sound reduction index of the paths and the measured value was a maximum of 18 dB in the 160 Hz and the 315 Hz 1/3 octave bands.

The results of the EN12354 and ESEA predictions are compared to the measured intensity flanking sound reduction index in Figure 8.9.

The figure shows that the EN12354 estimate and the ESEA estimate differed from each other over most of the frequency range. The largest difference between the predictions was 11 dB in the 250 Hz 1/3 octave band. Since the first order path of the ESEA model was dominant, the differences between the EN12354 and the ESEA estimates were caused primarily by the differences in the velocity level difference used by the predictions. The EN12354 estimate uses the direction averaged velocity level difference between element 4 and element 6.
whereas the ESEA estimate uses the velocity level difference (not direction averaged) between elements 2 and 6.

The EN12354 prediction overestimated the flanking sound reduction index in the 1/3 octave bands below 4000 Hz. The overestimation was up to 29 dB at the low frequencies and the measured value was outside the confidence interval of the estimate up to the 1600 Hz 1/3 octave band. The ESEA estimate also overestimated the flanking sound reduction index up to the 2500 Hz 1/3 octave band. Although both estimates overestimated the measured flanking sound reduction index, the ESEA estimate showed less deviation from the measured value.

The figure shows that predictions by the EN12354 and the ESEA models which used the total, measured sound reduction index instead of the resonant component had the least deviation from the measured flanking sound reduction index. In the 1/3 octave bands where the modal overlap factor was greater than one, the maximum deviation of the EN12354 calculation with no correction was 8 dB whereas the maximum deviation of the ESEA model with no correction was 14 dB as shown in Figure 8.10.
The large difference between the magnitudes of the deviations of the predictions using the total sound reduction index and the predictions using the calculated resonant component of the sound reduction index suggests that the calculation of the resonant component was not a good estimate of the true value. The calculation of the resonant component by Leppington was derived for a single-leaf panel set in a rigid baffle and therefore may not be accurate for double-leaf elements. It is possible that a better estimate of the resonant component of the sound reduction index of double-leaf constructions could result in a better approximation of the flanking sound reduction index.

To evaluate other possible sources of error, corrections were applied to the EN12354 predictions. The corrections included the proposed calculation of the direction averaged velocity level difference using a log-normal PDF as described in Chapter 5 and the proposed correction factor for the assumption of reciprocity as discussed in Chapter 6. The prediction including the correction factors is compared to the ESEA prediction and the measured flanking sound reduction index in Figure 8.11.
Figure 8.11: Comparison between the EN12354 prediction inclusive of correction factors, the ESEA prediction and the measured flanking sound reduction index of the double-leaf MDF L-panel.

The corrections are shown to decrease the EN12354 prediction by up to 2.4 dB in the 250 Hz 1/3 octave band. However, the magnitude of the corrections is small compared to the deviations between the EN12354 prediction and the measured flanking sound reduction index. Therefore, errors due to the approximation of a Gaussian PDF or due to reciprocity are most likely not the primary source of the differences between the predicted and the predicted and the measured values. However, the corrections can not account for the failure of the element to meet the requirements of a SEA subsystem or the limitations to the use of SEA if reciprocity does not hold. Other factors contributing to the deviations between the predictions and the measured values include the low modal overlap factors at the low frequencies and the non-uniform energy density on the elements.

8.4.4. Double-Leaf Gypsum Board L-Panel
The calculated flanking sound reduction indices of the paths of the ESEA model of the double-leaf Gypsum Board L-panel are compared in Figure 8.12.
The figure shows that path 1 was the dominant path up to the critical frequency after which path 2 was the dominant path. However, in the 3150 Hz 1/3 octave band and above, the contribution from path 2 is shown to underestimate the flanking sound reduction index when compared to the measured value whereas path 1 shows good agreement. Therefore the prediction is an underestimate of the true value above the 3150 Hz 1/3 octave band.

The results of the EN12354 and ESEA predictions are compared to the measured intensity flanking sound reduction index in Figure 8.13.
Figure 8.13: Comparison between the values of the flanking sound reduction index of the double-leaf Gypsum Board L-panel. The error bars are the 95% confidence. The shaded 1/3 octave bands had a modal overlap factor less than 1.

As with the double-leaf MDF L-panel, both the EN12354 estimate and the ESEA estimate overestimate the flanking sound reduction index over most of the frequency range. At the coincidence frequency and above, the ESEA estimate underestimates the flanking sound reduction index due to path 2 of the ESEA model.

The difference between the calculated and the measured flanking sound reduction index shown in Figure 8.14 indicated that the EN12354 estimate using the total sound reduction was the best estimate of the measured value.
Figure 8.14: Deviation (Calculated - Measured) of the estimates of the flanking sound reduction index of the double-leaf gypsum board L-panel. The shaded 1/3 octave bands had a modal overlap factor less than 1.

The figure suggests that as with the double-leaf MDF L-panel, the estimate of the resonant sound reduction index was not a good estimate of the true value. The highest deviations of the estimates from the measured value are shown to occur in the 1/3 octave bands with low modal overlap. However, even the EN12354 estimate using the total sound reduction index still overestimated the flanking sound reduction index by up to 5 dB, possibly due to the non-diffuse vibratory field.

The results of the prediction which used the total sound reduction index instead of the calculated resonant component suggest that a more accurate method of calculating the resonant component of double-leaf constructions may result in a better estimate by the SEA models. As with the double-leaf MDF L-panel, other errors due to the failure to meet the requirements of a SEA subsystem may also have contributed to the deviations between the predictions and the measured values. The deviations by the EN12354 method may also have been caused by the exclusion of higher order flanking paths which include the flanking path through the cavities of the panels.
8.5. Discussion of L-Panel Results

Only four L-panels were included in this study so the conclusions which can be drawn from the results are limited in their scope. More general conclusions regarding the accuracy of the EN12354 method would require a study of a number of other L-panels constructed from other common lightweight elements both with and without insulation in the cavity.

The L-panels measured in this study were chosen due to the simplicity of the design and were constructed carefully to reduce the influence of workmanship. The loss factors and the velocity level differences were measured in situ under laboratory conditions. Therefore, the measured values perfectly fit the system being evaluated by the EN12354 prediction. Possible errors such as incorrectly matching the vibration reduction index of elements tested in the laboratory according to ISO10848 and the actual elements or estimates of the structural reverberation time were mitigated by the use of in situ data. Therefore, the conditions for the evaluation were as ideal as possible.

The steel L-panel showed the best fit between the predictions using the EN12354 method, the ESEA model and the measurements. In this case, the direct path was the same for both the EN12354 method and the ESEA model and the results indicated that higher order paths did not contribute significantly. Furthermore, the calculated resonant component appeared to be a good estimate of the true value. However, even in this case, the predictions by the EN12354 method still differed from the measured value by up to 3 dB. Furthermore the uncertainty of the estimate was up to ±38 dB, indicating the non-diffuse vibratory field affected the accuracy of the results. The high uncertainty of the predictions would limit the meaning and the usefulness of the predictions.

The use of the EN12354 method resulted in inaccurate predictions for the single-leaf MDF L-panel and both of the double-leaf L-panels. In the case of the single-leaf MDF L-panel the prediction differed from the measured value by an average of 8.4 dB over the frequency range. The predictions for the double-leaf MDF L-panel and the double-leaf gypsum board L-panel differed from the measured values by an average of 15 dB and 8.7 dB, respectively. The reasons for the errors in the predictions may include the non-diffuse vibratory field
which is in violation of the requirements of a SEA subsystem, the low modal overlap at the low frequencies, which is also in violation of the requirements of a SEA subsystem, the exclusion of higher order flanking paths which proved important even for the simple single-panel system and the difficulty of accurately predicting the resonant component of the sound reduction index. The later presents a real challenge to using the EN12354 method to predict the flanking sound reduction index of lightweight structures as discussed in Chapter 7. The calculations by Leppington or by Lee were based on single-leaf panels and appear to be unsuitable for application to double-leaf constructions. An accurate method of calculating the resonant component of the sound reduction index for double-leaf elements is needed.

The evaluations showed that the use of the total, measured sound reduction of the elements in the EN12354 predictions resulted in more accurate prediction of the flanking sound reduction index of the systems evaluated than the predictions using only the resonant component of the sound reduction index. Although it would be tempting to conclude that the total, measured sound reduction index should be used in the EN12354 predictions instead of the resonant component, the use of the total sound reduction index is not recommended. The EN12354 method is a simplified SEA model and therefore includes resonant transmission only. The inclusion of the non-resonant component would be in violation of the assumptions for SEA and can lead to an underestimation of the flanking sound reduction index. This study included an evaluation of only four systems. Other systems may show a larger difference from the measured flanking sound reduction index than the systems included in this study. Although an underestimation may seem advantageous in that it would be a conservative estimate of the true flanking sound reduction index, the underestimation could result in a considerable overestimation of the significance of the flanking path [26].
The use of a full ESEA model resulted in better predictions than the EN12354 method for the single-leaf MDF L-panel and the double-leaf MDF L-panel. However, the ESEA predictions for these L-panels still differed from the measured values. In the case of the double-leaf MDF L-panel, the difference between the prediction and the measured value was an average of 10 dB over the frequency range. The predictions by the ESEA model of the double-leaf gypsum board L-panel were less accurate than the EN12354 predictions. Therefore, even a more complex SEA model could not accurately predict the flanking sound reduction index of the simple L-panels. The results suggest that the use of SEA may be inappropriate for the lightweight elements tested in this study.
9. Evaluation of the EN12354 Method - Field Testing

9.1. Introduction

The EN12354 method was evaluated by comparing the predicted and the measured apparent sound reduction index between two adjoining rooms in a building. The elements in the rooms included a mixture of lightweight, double-leaf gypsum board and heavy, concrete elements. This study differed from prior studies [30, 31] by including the measurement of the intensity flanking sound reduction index of the elements in the receiving room. Measurement of the intensity flanking sound reduction index allowed for the assessment of the EN123554 prediction of each flanking path and for the ranking of the flanking paths.

9.2. Summary of the Assessed Flanking Paths

The elements in the source and the receiving rooms which are considered in this study are shown in Figure 9.1 and Figure 9.2.

![Figure 9.1: Elevation view of the rooms showing the common wall dD and the flanking element e in the source room, E519.](image-url)
Figure 9.2: Plan of the rooms showing the elements on the walls.

Figure 9.2 shows a wall in room E519 described as both element h and elements h1 and h3. The wall was evaluated in this study both as a single element and as two smaller elements, the geometry of which were based on the wall construction as shown in Figure 9.3.

Figure 9.3: Elements h1 and h3 in the source room E519.

The velocity level measured on wall h was expected to be the highest at locations near the excitation and much lower elsewhere on the wall. Therefore, the measured velocity level on wall h would be expected to have a large variance. The motivation for breaking up the wall into separate elements based on geometry and vibratory response was to reduce the variance of the observations of the velocity level on the wall. However, if h1 and h3 were used in the EN12354 calculations then the area under the window and to the right of the window as
shown in Figure 9.3 could not be included as an element in the predictions since they would require higher order flanking paths.

Wall H in the receiving room was also considered both in its entirety and as a section as shown in Figure 9.4.

During the excitation of the elements in the source room using both an electromagnetic shaker or using a sound source, the loudest noise in the receiving room appeared by ear to be coming from the element H1. Intensity measurements of the noise radiated from the wall H were negative unless the measurement surface included the sound intensity from element H1. A possible explanation for the impression that element H1 was the dominant transmission path may have been the transmission of noise through the cavity of the walls as discussed in Chapter 10. The complete wall H and the section H1 were analyzed separately to investigate the effect of only measuring close to the junction.

The flanking paths evaluated in this study are summarized in Table 9.1 for the calculations considering only the walls and Table 9.2 for the calculations which included the smaller elements.
### Table 9.1: List of the nine flanking paths for the analysis when only walls were considered.

<table>
<thead>
<tr>
<th>Source Element</th>
<th>Receiving Element</th>
</tr>
</thead>
<tbody>
<tr>
<td>h</td>
<td>H</td>
</tr>
<tr>
<td>d</td>
<td>H</td>
</tr>
<tr>
<td>e</td>
<td>H</td>
</tr>
<tr>
<td>h</td>
<td>D</td>
</tr>
<tr>
<td>e</td>
<td>D</td>
</tr>
<tr>
<td>k</td>
<td>D</td>
</tr>
<tr>
<td>d</td>
<td>K</td>
</tr>
<tr>
<td>e</td>
<td>K</td>
</tr>
<tr>
<td>k</td>
<td>K</td>
</tr>
</tbody>
</table>

### Table 9.2: List of the eleven flanking paths when the smaller elements h1, h3 and H1 were considered.

<table>
<thead>
<tr>
<th>Source Element</th>
<th>Receiving Element</th>
</tr>
</thead>
<tbody>
<tr>
<td>h1</td>
<td>H1</td>
</tr>
<tr>
<td>h3</td>
<td>H1</td>
</tr>
<tr>
<td>d</td>
<td>H1</td>
</tr>
<tr>
<td>e</td>
<td>H1</td>
</tr>
<tr>
<td>h1</td>
<td>D</td>
</tr>
<tr>
<td>h3</td>
<td>D</td>
</tr>
<tr>
<td>e</td>
<td>D</td>
</tr>
<tr>
<td>k</td>
<td>D</td>
</tr>
<tr>
<td>d</td>
<td>K</td>
</tr>
<tr>
<td>e</td>
<td>K</td>
</tr>
<tr>
<td>k</td>
<td>K</td>
</tr>
</tbody>
</table>

Elements not listed in the tables including the floors and ceiling of the rooms either had negative intensity, negligible velocity level difference or were parts of higher order flanking paths and therefore were not considered. Further details about the rooms and the elements are discussed in Appendix J.

### 9.3. Summary of the Measurements

Details about the testing procedure are described in Appendix L.
9.3.1. Velocity Level Difference

The velocity level difference was measured *in situ* between the elements in the source and the receiving room. Ideally, the velocity level difference of the elements and the junctions between them should have been measured in the laboratory according to ISO10848 and converted into invariant terms for use in the EN12354 method. However, it was not possible to first test the assemblies in the laboratory for this study. The use of the *in situ* values reduced possible errors in the EN12354 method as opposed to using laboratory data. The possible errors include incorrectly matching laboratory assemblies and assemblies used at the test site or errors in the estimation of the *in situ* equivalent absorption lengths.

The velocity level difference measured between the common wall d and the elements on walls H and K was found to be negative above the 315 Hz and 1000 Hz 1/3 octave bands, respectively. In addition, element e had a negative velocity level difference in one 1/3 octave band to the smaller element H1, but not to any of the walls. A negative velocity level difference would indicate power flow from the element in the receiving room to the element being excited with the shaker which is not possible.

If the element being tested is a double-leaf construction, ISO10848 requires that the response of the element to excitation by an electromagnetic shaker is measured on the leaf on the opposite side of the wall from the excitation as shown in Figure 9.5.

![Figure 9.5: Measurement of the velocity level difference of the common wall.](image)
Since element d was only 16 mm thick with a mass per unit area of $\rho_s = 14.1 \text{ kg/m}^2$ and element D was 46 mm thick with mass per unit area of $\rho_s = 34.4 \text{ kg/m}^2$, it would be reasonable to expect that the response of element D would be less than that of element d when element d was excited with an electromagnetic shaker. Element D was also thicker and heavier than any of the other leafs in the rooms. The negative values of the velocity level difference were therefore most likely a consequence of the thinner flanking elements being easier to excite than the much heavier element D. It is hypothesized that double-leaf elements with one leaf which has a much higher mass per unit area than the other leaf may not be as well suited to predictions using the EN12354 method as symmetric, double-leaf structures.

Alternatively, the first order flanking path of a full SEA model would be from element d to the elements in the receiving room. Element D would still be considered, but as a higher order flanking path, similar to those shown for the L-shaped panels in Chapter 8. Therefore a full SEA model would not be as affected as the EN12354 method by the difference in the magnitude of the velocity level measured on element D and the rest of the elements.

### 9.3.2. Intensity Flanking Sound Reduction Index

The intensity flanking sound reduction index $R_{IF}$ was measured for each element in the receiving room according to ISO15186-2:2003 [119] and as discussed in Appendix M. The accuracy of sound intensity measurements depends very much on the sound field under study [122]. The intensity flanking sound reduction index of several of the elements in the receiving room was negative over the frequency range. The negative intensity indicated the difficulty in making measurements when the direct path was still dominant despite the modifications to the common wall to reduce the direct transmission between the rooms.

A review of the pressure-intensity index from the measurements confirmed that the measurements were affected by the strength of the direct transmission compared to the lower intensity levels measured over the flanking elements. Therefore, the accuracy of the measurements was not expected to be as high as intensity measurements made in the semi-anechoic chamber, for example. The uncertainty associated with the intensity flanking sound
reduction index in this study is the Type A uncertainty calculated from the standard deviation of repeat measurements. Other sources of uncertainty such as the influence of the direct transmission on the measurements of the intensity flanking sound reduction index could not be evaluated and therefore were not included.

9.3.3. Apparent Sound Reduction Index
In addition to the intensity measurements, the apparent sound reduction index between the rooms was measured according to ISO140-4 [123]. The measurements in each transmission direction were within the expanded uncertainty of the measurements in the opposite transmission direction as shown in Figure 9.6.

![Figure 9.6: Comparison of the apparent sound reduction index measured in each transmission direction. The error bars are the 95% confidence.](image)

In addition, the apparent intensity sound reduction index $R'_I$ was calculated from the intensity flanking sound reduction index of each element in the receiving room according to ISO15186-2:2003.
9.4. Summary of the Calculations

9.4.1. Sound Reduction Index

The EN12354 method requires as inputs the resonant sound reduction index of the flanking elements and the total sound reduction index of the common element. As discussed in Chapter 7, the choice of the method of calculating the resonant sound reduction index can have a large affect on the predictions using the EN12354 method. The theory of Leppington [108] was used to calculate the resonant component of the sound reduction index for the flanking elements. The method of Sharp [124] was used to calculate the total sound reduction index of the separating element. Further discussion regarding the calculation of the sound reduction indices can be found in Appendix J.

9.4.2. Velocity Level Difference

The velocity level difference was calculated both according to ISO10848-1 and using the proposed velocity level difference for a log-normal PDF as proposed in Chapter 5. The apparent sound reduction index was calculated using both methods.

9.4.3. Uncertainty

As discussed in Chapter 4, the uncertainty of the apparent sound reduction index can not be calculated using the method of GUM due to the log-normal PDF of the flanking transmission factor. For this study, the uncertainty of the calculated values of the apparent sound reduction index was calculated using Monte Carlo simulations.

9.4.4. Modal Overlap

The values of the modal overlap factors of the elements are shown in Table 9.3.
Table 9.3: Comparison between the values of the modal overlap factors $M$ of the elements where $M = 2\pi f \eta n(f)$. Values less than 1 are highlighted.

<table>
<thead>
<tr>
<th>Element</th>
<th>Complete Walls</th>
<th>Small Elements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$H$</td>
<td>$K$</td>
</tr>
<tr>
<td>Area ($m^2$)</td>
<td>3.2</td>
<td>4.0</td>
</tr>
<tr>
<td>100</td>
<td>1.1</td>
<td>0.7</td>
</tr>
<tr>
<td>125</td>
<td>1.1</td>
<td>0.6</td>
</tr>
<tr>
<td>160</td>
<td>1.8</td>
<td>0.6</td>
</tr>
<tr>
<td>200</td>
<td>2.1</td>
<td>0.8</td>
</tr>
<tr>
<td>250</td>
<td>2.7</td>
<td>1.0</td>
</tr>
<tr>
<td>315</td>
<td>2.8</td>
<td>1.3</td>
</tr>
<tr>
<td>400</td>
<td>3.2</td>
<td>1.4</td>
</tr>
<tr>
<td>500</td>
<td>3.2</td>
<td>1.7</td>
</tr>
<tr>
<td>630</td>
<td>3.3</td>
<td>1.6</td>
</tr>
<tr>
<td>800</td>
<td>3.5</td>
<td>1.8</td>
</tr>
<tr>
<td>1000</td>
<td>3.5</td>
<td>2.2</td>
</tr>
<tr>
<td>1250</td>
<td>3.6</td>
<td>2.2</td>
</tr>
<tr>
<td>1600</td>
<td>3.6</td>
<td>2.4</td>
</tr>
<tr>
<td>2000</td>
<td>3.6</td>
<td>2.3</td>
</tr>
<tr>
<td>2500</td>
<td>3.6</td>
<td>2.2</td>
</tr>
<tr>
<td>3150</td>
<td>3.5</td>
<td>2.0</td>
</tr>
<tr>
<td>4000</td>
<td>3.5</td>
<td>2.2</td>
</tr>
<tr>
<td>5000</td>
<td>3.4</td>
<td>2.2</td>
</tr>
</tbody>
</table>

Of the complete walls, element K, h and e had modal overlap values less than 1 in some of the 1/3 octave bands. The small modal overlap could result in errors in the predictions by the EN12354 method in these 1/3 octave bands. All of the smaller elements had low modal overlap values at the lower frequencies. In the case of element h3, the modal overlap was less than 1 over the entire frequency range. In this study, the use of the smaller elements rather than the complete walls was expected to reduce the accuracy of the predictions using the EN12354 method due to the lower modal overlap factors. However, even if the smaller elements better address the requirement of a uniform energy density, the elements still fail the requirements of a SEA subsystem at the low frequencies due to the modal overlap factors less than 1.
The geometric mean of the modal overlap factors \( M = \sqrt{M_1 M_2} \) \(^{[51]}\) was calculated for each of the flanking paths. All of the values of the geometric means of the modal overlap factors were greater than 1 above the 250 Hz 1/3 octave band with several of the paths having values greater than 1 in the entire frequency range.

### 9.4.5. Number of Modes

The estimated number of modes for each element in each of the 1/3 octave bands are shown in Table 9.4.

<table>
<thead>
<tr>
<th>Element (m²)</th>
<th>Complete Walls</th>
<th>Small Elements</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( H )</td>
<td>( K )</td>
</tr>
<tr>
<td>100</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>125</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>160</td>
<td>6</td>
<td>8</td>
</tr>
<tr>
<td>200</td>
<td>8</td>
<td>10</td>
</tr>
<tr>
<td>250</td>
<td>10</td>
<td>12</td>
</tr>
<tr>
<td>315</td>
<td>12</td>
<td>16</td>
</tr>
<tr>
<td>400</td>
<td>16</td>
<td>20</td>
</tr>
<tr>
<td>500</td>
<td>20</td>
<td>25</td>
</tr>
<tr>
<td>630</td>
<td>25</td>
<td>31</td>
</tr>
<tr>
<td>800</td>
<td>31</td>
<td>40</td>
</tr>
<tr>
<td>1000</td>
<td>39</td>
<td>50</td>
</tr>
<tr>
<td>1250</td>
<td>49</td>
<td>62</td>
</tr>
<tr>
<td>1600</td>
<td>62</td>
<td>79</td>
</tr>
<tr>
<td>2000</td>
<td>78</td>
<td>99</td>
</tr>
<tr>
<td>2500</td>
<td>98</td>
<td>124</td>
</tr>
<tr>
<td>3150</td>
<td>123</td>
<td>156</td>
</tr>
<tr>
<td>4000</td>
<td>156</td>
<td>198</td>
</tr>
<tr>
<td>5000</td>
<td>195</td>
<td>248</td>
</tr>
</tbody>
</table>

Table 9.4: Comparison between the estimates of the number of modes. Values less than 5 are highlighted.

The elements with fewer than 5 modes in the frequency band include those with a modal overlap factor less than 1.
9.5. Measured Flanking Sound Reduction Index

The measured flanking intensity sound reduction indices of elements H, D and K are compared versus the apparent intensity sound reduction index \( R'_j \) and the apparent sound reduction index \( R' \) in Figure 9.7.

![Figure 9.7: Comparison between the flanking intensity sound reduction indices of the elements, the apparent intensity sound reduction index and the apparent sound reduction index between rooms E519 and E517. The error bars are the 95% confidence as calculated using Monte Carlo simulations.](image)

The value of the intensity flanking sound reduction index of element H in the 100 Hz 1/3 octave band is omitted from the figure due to the measurement of negative intensity in this frequency band.

