Low frequency flanking sound transmission in lightweight buildings

Wouter Meijers
Preface

This thesis is the graduation rapport to obtain the degree of Master of Science degree in Building Engineering with a specialization in structural design, at the Technical University in Delft at the faculty of Civil Engineering. Its contents are the result of nine months of research into the flanking sound transmission of low frequencies in lightweight buildings.

Although I thoroughly enjoyed my time as a student, the time has come to become an engineer. I would like to thank Ir. A. Broersma for offering me a place to work on my thesis, and the many laughs at the coffee table. I would also like to thank the rest of my graduation committee for evaluating my work, guidance and critical comments.

And thank you Francine, for your support and those pretty blue eyes.

Delft, May 2015

Wouter Meijers
Buildings are being designed that are lighter and more efficient than before. Lightweight structures have many advantages such as increased flexibility, sustainability, lower costs and faster building times. However, some inherent building characteristics that we have grown accustomed to are lost. One of those characteristics is the sound insulation provided by the building.

A large portion of the sound reduction comes from mass, which reflects the sound waves by having a higher specific impedance than the surrounding air. To obtain sufficient sound reduction in lightweight structures, a multitude of these impedance jumps are used, usually in the form of double leaf walls. Although this is an efficient way to increase the sound reduction over a large frequency range, it is less effective for low frequency sounds. An important prerequisite of such systems is that the layers are kept separated. Connections between layers are often needed for structural reasons, so there always exists a connection between the two that decreases the sound reduction capabilities of the system.

Sound transmission through connections and junctions is called flanking sound transmission. The sound path is longer than direct sound transmission, and involves multiple elements. Figure 1.1 illustrates the difference between direct sound transmission and flanking sound transmission. In lightweight structures, flanking sound can travel through the connections between separated layers, and effectively bypass part of the direct sound reduction. This makes flanking sound transmission a bigger problem for lightweight structures than for heavyweight structures. The formula for the flanking sound reduction index is given in Eq. (1.1).

\[
R_{ij} = 0.5R_i + 0.5R_j + D_{ij} + 10\log \left[ \frac{S_s}{S_i S_j} \right]
\]  

(1.1)

Where:

- \(R_i\) = Direct sound reduction index of element \(i\) [dB]
- \(R_j\) = Direct sound reduction index of element \(j\) [dB]
- \(D_{ij}\) = Junction reduction index in situ [dB]
- \(S_s\) = Surface area of separating element [m]
- \(S_i\) = Surface area of element \(i\) [m]
- \(S_j\) = Surface area of element \(j\) [m]

Figure 1.1 Sound transmission paths between two rooms. Source: http://www.nrc-cnrc.gc.ca/
It is therefore needed that junctions and connections in lightweight structures provide high sound reduction indices, to achieve sufficient sound insulation. Acoustic decoupling with mass-spring-mass systems involving rubbers and springs is used to increase the sound reduction indices of junctions. An important factor hereby is the resonance frequency of the rubber or spring, where frequencies closer to the resonance frequency are more easily transmitted.

Low frequencies are thus less attenuated by both the direct sound reduction part of the flanking sound reduction, and the junction reduction index part. Moreover, the wavelengths of low frequency sounds are larger than most building elements, making separation a less effective method to increase the sound reduction. Low frequency sound reduction is thus a larger problem for lightweight elements than for heavyweight elements.

To improve the overall sound reduction of lightweight elements and structures, it is thus important to decrease the flanking sound transmission of low frequency sounds. Preferably without impairing the structural system. Therefore it is tried to decrease the flanking sound transmission by making changes to the floor system.

Three floor variants were designed with the aim to reduce the bending wave propagation, the main wave form of flanking sound transmission. From the formula for bending wave impedance, Eq.(1.2), it appears that both stiffness and mass have influence. Where the stiffness is for a large part determined by the dimensions of the element. Increasing the difference in bending impedances between elements will decrease the propagation of bending waves, and thus decrease the flanking sound transmission.

\[ Z_{M0} = \frac{\sqrt{m} \sqrt{B^3}}{\sqrt{\omega}} \]  

With:
- \( Z_{M0} \) = Bending wave impedance [Ns/m]
- \( m \) = Mass per unit area [Kg/m²]
- \( B \) = Bending stiffness [Nm²]
- \( \omega \) = Angular frequency [Hz]

The variants were tested in the finite element program ANSYS, which is excellently suited for the evaluation of low frequency sounds. An abstract case was formulated in the form of a movie theater, from which a junction is extracted to be modelled. As validation, the junction was first modelled with homogenous slabs of concrete as floors, from which the results were verified with measurements found in the literature. This model was named the basic case, and is used to compare and evaluate the variants.

In the model, the velocity level difference between two floors was calculated. Which makes up the majority of the junction reduction, as can be seen in Eq.(1.1). Increasing the velocity level difference will thus decrease the flanking sound transmission. The frequency range in which these calculations were done is 25-125Hz. Any improvements in this region are assumed to also increase flanking sound reduction for the entire frequency range.

Table 1.1 Results of the variants from the finite element models. Table 1.1 shows an overview of the results from the basic case as well as the variants. Each variant improved the flanking sound reduction in a different way. Improvements of 15+ dB are quite large and might not be achieved in practice. However, a velocity level difference of around 40 dB should be more than enough for most applications.
Table 1.1 Results of the variants from the finite element models.

Initial results showed that every variant increased the velocity level difference compared to basic case. To understand where the improvements were coming from, a sensitivity study was conducted. Parameters that defined the variants, as well as material properties, were varied one by one around the initial set of values. From these calculations a number of interesting effects were observed.

The way that the model is excited, and where the plate velocity is calculated, greatly influences the velocity level difference. This effect is also observed in laboratory measurements. So, modelling has an uncertainty, that can be removed by separating the place of excitation and plate velocity measurement.

Velocity level differences vary over the frequency range, and show modal behavior. Peaks and dips alternate each other around an average value. This average value is not always constant over the frequency range, as is suggested by (NEN-EN12354-1, 2000). However, if the effects from each flanking path are combined, these peaks and dips might be averaged. The full effect of these peaks and dips is still unclear.

Mass has the same effects for more complex lightweight junction as for a simple heavy weight junctions. However, the effects that mass had on the resonance frequency, and thus the modal coupling of the elements, seemed to have more effect that the impedance jumps.

What already was predicted by Eq.(1.2), the stiffness had a significant influence on the flanking sound transmission. However, increasing the stiffness had an opposite effect as increasing the mass. Increasing the stiffness differences between elements created a larger bending impedance jump, and increased the velocity level difference. Adding impedance jumps in the floor element had positive effects as well. However, local optima were found, meaning that the added bending impedance jumps influenced each other.
By adding hinged connections in the flanking sound path, the velocity level difference was increased the most. Whether the bending impedance jump was created by the material that acted as a plastic hinge, or the interaction of forces in the connections, is unclear. Another interesting observation was that an increased number of hinged connections in the floor increased the velocity level difference even further. When the flanking sound transmission has to be reduced, multiple impedance jumps at the edge of floor and wall could be used.

Geometric differences had less direct effect than mass or stiffness differences. The dimension of the plate has influence on both the mass and stiffness, which have shown to have opposite effects. These effects cancel each other out. The dimensions do have a clear influence on the bending wave speed, which can make damping more effective. As a result, the modal behavior is damped and the velocity level difference is more constant. Adding damping in the system has the same effect.

It should be clear that a lot of factors influence the flanking sound transmission of low frequency sounds in lightweight buildings, besides mass and mass difference. This has led to the conclusion that it is possible for lightweight buildings to have sound reduction indexes as high as heavy weight buildings. However, this will require a complex design that might not be optimal for some applications. Lightweight elements make smart use of the characteristics that increase the sound reduction, and the low bending wave speed of small elements make low frequency sounds easier to attenuate. Making them potentially better than heavyweight elements.
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Figure 9.1 Schematized sound transmission through a wall
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<td>$A$</td>
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<td>Acceleration</td>
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<td>Bending stiffness</td>
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<td>$b$</td>
<td>Width</td>
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<td>Sound reduction index in coincidence controlled region</td>
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<tr>
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<td>$R_j$</td>
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<td>$\Delta R_j$</td>
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<tr>
<td>$R_f$</td>
<td>Sound reduction index for flanking sound transmission</td>
</tr>
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<td>Bending wave impedance</td>
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<tr>
<td>$&lt;\dot{\psi}&gt;$</td>
<td>Mean square normal surface vibration velocity</td>
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Greek alphabet

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CHAPTER 1. Introduction

Buildings are being designed that are lighter and more efficient than before. Lightweight structures have many advantages such as increased flexibility, sustainability, lower costs and faster building times. However, some inherent positive building characteristics that we have grown accustomed to are lost. One of those is the sound insulation provided by the building.

Lightweight structures have less inherent sound reduction than heavyweight structures, due to a lack of mass. Therefore lightweight structures make use of multiple smart impedance jumps to obtain sufficiently high sound reduction. This however creates a shortcutting effect for flanking sound transmission, which can severely diminish the sound reduction of elements. It is therefore needed that junctions and connections have high sound reduction indices. This is usually accomplished by acoustic decoupling with rubbers, springs or dilatations.

Low frequencies are hard to insulate for both heavyweight and lightweight structures. However, the smart impedance jumps that provide the sound insulation for lightweight structures is less effective for low frequencies sounds. This is because their wavelengths are comparable or larger than the building elements and their separation. If the flanking sound transmission of low frequency sounds could be reduced, this would increase the overall sound reduction of lightweight elements significantly.

1.1. Problem statement

Lightweight structures are considered to have lesser sound reduction qualities than their heavyweight counterparts. Therefore, the use of lightweight structures and elements are less used for buildings that have high sound reduction requirements. Building types such as homes, movie theaters and music halls make predominantly use of heavy building elements. If the sound reduction qualities of lightweight elements can be increased, such buildings could also be build lighter.

Especially for larger buildings, a lightweight structure with lightweight elements could provide lots of benefits. However, the separation of elements where the sound reduction of lightweight elements relies on, makes the building less stable. This creates an dilemma between the structural engineer, and the acoustic advisor. Structural elements that provide stability also connect rooms, making flanking sound transmission an even larger problem.

By increasing the flanking sound reduction of lightweight elements, the total sound reduction can be increased the most. Guidelines for flanking sound transmission stated in (NEN-EN12354-1, 2000) are based on statistical energy analyses, and focuses on heavy homogenous elements. This method is however not accurate for lightweight elements and low frequencies.

The amount and intensity of low frequencies sounds is also increasing. Heavy bass in music and movies, machinery and traffic are examples of low frequency sound sources. Low frequency sounds can be experienced as very annoying, and even lead to health issues. Increasing the need for low frequency sound insulation.
However, there seems to be a gap in the knowledge when it comes to flanking sound transmission of low frequency sounds in lightweight elements. To fill this gap, research has to be done to better understand the factors that define the flanking sound reduction. Moreover, solutions are researched that increase the flanking sound reduction, without causing interference with the structural design. The following research question is formed to achieve this:

*How can the flanking sound transmission be reduced in lightweight elements without impairing the structural system, so that the sound insulation of lightweight structures can be improved?*

### 1.2. Objectives

To answer the research question stated in the problem statement a number of objectives has been formulated to guide the process. This will roughly describe the outline of the thesis.

1. Investigating what acoustic properties of sound are important for the propagation and quantification of sound.
2. To determine what factors play a role in direct sound transmission, and how lightweight elements differ from their heavyweight counterparts.
3. Finding out how relevant flanking sound transmission is for the total sound insulation.
4. Applying acquired knowledge to design three floor variants to decrease the flanking sound transmission in lightweight structures.
5. Modelling the variants in a finite element program to evaluate the designs, and study the effects that affect the flanking sound reduction.
6. Drawing conclusions to improve the sound reduction capabilities of lightweight elements.

### 1.3. Methodology

The foundation of this thesis is the literature review. The literature review will be used to gain a basic understanding in the principles used in acoustics. This is necessary to understand what exactly provides the sound reduction of an element. Various papers, books and online knowledge centers were consulted and the relevant aspects concerning flanking sound transmission described.

The literature review is divided into three sections. Starting at the base, the basic principles and acoustic properties were studied. Secondly, the direct sound transmission of both heavy and lightweight elements was researched. These two sections were well documented by established scientific organizations. Therefore, no critical review is given over these subjects.

Thirdly, the literature review was used to study flanking sound transmission. Both the flanking transmission in a double leaf element as the flanking transmission between two rooms is discussed. For lightweight elements, both types are of interest.

A number of building physic consultancies were visited to find out how flanking sound transmission is dealt with in practice. Also, their design strategies and concerns about the subject were asked.

Three floor variants have been designed to increase the flanking sound reduction. These were tested in an abstract case, that simulates the conditions of a large movie theater room. The modelled junction consisted out of a beam, two floors and two double leaf walls. A sensitivity study was conducted by varying the parameters that define the variants around a reference case, and studying the effects.
Testing and modeling of the variants is done in the finite element program ANSYS. The finite element method is excellently suited for low frequency sounds, in contrast to programs based on statistical energy analysis. A basic junction, from which several measurements exist in the literature, was designed to validate the model.

From the model, the average surface velocity of the two floor was obtained. With these values, the velocity level difference could be calculated and evaluated. The variants were compared to each other and the basic case. In the conclusions the results are discussed.

1.4. Scope and limitations of the thesis

The research will focus on the flanking sound transmission of low frequency sounds. By doing so, it ignores a large part of the frequency range that is interesting for acoustics. However, according to various literature works, the junction reduction index is constant over the frequency. So, any results should be applicable for the entire frequency range.

Only first order flanking paths will be researched, as it is assumed that second and higher order flanking paths are effected by the same characteristics. Any improvements found should thus affect all flanking sound paths.

Although the focus lies on flanking sound transmission, part of the research is dedicated to direct sound transmission. Primarily because the direct sound transmission influences the flanking sound transmission, and secondly to understand how the direct sound transmission affects the design of lightweight elements.

Experiments will be conducted in a finite element program, without doing real world tests to confirm them. The finite element program itself has limitations that impact the model, as well as the results. Limitations concerning the model are the size and the needed simplifications. This means that only a small part of the building could be modelled, and that assumptions had to be made to reduce size of the model. Also, plate elements were used instead of solid elements to stay within the size limit of ANSYS.

Because of the size limit of the program, the radiation and excitations of the floors was not modelled. In reality these aspects are quite important. Instead, a harmonic pressure was applied to the floor.

For the limitations of the program itself, the calculated results are still based on formulas that were manmade. This means any results are still only an approximations of the reality. Also, because of these formula’s, restrictions in the program prohibited certain configurations.

The implications of the variants on the direct sound transmission are not researched. They are however briefly discussed in the conclusion.

Time limitations will make it impossible to study all the effects, that is why only a limited amount of parameters is varied over a certain range.
1.5. Overview of the contents and structure of the thesis

In chapter 2 the basic principles of sound are discussed. First sound in general, sound propagation and acoustics are discussed. This will then be related to the human hearing range, and to what is considered noise annoyance. Quantification of sound and the implications for low frequency sounds are discussed as well.

The direction sound transmission for the entire frequency range is discussed in chapter 3. Followed by the direct sound transmission of lightweight double leaf walls. Issues concerning the sound reduction of low frequency sounds will become clear.

Chapter 4 will consist of the theory for flanking sound transmission. First the sound insulation in general is discussed, and the relevance of flanking sound transmission will be established. A small note on how low frequency differs from these theories is made before continuing into more detail of flanking sound. Effects such as resonance and vibration reduction coefficients will be explained next.

A short summary of the sound insulation techniques and their effects will be given in chapter 5. Also, some new innovative ideas are discussed that could be adapted for the variants in chapter 6. The results from the interviews with the building physic consultancies will be presented as well.

In chapter 6 the knowledge of chapter 3 - 5 is applied to design floor variants that reduce flanking sound transmission. The abstract case is described to provide boundary conditions, as well as to create a realistic situation.

Chapter 7 will describe the setup of the ANSYS model, as well as the design choices and assumptions that had to be made due to the limitations of ANSYS. Results from the basic case are presented here.

In chapter 8 there results for the variants will be given and discussed. The effects of the characteristics that define the flanking sound reduction are discussed as well.

The conclusions and recommendations are given in chapter 9.
CHAPTER 2. General sound principles

In this chapter a brief summary is given of general sound principles that will be used throughout this thesis. Besides describing the theoretical basis of propagating sound, the quantification is discussed as well. This is important to understand the results of the experiments, in chapter 7 and 8.

2.1. Sound propagation

Sound can be described as a propagating vibration in a medium. This medium can be a gas such as air, but also a liquid or a solid. A vibrating object produces sound by moving the surrounding medium around it. Particles that where pushed away collide with other particles and the vibration is passed along. When the object flexes back a zone is created with low pressure, pulling the particles back. This is called rarefaction. The vibration is passed along by these pressure fluctuations, and a sound wave is formed, see Figure 2.1.

These sound waves are usually illustrated as longitudinal waves, where the particles and the propagating sound move in the same direction. Longitudinal waves can form in gasses, liquids and solids.

In transverse waves the particles move perpendicular to the direction of the wave. An example are the strings of a guitar when they are excited by a plectrum, see Figure 2.2. It is clear that the strings can only move sideways, and not in the direction of the material. For transverse waves to exist, a returning force is needed to the relaxed state is needed. In most materials, this is provided by the transverse contraction or Poisson’s ratio. Therefore, transverse waves can only propagate in solids, and not in liquids and gasses.
2.1.1. Sound waves and frequency

Sound waves can be described by considering the amplitude and wavelength, where the amplitude defines the loudness and the wavelength defines the frequency. Humans experience the latter as pitch, where high frequencies are high notes and low frequencies are low notes. Sound frequency is defined as the number of vibrations per second. In a formula form:

\[ f = \frac{c}{\lambda} \]  

(2.1)

With:
- \( f \) = Frequency \([1/s]\)
- \( c \) = Wave velocity \([m/s]\)
- \( \lambda \) = Wave length \([m]\)

Notice that for a constant sound speed, lower frequencies will have longer wavelengths. To put this in perspective, a frequency of 100 Hz will have a wavelength of 34 meter in air. It also means that the amount of pressure fluctuations per meter is low, as can be seen in Figure 2.3. This phenomena makes low frequencies harder to insulate, dissipate and measure, compared to higher frequencies.

![High Frequency Wave](image1)

![Low Frequency Wave](image2)

Figure 2.3 High and low frequencies. Taken from: [http://www.physicsclassroom.com/](http://www.physicsclassroom.com/)

The wave speed, or speed of sound, varies depending on the medium and can be found with Eq.(2.2). For building acoustics a temperature of 20 °C is usually taken, so that the density is constant. The coefficient of stiffness can vary for different wave forms, materials and boundary condition. However, the stiffness of air inside a room is constant, resulting in a sound speed of roughly 343 m/s.

\[ c = \sqrt{\frac{K}{\rho}} \]  

(2.2)

With:
- \( K \) = Coefficient of stiffness \([N/m^2]\)
- \( \rho \) = Density \([kg/m^3]\)

Another value regularly used to describe waves is the (angular) wavenumber. This value is useful for discussing aspects related to spatial variations caused by sound and vibrations fields.
\[ k = \frac{\omega}{c} = \frac{2\pi}{\lambda} \]  

(2.3)

Where:
- \( k \) = Angular Wavenumber  \([1/m]\)
- \( \omega \) = Angular frequency \((2\pi f)\) \([1/s]\)

For sound waves the principle of superposition holds, meaning that individual sound waves can be summed up into one wave. Figure 2.4 illustrates this principle and also shows that sound waves can amplify or diminish each other, also called destructive and constructive interference. Basically every sound you hear is a combination of different frequencies and amplitudes, resulting in a complex wave. For sound insulation evaluation purposes this does not matter, and individual frequencies can be used for calculations.

Figure 2.4 Principle of superposition for sound waves. Taken from: www.boundless.com

So far the sound waves considered have been one-dimensional. To evaluate sound in three dimensions the following equation in a xyz-plane can be used.

\[
\frac{\Delta^2 P}{\Delta x^2} + \frac{\Delta^2 P}{\Delta y^2} + \frac{\Delta^2 P}{\Delta z^2} - \frac{1}{c_0^2} \frac{\Delta^2 P}{\Delta t^2} = 0
\]  

(2.4)

With:
- \( P \) = Pressure \([N/m^2]\)
- \( t \) = Time \([s]\)

From this equation the wavenumber, \( k \), can be related to one-dimensional wavenumbers in every direction according to the following equation.

\[ k^2 = k_x^2 + k_y^2 + k_z^2 \]  

(2.5)

The pressure from a sound wave at a distance \( x \) from the source can now be described by the following formula:

\[ P(x, t) = \hat{\rho} e^{i\omega t} e^{ikx} \]  

(2.6)

With:
- \( \hat{\rho} \) = Amplitude of pressure wave \([N/m^2]\)
- \( i \) = \( \sqrt{-1} \) \([-]\)
- \( k \) = Wavenumber \([1/m]\)
- \( \omega \) = Angular frequency \([1/s]\)
2.1.1. Bending-, transverse and quasi-longitudinal waves in solids

Three important types of sound waves can be distinguished in solids. Longitudinal waves are most important for direct sound transmission, while bending and transverse waves are dominant for flanking sound transmission. In Figure 2.5 the three wave types are illustrated with greatly exaggerated deflections.

Longitudinal waves can only exist in elements that are much larger than the wavelength. Since the wave speed in solids is higher than in air, elements are usually smaller than the wavelength. However, a different type of longitudinal wave can exist in building elements called quasi-longitudinal waves. This waveform differs from longitudinal waves in air by having a shear modulus. When the wave compresses the element in the direction of the wave, particles will move normal to the propagation of the wave (similar effects can be observed by compressing a banana). This effect increases the stiffness and changes the wave speed. The following formulas describe the longitudinal wave speed in common building elements:

\[
c_{L,b} = \frac{E}{\sqrt{\rho}} \tag{2.7}
\]

\[
c_{L,p} = \frac{E}{\sqrt{\rho(1 - v^2)}} \tag{2.8}
\]
With:

\[ E = \text{Young's modulus} \quad [\text{N/m}^2] \]
\[ \rho = \text{Density} \quad [\text{kg/m}^3] \]
\[ \nu = \text{Poisson's ratio} \quad [-] \]
\[ c_{lb} = \text{Longitudinal wave speed in a beam} \quad [\text{m/s}] \]
\[ c_{lp} = \text{Longitudinal wave speed in a plate} \quad [\text{m/s}] \]

Shear waves, also known as transverse waves, propagate through shear deformations in a solid medium. The speed of sound is now related to the shear modulus.

\[ G = \frac{E}{2(1 + \nu)} \quad (2.9) \]
\[ c_{t,b} = \sqrt{\frac{T}{\rho I_b}} \quad (2.10) \]
\[ c_{t,p} = \sqrt{\frac{G}{\rho}} \quad (2.11) \]

With:

\[ G = \text{Shear modulus} \quad [\text{N/m}^2] \]
\[ T = \text{Torsional stiffness} \quad [\text{Nm}^2] \]
\[ I_b = \text{Polar moment of inertia} \quad [\text{m}^4] \]
\[ c_{t,b} = \text{Torsional wave speed in a beam} \quad [\text{m/s}] \]
\[ c_{t,p} = \text{Transverse wave speed in a plate} \quad [\text{m/s}] \]

Bending waves are caused by transverse deflection of structural elements, and occur when the dimensions of the element are small compared to the wavelength. Longitudinal and shear waves, however, do not occur in small elements compared to their wavelengths. This implies that relatively thin elements such as walls are predominantly susceptible to bending waves. The particle velocity is normal to the plate and the direction of propagation, which means it can radiate sound effectively.

The bending wave velocity is related to the bending stiffness of the element, but is also frequency dependent, in contrast to longitudinal and shear waves. Since the wavenumber is depended on the wave velocity, the formula for the wavenumber is redefined.

\[ c_b = 4 \frac{B}{m} \cdot \sqrt{\omega} \quad (2.12) \]
\[ k_B = 4 \frac{m}{B} \cdot \sqrt{\omega} \quad (2.13) \]
\[ B = \frac{E}{1 - \nu^2} I \quad (2.14) \]

With:

\[ \omega = \text{Angular frequency} \quad [1/\text{s}] \]
\[ B = \text{Bending stiffness} \quad [\text{Nm}^2] \]
\[ m = \text{Mass per unit area} \quad [\text{kg/m}^2] \]
\[ I = \text{Moment of inertia for cross section} \quad [\text{m}^4] \]
\[ k_s = \text{Bending wavenumber} \quad [1/\text{m}] \]
As the wave speed is frequency dependent, higher frequencies will travel faster and outrun lower frequencies, see Figure 2.6. Sound thus gets deformed when transmitted through bending waves, resulting in the typical “muffled” sound when the source is covered or shielded by solid matter. Also note that the bending wave speed for low frequencies is relatively low compared to longitudinal wave forms, and that the wavelength is smaller as well.

Above equations describes the wave speed only for isotropic plates and beams, for anisotropic plates the stiffness in each direction has to be calculated to obtain the correct wave speeds. This is different for every case, and should be supported by experiments to obtain the correct equivalent stiffness’s.

![Figure 2.6 Bending wave speed of a steel plate with a thickness of 1, 2.5, and 10 mm. The horizontal line represents the wave speed in air.](image)

### 2.2. Sound distribution

To describe the propagation of sound waves in a room with formulas, a monopole source is used. Which is a source that radiates sound equally in every direction. For low frequencies, a monopole or dipole describes the sound source the most accurately. An illustration of a monopole sound source can be found in Figure 2.7.

![Figure 2.7 2D Illustration of an acoustic monopole. Source: www.southampton.ac.uk](image)

The sound waves propagate in every direction, distributing its energy over an increasing area. Assuming the monopole sound source produces a given intensity $W$, the effective sound pressure $P$ at a distance $r$ can be described by the following formula.
\[ P = \frac{\rho c W}{4\pi r^2} \]  

(2.15)

Where:

\( \rho \) = Density [kg/m\(^3\)] for air (20 °C) this is 1.204 kg/m\(^3\)

\( c \) = Wave velocity [m/s] for air (20 °C) this is 343.21 m/s

When a sound wave comes across an object that is relevant to its wave length, it is partially reflected back, and partially absorbed. The reflected sound wave will travel further until it hits another object. This process repeats itself until the sound wave is completely absorbed. Figure 2.8 illustrates reflected sound waves. Note that there are in fact an infinite amount of reflections.

![Figure 2.8 Direct and reflective sound waves. Taken from: http://continuingeducation.construction.com](image)

By absorbing sound waves, the element itself is set into vibration. Part of the vibration is converted into heat by the element, depending on the amount of damping. The other part is transmitted by radiating the vibration into the air, as can be seen in Figure 2.9. When improving the sound reduction, it is sought after to transmit as little sound energy through an element as possible.

