Structure-borne sound transmission in wall-floor timber junctions with damping elastomers
Abstract

The wooden multi-storey building industry is facing persistent sound transmission problems at low frequencies. Inserting elastomers within wall-floor junctions is common usage nevertheless no accurate studies have elucidated the real behavior of those nonlinear combined materials yet. Deeper knowledge is needed to create a relevant FE model which will help industry to use those materials efficiently.

The nonlinear dynamic behavior of the elastomers inserted in wooden junctions is analyzed while the static load acting on them is varying. The specific situation where those elastomers were tested is a scaled room made of two walls and one floor.

An experimental study was conducted on this prototype wooden construction and a numerical analysis was performed on the Finite Element model of it. The frequency response functions of several positions were measured on the physical setup.

The study showed that loaded structures (up to 2 times the load of the floor) had a lower damping ratio. Having the structure standing on really stiff or elastic material does not differ when comparing experimental and analytical modal parameters.

Those results depict the behavior of elastomers for different load cases and are definitely a step forward for the conception of a reliable FE model.

Key words: Elastomers, Sound Flanking Transmission, Vibrations Flanking Transmission, Wall-floor junctions, Structural Dynamics
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1. Introduction

Wooden framed buildings are increasing in popularity in Sweden. The reason for this can mainly be explained by two phenomena; changes in building regulations in 1994 and an increasing command for environmentally sustainable buildings (Negreira, 2013). Wood is the only structural material that is considered to be a renewable and carbon dioxide neutral material, viewed from a longer time perspective (Jarnerö, et al., 2015).

Two of many aspects to consider when designing wooden buildings are vibrations and structure-borne sound. These aspects can have negative effects on the residents living or working in the buildings (Simmons, et al., 2011). Problems with disturbing sound and vibrations can reduce comfort and efficiency (Hassan, 2009).

Nowadays there is a demand for buildings with flexible design that easily can be changed to different residents. One of Boverket (the Swedish authority for urban planning, construction and housing) visions for sustainable construction is that all new buildings in the future should be built with components that can be moved or removed. This is so that the buildings easily can be adapted to changes in housing need, household sizes and rapid changes in businesses (Boverket, 2012). This demand results in structures with fewer load carrying internal walls and larger floor spans. With increasing floor span the natural frequencies and damping of the floor decreases (Hassan, 2009) which has a negative effect on the floor vibrations.

1.1 Background

Vibrations in timber floors is a subject that has been studied greatly in recent years, see for example (Bolmsvik, 2013), (Jarnerö, et al., 2015) and (Negreira, 2013).

Emms, et al. (2006) points out that the largest problems with light-weight timber floor systems is impact sound transmission in the low frequency span, below 100 to 200 Hz. Impact sound is defined by Hassan (2009) as sound that occurs when two objects clashes. Sound from footsteps, dropped objects or people jumping that travels through the floor structure are examples of impact sound transmission. Structure-borne sound is another denotation for sound that travels through a structure. The latter term is a much wider denotation that covers the whole load carrying structure and impact sound is more related to sound transmission through floors.

There are suitable design methods that can be used to avoid higher frequency impact sound transmission through timber floors. In a study done by Emms, et al. (2006) building technics using either floating floor or
suspended ceiling was found to have a good mid to high frequency (100 to 3150 Hz) floor impact insulation. However, good floor impact insulation for lower frequency (below 60 Hz) was found harder to achieve. Light-weight timber floors have a lower natural frequency than steel or concrete floors making it more sensitive for low frequency excitations (Negreira, 2013). According to Emms, et al. (2006) there are not any standard impact insulation measurement techniques for lower frequencies, making this a complex subject to study.

Junctions have a large effect on the overall dynamic behavior of a timber structure (Bolmsvik, et al., 2012). In the study performed by Emms, et al. (2006) it was found that increasing damping in wall-floor joints has an influence on the resonance peaks. One way to increase the damping in a joist is to place elastomers between the load carrying walls and the floor structure (Bolmsvik, 2013; Jarnerö, et al., 2015). Elastomers are a damping material that is usually made of mixed cell polyurethane (PUR) and used to insulate flanking transmission. Flanking transmission is the part of the impact sound that goes through the supporting walls (Hassan, 2009).