The figure shows that the common wall (element D) had the lowest flanking sound reduction index of the three elements in the receiving room up to the 1600 Hz 1/3 octave band. Element H performed better than element K up to the 630 Hz 1/3 octave band after which element K had the highest flanking sound reduction index. Element H had the lowest flanking sound reduction index above the 1600 Hz 1/3 octave band. In this frequency range, the flanking sound reduction index of element H was even lower than the sound reduction index of the
direct transmission, possibly due to the transmission path through the cavity as discussed in Chapter 10.

The apparent intensity sound reduction index $R'_I$ was calculated by summing the contribution from the three flanking elements. A comparison between $R'_I$ and the apparent sound reduction index $R'$ shows that $R'_I$ overestimated the apparent sound reduction index between the 500 Hz and the 1600 Hz 1/3 octave bands by up to 8 dB. The overestimation could indicate that the calculation of $R'_I$ did not include all of the transmission paths. Although the paths which had negative intensity were excluded from the analysis, localized sources such as the ceiling near the junction with the common wall could still have been possible. Alternatively, the difference could be due to measurement errors as evidenced by the high pressure intensity index values.

Below the 160 Hz 1/3 octave band, the value of $R'_I$ is shown to be lower than $R'$ due to the low intensity flanking sound reduction index of element D and the omission of the intensity flanking sound reduction index of element H due to negative intensity. However, the 95% confidence intervals of $R'_I$ and $R'$ are shown to overlap within this frequency range.

9.6. Evaluation of Each Flanking Path

9.6.1. Element H

The predicted contribution of each flanking path to element H are compared versus the total flanking sound reduction index and the measured intensity flanking sound reduction index in Figure 9.8.
The flanking sound reduction index of path d H could not be calculated above the 315 Hz 1/3 octave band due to the measurement of negative values of the velocity level difference. The figure shows that Path e H had lowest flanking sound reduction index in the frequencies above the 315 Hz 1/3 octave band.

The 95% confidence interval of the total, calculated flanking sound reduction index was ±6.8 dB at the lower frequencies and an average of ±4.4 dB over the remainder of the frequency range. The magnitude of the expanded uncertainty was due to the large variance of the average velocity level measured on the elements. The figure shows that the 95% confidence intervals of the total, calculated flanking sound reduction index and of the intensity flanking sound reduction index overlapped over much of the frequency range with the exception of the 125 Hz 1/3 octave bands where the EN12354 method overestimates the measured value.

The total flanking sound reduction index is shown to overestimate the measured value up to the 630 Hz 1/3 octave band and above the 2500 Hz 1/3 octave band. Above the critical frequency the overestimation by the EN12354 method may be due to the omission of the higher order flanking path through the cavities of the double-leaf walls. To investigate the overestimation below the critical frequency, the EN12354 estimate of the flanking sound
reduction index using the total sound reduction index is compared to the EN12354 prediction using the calculated resonant sound reduction index in Figure 9.9.

The EN12354 prediction using the total sound reduction index underestimates the measured value in most of the 1/3 octave bands below the critical frequency as would be expected. Other errors due to the non-diffuse vibratory field measured on the elements, for example, may affect the EN12354 predictions, but it appears that part of the error in the prediction may be due to the estimate of the resonant component of the sound reduction index of the elements.

The calculation of the total sound reduction index using the estimate of the velocity level difference according to ISO10848 is compared to that using the proposed log-normal estimate in Figure 9.10.
Figure 9.10: Comparison between the flanking sound reduction index of element H calculated using the ISO10848 estimate of $D_{nij}$ and the proposed log-normal estimate $D_{nij,proposed}$. The error bars are the 95% confidence of the ISO10848 estimate.

The difference between the estimates was over 2 dB in the 100 Hz and between the 630 Hz and the 1250 Hz 1/3 octave bands and 1 dB or less over the rest of the frequency range. It is interesting to note that the ISO10848 estimate and the proposed log-normal estimate show different values for the critical frequency with the log-normal estimate matching the critical frequency of the measured data. A comparison between the uncertainties of the estimates showed that the uncertainty of the proposed estimate was slightly less than that of the ISO10848 estimate. Overall, the prediction using the proposed estimate was slightly more accurate than the prediction using the ISO10848 estimate, but both under predicted the flanking sound reduction index at the critical frequency and over predicted at the lower frequencies.
### 9.6.2. Element K

The predicted contributions of each flanking path to element K are compared versus the total flanking sound reduction index and the measured intensity flanking sound reduction index in Figure 9.11.

![Figure 9.11: Comparison between the calculated contribution from each flanking path to the total flanking sound reduction index of element K and the measured value.](image)

The flanking sound reduction index of path d K had the lowest value up to the 1000 Hz 1/3 octave band above which it could not be calculated due to the negative velocity level difference. Path e K had the next highest value followed by path k K.

Overall, the measured flanking sound reduction index did not agree with the prediction by the EN12354 method. The figure shows that the predicted flanking sound reduction index overestimated the measured value by up to 20 dB in the 100 Hz 1/3 octave band. Furthermore, the predictions are shown to have the wrong slope up to the 1600 Hz 1/3 octave band when compared to the measured values in this frequency range. Around the critical frequency, the EN12354 prediction is shown to underestimate the flanking sound reduction index by over 10 dB.
The calculation of the total sound reduction index using the estimate of the velocity level difference according to ISO10848 is compared to that using the proposed log-normal estimate in Figure 9.12.

![Figure 9.12: Comparison between the flanking sound reduction indices of element K calculated using the ISO10848 estimate of $D_{\nu ij}$ and the proposed log-normal estimate $D_{\nu ij,\text{proposed}}$. The error bars are the 95% confidence of the ISO10848 estimate.](image)

The difference between the estimates was over 3 dB at the high frequencies and 1 dB or less over the rest of the frequency range. The proposed estimate is more accurate in the 1/3 octave bands around the critical frequency, but the ISO10848 estimate was more accurate over the rest of the frequency range. Neither estimate was a good fit to the measured data over the entire frequency range.
9.6.3. Element D

The intensity flanking sound reduction index measured for element D included the contributions from the flanking paths as well as the contribution of the direct transmission through path d D. In order that the predicted value could be compared to the measured intensity flanking sound reduction index, the direct sound reduction index was included in the total calculated sound reduction index calculated according to EN123354 as shown in Figure 9.13.

![Figure 9.13: Comparison between the calculated contribution from each flanking path to the total flanking sound reduction index of element D and the measured value.](image)

The difference between the calculated flanking sound reduction index and the sound reduction index of the direct path was an average of 24 dB. Therefore the contribution of the flanking paths to the total flanking sound reduction index of element D was negligible. The critical frequency of the EN12354 prediction is correct, but between the 500 Hz and the 3150 Hz 1/3 octave bands the prediction underestimates the sound reduction index by up to 14 dB.

The 95% confidence intervals of the calculated values were ±10 dB in the 100 Hz 1/3 octave band and an average of ±5 dB over the rest of the frequency range. Up to the 500 Hz 1/3 octave band the confidence interval of the total predicted value overlapped that of the
measured value. Over the rest of the frequency range, the total predicted value underestimated the flanking sound reduction index by up to 18 dB.

The calculation of the total predicted sound reduction index using the estimate of the velocity level difference according to ISO10848 is compared to that using the proposed log-normal estimate in Figure 9.14.

![Graph showing comparison between the flanking sound reduction index of element D calculated using the ISO10848 estimate of $D_{\nu,ij}$ and the proposed log-normal estimate $D_{\nu,ij,proposed}$. The error bars are the 95% confidence of the ISO10848 estimate.](image)

The figure shows no difference between the estimates due to the dominance of the direct transmission path as compared to the flanking paths.

9.6.4. Comparison of the Measured and Calculated Flanking Sound Reduction Indices
The measured and calculated flanking sound reduction indices of the elements in the receiving room are compared in Figure 9.15.
Figure 9.15: Comparison between the measured and the calculated flanking sound reduction indices of the elements. Also included is the sound reduction index of the separating wall. The measured intensity flanking sound reduction index of each element is shown in the figure on the right for comparison.

The predicted values underestimate the contribution of element D above the 500 Hz 1/3 octave band. Element D is predicted to be the dominant path above the 1000 Hz 1/3 octave band but the measurements show element H as the dominant path. The error in the predictions may be due in part to errors in the calculation of the sound reduction index of the direct path and in part to the exclusion of higher order flanking paths from the EN12354 predictions.

The figure shows that only the common separating element had a critical frequency in the 2000 Hz 1/3 octave band. The other elements had critical frequencies in the 2500 Hz 1/3 octave bands. The difference in the critical frequencies may indicate that the walls were not built according to the drawings for the building. The drawings also indicated wood studs instead of metal studs and a drain pipe in the common wall which did not actually exist.

Overall, the predictions using the EN12354 method did not accurately predict the flanking sound reduction indices of the elements.
9.7. Evaluation of the Apparent Flanking Sound Reduction Index

The calculated flanking sound reduction calculated according EN12354 assuming a Gaussian distribution is compared the value calculated assuming a log-normal distribution and to the measured apparent sound reduction index in Figure 9.16.

![Comparison between the apparent sound reduction indices calculated according to EN12354, measured according to ISO140-4 and according to ISO15186-2.]

A comparison of only the measured and calculated apparent sound reduction indices as shown in Figure 9.16 would give the impression that the EN12354 method gave a reasonable prediction in fourteen of the eighteen 1/3 octave bands evaluated. The figure shows an overlap of the 95% confidence intervals up to the 800 Hz 1/3 octave band and above the 2000 Hz 1/3 octave band. The largest difference between the calculated and the measured apparent sound reduction index was 8 dB in the 1600 Hz 1/3 octave band.

Although the flanking sound reduction index for each of the elements was over predicted by the EN12354 method below the 1000 Hz 1/3 octave band, the dominance of the direct transmission path masked the over predictions in the calculation of the apparent sound
reduction index. If the calculation of the direct transmission did not under predict the measured value above the 500 Hz 1/3 octave band, it would have appeared that the EN12354 method accurately predicted the apparent sound reduction index.

The dominance of the direct path could explain how other studies, especially those only evaluating single numbers calculated according to ISO717 [32] have shown agreement within a few dB between the EN12354 predictions and the measurements of the apparent sound reduction index. For example, despite the differences between the measured and the predicted values of the flanking sound reduction indices which were in excess of 20 dB in some of the 1/3 octave bands, the weighted apparent sound reduction index of the EN12354 prediction was 40 dB, only 1 dB lower than that of the measured apparent sound reduction index of 41 dB. Ignoring the flanking paths altogether and only calculating the sound reduction index of the direct transmission path also resulted in a weighted sound reduction index of 41 dB. The use of single numbers and only evaluating the apparent sound reduction index rather than the individual flanking intensity sound reduction indices of the elements in the source room can lead to false confidence in the results of the EN12354 predictions.

9.8. Walls Versus Smaller Elements

9.8.1. Measured Apparent Sound Reduction Index

The calculated flanking sound reduction indices of each element in the receiving room are compared to the apparent sound reduction index calculated according to EN12354 using the smaller elements in Figure 9.17.
Figure 9.17: Comparison between the apparent sound reduction indices calculated according to EN12354 using the smaller elements (R’ elements), using the walls (R’ Calculated), measured according to ISO140-4 and according to ISO15186-2.

The figure does not include the calculated values using the smaller elements in the 4000 Hz 1/3 octave band due to the negative velocity level difference measured for path e H1. The dominance of the direct path prevents comparisons between the results of using the smaller elements versus walls up to the 1000 Hz 1/3 octave band. Over the frequency range where the difference could be assessed, the predictions using the smaller elements was slightly less accurate than the predictions using the wall elements.

9.8.2. Comparison Between H1 and H

The flanking sound reduction index was calculated according to EN12354 when the entire wall H was considered instead of the element H1. The results of the calculations are compared to the measured flanking intensity sound reduction indices in Figure 9.18.
The value of the flanking sound reduction index of H1 is missing from the 4000 Hz 1/3 octave band due to a negative velocity level difference. The figure shows that the measured flanking sound reduction index of H1 was greater than that of H. The measured intensity flanking sound reduction index of element H included element H1 which indicates that the rest of the wall had a lower flanking sound reduction index than element H1. However, the calculated value of H1 was less than that of H with the exception of the 4000 Hz 1/3 octave band.

The differences between the measured and the calculated values are compared in Figure 9.19 where $\Delta$ is the absolute value of the difference between the measured and the predicted values.
Figure 9.19: Difference between the calculated flanking sound reduction index for each element and the measured value.

The value of ΔH1 in the 4000 Hz 1/3 octave band is not included in the figure because it was overestimated due to a negative velocity level difference between elements e and H1. Although the value of ΔH was greater at the low frequencies, the use of the complete wall H versus the small element H1 was slightly more accurate. However, considering element H1 instead of H increased the flanking sound reduction index of the path, thereby increasing the value of the apparent intensity sound reduction index as shown in Figure 9.20.
Figure 9.20: Comparison between the apparent sound reduction index measured according to ISO140-4 using the smaller elements $R'_1(H1)$, measured according to ISO140-4 using the walls and according to ISO15186-2.

The figure shows that replacing the measured intensity flanking sound reduction index of H with that of element H1 increased the value of the apparent intensity sound reduction index, leading to a greater over prediction of the apparent sound reduction index when compared to the value measured according to ISO140-4. Therefore, the inclusion of the smaller element H1 instead of H appears to have overestimated the flanking sound reduction index of the wall.
9.8.3. Path h to H Versus h1 and h3 to H1

The absolute value of the difference between the measured flanking sound reduction index and the EN12354 prediction between the walls h H and the smaller elements h1 H1 and h3 H1 are compared in Figure 9.21.

Figure 9.21: Absolute value of the difference between the calculated values and the measured value.

The use of the walls versus the smaller elements gave better predictions at the low frequencies, possibly due to the low modal overlap factors of the smaller elements. Above the 500 Hz 1/3 octave band, the use of the smaller elements was more accurate. Overall, the use of the observations of the mean square velocity of the small elements versus using the observations of the entire wall had little impact on the overall accuracy of the calculation of the flanking sound reduction index.
9.9. Discussion of Field Testing Results

It would have been interesting to have also evaluated the apparent sound reduction index using a full ESEA model as was done for the L-shaped panels. However, a SEA model was not planned at the time of the field measurements. The mean square velocity was only measured according to ISO10848 and therefore not on the excited side of the double-leaf elements. Without the additional velocity data, calculation of the first order paths of the full ESEA model would not have been possible. In hind sight, more measurements of the mean square velocity on the elements should have been made.

The negative velocity level difference measured for flanking paths originating from the common wall presents a possible difficulty with applying the EN12354 method to double-leaf constructions with leaves of different masses. In hind sight, the source and the receiver rooms should have been swapped for the field testing so that the thicker leaf was exposed to the sound source.

The results of the study showed that the EN12354 method did not accurately predict the flanking sound reduction indices of the elements in the source room. The EN12354 method over predicted the flanking sound reduction index at the low frequencies by as much as 20 dB and under predicted the value around the coincidence dip frequencies of the elements. Possible sources of the errors in the predictions include the non-diffuse vibratory field measured on the elements in violation of the requirements of a SEA subsystem. Another source could be the calculation of the resonant component of the sound reduction index for the double-leaf elements.

The critical frequency of some of the elements was higher than expected. The calculation of the sound reduction index for use in the EN12354 method could have been changed to better reflect the critical frequencies of the measurements. However, using a different critical frequency once the problem was found would have been “cheating” since in a real application of the EN12354 method, such errors might not be known. Deviations between the blueprints and what is built represents a problem to any of the prediction models, not just EN12354.
Breaking up the walls into smaller elements in this study was found to over predict the flanking sound reduction index at the low frequencies, possibly due to the low modal overlap factors of the smaller elements. At the high frequencies, the calculations using the smaller elements were found to better predict the flanking sound reduction index than the calculations using the complete walls. However, when taking into account the over and under predictions, the overall effect of using the smaller elements in the calculations was negligible.

The dominance of the direct flanking path in the calculation of the apparent sound reduction index masked most of the deviations between the EN12354 method and the measured values of the flanking sound reduction index. The effect of the direct path could explain how other studies, especially those only evaluating single numbers calculated according to ISO717 have shown agreement within a few dB between the EN12354 predictions and measurements.
10. Observations Regarding Wood versus Metal Studs

During the course of the field testing, it was observed that element H1 sounded like the dominant source of the noise transmitted into the receiving room. The noise from element H1 sounded higher frequency and was caused by the excitation of the elements in the source room by either airborne noise or by an electromagnetic shaker. A comparison between the measured flanking sound reduction indices of the elements in the receiving room showed that element H1 had the lowest measured value above the 2000 Hz 1/3 octave band. The intensity flanking sound reduction index of H1 was even lower than that of the common wall which included the direct transmission path. These results indicate that flanking paths ending with element H1 were the dominant source of noise transmitted to the receiving room above the 2000 Hz 1/3 octave band.

The exact construction of the junction between wall H, h and D could not be found in the stacks of architectural drawings available for the field testing site. However, it is believed that the junction was similar to a typical T junction [125] that shown in Figure 10.1.

![Figure 10.1: Common T junction using metal studs.](image)

Therefore, the cavity of the double-leaf walls was believed extended between all of the double-leaf walls. Since the cavity of the double-leaf walls did not include insulation, the only mechanism for stopping the transmission of airborne noise through the cavity was the thin, metal studs. The sound reduction index of the metal studs was not expected to be high.
and was reduced further by periodic holes in the studs to allow wire to be run through the wall. The transmission through the cavity would include both resonant and non-resonant components.

The cavity therefore offers a transmission path between all of the connected double-leaf walls in the rooms. For example, when the electromagnetic shaker was attached to element k as shown in Figure 10.2, the noise being radiated by element H1 was clearly audible in the receiving room most likely due to the transmission through the cavity.

![Figure 10.2: Drawing showing the cavity of the double-leaf wall that extended around the rooms. The figure shows the path of airborne noise caused by the excitation of element k in the source room to element H1 in the receiving room.](image)

The loud, high frequency noise was also heard during the testing of the double-leaf gypsum board on metal studs L-shaped panel. However, the noise was not as noticeable during the testing of the double-leaf MDF on wood studs L-shaped panel shown in Figure 10.3.
Observations Regarding Wood versus Metal Studs

Figure 10.3: Assembly of the double-leaf MDF L-panel with wood studs.

The location of the wood studs in the corner between the panels was chosen so that the two panels would be symmetrical. The metal studs of the double-leaf gypsum board panel had an identical configuration. As with the elements of the field testing site, the L-panels did not have insulation in the cavity between the leaves. Therefore, the only impediment to the transmission of airborne noise through the cavity was the thin, metal studs as shown in Figure 10.4

Figure 10.4: Corner assembly of the double-leaf gypsum board L-panel with metal studs.

The metal studs are shown in different colors so that they can each be distinguished. The arrow shows the transmission through the cavity.
Observations Regarding Wood versus Metal Studs

It is hypothesized that although the use of metal studs versus wood studs can reduce the direct transmission of noise through a panel, the solid, wood studs in the wall cavity may offer a higher flanking sound reduction index for flanking paths through the cavity without insulation. If this hypothesis is correct, it would be advisable to put a blocking mass in the junctions between double-leaf elements that are constructed using metal studs without insulation in the wall cavity.

Ideally, an experimental evaluation of this hypothesis would have included measurements on a double-leaf gypsum board with wood studs L-panel. Unfortunately, a L-panel of that construction was not included amongst those built for this study. Future work should include the measurement of the intensity flanking sound reduction index on L-panels including wood and metal studs to compare the effect of the stud material on the transmission of noise through the cavity and into the receiving room.
11. Conclusions and Recommendations

11.1. Conclusions

As part of this study, the probability density functions of the terms described by the EN12354 method were determined. Knowledge of the PDF’s of the terms was necessary for the calculation of the propagated uncertainty of EN12354-1 and ISO10848-1 according to the method of GUM. GUM was also shown to be applicable to terms which had a log-normal distribution, but were converted into logarithmic values for the EN12354 predictions. The uncertainty of the EN12354 method was found to depend on the standard deviation of reproducibility of the sound reduction index according to ISO140-2, the uncertainty of the average velocity levels measured on the elements according to ISO10848 and the uncertainty of the dimensions and the structural reverberation times of the elements. Input data with small variances and small Type B uncertainty would lead to low uncertainty in the EN12354 predictions. The uncertainty calculations will be a valuable tool to those employing the EN12354 since the uncertainty calculations will help to quantify the accuracy of the predictions, something which has been sorely lacking until this study.

Furthermore, knowledge of the PDF’s led to alternative calculations of the average velocity level in cases where the PDF can not be approximated as a Gaussian distribution. Errors due to the approximation of a Gaussian distribution were shown to propagate directly into the prediction of the flanking sound reduction index.

Changes to ISO10848-1 were proposed to improve the predictions for lightweight construction and to result in an accurate statement of the uncertainty. Also proposed was a correction factor to correct for errors due to the assumption of reciprocity between the flanking sound reduction index in each transmission direction. The use of the correction factor is limited by the inclusion of the radiation efficiency terms.
The requirement that only the resonant component of the sound reduction index be used in the EN12354 calculations was shown to be a real challenge to the application of EN12354 as well as full SEA models to lightweight constructions. A review of different methods of calculating the resonant component of the sound reduction index either from data measured according to international standards or from theory showed that the different methods result in a wide range of values. Directly calculating the resonant component using the theory of Lee or Leppington was shown to be the best method of determining the resonant component of the sound reduction index. However, these theories were based on homogeneous, single panels and were shown to give inaccurate results if they were applied to the double-leaf constructions which are typical of lightweight elements. If the EN12354 method is to be applied to lightweight, double-leaf constructions, a new theoretical model to calculate the resonant component of the sound reduction index for double-leaf panels is needed.

The experimental evaluation of the application of the EN12354 method to lightweight building constructions presented in this study included laboratory and field testing. The testing was unique compared to prior studies in that intensity measurements allowed for the measurement of the flanking sound reduction index of each of the flanking paths.

The use of the L-panels in the laboratory had the advantage over field testing in that possible errors such as mismatching of velocity level difference data or the effect of workmanship were mitigated. The evaluation of the single-leaf MDF L-shaped panel showed the prediction by the EN12354 method differed from the measured value by up to 14 dB. The results of the EN12354 predictions for the double-leaf L-shaped panels differed from the measurements by as much as 29 dB. Better accuracy can not be expected for more complex elements in buildings. Furthermore, the full ESEA model also failed to accurately predict the flanking sound reduction index for the double-leaf L-panels indicating that the use of SEA may not be appropriate for this type of building element.

The field measurements included the measurement of the flanking sound reduction index of each flanking path in addition to the measurement of the apparent sound reduction index between the source and the receiving rooms. The path by path analysis showed errors
between the measured and the predicted flanking sound reduction indices in excess of 10 dB in some of the 1/3 octave bands. However, due to the dominance of the direct path, the difference between the measured and the predicted weighted apparent sound reduction index was only 1 dB. Evaluations including only single number ratings or only the evaluation of the apparent sound reduction index were concluded to be inaccurate assessments of the accuracy of the EN12354 method if the direct transmission was the dominant path.

Reasons for the potentially large differences between the measured and the predicted values include the limitations of the simplified calculations according to EN12354 such as the inclusion of only first order flanking paths. Furthermore, the elements evaluated in this study failed to meet the requirements of an ideal SEA subsystem. For example, many of the elements had modal overlap factors less than one at the low frequencies in violation of the requirements of a SEA subsystem and limiting the accuracy of the predictions in these frequency bands.

The requirement of a SEA subsystem that the elements support a diffuse vibratory field can be problematic for lightweight elements. Attempts to reduce the non-diffuse vibratory fields in this study by breaking up walls into smaller elements was found to improve the predictions at the higher frequencies, but at the cost of less accurate predictions at the lower frequencies. Therefore, the overall effect of the smaller elements on the accuracy of the predictions in this study was negligible. However, the ability to break up walls into smaller elements is limited due to the exclusion of higher order paths.

Based on the systems tested as part of this study, the current version of the standard, EN12354 can not be endorsed as a reliable means of predicting the apparent sound reduction index for lightweight constructions. If EN12354 is used in its current form as an estimate of the apparent sound reduction index in buildings which include lightweight elements, it should only be with the understanding that the predictions can potentially include significant errors.
11.2. Recommended Changes to EN12354 and ISO10848

Recommended changes to future versions of EN12354-1 include the addition of:

- A section explaining the calculation of the modal overlap factor and the modal density and the potential effect on the accuracy of the EN12354 method when the modal overlap factor is less than one. The section will be of great assistance to people who are not familiar with SEA, but use EN12354 without understanding the possible frequency limitations of the calculations.