![Figure 2.9 Sound transmission, absorption and reflection](image)

### 2.2.1. Acoustic impedance

For the evaluation of sound absorption and transmission it is important to know what sound does at a boundary. Acoustic impedance describes the relation between an applied pressure and the resulting flow in a medium, as Eq. (2.16). This quantity is used most to describe the behavior of
sound waves at a boundary. Large differences in acoustic impedances at a boundary results in reflected sound waves, while a low difference will transmit most of the sound waves.

\[ Z_a = \frac{P}{u} \]  
\[ (2.16) \]

When it is assumed that the wave impinging on the wall is a simple one dimensional wave the equation becomes:

\[ Z_{a,s} = \rho c \]  
\[ (2.17) \]

With:
- \( P \) = Pressure \([N/m^2]\)
- \( u \) = Particle velocity \([m/s]\)
- \( Z_{a,s} \) = Specific acoustic impedance \([Ns/m^3]\)
- \( c \) = Wave speed \([m/s]\)
- \( \rho \) = Density \([kg/m^3]\)

Using the expression for a spherical pressure wave from Eq.(2.6), the equation for the acoustic impedance becomes, according to (Vigran, 2008):

\[ Z_s = \rho c \frac{i k r}{1 + i k r} \]  
\[ (2.18) \]

With:
- \( i = \sqrt{-1} \) \([-\]
- \( r \) = Distance from origin \([m]\)

Here, the impedance is based on the complex wavenumber and distance. The impedance is no longer linear for a (wall)surface, but has a phase difference between sound pressure and velocity. When the distance or the frequency is large enough \((kr>>1)\), the acoustic impedance approaches the specific acoustic impedance \(\rho c\). This simplification is used in many derivations, resulting in a simpler wave model.

### 2.2.2. Absorption

When a sound reaches a boundary, it is partially reflected because the acoustic impedance of the two mediums is different. The amount of sound energy that is absorbed is defined as the absorption coefficient \(\alpha\). This coefficient can vary between 0 and 1 for elements, where a value of one represents 100% absorption.

From the absorbed sound energy, a part is transmitted and a part is dissipated. Materials with high absorbing coefficients are often used to dissipate the sound, preventing effects such as echoes. Three commonly used type of absorbers are briefly discussed and presented in Figure 2.10. As can be seen, lower frequencies are often absorbed less, but also dissipated less than higher frequencies.
Porous materials are often good absorbers. The amount of absorption depends on the permeability of the material. Air can flow into the canals and the material creates friction that dissipates the sound into heat. The size of the pores are critical for the absorption rate, too small pores do not let enough air through and too wide pores do not create enough friction. Fibrous material such as glass wool are good absorbers, with long canals and a flow resistance between 10 and 50 rayls/mm, depending on the thickness.

Panel absorbers create friction by restraining the vibration of the panel. They have a low range of frequencies in which they are effective, namely around the resonance frequency. It thus has a very specific absorbing function.

Helmholtz resonators are panels with apertures in them and a cavity of air behind that. Pressure from sound waves in the apertures compresses the air in the cavity. The air in the cavity then acts as a spring moving air at a certain frequency. The dimensions of the air cavity determine the resonance frequency. Putting absorption material in the cavity or aperture increases the amount of absorbed sound energy and the frequency range in which it is effective.

If a reflective surface is placed behind an absorbing material the reflected sound waves travel back through the absorbing material. This increases the effectiveness of the absorbing material by letting the sound wave pass by twice.

### 2.2.1. Transmission

Sound energy that is not dissipated into heat by the material, is transmitted. The transmitted can then be radiated into the next room. Limiting the amount of transmitted and radiated sound energy is the main goal of sound reduction.

Sound absorption is often confused with sound reduction. While sound absorption affects the sound level and acoustics inside a room, it does little to nothing for the sound reduction from one room to another. Absorbing materials are used to dissipate sound into heat, usually in small increments. Reflection is used to reduce the amount of sound transmitted, and thus increase the sound insulation.
2.2.1. Reflection

Sound that is not absorbed is reflected back into the room. Hard and smooth surfaces are highly reflective. Also, the mass of the element increases the reflective capabilities of the element by increasing the acoustic impedance of the material. The larger the impedance difference, or 'impedance jump', the more sound is reflected and the higher the sound reduction is.

Depending on the surface and the wavelength of the sound, impinging sound waves are either scattered or mirrored. If the wavelength is much larger than the imperfection of a surface the sound is reflected as Snell’s law describes. Figure 2.11 illustrates the reflection of a sound wave for smooth and rough surfaces. The same is true for the entire element, where frequencies with large wavelengths bend around small objects and higher frequencies are reflected. This principle is called diffraction.

![Figure 2.11 Diffuse and non-diffuse reflection. Taken from: http://usra.ca/echoreflection.php](image)

Reflected waves are an important aspect of acoustics, and can make it easier to understand each other in a room. The reflected waves that are registered up to 50-80ms after the direct sound wave are considered early reflections and are heard as direct sound. Later reflections can hinder intelligibility and are experienced as echoes if they are loud enough.

![Figure 2.12 Early and late reflections Source: http://www.moultonlabs.com/](image)

Reflected sound waves create a diffuse sound field in a room, which increases the intelligibility. It also makes it possible to hear someone at a distance by increasing the total sound pressure in the 50ms timeframe. However, too much reflection results in long fading times of sound, making
intelligibility worse. It becomes harder to distinguish different sounds, and everything becomes a blur. Locally high reflections can form echoes, for example in water wells. Figure 2.12 illustrates early and late reflections.

2.2.1. Diffuse and non-diffuse sound

Figure 2.7 shows a non-diffuse sound field where the sound waves are equally distributed in space but vary in intensity. When sound waves hit objects or walls, they reflect and scatter, resulting in reflected sound waves in every direction. The created sound field from direct and reflected waves is called diffuse, which means the relation between the produced sound and the resulting sound pressure from Eq. (2.15) is no longer correct. However, by adding a correction for the reverberation, the equation can still be used.

Diffuse sound is wanted in rooms because it equalizes the sound level in the room and increases intelligibility. It also increases the sound quality by reducing the chance for standing waves and distinct echoes in a room. In most cases the furniture and walls of a room are enough to create a diffuse sound field.

A diffuse sound field simplifies the calculation for the absorption and transmission of a wall. It is therefore assumed that the sound waves come from every direction and are equal in strength. This assumption leads to an easy way to evaluate omnidirectional incident sound, by taking the sum of incident sound waves over half a sphere.

2.2.2. Reverberation time

The reverberation time is defined as the time it takes for a sound to diminish 60 dB(A). Sabine derived an expression for the reverberation time in the 1890's for a cubic room with equally spread absorption. Although the formula is not entirely accurate, it is widely used because the error is small compared to measurement errors. Moreover, Sabine assumed a perfect diffuse sound field. For large rooms and rooms where absorption is not equally distributed, this may not be the case and other formulas are needed.

\[
T = \frac{V}{6A}
\]

(2.19)

With:
- \( T \) = Reverberation time [s]
- \( V \) = Volume of the room \([m^3]\]
- \( A \) = Total absorbing area \([m^2 Sabin]\]
- 6 = Simplification \([s/m]\]

Sabine’s formula is used in the calculation of sound reduction because there has to be compensated for the increased sound level in the room due reverberation. It is also used to describe the acoustic quality of a room, relating reverberation to intelligibility, warmness and loudness.

Movie theaters have large rooms where speech and music are combined. In order to provide the best sound experience, a diffuse sound field is created with a lot of absorption and relative low reverberation time. This also ensures that the sound level throughout the room is consistent, which is equally important for the listener.
2.3. Quantification of sound

To put the results of the formulas and experiments into perspective, the quantification of sound is briefly discussed.

2.3.1. Sound levels in Decibel

In a previous chapter the loudness of a wave was defined by the amplitude. A higher amplitude means a higher effective pressure. This is the result of a larger vibrations induced by a greater vibration of the sound origin. Figure 2.13 illustrates this principle. Compression of particles can be seen as potential energy that is passed along the direction of the wave, spreading out more the further away from the source. To quantify the amount of pressure of a sound wave, the sound intensity is defined as power over area.

![Diagram of sound waves and pressure fluctuations](http://www.sarahtulga.com/panpipes.htm)

Figure 2.13 High and low amplitude of a sound wave. Taken from: http://www.sarahtulga.com/panpipes.htm

The human ear registers pressure fluctuations as sound. The lowest difference recognizable by the ear is 20 micro Pascal, which translates to an intensity of $1 \times 10^{-12}$ Watt/m². Also known as the threshold of hearing. At a pressure of 200 Pascal, or an intensity of $1 \times 10^1$ Watt/m², the threshold of pain is reached. Since this range is very large, a logarithmic scale is introduced called decibel. The threshold of hearing is used as reference intensity and is defined as 0 dB. Every 10 dB increase means an intensity increase of factor 10.

Figure 2.14 show the hearing threshold for humans over the frequency range. Humans can hear sound between roughly 20hz and 20khz. In Figure 2.14, phon is used as a measurement for loudness, which can be related to decibel. It is clear that we do not perceive a 125hz 50 dB sound equally loud as a 4000hz 50dB sound.
This non-linear relation between perceived loudness and pressure difference from sound waves makes it impossible to give a 1-number value to sound. That is why several weighted curves have been made to relate sound to perceived loudness. In Figure 2.15 a few of those curves are presented. The most used curve is the A-weighted curve. Applying this weighting to a sound results in an almost equal loudness across the frequency range. The notation for sounds that are weighted with the A-curve is dB(a).
2.3.2. Hearing and noise annoyance

To create a feeling for the loudness of a sound, Table 2.1 gives the dB(A) values for a variety of sounds. As a rough estimate, it could be said that an increase of 10 dB(A) is experienced as twice as loud. This makes a motorcycle 8 times as loud as a normal conversation, and a rock concert 32 times louder than normal conversation.

Noise annoyance is experienced when a sound is louder than the activity one is listening to. In order to diminish the annoyance, the sound you are listening to has to be amplified or the annoying sound has to be reduced. In most cases the latter is wanted. Between most residences a sound reduction of 52 dB(A) is required according to NEN1070.

<table>
<thead>
<tr>
<th>Source</th>
<th>dB(A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Threshold of hearing</td>
<td>0</td>
</tr>
<tr>
<td>Breathing</td>
<td>10</td>
</tr>
<tr>
<td>Whisper</td>
<td>20</td>
</tr>
<tr>
<td>Background noise quiet room</td>
<td>30</td>
</tr>
<tr>
<td>Sound of a fridge</td>
<td>40</td>
</tr>
<tr>
<td>Small office</td>
<td>50</td>
</tr>
<tr>
<td>Normal conversation</td>
<td>60</td>
</tr>
<tr>
<td>Vacuum cleaner</td>
<td>70</td>
</tr>
<tr>
<td>Busy traffic road</td>
<td>80</td>
</tr>
<tr>
<td>Motorcycle</td>
<td>90</td>
</tr>
<tr>
<td>MP3 player at full volume</td>
<td>100</td>
</tr>
<tr>
<td>Front row rock concert</td>
<td>110</td>
</tr>
<tr>
<td>Thunder clap</td>
<td>120</td>
</tr>
<tr>
<td>Threshold of pain</td>
<td>130</td>
</tr>
<tr>
<td>Yet aircraft at 50m distance</td>
<td>140</td>
</tr>
</tbody>
</table>

Table 2.1 A-weighted decibel levels for a variety of sounds

2.3.3. Low frequency sounds in movie theaters and other places

Low frequency sounds are somewhat modern in the sense that these sounds have become more frequent over the years. Machinery and installations only recently started producing low frequency sounds. An example found in most homes is the washing machine or the fridge. Also, music has evolved and contains more low frequency sounds. Music styles such as pop, rock and house are examples. Newer and bigger sound installations make it possible to produce harder and heavier beats by producing louder low frequency sounds. Movie makers have a wider arsenal of sound effects available due computers, and these effects are widely used in explosions, impacts and space scenes. The last two sources can also be found in homes, with increasingly better sound systems becoming available for low prices.

There is a special niche of noise annoyance complaints, where people hear low frequency sounds all day long from factories or installations kilometers away. Because these sounds are so low frequency, they can even be felt as vibrations by some people. Most of these people feel miserable because of this. It should be noted that these people are more sensitive to low frequency sounds.

Since there were used to be less low frequency sounds, most older buildings were not designed to insulate these sound well. That is partially the reason why noise annoyance is dominated by low
frequency sounds. The other reason being, that low frequencies are harder to insulate than other frequencies.

It also means that new buildings have to take into account to insulate low frequency sounds. And maybe, with an eye on the future, add some extra insulation to compensate for a possible increase of low frequency sounds. Particularly because chest vibrations due low frequency sounds are experienced as enjoyable in concerts nowadays, and perhaps also in movies in the future.

Building types were low frequency sounds are dominant, are music halls, factories and movies. Contact sound such as footsteps are also low frequency, and are experienced most often in residential buildings. Contact sound is somewhat harder to control, especially in simple buildings such as homes.

Low frequency sounds in movies are heavily used by film directors to create that epic feel in a scene. Frequencies between 50-100 Hz are most used to create these effects. The regulations only specify an A-weighted average of 85 dB, with 30 dB headroom. This means low frequency sounds can become very intense, and even vibrate the chairs of the viewers. Often those sound effects are also heard in adjacent rooms.
CHAPTER 3. Direct sound transmission

The total apparent sound reduction between two rooms is determined by the sum of direct and flanking sound transmission, as described in Eq. (3.1) from (NEN-EN12354-1, 2000). To improve the total apparent sound reduction it is thus important to improve both the direct and flanking contributions.

$$R' = -10 \log \left[ 10^{\frac{R_d}{10}} + \sum 10^{\frac{R_f}{10}} \right]$$  \hspace{1cm} (3.1)

Where :

- $\text{R}_d$ = Sound reduction index for direct sound transmission \hspace{1cm} [dB]
- $\text{R}_f$ = Sound reduction indexes for flanking sound transmission \hspace{1cm} [dB]

If a path has a significantly lower sound reduction index, it becomes the dominant path. Limiting these paths will give the largest improvement in sound reduction. As an example, Table 3.1 shows the total apparent sound reduction index for three scenarios with varying direct sound reduction indexes. It also illustrates the how difficult it is to achieve a high sound reduction, and the limitations when flanking and direct sound transmissions contributions are relatively equal.

<table>
<thead>
<tr>
<th></th>
<th>Example 1</th>
<th>Example 2</th>
<th>Example 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{R}_d$</td>
<td>50 dB</td>
<td>40 dB</td>
<td>30 dB</td>
</tr>
<tr>
<td>$\text{R}_{f1}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f2}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f3}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f4}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f5}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f6}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f7}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f8}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f9}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f10}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f11}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>$\text{R}_{f12}$</td>
<td>50 dB</td>
<td>50 dB</td>
<td>50 dB</td>
</tr>
<tr>
<td>Total apparent sound reduction:</td>
<td>38.86 dB</td>
<td>36.58 dB</td>
<td>29.51 dB</td>
</tr>
</tbody>
</table>

Table 3.1 Example of total apparent sound reduction index

Most of the formulas for sound reduction values are based on heavyweight structures. As will become apparent in this chapter, lightweight structures obtain their (direct) sound reduction in a different way than heavyweight structures. This has consequences for the flanking sound reduction, which will be discussed in the next chapter.

3.1. Sound insulation principles

As explained in the previous chapter, sound reduction relies on increasing the reflection and reducing the radiation of an element. High reflection is obtained by large differences in acoustic impedance, for example with hard dense surfaces. Absorbing materials are used to dissipate sound energy, influence the acoustics inside a room or as damping in cavity.
The sound reduction of a homogenous wall varies over the frequency range, and can be divided into three regions (See Figure 3.1). Below the resonance frequency of a wall the sound reduction is governed by stiffness (region 1). Above the resonance frequency, the sound reduction is governed by mass (region 2), up to half of the coincidence frequency. Above that, the sound reduction is also controlled by damping (region 3). Although the formulas describing these effects should be seen as predictions, only small differences have been observed in comparison with measurements.

![Figure 3.1 Extended Mass law. Taken from: http://personal.inet.fi/koti/juhladude/soundproofing.html](http://personal.inet.fi/koti/juhladude/soundproofing.html)

Important frequencies for building acoustics lie between 125 and 4000Hz, which is comparable to our hearing range. This area corresponds roughly to the mass controlled area, and the coincidence controlled area. However, the sound reduction at lower frequencies is significantly less, making this the problem area.

3.2. **Resonance controlled region**

When the wavelength of a sound is much larger than the wall, the wall can be seen as very thin. As a result, the thickness of the element is not considered to be relevant for the total sound insulation, and will not be found in the derivation (directly). Very low frequencies, with large wavelengths, are only reduced by the stiffness of a wall, as can be seen in Figure 3.2. This dependence is finite, but has some overlap with the resonance controlled area. In the following derivation the sound vibrations inside the plate are neglected.
In order to obtain the sound insulation value, the air pressure ratio between the origin and receiving room is needed. Following the derivation of (Hassan, 2009), the expression for the sound pressure:

\[ P_1(x, t) = \hat{p}_1 e^{i\omega t} e^{-ik_1x} + \hat{p}_1 e^{i\omega t} e^{+ik_1x} \quad (3.2) \]
\[ P_2(x, t) = \hat{p}_2 e^{i\omega t} e^{-ik_2x} \quad (3.3) \]

And the particle velocity:

\[ v_1(x, t) = \frac{1}{Z_1} (\hat{p}_1 e^{i\omega t} e^{-ik_1x} - \hat{p}_1 e^{i\omega t} e^{+ik_1x}) \quad (3.4) \]
\[ v_2(x, t) = \frac{1}{Z_2} (\hat{p}_2 e^{i\omega t} e^{-ik_2x}) \quad (3.5) \]

Where:

- \( Z_1 = Z_3 \) = Acoustic impedance of air \([\text{Ns/m}^2]\)
- \( k_1 = k_2 \) = Wavenumber \([1/\text{m}]\)
- \( i = \sqrt{-1} \) \([-]\)
- \( \omega \) = Angular frequency \([1/\text{s}]\)
- \( t \) = time \([\text{s}]\)
- \( \hat{p} \) = Amplitude of pressure wave \([\text{N/m}^2]\)

At the boundary of the plate the particle velocities are equal to the velocity of the plate. So the system can be described as a mass-spring system. The motion of the entire system can then be rewritten to:

\[ F(t) = m \frac{dV}{dt} + r_m V(t) + k_s \int V(t) dt \quad (3.6) \]

With:

- \( r_m \) = Coefficient of viscous damping \([\text{Ns/m}]\)
- \( m \) = Mass \([\text{kg}]\)
- \( F \) = Applied force \([\text{N}]\)
- \( k_s \) = Stiffness \([\text{N/m}]\)
- \( V(t) \) = Plate velocity \([\text{m/s}]\)
Assuming the applied load is sinusoidal and the mass and damping have little effect in this frequency range compared to the stiffness, the relation between the sound pressure on each side of the thin plate can be obtained. A complete derivation can be found in the appendix.

\[
\frac{p_{2+}}{p_{1+}} = \frac{1 - \frac{ik_s}{2\omega\rho_0 c}}{1 + \left(\frac{k_s}{2\omega\rho_0 c}\right)^2} \tag{3.7}
\]

And the sound power transmission coefficient:

\[
\tau_n = \frac{1}{1 + \left(\frac{k_s}{2\omega\rho_0 c}\right)^2} \tag{3.8}
\]

And with the angle of incidence, \( \varphi \):

\[
\tau(\theta) = \frac{1}{1 + \left(\cos\theta \frac{k_s}{2\omega\rho_0 c}\right)^2} \tag{3.9}
\]

The stiffness for a rectangular plate is given by the following formula:

\[
k_s = \frac{\pi^8 E d^3}{768(1 - \nu^2)} \left(\frac{1}{b^2} + \frac{1}{l^2}\right)^2 \tag{3.10}
\]

With:

- \( E \) = Young's Modulus [N/m²]
- \( \nu \) = Poisson’s ratio [-]
- \( d \) = Thickness [m]
- \( b \) = Width [m]
- \( l \) = Length [m]
- \( \theta \) = Angle [rad]

Eq. (3.8) assumes a normal incidence of sound. If a diffuse sound field is assumed, where sound can strike at all angles of incidence, the Paris formula can be used.

\[
\bar{r} = 2 \int_0^{2\pi} \tau(\theta) \cos\theta \sin\theta \, d\theta \tag{3.11}
\]

Leading to the power transmission coefficient for oblique incident sound:

\[
\bar{r} = -\left(\frac{2\omega\rho_0 c}{k_s}\right)^2 \ln(\tau_n) \tag{3.12}
\]

The power transmission coefficient can be used to obtain the sound reduction index with the following formula:

\[
R = 10\log\frac{1}{\bar{r}} \tag{3.13}
\]
The sound transmission coefficient in Eq.(3.12) is valid for frequencies below the first bending resonance frequency of the wall. For most building elements, the resonance frequency lies between 10 and 50 Hz, making this region of less importance for sound insulation. For lighter buildings the resonance frequency goes down and can be less than 10 Hz. However, lightweight structures are made of composite walls resulting in difficult calculations where the resonance frequency may turn out to be different. Figure 3.3 shows the sound reduction from stiffness only for a concrete plate. The sound reduction decreases rapidly for larger, less stiff plates.

![Figure 3.3 Sound reduction from stiffness only for a concrete wall of 6x3x0,2m. E=15e9, ν=0,25 ρ =2200.](image)

**3.2.1. Resonance frequency**

Every element in a building has his own resonance frequency (also called natural or Eigen frequency), at which it is most vulnerable to excitation. At this frequency an equilibrium between force and vibration is found, so that the system can vibrate freely. If a system is set into vibration by a harmonic force, the system will be subjected to a forced vibration. When the force is removed, the system will return to the nearest free vibration.

If a harmonic force is applied at the resonance frequency, the amplitude of the vibration will increase. Without damping, the amplitude can become infinitely large in theory. However, in most situations some form of damping is present, and the amplitude of the vibrations is limited. At the resonance frequency the sound insulation of an element is lowered significantly. This dip in sound insulation can be quite large, and is only limited by the damping of the element.

Combined systems with elements that have the same resonance frequencies can pass along vibrations relatively easy. On the other hand, combined systems with elements that have different
resonance frequencies act as damping on each other. It is therefore advised to give connected elements different resonance frequencies when trying to limit the sound transmission.

The bending resonance frequencies of elements can be calculated with the following formula:

\[
\begin{align*}
    f_{n,m} &= \frac{c^2}{4f_c} \left[ \left( \frac{n}{b} \right)^2 + \left( \frac{m}{l} \right)^2 \right] \\
    (3.14)
\end{align*}
\]

Where:
- \( f_{n,m} \): Resonance frequency \( n,m \) [1/s]
- \( c \): Speed of sound in air [m/s]
- \( f_c \): Critical frequency [1/s]
- \( n, m \): Integers for the first, second etc resonant frequency. [-]
- \( b \): Width [m]
- \( l \): Length [m]

From Eq. (3.14) follows that the resonance frequency depends on the critical frequency and the dimensions of an element. The critical frequency will be discussed in paragraph 3.4, and depends on the stiffness and mass of an element. Increasing either the dimensions or the critical frequency of an element will result in a lower resonance frequency. Since sound reduction is diminished at and slightly above the resonance frequency, it is best if this is kept as low as possible. Sound transmission in the very low frequency range is accepted because the hearing threshold is a lot higher, and almost no sounds are produced at this frequency.

### 3.3. Mass controlled region

Sound transmission in region 2 can be described by using a thick wall. From the following derivation it will become clear that the jump in acoustical impedance, \( Z \), is determining for the sound reduction.

![Figure 3.4 Schematized sound transmission through a wall](image)

If a normal wave is taken with an angle of zero, the behavior of sound at a boundary can be illustrated by Figure 3.4. Following the derivation of (Hassan, 2009) the sound pressure in each of the materials can be described by:
\[ P_1(x,t) = \hat{p}_1 e^{i\omega t} e^{-ik_1x} + \hat{p}_1 e^{i\omega t} e^{+ik_1x} \]  
(3.15)

\[ P_2(x,t) = \hat{p}_2 e^{i\omega t} e^{-ik_2x} + \hat{p}_2 e^{i\omega t} e^{+ik_2x} \]  
(3.16)

\[ P_3(x,t) = \hat{p}_3 e^{i\omega t} e^{-ik_3(x-d)} \]  
(3.17)

And the particle velocity:

\[ v_1(x,t) = \frac{1}{Z_1} (\hat{p}_1 e^{i\omega t} e^{-ik_1x} - \hat{p}_1 e^{i\omega t} e^{+ik_1x}) \]  
(3.18)

\[ v_2(x,t) = \frac{1}{Z_2} (\hat{p}_2 e^{i\omega t} e^{-ik_2x} - \hat{p}_2 e^{i\omega t} e^{+ik_2x}) \]  
(3.19)

\[ v_3(x,t) = \frac{1}{Z_3} (\hat{p}_3 e^{i\omega t} e^{-ik_3(x-d)}) \]  
(3.20)

With:

- \( Z \) = Acoustic impedance  \([\text{Ns/m}^3]\)
- \( k \) = Wavenumber  \([1/\text{m}]\)
- \( d \) = Thickness of wall  \([\text{m}]\)
- \( \omega \) = Angular frequency  \([1/\text{s}]\)
- \( i = \sqrt{-1} \)
- \( t \) = Time  \([\text{s}]\)
- \( \hat{p} \) = Amplitude of pressure wave  \([\text{N/m}^2]\)

Using the matching conditions at the boundaries so that pressure and particle velocity are equal in both materials, the following relation between the pressure in room 1 and room 3 can be found. The complete derivation can be found in the appendix.

\[ \frac{\hat{p}_1}{\hat{p}_3} = \frac{1}{2} \left( 1 + \frac{Z_1}{Z_3} \right) \cos(k_2d) + i \frac{1}{2} \left( \frac{Z_1}{Z_2} + \frac{Z_2}{Z_3} \right) \sin(k_2d) \]  
(3.21)

Which is used to find the sound power transmission coefficient, \( \tau_n \):

\[ \tau_n = \frac{4}{4 \cos^2(k_2d) + \left( \frac{Z_1}{Z_2} + \frac{Z_2}{Z_1} \right)^2 \sin^2(k_2d)} \]  
(3.22)

Where:

- \( Z_1 = Z_3 \) = Acoustic impedance of air  \([\text{Ns/m}^3]\)
- \( Z_2 \) = Acoustic impedance of the wall  \([\text{Ns/m}^3]\)
- \( k_2 \) = Wavenumber of the wall  \([1/\text{m}]\)
- \( d \) = Thickness of the wall  \([\text{m}]\)

This can be simplified to the mass law when the term \( k_2d \) is smaller than 0.3 radians. For low to medium frequencies and normal walls, this is true. Also \( Z_2 \), the acoustic impedance of the wall, is much higher than \( Z_1 \). Using these approximations the mass law is obtained:

\[ \frac{1}{\tau_n} = 1 + \left( \frac{Z_2}{Z_1} \right)^2 (k_2d)^2 = 1 + \left( \frac{\pi f \rho_2 d}{\rho_1 c_1} \right)^2 \]  
(3.23)

With:

- \( \rho_1 \) = Density of air  \([\text{kg/m}^3]\)
- \( c \) = Speed of sound in air  \([\text{m/s}]\)
- \( \rho_2 d \) = Mass of the wall \((\text{m})\)  \([\text{kg/m}^2]\)
The sound power reduction coefficient, $\tau$, can be used to calculate the sound for frequencies with wavelengths that are comparable to the wall thickness. Also known as region 2, where the sound reduction is governed by mass.

$$R = 10 \log \left( \frac{1}{\tau} \right)$$ (3.24)

Above equations are valid for normal incidence of sound. When a diffuse sound field is assumed, oblique incident sound waves have to be considered to. In Eq. (3.25) the sound reduction for an oblique incidence of sound waves is given. Eq. (3.26) averages the Eq. (3.25) over half a sphere, resulting in a 5dB decrease in sound reduction for a diffuse sound field. This is used to obtain the simple theoretical mass law of Eq. (3.27), an example of the sound reduction due to mass is given in Figure 3.5 for a concrete plate.

$$R(\theta) = 10 \log \left[ 1 + \left( \frac{\pi f \rho_2 d \cos(\theta)}{\rho_1 c_1} \right)^2 \right]$$ (3.25)

$$10 \log \left[ \frac{1}{2\pi} \int_0^{\pi} 2\pi \sin \theta \cos^2 \theta d\theta \right] \approx 5$$ (3.26)

$$R = 20 \log(m) + 20 \log(f) - 47.3$$ (3.27)

With:

- $m$ = Mass per unit area [kg/m$^2$]
- $f$ = Frequency [1/s]
- $R$ = Sound reduction index [dB]

Figure 3.5 Sound reduction from mass only for a concrete wall of 6x3x0,2m. $E=15e9$, $\nu=0,25$ $\rho=2200$. 