1.2 Aim and Purpose

The aim of this study is to extract the eigenmodes and damping ratio for a wall-floor junction with damping elastomers, for different frequencies and pre-loads. The aim is also to develop a FE-modal to analyze how elastomers effect sound flanking transmission, at low frequencies, in wall-floor timber junctions.

The purpose of this study is to find an efficient and validated FE-model that, after some additional work, can be used by the industry when designing wall-floor junctions with elastomers in timber buildings.

1.3 Limitations

Sound and vibrations may arise from many different sources. This study is limited to sound from internal sources, caused by vibration of floor structures. In other words, this thesis focuses on structure-borne sound, caused by low frequency mechanically-excitation of the floor.

Further, different approaches can be taken to reduce floor vibrations and other variables (e.g. pre-load, workmanship, geometry) might have an effect on the dynamic properties. In this thesis, an elastomer with a given stiffness is used as damping material; the pre-load on the floor is used as a variable during the experiment.
Elastomer comes in various forms and can be mounted on top of the walls in different ways, e.g. continuous, intermitted and half-embedded. Within this thesis only continuous elastomer strips are studied.

1.4 Reliability, validity and objectivity

The experimental modal analysis procedures that are used in this thesis has been validated and used for several years by the structural engineering laboratory of Linnaeus University and by researchers within the field. It is therefore believed to be reliable.

The method used to find the material properties of the single beams have been used in other research studies at Linnaeus University and will be tested also within this field. The material properties for the longitudinal direction of the beams will then be used to calibrate the radial and tangential directions. It is not fully reliable, but believed to give data good enough for this purpose.

The object studied is a scaled model of a real assembly. The parts are bought from the sawmill without putting any care of how the specimens looks like, therefore the objectivity of this wooden structure is believed to be high.
2. Literature Review

2.1 Impact loads

The structure-borne sound or impact sound in timber buildings is mainly caused by a vertical impact load on the floor above, e.g. by people walking. The impact causes the floor to vibrate and the vibration may continue to the loadbearing walls, setting them into vibration as well (Bolmsvik, et al., 2012).

Human induced impact loads varies in excitation frequency. Walking is assumed to have the activity rate from 1.7 to 2.3 steps per seconds, which is a low frequency (Smith, 2003). A common assumption is that a human footfall contains force components up to 50 Hz (Jarnerö, et al., 2015) but the most dominated components is within the low frequency range 0-6 Hz (Smith, 2003).

Smith (2003) reports that humans tend to be sensitive to vibrations at the frequency range 4-8 Hz which is the eigenfrequency for the internal organs.

2.2 Elastomers

In Bolmsvik and Linderholt (2015) dynamic properties for elastomers strips was investigated by doing dynamic experiments and finite element analysis.

To model the elastomers in Abaqus (a FE-tool) they used springs and dashpots having stiffness in the x, y and z directions. By modelling the springs in a row the stiffness in the x- and y-rotational directions was fulfilled, but the rotational stiffness in z-direction had to be added separately.

When using different pre-loads on the elastomer they found that it had a significant effect on the first eigenmode, both in the FE-analysis and in the experimental measurements.

They concluded that the static stiffness for the elastomers, given by the supplier, did not give a good correlation between FE-analysis and experiments. Instead properties found by dynamic testing should be used.

2.3 Using elastomers in buildings

According to Cristian Berner (2013), a supplier of elastomers, the correct way to mount the elastomers is to point glue the elastomers to the wood with the distance c/c 600 mm or to have it free, half-embedded on top of the
walls. They stresses that it is important not to use nails or screws to mount the elastomers.

In a study, Bolmsvik and Brandt (2013) compared two types of wall-floor joints, one with the floor screwed to the underlying load-carrying walls and one with elastomers (of the type Sylomer®) between the floor and the walls. They found out that the joist configuration with elastomers had an average relative viscous damping of 2.1 % which was larger than the screwed with 1.2 %.

2.3.1 Vibration response of floors

Jarnerö, Brandt and Olsson (2015) have summarized many laboratory dynamic measurements on timber floors conducted by different researchers and found that the reported dynamic properties varied a lot. They found for example that the reported damping ratios varied from 1 % to 5 %. The differences in results depends on how the experiments were carried out, e.g., boundary conditions (i.e. if the floor is supported on two, three and four sides), joists spacing, floor span, type of material used and top flooring have an influence on the floor vibration.