- A section about the uncertainty of the predictions. The section should include an explanation of how to calculate the uncertainty and the expanded uncertainty from the uncertainty of the input data as was shown in this thesis. Other sources of uncertainty such as workmanship should also be discussed with sources listed if estimates are given. The addition of the uncertainty analysis will greatly improve the confidence in the predictions. Furthermore, a person calculating the flanking sound reduction index according to the EN12354 method would be able to determine how the quality of the input data affected the uncertainty of the EN12354 predictions.

- A requirement that measurements of the velocity level difference, the structural reverberation time, the sound reduction index and other inputs into the EN12354 method include an expression of the uncertainty of the measurements. The uncertainty of the measurements is a necessary part of any measurement which is lacking from many acoustics standards. The determination of uncertainty is an important topic which needs to be addressed by the ISO and CEN standard committees when revisions of standards such as ISO140, ISO10848 and EN12354 are published in the future.

- More equations to account for higher order flanking paths to the EN12354 calculations to make it more applicable to lightweight, double-leaf constructions. Instead of treating double-leaf walls as homogeneous elements, the EN12354 method should address the multiple elements in the wall. The inclusion of more flanking paths which use experimental data to define the coupling loss factors will increase the complexity of the EN12354 calculations, but with the benefit of more accurate predictions.
Conclusions and Recommendations

- A statement regarding how the resonant component of the sound reduction index is to be calculated. Failure to specify which calculation to use can result in a wide range of predicted values when different calculation methods are used.

Most importantly, if the EN12354 method is to be applicable to lightweight elements, a reliable theory for calculating the resonant sound reduction index of double-leaf constructions is needed. Until such a theory is published, the use of the EN12354 method should be restricted to elements with critical frequencies below the frequency range of interest.

It is recommended that future versions of ISO10848-1 include:

- The proposed estimate for the direction averaged velocity level difference. The error in the ISO10848 estimate is propagated into the calculation of the apparent sound reduction index. The proposed estimate was shown in some cases to improve the prediction of the apparent sound reduction index by several dB.

- Definite limits on the validity of the assumption of a diffuse vibratory field. The limits should include a statement that if the limit is exceeded then the EN12354 method should not be used.

- A section about the uncertainty of the predictions as described in this thesis.

11.3. Proposed Future Work

This study resulted in equations to describe the propagation of uncertainty in the EN12354 predictions due to the uncertainty of the inputs. Other sources of uncertainty such as workmanship also need to be quantified. There have been other studies made and numbers for the uncertainty due to workmanship are often mentioned at conferences. However, without reference to a large scale study of multiple field measurement sites, a true indication of the uncertainty due to workmanship can not be known.

Future work to evaluate the flanking transmission through double-leaf constructions with metal studs should include building and testing L-shaped panels identical to the gypsum board on metal studs L-shaped panel evaluated in this study, but with wood studs. The
measurements from the new L-shaped panel will allow for the evaluation of the hypothesis that metal studs allow for greater flanking transmission in double-leaf constructions than wood studs.

Additional field testing with additional measurements to allow for a full ESEA model may allow for greater insight into the differences between the predicted and measured flanking sound reduction indices. The future measurements should include the intensity flanking sound reduction index of the elements in the receiving room since the use of intensity measurements proved to be insightful in this study to distinguish between the dominant paths and the difference between the flanking and the direct transmission.
Appendix A: Derivation of the EN12354 Method

A.1. Derivation of the Flanking Transmission Factor

If an element is mounted between two reverberant rooms and the only transmission between the rooms is through the element, the sound reduction index and the transmission factor of the element are defined to be [47]:

\[ R = 10 \log \left( \frac{\langle p_{inc}^2 \rangle}{\langle p_{trans}^2 \rangle} \right) \]  
\[ \tau = \frac{\langle p_{trans}^2 \rangle}{\langle p_{inc}^2 \rangle} \]  

where \( \langle p_{inc}^2 \rangle \) and \( \langle p_{trans}^2 \rangle \) are the spatially averaged mean square pressure in the source room and the receiver room, respectively, \( S \) is the area of the element and \( A \) is the equivalent absorption area in the receiver room. The transmission factor in eqn (A.2) may be rewritten in terms of the incident and transmitted sound power level such that:

\[ \tau = \frac{P_{trans}}{P_{inc}} \]  

where the sound power radiated from the element \( P_{trans} \) may be written in terms of the spatially averaged mean square velocity measured on its surface \( \langle v^2 \rangle \) such that [59]:

\[ P_{trans} = \rho_o c_o \langle v^2 \rangle S \sigma \]  

where \( \rho_o c_o \) is the characteristic impedance of air (413 kgm\(^2\)s\(^{-1}\) at 20°C, 101.3 kPa) and \( \sigma \) is the radiation efficiency of the element. The sound power incident on the element may be written as [59]:
Substituting eqns (A.4) and (A.5) into eqn (A.3) yields:

\[ \tau = \frac{4\rho c_2^2 A_2 \sigma(v^2)}{\langle p_{inc}^2 \rangle} \]  

(A.6)

which is a relationship between the transmission factor and the mean square velocity of the element due to an incident mean square sound pressure level.

Now consider the more general case of the sound transmission between two rooms shown in Figure A.1 which is inclusive of the direct transmission path (shown as path \( d \) in the figure) and \( n \) flanking paths, some of which are shown in the figure.

![Figure A.1: Flanking paths between two rooms](image)

From eqn (A.24), the sound power radiated from element \( j \) into Room 2 is:

\[ P_{2,j} = \rho c_\sigma \langle v_j^2 \rangle S_j \sigma_j \]  

(A.7)

The steady state sound power in Room 2 is described by [126]:

\[ P_2 = \frac{\langle p_2^2 \rangle A_2}{4 \rho c_\sigma} \]  

(A.8)
Equating eqns (A.7) and (A.8) and solving for the spatial average of the sound pressure level in Room 2 due to element $j$ yields:

$$\langle p_{2,j}^2 \rangle = \frac{4 \rho_s \bar{V}^2 \sigma_j \sigma_i}{A_2} \quad (A.9)$$

Substituting eqn (A.6) into eqn (A.9) yields:

$$\langle p_{2,j}^2 \rangle = \frac{\tau_i \sigma_i}{\tau_j \sigma_j} \quad (A.10)$$

The flanking transmission coefficient is defined as the ratio between the radiated sound power via a certain transmission path and the sound power incident on the common partition between the rooms such that [22]:

$$\tau_{ij} = \frac{P_{2,ij}}{P_{1,o}} = \frac{\langle p_{2,j}^2 \rangle A_2}{\langle p_i^2 \rangle} S_o \quad (A.11)$$

where $P_{1,o}$ is the incident sound power on the partition wall area $S_o$ and $P_{2,ij}$ is the power radiated into the receiving room from the transmission path under consideration. The partition $o$ is chosen as the reference area for all flanking transmission coefficient calculations so that they can be easily added together without further area corrections. This would not be the case if each flanking transmission coefficient used for a reference the incident sound power $P_{1,i}$ on the flanking surface $S_i$, for example [15]. In cases where the rooms do not adjoin, and thus have no common separation area, $S_o$ is set to a fixed reference value of 10 m$^2$ [127].

Substituting the sound pressure level in the receiving room from eqn (A.10) into eqn (A.11) yields:

$$\tau_{ij} = \frac{\langle v_i^2 \rangle \sigma_i \sigma_i}{\langle v_i^2 \rangle \sigma_i S_o} \quad (A.12)$$
which describes the flanking transmission coefficient between two elements in terms of the mean squared velocity, the radiation efficiencies and the areas of the elements. A vibration velocity factor $d_{ij}$ is introduced as the ratio of the time and spatially averaged mean square velocities on each element such that [100]:

$$d_{ij} = \frac{\langle v_i^2 \rangle}{\langle v_j^2 \rangle}$$

(A.13)

Substituting the vibration velocity factor into eqn (A.12) yields:

$$\tau_{ij} = \tau_i d_{ij} \frac{\sigma_j s_j}{\sigma_i s_o}$$

(A.14)

Likewise, the flanking transmission factor in the opposite direction between elements $j$ and $i$ is defined as:

$$\tau_{ji} = \tau_j d_{ji} \frac{\sigma_i s_i}{\sigma_j s_o}$$

(A.15)

Eqns (A.14) and (A.15) show the flanking transmission coefficients which Gerretsen showed in his 1979 [21] and 1986 [22] articles.

### A.2. Estimate of the Flanking Transmission Factor

Of the terms in eqns (A.14) and (A.15) the resonant radiation efficiencies of the elements are often the least known [97]. The values are often not readily available [45] and can be determined correctly only if the velocity amplitudes and the radiation efficiencies of all participating modes are known [41]. Therefore, it would be advantageous to remove the terms from the equations and Gerretsen used reciprocity between the calculated flanking transmission factor in each direction to achieve this goal [86].

Kihlman [44] found that for structures excited by airborne noise above the critical frequency and assuming a diffuse field that:
Therefore, the flanking transmission factor may be estimated by:

\[
\hat{\tau}_{ij} = \sqrt{\tau_{ij} \tau_{ji}} = \sqrt{\frac{\tau_{ij} d_{ij} d_{ji} S_i S_j}{S_0^2}} \tag{A.17}
\]

where \( \hat{\tau}_{ij} \) is the estimate of the measurand.

\[ A.3. \text{Estimate of the Flanking Sound Reduction Index} \]

The flanking sound reduction index \( \hat{R}_{ij} \) is defined as minus ten times the common logarithm of the flanking transmission coefficient such that:

\[
\hat{R}_{ij} = -10 \log \hat{\tau}_{ij} \tag{A.18}
\]

Substituting eqn (A.17) into eqn (A.18) yields the estimate of the flanking sound reduction index:

\[
\hat{R}_{ij} = \frac{R_i + R_j}{2} + \overline{D_{v,ij}} + 10 \log \left( \frac{S_0}{\sqrt{S_i S_j}} \right) \tag{A.19}
\]

where \( \overline{D_{v,ij}} \) is the direction averaged velocity level difference such that:

\[
\overline{D_{v,ij}} = \frac{D_{v,ij} + D_{v,ji}}{2} \tag{A.20}
\]

and \( D_{v,ij} \) and \( D_{v,ji} \) are the velocity level differences between elements \( i \) and \( j \) and \( j \) and \( i \), respectively. The velocity level difference is defined as:

\[
D_{v,ij} = 10 \log \frac{1}{d_{ij}} \tag{A.21}
\]
Eqn (A.19) is as Gerretsen derived in his 1986 paper for the prediction of flanking noise [22]. However, Gerretsen later reported that there are some problems with the equation including that the equation is only valid above the critical frequency, the sound reduction indices and the velocity level differences are situation dependent and the equation does not give the same result when applied to the opposite transmission direction [100].

A.4. Modified Flanking Sound Reduction Index

A.4.1. Resonant Component

It is possible to measure the sound reduction indices of the flanking elements in eqn (A.19) according to ISO 140 or ISO 15186. However the sound reduction index measured according to these standards includes contributions from both the resonant and the non-resonant components below the critical frequency [128]. However, only the resonant transmission is present in flanking paths that have a completely structure-borne path and thus the sound reduction index used in eqn (A.19) must only include the resonant component of the sound reduction index [21]. The use of a sound reduction index which includes the non-resonant components will tend to underestimate the predicted apparent sound reduction index [86]. For monolithic structures with critical frequencies at the low end of the frequency range of interest, the inclusion of the non-resonant components may result in a conservative estimation of the apparent sound reduction index since the sound reduction index above the critical frequency is dominated by the resonant contributions [110]. However, in the case of lightweight structures which may have critical frequency above the frequency range of interest, the majority of the measured sound reduction index will be due to the non-resonant component. The inclusion of the non-resonant component for these materials may lead to a considerable underestimation of the apparent sound reduction index [26].

A.4.2. In Situ Resonant Sound Reduction Index

If the resonant component of the sound reduction index is measured according to ISO140 as may be the case for heavy, monolithic constructions, the sound reduction index may be influenced by the boundary conditions of the test facility. The total loss factor and therefore the sound reduction index of an element is dependent not only on its material properties, but also on the constructions at the boundaries [129]. For example, Kihlman [44] writes that for
diffuse bending waves in plates, if the radiation losses are neglected the total loss factor can be written as:

\[
\eta_{\text{tot}} = \frac{2.2c_g}{13.8\pi S} \left[ \sum \gamma_i l_i + \frac{\pi}{2} S\eta k \right]
\]

(A.22)

where \(c_g\) is the group velocity of the waves, \(\gamma_i\) is the absorption factor of the \(i^{\text{th}}\) part of the boundary, \(l_i\) is the length of the \(i^{\text{th}}\) part of the boundary, \(\eta\) is the internal loss factor and \(k\) is the bending waves number. The boundaries of the element under test and therefore the losses at the edges differ between laboratory facilities. The boundaries even differ between facilities that comply with ISO 140-1 as demonstrated by a German round robin test in eleven laboratories [130]. The variation in boundaries is even larger in buildings. Therefore, the use of sound reduction index measured in the laboratory may reduce the accuracy of the predictions.

It is desirable to express the flanking sound reduction index in terms of measured data by taking into consideration the differences in the structural damping between the field and laboratory situations. Gerretsen [98] suggested that laboratory results could be made invariant and could be more easily be transferred to relevant field situations by converting the laboratory data for each element into \textit{in situ} data such that:

\[
R_{R,\text{situ}} = R_R - 10 \log \frac{T_{s,\text{situ}}}{T_{s,\text{lab}}}
\]

(A.23)

Where \(R_{R,\text{situ}}\) is the \textit{in situ} resonant component of the sound reduction index, \(R_R\) is the resonant component of the sound reduction index, \(T_{s,\text{situ}}\) is the structural reverberation time of the element in the actual field situation in seconds and \(T_{s,\text{lab}}\) is the structural reverberation time of the element in the laboratory in seconds. Annex E of ISO 140-3:1995 [47] gives guidance on the measurement of the reverberation time. However, EN12354-1 stipulates that the values of \(T_{s,\text{situ}}\) and \(T_{s,\text{lab}}\) should be considered to be equal and therefore no correction should be applied to the sound reduction index for certain elements including [23]:
Appendix A: Derivation of the EN12354 Method

- lightweight, double leaf elements such as timber framed or metal wall studs.
- elements with an internal loss factor greater than 0.03.
- elements which are much lighter than the surrounding structural elements (by a factor of at least three).
- elements which are not firmly connected to the surrounding structural elements.

Eqn (A.19) is rewritten to include the in situ resonant component of the sound reduction index such that:

\[
\bar{R}_{ij} = \frac{R_{ij,\text{situ}} + R_{ij,\text{situ}}}{2} + D_{\nu,ij} + 10 \log \left( \frac{s_0}{s_i s_j} \right) \quad (A.24)
\]

where \( R_{ij,\text{situ}} \) is the resonant, in situ sound reduction index of element \( j \). However, the direction averaged velocity level difference is still dependant on the boundary conditions during testing and therefore should be replaced by the invariant vibration reduction index.

A.5. Vibration Reduction Index

A.5.1. Power Transmitted between Elements

Consider the case of two elements connected on one edge by a structural joint as shown in Figure A.2:

![Figure A.2: Elements connected by a common boundary element.](image-url)
If element $i$ is excited with structure-borne excitation, there will be power transmitted between the elements across the boundary as shown in the block diagram in Figure A.3:

![Block diagram of the two coupled elements.](image)

Figure A.3: Block diagram of the two coupled elements.

The energy balance equation for element $j$ is:

$$ P_{lj} = P_{j,d} + P_{ji} $$  \hspace{1cm} (A.25)

where $P_{lj}$ is the power transferred from element $i$ to element $j$, $P_{j,d}$ is the dissipated power of element $j$ and $P_{ji}$ is the power transferred from element $j$ to element $i$. The power transferred from element $i$ to element $j$ may be written as [49]:

$$ P_{lj} = E_i \eta_{ij} \omega $$  \hspace{1cm} (A.26)

where $E_i$ is the current, total energy in element $i$, $\eta_{ij}$ is the coupling loss factor between the elements and $\omega$ is the frequency in radians per second. Eqn (A.25) may then be rewritten as:

$$ E_i \eta_{ij} \omega = E_j \eta_{j,\text{tot}} \omega $$  \hspace{1cm} (A.27)

where $E_j$ is the energy in element $j$, $\eta_{ij}$ is the coupling loss factor between elements $i$ and $j$ and $\eta_{j,\text{tot}}$ is the total loss factor of element $j$. The total loss factor may be found experimentally from the structural reverberation time such that [131]:

$$ \eta_{j,\text{tot}} = \frac{2.2}{T_{r,kj}} $$  \hspace{1cm} (A.28)
where $T_j$ is the structural reverberation time of element $j$.

The total energy in a plate may be expressed as twice the kinetic energy such that [15]:

$$E = \rho_s \langle v^2 \rangle S$$  \hspace{1cm} \text{(A.29)}

where $\rho_s$ is the mass per unit area of the plate. Substituting eqns (A.28) and (A.29) into eqn (A.27) and rewriting in terms of the coupling loss factor yields:

$$\eta_{ij} = \frac{\rho_{s,i} \langle v_i^2 \rangle S_i}{\rho_{s,j} \langle v_j^2 \rangle S_j \ T_{S,i}}$$  \hspace{1cm} \text{(A.30)}

Therefore, from eqn (A.26) the power transmitted between elements $i$ and $j$ may be written:

$$P_{ij} = E_i \frac{\rho_{s,j} \langle v_j^2 \rangle S_j \ 4.4\pi}{\rho_{s,i} \langle v_i^2 \rangle S_i \ T_{S,i}}$$  \hspace{1cm} \text{(A.31)}

### A.5.2. Transmission Coefficient

Rindel [114] describes a transmission coefficient between two plates to be:

$$\gamma = \frac{P_{\text{transmitted}}}{P_{\text{incident}}}$$  \hspace{1cm} \text{(A.32)}

Under the assumption of a diffuse field in element $i$, the incident power on the connection edge between the elements may be written as [114]:

$$P_{\text{inc}} = \frac{E_i c_{\text{eq}} l_{ij}}{\pi S_i}$$  \hspace{1cm} \text{(A.33)}

where $l_{ij}$ is the length of the boundary between the elements. Substituting eqns (A.31) and (A.33) into eqn (A.32) yields the transmission coefficient between elements $i$ and $j$ such that:
If it is assumed that the energy is transmitted by bending waves only such that \( f < \frac{f_s}{4} \) where \( f_s \) is the cross-over frequency, then the group speed will be twice the phase speed such that [114]:

\[
\gamma_{ij} = \frac{\rho_{s,i} \overline{\psi_i^2}}{\rho_{s,j} (\overline{\psi_j^2})} \frac{4,6\pi^2}{T_{s,j}^i a_{ij}}
\]  

(A.34)

\( \gamma_{ij} \) is the cross-over frequency, then the group speed will be twice the phase speed such that [114]:

\[
c_b = 2c_b
\]  

(A.35)

where \( c_b \) is the phase speed of the bending wave. However, Craik [42] notes that if additional waves types are to be considered eqn (A.35) would not hold.

An equivalent absorption length is introduced such that [97]:

\[
a_j = \frac{2,2\pi^2 s_j}{c_b f_{s,j}}
\]  

(A.36)

Substituting eqns (A.35) and (A.36) into eqn (A.34) yields:

\[
\gamma_{ij} = \frac{\rho_{s,i} \overline{\psi_i^2} c_b_j a_j}{\rho_{s,j} \overline{\psi_j^2} c_b_i a_{ij}}
\]  

(A.37)

Introducing the vibration velocity factor \( d_{ij} \) into eqn (A.37) and taking the inverse of the equation yields:

\[
\frac{1}{\gamma_{ij}} = \frac{1}{d_{ij}} \frac{\rho_{s,i} c_b_i a_{ij}}{\rho_{s,j} c_b_j a_j}
\]  

(A.38)

Rewriting eqn (A.38) in terms of logarithmic values yields:

\[
-10 \log \gamma_{ij} = D_{v,ij} + 10 \log \left( \frac{\rho_{s,i} c_b_i a_{ij}}{\rho_{s,j} c_b_j a_j} \right) + 10 \log \left( \frac{a_{ij}}{a_j} \right)
\]  

(A.39)
where term $\rho_s c_b$ is described by Cremer as the bending wave impedance [41].

Gerretsen defines the structural reduction index:

$$R_{s,ij} = -10 \log \gamma_{ij}$$

(A.40)

which he suggests is more invariant than the velocity level difference over the junction $D_{v,ij}$ alone because $D_{v,ij}$ has been found to yield different values for measurements made on different constructions using the same materials and construction techniques [132].

Because the velocity level difference in each direction may be different ($D_{v,ij} \neq D_{v,jl}$) [133], the term $D_{v,ij}$ in eqn (A.39) is replaced by the direction averaged velocity level difference $\overline{D_{v,ij}}$ which was defined in eqn (A.20) as the arithmetical average of the velocity level difference in each transmission direction. Therefore the direction averaged structural reduction index is written as:

$$\overline{R_s} = \frac{R_{s,ll}}{R_{s,li}} = \frac{\overline{D_{v,ij}} + 10 \log \left( \frac{l_{ij}}{\sqrt{a_i a_j}} \right)}{2}$$

(A.41)

Gerretsen refers to the term $\overline{R_s}$ as the vibration reduction index $K_{ij}$ such that [132]:

$$K_{ij} = \overline{D_{v,ij}} + 10 \log \left( \frac{l_{ij}}{\sqrt{a_i a_j}} \right)$$

(A.42)

Through the use of the vibration reduction index to describe the velocity level difference, the attenuation of the vibrational power flow through the junction is expressed as an invariant quantity [101]. The term is described as being applicable to both monolithic and lightweight constructions [30].

The vibration reduction index incorporates the equivalent absorption lengths which include the critical frequency. Although the critical frequency may be calculated for homogenous
materials, this may not be the case in general. Therefore, the critical frequency in the equivalent absorption lengths is replaced by a fixed reference frequency such that [132]:

\[ a_i = \frac{2.2\pi^2 s_i}{c_o T_{sk,i} \sqrt{\frac{f}{f_{ref}}}} \]  

(A.43)

The NORTEST method NT ACOU 090 [134] notes that the interpretation of \( a_i \) and \( a_j \) as the equivalent absorption lengths is only physically correct for diffuse vibration fields in homogeneous plates with critical frequency \( f_c = f_{ref} = 1000 \) Hz. Nightingale has pointed out that earlier articles by Gerretsen [97] used a value of \( f_{ref} = 500 \) Hz and there is some question as to whether the value was changed to better fit measured data.

ISO10848-1 notes that for lightweight, well-damped types of elements (for example, timber or metal-framed stud walls or wooden floors on beams) where the actual situation has no real influence on the sound reduction index and damping of the elements, the equivalent absorption area is taken as numerically equal to the surface area of the element \( S \) such that:

\[ a_i = \frac{s_i}{l_o} \]  

(A.44)

where the reference length \( l_o = 1 \) m. ISO18048-1 section 4.3.2 also notes that the vibration reduction index is often not relevant for lightweight elements because the vibration fields are not reverberant and the application of \( K_{ij} \) for lightweight elements in prediction models such as EN12354-1 has in several cases been shown to be inaccurate [25].

The use of eqn (A.44) for lightweight elements removes the structural reverberation time from the calculation of the equivalent absorption lengths. This removes the problem of determining which structural reverberation time to use in the calculations since elements such as double-leaf walls may have different structural reverberation times depending on which side of the element is excited or the response is measured.
**A.5.3. Measurement of the Vibration Reduction Index**

EN12354-1 notes that the vibration reduction index is to be determined in accordance with ISO10848-1. If the elements under test are double-leaf constructions then the velocity of the excited element is to be measured on the not excited side (“outside”) and the velocity of the receiving element is to be measured on the radiating side (“inside”) as shown in Figure A.4 [25].

![Figure A.4: Measurement of the response of the double-leaf elements.](image)

However, for homogeneous constructions, the side of the construction is irrelevant, but not so for double-leaf constructions.