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3.4. Coincidence controlled region

In region 3, containing high frequencies, coincidence is the reason that the mass law does not correctly predict the sound reduction. Coincidence occurs when the vibration of the element is the same as the vibrations in the air. For sound, this only happens when the sound speeds are equal in both mediums. As the speed of sound varies only for bending waves, the critical frequency is defined as the frequency where bending waves have the same speed as sound in air, Eq. (3.28). Figure 3.6 illustrates the effect of a sound wave with a coincidence frequency impinging on a plate. Only sound waves with an angle of incidence unequal to 0 degrees can have this effect. The lowest frequency coincidence frequency occurs at an angle of 90 degrees.

\[ \frac{c_{\text{air}}}{\sin(\theta)} = c_b(f_c) \]  

(3.28)

Dips in the sound reduction graph from Figure 3.1 are caused by increased sound radiation due to coincidence. In Figure 3.6 only one angle of incidence is considered, but in reality the complete range of angles of incidence result in a range of critical frequencies. This causes the wide dip in the sound reduction index, which is also sometimes drawn as a plateau, if there is enough damping in the system.

Figure 3.6 Coincidence effect. Source: Kennisbank bouwfysica A_17. A line is added to illustrate a impinging sound wave.

After the first coincidence frequency, region 3 starts, where damping governs the sound reduction. The coincidence frequency is also called the critical frequency and is given by the following formula for plates:

\[ f_c = \frac{c^2}{2\pi \sqrt{B}} \]  

(3.29)

With:

- \( f_c \) = Critical frequency [1/s]
- \( c \) = Speed of sound in air [m/s]
- \( m \) = Mass per unit area [kg/m²]
- \( B \) = Bending stiffness of plate [N/m²]

Which can be simplified to:

\[ f_c = \frac{k_b^2 c^2}{4\pi^2 f} = \frac{c^2 \sqrt{3}}{\pi c_l d} \]  

(3.30)

With:

- \( d \) = Thickness [m]
- \( c_l \) = Longitudinal wave speed in the plate [m/s]
- \( k_b \) = Bending wave number [1/m]
For sound insulation purposes it is beneficial to have a high critical frequency, preferably above frequency range of importance in building acoustics. From Eq. (3.29) and Eq. (3.30), it follows that the critical frequency increases with density and decreases with thickness and stiffness. Thin lightweight elements thus have higher critical frequencies, making the region in which the mass law can be applied larger.

To calculate the sound reduction in the frequency range above the critical frequency, the following formula can be used (Hassan, 2009):

\[
R = R_m(f_c) + 10 \log(\eta) + 33.22 \log\left(\frac{f}{f_c}\right) - 5.7 - 10 \log \frac{S}{S_{tot}}
\]

(3.31)

With:
- \(\eta\) = Total loss factor [-]
- \(R_m(f_c)\) = Sound reduction of mass law at \(f_c\) [dB]
- \(f_c\) = Critical frequency [1/s]
- \(S\) = Area \([m^2]\]

The last term in Eq. (3.31) is called the surface factor, and only affects the sound reduction when the exited part of the plate is part of the construction, and less than than the area of the whole wall. This is often the case in lightweight constructions where walls are made of plasterboard walls.

Figure 3.7 shows the sound reduction in the coincidence region. Note that the formula is only valid above the critical frequency and thus shows a sound reduction increase of around 10dB/octave. It does not account for the sound reduction dip around the coincidence frequency, which is a dip of \(\sim 20 \log(\eta)\). An example of such a dip is shown in Figure 3.8.
3.4.1. Sound radiation

An essential part of transmitting sound is radiating that sound into the receiving room. The amount of sound radiated is determined by the critical frequency, and expressed as the radiation factor. As a guideline, this factor can be taken as 0.1 below the critical frequency and as 1 above the critical frequency.

In reality the radiation factor of a plate consists out of the vibrational modes of the Eigen frequencies of the plate. The vibration pattern is a combination of natural modes that depend on the boundary conditions. It is assumed that these modes are inside the frequency band of the exciting force. Although it is possible to calculate the radiation factor of one specific frequency, caused by a specific vibration pattern, it is more useful to take the average over a small frequency band. Therefore the sound radiation factor is often expressed as the average radiation factor over an octave band. Three areas are distinguished, and the radiation factor is given by (Vigran, 2008):

\[
\sigma = \frac{2(a + b)c_0}{2\pi \sqrt{f \cdot f_c} \sqrt{\chi^2 - 1}} \left[ \ln \left( \frac{\chi + 1}{\chi - 1} \right) + \frac{2\chi}{\chi^2 - 1} \right] \quad f < f_c
\]

\[
\sigma = \frac{2\pi f}{c_0} \sqrt{\alpha} \left( 0.5 - 0.15 \frac{a}{b} \right) \quad f = f_c
\]

\[
\sigma = \frac{1}{\sqrt{1 - \frac{f_c}{f}}} \quad f > f_c
\]

\[
\chi = \sqrt{\frac{f_c}{f}}
\]

With:
- \(a\) = length of plate, \(a < b\) [m]
- \(b\) = width of plate, \(a < b\) [m]
- \(S\) = area of plate [m²]
- \(\sigma\) = Radiation factor [-]
Above formulas apply when the plate is vibrated in a (combination of) natural resonant mode(s) and thus vibrates freely. When a vibration is forced upon the plate, above equations are no longer valid. (Sewell, 1970) proposed a radiation factor for forced vibrations for a diffuse incident sound field.

\[
\sigma_f = \frac{1}{2} \left( \ln(k\sqrt{S}) + 0.16 - F(\Lambda) + \frac{1}{4\pi k^2 S} \right) \tag{3.36}
\]

\[
\Lambda = \frac{b}{a} \quad (\Lambda > 1) \tag{3.37}
\]

Where:
- \(F(\Lambda)\) = Polynomial that will be discussed later [-]
- \(k\) = wavenumber \([1/m]\)
- \(S\) = Area \([m^2]\)
- \(\sigma_f\) = Radiation factor for forced vibrations [-]

For frequencies below the critical frequency the forced vibration field is dominant, and above that, the free vibration field. However, for direct sound radiation it is best to consider both forms of vibration. Radiation from flanking sound transmission is purely radiated by resonant waves, since only resonant waves are transmitted via junctions. Figure 3.9 shows an example of the radiation factor for an undefined element.

![Figure 3.9 Radiation factor example. Source: http://vibrationacoustics.asmedigitalcollection.asme.org/](image)

**3.4.1. Internal damping**

Internal damping is the energy loss from the propagation of the sound wave through a medium. As mentioned earlier, sound waves in air can be described by an alternating pattern of compression and rarefactions in the direction of the wave. The pressure differences create temporal density variations in the air compared to the equilibrium state. Between the pressure and density differences is a small time delay. According to ideal gas law, this means that the temperature increases. However, these temperature increases are small and only become relevant when there is no other source of energy.
loss or when there are a great number of pressure differences, e.g. high frequencies or long distances. In other mediums the internal damping is caused by similar effects.

According to (NEN-EN12354-1, 2000) the following formula can be used for the total loss factor, which is a simplification of the real situation. Damping caused by support conditions or radiation is not taken into account.

\[
\eta_{\text{tot}} = \eta_{\text{int}} + \frac{m}{485\sqrt{f}}
\]  

(3.38)

\[\eta_{\text{int}} \approx 0.01 \text{ (or see appendix)} \]  

\[m = \text{mass per area} \quad \text{[kg/m}^2\text{]}\]

\[f = \text{frequency} \quad \text{[1/s]}\]

### 3.5. Total direct sound transmission

Several researchers have tried to combine the various effects described in this chapter to obtain one description of the sound reduction index, with varying degrees of success. However, the NEN norm (NEN-EN12354-1, 2000), specifies the following formulas for the sound reduction index:

\[R_d = 10 \log \left( \frac{\pi f m}{\rho_0 c_0} \right)^2 + 10 \log \left( \frac{2 f \eta_{\text{tot}}}{\pi f_c \sigma^2} \right) \quad f \geq f_c \]  

(3.39)

\[R_d = 10 \log \left( \frac{\pi f m}{\rho_0 c_0} \right)^2 + 10 \log \left( \frac{2 \eta_{\text{tot}}}{\pi \sigma^2} \right) \quad f \approx f_c \]  

(3.40)

\[R_d = 10 \log \left( \frac{\pi f m}{\rho_0 c_0} \right)^2 + 10 \log \left( \frac{1}{2 \sigma_f} + \frac{\eta_{\text{tot}}}{\sigma^2} \cdot \frac{f}{f_c} \cdot \frac{a^2 + b^2}{(a + b)^2} \right) \quad f < f_c \]  

(3.41)

Where:

\[m = \text{Mass per unit area} \quad \text{[kg/m}^2\text{]}\]

\[f_c = \text{Critical frequency} \quad \text{[1/s]}\]

\[\eta_{\text{tot}} = \text{Total loss factor} \quad \text{[-]}\]

\[\sigma = \text{Radiation factor free bending waves} \quad \text{[-]}\]

\[\sigma_f = \text{Radiation factor for forced transmission} \quad \text{[-]}\]

\[a = \text{Length} \quad \text{[m]}\]

\[b = \text{Width} \quad \text{[m]}\]

These formulas do not consider the resonance region, and are thus inaccurate for low frequencies. It is assumed that the resonance frequency is low enough that it does not affect the sound reduction in the frequency range of interest.

A more elaborated formula to describe the sound reduction index is given by (Sewell, 1970), Eq.(3.42)-(3.45). Although it also does not consider the resonance region, the coincidence region is described better that in the NEN Norm. In Figure 3.10 the sound reduction for both methods are presented. For the mass controlled region both methods predict a 6dB/octave increase, while the sound reduction in the coincidence region is clearly different.
\[ R_d = 10 \log \frac{1}{\tau} \]  
\[ \tau = \frac{\ln(k\sqrt{A}) + 0.160 - U(\Lambda) + \frac{1}{4\pi k^2 A}}{\left[ \frac{m\omega}{2\rho c} \left( 1 - \frac{\omega^2}{\omega_f^2} \right) \right]^2} \]  
\[ U(\Lambda) = \frac{1}{2\pi} \left( \Lambda + \frac{1}{\Lambda} \right) \ln(1 + \Lambda^2) - \left( \frac{1}{2} - \frac{\Lambda}{\pi} \right) \ln\Lambda - \frac{1}{\pi} \ln 2 - \frac{2}{\pi} \int_0^1 \tan^{-1} t \frac{dt}{t} \]  
\[ \Lambda = \frac{b}{h} \]

Where:
- \( b \) = Width of element  \([\text{m}]\)
- \( h \) = height of element  \([\text{m}]\)
- \( A \) = Area of element  \([\text{m}^2]\)
- \( k \) = wavenumber  \([1/\text{m}]\)
- \( m \) = mass per area  \([\text{kg}/\text{m}^2]\)
- \( \omega \) = Angular frequency  \([1/\text{s}]\)
- \( c \) = Speed of sound in air  \([\text{m}/\text{s}]\)
- \( \rho \) = Density of air  \([\text{kg}/\text{m}^3]\)

Alternatively there are tables for \( U(\Lambda) \) based on the width and height of the element.

![Graph](image)

Figure 3.10 Sound reduction index according to (NEN-EN12354-1, 2000) and (Sewell, 1970) for a concrete element of 6x3x0.2m. \( E=15\times10^9, \nu=0.25, \rho =2200. \)

Following this set of formulas, the best way to increase the sound reduction is using heavy and limp materials. Each doubling of the mass results in a 6dB increase in sound reduction.
3.5.1. Double leaf walls

Lightweight constructions are light, and thus have a lower sound reduction than heavier constructions according to the previous paragraph. To obtain enough sound reduction, lightweight constructions use double leaf walls (see Figure 3.11), which doubles the sound reduction if they are completely separated. Figure 3.12 illustrates the difference between doubling the mass and using separated walls with the Sewell method. Note however, that the calculated sound reduction for a double wall is limited by effects discussed below.

![Double leaf wall](source.png)

Figure 3.11 Double leaf wall. Source www.tmsoundproofing.com.

In the mass region a 6dB increase can be found when doubling the mass, in contrast to the double of the sound reduction for the separated walls. The change in mass changes the critical frequency, making the comparison in the coincidence region inaccurate. However, it should be clear that the separation of walls increases the sound reduction far more than adding mass.

![Sound reduction index](source.png)

Figure 3.12 Sound reduction index according to (Sewell, 1970), comparing doubling off mass to two separated walls. Concrete wall with $E=15e9$, $\nu=0.25$, $\rho=2200$ and 6x3x0.2m(R Sewell), $\rho=4400$ (R Sewell 2xMass) and 2x 6x3x0.2m(R Sewell 2xWall).
For walls to be completely separated, there can be no link between the two walls. This means no structural connection, no common foundation or edge support and an air gap of at least 1 meter. This is practically impossible in a building, but striving to complete separation can greatly increase sound reduction values.

In reality, double leaf walls are connected to the structure or each other. Often the path via those connections (see Figure 3.14) will transmit more sound than the direct, airborne path. This form, which can be seen as localized flanking, limits the effectiveness of sound reduction from the double leaf wall.

Standing waves inside the cavity also result in a decrease in sound reduction, as well as varying resonance and critical frequencies between leaves and elements in the system. Having enough damping in the double leaf wall will counter these effects, and is often done by adding absorbing materials in the cavity.

Sound reduction for a double leaf wall without structural connections and absorbing materials in the cavity (Hassan, 2009). The formula’s below assume that there is enough absorption in the cavity that the oblique incident sound waves are damped more than perpendicular waves, and that the latter determines the sound reduction (Sharp, 1978).

\[
R = 20 \log(m_1 + m_2) + 20 \log(f) - 47.3 \quad \text{for } f < f_0 \tag{3.46}
\]

\[
R = R_{m1} + R_{m2} + 20 \log(f d) - 29 \quad \text{for } f_0 < f < f_a \tag{3.47}
\]

\[
R = R_{c1} + R_{c2} + 6 \quad \text{for } f_a > f \tag{3.48}
\]

Where:

- \(m\) = Mass per unit area \([\text{kg/m}^2]\)
- \(f\) = Frequency \([\text{1/s}]\)
- \(R_{m1}\) = Sound reduction according to the mass region for wall 1 \([\text{dB}]\)
- \(R_{c1}\) = Sound reduction according to the coincidence region for wall 1 \([\text{dB}]\)
- \(d\) = Thickness of cavity \([\text{m}]\)
- \(f_0\) = First mass-spring-mass resonance frequency \([\text{1/s}]\)
- \(f_a\) = Lowest order resonance frequency across the cavity \([\text{1/s}]\)

\[
f_a = \frac{c}{2\pi d} \tag{3.49}
\]

\[
f_0 = \frac{c}{2\pi} \left[ 1.8 \frac{\rho_0}{d} \left( \frac{1}{M_1} + \frac{1}{M_2} \right) \right]^{1/2} \tag{3.50}
\]

With:

- \(c\) = Sound speed in air \([\text{m/s}]\)
- \(\rho_0\) = Density of air \([\text{kg/m}^3]\)

Using above formulas, the sound reduction is calculated for a double leaf wall with varying cavity thicknesses. In Figure 3.13 the effect of increasing the cavity is clearly visible. The frequency range that is affected lies between the mass-spring resonance frequency and about 500 Hz. Frequencies above 500 Hz are susceptible to standing waves, but those are assumed to be properly damped by absorbing materials inside the cavity. Also note that absorbing materials affect only the sound reduction of higher frequencies and resonance dips, making placement inside the cavity very effective. Below the mass-spring-mass resonance frequency, the double leaf wall acts as one element, because the wavelength is significantly greater than the total wall. This means that only an increase in sound reduction due to mass can be seen.
Connections between the leaves of double walls, or between leaves and the constructions reduce the sound reduction index of the wall. The connections are needed for the stiffness of the walls, and stability of the construction. To evaluate this reduction the term \( \Delta R_m \) is added to Eq.(3.46), which then is used instead of Eq.(3.47) whenever it has a lower value. For the simplest rigid connections between leaves the following formulas (Hassan, 2009) can be used. An illustration of the connection can be found in Figure 3.14, where both leaves are connected rigidly to a stud.

\[
\Delta W_{	ext{fD}} = 20 \log\left(\frac{R_m + R_{m\theta}}{R_m - R_{m\theta}} + 0.9\right)
\]

\[
\Delta W_{	ext{f@}} = 10 \log\left(\frac{R_m + R_{m\theta}}{R_m - R_{m\theta}}\right) - 10 \log\left(\frac{n\psi}{s}\right)
\]

\[
\Delta R_m = 10 \log(f_{cl}) + 10 \log(b) - 23.4
\]

\[
f_{cl} = \frac{M_{s1} \sqrt{f_{c2}} + M_{s2} \sqrt{f_{c1}}}{M_{s1} + M_{s2}}
\]

\[
\psi = \frac{8c^2}{\pi^2 f_{cl}^2}
\]
\[ Z = \frac{4c^2 m_s}{\pi f_c} \]  
(3.55)

Where:

- \( c \) = speed of sound \[ \text{[m/s]} \]
- \( Z \) = Acoustic impedance \[ \text{[Ns/m}^3] \]
- \( m \) = Mass per unit area \[ \text{[kg/m}^2] \]
- \( n \) = Number of points \[ [-] \]
- \( S \) = Area \[ \text{[m}^2] \]
- \( b \) = Spacing between studs or line connections \[ \text{[m]} \]

The cut-off frequency, \( f_b \), is introduced, which is the frequency where the sound reduction of Eq. (3.56) and (3.57) are equal.

\[
\begin{align*}
R &= 20 \log(M_1 + M_2) + 20 \log(f) - 47.3 + \Delta R_m \\
&\quad \text{for } f_b < f < 0.5f_c \\
R &= R_{m1} + R_{m2} + 20 \log(fd) - 29 \\
&\quad \text{for } f_b < f < f_b
\end{align*}
\]  
(3.56)  
(3.57)

With:

- \( \Delta R_m \) = Difference in sound reduction from rigid studs \[ \text{[dB]} \]

Resilient studs can be used to decrease the sound reduction loss from the connections even further. Similar, double stud systems, adding rubbers or limiting the number of connections can increase the separation between the leaves of a double wall and thus decrease the direct sound transmission.
CHAPTER 4. Flanking sound transmission

4.1. Flanking sound considerations

Buildings make use of smart impedance jumps and separation to increase the sound reduction index. However different building parts still have to be connected to each other in order to create a stable building. When sound reduction relies heavily on the separation of elements, or smart impedance jumps, connections become critical. This is especially the case for lightweight buildings and buildings with high sound reduction requirements.

Flanking sound paths that travel through connections can be illustrated as in Figure 4.1 on a larger scale, or as in Figure 3.14 on a smaller scale. However, calculating the flanking sound reduction index is similar for both. Figure 4.2 illustrates the simplification made for the calculation of flanking sound transmission. Basically, a flanking path is defined as a building element in the source room, a building element in the receiving room and a junction. Eq. (4.1) defines the flanking sound reduction index of sound travelling through all these elements.

\[
R_{ij} = \frac{R_i + R_j}{2} + \Delta R_i + \Delta R_j + K_{ij} + 10 \log \frac{S}{L_0 L_{ij}}
\]  

(4.1)

Where:
- \( R_{ij} \) = Flanking sound reduction index  [dB]
- \( R_i \) = Sound reduction index of wall i  [dB]
- \( R_j \) = Sound reduction index of wall j  [dB]
- \( \Delta R_i \) = Extra sound reduction index of wall i  [dB]
- \( \Delta R_j \) = Extra sound reduction index of wall j  [dB]
- \( K_{ij} \) = Vibration reduction index  [dB]
- \( S \) = Area of separating wall (10 m² if the walls have no common separating wall)  [m²]
- \( L_{ij} \) = Connection length between wall i and j  [m]
- \( L_0 \) = Reference length (1 m)  [m]
The extra sound reduction gained from added linings and floating floors is less effective for flanking sound reduction than for direct sound reduction. Taking half the value of the extra reduction for direct sound as extra flanking sound reduction is a fair assumption (Bron van der Jagt, et al., 2011).

Although the flanking path from Figure 4.1 travels through more concrete than the direct path, this does not add to the flanking sound reduction index according to Eq. (4.1) Taking a closer look, the sound path in the concrete can be split in a perpendicular and a parallel part. Where perpendicular sound propagation is accomplished by longitudinal waves, and parallel sound propagation by transverse and bending waves. Without bending wave impedances, the sound reduction part of the elements i and j can be described by the average of their direct sound reduction.

If junctions provide any sound reduction, direct transmission will almost always have the lowest sound reduction index. However, most flanking sound transmission problems exist due short circuiting. Where the flanking path takes a shorter route, skipping part of the direct sound insulation. This happens almost exclusively in buildings with double leaf elements. A common mistake is illustrated in Figure 4.3, where the floating floor is not dilated between rooms.

Figure 4.3 Short circuit flanking sound transmission.

When designing to reduce flanking sound transmission there are two key points, separation and sound reduction in the junction. To reach higher sound reduction values, separation of walls is also used for direct sound transmission. For a good separation, junction detailing is very important.

**4.1.1. Ventilation pipes and other air paths**

An airtight seal is the most important factor in sound insulation. In modern buildings high air tightness can be achieved. However, plumbing, wiring and ventilation require open connections between rooms, creating possible flanking sound paths. Separating the climate systems of each room can greatly decreases the sound transmission in these paths.

Another possible air leak is located at the connection between wall and floor elements. The rough surfaces of the floor and wall leave air gaps between them, which create possible air paths. Caulking the connection takes care of these air gaps. However, temperature and load differences can cause movement between building elements and tear the caulk. Thus maintenance is required, and neglect can increase the sound transmission. Short circuiting should also be prevent when caulking, meaning unwillingly connecting double leaf constructions with caulk.
4.1.2. Relevance of flanking sound transmission

Differences between sound insulation values for separating walls between laboratory tests and in situ tests indicate that flanking sound contributes significantly to the sound transmission. Sound reduction differences of up to 10 dB have been found between laboratory tests and in situ measurements. Figure 4.4 presents differences found for a separating wall in a dwelling by (Gerretsen, 1979).

Residential homes have very simple junctions that are cheap and easy to build. However, they provide little sound reduction. The numerous flanking sound paths thus add up to a considerable decrease in sound reduction as can be seen in Figure 4.4.

![Figure 4.4 Sound reduction according to theory and measurements. Several calculations are made for a 425 kg/m² brick wall. From top to bottom: Mass law reduction index, 'light' surroundings, laboratory situation, 'Heavy' surroundings, and only internal damping and radiation. Source: (Gerretsen, 1979)](image)

Lightweight buildings often have rigid junctions to gain the necessary stability. In contrast to heavyweight buildings which gain stability from large elements. If no consideration is put into connections between floors and walls in lightweight buildings, flanking transmission becomes the dominant path over direct transmission.

In buildings where higher sound reduction values are required, flanking sound can become a problem. Separation used to increase the sound reduction while keeping the amount of materials used low. Similar to separation in lightweight buildings, flanking sound transmission can then become the dominant path.

4.1.3. Categorizing different transmission paths

If a simple room is considered, as in Figure 4.5, there are 12 first order flanking paths between two rooms. A complete list is given in Table 4.1. Out of these 12 flanking paths, there will be a handful critical paths. While walls are easily separated, most problems occur with floors and ceilings. To indicate different paths, subscripts are used of the elements they pass. For example, $R_{37}$ is the
flanking path that passes through the floor in the origin room and the floor in the receiving room as indicated in Figure 4.5. Every element in the origin room (#1-5) can pass along a sound to two elements in the receiving room, except for element 1 which can excite 4 elements. This sums up to a total of 12 flanking paths.

Figure 4.5 Flanking sound transmission paths

Considering the transmission between two horizontal adjacent and two vertical adjacent rooms will result in a complete evaluation. Sound reduction values for mirror situations are the same. For symmetrical rooms with the same boundary condition this can decrease the amount of flanking transmission paths that have to be evaluated.

<table>
<thead>
<tr>
<th>Path name</th>
<th>Elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>R_{16} Separating wall-side wall</td>
</tr>
<tr>
<td>2</td>
<td>R_{17} Separating wall-floor</td>
</tr>
<tr>
<td>3</td>
<td>R_{18} Separating wall-side wall</td>
</tr>
<tr>
<td>4</td>
<td>R_{19} Separating wall-Ceiling</td>
</tr>
<tr>
<td>5</td>
<td>R_{26} Side wall - Side wall</td>
</tr>
<tr>
<td>6</td>
<td>R_{21} Side wall - Separating wall</td>
</tr>
<tr>
<td>7</td>
<td>R_{37} Floor-Floor</td>
</tr>
<tr>
<td>8</td>
<td>R_{31} Side wall - Separating wall</td>
</tr>
<tr>
<td>9</td>
<td>R_{34} Side wall - Side wall</td>
</tr>
<tr>
<td>10</td>
<td>R_{51} Side wall - Separating wall</td>
</tr>
<tr>
<td>11</td>
<td>R_{59} Ceiling - Ceiling</td>
</tr>
<tr>
<td>12</td>
<td>R_{51} Side wall - Separating wall</td>
</tr>
</tbody>
</table>

Table 4.1 Flanking sound transmission paths
4.2. Detailed Flanking sound transmission

Specific for the flanking sound reduction is the vibration transmission coefficients and the modal coupling between elements. Both effects will be discussed, as well as the loss factors that can reduce the effect of modal coupling.