In situ measurements have showed that the overall damping ratio for timber floors increases compared to damping ratio measured in laboratories (Jarnerö, et al., 2015). This is due to that the dynamic properties changes when the floor is integrated with the rest of the structural system and when additional surface layers, non-bearing internal walls and fixtures are installed. By measuring the vibration performance, both in laboratory and in situ during different steps of the erection of a timber framed residential building this was studied by Jarnerö, et al (2015). The floor system tested was a prefabricated floor element that consisted of a cross laminated board on top of beams made of glulam and with mineral wool in the space between the beams, see Figure 1.
As shown in Figure 1, the ceiling was separated from the floor and there was an elastomer (of the types Sylomer® and Sylodyn®) placed half embedded in the wall-floor junction.

One of the conclusions drawn in the paper is that the floor in situ showed good vibration performance. When measuring the third floor response in an almost finished eight story building they found that the first natural frequency occurred at 21.7 Hz with the damping ratio 6.5 %.

2.3.2 Vibration response of walls

Bolmsvik, Linderholt and Jarnerö (2012) did laboratory experiments on a floor construction consisted of a three-layer cross-laminated timber board (CLT board) resting on two high glulam beams representing loadbearing walls. The tests were performed using different test setups with different connections between the floor and the beams. The outcome showed that having the floor screwed to the walls might result in lower acceleration amplitudes out of the plane, of the top of the walls, compared to connections with elastomers in the frequency range 30 to 80 Hz.

Similar results have also been observed by Bolmsvik and Brandt (2013) and by Ljunggren and Ågren (2011). Bolmsvik and Brandt (2013) showed that the vibrations might increase perpendicular to the load direction in the low frequency range, when using elastomers between floor and walls in timber junctions. Ljunggren and Ågren (2011) found that the elastomer sometimes can increase the impact sound transmission. However the phenomenon was not noticed when Bolmsvik and Linderholt (2015) later did experiments on elastomers, half-embedded in steel.
3. Theory

3.1 Structure-borne sound and vibrations

Sound is defined as pressure waves that travel through a medium and the structure-borne sound is pressure waves that travels through structure elements in form of vibrations (Hassan, 2009). The pressure waves carry energy and can have varying configurations. According to Bolmsvik (2013) are the bending or flexural waves particularly interesting from an acoustic point of view since they gives large deflections in the perpendicular direction. This sideways deflection can explain the flanking transmission through the walls.

How much a floor vibrates depends on the mass, stiffness and damping of the entire floor system and its connections (Jarnerö, et al., 2015). The heavier the floor is, the less vibration will occur and together with the stiffness it determines the floors natural frequency. The floor damping has an impact on how long it takes for an induced vibration to decay.

Vibrations in timber structures shows an orthotropic behavior which means that it differs in the longitudinal and transversely directions (Quirt, 2009). Figure 2 represents the resulting variation of velocity of an impact blow on a wooden floor.

![Figure 2: Repartition of velocity on axial and transversal direction in wooden structure, figure from (Quirt, 2009)](image)

It is clear from Figure 2 that most of the distribution happens along the beam, but some vibrations spreads in the perpendicular direction as well.
3.2 Structural dynamics

The ambition of the following three subchapters is to remind the reader some fundamental equations and principles of structural dynamics. Those will be needed to understand the theoretical background for the experiments of this thesis. The theory in the following subchapters is mainly based on Craig and Kurdila (2006) and further insight could be found there.

3.2.1 Single degree-of-freedom systems

In order to introduce the fundamental equation in structural dynamics and linear vibration theory, Figure 3 is introduced. An external force, represented by $p(t)$, is applied to the mass $m$. Note that the spring is considered as linear and that only vertical motions are regarded. Also, the mass of the spring, damping in it and the air resistance are neglected.

![Diagram of a simple mass-spring-dashpot system](image)

*Figure 3: A simple mass – spring – dashpot system.*

By use of the Newton’s Laws, the second-order differential equation is given as,

$$m \cdot \ddot{u} + c \cdot \dot{u} + k \cdot u = p(t) \quad (1)$$

where $m$ is the mass in [kg], $c$ the coefficient of viscous damping in [N·s/m], $k$ the spring constant in [N/m]. The time dependent load is represented by $p(t)$.

Two important basic responses of this system can be illustrated when the mass is statically loaded and when the load varies with time; see the difference in Figure 4.
Figure 4: In a) a statically loaded system and in b) a system loaded with time-varying load.