Alternatively, catalogs of vibration reduction index values have been created based on material dimensions and practices [29]. The standard also notes that the values for $K_{ij}$ can be taken from Annex E of EN12354-1. However, Schneider [35] found that the $K_{ij}$ measured in the laboratory does not match the behavior of the $K_{ij}$ calculated from EN12354. Hopkins [34] found that measured values of $K_{ij}$ were lower than the predicted values, in part because the term may be biased by uneven velocity distributions and the term is prone to errors in the structural reverberation times.

The value of the vibration reduction index may also be predicted from the junction transmission coefficient such that:
However, Gerretsen writes that method can not be used for surfaces that are not homogeneous and isotropic (e.g. surfaces that have eccentric beams). Doing so may result in significant errors as it can no longer be assumed that in-plane waves are insignificant and reciprocity may not hold because the surfaces may actually behave as multiple subsystems [26].

Before it can be used, the vibration reduction index $K_{ij}$ still needs to be converted into an in situ term. Using a method similar to that used to correct the sound reduction index, the in situ mean difference in the normal surface velocity between the source and receiver surfaces connected to the junction $\overline{D_{v,ij}}$ is obtained from the vibration reduction index, taking into account the damping of the vibrations in the surfaces by the equivalent absorption length such that [26]:

\[
\overline{D_{v,ij,\text{situ}}} = K_{ij} - 10 \log \left[ \frac{l_{ij}}{\sqrt{a_{i,\text{situ}}a_{j,\text{situ}}}} \right] \quad (A.46)
\]

and

\[
a_{i,\text{situ}} = \frac{2\pi^2 S_i}{c_0 T_{s,i,\text{situ}}} \sqrt{\frac{f_{\text{ref}}}{f}} \quad (A.47)
\]

where $T_{s,i,\text{situ}}$ is the structural reverberation time of element $i$ in the actual field situation.

The value of the structural reverberation time of the actual field situation may be difficult to determine in practice. However, Gerretsen [132] states that for well damped constructions like light weight double elements, the dampening is only determined by the element itself, not by the surrounding constructions and is therefore invariant between laboratory and the different field situations. For well damped constructions, the value for the equivalent absorption length should be taken as the area of the construction or the area of the part of the construction over which the average velocity level is determined. Therefore, for the
Appendix A: Derivation of the EN12354 Method

following building materials, the equivalent absorption length is taken as numerically equal to the area of the element [23]:

- lightweight, double leaf elements such as timber framed or metal wall studs.
- elements with an internal loss factor greater than 0.03.
- elements which are much lighter than the surrounding structural elements (by a factor of at least three).
- elements which are not firmly connected to the surrounding structural elements.

Note that the vibration reduction index is directly related to the loss factor used in SEA such that [45]:

\[
K_{ij} = -10 \log \left[ \eta_{ij} \frac{\pi^2 s_i}{\omega_{ij} \sqrt{f_{ref} f_{c,j}}} \right]
\]  

(A.48)

where \( f_c \) is the critical frequency of the surface indicated.

A.6. EN12354 Estimate of the Flanking Sound Reduction Index

Substituting eqn (A.46) into eqn (A.24) yields the EN12354 estimate of the flanking sound reduction index:

\[
R_{ij,EN12354} = \frac{R_{R,i,situ} + R_{R,j,situ}}{2} + D_{v,i,j,situ} + 10 \log \left( \frac{S_o}{S_i} \right)
\]  

(A.49)

EN12354-1:2006 notes that eqn (A.49) may alternatively written as:

\[
R_{ij} = R_{R,i,situ} + D_{v,i,j,situ} + 10 \log \left( \frac{S_o}{S_i} \right) + 10 \log \left( \frac{\sigma_{i,situ}}{\sigma_{j,situ}} \right)
\]  

(A.50)

However, since the junction velocity level difference \( D_{v,i,j,situ} \) is not an invariant quantity and the radiation factors of the field situation, \( \sigma_{i,situ} \) and \( \sigma_{j,situ} \) are often not known, eqn (A.50) is less suited for predictions than eqn (A.49). Eqn (A.50) could however be used in existing field situations to estimate flanking transmission if appropriate data for the junction velocity
level difference and the radiation efficiencies of the elements for that field situation are available [23]. It may be possible to assume that $\sigma = 1$ for the case of massive structures with low critical frequencies. However, this may not be possible even for heavy constructions if the frequency range is extended below 100 Hz.

### A.7. EN12354 Estimate of the Flanking Transmission Factor

The EN12354 estimate of the flanking transmission factor is calculated from eqn (A.49) such that:

$$
\tau_{ij,EN12354} = \sqrt{\frac{\tau_{R,i,situ} \tau_{R,j,situ} d_{ij,situ} d_{ji,situ} S_i S_j}{S_0^2}}
$$

where $\tau_{R,i,situ}$ is the resonant component of the in situ transmission factor of element $i$ and $d_{ij,situ}$ is the in situ vibration velocity factor between elements $i$ and $j$.

### A.8. Calculation of the Apparent Sound Reduction Index

The EN12354 method determines the apparent transmission factor such that:

$$
\tau' = \tau_d + \sum_{f=1}^{n} \tau_f
$$

which includes the contribution of the direct transmission $\tau_d$ and from $n$ flanking paths $\tau_f$. For example, for the transmission paths shown in Figure A.5,
the apparent transmission factor is written as:

\[
\tau' = \tau_{D} + \tau_{F} + \tau_{Df} + \tau_{Fd}
\]  \hspace{1cm} (A.53)

The main assumption to this approach is that the transmission paths can be considered to be independent and that sound and vibration fields behave statistically [23].

The direction transmission factor \( \tau_d \) includes both the resonant and the non-resonant components of the transmission factor as measured according to ISO140 or ISO15186. The transmission factor of each flanking path is calculated from eqn (A.49) where:

\[
\tau_{ij} = 10^{\frac{R_{ij}}{10}}
\]  \hspace{1cm} (A.54)

Lastly, the apparent sound reduction index is determined from eqn (A.52) such that:

\[
R' = -10 \log \tau'
\]  \hspace{1cm} (A.55)

A single number rating may be calculated from eqn (A.55) in accordance with ISO 717-1.
Appendix B: Description of Panels for this Study

B.1. Single Panels

The single panels included both single-leaf and double-leaf constructions. The single panels were designed to be used for the measurement of the sound reduction index of the L-shaped panel elements.

A list of the single panels used for the measurement of the sound reduction index is shown in Table B.1.

<table>
<thead>
<tr>
<th>Number of Leaves</th>
<th>Leaf Material</th>
<th>Stud Material</th>
<th>Width (m)</th>
<th>Height (m)</th>
<th>Depth (m)</th>
<th>$\rho$ (kg/m$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single</td>
<td>Steel</td>
<td>-</td>
<td>0.950</td>
<td>1.546</td>
<td>0.002</td>
<td>12.51</td>
</tr>
<tr>
<td>Single</td>
<td>MDF</td>
<td>-</td>
<td>0.946</td>
<td>1.545</td>
<td>0.004</td>
<td>3.23</td>
</tr>
<tr>
<td>Single</td>
<td>Gypsum Board</td>
<td>-</td>
<td>0.949</td>
<td>1.548</td>
<td>0.010</td>
<td>6.78</td>
</tr>
<tr>
<td>Double</td>
<td>Gypsum Board</td>
<td>Metal</td>
<td>0.945</td>
<td>1.546</td>
<td>0.091</td>
<td>17.62</td>
</tr>
<tr>
<td>Double</td>
<td>MDF</td>
<td>Wood</td>
<td>0.943</td>
<td>1.545</td>
<td>0.078</td>
<td>12.62</td>
</tr>
</tbody>
</table>

Table B.1: List of the single panels made for the study.

The dimensions shown in the table were an average of ten measurements.

B.1.1. Panel Construction

The single panels were designed so that they were identical to each side of the L-shaped panels. For example the assembly of metal studs of the double-leaf gypsum board on steel studs panel is shown in Figure B.1.
Figure B.1: Drawing of the assembly of metal studs for the double-leaf gypsum board panel.

The figure shows three studs being used in on one side. Three studs were used so that the stud assembly was identical to the stud assembly of the double-leaf L-shaped panel.

B.1.2. Material Properties

The material properties of the single-leaf steel, MDF [135, 136] and gypsum board panels are shown in Table B.2.

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness (mm)</th>
<th>ρ (kg/m²)</th>
<th>$f_c$ (Hz)</th>
<th>E (Pa)</th>
<th>ν</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>1.6</td>
<td>12.51</td>
<td>8000</td>
<td>2.0E+11</td>
<td>0.28</td>
</tr>
<tr>
<td>MDF</td>
<td>4</td>
<td>3.2</td>
<td>8000</td>
<td>3.6E+09</td>
<td>0.2</td>
</tr>
<tr>
<td>Gypsum Board</td>
<td>10</td>
<td>6.78</td>
<td>8000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table B.2: Material properties of the panels.

The critical frequencies shown in the table represent the center frequency of the 1/3 octave band in which the critical frequency was located.
Appendix B: Description of Panels for this Study

B.2. L-Shaped Panels

L-shaped panels were built by joining two identical panel elements at a corner. Each of the panel elements was 1.548m x 0.948m so that either side of the L-panel could be inserted into the transmission loss rig at the University of Canterbury. A list of the panels is shown in Table B.3.

<table>
<thead>
<tr>
<th>Number of Leafs</th>
<th>Leaf Material</th>
<th>Stud Material</th>
<th>Construction</th>
<th>Depth (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single</td>
<td>2mm Steel</td>
<td>-</td>
<td>Panels spot welded to angle iron</td>
<td>0.002</td>
</tr>
<tr>
<td>Single</td>
<td>4mm MDF</td>
<td>-</td>
<td>Panels glued and screwed to a 1”x1” wood bar</td>
<td>0.004</td>
</tr>
<tr>
<td>Double</td>
<td>10mm gypsum board</td>
<td>Metal</td>
<td>Studs crimped together. Gypsum board screwed to studs.</td>
<td>0.091</td>
</tr>
<tr>
<td>Double</td>
<td>4mm MDF</td>
<td>Wood</td>
<td>Studs nailed together. MDF glued and screwed to studs.</td>
<td>0.078</td>
</tr>
</tbody>
</table>

Table B.3: List of L-shaped panel constructions used for the study.

Although single panels of steel and MDF are not commonly used in buildings, they were chosen due to the homogeneity of the material. It was important to remove sources of uncertainty in the measurements such as discontinuities that could affect the measurement results. Custom metal studs were fabricated to ensure that the studs did not include holes for electrical cables. Metal studs were thought to be preferable to wood studs for the purpose of this study since they are commonly used and wood is not homogeneous due to the presence of knots and grains.

Photographs of the steel and the MDF single L-shaped panels are shown in Figure B.2.
Appendix B: Description of Panels for this Study

Figure B.2: Steel and MDF L-shaped panels.

The photographs and Table B.3 show that the single panels included an additional element in the corner to which the panels were fastened. In the case of the steel L-panel, the panels were spot welded to a piece of angle iron. The single-leaf MDF L-panel was assembled by screwing and gluing the sheets of MDF to a 1” x 1” wood bar.

A drawing of the assembled, metal studs of the double-leaf gypsum board L-panel is shown in Figure B.3.

Figure B.3: Drawing of the assembled metal studs for the double-leaf gypsum board L-panel and a detailed view of the corner assembly where each stud is noted.
The figure shows that the corner of the assembly was created using four studs so that the two sides of the L-panel were symmetric.

**B.2.1. Measured L-Panel Properties**

The loss factors of the double-leaf MDF on wood studs L-panel and the double-leaf gypsum board on metal studs L-panel are compared in Figure B.4 and Figure B.5, respectively.

![Figure B.4: Comparison of the loss factors of each of the leaves of the double-leaf MDF L-panel.](image-url)
The figures show that elements 2 and 4 (the side of the L-panel mounted into the space between the reverberant and the semi-anechoic chamber) had higher loss factors than elements 6 and 9.

The velocity level difference measured on the double-leaf MDF on wood studs L-panel and the double-leaf gypsum board on metal studs L-panel are compared in and, respectively.
Figure B.6: Comparison of the velocity level difference between the leafs of double-leaf MDF L-panel.

Figure B.7: Comparison of the velocity level difference between the leafs of double-leaf gypsum board L-panel.

The figures show that the velocity level difference between leafs of the same panel was lower than between leafs on the other side of the junction as would be expected.
B.3. Damped Panels

Steel panels which were 1.6mm thick and had a mass per unit area of 12.51 kg/m² were used to make a series of panels with different damping properties. The critical frequency of the steel panels was calculated to be 7751 Hz and measurements confirmed that the critical frequency was in the 8000 Hz 1/3 octave band which is above the frequency range of interest. Two different types of damping materials were applied to the panels. The first material was Ultralon EVA 30 closed cell foam which was attached to the steel with an adhesive. The second material was a water based damping treatment (Vibradamp). The Vibradamp was poured into a mold constructed around the panel and allowed to cure. Once the Vibradamp hardened, the surface of the damping material was sanded to ensure a uniform thickness.

The properties of the damped panels are shown in Table B.4.

<table>
<thead>
<tr>
<th>Damping Material</th>
<th>Material Thickness</th>
<th>Width (m)</th>
<th>Height (m)</th>
<th>$\rho$ (kg/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>-</td>
<td>0.950</td>
<td>1.546</td>
<td>12.51</td>
</tr>
<tr>
<td>Ultralon</td>
<td>3mm</td>
<td>0.947</td>
<td>1.545</td>
<td>13.25</td>
</tr>
<tr>
<td></td>
<td>6mm</td>
<td>0.946</td>
<td>1.548</td>
<td>13.86</td>
</tr>
<tr>
<td></td>
<td>12mm</td>
<td>0.947</td>
<td>1.549</td>
<td>15.14</td>
</tr>
<tr>
<td>Vibradamp</td>
<td>3mm</td>
<td>0.945</td>
<td>1.549</td>
<td>15.85</td>
</tr>
<tr>
<td></td>
<td>6mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>12mm</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table B.4: List of the damped, steel panels used for the study.

The panels with 6mm and 12mm of Vibradamp shown in the table were made for the study but were not used. The addition of the damping material changed the mass of the panels so much that there was concern that the magnitude of the non-resonant velocity of the panels would be affected.
The panels were each mounted into an opening between a reverberant chamber and a semi-anechoic chamber. The structural reverberation time of the panel was measured *in situ* using an impact hammer and the integrated impulse response method described in ISO 3382 with backward integration of the square impulse response. The panel was impacted in fifteen locations and the surface velocity of the panel was simultaneously measured with four accelerometers which were moved to different, random locations for each impact.

The value which was measured was the total loss factor, inclusive of internal losses in the panel, edge losses at the boundaries and the losses due to radiation. The edge losses were kept consistent between the different panels by using a torque wrench to tighten the bolts which clamped the panel into the rig. However, the radiation losses due to mechanical energy being converted to sound could differ between panels due to the influence of the damping material. The total loss factors of the panels are compared in Figure B.8.

![Figure B.8: Comparison of total loss factors of the damped panels.](image-url)
The figure shows that all of the damping materials increased the total loss factor of the panels compared to the untreated, steel panel. The figure also shows that the magnitude of the total loss factor was dependant on the thickness and mass of the damping material. The Ultralon panels are shown to have a total loss factor which increased with the thickness of the damping layer at frequencies below the 1250 Hz 1/3 octave band. Above the 1250 Hz 1/3 octave band, the thickness of the Ultralon material is shown to have little impact on the total loss factor.
Appendix C: Testing Procedure for L-Shaped Panels

C.1. Introduction

The testing of the L-panels was done in accordance with ISO10848-1 [25] and ISO10848-2 [137]. The L-panels were tested by mounting one panel element into the opening between the reverberant and the semi-anechoic chambers. The tests on each L-panel typically took thirty hours to complete and were conducted during weekend nights to take advantage of the relative quiet of the laboratory. As much of the testing as possible was done at one time to avoid any possible changes in the conditions.

The steel L-panel was tested first and needed to be tested several times before the test procedure was worked out. The brace around the panel which extended into the semi-anechoic chamber and the additional sound insulation around the test rig were outcomes from the trial and errors with the steel L-panel.

A list of the equipment used for the testing is shown in Appendix N.

The following is a summary of how the measurements were setup to test according to the standards.

C.2. Preparation for Testing

The following steps were taken prior to the insertion of the L-panel into the test rig. Since the reverberant chamber would be sealed until the panel was removed (typically a week after being installed), care had to be taken to ensure that all of the equipment in the reverberant chamber was correctly set up and working.

- The microphones were mounted in tripods. Four of the microphones were located in random positions in the reverberant chamber. The fifth microphone was positioned a meter from the opening into the semi-anechoic chamber.
• The microphone cables were run through the pass through between the control room and the reverberant chamber
• All microphones were calibrated
• Additional sound insulation material was added to the semi-anechoic chamber. This was done by packing fiberglass insulation around the test rig and along the walls of the chamber.
• A brace is mounted into the test rig. The purpose of the brace was to hold the test panel in the rig flush with the outer lip of the rig as shown in Figure C.1.

![Figure C.1: Front and section view of the test rig with the brace inserted into the opening.](image)

Two different braces were made for the testing from 25mm thick MDF. All of the sides of the brace were lined with resilient material so that the brace fit snugly into the test rig as shown in Figure C.2.
In addition, resilient material was added between panel A and the brace to ensure that the side of panel A facing into the semi-anechoic chamber was flush with the edge of the test rig as shown in Figure C.3.

Figure C.3: Photograph showing the double-leaf gypsum board L-shaped panel mounted in the test rig. The side of panel A facing into the semi-anechoic chamber is flush with the side of the test rig as indicated in the photograph.
If a double-leaf L-panel was to be tested, then the mean square velocity on the side of the panel facing into the reverberation room would need to be measured. Since the room was not accessible during testing, the accelerometers needed to be glued to the panel surface prior to its insertion into the test rig. The procedure included the following steps:

- Ten coaxial cables for the accelerometers were run through the pass through.
- After being connected to the coaxial cables, the accelerometers were calibrated.
- A minimum of eight accelerometers were glued to the side of the panel that faced into the reverberant chamber. Care was taken to ensure that the accelerometers were not mounted within 10cm of the edges or discontinuities in the panels.
- The accelerometer cables were taped to the door frame in multiple locations as shown in Figure C.4.

![Figure C.4: Accelerometers attached to the reverberant side of panel A.](image)

The tape shown in the figure ensured that none of the cables came loose during the testing. Careful arrangement also ensured that none of the cables contacted the panel during testing. This was ensured by looking past a small opening between the panel and the test rig using a flashlight to ensure that none of the cables contacted the panel prior to the panel being inserted fully into the rig.
C.3. Mounting of the Panels into the Test Rig

The panel element inserted into the chamber was noted as panel A. The mounting of the L-panel included the following steps:

- A lift was used to position the panel at the correct height before it was pushed into the test rig. The panel typically needed to be rocked back and forth to get it into place. Care was taken to ensure that the accelerometer cables were not contacting the panel or pinched between the panel and the rig.
- With the lift still in place, metal bars were fixed into position around the panel as shown in Figure C.5.

![Figure C.5: Photograph of the metal bars and clamps to hold the panel into the rig.](image)

The bolts shown in the figure were tightened enough to hold the metal bars in place, but were not tightened fully. The metal bars shown in the figure have a rectangular profile. It was realized earlier in the study that a square profile would have been better to provide a uniform pressure along the edge of the panel, thus affecting the edge losses of the panel. Material for rectangular bars was chosen and purchased but a year on, the lab technician has yet to cut them.
- The lift was moved out of place and the final metal bar was put in place.
• A torque wrench was used to tighten the bolts of the braces. Starting in one corner, the bolts were all tightened to a preset torque starting with the bolts in the corners and then proceeding clockwise around the rig. The clockwise sequence of tightening continued for several passes around the bolts until all none of the bolts required further tightening.

• The sound source in the reverberant room was turned on. By making a check of the perimeter of the panel in the test rig by ear, one could tell if the panel was correctly mounted. If it sounded like noise was getting past the panel, the panel was removed and additional layers of resilient material were added to fill the gaps before reinstalling the panel.

C.4. Installation of a Frame around Panel B

A frame was installed around panel B. The frame was made of 25mm thick MDF as shown in Figure C.6.

![Frame attached to panel B.](image)

The figure shows that the frame extended beyond the surface of panel B. This was done to create an enclosure over which the sound intensity could be measured. On the opposite side of the panel, wooden bars were placed around the side of the panel as shown in Figure C.7.
Appendix C: Testing Procedure for L-Shaped Panels

Figure C.7: Bars placed around the perimeter of the outside of panel B.

C-clamps were used to hold the parts of the frame to the panel as shown in the figures. The purpose of the frame was to allow for the measurement of the intensity from panel B and to meet the requirement of ISO10848 that panel A and panel B be weakly coupled. Early testing with the steel L-shaped panel showed that if panel B was not supported by the frame, the two panels were strongly coupled according to the definition in ISO10848 section 4.3.3.

such that:

\[ D_{n,ij} \geq 3 - 10 \log \left( \frac{m_i f_{ej}}{m_j f_{el}} \right) \]  \hspace{1cm} (C.1)

where \( f_{ej} \) is the critical frequency of element \( j \) and \( m_i \) is the mass per unit area of element \( i \). If the condition in eqn (C.1) is not met, the standard recommends adding damping material to the edges of the panels or connecting them to other structures as has been done using the frame.
C.5. Panel Notation

EN10848-1 requires that for the testing of double-leaf panels, the response of the panel is to be measured on the “not excited side of the source panel” (noted as the “outside”) and the radiating side of the receiving panel (noted as the “inside”). For homogeneous constructions such as single-leaf panels, the side of the panel is irrelevant. Two different scenarios were tested for the double-leaf L-panels requiring that sides of panel A be noted as the “reverberant side” and the “semi-anechoic side” rather than the “inside” or “outside” to avoid confusion. The sides of panel B are noted as inside and outside.

C.6. Double-Leaf Panel Testing Scenarios

The testing of the panels would be done for two different scenarios as shown in Figure C.8 and Figure C.9.

![Diagram of testing scenarios](image)

**Figure C.8:** Testing scenario 1: panel A is excited on the semi-anechoic side and response is measured on the inside of panel B. The shaded squares represent the accelerometers.
Scenario 2 would allow the L-panel to be characterized for an excitation on the reverberant side of panel A. Therefore, if the reverberant side of panel A was excited by a diffuse sound field, the sound power radiated from the inside of panel B could be measured using the intensity method. Therefore, the sound power in the reverberation room and the sound power on the inside of panel B could be used to calculate the measured flanking sound reduction index to compare versus the predictions according to the EN12354 method.

The measurements made for scenario 1 could not be used for the measurement of the flanking sound reduction index, but since scenario 1 only required a few additional measurements, it was thought worthwhile to accumulate as much data as possible.

### C.7. Reverberation Time Measurements

The structural reverberation time was measured using an impact hammer and backwards integration of the response of four accelerometers. The backwards integration was done using the template in PULSE. The structural reverberation time measurements were repeated for a total of fifteen measurements. For each measurement, the impact location or the accelerometer locations were changed.
The reverberation time was measured on each panel in the case of single-leaf panels or on each side of the panel in the case of double-leaf panels. The measurements on the double-leaf panels included different combinations of which side was hit with an impact hammer and which side. For example, panel B was impacted on the inside and the response was measured both on the outside and on the inside.

C.8. Attachment of Electromagnetic Shaker

A Brüel & Kjær 4810 10N shaker was used for the measurements. The shaker was driven by a Brüel & Kjær 2718 amplifier and PULSE generated the pink noise. An impedance head was glued to the panel and a 2mm diameter stinger was connected between the impedance head and the shaker as shown in Figure C.10.

![Figure C.10: Connection between the 10N shaker and the panel including the stinger and the impedance head.](image)

For testing of the double-leaf panels under scenario 2, panel A needed to be excited on the reverberant side of the panel. To accomplish this, a hole was drilled in the semi-anechoic side of the panel to allow the stinger to be passed through as shown in Figure C.11.
The mean squared velocity of the semi-anechoic side of panel A due to the excitation of panel B was measured before and after the holes were drilled in the panel. The spatial average of the mean square velocity both with the holes and without were similar in value in each 1/3 octave band as shown in Figure C.12.
Therefore, the holes were concluded to have no noticeable effect on the mean square velocity.