4.2.1. Vibration transmission coefficients in junctions

The vibration reduction index describes the sound reduction of a junction. \( K_{ij} \) from Eq.(4.2) can be used for all junctions, even lightweight junctions. \( K_{ij,s} \) from Eq. (4.3) is a simplification in which no correction for the reverberation time has to be added, this is also valid for lightweight junctions, but only for lightly damped boundaries (Bron van der Jagt, et al., 2011). As can be seen from the formulas, the sound reduction of the junction largely depends on the vibration velocity level difference. Damping inside the elements connected to the junction are evaluated by using the structural reverberation time.

\[
K_{ij} = \frac{D_{v,ij} + D_{v,ji}}{2} + 10 \log \frac{l_{ij}}{\sqrt{a_i a_j}} \tag{4.2}
\]

\[
K_{ij,s} \approx \frac{D_{v,ij} + D_{v,ji}}{2} + 10 \log \frac{l_{ij}}{\sqrt{S_i S_j}} \tag{4.3}
\]

\[
a_i = \frac{2.2 \pi^2 S_i}{c_0 T_{S,i}} \sqrt{\frac{f_{ref}}{f}} \tag{4.4}
\]

\[
T_S = \frac{2.2}{f \eta} \tag{4.5}
\]

With:

- \( K_{ij} \) = Vibration reduction index [dB]
- \( D_v \) = Vibration velocity level difference [dB]
- \( l_{ij} \) = Junction coupling length [m]
- \( a \) = Equivalent absorption length [m]
- \( S \) = Area \([m^2]\)
- \( c_0 \) = Speed of sound in air \([m/s]\)
- \( T_S \) = Structural reverberation time [s]
- \( f_{ref} \) = 1000 Hz [Hz]
- \( f \) = Frequency [Hz]
- \( \eta \) = Total loss factor [-]

The vibration velocity level difference is the normalized ratio of vibrations transmitted via the joint to the radiating element. For simple connections consisting out of homogenous slabs of equal size there are formulas that give good estimations. When more complicated, or lightweight junctions are considered, these formulas are prone to errors. That is why for new junctions measurements are taken to create empirical formulas specific to that type of junction. The obvious downside is that this is costly and ineffective. Measurements are based on the following formulas and make use of impact sound.

\[
D_{v,ij} = 10 \log \frac{\langle \psi_i^2 \rangle}{\langle \psi_j^2 \rangle} \tag{4.6}
\]
\[
< \ddot{v}_i^2 > = \frac{F^2 \text{Re} \left\{ \frac{1}{Z_i} \right\}}{\omega m_i M_{si} S_i}
\]
(4.7)

\[
< \ddot{v}_i^2 > = \frac{M_{si} S_i \eta_{il}}{M_{sj} \eta_{lj} \eta_{tot}}
\]
(4.8)

Where:

\(< \ddot{v} >\) = Mean square normal surface vibration velocity [m/s]

\(Z\) = Impedance [Ns/m^3]

\(\omega\) = Angular frequency [Hz]

\(\eta_{ii}\) = Coupling loss factor [-]

\(m\) = Mass per unit area [kg/m^2]

\(F\) = Force [N]

The mean square normal surface vibration velocity of the plates i and j has to be measured when using Eq. (4.6). A rough estimation can be made with Eq. (4.8), but is limited to homogenous plates that are simply connected. From the formula follows that if the vibration is passed along to an element half the weight of the first element, that the lighter element vibrates twice as much (if there is no damping). However, even with damping, this can cause problems in lightweight buildings where lightweight elements radiate more sound by having induced vibration of heavier elements.

For the rigid junction of heavy elements there are empirical formulas to obtain \(D_{\nu,ij}\). Although they are not based on lightweight elements, the principles should still be valid for lightweight elements. The sound reduction is frequency independent for rigid junctions, and depends for a large part on the mass difference between elements. The transmission losses of the junction in Figure 4.6 are presented in Figure 4.7 and Figure 4.8. These estimations concur reasonably with measurements, and are related to the mass and critical frequency difference between the elements via a factor \(\psi\). When the elements are made of the same material, \(\psi\) depends solely on the dimensions of the plate.

\[
\chi^2 = \frac{f_{c2}}{f_{c1}}
\]
(4.9)

\[
\psi = \frac{m_{s2} f_{c1}}{m_{s1} f_{c2}}
\]
(4.10)

When plates have the same materials the equation reduces to:

\[
\psi = \left( \frac{h_2}{h_1} \right)^2
\]
(4.11)

With:

\(f_c\) = Critical frequency [Hz]

\(m\) = Mass per unit area [kg/m^2]

\(h\) = Thickness [m]
Figure 4.6 Plate numbering at the junction. Taken from (Craik, 1981).

Figure 4.7 Transmission loss from plate 1 to plate 2. Taken from (Craik, 1981).
In (NEN-EN12354-1, 2000) several empirical formulas to calculate the transmission loss are given for basic junctions. Compared to calculations based on measurements from Craik, they seem to agree well. For a junction resembling the one in Figure 4.6, the following formulas are given:

\[
K_{1,3} = 8.7 + 17.1M + 5.7M^2
\]
\[
K_{1,2} = 8.7 + 5.7M^2
\]
\[
M = \log \frac{m_{s,i}}{m_{s,j}}
\]

When elastic layers are added, a frequency dependent term is added. Where lower frequencies are less attenuated than higher frequencies. (NEN-EN12354-1, 2000) gives:

\[
K_{1,3} = 8.7 + 14.1M + 5.7M^2 + 2\Delta_1
\]
\[
K_{2,4} = 3.7 + 14.1M + 5.7M^2 ; 0 \leq K_{2,4} \leq -4
\]
\[
K_{1,2} = 5.7 + 5.7M^2 + \Delta_1
\]
\[ \Delta_1 = 10\log\left(\frac{f}{f_1}\right) \]  
If:

\[ \frac{E_i}{\tau_1} \approx 110 \frac{MN}{M^3} \]  

\[ f_1 = 125 \text{ Hz} \]

Figure 4.10 Symmetric junction with elastic interlayers. Taken from (Pedersen, 1995).

Where \( f_1 \) is the crossover frequency between calculation methods for low frequencies and higher frequencies, for which the transmission loss increases with the frequency. A more precise calculation for \( f_1 \) is given by (Pedersen, 1995):

\[ f_1 = 2.5 \cdot 10^{-6} \left( \sqrt{\frac{\rho_1 \rho_2 d_1 h_1}{G}} \right)^{\frac{3}{2}} \]  

With:

- \( w \) = Thickness of elastic layer [m]
- \( h_1 \) = Thickness of plate [m]
- \( G \) = Shear modulus of elastic layer, making use of the dynamic stiffness \([N/m^2]\)
- \( \rho \) = Density \([kg/m^3]\)

(Pedersen, 1995) also states that values of \( f_1 \) below 50 Hz should not be accepted, and that for frequencies below \( f_1 \) the junction behaves like a rigid junction. This implies that elastic junctions for low frequencies have no additional effect. A similar conclusion was found for the resilient connections of double leaf walls. However, measured crossover frequencies do not always agree with the proposed formula, and there is still a uncertainty for the crossover frequency.

For lightweight junctions with double leaf walls the following empirical formulas are given by (NEN-EN12354-1, 2000):

\[ K_{1,3} = 10 + 20M - 3.3\log\frac{f}{f_k} \]  

\[ K_{2,4} = 3 + 14.1M + 5.7M^2 \; ; \; 3 \leq \frac{m_2}{m_1} \]  

\[ K_{1,2} = 10 + |10M| + .3\log\frac{f}{f_k} \]  

With:

\( f_k = 500 \text{ Hz} \)
In contrast to heavy weight junctions, lightweight junctions are frequency dependent. Surprisingly, the sound reduction increases for lower frequencies in lightweight junctions. However, for added elastic layers in lightweight junctions there are no formulas given by (NEN-EN12354-1, 2000). Above formulas are based on a SEA approach, which is erroneous for lightweight elements and low frequencies, and thus might be wrong. FEM analysis will provide a better insight in what parameters influence the sound reduction of junctions in chapter 7 and 8.

4.2.2. Internal and total loss factor

Damping is used to attenuate sound, and to reduce the dips in the sound reduction caused by modal phenomena. Increasing the total loss factor increases the amount of dampening a system has. Besides that, it is important to vary resonance frequencies of different elements to prevent modal coupling of elements. Otherwise the vibrations can propagate very easy from one element to another, and even amplify each other. For structural considerations the resonance frequency of loadbearing elements should not be below 3 Hz (NEN6702). This can be done by varying dimensions and stiffness of elements (and rooms), which prevents overlapping resonance frequencies and modal coupling.

There is a key difference between the internal and the total loss factor, which is often missed. Where the internal loss factor describes the losses from one structural element, the total loss factor describes the losses from a system of elements. However, measurements cannot distinguish each loss factor accurately and thus the definitions are blurred. The internal loss factor is usually seen as a material property.

Following (Norton & Karczub, 2012) definition the total loss factor is defined as:

\[
\eta_{\text{tot}} = \eta_{\text{int}1} + \frac{\eta_2}{\eta_1} \eta_{\text{int}2} \eta_{21}
\]

\[
\eta_{\text{int}} = \eta_s + \eta_{\text{edge}} + \eta_{\text{rad}}
\]
Where:
\[
\eta_{\text{int}} = \text{Internal loss factor} \quad [-]
\]
\[
\eta_{\text{tot}} = \text{Total loss factor} \quad [-]
\]
\[
\eta_s = (\text{internal) Loss factor due to friction in structural element} \quad [-]
\]
\[
\eta_{\text{edge}} = \text{Edge loss factor} \quad [-]
\]
\[
\eta_{\text{rad}} = \text{Radiating loss factor} \quad [-]
\]
\[
\eta_{21} = \text{Coupling loss factor} \quad [-]
\]
\[
n = \text{Modal density} \quad [-]
\]

For rigidly connected structures it is found that \( \eta_{\text{edge}} < \eta_s \), and is thus often ignored. \( \eta_{\text{rad}} \) can become the dominant term in lightweight structures, although for heavyweight structures it is often neglected as well. That is why in (NEN-EN12354-1, 2000) the following formula is presented for the total loss factor, which is valid for masses below 800 kg/m\(^2\):

\[
\eta_{\text{tot}} \approx \eta_{\text{int}} + \frac{m}{485 \sqrt{f}} \quad (4.23)
\]

\[
\eta_{\text{tot}} \approx \eta_{\text{int}} + \frac{m}{485 \sqrt{f}}
\]

Where:
\[
\eta_{\text{int}} = \text{Internal loss factor (can be assumed to be 0.01)} \quad [-]
\]
\[
m = \text{Mass per m}^2 \quad [-]
\]

Although a value of 0.01 can be assumed for \( \eta_{\text{int}} (= \eta_s) \) according to (NEN-EN12354-1, 2000), this value varies greatly for different materials and situations (See Appendix for values). For example, adding a friction layer between two gypsum board plates should increase the damping and thus the sound reduction index. However, this can only be found by doing experiments.

For a more detailed loss factor (NEN-EN12354-1, 2000) gives the following formula

\[
\eta_{\text{int}} = \eta_s + \frac{\rho_0 c_0}{\pi f m} \sigma + \frac{c_0}{\pi^2 \sqrt{f} \cdot f_c} \sum_{k=1}^{4} l_k \alpha_k \quad (4.24)
\]

With:
\[
a_k = \text{Absorption factor (0.05 - 0.5)} \quad [-]
\]
\[
l_k = \text{Edge length} \quad [\text{m}]
\]
\[
\sigma = \text{Radiation factor} \quad [-]
\]
\[
\rho = \text{Density} \quad [\text{kg/m}^3]
\]
\[
c = \text{Speed of sound in air} \quad [\text{m/s}]
\]
\[
S = \text{Area} \quad [\text{m}^2]
\]

Which also can be found by measuring the structural reverberation time:

\[
T_s = \frac{2.2}{f \eta_{\text{tot}}} \quad (4.25)
\]

However, the coupling factor is now neglected. This factor is very difficult to quantify, again because measurements cannot keep each factor separated accurately. Nonetheless it is clear that increasing the internal loss factor of each element will lead to a higher sound reduction index, and a reduction of dips in the sound reduction around the resonance frequencies.
4.2.3. Bending impedance

If a junction is looked at as a boundary, the same way the boundary between an element and air is looked at, a large impedance jump will result in a high sound reduction. As described before, bending waves are dominant in elements, so instead of the longitudinal impedance, the bending impedance has to be used. According to (Linjama & Lahti, 1993) the bending wave impedance is:

\[
Z_{M0} = \frac{kB}{\omega} \tag{4.26}
\]

Inserting the wavenumber for bending waves, the equation can be reduced to:

\[
Z_{M0} = \frac{\sqrt{m} \sqrt{B^3}}{\sqrt{\omega}} \tag{4.27}
\]

With:

- \(k\) = Wavenumber \([1/m]\)
- \(B\) = Bending stiffness \([Nm^2]\)
- \(m\) = Mass per unit area \([kg/m^2]\)
- \(\omega\) = Angular frequency \([Hz]\)
- \(Z_{M0}\) = Bending wave impedance \([Ns/m]\)

As can be seen in Eq. (4.27), the bending wave impedance of an elements thus largely depends on the stiffness and the mass. A bending wave impedance jumps depends on the difference in stiffness and mass between two elements. For elements of the same material, the impedance jump depends solely on the dimension differences between the elements. Earlier, the junction velocity difference was based on the mass difference, which is also solely based on dimension differences between elements if these elements are the same material. This means that the velocity level difference might be described as an bending wave impedance jumps, and be increased by the stiffness differences as well as the mass difference.
CHAPTER 5. Sound insulation in practice and new developments

Previous chapters have given an overview of sound insulation from the existing literature. This chapter will summarize the measures that can be taken to increase the sound insulation. Also, it is investigated how sound insulation is achieved in practice, by interviewing building physic consultancies. Lastly, some experimental sound insulation methods are discussed.

5.1. Sound insulation in case studies

To gain a better understanding of how sound insulation is dealt with when designing a building, several building physic consultancies where visited. The interviews consisted partly of open questions and partly of questions about a specific case. In APPENDIX D the complete interviews can be found, and a short summary is given below.

Most building physic consultancies agree with each other, and named the same important factors. Where some emphasized the use of mass, most thought separation was the best option. This was valid for general sound reduction as well as for low frequency sounds. For the latter, stiffness became important as well.

However, flanking was never seen as a large problem. The main reason for this was that flanking had to travel through the same elements as the direct path, but more of it. Therefore it is curious that separation, dilatation and springs were mentioned that often, since that has to do with flanking.

All but one office was convinced that a sufficient sound reduction could be obtained with a lightweight construction. For higher sound reduction requirements all offices suggested a box-in-box system made of concrete.

In APPENDIX E a floor detail from a move theater in Dordrecht is discussed, and recommendations are made based on the results from this research.

5.2. Experimental sound insulation methods

Several new ideas are being researched at the moment, that can greatly increase the sound reduction. New fabrication techniques make it possible to produces these ideas, and the initial results are very promising. These experimental methods are not yet ready to be used in buildings, but are worth mentioning. A short summary is given of the most promising ideas.

Acoustic black holes are a recent development where bending waves in plates are almost 100% absorbed (Krylov & Bowyer, 2013). The thickness of the plate is gradually decreased to zero, so that the bending wave speed approaches zero. In doing so, the bending waves are essentially trapped, and can be easily absorbed by adding a small amount of absorbing material. An example plate is shown in Figure 5.1. Downsides are that the edges are razor sharp, and that the plates cannot be used as structural elements due their low stiffness.
Metamaterials are manmade materials that have properties that cannot be found in nature. One of the properties that is interesting for acoustic purposes is the negative refraction, which can be used to allow only certain frequencies to be transmitted (Liu, et al., 2008). Also, the ability to control the redirection of sound can have its purposes.

Another metamaterial that is up and coming, is the locally resonating sonic material (Liu, et al., 2000). These materials are made with an array of weights within a holding structure, such as illustrated in Figure 5.2. The spatial arrangement of resonators can be tuned so that the material behaves as a material with negative elastic constants. Initial test have shown sound reduction far greater than achievable by the conventional mass law, including the low frequency range.

5.3. Guidelines for improving sound reduction

In APPENDIX C a table can be found with an overview of how the difference characteristics have influence on the sound reduction. A short summary of the most important tools for sound insulation is given below.

Sound insulation is achieved by impedance jumps, that reflect the sound back instead of transmitting it. Materials with high density have a higher specific impedance. Multiple impedance jumps can be used to increase the sound insulations further. It is thereby important that the impedance jumps are separated, or at least connected with a material that has a low specific impedance. Box-in-box
systems are used for buildings with very high sound reduction requirements, and rely on both large impedance jumps due to mass, and separation. Separation is done with springs and rubbers, that both have a very low stiffness and impedance.

Flanking sound transmission relies on many of the same principles, where both impedance jumps and separation decrease the amount of sound transmitted. An important characteristic of flanking sound is the modal coupling between elements. Elements are more easily set into vibration near their natural frequency, and will transmit vibrations more easily as well. When the natural frequencies of connected elements are too close together, the flanking sound transmission can increase. This should be avoided by making adjacent rooms asymmetric.
CHAPTER 6. Experiment description

To investigate low frequency flanking sound transmission in lightweight buildings, a case study is done via a finite element method (FEM) model. FEM models are excellently suited for low frequency studies, in contrast to the commonly used statistic energy analysis models, which are inaccurate in that area. FEM models are however, not suited for higher frequency studies.

The model itself is kept abstract, and does not correspond to real situation. In this way, effects such as stiffness, density and geometry differences on the vibration reduction index, \( k_{ij} \), can be researched. By keeping the model abstract, the important factors for flanking sound transmission can be isolated and improved upon. Moreover, it creates the opportunity to study the effect of material properties instead of materials with set properties. The research focusses on variations in floor elements, instead of varying the geometry of the complete system (e.g. Box-in-box, sound buffer zone, changing structural scheme).

Since flanking sound transmission mostly becomes a problem with higher sound insulation requirements, a movie theater is taken as case study. Movie theaters often have large spans, making a lightweight structure an attractive choice. By keeping the case abstract, the results should be applicable in most building types.

6.1. Case design

In a movie theater, the movie room with the highest sound reduction requirements is often the largest room. The dimensions of the room in the case study should reflect this, and are chosen as 40x25x10m. To keep the case simplistic, the room is modelled as horizontal and vertical plates, without the interior or the tribune. The schematic drawings in this chapter can also be found in the appendix, where they are presented in a more appropriate scale.

To incorporate the surrounding structure, the structure supporting 12 movie rooms with identical dimensions is modelled. Although this is not a realistic scenario, it will provide insight in the effect of the surrounding structure. The supporting structure will consist out of steel beams and columns. Since there can be no columns inside the movie room, the largest beams span 25 meter. Columns are HEA750 elements, beams are HEA1000. Figure 6.2-6.3 gives a structural schematization of the abstract case.

Figure 6.1 Structural schematization of the movie theater case, cross section AA'.
In the above figures the modelled part in ANSYS is indicated by dotted lines. Only a part of the two floors will be modelled, as well as four walls and one beam. Hereby it is assumed that the floors are structurally disconnected every 10 meters. Due to earlier incorrect assumptions, the modelled floors span 10 meter and the beam is HEA750 in the ANSYS model. The consequences of these incorrect assumptions are briefly discussed in the next chapter.

Further simplifications of the case are needed to model the case in the finite element program, and will follow the guidelines proposed by lichterbouwen.nl (TNO & lichterbouwen.nl, sd). Assuming a lightweight floor, a system like Figure 6.4a can be used. The connection between the floor and beam
is considered a hinge when no precautions are taken to guide vibrations into the beam, for example via a rubber. Vibrations deform the beam as in Figure 6.4b (greatly exaggerated), so that for vibrational analysis the connection can be seen as hinged. This is confirmed with measurements by (TNO & Lichterbouwen.nl, sd).

Floors and walls are taken as homogenous plates of 200mm concrete. Finishing layers or additive linings such as floating floors or suspended ceilings are assumed to provide additional sound reduction as (NEN-EN12354-1, 2000) describes, without influencing the vibration reduction index. Furthermore, only acoustic loads will be considered, any additional loads are ignored, which should not matter for the vibrational analysis as long as no springs are used. Schematization of both the ANSYS model and a more realistic junction are given in Figure 6.5. The connections in ANSYS will be modelled as in Figure 6.4a.
6.2. Experiment set-up

In the model, the velocity level difference, $D_{v,ij}$, of one of the flanking paths between two rooms will be calculated. This means that only a part of the total flanking sound transmission is looked at, see Eq.(6.1). Because the wall and the floor areas are kept the same throughout the experiments, the last term is constant. The direct sound reduction, $R_i$ and $R_j$, will vary, but is assumed to be relatively constant during the modelling phase. Any effects that could change the direct sound reduction are discussed afterwards. An increase in the velocity level difference, $D_{v,ij}$, will result in an increase in the vibration reduction index $k_{ij}$. This value is often used as well, the two are related by Eq. (6.2).

$$ R_f = \frac{R_i + R_j}{2} + D_{v,ij,\text{situ}} + \log \left[ \frac{S_s}{\sqrt{S_i S_j}} \right] $$ (6.1)

$$ K_{ij} = \frac{D_{v,ij} + D_{v,ji}}{2} + 10 \log \frac{l_{ij}}{\sqrt{a_i a_j}} $$ (6.2)

With:
- $R_f$ = Flanking sound reduction [dB]
- $R_i$ = Direct sound reduction of plate i [dB]
- $R_j$ = Direct sound reduction of plate j [dB]
- $D_{v,ij,\text{situ}}$ = Vibration velocity level difference in field situation [dB]
- $S_s$ = Area of separating wall $[m^2]$
- $S_i$ = Area of element i $[m^2]$
- $S_j$ = Area of element j $[m^2]$
- $l_{ij}$ = Connection length [m]
- $a_i$ = Equivalent absorption length of element i [m]
- $a_j$ = Equivalent absorption length of element j [m]

A higher velocity level difference will thus result in a higher flanking sound reduction. To do so, three variants for the floor are made that should increase $D_{v,ij}$, which are discussed below. The floor element is chosen because it reduces the weight of the structure the most, and walls are often made of lightweight materials already. Also, it is interesting to investigate the effect of floors on $D_{v,ij}$ because normally, only the junction and mass difference is looked at to increase the velocity level difference. As the frequency range is near various resonance frequencies, the decoupling of the floor-beam-floor can yield great results.

Because FEM is only accurate for low frequencies, the model will evaluate low frequency sound transmission from 25 to 125Hz. The velocity level difference should be constant (or slightly increase) over the frequency range according to the literature, making the results valid for the sound transmission in the complete frequency range. Because it is still a simulation of the reality, the results should not be taken literally, and small errors of ~3dB are expected for similar calculations.

A sensitivity study is conducted to study the effects of a number of parameters, to gain insight into the factors influencing the velocity level difference. This is done by varying the parameters and comparing the results around a basic case with set values. Significant changes in sound transmission are discussed, and used to optimize the total sound reduction.

An increase of 10-20 dB of the vibration reduction index, without decreasing the direct sound reduction of the system, would greatly increase the total sound reduction. Identifying problematic areas or irregularities will lead to a greater understanding, and eventually to better solutions. The factors that affect the abstract case should also affect real scenarios. Moreover, any improvements found should be applicable to the other flanking sound transmission paths as well.
To make the setup clear, only the floor element is changed while the rest of the structure remains the same. The velocity level difference is calculated between the two floors. How this is done exactly, is described in the next chapter. Any improvements found are relative, and should thus eliminate most modelling faults or errors.

6.3. Floor variants

To increase the vibration reduction index, three variants were designed for the floor. These designs are aimed at reducing the propagation of bending waves, which are mainly responsible for flanking sound transmission. Based on the theory, this can be done via impedance jumps and mass differences. However, decoupling, varying the resonance frequencies, stiffness differences and controlling the bending wave speed may have better results. Because these effects are interrelated, a local optima might be found that balances the effects of all above mentioned factors.

6.3.1. Honeycomb model

A grid of stiffeners is made, where plates can be added in between or on top of the stiffeners, as is shown in Figure 6.6 and Figure 6.7. This increases the total floor height, but makes it lighter and relatively stiffer. The connection between the stiffeners and the plate provides an excellent way to increase the damping, by adding material with a high friction coefficient (Zegers, 2011). Dimensions shown in the drawings are used as a reference case in chapter 8.

Figure 6.6 Floor design for the honeycomb variant.
The stiffeners themselves provide impedance jumps to decrease sound transmission and bending wave propagation. This is accomplished by increasing the thickness, using a material with a higher density or with a lower Young's modulus. Geometry differences also influence the bending wave speed, which can be used to damp vibrations more effectively (Varanasi, et al., 2013).

Varying the geometry, stiffness and density also has effect on the resonance frequency of the plate and stiffeners. When the resonance frequency of the plate is significantly higher than the stiffeners, similar effects as with locally resonating sonic materials (LRSM's) could decrease the sound transmissions in a small frequency range (Liu, et al., 2000). Also, varying the resonance frequency can influence the modal density and behavior of the plate, resulting in a pass band – stop band phenomena similar to what LRSM'S have, see Figure 6.8. Designing the honeycomb plate in such a way that the stop band lies in the relevant low frequency range can greatly increase the (overall) sound reduction.

Figure 6.7 Detail drawing of the one grid block.

Figure 6.8 Transmission coefficient as function of the frequency. Opened dots are the measured transmission amplitude for a locally resonant sonic material. The solid squares are the transmission coefficients for a reference sample. Source: (Sheng, et al., 2003)
Adding ribs that stiffen the plate also creates a non-diffuse sound field, resulting in a directional attenuation of the vibration. Most likely the direction perpendicular to the ribs is affected the most by this effect, meaning that the amount of ribs can influence the sound reduction in a certain direction.

Figure 6.9 shows several floor systems that could be made similar to the honeycomb variant. Important hereby is the connection between the stiffeners and the floor. Adding stiffeners in the other direction would complicate the floor systems, but would be possible if it significantly improves the flanking sound reduction. Another option would be making precast concrete slabs with the actual honeycomb pattern, by creating cut-outs in the casting.

6.3.2. Wave Model

The wave model consist out of two coupled plates that have varying thickness, that is described by a sine function. This creates a wave like shape as is illustrated in Figure 6.10. Because the floor can become very thin in some places, stiffeners are added for stability. They also couple the two floors together, so that they can work together.

By varying the thickness of the plate, the mass and stiffness are varied inside the floor. The impedance jumps from the mass difference should increase the sound reduction, while the stiffness
will vary the resonance frequency. The latter could make the sound reduction significantly worse, or reduce the modal behavior of the plate in the low frequency region. While the variant has similarities to the honeycomb model, the waves are only made into one direction. Also, the transition is smoother which may have different results.

Figure 6.11 and Figure 6.12 show how the floor would look like for the wave variant. The dimensions that are given will be used as a reference case in chapter 8. Larger drawings can be found in the appendix, but it should be clear that only in the direction of the stiffeners the thickness is varied.

Figure 6.11 Floor design for the wave variant.

Figure 6.12 Detailed drawing of 1 wave.
6.3.3. Hinged model

The hinged model consist of a floor that is split up, and connected via hinges. A rough sketch is given in Figure 6.13 of a hinged connection. These can either be physical hinges, or something material that has little to no bending resistance. Since flanking sound transmission is dominated by bending waves, perhaps adding connections that cannot transfer bending forces increase the sound reduction. In ANSYS the hinges are modelled by making use of plastic hinges.

![Figure 6.13 Elements connected with hinges and resilient materials.](image)

By splitting the floor up, the resonance frequencies of the plate will increase. This is sometimes useful for very large surfaces, where the resonance frequency will become too low. Also, the amount of surface area that is excited, as well as the radiating surface, is severely reduced. This is however not modelled.

Figure 6.14 shows the placement of the hinged connections in the floor, while Figure 6.15 shows the size of the plastic hinge. Dimensions shown in the drawings will be used as a reference case in chapter 8. The placement of the hinged connections in Figure 6.15 will create an unstable floor, that needs to be supported by beams. These are however not modelled, since the acoustic pressure loads will be too small to cause any significant deformations.

![Figure 6.14 Floor design for the hinged variant.](image)

![Figure 6.15 Detailed drawing of hinged connection, as modeled in ANSYS.](image)
Whether hinges as small as 1 cm are large enough to affect low frequency sounds, is not yet know. However, if the bending wave speed is sufficiently low, the hinges might be relatively large enough. In Figure 6.16 some systems are shown that are similar to the hinged variant. The floor systems are mostly supported beams, while a hanging wall system could be connected by cables.