**C.9. Attachment of Accelerometers**

The mean square velocity of the panels in the semi-anechoic chamber was measured in 10 to 15 positions per panel. The accelerometers were attached to the panels using beeswax. The mean square velocity was averaged for forty-five seconds per measurement for a total of three measurements. The repeat measurements allowed for an assessment of the quality of the measurement. If the standard deviation of the repeat measurements was greater than 0.1 dB, then the mounting of the accelerometer was checked and the measurements repeated. Typically, the measurements were repeatable, but if the beeswax was not adequately applied or if a connection was loose, the poor repeatability would highlight the problem.
C.10. Measurement of the Intensity Sound Reduction Index of the Flanking Element

The flanking intensity sound reduction index of the L-shaped panel was measured on panel B when panel A was exposed to an airborne noise source in the reverberant room as shown in Figure C.13.

![Figure C.13: Measurement of the flanking intensity sound reduction index.](image)

The intensity sound reduction index is defined in ISO15186-2 as:

\[ R_{IF} = L_{p1} - 6 - L_{inB} + 10 \log \left( \frac{S}{S_{Mj}} \right) \]  \hspace{1cm} (C.2)

where \( L_{p1} \) is the average sound pressure level in the reverberant room, \( L_{inB} \) is the average normal sound intensity level measured over the measurement surface for element B, \( S \) is the reference area and \( S_{Mj} \) is the total area of the measurement surface of element B.
Appendix D: Comparison of Reverberant Room Data

Measurements of the mean square pressure were made in three reverberant rooms as part of the study of the PDF’s of the EN12354 method. The rooms are summarized in Table D.1.

<table>
<thead>
<tr>
<th>Room Designation</th>
<th>Reverberant Room</th>
<th>Volume (m³)</th>
<th>Number of Measurement Positions</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>University of Canterbury</td>
<td>217</td>
<td>50</td>
</tr>
<tr>
<td>B</td>
<td>RMIT University</td>
<td>199.9</td>
<td>52</td>
</tr>
<tr>
<td>C</td>
<td>University of Adelaide</td>
<td>179.9</td>
<td>51</td>
</tr>
</tbody>
</table>

Table D.1: Summary of reverberant rooms included in this study.

The results of the $\chi^2$ goodness-of-fit test on the mean square pressure are shown in Table D.2 to Table D.4 and results of the $\chi^2$ goodness-of-fit test on the sound pressure level are shown in Table D.5 to Table D.7. Note that measurements were not made above the 5000 Hz 1/3 octave band in the University of Adelaide reverberant room.
### Table D.2: Summary of the results of the $\chi^2$ goodness-of-fit test for the University of Canterbury reverberant room. The volume of the room is $217\text{m}^3$.  

<table>
<thead>
<tr>
<th>1/3 Octave Band (Hz)</th>
<th>$\chi^2$ Goodness-of-Fit Test Results - University of Canterbury</th>
<th>Coefficient of Variation $c$</th>
<th>Shape Parameter $\tilde{a}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Gaussian</td>
<td>Log-Normal</td>
<td>Gamma</td>
</tr>
<tr>
<td></td>
<td>Result Probability</td>
<td>Result Probability</td>
<td>Result Probability</td>
</tr>
<tr>
<td>50</td>
<td>Reject 4.3%</td>
<td>Reject 0.0%</td>
<td>Reject 0.2%</td>
</tr>
<tr>
<td>63</td>
<td>Accept 17.2%</td>
<td>Accept 25.7%</td>
<td>Accept 26.9%</td>
</tr>
<tr>
<td>80</td>
<td>Accept 16.0%</td>
<td>Accept 22.3%</td>
<td>Accept 28.1%</td>
</tr>
<tr>
<td>100</td>
<td>Accept 9.9%</td>
<td>Accept 52.5%</td>
<td>Accept 46.0%</td>
</tr>
<tr>
<td>125</td>
<td>Accept 68.1%</td>
<td>Accept 30.4%</td>
<td>Accept 41.6%</td>
</tr>
<tr>
<td>160</td>
<td>Accept 41.3%</td>
<td>Accept 90.2%</td>
<td>Accept 69.4%</td>
</tr>
<tr>
<td>200</td>
<td>Reject 0.1%</td>
<td>Reject 0.0%</td>
<td>Reject 0.0%</td>
</tr>
<tr>
<td>250</td>
<td>Accept 33.8%</td>
<td>Accept 44.2%</td>
<td>Accept 46.1%</td>
</tr>
<tr>
<td>315</td>
<td>Reject 2.4%</td>
<td>Accept 6.5%</td>
<td>Accept 10.7%</td>
</tr>
<tr>
<td>400</td>
<td>Accept 76.4%</td>
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<td>Accept 46.1%</td>
</tr>
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<td>Accept 70.3%</td>
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<td>630</td>
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<td>Accept 54.3%</td>
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<tr>
<td>800</td>
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<td>Accept 31.6%</td>
</tr>
<tr>
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<td>Accept 43.8%</td>
</tr>
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<td>Accept 49.1%</td>
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</tr>
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</tr>
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<td>Accept 24.7%</td>
</tr>
<tr>
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</tr>
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<td>Reject 0.4%</td>
</tr>
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<td>Reject 2.4%</td>
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<tr>
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</tr>
<tr>
<td>10000</td>
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<td>Reject 0.7%</td>
<td>Reject 0.6%</td>
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</tbody>
</table>
Table D.3: Summary of the results of the $\chi^2$ goodness-of-fit test for the RMIT University reverberant room. The volume of the room is 199.9 m$^3$. 

<table>
<thead>
<tr>
<th>$1/3$ Octave Band (Hz)</th>
<th>$\chi^2$ Goodness-of-Fit Test Results - RMIT University Reverberant Room</th>
<th>Coefficient of Variation $\sigma$</th>
<th>Shape Parameter $\tilde{\sigma}$</th>
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<tbody>
<tr>
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<td>Gamma</td>
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<td>0.0%</td>
<td>Accept</td>
</tr>
<tr>
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<td>0.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>80</td>
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<td>0.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>100</td>
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<td>Accept</td>
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</tr>
<tr>
<td>1/3 Octave Band (Hz)</td>
<td>$\chi^2$ Goodness-of-Fit Test Results - University of Adelaide</td>
<td>Coefficient of Variation $C$</td>
<td>Shape Parameter $\tilde{\alpha}$</td>
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<td>----------------------</td>
<td>---------------------------------------------------------------</td>
<td>----------------------------</td>
<td>---------------------------------</td>
</tr>
<tr>
<td></td>
<td>Mean Square Sound Pressure $p^2$ (Pa$^2$)</td>
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<tr>
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<td>3150</td>
<td>Accept</td>
<td>0.25</td>
<td>Accept</td>
</tr>
<tr>
<td>4000</td>
<td>Reject</td>
<td>0.04</td>
<td>Reject</td>
</tr>
<tr>
<td>5000</td>
<td>Reject</td>
<td>0.03</td>
<td>Accept</td>
</tr>
<tr>
<td>6300</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>8000</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>10000</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table D.4: Summary of the results of the $\chi^2$ goodness-of-fit test for the University of Adelaide reverberant room. The volume of the room is 179.9 m$^3$. The measurements were made in the 50 Hz to the 5000 Hz 1/3 octave bands only.
<table>
<thead>
<tr>
<th>1/3 Octave Band (Hz)</th>
<th>$\chi^2$ Goodness-of Fit Test Results - University of Canterbury Sound Pressure Level $L_p$ (dB)</th>
<th>Coefficient of Variation $\hat{\sigma}$</th>
<th>Shape Parameter $\hat{\gamma}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Gaussian</td>
<td>Log-Normal</td>
<td>Gamma</td>
</tr>
<tr>
<td>50</td>
<td>Reject</td>
<td>0.0%</td>
<td>Reject</td>
</tr>
<tr>
<td>63</td>
<td>Accept</td>
<td>25.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>80</td>
<td>Accept</td>
<td>22.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>100</td>
<td>Accept</td>
<td>52.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>125</td>
<td>Accept</td>
<td>30.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>160</td>
<td>Accept</td>
<td>86.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>200</td>
<td>Reject</td>
<td>0.0%</td>
<td>Reject</td>
</tr>
<tr>
<td>250</td>
<td>Accept</td>
<td>44.3%</td>
<td>Accept</td>
</tr>
<tr>
<td>315</td>
<td>Accept</td>
<td>6.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>400</td>
<td>Accept</td>
<td>31.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>500</td>
<td>Accept</td>
<td>59.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>630</td>
<td>Accept</td>
<td>54.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>800</td>
<td>Accept</td>
<td>31.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>1000</td>
<td>Accept</td>
<td>44.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>1250</td>
<td>Accept</td>
<td>60.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>1600</td>
<td>Accept</td>
<td>59.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>2000</td>
<td>Accept</td>
<td>37.7%</td>
<td>Accept</td>
</tr>
<tr>
<td>2500</td>
<td>Accept</td>
<td>21.1%</td>
<td>Accept</td>
</tr>
<tr>
<td>3150</td>
<td>Accept</td>
<td>49.3%</td>
<td>Accept</td>
</tr>
<tr>
<td>4000</td>
<td>Accept</td>
<td>7.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>5000</td>
<td>Reject</td>
<td>0.3%</td>
<td>Reject</td>
</tr>
<tr>
<td>6300</td>
<td>Reject</td>
<td>2.4%</td>
<td>Reject</td>
</tr>
<tr>
<td>8000</td>
<td>Accept</td>
<td>20.9%</td>
<td>Accept</td>
</tr>
<tr>
<td>10000</td>
<td>Reject</td>
<td>0.7%</td>
<td>Reject</td>
</tr>
</tbody>
</table>

Table D.5: Summary of the results of the $\chi^2$ goodness-of-fit test for the University of Canterbury reverberant room. The volume of the room is 217m$^3$. 

### Table D.6: Summary of the results of the $\chi^2$ goodness-of-fit test for the RMIT University reverberant room. The volume of the room is 199.9 m$^3$. 

<table>
<thead>
<tr>
<th>1/3 Octave Band (Hz)</th>
<th>$\chi^2$ Goodness-of Fit Test Results - RMIT University Reverberant Room</th>
<th>Coefficient of Variation $\gamma$</th>
<th>Shape Parameter $\tilde{\alpha}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Gaussian</td>
<td>Probability</td>
<td>Log-Normal</td>
</tr>
<tr>
<td>50</td>
<td>Accept</td>
<td>37.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>63</td>
<td>Accept</td>
<td>31.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>80</td>
<td>Accept</td>
<td>5.1%</td>
<td>Accept</td>
</tr>
<tr>
<td>100</td>
<td>Accept</td>
<td>50.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>125</td>
<td>Accept</td>
<td>74.3%</td>
<td>Accept</td>
</tr>
<tr>
<td>160</td>
<td>Accept</td>
<td>33.1%</td>
<td>Accept</td>
</tr>
<tr>
<td>200</td>
<td>Accept</td>
<td>84.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>250</td>
<td>Accept</td>
<td>35.3%</td>
<td>Accept</td>
</tr>
<tr>
<td>315</td>
<td>Accept</td>
<td>55.3%</td>
<td>Accept</td>
</tr>
<tr>
<td>400</td>
<td>Accept</td>
<td>9.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>500</td>
<td>Accept</td>
<td>80.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>630</td>
<td>Reject</td>
<td>4.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>800</td>
<td>Accept</td>
<td>83.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>1000</td>
<td>Accept</td>
<td>98.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>1250</td>
<td>Accept</td>
<td>77.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>1600</td>
<td>Accept</td>
<td>62.9%</td>
<td>Accept</td>
</tr>
<tr>
<td>2000</td>
<td>Reject</td>
<td>0.3%</td>
<td>Reject</td>
</tr>
<tr>
<td>2500</td>
<td>Accept</td>
<td>7.7%</td>
<td>Accept</td>
</tr>
<tr>
<td>3150</td>
<td>Accept</td>
<td>5.2%</td>
<td>Accept</td>
</tr>
<tr>
<td>4000</td>
<td>Reject</td>
<td>3.8%</td>
<td>Reject</td>
</tr>
<tr>
<td>5000</td>
<td>Accept</td>
<td>34.8%</td>
<td>Accept</td>
</tr>
<tr>
<td>6300</td>
<td>Reject</td>
<td>3.9%</td>
<td>Reject</td>
</tr>
<tr>
<td>8000</td>
<td>Accept</td>
<td>9.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>10000</td>
<td>Reject</td>
<td>0.0%</td>
<td>Reject</td>
</tr>
</tbody>
</table>
### Appendix D: Comparison of Reverberant Room Data

Table D.7: Summary of the results of the $\chi^2$ goodness-of-fit test for the University of Adelaide reverberant room. The volume of the room is 179.9m$^3$. The measurements were made in the 50 Hz to the 5000 Hz 1/3 octave bands only.

<table>
<thead>
<tr>
<th>1/3 Octave Band (Hz)</th>
<th>Gaussian</th>
<th>Log-Normal</th>
<th>Gamma</th>
<th>Coefficient of Variation $\gamma$</th>
<th>Shape Parameter $\alpha$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Result</td>
<td>Probability</td>
<td>Result</td>
<td>Probability</td>
<td>Result</td>
</tr>
<tr>
<td>50</td>
<td>Reject</td>
<td>0.2%</td>
<td>Reject</td>
<td>0.3%</td>
<td>Reject</td>
</tr>
<tr>
<td>63</td>
<td>Reject</td>
<td>0.0%</td>
<td>Reject</td>
<td>0.2%</td>
<td>Reject</td>
</tr>
<tr>
<td>80</td>
<td>Reject</td>
<td>0.0%</td>
<td>Reject</td>
<td>0.0%</td>
<td>Reject</td>
</tr>
<tr>
<td>100</td>
<td>Reject</td>
<td>0.7%</td>
<td>Reject</td>
<td>0.3%</td>
<td>Reject</td>
</tr>
<tr>
<td>125</td>
<td>Accept</td>
<td>52.8%</td>
<td>Accept</td>
<td>52.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>160</td>
<td>Reject</td>
<td>0.3%</td>
<td>Reject</td>
<td>0.3%</td>
<td>Reject</td>
</tr>
<tr>
<td>200</td>
<td>Accept</td>
<td>35.9%</td>
<td>Accept</td>
<td>35.9%</td>
<td>Accept</td>
</tr>
<tr>
<td>250</td>
<td>Accept</td>
<td>15.4%</td>
<td>Accept</td>
<td>15.4%</td>
<td>Accept</td>
</tr>
<tr>
<td>315</td>
<td>Accept</td>
<td>31.9%</td>
<td>Accept</td>
<td>19.6%</td>
<td>Accept</td>
</tr>
<tr>
<td>400</td>
<td>Accept</td>
<td>23.0%</td>
<td>Accept</td>
<td>22.9%</td>
<td>Accept</td>
</tr>
<tr>
<td>500</td>
<td>Accept</td>
<td>11.8%</td>
<td>Accept</td>
<td>11.8%</td>
<td>Accept</td>
</tr>
<tr>
<td>630</td>
<td>Reject</td>
<td>0.4%</td>
<td>Reject</td>
<td>3.4%</td>
<td>Reject</td>
</tr>
<tr>
<td>800</td>
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<td>54.6%</td>
<td>Accept</td>
<td>54.7%</td>
<td>Accept</td>
</tr>
<tr>
<td>1000</td>
<td>Reject</td>
<td>2.4%</td>
<td>Reject</td>
<td>2.4%</td>
<td>Reject</td>
</tr>
<tr>
<td>1250</td>
<td>Reject</td>
<td>0.8%</td>
<td>Reject</td>
<td>0.8%</td>
<td>Reject</td>
</tr>
<tr>
<td>1600</td>
<td>Reject</td>
<td>3.4%</td>
<td>Reject</td>
<td>3.4%</td>
<td>Reject</td>
</tr>
<tr>
<td>2000</td>
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<td>9.7%</td>
<td>Accept</td>
<td>9.5%</td>
<td>Accept</td>
</tr>
<tr>
<td>2500</td>
<td>Reject</td>
<td>0.2%</td>
<td>Reject</td>
<td>0.2%</td>
<td>Reject</td>
</tr>
<tr>
<td>3150</td>
<td>Accept</td>
<td>46.1%</td>
<td>Accept</td>
<td>46.0%</td>
<td>Accept</td>
</tr>
<tr>
<td>4000</td>
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<td>Reject</td>
<td>0.0%</td>
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</tr>
<tr>
<td>5000</td>
<td>Accept</td>
<td>10.4%</td>
<td>Accept</td>
<td>36.7%</td>
<td>Accept</td>
</tr>
<tr>
<td>6300</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
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<tr>
<td>8000</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>10000</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
Appendix E: Generation of Random Observations

E.1. Introduction

Monte Carlo simulations were used in this study to determine the probability density functions of the terms defined by the EN12354 method, to determine the uncertainty of the terms and to test different hypothesis. In each case, the random observations were synthesized in MATLAB using the Wichmann-Hill algorithm [138-140] to generate pseudo-random values uniformly distributed in the interval between 0 and 1. The Wichmann-Hill algorithm has proven to be a robust pseudo-random number generator by passing the Big Crush battery of statistical tests [141]. The synthesized values from the uniform distribution could then be transformed into values from other distributions as detailed in the following sections.

E.2. Generation of Random Values from a Gaussian Distribution

E.2.1. Standard Gaussian Distribution

The Box-Muller method [142, 143] was used to transform uniformly distributed random numbers on the interval (0,1) into the standard Gaussian distribution.

E.2.2. Scaled and Shifted Gaussian Distribution

Observations from a scaled and shifted Gaussian distribution with mean \( \mu \) and standard deviation \( \sigma \) could be obtained from the standard Gaussian distribution such that [143]:

\[
X_{\text{Gaussian}} = sR_N + \mu
\]  
(E.1)

where \( X_{\text{Gaussian}} \) is a random value from the scaled and shifted Gaussian distribution and \( R_N \) is a random value from the standard Gaussian distribution.

E.3. Generation of Random Values from a Log-Normal Distribution

Hahn [74] showed that values from a log-normal distribution may be obtained from random values of the standard Gaussian distribution such that:
$X_{LN} = \exp(s_Y R_N + \mu_Y)$ \hspace{1cm} (E.2)

where $X_{LN}$ is a random value from the log-normal distribution, $R_N$ is a random value from the standard Gaussian distribution and $\mu_Y$ and $s_Y$ are the mean and the standard deviation of the log shifted values $Y = \ln(X)$ where $X$ is from a log-normal distribution.

**E.4. Generation of Random Values from a Gamma Distribution**

Hahn [74] showed that values from a gamma distribution may be obtained from random values of the uniform distribution over the interval (0,1) such that:

$$X_{\text{Gamma}} = -\frac{1}{\hat{\lambda}} \left[ \sum_{k=1}^{n} \ln(1 - R_{Uk}) \right]$$ \hspace{1cm} (E.3)

where $X_{\text{Gamma}}$ is a random value from the gamma distribution, $\lambda$ is the scale parameter which may be estimated as:

$$\hat{\lambda} = \frac{\bar{X}}{S^2}$$ \hspace{1cm} (E.4)

where $\bar{X}$ and $S$ are the mean and the standard deviation of the gamma distribution, $\alpha$ is the shape parameter which may be estimated as:

$$\hat{\alpha} = \left( \frac{\bar{X}}{S} \right)^2$$ \hspace{1cm} (E.5)

and $R_{Uk}$ are random values from the uniform distribution over the interval (0,1). Note that $\alpha$ number values from the uniform distribution are required for each synthesized gamma value.
Appendix F: Explanation of the Uncertainty Analysis

F.1. Summary of the GUM Method

F.1.1. Propagation of Uncertainties

The Guide to the Expression of Uncertainty in Measurement (GUM) [64] implements the law of propagation of uncertainty and the application of the central limit theorem to determine the propagated uncertainty of a clearly defined measurand. GUM describes the relationship $f$ between the estimate of the measurand $y$ and $N$ input estimates $x_1, x_2, \ldots, x_n$ such that:

$$y = f(x_1, x_2, \ldots, x_n)$$  \hspace{1cm} (F.1)

The input quantities upon which the output quantity depends may themselves be viewed as measurands and may themselves depend on other quantities. The uncertainty of the estimate of the measurand may be determined by appropriately combining the uncertainties of the input estimates according to the law of propagation of uncertainty such that:

$$u^2(y) = \sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} \right)^2 u^2(x_i) + 2 \sum_{i=1}^{N-1} \sum_{j=i+1}^{N} \frac{\partial f}{\partial x_i} \frac{\partial f}{\partial x_j} u(x_i)u(x_j) r(x_i, x_j)$$ \hspace{1cm} (F.2)

where $u(y)$ is the uncertainty of the estimate of the measureand, $c_i = \frac{\partial f}{\partial x_i}$ is the sensitivity coefficient of the input term $x_i$, $u(x_i)$ is the uncertainty of $x_i$ and $r(x_i, x_j)$ is the correlation coefficient between $x_i$ and $x_j$. If the input terms are independent and uncorrelated, then eqn (F.2) may be reduced to:

$$u^2(y) = \sum_{i=1}^{N} c_i^2 u^2(x_i)$$ \hspace{1cm} (F.3)

The uncertainty $u^2(y)$ can therefore be viewed as a sum of the uncertainty and sensitivity coefficients of the input estimates.
F.1.2. Categories of Uncertainty

GUM categorizes uncertainty as either Type A or Type B uncertainty. The types of uncertainty do not differ in essence, but are categorized separately to indicate that they are evaluated in different ways [96]. Both types of evaluation are based on probability distributions and the uncertainty components resulting from either type are quantified by variances or standard deviations. Type A and Type B uncertainty are treated in the same manner when combining the uncertainties according to eqn (F.2) or eqn (F.3).

Type A Uncertainty

Type A uncertainty indicates a method of evaluation of uncertainty by the statistical analysis of a series of observations. Type A standard uncertainty is obtained from the probability density function of $n$ repeated observations of the measurand $y$. For a measurand with a Gaussian PDF, the best estimate of the uncertainty of the mean is calculated from the unbiased estimate of the variance of the repeated observations such that [96]:

$$u_{TypeA}(\bar{y}) = \frac{s^2_y}{\sqrt{n}}$$

where $u_{TypeA}$ is the Type A standard uncertainty of the mean $\bar{y}$, and $s^2_y$ is the unbiased estimate of the variance. Therefore, the greater the variance of the individual observations, the greater the Type A uncertainty of the measurand. The effective degrees of freedom associated $u_{TypeA}(\bar{y})$ is given by $\nu_{eff} = n - 1$.

Type B Uncertainty

Type B uncertainty indicates a method of evaluation of uncertainty of the measurand by means other than the statistical analysis of a series of observations. Type B uncertainty is obtained from an assumed PDF evaluated by scientific judgment based on all of the available information such as previous measurement data, experience or general knowledge, the manufacturer’s specifications, calibration data or reference data from handbooks. If a quoted uncertainty is given as a level of confidence, it may be assumed that a Gaussian distribution was used to calculate the quoted uncertainty to recover the standard uncertainty [64]. In the
case of a standard uncertainty obtained from a Type B evaluation from an a priori probability distribution, the effective degrees of freedom may be assumed to be $u_{eff} = \infty$ [64].

**Combining Types of Uncertainty**

The reported combined uncertainty of a measurand is likely to include both Type A and Type B components. For example, the uncertainty of an average of $n$ measurements of the length of an element may include the Type A uncertainty due to the variance of the repeated measurements as well as the type B uncertainty of the resolution of the measurement device.

The combined uncertainty becomes wholly Type B uncertainty when subsequent use is made of it [96]. For example, if the measurand $x_1$ is calculated from an average of inputs, it would have Type A uncertainty. But, if the measurand $y$ can be described in terms of $x_1$ such that

$$y = f(x_1)$$

then the uncertainty $u(x_1)$ is described as being Type B uncertainty in the calculation of the uncertainty $u(y)$ since $u(x_1)$ is based on prior knowledge.