Figure 6.16 Examples of realistic systems similar to the hinged variant.

Figure 6.17 Detailed examples of hinged floor system
CHAPTER 7. ANSYS Model

The case will be evaluated with the finite element program called ANSYS. Calculations from this program should approximate the actual situation and sound reduction values. However, the results are still based on known formulas and only give an indication of the real world.

With the finite element it is tried to increase the sound reduction obtained specifically from the vibration velocity level difference $D_{v,ij}$ of Eq. (7.1). In other words, the sound reduction obtained from the junction. The contribution of the direct sound reduction and the normalization terms are considered as constant, so that the relative increase in flanking sound reduction is found. Results will be presented in the form of graphs that display the vibration velocity level difference, $D_{v,ij}$.

\[ R_f = \frac{R_i + R_j}{2} + D_{v,ij,\text{situ}} + \log \left[ \frac{S_i}{\sqrt{S_i S_j}} \right] \]  \hspace{1cm} (7.1)

With:
- $R_f$ = Flanking sound reduction [dB]
- $R_i$ = Direct sound reduction of plate i [dB]
- $R_j$ = Direct sound reduction of plate j [dB]
- $D_{v,ij,\text{situ}}$ = Vibration velocity level difference in field situation [dB]
- $S_s$ = Area of separating wall [m$^2$]
- $S_i$ = Area of element i [m$^2$]
- $S_j$ = Area of element j [m$^2$]

To obtain the velocity level difference, a harmonic response analysis is used in ANSYS. This analysis has the option to use modal superposition, which needs the output of a modal analysis. Input data and design considerations for both analyses are discussed in this chapter, starting with the modal analysis. Validation of the model is done via simple cases from which the results are known.

A basic model is defined, so that the variants in chapter 8 can be evaluated. For this case, the boundary conditions and limitations are discussed thoroughly, as the same boundary conditions and limitations apply to the variants.
7.1. ANSYS limitations

7.1.1. Model and element size

ANSYS Workbench 14.5 with an academic license is used to model and calculate the velocity vibration difference. While the academic license has access to almost all functions of ANSYS, there are size limits for nodes and elements. According to the website of ANSYS, the limit is 32,000 nodes/elements. As a result the maximum model size is relatively small. ANSYS recommends a minimum of 6, but rather 10, nodes per wave for accurate vibration analysis. To illustrate this, Figure 7.1 shows 3 sets of waves with decreasing number of nodes. It is clear that a minimum of 6 nodes is needed for multiple waves, excluding the end node of the system.

![Figure 7.1 Waves with 3, 6 and 12 data points.](image)

To obtain the minimum element length for the meshing of the model, the amount of waves has to be known. Which is determined by the wave speed for forced waves, and determined by the resonant modes for free waves. Because ANSYS uses modal superposition to calculate the solution, the amount of free waves is governing.

Modal superposition involves uncoupling the equation for vibration by using free vibrations mode shapes. The uncoupled equations are then solved, resulting in so called modal coordinates. Superposition of these modal coordinates are used to obtain the solution of the original equations (Xiao, 2013). This method severely reduces computation time compared to the full equation. To obtain an accurate solution, enough mode shapes are needed around the requested frequency.

Using formulas for the Eigen frequency from (Blevins, 1995), an estimation can be made of the modal shapes and location of the Eigen frequencies in the model. However, the book only describes the first 6 modes, which is not enough for a large frequency range. Using the formulas stated earlier, Eq. (7.2) and (7.3), similar results are found for the first 6 modes and thus will be used instead.

\[
\begin{align*}
    f_{nm} &= \frac{c^2}{4f_c} \left[ \left( \frac{n}{b} \right)^2 + \left( \frac{m}{n} \right)^2 \right] \\
    f_c &= \frac{c^2}{2\pi} \sqrt{\frac{m}{b}}
\end{align*}
\]  

In Table 7.1, the Eigen frequencies are calculated for a concrete plate of 200mm thick, with the following properties. The same properties will be used for the basic model that will be used as a reference.
To obtain an accurate solution in the harmonic response analysis for a certain frequency range, a modal analysis is needed with a slightly wider frequency range. If the largest frequency considered in the harmonic response analysis is chosen as 125Hz, the modal analysis has to consider frequencies up to 150Hz.

From the table can be seen that mode shape with the most knots in the considered frequency range, occurs at 143.71 Hz and has 19 knots in one direction, and 1 in the other (19,1). Which means 9,5
waves in the longest direction. Translating this into element size gives a minimum element length of 0.4629 meter \( \left( \frac{25}{60.9} \right) \).

From the modal analysis in ANSYS the highest modal shape found is (15,5) at 142.92 Hz, see Figure 7.2. Table 7.1 predicted that particular shape to occur at 149.19 Hz. The difference is caused by the support conditions of the plate in the model, and possibly the presence of the rest of the structure. Figure 7.3 shows a more complex modal shape, in which the entire structure is in motion. Again, Table 7.1 seems to predict the Eigen frequency with a small error. However, for the choice of mesh size the prediction suffices.
7.1.2. Frequency range

With a maximum modal frequency of 150Hz, the harmonic response up until 125 Hz can be calculated. This is chosen so that the first 8 one-third octave bands can be examined (25, 31.5, 40, 50, 63, 80, 100, 125 Hz). Attempting to calculate a wider frequency band containing higher frequencies would result in a very large and inaccurate model with long computation times.

However, the sound reduction of the junction should be relatively frequency independent according to the literature. For lightweight elements, a slight increase with the frequency is even found (NEN-EN12354-1, 2000). Any improvements found in the frequency range of 25-125Hz should thus be expected to improve the flanking sound reduction over the entire frequency range.

7.1.3. Plate elements

ANSYS has the possibility to create solids, plates and lines. Ideally, everything should be modelled with solids. However, to accurately calculate the (local) force distribution with solid elements would result in a very large number of elements and nodes. Also, to link adjacent elements the elements themselves should be comparable in size.

Plates are simplified elements in ANSYS that use less nodes and elements than solids without sacrificing accuracy. However, it does come with limitations for the design of the model. Stacking of plates becomes impossible, and out of plane plates can only be connected at the edge. Plate elements are created through a surface in the design modeler, and are given a thickness later in calculation.

The thickness of a plate can be given a direction (bottom, top or middle), that influences the characteristics of the plate, and creates a 3D shape. In the model the thickness direction is chosen so that the 3D model resembles the schematization of Figure 7.5.

Since the academic license has an element/node limit, plate elements will be used in the model. Also, mid-side nodes are dropped since no curved elements are used. Figure 7.4 shows the difference in nodes and element size between the discussed elements.

![Figure 7.4 Difference between solids and plates, with and without mid-side nodes](image)
7.1.4. Design simplification

For the calculations to be accurate and feasible, the real situation is simplified. The junction is modelled as schematized in Figure 7.5. Only the most important elements for the flanking sound reduction are modeled, neglecting finishing layers and substructures. Due the symmetric shape of the movie room, the velocity level difference only has to be calculated once.

![Figure 7.5 Schematization of the junction](image)

The floor and walls will be modelled as rigidly connected, while the rest of the connections are hinged. Floors and walls will be 25m x 10m and made of homogenous material. The symmetry axis of the model simplifies the model further, the modal coupling of the floors that follows will be studied separately.

7.2. Model build up

In ANSYS Workbench two linked analyses will be used to evaluate the model, as is shown in Figure 7.6. The modal analysis is used to find all the Eigen frequencies in the frequency range of 0-150Hz. In the harmonic response analysis, the vibrations due a harmonic pressure load with a frequency range of 25-125Hz will be calculated. Input data needed for these analyses are the geometry, material properties and boundary conditions.
The same geometry, material properties and boundary conditions are used for both analyses, which is indicated by the straight lines between the branches. The found Eigen frequencies of the modal analysis are used in the harmonic response analysis, indicated by the curved line.

![Figure 7.6 Linked analyses in ANSYS](image)

### 7.2.1. Geometry

The geometry model is setup parametrically as far as possible, to make a sensitivity or optimization study possible. Because of this, the geometry is made in the design modeler of ANSYS Workbench, instead of importing AutoCAD or Solidwork models. It also means that the geometry design might appear as overly complicated.

To create the geometry, several points and lines are specified based on the parameters below. The parameters These lines are then extruded and copied in a pattern to make a 2D floor, as is shown in Figure 7.7. The complete floor is then used multiple times to create the rest of the model, after that, the beam is constructed between the floors. Figure 7.8 shows the complete model, which consists out of four walls, two floors and one beam. The line bodies are suppressed because they were only used to construct the geometry and have no purpose in further analyses.

Parameters:

- Floor Width = 10 [m]
- Wall Height = 10 [m]
- Beam Length = 25 [m]
- Beam Width = 0.268 [m]
- Beam Height = 0.770 [m]
- Beam $t_f$ = 0.02 [m]
- Beam $d_w$ = 0.01 [m]
In the model, the homogenous floors are split up in 20 plates. This is done for reasons explained in the section Model validation. These 20 elements are made into 1 part so that it behaves as 1 plate. The connection between the wall and floor is assumed to be rigid, and the two elements are made into 1 part as well. The end result is 5 parts containing multiple surface bodies, as can be seen in the bottom of the model branch in Figure 7.8. The parts consist of: 2x wall+floor, wall, and the beam.
7.2.2. Material properties

The material properties are kept the same for every analysis by linking to a common engineering data branch. Instead of using the database provided by ANSYS, material properties are used defined, to have more control over the variables. To specify the material properties, the toolbox options density and isotropic elastic are used. This gives three parameters to define the material: density, young’s modulus and poison’s ratio, also see Figure 7.9. In table 4 the properties of the different materials that were used can be found. To conduct the sensitivity study, density and young’s modulus was often varied by a factor 5.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density [kg/m$^3$]</th>
<th>Young’s modulus [N/m$^2$]</th>
<th>Poisson’s ratio [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>7850</td>
<td>2e12</td>
<td>0,3</td>
</tr>
<tr>
<td>Concrete</td>
<td>2200</td>
<td>15e9</td>
<td>0,25</td>
</tr>
<tr>
<td>Concrete E/5</td>
<td>2200</td>
<td>3e9</td>
<td>0,25</td>
</tr>
<tr>
<td>Concrete E*5</td>
<td>2200</td>
<td>75e9</td>
<td>0,25</td>
</tr>
<tr>
<td>Concrete P/5</td>
<td>11000</td>
<td>15e9</td>
<td>0,25</td>
</tr>
<tr>
<td>Concrete P*5</td>
<td>440</td>
<td>15e9</td>
<td>0,25</td>
</tr>
</tbody>
</table>

Table 7.2 Material properties of used materials

![Figure 7.9 Material properties in ANSYS Workbench](image)

7.2.3. Boundary conditions

In the model, boundary and matching conditions are needed to solve the analysis. As this is only part of a structure, the floors are given artificial support conditions at the edges, see Figure 7.10. In reality, they are supported by a beams, and the floor fields are connected to each other. It is assumed the floor is separated every 10 meters, and that the support condition can be modelled as simply supported. In the other direction, the floors do not have a support condition, otherwise the floor would be over constrained. Walls are given the same support conditions.

At the junction between the floor and the beam, matching conditions are needed. This is done with a revolute joint, which simulates a hinged connection between the beam and the floor.
The beam itself is supported at the front and back side, as indicated in Figure 7.11. A structural model was made of the surrounding structure to simulate the support conditions of the beam. The obtained stiffness coefficients were used to define the spring support for the beam (bushing joint in ANSYS), that is founded on the ground. In Table 2.1, the used stiffness coefficients are given. These are obtained by removing the beam from the structural model, and applying a unit force or moment. The resulting deformations were used to deduct the stiffness.
<table>
<thead>
<tr>
<th>Node1</th>
<th>Per x [m]</th>
<th>Per y [m]</th>
<th>Per z [m]</th>
<th>Per θx [rad]</th>
<th>Per θy [rad]</th>
<th>Per θz [rad]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Δ Force x[N]</td>
<td>3.31E+07</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Δ Force y[N]</td>
<td>-8.16E+04</td>
<td>3.90E+08</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Δ Force z[N]</td>
<td>1.07E+03</td>
<td>-6.27E+04</td>
<td>8.26E+06</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Δ Moment [Nm]</td>
<td>-2.03E+06</td>
<td>9.74E+09</td>
<td>8.33E+07</td>
<td>2.44E+11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Δ Moment [Nm]</td>
<td>-8.31E+08</td>
<td>4.55E+06</td>
<td>-3.43E+08</td>
<td>-3.28E+09</td>
<td>3.50E+10</td>
<td></td>
</tr>
<tr>
<td>Δ Moment [Nm]</td>
<td>-3.21E+08</td>
<td>1.56E+10</td>
<td>-2.52E+06</td>
<td>3.90E+11</td>
<td>8.18E+09</td>
<td>6.27E+11</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Node2</th>
<th>Per x [m]</th>
<th>Per y [m]</th>
<th>Per z [m]</th>
<th>Per θx [rad]</th>
<th>Per θy [rad]</th>
<th>Per θz [rad]</th>
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<tr>
<td>Δ Force x[N]</td>
<td>3.28E+07</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Δ Force y[N]</td>
<td>-4.63E+00</td>
<td>3.89E+08</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Δ Force z[N]</td>
<td>-4.36E+00</td>
<td>6.26E+04</td>
<td>8.56E+06</td>
<td></td>
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<td>Δ Moment [Nm]</td>
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</tr>
<tr>
<td>Δ Moment [Nm]</td>
<td>-1.64E+09</td>
<td>-2.50E+06</td>
<td>-3.42E+08</td>
<td>-3.52E+09</td>
<td>9.57E+10</td>
<td></td>
</tr>
<tr>
<td>Δ Moment [Nm]</td>
<td>-3.15E+08</td>
<td>-1.557e10</td>
<td>2.50E+06</td>
<td>7.79E+11</td>
<td>1.56E+10</td>
<td>6.26E+11</td>
</tr>
</tbody>
</table>

Table 7.3 Found stiffness coefficients of the ends of the beam.

### 7.2.4. Damping

Damping specifies the amount of energy that is converted to heat by the structure. ANSYS has several ways to add damping to the system, such as constant damping and stiffness damping. As the model itself should provide the edge damping, and radiation damping is assumed to be very small, only internal damping has to be specified. This is done via adding a constant damping ratio in the harmonic response analysis, which is Rayleigh damping. Damping will be one of the parameters that will be varied, but is at first assumed to be 0.01% of the critical damping, as specified in (NEN-EN12354-1, 2000).

### 7.2.5. Other design considerations

Joints in ANSYS have the option to set the behavior of the participating elements to rigid or deformable. While the best results are obtained with a rigid behavior, all the edges of the beam are participating in a joint. This means the whole beam acts rigidly, and does not deform. Because of this, the beam behavior is set to deformable in the bushing joint.

For the meshing a quadratic form is chosen, since the model contains only simple plate elements. However, when the element size nears the minimum element size of the mesh, the mesh sometimes has to resort to triangles. This could be the cause of small difference in results. Figure 7.12 shows the meshing of the model.
7.3. Calculation procedure

In the harmonic analysis a pressure force is applied over a frequency range, which will bring the structure in motion. ANSYS can then compute the average directional deformation of an element or surface. Placement of the harmonic pressure load and measured surfaces on the floor can be seen in Figure 7.13. These surfaces are deliberately placed apart from each other to get the most realistic results, based on results from the validation at the end of the chapter.
Using Eq. (7.6) the Vibration velocity level difference between floor 1 and floor 2 can be calculated. Note that ANSYS calculates the average deformation instead of the root mean square velocity. Because there is a linear relation between deformation and velocity, the deformation can be used instead of the velocity. ANSYS seems to take the average of the absolute deformation instead of the mean square of the deformation, which leads to a small error in the system.

With the vibration velocity level difference, the junction vibration transmission coefficient $K_{ij}$ can be calculated. The correction term for the in-situ situation can be calculated analytically, or estimated to be -5dB according to (NEN-EN12354-1, 2000). For evaluation purposes, only the velocity level difference, $D_{v,ij}$, is used. In symmetric models $D_{v,ij}$ is only calculated in one direction, since the other direction should give exactly the same value.

A larger difference between the plate velocities of floor 1 and 2 will result in a higher $D_{v,ij}$, and thus less flanking sound transmission. For rigid junctions in heavy structures the value of $D_{v,ij}$ between 10 dB and 20 dB. In lightweight structures this value has to be significantly higher, preferably 30-40 dB.

$$v = id\omega$$
$$a = iv\omega$$
$$D_{v,ij} = 10\log \frac{<\hat{v}_i^2>}{<\hat{v}_j^2>}$$
$$K_{ij} = \frac{D_{v,ij} + D_{v,ji}}{2} + 10\log \frac{l_{ij}}{a_i a_j}$$
$$a_i = \frac{2\pi^2 S_i}{c_0 T_{s,i}} \sqrt{f_{\text{ref}}}$$
$$T_s = \frac{2.2}{f \eta}$$

With:
- $v$ = velocity [m/s]
- $d$ = Deformation [m]
- $a$ = Acceleration [m/s$^2$]
- $i$ = phase shift 90 degrees [rad]
- $\omega$ = Angular frequency [1/s]
- $<\hat{v}^2>$ = Mean square normal surface vibration velocity [m/s]
- $K_{ij}$ = Vibration reduction index from excited plate $i$ to radiating plate $j$ [dB]
- $K_{ji}$ = Vibration reduction index from excited plate $j$ to radiating plate $i$ [dB]
- $D_{v}$ = Vibration velocity level difference [dB]
- $l_{ij}$ = junction coupling length [m]
- $a$ = Equivalent absorption length [m]
- $S$ = Area [m$^2$]
- $c_0$ = Speed of sound in air [m/s]
- $T_s$ = Structural reverberation time [s]
- $f_{\text{ref}}$ = 1000 Hz [1/s]
- $f$ = Frequency [1/s]
- $\eta$ = Total loss factor [-]

Figure 7.14 and Figure 7.15 are examples of the frequency response calculation, showing the average deformation for plate S in floor 1 and plate R in floor 2. These plates are indicated in Figure 7.13, and stand for source plate and receiving plate.
The tabular data is exported to excel for post-processing as described above. Graphs are made from the found vibration velocity level differences. With 100 data points, this leads to one vibration velocity level difference value per frequency. Because the calculated frequency range has large wavelengths, $D_{v,ij}$ shows modal behavior, with numerous peaks and dips. A mathematically averaged $D_{v,ij}$ is added to increase readability. Also, the velocity is averaged over each one-third octave band to obtain the velocity averaged $D_{v,ij}$ as described in Eq.(7.10). These points are connected with lines to increase readability. Although this is not an energetic summation or average, it does devalues peaks. A large difference between the averaged $D_{v,ij}$ and the velocity averaged $D_{v,ij}$ per one-third octave band thus indicates modal behavior.
\[ \bar{D}_{v,ij} = 10 \log \left[ \frac{\frac{1}{n} (v_{i1} + v_{i2} + v_{i3} + \ldots)}{\frac{1}{n} (v_{j1} + v_{j2} + v_{j3} + \ldots)} \right]^2 \]  

(7.10)

With:
\[ \bar{D}_{v,ij} \] = Velocity averaged \( D_{v,ij} \) per one-third octave band
\[ v_i \] = Velocity of plate i
\[ v_j \] = Velocity of plate j

7.4. Model validation

In order to proof that the model is correct, a few validation tests were made. Measurements and thereof deducted formulas exist for simple junctions. Recreating those junctions in ANSYS and comparing them to the expected values according to (NEN-EN12354-1, 2000) will proof that the model is correct. Only the model containing three plates rigidly connected in a T-junction is briefly discussed, see Figure 7.16.

Figure 7.16 Model of T-Junction, 3D impression on the left and side view on the right

In these models the total mass, mass difference between elements and the young’s modulus was varied and the results compared to (NEN-EN12354-1, 2000). Based on these results the model was refined. The most important observation was that the placement of exciting and “measuring” had great impact on velocity level difference between the two floors. This is also a problem in real measurements, where distance, frequency and near field effects influence the flanking sound transmission measurements in lightweight or complex building components (Buchegger, et al., Inter noise 2014). Element attenuation and modal phenomena make the assumption that there is a diffuse sound field incorrect.

This was observed when the entire surface of floor1 is loaded with a harmonic pressure load, and the plate velocity is averaged over the entire surface of floor1 and floor 2 respectively. Figure 7.17 gives the expected value for \( K_{ij} \) for a T-junction of homogenous plates. When the mass over every element is equal, the vibration level difference, \( D_{v,ij} \), should be around \( \sim 10 \)dB. Figure 7.19 shows the vibration velocity level difference when the entire floor is excited and “measured” while in Figure 7.18 the loading and measuring is done as is indicated in Figure 7.16. It is clear that Figure 7.18 is the correct one.
Figure 7.17 Expected flanking sound reduction. Source (NEN-EN12354-1, 2000)

Figure 7.18 Calculated vibration velocity level difference of T-junction, loading and measuring is done as in Figure 7.16. Expected value is ~10dB for $D_{v,ij}$.

\[
K_{13} = 5.7 + 14.1 M + 5.7 M^2 \text{ dB / octave}
\]

\[
K_{12} = 5.7 + 5.7 M^2 ( = K_{23}) \text{ dB / octave}
\]  

(E.4)
A possible explanation could be that the source floor is subjected to both resonant and non-resonant vibrations, and the receiving floor is only subjected to resonant vibrations. While only resonant vibrations are assumed to be of importance for flanking sound transmission in (NEN-EN12354-1, 2000), in lightweight constructions non-resonant vibrations can contribute as well. Which are affected by distance, frequency and near field effects.

It is chosen to exclude any near field effects by separating the placement of the pressure load and the placement of the measurement. This is also specified in (NEN-EN-ISO10848, 2006), which provides guidelines for flanking sound transmission measurements. This was done by splitting the floor up in 20 elements, and loading and measuring only $1/20$th of the floor.

### 7.5. Basic model results

A short overview is given of the results from the basic model. These results provide a baseline for the evaluation of the variants. The result of the basic case is shown in Figure 7.20, and the reference values for the parameters can be found below. As a reminder, the model is shown again in Figure 7.21. Preliminary observations of the variations of these parameters are discussed as well.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2200 [kg/m$^3$]</td>
</tr>
<tr>
<td>Young’s modulus</td>
<td>15e9 [N/m$^2$]</td>
</tr>
<tr>
<td>Plate thickness</td>
<td>0.2 [m]</td>
</tr>
<tr>
<td>Floor width</td>
<td>10 [m]</td>
</tr>
<tr>
<td>Wall Height</td>
<td>10 [m]</td>
</tr>
</tbody>
</table>
Beam Length = 25 [m]
Beam Width  = 0.268 [m]
Beam Height = 0.770 [m]
Beam $t_f$   = 0.02 [m]
Beam $d_w$   = 0.01 [m]
Damping     = 0.01 [-]

Figure 7.20 Results of basic mode, will be referred to as the basic case.

Figure 7.21 3D impression of, and cross section of the basic model in ANSYS.
The effect of mass on the velocity level difference was studied by varying the density of the floors, while the rest of the parameters remained the same. Both increasing and decreasing the density reduces the velocity level difference, see Figure 7.22. The literature suggest that when the floor is lighter compared to the wall, the velocity level difference should be higher. Due modal phenomena, this is not the case. However, increasing the density does decrease the velocity level difference more, and has significantly less modal behavior.

Figure 7.22 Results of basic model with varying density for the floor. Left has density of 440 kg/m$^3$ and right a density of 11000 kg/m$^3$.

When the young's modulus was varied, similar effects were observed. Increasing the young's modulus had the same effect as decreasing the density, heavier modal behavior and a slight decrease in velocity level difference. The natural frequencies of the plate are closer to the frequency of the excitation, and the plates are more easily set into vibration.

Figure 7.23 Results of basic model with varying young's modulus for the floor. Left has young's modulus of 75e9 N/m$^2$, and right has a young's modulus of 3e9 N/m$^2$.

Increasing the damping decreases the modal behavior, as can be seen in Figure 7.24. Here, the damping is varied for the basic model. Increasing the damping reduces the amount of peaks and gives a more constant, and higher, velocity level difference. Reducing the amount of damping does exactly the opposite.
That the resonance frequency of the floors has a large impact on the velocity level difference is further investigated by varying the dimensions of the floor. Hereby, the floor is still split up in 20 plates. The floor width had the most influence on the velocity level difference. By reducing the width of the floor, the acoustic pressure load came closer to the surface were the plate velocity was calculated. This might have had an influence on the results, because when the floor width was increased, the influence was significantly less. Although the effects are less pronounced than for varying the stiffness and mass, the same results are found.

In the basic model the junction is perfectly symmetrical, which is unfavorable for the velocity level difference. Modal coupling between the two rooms are resulting in relatively large peaks and dips. By varying the resonance frequency of the receiving floor compared to the source floor, the rooms should be modally decoupled. This effect is investigated in Figure 7.26 and Figure 7.27 by varying the dimensions and the density of the receiving floor.
Figure 7.26 Results of basic model with modal decoupling through varying the dimensions of the receiving floor. Left has a floor of 25m by 5m, right of 25m by 7.5m.

While varying the dimensions of the floor has predominantly influence on the size of the peaks, varying the density has more effects. Because the receiving floor is lighter compared to the source floor, it is easier set into vibration. As a result, the velocity level difference is significantly less in the left graph of Figure 7.27. The opposite holds for the right graph. Modal decoupling thus influences both the modal behavior, and the amount of velocity level difference.

Figure 7.27 Results of basic model with modal decoupling through varying the density of the receiving floor. Left has density of 440 kg/m$^3$, right a density of 11000 kg/m$^3$

The initial results show that it is difficult to increase the velocity level difference by making changes to a homogenous floor. Damping and modal decoupling do have a positive influence on the velocity level difference, but the latter can have unforeseen effects. However, it is already clear that the flanking sound transmission depends on more factors than just the mass difference between elements, as (NEN-EN12354-1, 2000) suggests.
CHAPTER 8. Model variants

The three variants presented in a previous chapter were modeled and compared to the basic case that was described in the previous chapter. Boundary conditions are kept the same where ever possible, so the sound reduction improvement is only due the changes from the variants. In a similar way as the model validation, certain parameters will be varied to understand where the sound reduction improvement is coming from, and to find any mistakes in the model.

First, the model buildup of the variants will be shortly discussed, followed by the most interesting results. A large number of calculation has been done to support the found results, it is however impractical to present all of them. Therefore the rest of the calculation results can be found in the Appendix. Comparisons will be made with the mathematical averaged vibration reduction and the average of the energetic summation.

The same calculation method is used as in the previous chapter. Where the floor is divided into 20 plates, so that the placement of applied force and measurement can be separated. Where measurement refers to the calculation of the averaged plate vibration velocity in the direction normal to the plate is meant.

A sensitivity study is conducted for each variant to provide insight into the effects that increase or decrease the vibration reduction index. This is done by varying a single parameter and comparing that to the reference case with set values. Usually the parameter is increased by a factor 2 or 5, depending on the parameter, which will result in unreal properties and dimensions. However, since this is an abstract case, it is possible to study these effects and find a theoretical optimum.

After the parameters are evaluated separately, multiple parameters are varied together to find the maximum vibration reduction index. This will also reveal any non-linearity between separate effects, which is important to know.