**F.1.3. Degrees of Freedom**

The degrees of freedom associated with the combined uncertainty may be calculated according to the Welch-Satterthwaite formula such that [96]:

$$u_{eff}(y) = \frac{u_s(y)}{\sum_{i=1}^{n} \frac{c_i^2 u_s(x_i)}{\nu_{eff}(x_i)}}$$  \hspace{1cm} (F.5)

where $u_{eff}(x_i)$ are the effective degrees of freedom of input term $x_i$. 
F.1.4. Expanded Uncertainty

It is often necessary to state the uncertainty that defines an interval about the measurand that may be expected to encompass a large, specified fraction (e.g. 95%) of the distribution of values that could be reasonably attributed to the measurand [63]. The GUM method determines expanded uncertainty $U$ from a coverage factor $k$ and from the uncertainty of the measurand $y$ such that:

$$U = ku(y)$$  \hspace{1cm} (F.6)

The coverage interval may be expressed as [64]:

$$y - U \leq y \leq y + U$$  \hspace{1cm} (F.7)

The coverage factor is determined from the number of degrees of freedom associated with the uncertainty and from the desired level of confidence. For this study the coverage factor will be calculated for a 95% confidence. In the determination of the coverage factor, the GUM method assumes a t-distribution for PDF of the measurand. A t-distribution is virtually identical to a Gaussian distribution when a sufficiently large number of observations are included in the calculation (such as more than 20 measurements [66]). However, as fewer observations are included, the shape of the t distribution becomes flatter and broader, reflecting that fewer observations mean that less is known about the population [144]. When the number of observations is small, the coverage factor can be very large [145].

F.2. Basis for Uncertainty Calculations in this Study According to GUM

F.2.1. Averaging

The uncertainty of an average value $y$ where $y = \frac{\sum_{i=1}^{n} x_i}{n}$ includes not only the uncertainty of the input quantities, $x_1, x_2, \ldots, x_n$ (Type B uncertainty) but also the uncertainty due to the variance of the values being averaged (Type A uncertainty). That is,

$$u^2(y) = \frac{s^2}{n} + \left(\frac{1}{n}\right)^2 \sum_{i=1}^{n} u^2(x_i)$$  \hspace{1cm} (F.8)
where $s^2$ is the variance between the values of the inputs. The inclusion of both the Type A and the Type B uncertainty follows from prior studies by Bloembergen [146] and Desenfant [68].

For example, consider the case of the average of several values of $x$ as shown in Figure F.1 and Figure F.2.

![Figure F.1: Example 1 of an uncertainty calculation only considering the Type B uncertainty.](image)

![Figure F.2: Example 2 of an uncertainty calculation only considering the Type B uncertainty.](image)

If only the Type B uncertainty is considered, the uncertainty of $y_1$ is the same as the uncertainty of $y_2$ even though the second example shows more variance between the means.
However, if the Type A uncertainty is considered, the uncertainty of $y$ will be increase as the variance in $x$ increases as shown in Figure F.3.

![Figure F.3](image)

Figure F.3: Example 3 of an uncertainty calculation which includes both the Type A and the Type B uncertainty.

Therefore, the inclusion of both types of uncertainty gives a more realistic indication of the potential spread of the values than the inclusion of Type B uncertainty only. In this study, the calculation of the uncertainty of averaged terms includes both the Type A and the Type B uncertainty.

**F.2.2. Correlated Type B Uncertainty of Spatial Averages**

The calculations of the uncertainty of the EN12354 method presented in this study were written under the assumption that the correlated uncertainty of the input terms could be neglected. While repeat measurements of the mean square pressure and the mean square velocity have been shown to have correlation coefficients as high as ±1, it has been found that the uncertainty due to the correlated terms is negligible if best practices are followed during the measurements. Best practices for the velocity measurements would include proper mounting of the accelerometers, calibration and properly securing the cables.

Measurements of the mean square velocity for this study included a minimum of three measurements of forty-five seconds. If the standard deviation of repeatability for any one accelerometer was greater than 0.1 dB, the accelerometer was checked because the thin film...
of beeswax between the accelerometer and the element was most likely not adequately holding the accelerometer in place. Therefore, the standard deviation of repeatability of the repeat measurements at the same measurement position was very small.

For example, the velocity was measured at twelve positions on a double-leaf gypsum board panel. The measurements were repeated three times each and the correlation coefficients between the measurement positions are shown in Figure F.4.

![Figure F.4: Correlation coefficients calculated for twelve measurement positions.](image)

The figure shows that several correlation coefficients were as high as ±1 indicating in these cases, the measurements were perfectly mutually correlated [96]. A comparison of the uncertainty calculated with and without the inclusion of the correlated terms is shown in Figure F.5.
Figure F.5: Comparison of the Type B uncertainty when correlated terms are ignored and when they are included in the uncertainty calculation.

The figure shows that ignoring the correlated uncertainty can have a large effect on the Type B uncertainty. In some 1/3 octave bands, the uncertainty more than doubled when the correlated terms were included. In other 1/3 octave bands such as 1000 Hz, the uncertainty decreased significantly when the correlated terms were included.

However, if the repeat measurements in each measurement position were made as part of a measurement of a spatial average, then the uncertainty of the repeat measurements become Type B uncertainty which was much smaller than the Type A uncertainty of the spatial average. For example, the Type A and Type B uncertainty from a number of observations of the mean square velocity on a panel are compared in Figure F.6.
Figure F.6: Comparison between the Type A and the Type B uncertainty for the velocity measurements. The figure shows the Type B uncertainty to represent a small fraction of the Type A uncertainty.

The figure shows that the Type B uncertainty was much less than the Type A uncertainty and therefore could be neglected. Therefore the calculation of the correlated uncertainty for the repeat measurements could be neglected.
Appendix G: Derivation of Correction Factors

G.1. Introduction

In Chapter 5, a correction factor $C$ was described which may be applied to the total, measured sound reduction index to obtain the resonant component. The correction factor was written as:

$$C = 1 + \frac{\tau_{NR}}{\tau_R} \quad (G.1)$$

where $\tau_{NR}$ is the non-resonant component of the transmission factor and $\tau_R$ is the resonant component of the transmission factor. This appendix shows the derivation of two of the correction factors described in this study.

G.2. Proposed Correction Factor Based on Mean Square Velocity

The sound power radiated from a vibrating panel may be written as [17]:

$$P = S \rho_o c_o (\langle v_R^2 \rangle \sigma_R + \langle v_{NR}^2 \rangle \sigma_{NR}) \quad (G.2)$$

where $S$ is the surface area of the panel and $\langle v_R^2 \rangle$ and $\langle v_{NR}^2 \rangle$ are the resonant and non-resonant components of the mean square velocity, respectively. The transmission factor may therefore be written as:

$$\tau = \frac{\text{Transmitted Power}}{\text{Incident Power}} = \frac{S \rho_o c_o (\langle v_R^2 \rangle \sigma_R + \langle v_{NR}^2 \rangle \sigma_{NR})}{\text{Incident Power}} \quad (G.3)$$

If the non-resonant component of the transmission factor is divided by the resonant component, the result is:

$$\frac{\tau_{NR}}{\tau_R} = \frac{\langle v_{NR}^2 \rangle \sigma_{NR}}{\langle v_R^2 \rangle \sigma_R} \quad (G.4)$$

Substituting eqn (G.4) into eqn (7.6) yields:
Appendix G: Derivation of Correction Factors

\[ C_{\text{proposed}} = 1 + \frac{\sigma_{\text{NR}} \langle v_{\text{NR}}^2 \rangle}{\sigma_{\text{R}} \langle v_{\text{R}}^2 \rangle} \]  \hspace{1cm} (G.5)

which is the proposed correction factor based on the components of the mean square velocity.

The radiation efficiencies in eqn (G.5) may were determined from Annex B of EN12354.

G.3. Correction Factor Appendix G

The total transmission factor of a panel may be predicted as [17]:

\[ \tau_T = \left( \frac{2 \rho_0 c_0}{2\pi f \rho_s} \right)^2 \left( 2 \sigma_{\text{NR}} + \frac{f_c \pi \sigma_{\text{R}}^3}{2 f \eta_{\text{tot}}} \right) \]  \hspace{1cm} (G.6)

This equation may be separated into its non-resonant and resonant components such that:

\[ \tau_{\text{NR}} = \left( \frac{2 \rho_0 c_0}{2\pi f \rho_s} \right)^2 (2 \sigma_{\text{NR}}) \]  \hspace{1cm} (G.7)

\[ \tau_{\text{R}} = \left( \frac{2 \rho_0 c_0}{2\pi f \rho_s} \right)^2 \left( \frac{f_c \pi \sigma_{\text{R}}^3}{2 f \eta_{\text{tot}}} \right) \]  \hspace{1cm} (G.8)

Dividing the non-resonant component of the transmission factor by the resonant component of the transmission factor yields:

\[ \frac{\tau_{\text{NR}}}{\tau_{\text{R}}} = \frac{(2 \sigma_{\text{NR}})}{\left( \frac{f_c \pi \sigma_{\text{R}}^3}{2 f \eta_{\text{tot}}} \right)} = \frac{4 f \eta_{\text{tot}} \sigma_{\text{NR}}}{\pi f_c \sigma_{\text{R}}^3} \]  \hspace{1cm} (G.9)

Substituting eqn (G.9) into eqn (7.6) yields:

\[ C_{\text{Appendix G}} = 1 + \frac{4 f \eta_{\text{tot}} \sigma_{\text{NR}}}{\pi f_c \sigma_{\text{R}}^3} \]  \hspace{1cm} (G.10)
Alternatively, eqn (G.10) may be derived from the theoretical equations for the resonant and non-resonant components of the mean square velocity which are found using both SEA [59] and the work of Heckl [1]. The resonant component of the mean square velocity is:

\[
\langle \nu^2_R \rangle = \frac{\langle p^2 \rangle f_c \sigma_R}{8\pi f^2 \eta_{tot} \rho_T^2}
\]  
(G.11)

where \( \langle p^2 \rangle \) is the time and spatially averaged mean square pressure in the source room. The non resonant component is:

\[
\langle \nu^2_{NR} \rangle = \frac{\langle p^2 \rangle}{2\pi f^2 \rho_T^2}
\]  
(G.12)

Substituting eqns (G.11) and (G.12) into eqn (G.4) yields:

\[
\frac{\tau_{NR}}{\tau_R} = \frac{4f_{tot} \eta_{tot} \sigma_{NR}}{\pi f_c \sigma_R^2}
\]  
(G.13)

which is identical to eqn (G.9).
Appendix H: Separation of the Components of the Surface Velocity

H.1. Introduction

The European Committee for Standardization (CEN) standard, EN12354-1 describes methods for the prediction of the apparent sound reduction index between two rooms due to an airborne noise source inclusive of flanking paths [23]. The inputs for the prediction methods include the sound reduction index of the elements along the flanking paths in the source and the receiver rooms. The standard requires that only the resonant component of the sound reduction index be used in the predictions [97]. In the case of massive, monolithic constructions with critical frequencies below the frequency range of interest, the sound reduction index measured according to ISO140 or ISO15186 can be used without correction since the resonant component dominates above the critical frequency [147]. However, in the case of lightweight materials which may have a critical frequency above the frequency range of interest, a method of accurately separating the resonant component from the measured sound reduction index is required. Correction factors have been proposed by Nightingale [86] and in Chapter 5 of this study for the separation of the resonant and non-resonant components of the sound reduction index based on the resonant and non-resonant components of the time and spatially averaged mean square velocity (mean square velocity) of the elements. In order to employ these correction factors, a reliable method for separating the components of the mean square velocity is needed.

Two methods are proposed for separating the components of lightweight building elements with a critical frequency above the frequency range of interest. The first method involves using a diffuse sound field to excite a series of panels, each with a different degree of damping. The mean square velocity on the surface of the panels is measured and the resonant and non-resonant components of the mean square velocity are calculated using a modified least squares method. The second method uses the velocity level difference $D_{v,ij}$ measured between building elements according to ISO10848 to separate the components of the mean square velocity when the test panels are excited by a diffuse sound field.
H.2. Theory

H.2.1. Damped Panels

When a panel is excited by a diffuse sound field, the velocity measured on the surface of the panel is a combination of the resonant and non-resonant velocities such that [1]:

\[ \langle v_T \rangle = \langle v_R \rangle + \langle v_{NR} \rangle \]  \hspace{1cm} (H.1)

where \( \langle v_T \rangle \) is the total, measured time and spatially averaged velocity, and \( \langle v_R \rangle \) and \( \langle v_{NR} \rangle \) are the resonant and non-resonant components, respectively. The resonant component is proportional to the damping of the panel whereas the non-resonant component is a function of the mass and almost independent of damping [26]. If the velocity is measured on the surface of two panels that are almost identical in material and mass, but have different levels of damping, it is assumed that non-resonant component of the velocity remains the same for both panels. Any differences in the total, measured velocity of the panels due to the differences in damping will be due to the differences in the resonant component only such that:

\[ \langle v_T \rangle_A - \langle v_T \rangle_B = \langle v_R \rangle_A - \langle v_R \rangle_B \]  \hspace{1cm} (H.2)

where the subscripts indicate panels A and B.

The resonant component of the mean square velocity of the panel due to excitation by the diffuse sound field may be calculated using statistical energy analysis (SEA) [15]. From the energy balance equations, the power transmitted from the diffuse sound field to the panel may be written as [59]:

\[ P = \langle v_R^2 \rangle \omega \rho_s \eta_{tot} \]  \hspace{1cm} (H.3)

where \( P \) is the power transmitted from the sound field to the panel, \( \langle v_R^2 \rangle \) is the resonant component of the mean square velocity, \( \omega \) is the frequency in radians per second, \( \rho_s \) is the
mass per unit area of the panel and $\eta_{tot}$ is the total loss factor of the panel. If the damping material is only applied to one side of the panels and the undamped sides of the panels are exposed to the sound field, it is assumed that the power transmitted from the sound field to the panels is the same for panels $A$ and $B$ such that:

$$\langle v_R^2 \rangle_A \omega \rho_s A \eta_{tot,A} = \langle v_R^2 \rangle_B \omega \rho_s B \eta_{tot,B}$$  \hspace{1cm} (H.4)$$

Eqn (H.4) may be rewritten as a ratio of the resonant component of the mean square velocities of the panels such that:

$$\frac{\langle v_R^2 \rangle_B}{\langle v_R^2 \rangle_A} = M_{AB}^2$$  \hspace{1cm} (H.5)$$

The velocities are related by the squared $M$ Factor, $M_{AB}^2$ which is a ratio of the mass per unit area and the total loss factor such that:

$$M_{AB}^2 = \frac{\rho_s A \eta_{tot,A}}{\rho_s B \eta_{tot,B}}$$  \hspace{1cm} (H.6)$$

Therefore, the total, measured velocity of panel $B$ may be written in terms of the resonant component of the velocity of panel $A$ such that:

$$\langle v_T \rangle_B = \langle v_R \rangle_A M_{AB} + \langle v_{NR} \rangle$$  \hspace{1cm} (H.7)$$

If instead of just two panels, there were a series of panels, each with a different amount of damping material applied to one side, the total velocity of each panel could be written in terms of the resonant component of a panel without additional damping such that:

$$\langle v_T \rangle_i = \langle v_R \rangle_i M_{oi} + \langle v_{NR} \rangle$$  \hspace{1cm} (H.8)$$
where \( i = 1 \ldots n \) is the number of each panel and the subscript \( o \) indicates the panel without additional damping. Eqn (H.8) describes a straight line with a Y intercept at \( M = 0 \) as shown in Figure H.1.

![Figure H.1: Changes in the total velocity as the value of the M factor is increased.](image)

The Y intercept is the value of the non-resonant component of the velocity and the slope of the line is the resonant component of the velocity. The total velocity is shown to decrease as the value of \( M \) decreases from a maximum value of 1 which is the panel without additional damping layers to 0 which is a panel with so much damping that \( \langle v_R \rangle = 0 \). If the slope and intercept of eqn (H.8) can be determined from experimental data using least squares methods, the values of the components of the velocity can be determined [96].

Once the components of the total velocity are known in each 1/3 octave band, the total, calculated mean square velocity may be written in terms of the resonant and non-resonant components such that:

\[
\langle v_T^2 \rangle = \langle v_{NR}^2 \rangle + \langle v_R^2 \rangle + 2\langle v_{NR} \rangle \langle v_R \rangle
\]  

\( \text{(H.9)} \)

where \( \langle v_R^2 \rangle \) and \( \langle v_{NR}^2 \rangle \) are the calculated resonant and non-resonant components of the mean square velocity. If the velocity of the panel can be assumed to be diffuse, the resonant and non-resonant components of the bending waves are assumed to be incoherent and therefore
the cross-terms of eqn (H.9) may be ignored [1, 49]. However, for lightweight constructions the assumption of a diffuse field may not be applicable and therefore eqn (H.9) was used in this study to determine the total, calculated mean square velocity.

**H.2.2. Measurements of the velocity level difference \( D_{v,ij} \)**

EN10848 describes measurement techniques to characterize the structure-borne vibration through building elements and junctions [25]. For example, in the case of two elements separated by a junction, the velocity level difference \( D_{v,ij} \) between the elements may be determined by exciting element \( i \) in several positions with an electromagnetic shaker and measuring the average velocity level of both elements such that:

\[
D_{v,ij} = L_{v,i} - L_{v,j}
\]  

(H.10)

where \( L_{v,i} \) and \( L_{v,j} \) are the average velocity levels on elements \( i \) and \( j \), respectively.

Gerretsen [97] notes that theoretically, only resonant transmission is important to the calculation of flanking transmission. If this assumption is applied to the measurements made on the building component tested according to EN10848, then the mean square velocity of element \( j \) due to the airborne excitation of element \( i \) is theoretically due only to the resonant component of the mean square velocity measured on element \( i \). Therefore, the resonant component of the mean square velocity of element \( i \) may be calculated by exciting element \( i \) with a diffuse sound field and measuring the mean square velocity of element \( j \) such that:

\[
L_{vR,i} = D_{v,ij,mechanical} + L_{vR,j,airborne}
\]  

(H.11)

where \( L_{vR,i} \) is the resonant velocity level of element \( i \), \( D_{v,ij,mechanical} \) is the velocity level difference measured due to excitation with an electromagnetic shaker according to EN10848 and \( L_{vR,j,airborne} \) is the velocity measured on element \( j \) due to the airborne excitation of panel \( i \) by the diffuse sound field. Rewriting eqn (H.11) in terms of linear quantities:
\[ \langle v_R^2 \rangle_i = \langle v_T^2 \rangle_{j,\text{airborne}} \left[ \frac{\langle v_R^2 \rangle_{j,\text{mechanical}}}{\langle v_F^2 \rangle_{j,\text{mechanical}}} \right] \]  \hspace{1cm} (H.12)

The non-resonant component is found by subtracting the resonant component from the total, measured velocity such that:

\[ \langle v_{NR}^2 \rangle_i = \langle v_T^2 \rangle_{j,\text{airborne}} - \langle v_R^2 \rangle_i \]  \hspace{1cm} (H.13)

Therefore, the components of the mean square velocity of element \(i\) may be determined during the testing according to EN10848 by adding the additional step of exciting element \(i\) with a diffuse sound field.

**H.2.3. Theoretical**

Heckl [1] has described the total, theoretical mean square velocity of a panel in a baffle excited by an diffuse sound field to be:

\[ \langle v_T^2 \rangle = \left[ \frac{\langle p^2 \rangle f_c \sigma_R}{6n f^2 \eta_{\text{total}} \rho_s^2} \right] + \left[ \frac{\langle p^2 \rangle}{2n^2 f^2 \rho_s^2} \right] \]  \hspace{1cm} (H.14)

where \(\langle p^2 \rangle\) is the time and spatially averaged mean square pressure of the sound field, \(f_c\) is the critical frequency of the panel, \(\sigma_R\) is the resonant radiation efficiency and \(f\) is the frequency in Hz. The term in the left bracket of the equation is the resonant component \(\langle v_R^2 \rangle\) and the term in the right bracket is the non-resonant component \(\langle v_{NR}^2 \rangle\).

**H.3. Experimental Methods**

**H.3.1. Damped Panels**

Steel panels which were 1.6mm thick and had a mass per unit area of 12.51 kg/m\(^2\) were used for the damped panels study. The critical frequency of the steel panels was calculated to be 7751 Hz and measurements confirmed that the critical frequency was in the 8000 Hz 1/3 octave band which is above the frequency range of interest. Two different types of damping materials were applied to the panels. The first material was Ultralon EVA 30 closed cell
Appendix H: Separation of the Components of the Surface Velocity

foam which was attached to the steel with an adhesive. The second material was a water based damping treatment (Vibradamp). The Vibradamp was poured into a mold constructed around the panel and allowed to cure. Once the Vibradamp hardened, the surface of the damping material was sanded to ensure a uniform thickness.

The panels were each mounted into an opening between a reverberant chamber and a semi-anechoic chamber. The mean square velocity was measured on the exposed steel side of the panel which faced into the reverberant chamber. Twelve random measurement positions were used for the spatial averaging. The uncertainty of the velocity measurements was determined according to the ISO Guide to the Measurement of Uncertainty (GUM) [64].

The difference between the mean square velocities measured at the different positions on each panel was found to be as high as 8 dB between the 100 Hz and 200 Hz 1/3 octave bands for several of the panels. Above the 200 Hz 1/3 octave band, the difference was less than 5 dB.

The structural reverberation time of the panel was measured in situ using an impact hammer and the integrated impulse response method described in ISO 3382 with backward integration of the square impulse response. The panel was impacted in fifteen locations and the surface velocity of the panel was simultaneously measured with four accelerometers which were moved to different, random locations for each impact.

The calculation of the resonant and non-resonant components of the velocity was performed using a modified least squares method (MLS) which minimized the offsets in both coordinates. The goal of the calculations was to reduce the distance of a perpendicular line from the data point to the best fit line, shown as $\epsilon_1$ and $\epsilon_2$ in Figure H.2.
The goal of the MLS calculations was to reduce the distance $\epsilon$ between the data points (shown with 95% confidence intervals) and the best fit line. The calculations were performed in MATLAB.

**H.3.2. L-Panels**

L shaped panels made of steel were fabricated by spot welding two 1.6mm thick steel panels to a piece of angle iron. Each of the steel panels had a mass per unit area of 12.51 kg/m$^2$ and a critical frequency in the 8000 Hz 1/3 octave band. L shaped panels made of MDF were fabricated by gluing and screwing two 4mm thick panels to a rectangular section of wood. The MDF panels had a mass per unit area of 3.2 kg/m$^2$ and also had a critical frequency in the 8000 Hz 1/3 octave band.

Element $i$ of the L-panels was mounted to the concrete walls between the test rooms with a soft, resilient material inserted between the edges of the element and the test room walls according to ISO10848-3 [148]. Element $j$ was also resiliently mounted into a frame. Resilient material was added to the envelopes around the two elements. The setup met the requirement that the coupling between the elements be weak in each 1/3 octave band as defined by ISO10848-1.
Appendix H: Separation of the Components of the Surface Velocity

The structural reverberation time of the panel was measured *in situ* using an impact hammer and the integrated impulse response method described in ISO 3382 with backward integration of the square impulse response. The panel was impacted in fifteen locations and the surface velocity of the panel was simultaneously measured in four positions.

The values of $L_{v,i}$ and $L_{v,j}$ were measured according to EN10848 using structure-borne excitation. A Brüel & Kjær 4810 shaker was connected via a stinger to a PCB 288D01 impedance head which was fastened to the panel surface. Three excitation points per panel were used and the average velocity level was calculated from twelve measurement positions per panel. The uncertainty of the velocity measurements was determined according to GUM and the velocity measurements at each of the positions on the panel were treated as correlated inputs during the calculation of the uncertainty. The velocity level difference between the elements was determined according to EN10848-1.

A requirement of EN10848-1 and of SEA is that the vibration fields in the elements be diffuse. EN10848-1 quantifies a diffuse field by stating that there is a strong decrease in vibration across an element if the measured velocity level decreases by more than 6 dB over the allowed measurement area when the accelerometer is moved away from the stationary vibration source. During the measurements on the L-panels, it was found that the difference in the velocity level measured at the twelve measurement positions on each panel exceeded the 6 dB limit in most of the 1/3 octave bands. The variance of the velocity measurements was expected to affect the uncertainty of the velocity level measurements and the calculation of the velocity level difference between the elements.

The velocity level difference for the steel and the MDF L-panels are compared in Figure H.3.
The figure shows that the 95% confidence levels were as high as ±3dB in some of the 1/3 octave bands. The large uncertainty was expected to affect the calculation of the resonant and non-resonant components of the mean-squared velocity.

In addition to the measurements made according to EN10848-1, panel \( i \) of the L-panel was mounted into an opening between a reverberant and a semi-anechoic chamber. Panel \( j \) extended into the semi-anechoic chamber and was clamped into a frame. Panel \( i \) was excited using airborne excitation and the average velocity level was measured at twelve measurement positions on each panel. The sound power of panel \( j \) was measured using sound intensity.