8.1. Honeycomb panel

As a reminder what the honeycomb panel should look like, the model is shown in its entirety in Figure 8.2. Stiffeners are added every meter to form a honeycomb grid. In a real situation, the stiffeners should be on the underside of the floor, however for a vibrational analysis this should have no influence. The dimensions of the wall, floors and beams is kept the same as in the basic case. Any color differences in ANSYS models indicates separate body elements to increase visibility. This does not imply anything about material properties, connections or differences therein.

The drawing of Figure 8.1 is repeated for easier viewing. The dimensions in the drawing are the same as specified for the reference case, which will be given in section 8.1.2. In the drawings, the placement of the parameters is indicated as well.
Figure 8.1 Drawings of the honeycomb variant including parameter clarification.

Figure 8.2 3D Ansys model of the honeycomb model
8.1.1. Model build up

To compare the results between the variants and the basis case it is important that the same measurement and excitation placement is used. Therefore, the floor is divided into 20 plates, just like the basic case. As before, the model is setup parametrically so that any variations of the design can be made quickly.

Starting the same way as in the basic case, lines are made between a number of points. These points are based on the same parameters as before. Extrusions are made to create the floor elements, and copied with the pattern command into a complete floor consisting out of 20 plates (Figure 8.3). Plates for the stiffeners are made in the same manner, and are added in Figure 8.4.

Figure 8.3 Geometry build-up of the honeycomb model, floor partitioning according to the basic case

Figure 8.4 Geometry build-up of the honeycomb model, grid lines for the stiffeners are added

To prevent the overlap of the floor and the grid elements, the grid elements are united and intersected with the floor elements. This reduces the amount of body elements, and divides the total floor in 20 pieces of the same size as in the basic case(Figure 8.5). With the complete geometry formed, the elements can be mirrored and moved to obtain the entire model, see Figure 8.6.
Figure 8.5 Geometry build-up of the honeycomb model. The stiffeners, floor and grid are combined.

Because the model consist out of plates, the spatial geometry is limited to connected surface bodies. A thickness and a thickness direction is given to the plates so it resembles the basic case, and a realistic situation at the junction. This is the main reason that the stiffeners are on the top side of the floor. Boundary conditions are kept the same as the basic case. In Figure 8.6 the excited and measurement placement are highlighted in red.

Figure 8.6 Analysis set-up of the honeycomb model, red squares indicate the excited and measurement placements.

Meshing of structure is done by setting a maximum element size of 0,4m for the edges, and using the face mapping command. This provides a quadratic mesh, with some irregularities where the elements are smaller than the mesh. These irregularities will probably influence the result in a minor way, but increasing the mesh size further is not possible in the student version.

The honeycomb model is validated by setting the thickness of all the floor elements to 0,2 meter. This creates an homogenous floor that is exactly the same as the basic case, except for the meshing
and element distribution. The results can be seen in Figure 8.7, and closely resemble the results of the basic case. The small deviations are acceptable and allocated to the difference in meshing.

Figure 8.7 Result of the validation of the honeycomb variant.

8.1.2. Results

The following set of values are used as a reference case to evaluate the honeycomb variant. A sensitivity study is conducted to study the effects that influence the vibration reduction index. One parameter will be varied at a time, and compared to the reference case. When non-linear relations are found, multiple parameters are varied to explain the effects and find the local optimum. The most interesting effects are discussed below, and a complete overview of the conducted calculations can be found in the appendix.

Reference values:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Edge width (d1)</td>
<td>0.2 [m]</td>
</tr>
<tr>
<td>Edge thickness (h1)</td>
<td>0.2 [m]</td>
</tr>
<tr>
<td>Back plate thickness (h2)</td>
<td>0.05 [m]</td>
</tr>
<tr>
<td>Number of stiffeners x-direction</td>
<td>10 [-]</td>
</tr>
<tr>
<td>Number of stiffeners y-direction</td>
<td>25 [-]</td>
</tr>
<tr>
<td>Density edge</td>
<td>2200 [kg/m^3]</td>
</tr>
<tr>
<td>Young’s modulus edge</td>
<td>15e9 [N/m^2]</td>
</tr>
<tr>
<td>Density back plate</td>
<td>2200 [kg/m^3]</td>
</tr>
<tr>
<td>Young’s modulus back plate</td>
<td>15e9 [N/m^2]</td>
</tr>
<tr>
<td>Density beam</td>
<td>7850 [kg/m^3]</td>
</tr>
<tr>
<td>Young’s modulus beam</td>
<td>2e11 [N/m^2]</td>
</tr>
<tr>
<td>Damping</td>
<td>0.01 [-]</td>
</tr>
</tbody>
</table>
Results from the set of values above can be seen in Figure 8.8. Compared to the basic case, the reference case has 48% less mass, but the vibration reduction is significantly better.

Figure 8.8 Results of honeycomb variant with reference values (reference case).

By adding stiffeners, the floor becomes more stiff relative to its weight. As described in chapter 3, stiffness can influence the sound reduction in the low frequency range. By further increasing the stiffness, it becomes apparent that this indeed has a positive effect, see Figure 8.9 Results of honeycomb variant for varying stiffness. Left has concrete with a young’s modulus of 3e9 N/m², right with a young’s modulus of 75e9 N/m². Figure 8.9. Moreover, increasing the stiffness leads to a higher plate resonance frequency, resulting in modal behavior with larger dips and peaks in the vibration reduction. Decreasing the stiffness, and thus the plate resonance frequency, has a negative effect for low frequencies. However, it is not a linear relation, which means an optimum could be found around the reference case. When the density of the material is changed, almost identical results are found. Where increasing the stiffness has the same effect as decreasing the density.

Figure 8.9 Results of honeycomb variant for varying stiffness. Left has concrete with a young’s modulus of 3e9 N/m², right with a young’s modulus of 75e9 N/m².
Varying the stiffness and density seems to have the same effect as for the basic case, but it does not explain the increase in vibration reduction of the reference case compared to the basic case. By varying the thickness of the back plate, two limit cases can be examined. If the thickness goes to 0,2 meters, the basic case is obtained. And when the thickness goes to zero, only the grid of stiffeners remains. For the first case a decrease of the vibration reduction index was already observed. In Figure 8.10, the latter case is examined. It seems that there is a local optimum around a thickness of 0,01m.

The non-linear relationship makes it difficult to pinpoint where the improvement exactly comes from. Because of the geometry difference, there is a difference in plate resonance frequency between the back plate and the grid. By increasing the height of the back plate, these plate resonance frequencies are closer to each other. There could be a thickness where destructive interference between the resonance frequencies occurs. This would also explain the more constant vibration reduction index of the left figure.

Another possible explanation is that because the back plate of 0,01m thick has a resonance frequency (~80Hz) in the frequency range of interest (25-125Hz), the connection with the grid provides extra damping when the plate is excited near its resonance frequency. However, there seems to be no significant change at this frequency compared to the reference case.

Lastly, the thickness also influences the bending wave speed, and the bending wave impedance. By decreasing the thickness to such small dimensions, the impedance jump may be too large, reducing the damping effects of the back plate. The sound travels only through the grid, and a vibration reduction similar to the basic case is obtained. Moreover, the back plate might trap sound waves by being easily vibrated, without being able to pass sound waves on effectively.

To examine these effects further, the stiffness of the grid is varied. Lowering the young’s modulus makes the total floor less stiff, and decreases the difference in plate resonance frequency and bending wave impedance between the grid and the back plate. Figure 8.11 shows the results of varying the stiffness of the grid. A significant increase can be observed when the stiffness of the grid is lowered. Surprisingly, varying the density of the grid hardly seems to influence the vibration reduction index.
Figure 8.11 Results of honeycomb variant for varying edge stiffness. Left has concrete with a young’s modulus of 3e9 N/m², right with a young’s modulus of 75e9 N/m².

Figure 8.12 Sound transmission between honeycomb grid and back plate

Eq. (8.1) and (8.2) from (Craik, 1981) can be used to describe the sound transmission between plate 1 and 3 in Figure 8.12. In Eq. (8.3) and (8.4) the formulas are elaborated. An increase in $\psi$, or a decrease in $\chi$, results in an increase of the vibration reduction index according to Figure 8.13. The young’s modulus has slightly more influence than the density, because the young’s modulus also has influence on the difference between $\psi$ and $\chi$, Eq. (8.4). When $\psi$ is larger than $\chi$, the sound reduction will be higher. The formulas also explain the increase in vibration reduction when the thickness of the back plate is reduced. However, they do not explain the local optimum, nor the increase in vibration reduction index when the young’s modulus of the grid (plate 2) is lowered.

$$\psi = \frac{m_{s2}}{m_{s1}} \frac{f_{c1}}{f_{c2}}$$  \hspace{1cm} (8.1)

$$\chi^2 = \frac{f_{c2}}{f_{c1}}$$  \hspace{1cm} (8.2)

$$\psi = \frac{d_2^2}{d_1^2} \sqrt{\frac{E_{2} \rho_{2}}{E_{1} \rho_{1}}}$$  \hspace{1cm} (8.3)

$$\frac{\psi}{\chi^2} = \frac{E_2 d_2^5}{E_1 d_2^5}$$  \hspace{1cm} (8.4)
These formulas were meant for large homogenous plates, which gives rise to the question if the dimensions of the system are comparable to the wavelength. If the wavelength is too large, the floor can be considered as a homogenous plate with equivalent stiffness and mass. When this is not the case, each stiffener adds a small impedance jump that should increase the total vibration reduction index. Lowering the young’s modulus of the stiffeners should then increase the vibration reduction index.

According to the formulas above, varying the height of the stiffeners should greatly influence the vibration reduction index. However, both increasing and decreasing the height of the stiffeners results in a lower vibration reduction index. This again points to a local optimum, or a bandwidth of frequencies where the changes of variant are effective.

At the junction between floor and beam, similar effects are found. When the young’s modulus of the beam is drastically lowered (4e8 N/mm$^2$), the vibration reduction index increases to an average of 50 dB. This result illustrates the effect of rubbers, which also have a low young’s modulus. Surprisingly, increasing the stiffness young’s modulus of the beams also increases the vibration reduction index, although to lesser extents. Varying the geometry of the beam does not have significant effects.

Although varying the density of the grid did not have significant effects, varying the density of the back plate has, see Figure 8.14. Similar effects are found when the young’s modulus of the back plate are increased, which gave a higher vibration reduction index.

Figure 8.13 Transmission loss from plate 1 to plate 3. Taken from (Craik, 1981).

Figure 8.14 Results of honeycomb variant for varying back plate density. Left has concrete with a density of 11000 kg/m$^3$, right with a density of 440 kg/m$^3$. 
An explanation might be that the varying the density of the back plate affects only the transmission loss in the floor, while varying the density of the edge also affects the transmission loss at the junction between the floor and beam. Moreover, the density differences between wall and floor can have a large influence as well.

When the young’s modulus of the back plate is increased even further, the vibration reduction index decreases again. Similar results are found for decreasing the density further. Again, local optima are found.

By combining all the positive effects on the vibration reduction index, the vibration reduction index is less than when a single positive effect is used (Figure 8.15). However, it does increase the vibration reduction index significantly, and a local optimum might be found that is even better.

![Figure 8.15 Results of honeycomb variant for combing positive effects. The back plate has a young’s modulus of $75e9 \text{ N/m}^2$ and a thickness of 0.01m. The grid has a young’s modulus of $3e9 \text{ N/m}^2$.](image)

In practice, floors span in only one direction, and a honeycomb would thus be inefficient. Therefore another test was done, with stiffeners in one direction, see Figure 8.16. Contradictory to what is expected, the stiffeners in the x-direction increase the flanking sound transmission. The added mass impedance jumps in the flanking sound path apparently do not have the wanted effect. Stiffeners in the other direction make the floor more stiff in the spanning direction, and have a large positive influence on the flanking sound reduction. The orthotropic properties of the floor seem to have a large influence on the flanking sound transmission, which is something that is not yet considered.
8.1.3. Conclusions honeycomb variant

An increase in the vibration reduction index in the honeycomb variant is found due to a combination of effects, it is therefore difficult to assign the increase in the vibration reduction index to one effect. Impedance jumps between the back plates, edges and beam all influence each other, and the same applies to the modal coupling between elements. However, a significant improvement compared to the basic case of ~10 dB has been accomplished while reducing the weight of the floor. The interaction between the grid of stiffeners and the back plates results in additional flanking sound transmission loss, without changing the connection at the junction itself. Also, more variables than mass have an effect on the flanking sound transmission, at least for low frequencies. This is a promising result for lightweight structures with high sound reduction requirements.

The directional properties of flanking sound transmission are very interesting, and could be exploited in the acoustic design of buildings. However, when this aspect is not considered, the flanking sound transmission could be lower than expected in several paths.
8.2. Wave variant

The idea for the wave floor consisted of two plates with varying thickness following a sinus function, creating a waved shape. These two plates will have a phase shift in thickness of pi, resulting in a constant gap between them. As with the honeycomb variant, the varying height of the plate will result in differences in bending wave speed, impedance and resonant frequency. In this variant the changes are smoother, and only present in one direction. For structural considerations, baffles are added every 5 meters to span from the support to the beam (10m). These will also provide the connection between the two plates.

The drawing of Figure 8.17 is repeated for easier viewing. The dimensions in the drawing are the same as specified for the reference case, which will be given in section 8.1.2. In the drawings, the placement of the parameters is indicated as well.

![Figure 8.17 Drawings of the wave variant, including placement of parameters.](image)

8.2.1. Model Build up

Only a few elements have to be added to the basic case to obtain the wave variant. The floor can be copied to create the double floor, so only the baffles have to be added. This is done via an extrusion of an earlier created line, which is copied in a pattern similar to the floor in the basic model. An end plate is added at the support, in order to correctly support both floors. Floor elements are then copied and moved to obtain the complete model.
In the modal analysis a thickness branch is added to the geometry section. This allows the use of a function, or table, for the thickness of a selection of elements. Since the input of the function had limitations, such as not recognizing a sinus function, an excel sheet is used to create a table and import those into ANSYS. The thickness is assigned to the floor as in Figure 8.19, and a wave like shape is obtained. In Figure 8.20 the baffles are removed so the two wave shaped floors are recognizable.
8.2.2. Results

A sensitivity study is conducted around a reference case with values specified below. Comparisons were made by varying one parameter around the reference case. Although no validation calculation is made, initial results are comparable to both the honeycomb variant and the basic case. This paragraph will discuss the most interesting results, the complete list of results can be found in the Appendix.

Reference values:

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density floor</td>
<td>2200 [kg/m³]</td>
</tr>
<tr>
<td>Young’s modulus floor</td>
<td>15e9 [N/m²]</td>
</tr>
<tr>
<td>Density stiffeners</td>
<td>2200 [kg/m³]</td>
</tr>
<tr>
<td>Young’s stiffeners</td>
<td>15e9 [N/m²]</td>
</tr>
<tr>
<td>Density beam</td>
<td>7850 [kg/m³]</td>
</tr>
<tr>
<td>Young’s modulus beam</td>
<td>2e11 [N/m²]</td>
</tr>
<tr>
<td>Number of waves in floor</td>
<td>4 [-]</td>
</tr>
<tr>
<td>Average thickness bottom floor</td>
<td>0,06 [m]</td>
</tr>
<tr>
<td>Average thickness top floor</td>
<td>0,06 [m]</td>
</tr>
<tr>
<td>Total height floor</td>
<td>0,2 [m]</td>
</tr>
<tr>
<td>Amplitude (a)</td>
<td>0,02 [m]</td>
</tr>
<tr>
<td>Baffle thickness</td>
<td>0,05 [m]</td>
</tr>
<tr>
<td>Number of stiffeners</td>
<td>6 [m]</td>
</tr>
<tr>
<td>Junction symmetry</td>
<td>yes [-]</td>
</tr>
<tr>
<td>Damping</td>
<td>0,01 [-]</td>
</tr>
</tbody>
</table>

The results from the calculation with the reference values can be seen in Figure 8.21. Compared to the basic case, an increase in vibration reduction index of ~5dB is found. Moreover, the vibration reduction index is slightly more constant over the frequency range. Compared to the basic case, the floor has 39% less mass.
By varying the parameters that control the wave shape of the floor, the amplitude and number of waves in the floor, it became clear that the wave shape did not improve the vibration reduction index. Figure 8.22 shows that increasing the amplitude slightly lowers the average vibration reduction index. However, it has a similar effect on the vibration reduction index as damping, decreasing the size of dips and peaks.

When the average thickness was decreased, similar effects were found. With an average thickness of 0.04 meter for both plates, the vibration reduction index did not go below 21 dB. Where the thickness is very small, damping has more effect because the deformations are larger. Because the damping is kept constant, adding thin plates in the flanking path increases the damping.

Like the amplitude, varying the amount of waves in the floor had little effect on the average vibration reduction index. Lowering the amount of waves resulted in heavier modal behavior, with distinguishable dips and peaks. This is most likely caused by the modal coupling of the top and bottom plate.
Varying the young’s modulus and density of the floor plates had less effect on the vibration reduction index than for the honeycomb variant. However, increasing the young’s modulus and decreasing the density still led to an increase in vibration reduction index. Contrary to the honeycomb variant, density had a larger effect than the young’s modulus for the wave variant, see Figure 8.23.

Similar results were found when the young’s modulus and density of the stiffeners was varied. However, removing the stiffeners or bottom floor severely reduced the vibration reduction index. This indicates that the double leaf construction has a large impact on the vibration reduction index. Moreover, it is likely that the stiffeners decrease the flanking sound transmission similar as in the honeycomb variant.

To see if the found results are linear, a calculation is made with all the positive modifications, see Figure 8.24. As a control, a normal double leaf floor is calculated without wave shape (a=0). The right figure illustrates that the wave shape influences the modal shape while slightly lowering the average vibration reduction index. However, combining multiple effects increase the vibration reduction index significantly. The cavity has become larger, and the plates thinner, which has led to a more pronounced double leaf system as can be seen in Figure 8.25. Increasing the cavity further, by increasing the total height of the floor has the same result.
Figure 8.24 Results of wave variant with optimum parameters and normal situation. Left has stiffeners with a young’s modulus of $3\times10^9$ N/m$^2$, floors with a density of 440 kg/m$^3$, an average thickness of 0.04 m for, an amplitude of 0.01 m and 2 waves total. Right has an average thickness of 0.08 m and an amplitude of 0m.

Figure 8.25 Cross section of the ansys model with optimum parameters as specified in Figure 8.24.

The effects that increase the vibration reduction index do not interfere with each other, and agree with the known literature. A weaker connection between the two leaves and a larger mass difference between floor and wall, combined with the more effective damping for thin plates leads to an increase in vibration reduction index. Where the mass difference has the most effect.

Although the improvement seems to come from the separation between the leafs of the double leaf floor, the stiffeners play an important role. As Figure 8.26 illustrates, both increasing and decreasing the amount of stiffeners has a negative effect on the velocity level difference. Connecting the leafs increases the dampening of the system, but if the connection is to stiff, the floor acts as a whole and the basic case is returned. Not enough stiffeners leads to larger deformations, and thus a lower velocity level difference. Removing the stiffeners completely decreases the velocity level difference significantly.
Figure 8.26 Results of wave variant with varying amount of stiffeners connecting the top and bottom leaf of the floor. Left has 4 stiffeners, right has 11.

Changing the interaction between the two leaves of the floor by varying the average thickness of each plate has a large impact on the velocity level difference. The best results were obtained when the bottom plate was thicker than the top plate, as can be seen in Figure 8.27. This has to do with the impedance jump from the bottom leaf to the beam, which is larger in the left graph.

Figure 8.27 Results of wave variant with varying average thickness between the leaves of the floor. Left with d1=0.12 and d2=0.04, right with d1=0.04 and d2=0.12

Calculations that were made with the wave shape in the other direction, parallel to the wall, had the same dependencies as discussed for the floor with the wave shape in the x-direction. The overall velocity level difference did however not improve, but instead was worse. By adding impedance barriers parallel to the direction in which the velocity level difference is calculated, the sound waves are focused into a lane. Similar results were found for the hinged model, which is discussed next.

8.2.3. Conclusions wave variant

A wave shaped floor did not seem to have a positive effect on the velocity level difference. The increase in velocity level difference of the modal is due to the double leaf system and the stiffeners. Interesting hereby was that the connection between the two leaves decreased the flanking sound transmission between the two floors. However, this would most likely severely decrease the direct sound transmission. Modal decoupling of the floors did seem to have a positive effect.
8.3. **Hinged Model**

Since bending waves are the dominant wave form inside elements, the hinged model is designed to reduce the propagation of those waves. By adding hinges to the floor, no bending moments can be transferred, and hopefully also no bending waves. It also increases the resonance frequency by separating the floor in elements with smaller dimensions. Although this is generally a bad design choice, a resonance frequency that is too low can come to close to the frequencies of certain loads, such as walking people.

The floor will become unstable by adding hinges, and it will deform like a cable. In reality these hinges should thus be made above a support, or the floor should span in the other direction. However, the latter is impossible with hinges in two directions.

The drawing of Figure 8.28 is repeated for easier viewing. The dimensions in the drawing are the same as specified for the reference case, which will be given in section 8.1.2. In the drawings, the placement of the parameters is indicated as well.

![Figure 8.28 Drawings of the hinged model, including parameter placement.](image)

8.3.1. **Model build up**

After many attempts to modelling the hinged variant with the connection options provided by ANSYS, no correct model has been found. To simulate the hinged variant, a model is made that uses plastic hinges instead of hinged connections. This is done by adding strokes of material that have a lower young's modulus in one direction. By specifying an orthotropic material, the young's modulus could be changed while the shear modulus remained the same. However, this required the Poisson’s ratio to be zero. Because the shear modulus is unchanged, the deformations at the place of the plastic hinge are kept within reason.

The hinged model does not differ much from the basic model. Again, the floor is divided into 20 elements so that the same areas as in the basis case can be excited and measured. Thin strips are
added, that will later be given the properties of a plastic hinge. Figure 8.29 shows the strips of the plastic hinges.

![Figure 8.29 Geometry buildup of the hinged model, showing the strips to model the plastic hinges.](image)

The surface area of the strips is then subtracted from the surfaces that make up the floor. Floor parts and the strips are modelled as 1 part. Finishing the model is done in the same manner as for the basic case, where model is mirrored to obtain the entire junction.

![Figure 8.30 Complete geometry buildup of the hinged model.](image)

Hinged connections in Figure 8.31 are highlighted in green, and are assigned a different material than the rest of the floor in the design modeler. In the receiving floor, the hinged connections are in the same place.
8.3.2. Results

A sensitivity study is conducted around a reference case with values specified below. Comparisons were made by varying one parameter around the reference case. This paragraph will discuss the most interesting results, the complete list of results can be found in the Appendix. When the material of the plastic hinges was chosen to be concrete, the basic case was obtained again.

Reference values:

- Density floor $= 2200 \text{ kg/m}^3$
- Young’s modulus floor $= 15\times 10^9 \text{ N/m}^2$
- Density plastic hinge $= 2200 \text{ kg/m}^3$
- Young’s modulus plastic hinge $Ex = 15\times 10^7 \text{ N/m}^2$
- Young’s modulus plastic hinge $Ey = 15\times 10^9 \text{ N/m}^2$
- Young’s modulus plastic hinge $Ez = 15\times 10^9 \text{ N/m}^2$
- Density beam $= 7850 \text{ kg/m}^3$
- Amount of hinges y-direction $= 3$ [-]
- Width of plastic hinge $= 0.01 \text{ m}$
- Damping $= 0.01$ [-]

In Figure 8.32 the results for the hinged model with the reference value is given. The plastic hinges are only 10 times less stiff then the floor, and no significant increase in velocity level difference can be seen. There is however increased modal behavior, with larger and wider peaks and dips. This can be explained by the increased bending resonance of the floor, which is now divided into four plates.
Decreasing the young’s modulus of the plastic hinge will make rotation in the hinge easier. Because the acoustic pressure load is quite small, the young’s modulus must be very low to be able to function as an actual plastic hinge. Figure 8.33 shows that decreasing the young’s modulus increases the average velocity level difference by ~10dB. Most likely the plastic hinges provide a bending impedance jump that limits the sound propagation in the floor.

The width of the plastic hinges does not seem to have a large influence on the velocity level difference when strip of material already acts as a plastic hinge. However, when the young’s modulus of the material is not low enough to deform plastically, a wider strip of material will increase the
velocity level difference. Figure 8.34 illustrates this. That decreasing the width of the plastic hinges gives a slight increase in velocity level difference is due to modal phenomena.

Figure 8.34 Result of the hinged model for varying width of the plastic hinge. Left has a width of 0.002m, right a width of 0.05m.

Increasing the amount of plastic hinges gives an enormous increase in velocity level difference as can be seen in Figure 8.35. When the young’s modulus is not low enough for plastic deformation to occur, the amount of hinges seems to have a negative effect on the velocity level difference. This is probably a result of the change in resonance frequency and the many floor plates that are still coupled.

Figure 8.35 Result of the hinged model for amount of the plastic hinges. Left has a 10 plastic hinges with a young’s modulus of 15e8 N/m², right has 10 plastic hinges with a young’s modulus of 15 N/m².

The enormous increase in velocity level difference is contributed to the increased amount of impedance jumps between the excited plate and the measured plate. Because there are plastic hinges inside the measured areas, the decline in vibration over those hinges is measured. For lightweight elements where the bending wave speed is sufficiently low, additional bending wave impedances in the form of plastic hinges or rubbers could be introduced near the junction to increase the velocity level difference.
Adding hinges in the other direction, perpendicular to the wall drastically lowers the velocity level difference. Having hinges in both directions somehow focuses the energy in the measured areas of floor1 and floor2. Figure 8.36 shows the results for a floor with plastic hinges in both directions, with an appropriate low young’s modulus in the needed direction.

![Graph showing sound reduction](image)

**Figure 8.36** Result of the hinged model for 10 plastic hinges in x- and y-direction. Hinges in x-direction have an Ey of 15 N/m², hinges in the y-direction have an Ex of 15 N/m².

### 8.3.3. Conclusions of hinged variant

Hinges can provide extra impedance jumps in the floor, which increases the velocity level difference between floors. This effect is directional, and decreases the velocity level difference in the other direction. This directionality was also found with the honeycomb variant, and is most interesting. It provides a way to smartly decrease the flanking sound transmission, but could lead to problems when it is not considered.

Adding hinged connections only has an effect when the bending waves are already inside the floor, so the most effective place to add hinges would be near the junction. Because the hinged connections reduce the propagation of bending waves inside the floor, they may also have a positive effect on the radiation factor. Or at least reduce the surface area that is radiating sound.
8.4. Research Observations

Based on the finite element model, the following observations have been made.

1) The calculated velocity level difference depends heavily on where the force is applied, and at which points the plate velocity is measured or calculated. When the plate velocity is measured to close to where the force is applied, near field effects heavily influence the velocity level difference. Similar, plate velocity measurements on symmetric locations, close to edges or close to supports influence the velocity level difference greatly. This makes modeling of real situations in a finite element model difficult, and the results uncertain. Great care should thus be taken when interpreting the results of finite element models.

2) The vibration velocity level difference is far from constant in the low frequency range, due to modal phenomena. The modal behavior of the velocity level difference is something that is not exclusively for the low frequency range according to the model. However, the accuracy of the calculation is significantly worse for higher frequencies, and the effects may be less significant than observed. Since only a small frequency band was evaluated in the experiments, any change in the resonance frequency will shift velocity level difference curve, which might lead to incorrect conclusions. These inconsistencies have been filtered by only looking at relatively large overall changes to the velocity level difference.