**H.4. Results**

**H.4.1. Damped Panels**

The resonant and non-resonant components of the velocity were evaluated in each 1/3 octave band. For example, Figure H.4 shows the calculations for the 100 Hz 1/3 octave band.
Figure H.4: Total, measured velocity versus $M$ Factor for the 100Hz 1/3 octave band for the five steel panels with 95% confidence intervals. The linear curve fit was calculated using the modified least squares method.

The error bars in the figure are the 95% confidence intervals of the total, measured velocity and the $M$ factors. The calculated resonant and non-resonant components of the mean square velocity from the MLS method are compared to the theoretical components in Figure H.5.
Figure H.5: Comparison between the calculated and the theoretical resonant and non-resonant components of the mean square velocity.

The theoretical values shown in Figure H.5 are those from Heckl as described in eqn (H.14). The calculated and theoretical curves in the figure have similar slopes, but the calculated values are lower than the theoretical curves over most of the frequency range. The figure shows that the magnitude of the calculated non-resonant component was on average 14 dB less than that of the calculated resonant component. Likewise, the theoretical non-resonant component is shown to be on average 11 dB less than the theoretical resonant component.

The total, measured mean square velocity is compared in Figure H.6 to the total value calculated from theory and from the components of the mean square velocity calculated using the MLS method.
The figure shows that the total mean square velocity calculated from the sum of the components from the MLS method is in good agreement with the total, measured value over the entire frequency range. The good agreement suggests that the sum of the components calculated using the MLS method was correct.

Since the theoretical non-resonant component is on average 11 dB less than the resonant component over the frequency range, the magnitude of the total, theoretical mean square velocity is determined predominantly from the resonant component. For example, between the 1000 Hz and the 1600 Hz 1/3 octave bands, the values of the calculated, theoretical and measured mean square velocities are within 1 dB of each other and the total, measured mean square velocity. Figure H.5 shows that within this frequency range, the calculated and the theoretical resonant component are also within 1 dB of each other. Over the rest of the frequency range, theory appears to over predict the resonant component of the mean square velocity and therefore the total value over much of the frequency range. The over prediction
may be due to the non-diffuse wave field at the low frequencies, the finiteness of the panel and the edge losses of the panel.

**H.4.2. L-Panels**

The components of the mean square velocity for the steel and MDF L-panels as calculated from eqns (H.12) and (H.13) are shown in Figure H.7 and Figure H.8, respectively.

![Image of Figure H.7](image-url)

**Figure H.7:** Comparison between the calculated and the theoretical components of the mean square velocity for the steel L-panel. The error bars are the 95% confidence of the calculated resonant component. Also shown is the total, measured mean square velocity.
Appendix H: Separation of the Components of the Surface Velocity

Both figures show that the resonant component of the mean square velocity was calculated to be larger than the total, measured value in most of the 1/3 octave bands, resulting in a negative non-resonant component. The figures show gaps in the plots of the non-resonant component where it was calculated to be negative. The non-resonant components which are shown in the figures were calculated to have a large measurement uncertainty.

The separation of the components of the mean square velocity was not successful in most of the 1/3 octave bands. In most cases, the total measured mean square velocity is within the 95% confidence interval of the resonant component. Therefore, much of the overestimation of the resonant component may have been due to measurement uncertainty due to the non-diffuse velocity field.
Appendix H: Separation of the Components of the Surface Velocity

It is interesting to note that the theoretical resonant component is higher than both the total, measured mean square velocity above the 2000 Hz 1/3 octave band. A similar trend can be seen in the plots of the total mean square velocity for both L-panels in this frequency range as shown in Figure H.9 and Figure H.10.

Figure H.9: Total, mean square velocity of panel $i$ of the steel L-panel compared to the value from theory. Panel $i$ was excited by the diffuse sound field. The error bars are the 95% confidence interval of the measured data.
Appendix H: Separation of the Components of the Surface Velocity

Figure H.10: Total, mean square velocity of panel $i$ of the MDF L-panel compared to the value from theory. Panel $i$ was excited by the diffuse sound field. The error bars are the 95% confidence interval of the measured data.

As with the damped, steel panels, the theoretical equations appear to overestimate the resonant component at the higher frequencies.

**H.4.3. Velocity Ratio**

The ratio of the non-resonant to resonant components of the mean square velocity as calculated from the damped steel panels and the steel L-panel are compared versus theory in Figure H.11.
Appendix H: Separation of the Components of the Surface Velocity

Figure H.11: Comparison of the ratio of the non-resonant to the resonant component of the mean square velocity of the steel panels as calculated from the MLS method, the L-panels and from theory.

The theoretical ratio and the ratio calculated from the MLS methods show reasonable agreement up to the 630 Hz 1/3 octave band after which they differ by up to 13 dB in the 1600 Hz 1/3 octave band. Figure H.5 showed that between the 1000 Hz and the 1600 Hz 1/3 octave bands, the theoretical and calculated resonant components were in good agreement. Therefore, the difference in the velocity ratios in this frequency range is due to the difference in the non-resonant components. The difference between the calculated resonant and non-resonant components is an average of 18 dB in this frequency range while it is an average of 8 dB for the theoretical components.

The velocity ratio from the L-panels could only be calculated in some of the 1/3 octave bands. In the low frequencies, the theoretical, MLS and calculated L-panel velocity ratios were in agreement. However, it appears that overall, the calculated velocity level may underestimate the ratio in most of the frequency range. One possible reason for this is that the velocity field of the L-panels was not diffuse.
The ratio calculated from the MDF L-panel and from theory are compared in Figure C.12.

![Graph](image)

**Figure H.12:** Comparison of the ratio of the non-resonant to the resonant component of the mean square velocity of the MDF panels as calculated from the L-panels and from theory.

The figure only shows the result from the MDF L-panel since damped MDF panels were not evaluated in this study. The calculated ratio of the MDF panel data do not appear to agree with the theoretical ratio over the limited frequency range over which they could be compared.

**H.4.4. Flanking Sound Reduction Index**

The flanking sound reduction index was calculated for the steel L-panel according to EN12354-1 using the resonant component of the sound reduction index calculated from the proposed correction factor described in Part I of this study. The velocity ratios used for the correction factor are those shown in Figure H.11. The value of the vibration reduction index $K_{IJ}$ of the steel L-panel was determined according to EN10848-1.
The use of the predicted sound reduction index for the evaluation of the velocity ratios assumes that the vibration reduction index measured according to EN10848-1 is correct. However, since the measurements for the calculation of the vibration reduction index failed the EN10848-1 criteria for a diffuse vibration field on the elements, the predictions of the flanking sound reduction index are subject to large measurement uncertainty. Therefore, although the comparison of the predictions of the flanking sound reduction index may be applicable to the steel L-panel used in this study, conclusions of the overall accuracy of the different methods of separating the components of the mean square velocity may not be possible.

The predictions of the flanking sound reduction index are compared in Figure H.13.

![Figure H.13: Comparison between the flanking sound reduction indices. The proposed correction factor was used to calculate the resonant sound reduction index and the calculated and theoretical resonant and non-resonant components of the mean square velocity. Also shown are predictions based on other calculations of the resonant component of the sound reduction index. The error bars are the 95% confidence interval of the measured flanking reduction index.](image-url)
The number of data points for the L-panel are limited, but over the frequency range where it can be evaluated it appears to underestimate flanking sound reduction index for the steel panels used in this study at the higher frequencies. Further conclusions regarding the velocity ratio from the L-panels can not be made due to the limited frequency range over which they could be calculated.

The velocity ratio calculated from the MLS method shows better agreement with the measured flanking sound reduction index than the theoretical velocity ratio. The prediction using the theoretical ratio agrees less with the measured flanking sound reduction index, but does agree better with the other predictions of the flanking sound reduction index shown in the figure than the prediction using the MLS method.

**H.5. Discussion**

The results of this study suggest that the use of a series of damped panels and the MLS method could be a possible method for separating the resonant and non-resonant components of the mean square velocity. A difficulty in applying this method to lightweight materials is the non-diffuseness of the surface velocity of the elements. Further investigations with other materials are needed before the accuracy of this method can be determined. The use of a series of damped panels to determine the components of the mean square velocity may not be practical for panel constructions which can be solved by theory. However, accurate theoretical predictions for more complicated panel construction may not be as straightforward and in these cases, this technique may be useful.

A more practical experimental method of separating the components of the mean square velocity would be the use of velocity level difference measured according to EN10848 in addition to measurements made with a diffuse sound field. In this study, the panels were L shaped, but this need not be the case since other shapes could also be used. Using this technique requires only the additional step of exciting the wall construction with a diffuse sound field.
For an ideal SEA subsystem, there exists a uniform energy density and therefore there should not be a strong gradient in the velocity level [39]. EN10848 quantifies this in stating that the difference in the velocity level measured over the surface of the panel should not exceed 6 dB. It was found that during this study, this requirement was difficult to achieve for the L-panel and the difference between the velocity measurements was higher than 6 dB in almost all of the 1/3 octave bands. The variation in the velocity levels lends to a large uncertainty in the measurements which affects the calculation of the components of the mean square velocity. In this study, measurement uncertainty was pronounced and may have exceeded the quantities being subtracted, resulting in the calculation of negative values for one of the components.

The calculation of the velocity level difference for lightweight constructions presents other difficulties. Jauer [149] found in a study of junction attenuation of lightweight constructions that the velocity level difference exhibited a strong dependence on the measurement location. Furthermore the measured velocity level difference was not determined only to the juncture between the elements, but also by the attenuation in the elements themselves. Gerretsen [27] has noted that for excitation by point forces, the power incident on the junction will depend on the distance of the excitation point from the junction and the propagation attenuation (the attenuation rate corrected for geometrical spreading). Villot [115] has gone on to write that lightweight elements are often highly damped, the vibrational fields are no longer reverberant and existing standards often lose relevance, particularly in the case of mechanical excitation such as in vibration reduction index measurements of junctions. Therefore, to characterize a junction of such elements adequately also the velocity level reduction with distance (dB/m) is relevant. The addition of such a term may reduce the uncertainty in the calculations of the components of the mean square velocity. If the uncertainty of the measurements made on the L-panels could be reduced, using this technique would be a practical means of separating the components of the mean square velocity.
However, even if the components of the mean square velocity are separated successively, the use of the components to separate the sound reduction index may not be practical. The uncertainty in the velocity components and from the sound reduction index are both propagated into the uncertainty of the calculated resonant component of the sound reduction index. The uncertainty of each of these sources may be significant [98]. Therefore, other methods such as using theory or numerical techniques to predict the resonant component of the sound reduction index.
Appendix I: SEA Models of the L-Shaped Panels

I.1. Introduction to ESEA Models

The statistical energy analysis models used to calculate the flanking sound reduction index of the L-shaped panels are described in this appendix.

The coupling loss factors through the junctions between the panels of a double-leaf construction would be difficult to determine theoretically due to the complexity of the junctions and the possible influence of the construction (workmanship, screw spacing, etc). Therefore, the coupling loss factors through the junctions between the panels were determined experimentally. Mace [150] refers to the determination of the parameters of an SEA model from experimental measurements on a test structure as experimental SEA, or ESEA.

For example, in the case of the two panels (noted as subsystems 1 and 2) joined by a junction, the power balance equation is:

\[ E_2 \eta_{12} = E_2 \eta_2 \]  \hspace{1cm} (I.1)

Therefore, the coupling loss factor between the panels may be found from experimental data such that:

\[ \eta_{12} = \frac{\rho_2 S_2 \langle v_2^2 \rangle \eta_2}{\rho_1 S_1 \langle v_1^2 \rangle} \]  \hspace{1cm} (I.2)

where the total energy in a plate is \( E = \rho S \langle v^2 \rangle \). The values of the spatial average of the mean square velocity and the loss factors were determined experimentally. For this study, the values of the loss factor were determined by impacting the leaf included in the calculations with an impact hammer and measuring the vibratory response on the same leaf. Consistency for the measurement of the loss factor was important since the loss factor of the different leafs of a double-leaf construction can be different depending on which leaf is impacted.
I.2. Single Panel

A diagram of the single-leaf L-panel construction is shown in Figure I.1.

![Figure I.1: Schematic of the double-leaf L-shaped panel.](image)

The power flow diagram of the single-leaf L-panel is shown in Figure I.2.

![Figure I.2: Power flow diagram of the single-leaf L-shaped panel. The solid lines represent power flow and the dotted lines represent losses.](image)

Rather than to solve the all of the power balance equations for the system, a path by path analysis was used to evaluate independently the sound transmission along flanking paths between the source and the receiving rooms. Although the figure shows a number of possible transmission paths between rooms 1 and 7, only two transmission paths were considered in
the path by path analysis. The considered paths included the structure-borne noise between elements 2 and 4 as well as resonant and non-resonant transmission between rooms 1 and 5 and rooms 3 and 7.

**Path 1: 1 → 4 → 6 → 7**

\[ R_{17,1} = R_{15,R} + D_v,46 + 10 \log \left( \frac{S_2}{S_4} \right) + 10 \log \left( \frac{S_6}{S_4} \right) \]  \hspace{1cm} (I.3)

where \( R_{15,R} \) is the resonant component of the sound reduction index of the panel, \( D_v,26 \) is the velocity level difference between elements 2 and 6 and \( S_5 \) is the reference area.

**Path 2: 1 → 4 → 6 → 7**

\[ R_{17,2} = R_{15} + R_{57} + 10 \log \left( \frac{S_5 A_5}{S_2 S_6} \right) \]  \hspace{1cm} (I.4)

where \( A_5 \) is the absorption area of room 5. In the calculation of \( R_{15} \) and \( R_{57} \), both the resonant and the non-resonant components of the sound reduction index were included. Therefore, path 2 also included path 1 → 5 → 7.

**I.3. Double-Leaf Construction**

A diagram of the double-leaf L-panel construction is shown in Figure I.3.
The power flow diagram of the L-panel is shown in Figure I.4.

Following from the work of Schoenwald [121], a path by path analysis was used to evaluate independently the sound transmission along six flanking paths between the source and the receiving rooms.
Appendix I: SEA Models of the L-Shaped Panels

Path 1: 1 → 2 → 6 → 7

\[
R_{17,1} = R_{13,R} + D_{v,26} + 10 \log \left( \frac{\sigma_{23}}{\sigma_{67}} \right) + 10 \log \left( \frac{S_2}{S_6} \right) \tag{I.5}
\]

where \( R_{13,R} \) is the resonant component of the sound reduction index between elements 1 and 3, \( D_{v,26} \) is the velocity level difference between elements 2 and 6, \( \sigma_{23} \) is the radiation efficiency of leaf 2 into cavity 3. Craik [59] notes that the radiation efficiency of a panel into a cavity may not be the same as that of a panel into a free space. However, in a similar SEA model of a double-leaf wall [151], the radiation efficiency as described by Leppington [152] was used for the radiation of the leaf into the cavity, but with a correction. In this study, the same equations for the radiation efficiency of a panel into free space will be used to describe the radiation efficiency of the panel into the cavity.

Path 2: 1 → 2 (3) → 4 → 6 → 7

\[
R_{17,2} = R_{15,d} + D_{v,46} + 10 \log \left( \frac{\sigma_4}{\sigma_6} \right) + 10 \log \left( \frac{S_2}{S_6} \right) \tag{I.6}
\]

Schoenwald [121] describes \( R_{15,d} \) as the sound reduction index between rooms 1 and 5 for resonant transmission between the elements only to be:

\[
R_{15,d} = D_{24} + 10 \log \left( \frac{2\pi f^2 \eta_2 \rho_2^2}{f_2 \sigma_2 \sigma_4 \rho_4 \rho_2 \sigma_2} \right) \tag{I.7}
\]

where \( D_{24} \) is the corrected velocity level difference between elements 2 and 4. The correction must be applied since the excitation of element 2 with an electromagnetic shaker will cause both resonant and non-resonant velocity on panel 4 and \( R_{15,d} \) is for resonant transmission only. Schoenwald [121] describes the correction as:

\[
\bar{L}_{4,R} = \bar{L}_{4,R+NR} - D_{corr,R} \tag{I.8}
\]
where $L_{4,R}$ is the resonant component of the average velocity level and $L_{4,R+NR}$ is the total average velocity level. $D_{corr,R}$ is the correction factor given by:

$$D_{corr,R} = -10 \log \left[ \frac{\rho_{2}\rho_{3}c_{o}\sigma_{23}\sigma_{4}\eta_{4}\tau_{13}}{4\rho_{5}^{2}S_{2}f_{c,4}\eta_{4}(\Sigma \alpha)} \left( \frac{\nu_{2}^{2}}{\nu_{2,measured}^{2}} \right) \right] \quad (I.9)$$

where $S_{3}$ is the area of the cavity, $L$ is the length of the absorbing material and $\alpha$ is the absorption coefficient which is not the same as the coefficient when the material is placed in a room [59].

**Path 3: 1 $\rightarrow$ 3 $\rightarrow$ 4 $\rightarrow$ 6 $\rightarrow$ 7**

$$R_{17,3} = R_{15,c} + D_{v,46} + 10 \log \left( \frac{\sigma_{4}}{\sigma_{6}} \right) + 10 \log \left( \frac{S_{6}}{S_{6}} \right) \quad (I.10)$$

Schoenwald [121] describes $R_{15,c}$ as the resonant sound reduction index between rooms 1 and 5 such that:

$$R_{15,c} = 10 \log \left[ \frac{4(\Sigma \alpha)\eta_{4}\eta_{5}f_{c,4}^{2}\rho_{5}^{2}}{S_{6}S_{7}c_{o}\tau_{13}\rho_{5}^{2}f_{c,4}\eta_{4}^{2}} \right] \quad (I.11)$$

The transmission factor $\tau_{13}$ between elements 1 and 3 may is for non-resonant transmission and the non-resonant transmission factor of Leppington may be used to calculate this term.

**Path 4: 1 $\rightarrow$ 2 $\rightarrow$ 9 $\rightarrow$ (8) $\rightarrow$ 6 $\rightarrow$ 7**

$$R_{17,4} = R_{57,d} + D_{v,29} + 10 \log \left[ \frac{\alpha_{9}f_{c,9}\rho_{5}^{2}\eta_{5}}{\sigma_{5}f_{c,2}\rho_{5}^{2}\eta_{4}} \right] + 10 \log \left( \frac{S_{6}}{S_{6}} \right) \quad (I.12)$$

where
Appendix I: SEA Models of the L-Shaped Panels

\[ R_{57,d} = D_{s96} + 10 \log \left( \frac{2\pi f^2 \eta_s \rho_s^2}{f_{c,5}^2 \sigma_5 \sigma_9 \rho_5 \rho_9} \right) \]  
(I.13)

and \( D_{s96} \) is the corrected velocity level difference.

**Path 5: 1 \rightarrow 2 \rightarrow 9 \rightarrow 8 \rightarrow 7**

\[ R_{17,5} = R_{57,c} + D_{v,29} + 10 \log \left( \frac{\sigma_2 f_{c,2} \rho_2^2 \eta_2}{\sigma_5 f_{c,5} \rho_5^2 \eta_5} \right) + 10 \log \left( \frac{S_8}{S_9} \right) \]  
(I.14)

where \( R_{57,c} \) is the sound reduction index between rooms 5 and 7 for resonant transmission between the elements such that:

\[ R_{57,c} = 10 \log \left( \frac{4(\Sigma L_a) \eta_s \rho_s^2}{S_8 \rho_s \tau_{58} \rho_5 \rho_8} \right) \]  
(I.15)

The transmission factor \( \tau_{58} \) between elements 5 and 8 may is for non-resonant transmission and the non-resonant transmission factor of Leppington may be used to calculate this term.

**Path 6: 1 \rightarrow 2 \rightarrow (3) \rightarrow 4 \rightarrow 5 \rightarrow 9 \rightarrow (8) \rightarrow 6 \rightarrow 7**

\[ R_{17,6} = R_{15,d} + R_{57,d} + 10 \log \left( \frac{S_5 A_5}{S_4 S_9} \right) \]  
(I.16)

The total flanking sound reduction index between the source and the receiver room is then calculated by summing the contributions of the individual paths.
Appendix J: Description of the Field Testing Site

J.1. Selection of the Building Site

Early in this study, the need for field measurements was identified and a suitable site for the testing was sought. A student accommodation which was under construction at the University of Canterbury was suggested by the author as a potential site for the field testing and the supervisory team entered into discussions with the builders at the site. However, the permissions required for the testing were not obtained before the construction was completed and the buildings occupied.

The author delivered a departmental seminar on his research in April 2008 during which it was mentioned that a site for field testing was being pursued. After the seminar, Dr. Gregory A. MacRae of the Civil Engineering Department suggested to the author that rather than finding an offsite building for the field testing, adjacent offices in the University of Canterbury Civil/Mechanical building could be considered. Two offices on the fifth floor of the building were identified as being ideal for the study because they were only being used for meetings and seminars and therefore lacked permanent office furniture which would need to be removed for the testing. Furthermore, permission was granted to modify the common wall between the rooms to increase the sound reduction index of the direct transmission path.

J.2. Overview of the Field Testing Site

An outside view of the rooms used for the field testing is shown in Figure J.1 and the floor plan is shown in Figure J.2.
Figure J.1: View of the outside of the Civil/Mechanical building. The location of the offices on the fifth floor used for the testing is indicated in the figure.

Figure J.2: Floor plan showing the locations of offices E517 and E519 used for the testing.

The floor plan shows that each room includes one exterior wall, the common dividing wall and one wall shared with the corridor. E517 shares a wall with an elevator. Possible noise from the elevator during test was eliminated by requesting that the elevator repair service disable the elevator before the testing began. E519 partially shares a wall with the corridor and with another office. Both rooms share a concrete slab for the floor and E517 included a
Appendix J: Description of the Field Testing Site

drop ceiling whereas E519 did not. More details about each wall are given in the following sections.

J.3. Element Descriptions

J.3.1. Common Dividing Wall
The common dividing wall between the rooms was a double-leaf construction of 16mm GIB Fyreline® screwed to steel studs which were 450mm on center as shown in Figure J.3.

![Photographs of the original common dividing wall with a section of wall removed. The photograph shows the 16mm thick Fyreline gypsum board screwed to metal studs.](image)

The figures show that cavity of the common dividing wall did not included insulation. The dividing wall extended from the floor of the rooms to the T in the concrete ceiling as shown in Figure J.4.
Figure J.4: Photograph of the common dividing wall showing how it extends from the floor to single T ceiling section.

A cross section of the dividing wall from the architect’s drawings is shown in Figure J.5.

Figure J.5: Cross section of the common dividing wall showing the connection with the ceiling.
The cross section shows that there is a 15mm gap between the double-leaf wall and the concrete T. The gap is filled with what is noted in the drawing as an acoustic partition seal. The material used for the seal was similar to fiberglass.

The effectiveness of the acoustic partition seal was found to be lacking which may explain why the suspended ceiling in E517 was added after the building was constructed. During the course of setting up the rooms for testing, a layer of fiberglass insulation was added above the drop ceiling as explained in section J.3.2. The insulation was added by removing several ceiling panels and using a ladder to climb into the large space above the suspended ceiling. During the installation of the fiberglass insulation, it was possible to hear conversations being conducted in E519. The conversations could not be heard when the suspended ceiling was in place. By ear, it appeared the majority of the noise was coming through the acoustic partition seal. To make matters worse, the conversations which were being overheard were parts of a job interview.

To reduce the transmission of noise through the acoustic partition seal during the testing, fiberglass insulation was added along the gap between the double-leaf wall and the concrete T as shown in Figure J.6.
Appendix J: Description of the Field Testing Site

Figure J.6: Fiberglas insulation placed above the suspended ceiling and along the common dividing wall.

The suspended ceiling bisected the dividing wall as shown in Figure J.7.

Figure J.7: Cross section of the common dividing wall and the suspended ceiling. The common dividing wall is broken into three elements by the intersection of the suspended ceiling and the change in material.
EN12354-1 Part 4.2.4 gives guidelines when elements are constructed of different parts or horizontally displaced. Following these guidelines, the common dividing wall was broken into three elements as shown in Figure J.7. The EN12354 method only includes first order flanking paths [39] and transmission across two or more structural junctions is not considered. Therefore, only flanking transmission from elements e and d is to be considered in the EN12354 calculations and element f was neglected.

**J.3.2. Floor and Ceiling Construction**

The floor and ceiling of the rooms were made of precast concrete single Tee units as shown in Figure J.8.

![Figure J.8: Floor plan of the rooms including details regarding the floor and ceiling construction.](image)

A cross section of the single Tee floor unit is shown in Figure J.9.
The concrete sections were exposed in room E519, but a suspended ceiling was added in room E517 after the building was constructed as shown in the cross-section view of the rooms shown in Figure J.10 and in the photographs shown in Figure J.11.
The panels of the suspended ceiling were made of a heavy, ceramic material with a mass per unit area of 13.07 kg/m$^2$. The sound insulation of the suspended ceiling was increased for the study by adding a layer of fiberglass insulation on top of the ceiling tiles as shown in Figure J.6.

A sound source was placed in room 519 and the sound intensity from the suspended ceiling in room 517 with the additional insulation was measured with an intensity probe. The sound intensity level from the suspended ceiling with the additional insulation was found to be negative or at the same magnitude as the sound pressure level of the background noise in the rooms. It was concluded that the suspended ceiling in the receiving room should not be considered in this study as a flanking transmission path.

Likewise, the sound intensity level from the floor in the receiving room was found to be negative or at the same magnitude as the sound pressure level of the background noise in the rooms. Therefore, the floor in the receiving room was eliminated as a flanking transmission path for this study.
In the source room, structure-borne noise transmitted from the ceiling would need to first pass through one of the other elements in the source room prior to being transmitted to one of the elements in the receiving room. For example, the flanking path from the ceiling of the source room to the dividing wall in the receiving room is shown in Figure J.12.

![Flanking transmission from the ceiling of the source room to the dividing wall in the receiving room.](image)

Since only first order flanking paths are considered by the EN12354 method, the ceiling in the source room was not include as a flanking element in this study.

During the testing, a 100 N electromagnetic shaker was connected to the concrete floor in the source room via a stinger screwed into the floor and the response of the elements in the receiving room was measured. The velocity level measured on the elements was consistently at or below the background levels. Therefore, it was concluded that the floor in the source room was not part of any of the major flanking paths and could be neglected from this study.
Appendix J: Description of the Field Testing Site

J.3.3. Window Walls

The window walls of E517 and E519 each included a large window as shown in Figure J.13.

![Figure J.13: Photograph of the window in the exterior wall which was covered by insulating material.](image)

The photograph shows the window covered with insulation to avoid an exterior airborne flanking path between the window in the source room and the window in the receiving room. This follows from a study by Hongisto [14], in which it was found that noise being transmitted between two row houses was due in part to an exterior airborne flanking path between windows located next to the entrance doors of the houses.

Both wall h in the source room and wall H in the receiving room were evaluated in their entirety and as smaller elements. The purpose of evaluating the smaller elements was to investigate the effect of measuring the vibratory response close to the junction which was expected to reduce the potentially large variance in the measurements of the velocity level.

The elements of the window wall h of room E519 are shown in Figure J.14.
The smaller element h1 is shown to be the area of the wall between the common wall and the window. Element h3 was considered to be a separate element due to the wall construction. The rest of the wall was neglected when only the smaller elements were considered since only first order flanking paths are considered by the EN12354 method.

The elements of the window wall H of room E517 are shown in Figure J.15.
Appendix J: Description of the Field Testing Site

The smaller element H1 between the window and the common wall was considered for the study. The area above the window could have been an element, but the area as so small that it was neglected.

J.3.4. Entrance Wall
Both E517 and E519 had a double-leaf gypsum board wall which separated the rooms from the corridor, referred to in this study as the entrance wall. The wall included a door and a window as shown in Figure J.16.

![Figure J.16: Photograph of the entrance wall. The entrance walls in E517 and E159 were mirror images of each other. The clock above the window was removed during the testing due to the ticking noise.](image)

EN12354-1 Section 4.2.4 allows for the part of the wall behind the door to be neglected if the door is a complete discontinuity in the wall. However, the doors to the rooms used for this study did not present complete discontinuities in the walls and therefore, the entire surface of the wall was considered as one element. The door and the window elements were neglected as flanking paths in this study since they represented higher order paths which are not considered by the EN12354 method.
To prevent noise from being transmitted as airborne noise in the corridor as shown in Figure J.17, the windows and doors were blocked with insulation material as shown in Figure J.18.

![Figure J.17: Possible transmission of noise from the source room to the receiver room via the corridor.](image)

![Figure J.18: Photographs of the insulation material over the window and the door on the entrance wall to prevent the transmission of noise via the corridor.](image)

The material remained in place both during the measurement of the sound intensity due to the sound source and the measurements of the velocity level differences between the flanking elements due to the electromagnetic shaker.
J.4. Summary of the Transmission Paths

The transmission paths considered for the prediction of the apparent sound reduction index are summarized in Figure J.19 to Figure J.23.

![Figure J.19: Direct path dD and flanking path eD.](image1)

![Figure J.20: Flanking paths with element e as the first element.](image2)
Figure J.21: Flanking paths with the common wall as the first element.

Figure J.22: Flanking paths with element k as the first element.

Figure J.23: Flanking paths with element h1 or h3 as the first element.
In addition, the contribution from the direct path $dD$ was also included in the prediction of the apparent sound reduction index.

### J.5. Dimensions of the Elements and Rooms

The areas of the element are shown in Table J.1.

<table>
<thead>
<tr>
<th>Element</th>
<th>Area (m²)</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>H</td>
<td>3.16</td>
<td>E517 Window wall excluding window</td>
</tr>
<tr>
<td>H1</td>
<td>1.38</td>
<td>E517 Corner element of window wall</td>
</tr>
<tr>
<td>K</td>
<td>4.01</td>
<td>E517 Door wall</td>
</tr>
<tr>
<td>D</td>
<td>13.37</td>
<td>E517 Common wall</td>
</tr>
<tr>
<td>h</td>
<td>4.69</td>
<td>E519 Window wall</td>
</tr>
<tr>
<td>h1</td>
<td>1.66</td>
<td>E519 Corner element of window wall</td>
</tr>
<tr>
<td>h3</td>
<td>0.69</td>
<td>E519 Top element of window wall</td>
</tr>
<tr>
<td>k</td>
<td>6.10</td>
<td>E519 Door wall</td>
</tr>
<tr>
<td>d</td>
<td>13.37</td>
<td>E519 Door wall</td>
</tr>
<tr>
<td>e</td>
<td>2.72</td>
<td>E519 Flanking element on common wall</td>
</tr>
</tbody>
</table>

**Table J.1**: Areas of the flanking elements. Also included in the table is the area of element H which is the window wall in E517.

The volumes and the floor area of the rooms are show in Table J.2.

<table>
<thead>
<tr>
<th>Room</th>
<th>Volume (m³)</th>
<th>Floor Area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E517</td>
<td>38.12</td>
<td>15.40</td>
</tr>
<tr>
<td>E519</td>
<td>56.93</td>
<td>16.59</td>
</tr>
</tbody>
</table>

**Table J.2**: Volume and surface area of each room.

Since the rooms were not identical, diffusers were not needed during the testing to prevent coupling between the rooms.
J.6. Modal Densities

The modal densities of the elements are shown in Table J.3.

<table>
<thead>
<tr>
<th>Element</th>
<th>H</th>
<th>H1</th>
<th>K</th>
<th>D</th>
<th>h</th>
<th>h1</th>
<th>h3</th>
<th>k</th>
<th>d</th>
<th>e</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area (m²)</td>
<td>3.16</td>
<td>1.38</td>
<td>4.01</td>
<td>13.37</td>
<td>4.69</td>
<td>1.66</td>
<td>0.69</td>
<td>6.10</td>
<td>13.37</td>
<td>2.72</td>
</tr>
<tr>
<td>Modal Density</td>
<td>0.17</td>
<td>0.07</td>
<td>0.21</td>
<td>0.71</td>
<td>0.25</td>
<td>0.09</td>
<td>0.04</td>
<td>0.33</td>
<td>0.71</td>
<td>0.15</td>
</tr>
</tbody>
</table>

Table J.3: Modal density of each element.

J.7. Adaptation Terms for the Rooms

The Waterhouse correction may be used in the determination of the sound power in the rooms as discussed in Appendix M. ISO15186-2:2003 refers to the Waterhouse correction as the adaptation term \( K_c \). The values of the adaptation terms of rooms E517 and E519 are compared to the adaptation term from Annex A of ISO15816-2:2003 in Figure J.24.

![Figure J.24: Comparison between the adaptation terms of E517 and E519 and the adaptation term listed in ISO15186-2.](image)

The adaptation term of E517 is shown to be larger than that of E519 due to the smaller room volume.
Appendix K: Sound Reduction Indices for the Field Testing

K.1. Introduction

The EN12354 method requires as inputs the resonant sound reduction of the flanking elements and the total sound reduction index of the common wall between the rooms. All of the lightweight elements in the rooms were made of 16mm Fyreline gypsum board with metal studs. Although measured data was available for other wall constructions using Fyreline gypsum board, measured values of the specific constructions in the rooms were not available and therefore needed to be calculated.

The typical wall construction in the rooms was a double-leaf wall with 16 mm of Fyreline gypsum board on metal studs. The cavity was 88 mm deep and the studs were located 450 mm on center. No insulation was used in the cavity. The manufacturer listed the following properties for 16mm Fyreline gypsum board: $\rho = 883 \text{ kg/m}^3$, $\rho_s = 14.1 \text{ kg/m}^2$ and $E = 3.7 \text{ GPa}$.

K.2. Total Sound Reduction Index

The calculation of the total sound reduction index of the typical wall construction included calculations according to the equations in Annex B of EN12354. The equations of Annex B are based on monolithic walls and therefore are not likely to be as accurate for lightweight, double-leaf constructions. Therefore, the method of Sharp [124] with line-line support was also used to calculate the sound reduction index.

The manufacturer of the gypsum board was able to provide measurements made according to ISO140 for a double-leaf construction with a 100 mm cavity and wood studs 600 mm on center. The measured values are compared to predictions according to Annex B of EN12354 and the Sharp model in Figure K.1.
Also shown in the figure is the sound reduction index of a single sheet of Fyreline gypsum board, also measured according to ISO140. The manufacturer noted that Fyreline gypsum board is commonly chosen to be used due to its fire resistant properties rather than its noise reduction properties.

The figures show that the Fyreline gypsum board single sheet and the double-leaf construction both have a critical frequency in the 2000 Hz 1/3 octave band. The prediction using the equations from Annex B of EN12354 shows reasonable agreement with the measurements up to the 1250 Hz 1/3 octave band after which the measured sound reduction index is underestimated by approximately 15 dB. The prediction using the Sharp method is within 6 dB of the measured value in the 160 Hz 1/3 octave band and within less than 4 dB of the measured value over the rest of the frequency range.

The method of Sharp was used to calculate the sound reduction index of the modified common wall. The modified common wall was a double-leaf construction with 16 mm Fyreline gypsum board on one side and of 16 mm Fyreline gypsum board with three additional layers of 10 mm gypsum board on the other side for a total leaf thickness of 46 mm. The air gap was 88 mm and the distance between the metal studs was 450 mm. The
bending stiffness of the composite leaf was calculated based on Hamilton’s principle [41] to be 1.6 GPa. The total sound reduction index calculated using the method of Sharp and the equations in Annex B of EN12354 are compared in Figure K.2.

![Figure K.2: Comparison between results of calculating the sound reduction index of the modified common wall according to Annex B of EN12354 and the Sharp model.](image)

The critical frequency of the 16 mm Fyreline gypsum board leaf was calculated in the 2000 Hz 1/3 octave band but the critical frequency of the thicker leaf was in the 1000 Hz 1/3 octave band. The figure shows that the calculations are within 4 dB at the low frequencies, but that the Sharp model shows a different response above the 500 Hz 1/3 octave band. The calculated sound reduction index from the Sharp model was used in the calculation of the apparent sound reduction index between the rooms.

**K.3. Resonant Sound Reduction Index**

As discussed in Chapter 7, the choice of the method of calculating the sound reduction index can have a large affect on the predictions using the EN12354 method. The elements along the flanking paths were all calculated to have a critical frequency in the 2000 Hz 1/3 octave band and therefore the resonant component of the sound reduction index needed to be calculated below this frequency for use in the EN12354 predictions. The resonant sound
reduction indices of the elements were calculated according to the theory of Leppington [108].

The calculated resonant component of the sound reduction index of the flanking elements is compared to the total sound reduction index calculated according to the Sharp model and according to Annex B in Figure K.3.

![Figure K.3: Comparison between the total sound reduction index calculated according to the Sharp model and EN12354 and the resonant component calculated according to Leppington. The wall was a double-leaf construction with 16 mm Fyreline gypsum board on metal studs.](image)

Above the critical frequency, the total sound reduction index is dominated by the resonant component and the non-resonant component of the sound reduction index can be ignored [147]. Therefore, at the critical frequency and above the values of the total sound reduction index from the Sharp model were used for the values of the resonant component as shown in the figure.
**Appendix L: Field Testing Procedure**

**L.1. Introduction**

The field testing was conducted on week nights after 10PM or during the weekend to limit the sources of background noise and vibration in the building. The elevator next to E517 was shut down during the testing. The rooms used for the testing were modified by increasing the thickness of the common dividing wall and by adding fiberglass insulation above the suspended ceiling in room E517 as described in Appendix J.

**L.2. Preparation of the Rooms**

Sound absorbing material was fixed in front of the exterior windows in both rooms to reduce the airborne noise transmitted outside of the building. Gypsum board was fixed on the exterior of the windows between the rooms and the corridor. Sound absorbing material was fixed to the doors between the rooms and the corridor. The material on the doors needed to be removable so that the rooms could be accessed, but was always in place during the testing.

The desks, chairs and other furniture in the rooms were removed including the whiteboards on the walls. Kihlman [153] showed the importance of using diffusers when making sound pressure measurements in symmetric rooms. However, for this study diffusers were not added to the rooms since the rooms did not have the same volume or exactly the same dimensions.

**L.3. Sound Pressure Measurements**

A dodecahedron loudspeaker was placed in a corner of room E517. Five microphones were positioned on tripods in five positions in room E517. The following measurements were made:

- The sound pressure level in the room E517 was measured three times in five positions for forty five seconds.
- The reverberation time in room E517 was measured.
• The microphones were moved to E519 and the sound pressure level in room E519 due
to the source in E517 was measured.

The loudspeaker was moved to room E519.
• The sound pressure level in the room E519 was measured three times in five positions
for forty five seconds.
• The reverberation time in room E519 was measured.
• The microphones were moved to E517 and the sound pressure level in room E517 due
to the source in E519 was measured.

L.4. Sound Intensity Measurements

The dodecahedron loudspeaker was placed in a corner of the source room and the sound
intensity of all of the surfaces in the receiving room including the ceiling and floor were
measured. The intensity was measured by sweeping the probe back and forth approximately
10 cm from the element. Two sweeps were made, one up and down and one side to side.
The sets of measurement data were combined to create one measurement set. Three
measurement sets were made for each element for a total of six sweeps. If the sound intensity
index of the measurements was found to be large or if the measurements were not repeatable,
additional sweeps were made until the measurements were acceptable.

L.5. Velocity Level Difference Measurements

The velocity level difference for was measured using the following procedure.
• The electromagnetic shaker was connected to the first element of the flanking path using
a stinger as shown in Figure L.1
Figure L.1: Attachment of the electromagnetic shaker to the element via a stinger and an impedance head.

The figure shows the electromagnetic shaker mounted to a stand. A 3mm thick aluminum disk with a 10/32 threaded stud in the middle was glued to the wall. An impedance head was threaded onto the metal stud and a 2 mm diameter stinger connected the impedance head to the shaker head.

- The velocity of the first element was measured. If the first element was a double-leaf construction, a set of accelerometers was attached to the side of the element opposite from the excitation according to ISO10848-1 section 7.2.4. The accelerometers were attached to the element in random locations using beeswax. The surface velocity in each location was measured three times to ensure that the measurements were repeatable.

- The surface velocity of the second element of the flanking path was measured.

The above procedure was repeated three times for all of the elements included in the flanking paths considered in this study.
Appendix M: Calculation of the Apparent Sound Reduction Index

M.1. Predictions

The flanking sound reduction index of each flanking path is calculated according to EN12354-1 as:

\[ R_{ij,EN12354} = \frac{R_{Ri} + R_{Rj}}{2} + D_{v,ij,situ} + 10 \log \left( \frac{S_o}{\sqrt{S_i S_j}} \right) \]  \hspace{1cm} (M.1)

where \( R_{Ri} \) and \( R_{Rj} \) are the resonant components of the sound reduction index of elements \( i \) and \( j \), \( D_{v,ij} \) is the velocity level difference measured \textit{in situ}, \( S_o \) is the area of the separating element between the rooms and \( S_i \) and \( S_j \) are the areas of elements \( i \) and \( j \).

The total flanking sound reduction index for each element is calculated by summing the contributions from each flanking path. For example, for element D in the receiving room, the total flanking sound reduction index is:

\[ R_D = 10 \log \left[ 10^\left( \frac{-R_{Rd}}{10} \right) + 10^\left( \frac{-R_{Rd}}{10} \right) + 10^\left( \frac{-R_{R1d}}{10} \right) + 10^\left( \frac{-R_{R2d}}{10} \right) \right] \]  \hspace{1cm} (M.2)

Likewise, the apparent sound reduction index is calculated from the contributions from all of the flanking elements and from the direct transmission path. The direct transmission path includes both the resonant and the non-resonant components.

Since the calculation of the total flanking sound reduction index of each element and the apparent sound reduction index involve summing terms which have a log-normal PDF, the uncertainty of the calculations could not be determined using the method of GUM as explained in Chapter 3. Monte Carlo simulations were used instead to determine the uncertainty of the calculations.
Appendix M: Calculation of the Apparent Sound Reduction Index

M.2. Measured Data

M.2.1. Apparent Sound Reduction Index
Under the assumption of diffuse sound fields in the two rooms, the apparent sound reduction index may be calculated according to ISO140-4 [123]:

\[ R' = \bar{L}_1 - \bar{L}_2 + 10 \log \left( \frac{ST}{V} \right) \]  \hspace{1cm} (M.3)

where \( \bar{L}_1 \) and \( \bar{L}_2 \) are the average sound pressure level in the source and the receiver rooms, \( S \) is the area of the test specimen, \( T \) is the reverberation time in the receiving room and \( V \) is the volume in the receiving room.

M.2.2. Intensity Flanking Sound Reduction Index
The transmitted flanking noise from each element was measured using sound intensity in the source room according to ISO15186-2:2003 [119]. The intensity sound reduction index for flanking element \( j \) is defined as:

\[ R_{IF_j} = \bar{L}_{p1} - 6 - \bar{L}_{inj} + 10 \log \left( \frac{S}{S_{Mi}} \right) \]  \hspace{1cm} (M.4)

where \( \bar{L}_{p1} \) is the average sound pressure level in the source room, \( \bar{L}_{inj} \) is the average normal sound intensity level measured over the measurement surface for the flanking element \( j \) in the receiving room, \( S \) is the area of the separating building element and \( S_{Mi} \) is the total area of the measurement surface for the flanking element \( j \) in the receiving room. Eqn (M.4) differs from the equation for calculation of the intensity sound reduction index in ISO15186-1 in that it includes the ratio of the area of the separating element to the area of the flanking element.

M.2.3. Apparent Intensity Sound Reduction Index
The apparent sound reduction index is calculated from the intensity sound reduction index for each of the flanking elements according to ISO15186-2:2003 such that:
Appendix M: Calculation of the Apparent Sound Reduction Index

\[ R'_I = [L_p - 6 + 10 \log \left( \frac{S}{S_0} \right) - 10 \log \left( \sum_j S_{Mj} 10^{(0.1\tau_{n,j})} \right)] + K_C \]  

(M.5)

Where \( R'_I \) is the apparent intensity sound reduction index, \( S_o = 1 \text{ m}^2 \) and \( K_C \) is an adaptation term described in Annex A of ISO15186-2:2003. The standard states that the apparent intensity sound reduction calculated according to eqn (M.5) can be compared directly to the apparent sound reduction index measured according to ISO140-4 when none of the surfaces are shielded. The adaptation term \( K_C \) may be calculated as:

\[ K_C = 10 \log \left( 1 + \frac{S_{b2} A}{8V_2} \right) \]  

(M.6)

where \( S_{b2} \) is the area of all the boundary surfaces in the receiving room, \( \lambda \) is the wavelength of the mid-band frequency and \( V_2 \) is the volume of the receiving room. The term in the parenthesis is the Waterhouse correction [126]. The standard states that \( K_C \) is to be calculated if the traditional measurements according to ISO140 have been taken in a well-defined receiving room. If the receiving room is not well defined, values which may be used for \( K_C \) are listed in Annex A.

In addition, ISO15816-2:2003 includes an adaptation term \( K_2 \) to be applied to the calculation of the apparent intensity sound reduction index when it is to be compared versus measurements made according to ISO140-4. The purpose of the adaptation term \( K_2 \) is to reflect the biases between measurements made according to ISO151816 and ISO140. ISO15186-2 states that the bias depends on the receiving room and will not be the same for each situation. The standard is ambiguous in regard to how \( K_2 \) is to be determined. Instead, adaptation terms from repeat measurements on different wall constructions in different laboratories are presented in plot format. The choice of how to apply \( K_2 \) is left to the person conducting the test.
Appendix N: Equipment

N.1. List of Equipment Used for this Study

A list of the equipment used for this study is shown in Table N.1.

<table>
<thead>
<tr>
<th>Description</th>
<th>Manufacturer</th>
<th>Model</th>
<th>Serial Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Analyzer</td>
<td>Bruel &amp; Kjaer</td>
<td>PULSE C Frame with 7539 5 Channel Module</td>
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<td>Impact Hammer</td>
<td>PCB</td>
<td>T086C01</td>
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Table N.1: List of equipment with model numbers and serial numbers.
N.2. Defective Brüel & Kjær Accelerometers

In addition to the accelerometers listed above, the study also used the accelerometers shown in Table N.2 which proved to be defective.

<table>
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<tr>
<th>Manufacturer</th>
<th>Model</th>
<th>Serial Number</th>
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Table N.2: List of defective accelerometers.

Prior to testing, the accelerometers are calibrated by mounting the accelerometers onto the calibrator, calibrating, removing the accelerometer and then remounting in a different position. Due to the calibration procedure, it was discovered that the Brüel & Kjær 4517 accelerometers were very sensitive to mounting. Further testing was requested by the university and by Brüel & Kjær. In all, dozens of tests were conducted on the five accelerometers and each time the tests firmly showed that the acceleration measured by the accelerometers was not repeatable.

Two reasons for the problem were identified. The first was the very thin mounting surface of the accelerometer as shown in Figure N.1.

![Figure N.1: Schematic of the Brüel & Kjær 4517 accelerometers showing the thin mounting surface.](image-url)
The thin mounting surface made the accelerometer sensitive to mounting conditions. The author discovered that if the accelerometers were mounted upside down on the surface shown at the top of Figure N.1 the sensitivity issue was significantly reduced. In response to the sensitivity, Brüel & Kjær recommended that the accelerometer only be mounted to machined surfaces using a thin layer of special grease. However, this recommendation would be difficult to follow in any practical application of the accelerometer.

The second problem was a production issue with the accelerometers. According to a staff member at Brüel & Kjær in Denmark, the company has had issues with contamination in the clean room in which the accelerometers were manufactured. Initially, two accelerometers were sent back to Brüel & Kjær in Denmark around June 2007 and both were shown to be affected by contamination. In July 2007, Brüel & Kjær requested that the University of Canterbury return the rest of the accelerometers. The rest of the accelerometers were sent back in September 2007 for credit. The accelerometers were replaced with PCB accelerometers which were ordered in November 2007.

The problem with the Brüel & Kjær 4517 accelerometers required that all data measured prior to December 2007 be discarded. This included testing of L-panels and testing of the single panels including the series of damped panels used for the study of the separation of the resonant component of the velocity. Extensive retesting was carried out between December 2007 and February 2008.
Appendix O: List of Publications from this Study


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References


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[89] Das, C., Approximate Solution to the (Q,R) Inventory Model for Gamma Lead Time Demand, Management Science, 1976, 22(9), 1043-1047.


Appendix O: List of Publications from this Study


[108] Leppington, F. G., Heron, K. H., Broadbent, E. G., and Mead, S. M., Resonant and Non-Resonant Acoustic Properties of Elastic Panels. II. The Transmission Problem,
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