3) The mass difference between floors and walls has the same effects on the sound reduction for low frequency sounds as for higher frequency sounds. The mass differences between floors and walls influences the flanking sound transmission. Elements with lower mass can lose relatively more sound energy to heavy elements than vice versa, resulting in a higher velocity level difference when the mass of the floor is lowered. Although this relation is already proven for heavyweight junctions with fixed connections, it also seems to be true for lightweight elements with a more complicated junction. Mass impedance jumps can thus be used for both direct and flanking sound transmission.

4) Stiffness is just as important as, and sometimes even more than, mass for the velocity level difference. Stiffness differences between elements has an equally great effect on the velocity level difference, as mass differences. Whether the effect is indirect, by influencing the resonance frequency and the bending wave speed, is not clear. The latter is particularly effective for low frequencies, even more so in combination with damping mechanisms. Increasing the resonance frequency of the floor has complex consequences for the velocity level difference, but foremost increases the modal behavior. Which should be avoided to obtain the best total sound reduction index. However, studies have shown that with locally resonant sonic materials, the sound reduction can be increased over a small band width (Liu, et al., 2000). Either way, stiffness can also be used to create impedance jumps, and to decrease the flanking sound transmission.

5) Geometric differences can have a significant influence on the velocity level difference, mainly by reducing the bending wave speed and increased damping. The dimensions of an element have effect on a multitude of important characteristics, such as resonance frequency, stiffness and bending wave speed. Varying the young’s modulus and dimensions have different results, although they both influence the stiffness. Especially the combination of geometry differences with varying young’s modulus and density creates impedance jumps which increases the sound reduction. Interaction between the two leaves of a wall, or between an floor element and a stiffener, results in an increase of the velocity level difference. Adding damping in the connection can increase the sound reduction further (Zegers, 2011). Although
less effective than the stiffness and mass, geometric differences can be used to decrease the flanking sound transmission.

6) **Damping gives the most consistent increase in sound reduction, and limits the modal behavior.**
Increasing the (internal)damping of the floor increases the velocity level difference significantly. Moreover, it reduces the modal behavior, resulting in a more constant velocity level difference. A more consistent sound reduction makes the sound insulation design easier, and more reliable. It should thus be tried to increase the amount of damping in buildings, so that modal behavior is damped, and the flanking sound transmission is more constant.

7) **Adding separation measures such as hinges, or rubbers can greatly increase the vibration level difference.**
By far the most promising results were seen when hinges were introduced to create more separations in the floor. This creates extra impedance jumps in the flanking path, which can cause a direction dependent increase in velocity level difference. In practice this is already done in various forms, such as springs, buffer zones and box-in-box constructions. The addition of hinges can provide new design strategies.

8) **Flanking sound transmission is direction dependent due to the orthotropic properties of structural elements.**
When inhomogeneous floors were modelled, with stiffness varying in two directions, the flanking sound transmission also varied in two directions. While one direction became worse, the other improved. This directional flanking sound transmission appeared in both the honeycomb and the hinged model. Floor such as hollow core slabs and composite steel decks have directional stiffness, and would thus also have direction dependent flanking sound transmission.
CHAPTER 9. Conclusion and Recommendations

This chapter will discuss the results of the research into flanking sound transmission, focusing on lightweight structures and low frequency sounds. Finite elements models have been made to simulate the velocity level difference between two floors, with the goal to understand and improve the flanking sound transmission. Not a lot of studies have researched low frequency sounds in lightweight elements, which is a complex area subjected to modal behavior. Therefore, the conclusions made are based on limited observations, and very specific to the studied subject.

In the conclusion the following research question will be answered based on the finite element models.

*How can the flanking sound transmission be reduced in lightweight elements without impairing the structural system, so that the sound insulation of lightweight structures can be improved?*

### 9.1. Conclusions of research

From the results of the variants is concluded that sufficiently high junction vibration reduction indices can be obtained with lightweight floors, see Table 9.1. Compared to an homogenous slab of concrete, the lightweight configurations have a higher flanking sound reduction, and lower weight.

<table>
<thead>
<tr>
<th>Model</th>
<th>Weight</th>
<th>Practical $D_{v,ij}$</th>
<th>Found theoretical maximum $D_{v,ij}$</th>
<th>Improvement due to</th>
<th>Remarkable findings</th>
<th>Most influential parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Basic</td>
<td>100%</td>
<td>~30 dB</td>
<td>~30 dB</td>
<td>-</td>
<td>Simply edge supported slabs already have a high $D_{v,ij}$</td>
<td>Density and Young's modulus</td>
</tr>
<tr>
<td>Honeycomb</td>
<td>52%</td>
<td>~35 dB</td>
<td>~40 dB</td>
<td>Interaction between stiffeners and floor plates</td>
<td>Local optima and minima</td>
<td>Young's modulus and floor thickness</td>
</tr>
<tr>
<td>Wave</td>
<td>60%</td>
<td>~35 dB</td>
<td>~45 dB</td>
<td>Stiffeners and extra impedance jump</td>
<td>Connection between leafs is essential to improvement</td>
<td>Density and floor thickness</td>
</tr>
<tr>
<td>Hinged</td>
<td>100%</td>
<td>~40 dB</td>
<td>~50 dB</td>
<td>Extra bending impedance jumps</td>
<td>Direction dependant $D_{v,ij}$</td>
<td>Young's modulus and the amount of hinges</td>
</tr>
</tbody>
</table>

Table 9.1 Results of the variants from the finite element models.

A large part of the flanking sound reduction in the variants is obtained by bending impedance jumps, which can be achieved by either density, young's modulus or thickness differences between elements. Introducing such impedance jumps in flanking elements can improve the flanking sound reduction without complicated detailing. The relation between the differences and the flanking sound reduction is not linear, and multiple bending impedance jumps can influence each other. Moreover, multiple types of differences between two elements does not result in a higher sound reduction.
The non-linearity is mostly due to the resonance frequency of elements, and the modal coupling of those elements. Therefore, damping and modal decoupling show large improvements in the flanking sound reduction.

Another interesting non-linearity is that flanking sound is direction dependant when the floor as a whole has orthotropic properties. The sound energy is focussed into the weakest direction, depending on stiffness and amount of impedance jumps. Since structural elements can usually be described as orthotropic, this effect is often present.

To answer the research question:

Due to the complex nature of flanking sound transmission in lightweight buildings, there are only a few methods that consistently increase the sound reduction. These methods mostly consist of adding damping, impedance jumps and increasing separation. It is recommended that these methods are utilized first.

Separation is most easily accomplished by increasing the distance over which the sound should be attenuated. Placing noisy and quiet zones apart from each other is both logical and effective. Buffer zones can be used in between, which can be unoccupied zones or zones where the noise requirements are less strict. A smart building design can drastically reduce the amount of (flanking) sound reduction needed.

To increase the flanking sound reduction further, it is recommended to add extra impedance jumps in the flanking path. Stiffness, mass and geometry differences can help accomplish this, in forms such as springs, rubbers or physically disconnecting elements. Multiple impedance jumps can be used to achieve very high flanking sound reduction indexes. Great care has to be taken that no short cuts for the flanking path are designed, and that the design is properly executed.

Modal behaviour is a complex phenomenon, which can cause large dips in the sound reduction when the resonance frequency of elements are too close each other. These effects are hard to predict accurately, and should thus be treated carefully. Adding damping helps both with increasing the sound reduction, and reducing the modal behaviour. Lowering the bending wave speed in elements can increasing the effectiveness of damping.

Adding impedance jumps in the flanking element, such as floor and walls, could provide extra flanking sound reduction. These jumps can be placed anywhere on the flanking path, but are most effective on converging places. For direct sound insulation adding impedance jumps inside the element could increase the sound reduction as well. By reducing the amount of flanking sound in the separating wall, the effectiveness of the double wall is increased as well.

It should be clear that a lot of factors influence the flanking sound transmission of low frequency sounds in lightweight buildings, besides mass and mass difference. This has led to the conclusion that it is possible for lightweight buildings to have sound reduction indexes as high as heavy weight buildings. However, this will require a complex design that might not be optimal for some applications. Results have shown that the velocity level difference can have values as high as 50 dB, which means that the flanking sound transmission can be sufficiently reduced. Lightweight elements make smart use of the characteristics that increase the sound reduction, and the low bending wave speed of small elements make low frequency sounds easier to attenuate. Making them potentially better than heavyweight elements.
9.2. Recommendations

Before the results of this research can be used in practice, laboratory test are needed to confirm the results of the finite element models. However, assuming that the results from the finite element models are correct, the following recommendations are made.

The direction dependant flanking sound transmission is something that is not yet exploited in acoustic designs. Guiding the sound energy towards dampening systems or noisy areas would reduce the need for flanking sound reduction in the other directions. More research is needed in this area, especially to see how much this effect is present in structural elements in practice. Possibly new floor designs that exploit the direction dependant flanking sound transmission can be investigated.

Applying the results of the finite element models to existing floor types is difficult, due to the local optima and minima that were found when different elements were combined. Rough indications of floor designs are however given in the appendix, which should be thoroughly tested before being used.

Smaller floor spans would make the floor stiffer, and offer more places for separation and bending impedance jumps in the form of rubbers. For example, composite steel deck floors offer lightweight solutions, and the possibility to increase damping by adding rubbers between the beam and steel deck. Rubber separation strips could also be added in the concrete layer, to add bending impedance jumps in the floor.

Floating floors are already used in practice, and can also be used to improve the flanking sound reduction of low frequency sounds. If floating floors become the standard for finishing layers, acoustic requirements of structural elements could be lower. Important hereby is that the floor is dilated from the walls. Experiments to increase the effectiveness of the floating floor could include stiffening elements, larger cavities and modal decoupling.

New materials are being produced that have manmade properties. Materials with negative young’s modulus would provide excellent bending impedance jumps and high damping. Possible uses for such materials is being researched for direct sound transmission, but less so for flanking sound transmission.

Contractors should try to increase the separation between elements when more sound reduction is needed, and prevent accidental sound bridges. Connections that are not made resilient could become a dominant flanking path, and should be prevented when a high sound reduction is required. Adding buffer zones such as hallways between movie theatres rooms increases both the amount of bending impedance jumps and the length of the flanking path. As a result, the flanking sound reduction should improve drastically. When this is not possible, adding extra bending impedance jumps will improve the flanking sound reduction.

The use of lightweight elements for high sound reduction purposes is definitely possible, although a complex design is required. Moreover, the sound design of a lightweight structure is far more unforgiving than a heavyweight one. It is therefore advised that to make connections easily accessible, so that they can be inspected when the sound insulations is less than predicted.
BIBLIOGRAPHY


Buchegger, B., Ferk, H. & Meissnitzer, M., Inter noise 2014. *Flanking sound transmission in an innovative lightweight clay block building system with an integrated insulation used at multifamily houses*. Melbourne Australia, Graz University of Technology, austria.


Yang, Z. et al., 2010. *Acoustic metamaterial panels for sound attenuation in the 50-1000Hz regime*. Cairo, s.n.

## APPENDIX A Sound reduction improvement of added linings

### Table 9.1 Sound reduction index improvement from extra layers. Source: NEN-EN12354-1

<table>
<thead>
<tr>
<th>Construction additional layer</th>
<th>$\Delta R$ [dB] in octave bands [Hz]</th>
<th>$\Delta R_w$ [dB]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Basic wall 100 mm gypsum blocks, 80 kg/m$^2$:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>12.5 mm plaster board; 44 mm cavity with 25 mm mineral wool; no studs</td>
<td>0 2 14 23 24 19 18</td>
<td></td>
</tr>
<tr>
<td>12.5 mm plaster board; 73 mm cavity with 50 mm mineral wool; wooden studs</td>
<td>2 8 15 23 25 21 21</td>
<td></td>
</tr>
<tr>
<td>12.5 mm plaster board; 60 mm cavity with 50 mm mineral wool; metal studs, isolated from wall</td>
<td>2 8 15 24 25 20 21</td>
<td></td>
</tr>
<tr>
<td><strong>Basic wall 175 mm plastered porous concrete, 135 kg/m$^2$:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>12.5 mm plaster board; 40 mm mineral wool; metal studs$^1$</td>
<td>3 12 14 15 17 15 15</td>
<td></td>
</tr>
<tr>
<td>35 mm porous concrete; 50 mm mineral wool; no studs$^1$</td>
<td>3 11 14 16 14 13 14</td>
<td></td>
</tr>
<tr>
<td><strong>Basic wall 100 mm Calcium-Silicate blocks, 180 kg/m$^2$:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 mm x 12.5 mm gypsum board; 20 mm foam; no studs</td>
<td>2 5 19 30 41 42 23</td>
<td></td>
</tr>
<tr>
<td><strong>Basic wall 300 mm plastered hollow blocks, 240 kg/m$^2$:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>15 mm cement plaster; 30 mm mineral wool; no studs$^1$</td>
<td>0 -4 5 9 11 15 7</td>
<td></td>
</tr>
<tr>
<td>15 mm cement plaster; 50 mm mineral wool; no studs$^1$</td>
<td>0 -5 5 6 10 14 6</td>
<td></td>
</tr>
</tbody>
</table>

---

Table 9.1 Sound reduction index improvement from extra layers. Source: NEN-EN12354-1
### APPENDIX B Material properties

Table from (Hopkins, 2007).

<table>
<thead>
<tr>
<th>Material</th>
<th>Density, $\rho$ (kg/m$^3$)</th>
<th>Quasi-longitudinal phase velocity, $c_l$ (m/s)</th>
<th>Poisson’s ratio, $\nu$</th>
<th>Internal loss factor (bending waves), $\eta_{int}$</th>
<th>Critical frequency (m.Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concrete</td>
<td>2200</td>
<td>3800</td>
<td>0.2</td>
<td>0.005</td>
<td>17.1</td>
</tr>
<tr>
<td>Glass</td>
<td>2500</td>
<td>5200</td>
<td>0.24</td>
<td>0.003-0.006</td>
<td>12.5</td>
</tr>
<tr>
<td>Plasterboard (gypsum)</td>
<td>860</td>
<td>1490</td>
<td>0.3</td>
<td>0.0141</td>
<td>43.5</td>
</tr>
<tr>
<td>Plaster gypsum based</td>
<td>650</td>
<td>1610</td>
<td>0.2</td>
<td>0.012</td>
<td>40.3</td>
</tr>
<tr>
<td>Plywood</td>
<td>710</td>
<td>3850</td>
<td>0.3</td>
<td>0.016</td>
<td>16.8</td>
</tr>
<tr>
<td>Steel</td>
<td>7800</td>
<td>5270</td>
<td>0.28</td>
<td>0.0001</td>
<td>12.3</td>
</tr>
</tbody>
</table>
# APPENDIX C Overview of the effects element properties on the sound reduction

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>effect</th>
<th>Sound reduction effect</th>
<th>Measure</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Critical frequency</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>High critical frequency (4000+ Hz, lightweight elements)</td>
<td>Low radiation factor</td>
<td>Increases sound reduction for frequencies below $f_c$</td>
<td>Increasing mass, decreasing bending stiffness and thickness</td>
</tr>
<tr>
<td>Low critical frequency (125-Hz, Heavyweight elements)</td>
<td>Placing coincidence dips below the frequency range of interest. Possible because of enough damping</td>
<td>Increases sound reduction for frequencies above coincidence region</td>
<td>Decreasing mass, increasing bending stiffness and thickness.</td>
</tr>
<tr>
<td>Varying $f_c$ for connected elements</td>
<td>Smaller but wider dip in sound reduction due overlapping coincidence effects</td>
<td>Prevents a decrease in sound reduction for the coincidence dip/plateau</td>
<td>Varying mass, decreasing bending stiffness and thickness for elements</td>
</tr>
<tr>
<td>Damping in element/system</td>
<td>Smaller dip in sound reduction due coincidence</td>
<td>Increases sound reduction for the coincidence dip/plateau</td>
<td>Increasing the( edge) loss factor by rigidly fixing panels or other means</td>
</tr>
<tr>
<td>One $f_c$ in an element</td>
<td>Prevents that an element has multiple critical frequencies. This can happen in orthotropic elements</td>
<td>Increases the width and/or depth of coincidence dip/plateau</td>
<td>Avoid the use of orthotropic plates with multiple $f_c$’s in lightly damped structures.</td>
</tr>
<tr>
<td><strong>Resonance frequency</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Low resonance frequency (10-50hz)</td>
<td>Low sound reduction below the frequency range of interest</td>
<td>Increases sound reduction for frequencies above resonance region</td>
<td>Increasing element size and $f_c$</td>
</tr>
<tr>
<td>Varying $f_c$ for connected elements</td>
<td>Prevent the free propagation of sound and modal coupling between elements</td>
<td>Prevents a decrease in sound reduction for the resonance region</td>
<td>Varying element size and $f_c$ for connected elements</td>
</tr>
<tr>
<td>Damping in element/system</td>
<td>Reduction of resonance amplitude</td>
<td>Increases sound reduction for the resonance dips</td>
<td>Increasing the( edge) loss factor by rigidly fixing panels or other means</td>
</tr>
<tr>
<td><strong>Mass</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>High mass</td>
<td>Increases acoustic impedance and the amount of reflected sound energy</td>
<td>5-6dB increase per doubling of mass in the entire frequency range</td>
<td>Increasing density or thickness</td>
</tr>
<tr>
<td>Low mass</td>
<td>Lower acoustic impedance, and smaller impedance jumps</td>
<td>Lower sound reduction in the entire frequency range</td>
<td>Decreasing density or thickness</td>
</tr>
<tr>
<td>Separation</td>
<td>Total separation</td>
<td>Doubling of sound reduction if two identical walls are used</td>
<td>Increases sound reduction for entire frequency range</td>
</tr>
<tr>
<td>-----------------------------------</td>
<td>-----------------</td>
<td>----------------------------------------------------------</td>
<td>----------------------------------------------------</td>
</tr>
<tr>
<td>Cavity</td>
<td>Increases separation, more effective for high frequencies. Each doubling of the cavity depth is less effective.</td>
<td>Increases sound reduction for frequencies between $f_0$ and $f_a$</td>
<td>Adding space between elements</td>
</tr>
<tr>
<td>Absorption in cavity</td>
<td>Reduces standing waves inside cavity by adding damping to the system, also adds a little dissipation.</td>
<td>Increases sound reduction effect over entire frequency range</td>
<td>Placing absorbing materials in cavity without connecting the separated elements or filling the entire cavity.</td>
</tr>
<tr>
<td>Rigid connections</td>
<td>Creates a sound bridge between separated elements, but adds stability</td>
<td>Decreases sound reduction above the cut-off-frequency, $f_0$</td>
<td>Making mechanically rigid connections between elements</td>
</tr>
<tr>
<td>Resilient connections</td>
<td>Reduces the sound bridge between separated elements, but still adds stability</td>
<td>Decreases the loss in sound reduction above the cut-off frequency, $f_b$</td>
<td>Connecting elements flexible so that vibrations cannot be transmitted</td>
</tr>
<tr>
<td>Short circuiting</td>
<td>Decreases sound reduction by creating a sound bridge</td>
<td>Decreases sound reduction over entire frequency range</td>
<td>Connecting elements that ought to be separated. Prevent this.</td>
</tr>
<tr>
<td>Absorption</td>
<td>Dissipates a small amount of sound energy, and has influence on the acoustic climate.</td>
<td>Most effective for higher frequencies, but also applicable for lower frequencies</td>
<td>Using Helmholtz resonators, porous materials or panel resonators</td>
</tr>
<tr>
<td>Absorbing materials in origin room</td>
<td>Decreases the sound volume in the receiving room, and has influence on the acoustic climate</td>
<td>Effective when silence is required in receiving room</td>
<td>Using Helmholtz resonators, porous materials or panel resonators</td>
</tr>
<tr>
<td>Junction</td>
<td>Creates a impedance jump which increases the reflection of sound. Elements with the same mass can pass vibrations more easily</td>
<td>Increases sound reduction over the entire frequency range.</td>
<td>Using different size elements in dominant flanking paths</td>
</tr>
<tr>
<td>Mass difference</td>
<td>Decreases the amount of vibrations that can pass the junction by creating an impedance jump</td>
<td>Increases sound reductions for frequencies above $f_1$ in heavyweight elements.</td>
<td>Adding elastic layers or springs between elements where ever possible</td>
</tr>
<tr>
<td>Extra junctions</td>
<td>Junctions have a minimum sound reduction value, adding junctions will thus increase the total sound reduction</td>
<td>Increases the effects of a single junctions. Same principles apply</td>
<td>More structural dilations and connections</td>
</tr>
<tr>
<td>---------------------------------------------</td>
<td>------------------------------------------------------------------------------------------------------------</td>
<td>-------------------------------------------------------------------</td>
<td>-------------------------------------------</td>
</tr>
<tr>
<td><strong>Air Sealing</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Air tight room</strong></td>
<td>Sound travels easily through air, but gets attenuated by impedance jumps. Preventing air paths makes sure sound gets attenuated</td>
<td>Increases the sound reduction by forcing the sound to travel through impedance jumps. Most effective if there is good sound insulation</td>
<td>Caulking and sealing correctly</td>
</tr>
</tbody>
</table>
APPENDIX D Summary of interviews with building physic consultancies

Sound insulation in case studies

To gain a better understanding of how sound insulation is dealt with when designing a building, several building physic consultancies were visited. The interviews consisted partly of open questions and partly of questions about a specific case. In this chapter the views of each firm are briefly discussed and compared.

Peutz and Termeulen post
Termeulen Post is a residential building built on top of an existing shopping center in Rotterdam. Because it is built on top of an existing building, the structure is very light. During the construction phase it was observed that walking vibrations could be felt several apartments away from the origin.

The cause of this vibration leak was modal coupling of the resonance frequencies of the floors. Because the floors were lightweight, the vibration resonance could be excited more easily. As solution the floors have been stiffened in alternating direction, to prevent the excitation and the modal coupling.

In contrast, the sound insulation in the building was very good. Inside the steel skeleton, each apartment was executed as a box to create a box-in-box system. Between the box and the skeleton elastic layers were used to increase the separation.

Peutz agreed that the building regulations are a bit short on low frequency sounds, and how to reduce them. But it does cover a lot of the aspects and acts as a fine guidance. However, they almost always advice to add extra sound insulation for low frequency sounds.

Instead of increasing sound reduction values, it is often better to put distance between sound sources and quiet spaces. Placing corridors or utility rooms in the path of the dominant sound path reduces the sound reduction requirement.

When increased sound reduction is required, separation is suggested. A cavity increases the sound reduction a lot already, and further the flanking routes can be dilated. However, it is always a consideration between mass, space and money.

In lightweight buildings the building errors are even more important. It is thus very important that buildings are made precise as they are designed.

Scena-adviseurs and Theater Spijkenisse
The theater in Spijkenisse has a large theater hall and a smaller pop and house music hall. Besides that, several offices are made inside the building in the vicinity of the pop hall. This created the highest sound reduction requirement for the pop hall. To obtain that, a box-in-box construction was made with a lightweight inner box. For the larger hall a concrete was chosen as building material for the flexibility of the room.

Throughout the building several other lightweight elements have been used, such as the facade and in the foyer. However, it created a lot of extra complications, which lead to the conclusion that it may have been easier (and cheaper) to do it in concrete.

To increase the sound insulation in (lightweight) structures there exist several options according to Scena-adviseurs. With a cavity being the best option, which can increase the sound reduction
dramatically. Otherwise a larger buffer zone between room or more mass should provide the sound reduction needed.

For low frequency sounds stiffness becomes more important than mass. It is often neglected, because it has the largest requirements for space, thickness or mass. Complete separation is used in most cases to deal with low frequency sounds. Adjusting the springs resonance frequency is important as well.

Flanking however, is not seen as a large problem in heavy weight constructions. NEN-EN 12354 states that for the flanking reduction the average over the direct paths can be taken, and that it will always be less than direct. For lightweight constructions a small reduction has to be factored over the junction loss. Flanking can become a problem in older buildings, where the boundary conditions are set. Or during the building phase, when mistakes are made.

**DHV and TivoliVredenburg Concert building**

TivoliVredenburg is a concert building in Utrecht where pop, house and classic music halls can be found under one roof. After the opening a sound leak was found between two halls. After investigation it was found that an steel column was the cause, and the sound leak was fixed by separating the column with an elastic layer.

However the design of the building is fairly light, and makes use of separation and buffer zones. By spacing out the different halls, very large sound reduction values have been achieved. Where this was not possible, extra mass has been added.

In pop and house halls the sound spectrum is dominantly low frequency, even more so that in movie theaters. Low frequencies are always a problem because mass law provides little sound reduction there. Important is to design the first resonance frequency 4 times below the lowest considered octave band. This is to avoid the dips in sound reduction caused by resonance. Care should be taken that resonance frequencies do not overlap, and that structural elements require a resonance frequency above 4 Hz.

However, lightweight constructions can have high sound reduction values. In order for this to work, separation is key though. Likewise, if a higher sound reduction is required, a box-in-box system is proposed. Usually the outer box is executed heavy, and the inner box light. However, light elements can be exited in their resonance frequencies by amplified vibrations coming from the heavier box.

Another option is to complete dilate building parts from each other. This extreme form of separation has even higher sound reduction values. Note that the place of the room inside the building is critical for this to work. The same goes for the place of the springs, which should be founded on stiff or heavy elements.

Separation is thus the keyword for increasing sound insulation. However, mistakes in the build phase often damage the separation resulting in sound leaks.

**Pathe spui**

The movie theater Pathe Spuimarkt is made with a lightweight steel skeleton, and lightweight isover walls. Several other Pathe movie theaters are made in similar fashion. However, for each new building the advice of a building physics firm is requested. When designing a new movie theater, not much consideration is put into sound insulation. This has to be solved later, when critical design choices are already made.
While Pathé concerns itself more with the acoustic climate inside a movie hall, it is required to have at least 60 dB sound reduction between halls. Which can be obtained with a double leaf construction of triple gypsum board and some absorbing materials. Most important here is that the leaves are separated and the connections are resilient. Often mistakes are made in the building phase, which cause sound leaks.

The sound of movies is averaged on 85 dB(A), with 30 dB head room. This means that the max peak sound volume can be 115 dB(A). However, this is almost never reached. That the sound reduction is lacking in some areas is very clear in the coffee room, where you can follow the movie playing beside it quite well.

**DGMR**

According to DGMR flanking sound transmission becomes important for buildings with high sound reduction requirements. For normal situations, the sound reduction from the direct path is sufficient. When flanking does become a problem, this is usually the case for heavyweight structures and not so much for lightweight structures. This can be explained by the fact that lightweight structures are already based on separation, which improves flanking sound transmission as well.

Lightweight walls are however used in movie theaters, and reach a sound reduction of about 55 dB(A) using the movie theater spectrum. A common product that is used is called technostar walls from isover, which consists of a double leaf wall with gypsum boards and resilient connections.

However, they note that the movie spectrum is outdated and should consider louder low frequency sounds. Also, 55 dB(A) might not be enough for modern movies with an average sound of 85 dB(A).

When higher sound reduction values are needed that cannot be obtained by increasing the direct sound reduction, a heavier system is considered. For even higher sound reduction requirements a box-in-box system is considered. DGMR questions the effectiveness of the springs that separate the boxes, and suggests two concrete blocks might do the same.

To increase sound reduction of direct paths separation is used. Calculating the effects is difficult, but great improvements have been observed with it. The execution can however, mess things up and nullify the added sound reduction by separation. Building secure is viewed as very important.

For lower frequencies stiffness of elements and the resonance frequency becomes important. Where stiffness is influenced by boundary conditions as well. However, lighter elements have less internal damping which can be bad for the resonance region.
APPENDIX E Detailing case movie theater Dordrecht

A floor detail from the movie theater in Dordrecht is evaluated based on general knowledge, and based on the results of the finite element models. The figures on the next pages are definitive structural and architectural drawings of a slanting floor in the movie theater in Dordrecht.

The movie theater itself is made with a steel structure, floors are made from hollow core slabs and walls consist of gypsum plates. In the last figure, it can be seen that the great care has been taken to separate walls, floors and rooms in the movie theater. This decreases both the flanking sound transmission and the direct sound transmission. However, movie rooms are still placed above each other, and so the sound reduction requirements of the floors are still high. By separating the building elements and limiting the flanking sound transmission, a great deal of (acoustic) flexibility is added.

The architectural drawing on page 153 shows a floor detail where a window frame is connected to the hollow core slab via a metal stud. A lowered ceiling is connected resiliently to the floor and the multiplex mounting frame. The second lowered ceiling is hanged as well, and has a simple connection with the mounting frame. Between the floor and the supporting beam, CDM rubbers are used. The connection between the tribune and the floor is not specified, but is assumed to be simple.

Since the floor is relatively light weight, the direct sound reduction of the floor will not be high enough for a movie theater. Both the lowered ceiling(s) and the tribune help to increase the sound reduction by adding impedance jumps. Therefore it is important that they are acoustically separated. The connection between the tribune and the floor could be made resilient, and the cavity of the lowered ceiling could be larger to increase this separation. However, care should be taken to keep the resonance frequency of the mass-spring-mass system low.

Flanking sound will travel through the tribune and floor into the metal stud. Acoustically disconnecting the metal stud from the floor would greatly reduce this flanking path. Also, the sealing between the multiplex mounting frame and the floor could become a flanking path, keeping this sealing resilient is important. The connection between the second lowered ceiling and the multiplex is very simple, but should be sealed correctly, and would also benefit from a resilient connection.

Based on the finite element models, the floor can have significant influence on the flanking sound transmission. However, this is not something that is taken into account in this design. Because the floor itself can be considered as an orthotropic material, the flanking sound transmission is focused into one direction. Following from the honeycomb variant, the direction that is stiffer has a lower flanking sound transmission and vice versa. Looking at the whole design, the dominant flanking path is probably in the x-direction, and the floors should have spanned the that direction.

Alternatively, adding elastic material in the seams between the hollow core slabs can acoustically disconnect the slabs, and lead to a greatly reduced flanking sound transmission in the x-direction. The size of the gap depends on the properties of the elastic material, and should not affect the direct sound transmission as long as it has some mass. Because of the tribune, any deformation differences between the floor slabs should not be a large problem. In the structural drawings, acoustic material is already added to the dowel (afschuifnok), however it should be entirely covered.

In theory, the detailing of the floor with the proposed changes should provide excellent sound insulation. However, the detailing is quite complex, and building errors are easily made. It is therefore important that details are properly constructed and sealed. Care should be taken with the sealing, so that no sound bridges between separated elements are created.
Figure 9.1 Structural detailing of a slanting floor in the movie theater Dordrecht. Source: Broersma bv.
Figure 9.2 Architectural detailing of a slanting floor in the movie theater Dordrecht. Source: Snelders Architecten.
Figure 9.3 Architectural cross section of movie theater Dordrecht. Source: Snelders Architecten.
APPENDIX F Derivation of thin wall sound transmission

Figure 9.1 Schematized sound transmission through a thin wall

In order to obtain the sound insulation value, the air pressure ratio between the origin and receiving room is needed. Following the derivation of (Hassan, 2009), the expression for the sound pressure:

\[ P_1(x, t) = \hat{p}_1 e^{i\omega t} e^{-ik_1x} + \hat{p}_1 e^{i\omega t} e^{+ik_1x} \]  \hfill (9.1)

\[ P_2(x, t) = \hat{p}_2 e^{i\omega t} e^{-ik_2x} \]  \hfill (9.2)

And the particle velocity:

\[ v_1(x, t) = \frac{1}{Z_1} (\hat{p}_1 e^{i\omega t} e^{-ik_1x} - \hat{p}_1 e^{i\omega t} e^{+ik_1x}) \]  \hfill (9.3)

\[ v_2(x, t) = \frac{1}{Z_2} (\hat{p}_2 e^{i\omega t} e^{-ik_2x}) \]  \hfill (9.4)

Where:

- \( Z_1 = Z_3 \) = Acoustic impedance of air \ [Ns/m^3]\n- \( k_1 = k_2 \) = Wavenumber \ [1/m]\n- \( i = \sqrt{-1} \) \ [-]\n- \( \omega \) = Angular frequency \ [1/s]\n- \( t \) = Time \ [s]\n- \( \hat{p} \) = Amplitude of pressure wave \ [N/m^2]\n
At the boundary of the plate the particle velocities are equal to the velocity of the plate. So the system can be described as a mass-spring system. The motion of the entire system can then be rewritten to:

\[ p_{1+} - p_{1-} = p_{2+} \]  \hfill (9.5)

\[ V(t) = v_1(0, t) = v_2(0, t) = \frac{p_{2+} e^{i\omega t}}{\rho_0 c} \]  \hfill (9.6)
\[ F(t) = m \frac{dV}{dt} + \tau_m V(t) + k_s \int V(t) dt \]  

(9.7)

Assuming the applied load is sinusoidal, will result in a sinusoidal movement.

\[ F(t) = F_0 e^{i\omega t} \]  

(9.8)

\[ F(t) = i\omega m V(t) + \tau_m V(t) + \frac{k_s V(t)}{i\omega} \]  

(9.9)

Because the mass and damping have little effect in this frequency range compared to the stiffness, the equation reduces to:

\[ F(t) = \frac{k_s V(t)}{i\omega} \]  

(9.10)

Taking the total force balance over the plate, and substituting Eq. (9.5) and (9.6):

\[ p_1(0,t) - p_2(0,t) = -F = \frac{k_s p_{2+} e^{i\omega t}}{i\omega \rho_0 c} \]  

(9.11)

\[ p_{1+} + p_{1-} - p_{2+} = -F = \frac{ik_s p_{2+}}{\omega \rho_0 c} \]  

(9.12)

Combining Eq. (9.5) and (9.6) with Eq. (9.12), results in the sound power transmission coefficient by looking at the difference in pressure before and after the plate.

\[ \frac{p_{2+}}{p_{1+}} = \frac{1 - \left(\frac{ik_s}{2\omega \rho_0 c}\right)}{1 + \left(\frac{k_s}{2\omega \rho_0 c}\right)^2} \]  

(9.13)

\[ \tau_n = \left(\frac{p_{2+}}{p_{1+}}\right)^2 \]  

(9.14)

\[ \tau_n = \frac{1}{1 + \left(\frac{k_s}{2\omega \rho_0 c}\right)^2} \]  

(9.15)
APPENDIX G Derivation of thick wall sound transmission

Sound transmission in region 2 can be described by using a thick wall. From the following derivation it will become clear that the jump in acoustical impedance, $Z$, is determining for the sound reduction.

If a normal wave is taken with an angle of zero, the behaviour of sound at a boundary can be illustrated by Figure 3.4. Following the derivation of (Hassan, 2009) the sound pressure in each of the materials can be described by:

\[ P_1(x,t) = \hat{p}_1 e^{i\omega t} e^{-ik_1x} + \hat{p}_1 e^{i\omega t} e^{+ik_1x} \] \hspace{1cm} (9.1)

\[ P_2(x,t) = \hat{p}_2 e^{i\omega t} e^{-ik_2x} + \hat{p}_2 e^{i\omega t} e^{+ik_2x} \] \hspace{1cm} (9.2)

\[ P_3(x,t) = \hat{p}_3 e^{i\omega t} e^{-ik_3(x-d)} \] \hspace{1cm} (9.3)

And the particle velocity:

\[ v_1(x,t) = \frac{1}{Z_1} (\hat{p}_1 e^{i\omega t} e^{-ik_1x} - \hat{p}_1 e^{i\omega t} e^{+ik_1x}) \] \hspace{1cm} (9.4)

\[ v_2(x,t) = \frac{1}{Z_2} (\hat{p}_2 e^{i\omega t} e^{-ik_2x} - \hat{p}_2 e^{i\omega t} e^{+ik_2x}) \] \hspace{1cm} (9.5)

\[ v_3(x,t) = \frac{1}{Z_3} (\hat{p}_3 e^{i\omega t} e^{-ik_3(x-d)}) \] \hspace{1cm} (9.6)

With:

- $Z$ = Acoustic impedance \([\text{Ns/m}^2]\)
- $k$ = Wavenumber \([1/\text{m}]\)
- $d$ = Thickness of wall \([\text{m}]\)
- $\omega$ = Angular frequency \([1/\text{s}]\)
- $i = \sqrt{-1}$ \([-]\)
- $t$ = Time \([\text{s}]\)
- $\hat{p}$ = Amplitude of pressure wave \([\text{N/m}^2]\)

Using the matching conditions at the boundaries so that pressure and particle velocity are equal in both materials, the following relation between the pressure in room 1 and room 3 can be found.
\[
\hat{p}_{1+} + \hat{p}_{1-} = \hat{p}_{2+} + \hat{p}_{2-} \tag{9.7}
\]
\[
\frac{1}{Z_1}(\hat{p}_{1+} - \hat{p}_{1-}) = \frac{1}{Z_2}(\hat{p}_{2+} + \hat{p}_{2-}) \tag{9.8}
\]
\[
\hat{p}_{2+}e^{-ik_2d} + \hat{p}_{2-}e^{-ik_2d} = \hat{p}_{3+} \tag{9.9}
\]
\[
\frac{1}{Z_2}(\hat{p}_{2+}e^{-ik_2d} - \hat{p}_{2-}e^{-ik_2d}) = \frac{\hat{p}_{3+}}{Z_3} \tag{9.10}
\]

Combine above equations leads to:
\[
\frac{\hat{p}_{1+}}{\hat{p}_{3+}} = \frac{1}{4} \left( 1 + \frac{Z_1}{Z_2} \right) \left( 1 + \frac{Z_2}{Z_3} \right) e^{ik_2d} + \frac{1}{4} \left( 1 - \frac{Z_1}{Z_2} \right) \left( 1 - \frac{Z_2}{Z_3} \right) e^{-ik_2d} \tag{9.11}
\]

This can be rewriting with Eq.(9.12) to Eq.(9.13)
\[
e^{ix} = \cos(x) + i\sin(x) \tag{9.12}
\]
\[
\frac{\hat{p}_{1+}}{\hat{p}_{3+}} = \frac{1}{2} \left( 1 + \frac{Z_1}{Z_3} \right) \cos(k_2d) + i \frac{1}{2} \left( \frac{Z_1}{Z_2} + \frac{Z_2}{Z_3} \right) \sin(k_2d) \tag{9.13}
\]

Which is used to find the sound power transmission coefficient, \(\tau_n\):
\[
\tau_n = \frac{\frac{Z_1}{Z_3} \left( \frac{\hat{p}_{3+}}{\hat{p}_{1+}} \right)^2}{4} \tag{9.14}
\]
\[
\tau_n = \frac{4 \left( \frac{Z_1}{Z_3} \right)}{\left( 1 + \frac{Z_1}{Z_3} \right)^2 \cos^2(k_2d) + \left( \frac{Z_1}{Z_2} + \frac{Z_2}{Z_1} \right)^2 \sin^2(k_2d)} \tag{9.15}
\]

When a case is considered where both \(Z_1\) and \(Z_3\) are air, and thus the same, the equation can be reduced to:
\[
\tau_n = \frac{4}{\left[ 4 \cos^2(k_2d) + \left( \frac{Z_1}{Z_2} + \frac{Z_2}{Z_1} \right)^2 \sin^2(k_2d) \right]} \tag{9.16}
\]

Where:
\(Z_1=Z_3\) = acoustic impedance of air \([\text{Ns/m}^3]\)
\(Z_2\) = acoustic impedance of the wall \([\text{Ns/m}^3]\)
\(k_2\) = wavenumber of the wall \([1/\text{m}]\)
\(d\) = thickness of the wall \([\text{m}]\)
This can be simplified to the mass law when the term $k_2d$ is smaller than 0.3 radians. For low to medium frequencies and normal walls, this is true. Also $Z_2$, the acoustic impedance of the wall, is much higher than $Z_1$. Using these approximations the mass law is obtained:

$$\tau_n = 1 + \left( \frac{Z_2}{2Z_1} \right)^2 (k_2d)^2 = 1 + \left( \frac{\pi f \rho_2 d}{\rho_1 c_1} \right)^2 \quad (9.17)$$

With:

$\rho_1$ = Density of air [kg/m$^3$]
$c$ = Speed of sound in air [m/s]
$\rho_2d$ = Mass of the wall (M) [kg]

The sound power reduction coefficient, $\tau_n$, can be used to calculate the sound for frequencies with wavelengths that are comparable to the wall thickness. Also known as region 2, where the sound reduction is governed by mass.

$$R = 10\log \frac{1}{\tau} \quad (9.18)$$
APPENDIX H ANSYS Results

List of results for the honeycomb variant.

An overview is given of the calculations done for the honeycomb variant. The average velocity level difference is the green line in the graph, the energetic velocity level difference represent the average of the red line in the graph. When the two values are further apart, it usually indicates heavier modal behaviour.

Significant improvements in the sound reduction are indicated by a green block, significant decreases are indicated by a red block in the table. The results for the honeycomb variant with the reference values are given in the graph. The floor drawings are given again for readability.

For the different calculations only the parameters that were varied are mentioned, the rest of the model setup remains the same.
For comparison, the result for the base case:

<table>
<thead>
<tr>
<th>Velocity level difference avg</th>
<th>Velocity level difference energetic</th>
<th>floors</th>
<th>walls</th>
<th>Beam</th>
<th>Note</th>
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</thead>
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<td>Concrete</td>
<td>Steel</td>
<td>Base case</td>
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<td></td>
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<td>$E=15\times 9$</td>
<td>$E=2\times 11$</td>
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<td></td>
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<td>$p=2200\text{kg/m}^3$</td>
<td>$p=2200\text{kg/m}^3$</td>
<td>IPE 750</td>
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</tbody>
</table>

And the results for the honeycomb variant:

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<th>Velocity level difference energetic</th>
<th>Grid</th>
<th>Backplate</th>
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<th>Note</th>
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<td>50</td>
<td>46</td>
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<td>Steel E/500</td>
<td>Energetic is very low at low freq</td>
</tr>
<tr>
<td>42</td>
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<td>concrete E*5</td>
<td>Steel</td>
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<td>Steel</td>
<td>Large deviations energetic</td>
</tr>
<tr>
<td>30</td>
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<td>concrete E/5</td>
<td>Steel</td>
<td>Modal behaviour</td>
</tr>
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<td>Concrete p*5</td>
<td>Steel</td>
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<td>40</td>
<td>34</td>
<td>concrete</td>
<td>Concrete p/5</td>
<td>Steel</td>
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<td>32</td>
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<td>Concrete p/50 E*50</td>
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<td>Steel</td>
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<td>concrete E/5</td>
<td>steel</td>
<td>Steel</td>
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</tr>
<tr>
<td>33</td>
<td>30</td>
<td>concrete E*5</td>
<td>concrete E*5</td>
<td>Steel</td>
<td>Large deviations energetic</td>
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<td>29</td>
<td>concrete E*5</td>
<td>concrete E*5</td>
<td>Steel E*5</td>
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<td></td>
<td></td>
<td>Concrete E/5</td>
<td>Concrete E/5</td>
<td>Steel Walls</td>
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<td>--------------</td>
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<tr>
<td>26</td>
<td>22</td>
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<td>Concrete E/5</td>
<td>Steel E/5</td>
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<td>23</td>
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<td>Concrete E/5</td>
<td>Steel E/5</td>
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<td>27</td>
<td>24</td>
<td>Concrete E/5</td>
<td>Concrete E/5</td>
<td>Steel E/5</td>
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</tr>
<tr>
<td>29</td>
<td>27</td>
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<td>Concrete E/5</td>
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<table>
<thead>
<tr>
<th></th>
<th></th>
<th>Concrete</th>
<th>Stiffeners x=10</th>
</tr>
</thead>
<tbody>
<tr>
<td>24</td>
<td>19</td>
<td>Concrete</td>
<td>Stiffeners y=1</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
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<th>Concrete</th>
<th>Stiffeners x=1</th>
<th>Stiffeners y=25</th>
</tr>
</thead>
<tbody>
<tr>
<td>39</td>
<td>34</td>
<td>Concrete</td>
<td>Stiffeners y=25</td>
<td></td>
</tr>
<tr>
<td>27</td>
<td>21</td>
<td>Concrete</td>
<td>Stiffeners x=1</td>
<td></td>
</tr>
<tr>
<td>38</td>
<td>33</td>
<td>Concrete</td>
<td>Stiffeners x=1</td>
<td></td>
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</tbody>
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<table>
<thead>
<tr>
<th></th>
<th></th>
<th>Concrete</th>
<th>Stiffeners x=1</th>
<th>Stiffeners y=25</th>
</tr>
</thead>
<tbody>
<tr>
<td>38</td>
<td>36</td>
<td>Concrete</td>
<td>Thickness 0,01</td>
<td></td>
</tr>
<tr>
<td>44</td>
<td>42</td>
<td>Concrete</td>
<td>Thickness 0,01</td>
<td></td>
</tr>
</tbody>
</table>

Best of all experiments
Damping=0,05
Strong decline
List of results for the wave variant.

An overview is given of the calculations done for the wave variant. The average velocity level difference is the green line in the graph, the energetic velocity level difference represent the average of the red line in the graph. When the two values are further apart, it usually indicates heavier modal behaviour.

Significant improvements in the sound reduction are indicated by a green block, significant decreases are indicated by a red block in the table. The results for the honeycomb variant with the reference values are given in the graph. The floor drawings are given again for readability.

For the different calculations only the parameters that were varied are mentioned, the rest of the model setup remains the same.
For comparison, the result for the base case:

<table>
<thead>
<tr>
<th>Velocity level difference avg</th>
<th>Velocity level difference energetic</th>
<th>floors</th>
<th>walls</th>
<th>Beam</th>
<th>Note</th>
</tr>
</thead>
</table>
| 30                            | 29                                  | Concrete $E=15^9 \cdot 9$  
$p=2200$ kg/m$^3$ | Concrete $E=15^9 \cdot 9$  
$p=2200$ kg/m$^3$ | Steel $E=2^11$ IPE 750 | Base case |

And the results for the wave variant:

<table>
<thead>
<tr>
<th>Velocity level difference avg</th>
<th>Velocity level difference energetic</th>
<th>Stiffeners</th>
<th>Wave shape of floor</th>
<th>Beam</th>
<th>Note</th>
</tr>
</thead>
</table>
| 34                            | 31                                  | Concrete $E=15^9$  
$0,05 \cdot 10 \cdot 0,2$  
Stiffeners $x=6$  
Endplate at support | Concrete $E=15^9$  
$d1=d2=0,06$ m  
h1=0,2  
a=0,02$ m  
#waves=4 | Steel $E=2^11$ IPE 750 | Reference case  
Damping= 0,01  
Symmetric  
junction  
Ends at wave  
average |
<p>| 27                            | 26                                  | Concrete   | Concrete            | Steel | Measurement placement on bottom plate |
| 27                            | 21                                  | Concrete   | Concrete            | Steel | Bottom plate removed, strong increase over the frequency |
| 33                            | 31                                  | Concrete   | Concrete            | Steel | |
| 35                            | 32                                  | Concrete   | Concrete            | Steel | No dips below 20 dB |
| 32                            | 29                                  | Concrete   | Concrete            | Steel | Smaller modal behaviour |
| 35                            | 31                                  | Concrete   | Concrete            | Steel | |
| 34                            | 30                                  | Concrete   | Concrete            | Steel | Modal behaviour at 70Hz and 82Hz |
| 33                            | 30                                  | Concrete   | Concrete            | Steel | |
| 32                            | 30                                  | Concrete   | Concrete $E/5$      | Steel | |
| 37                            | 33                                  | Concrete   | Concrete $E<em>5$      | Steel | |
| 41                            | 36                                  | Concrete   | Concrete $p/5$      | Steel | Wide dips and peaks |
| 31                            | 29                                  | Concrete   | Concrete $p</em>5$      | Steel | |</p>
<table>
<thead>
<tr>
<th>No.</th>
<th>No.</th>
<th>Material</th>
<th>Material</th>
<th>Material</th>
<th>Material</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>35</td>
<td>32</td>
<td>Concrete E/5</td>
<td>Concrete</td>
<td>Steel</td>
<td>Dip below zero at 69 Hz</td>
<td></td>
</tr>
<tr>
<td>34</td>
<td>31</td>
<td>Concrete E/5</td>
<td>Concrete</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>28</td>
<td>26</td>
<td>Concrete E/5 (endplate) Stiffeners removed</td>
<td>Concrete</td>
<td>Steel</td>
<td>No coupling between top and bottom plate, except at the supports</td>
<td></td>
</tr>
<tr>
<td>32</td>
<td>30</td>
<td>Concrete</td>
<td>Concrete</td>
<td>Steel</td>
<td>Symmetric junction Ends at wave maximum</td>
<td></td>
</tr>
<tr>
<td>36</td>
<td>32</td>
<td>Concrete</td>
<td>Concrete</td>
<td>Steel</td>
<td>Symmetric junction Ends at wave minimum</td>
<td></td>
</tr>
<tr>
<td>36</td>
<td>32</td>
<td>Concrete</td>
<td>Concrete</td>
<td>Steel</td>
<td>Asymmetric junction Ends at wave minimum at source floor</td>
<td></td>
</tr>
<tr>
<td>32</td>
<td>30</td>
<td>Concrete</td>
<td>Concrete</td>
<td>Steel</td>
<td>Asymmetric junction Ends at wave maximum at source floor</td>
<td></td>
</tr>
<tr>
<td>35</td>
<td>32</td>
<td>Concrete</td>
<td>Concrete</td>
<td>Steel</td>
<td>Control as normal double leaf wall</td>
<td></td>
</tr>
<tr>
<td>44</td>
<td>42</td>
<td>Concrete E/5</td>
<td>Concrete p/5 d1=d2=0,04m a=0,01m #waves=2</td>
<td>Steel</td>
<td>Combination Walls are concrete</td>
<td></td>
</tr>
<tr>
<td>43</td>
<td>39</td>
<td>Concrete E/5</td>
<td>Concrete p/5 d1=0,08m d2=0,04m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>40</td>
<td>Concrete E/5</td>
<td>Concrete p/5 d1=0,04m d2=0,08m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>41</td>
<td>38</td>
<td>Concrete E/5</td>
<td>Concrete p/5 d1=0,04m d2=0,12m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>37</td>
<td>33</td>
<td>Concrete E/5</td>
<td>Concrete p/5 d1=0,12m d2=0,04m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Concrete</td>
<td>Concrete</td>
<td>Concrete p/5</td>
<td>Steel</td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td></td>
</tr>
<tr>
<td>38</td>
<td>34</td>
<td>Concrete E/5</td>
<td>d1=0.06m d2=0.06m h1=0.4m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>36</td>
<td>33</td>
<td>Concrete</td>
<td>d1=0.08m d2=0.04m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>28</td>
<td>Concrete</td>
<td>d1=0.04m d2=0.08m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>38</td>
<td>Concrete</td>
<td>d1=0.12m d2=0.04m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>28</td>
<td>Concrete</td>
<td>d1=0.04m d2=0.12m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>38</td>
<td>34</td>
<td>Concrete</td>
<td>h1=0.4m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>34</td>
<td>31</td>
<td>Concrete</td>
<td>h1=0.15m</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>26</td>
<td>24</td>
<td>Concrete Stiffeners x=4</td>
<td>Concrete</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>29</td>
<td>27</td>
<td>Concrete Stiffeners x=11</td>
<td>Concrete</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>33</td>
<td>28</td>
<td>Concrete Stiffeners x=0</td>
<td>Concrete Wave in y-direction #wave number=4</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
<tr>
<td>29</td>
<td>27</td>
<td>Concrete Stiffeners x=0</td>
<td>Concrete Wave in y-direction #wave number=10</td>
<td>Steel</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Deformations were confined in 1 lane.
List of results for the hinged variant.

An overview is given of the calculations done for the hinged variant. The average velocity level difference is the green line in the graph, the energetic velocity level difference represent the average of the red line in the graph. When the two values are further apart, it usually indicates heavier modal behaviour.

Significant improvements in the sound reduction are indicated by a green block, significant decreases are indicated by a red block in the table. The results for the honeycomb variant with the reference values are given in the graph. The floor drawings are given again for readability.

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For comparison, the result for the base case:

<table>
<thead>
<tr>
<th>Velocity level difference avg</th>
<th>Velocity level difference energetic</th>
<th>floors</th>
<th>walls</th>
<th>Beam</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>29</td>
<td>Concrete $E=15^5\text{9}$ $p=2200\text{kg/m}^3$</td>
<td>Concrete $E=15^5\text{9}$ $p=2200\text{kg/m}^3$</td>
<td>Steel $E=2^5\text{11}$ IPE 750</td>
<td>Base case</td>
</tr>
</tbody>
</table>

And the results for the hinged variant:

<table>
<thead>
<tr>
<th>Velocity level difference avg</th>
<th>Velocity level difference energetic</th>
<th>Plastic hinges</th>
<th>Floor</th>
<th>Beam</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>29</td>
<td>Concrete $w=0,01\text{m}$ $\text{Hinges }y=3$ Orthotropic material $Ex=15^5\text{7}$ $Ey=15^5\text{9}$ $Ez=15^5\text{9}$</td>
<td>Concrete $E=15^5\text{9}$ $P=2200\text{ kg/m}^3$</td>
<td>Steel $E=2^5\text{11}$ IPE 750</td>
<td>Base Damping= 0,01 Larger peaks than basic case</td>
</tr>
</tbody>
</table>

|                                |                                      |                |                |                | Control was exactly the same solution as basic case |
|                                |                                      |                |                |                | Control was exactly the same solution as basic case |

|                                |                                      |                |                |                | No noticeable difference with above calculation |

|                                |                                      |                |                |                | Deformation does not look like hinged connections |

|                                |                                      |                |                |                | No noticeable difference with first three calculations |

<p>| | | | | | |
|                                |                                      |                |                |                |                                          |</p>
<table>
<thead>
<tr>
<th>Concrete</th>
<th>Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concrete</td>
<td>Steel</td>
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<tr>
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<td>Concrete</td>
<td>Steel</td>
</tr>
<tr>
<td>Concrete</td>
<td>Steel</td>
</tr>
</tbody>
</table>

This is not a typo

Large peak (90dB) at 82 Hz

Plastic hinges have different young's modulus in each direction
APPENDIX I Drawings

In order:
1. Structural schematization
2. Junction schematization
3. Honeycomb model
4. Wave model
5. Hinged model
6. Floor examples hinged model
Ansys Model

Realistic schematic

Schematics junction
Scale: 1:20
Examples of floor systems

Examples of wall systems

Schematics Hinged variant
Scale 1:100