Figure 4a illustrates the final static displacement when all transient states have been absorbed by the dashpot. Note that $U_s = f_0/k$.

Figure 4b illustrates the response of the mass depending on a time-varying load.

For the requisite of this thesis the broad spectrum of cases included into the term time-varying load needs to be narrowed. As a matter of fact, the load which will be applied later on the wooden structure tested in the frame of this thesis is periodic. It actually tries to mimic the behavior of the load performed by a walking person. In that situation the load $p(t)$ of equation (1) can be replaced by a simple harmonic time varying load,

$$p(t) = p_0 \cdot \cos \Omega t \quad (2)$$

where $p_0$ is the amplitude and $\Omega$ is the driving frequency.

Even if the free vibration of damped and undamped cases might be interesting from a theoretical point of view, only the damped single degree of freedom (SDOF) will be exposed in details in the following chapter.

3.2.2 Viscous damped SDOF system

The first step of this development is to combine equation (1) and equation (2) to give the equation of motion of the system represented in equation (3) with viscous damper and harmonic excitation.

$$m \cdot \ddot{u} + c \cdot \dot{u} + k \cdot u = p_0 \cdot \cos \Omega t \quad (3)$$

Since the excitation is harmonic, the total response will be expressed by a combination of a homogenous part $u_h$ and a particular part $u_p$, see equation (9). The latter is called the steady-state response and will have the following form because of the action of the damper;
\[ u_p(t) = U \cdot \cos (\Omega t - \alpha) \] 

where \( U \) is the steady-state amplitude and \( \alpha \) is the phase lag angle of the steady-state of the corresponding excitation.

By use of rotating vectors, \( U \) can easily be determined as;

\[ U = \frac{U_s}{\left( \frac{1-r^2}{r} + (2 \cdot \xi \cdot r)^2 \right)^{1/2}} \] 

with respectively the frequency ratio \( r \) and the modal damping \( \xi \);

\[ r = \frac{\Omega}{\omega_n}, \quad \xi = \frac{c}{c_r} \] 

where \( c_r \) is the critical damping factor.

In the same way \( \alpha \) is determined as,

\[ \tan \alpha = \frac{2 \cdot r \cdot \xi}{1 - r^2} \] 

When \( r \approx 1 \), because both natural frequency \( \omega_n \) and forcing frequency \( \Omega \) are close to each other, the particular part is limited only by the damping action,

\[ U_p = \frac{U_s}{(2 \cdot \xi)} \] 

The total response has the following form,

\[ u(t) = u_h + u_p(t) \] 

Replacing equation (5) in equation (4) and adding the homogeneous solution, equation (9) can finally be rewritten,

\[ u(t) = \frac{U_s}{\left[ (1-r^2) + (2 \cdot \xi \cdot r)^2 \right]^{1/2}} \cdot \cos (\Omega t - \alpha) + e^{-\xi \omega_d t} \cdot (A_1 \cdot \cos \omega_d t + A_2 \cdot \sin \omega_d t) \]

where the damping circular frequency is here defined by \( \omega_d = \sqrt{1 - \xi^2} \) [rad/s] and both constancies \( A_1 \) and \( A_2 \) can be determined by use of boundary conditions.

It has to be reminded that this equation refers to the starting transient state. Indeed, the decaying part will fade as time increases giving finally,

\[ u(t) = \frac{U_s}{\left[ (1-r^2) + (2 \cdot \xi \cdot r)^2 \right]^{1/2}} \cdot \cos (\Omega t - \alpha) \]
From equation (10) it is easy to have the undamped case by inserting $\xi = 0$,

$$u(t) = \frac{U_s}{(1 - r^2)} \cdot \cos \omega_n t + (A_1 \cdot \cos \omega_n t + A_2 \cdot \sin \omega_n t) \quad (12)$$

### 3.2.3 Multiple degrees-of-freedom systems

In order to solve simplified geometry with great accuracy to the reality, continuous models can be used. However the physical model of this thesis is far from a simple geometry. That is why despite the fact that dynamical behavior of many systems can be unscrambled to SDOF, *Multiple degrees-of-freedom systems, MDOF*, are needed for the following.

The general MDOF equation of motion of a structure is given by,

$$M \ddot{u} + C \dot{u} + K u = f(t) \quad (13)$$

where $M$ is the $N \times N$ mass matrix, $C$ is the $N \times N$ viscous damping matrix and $K$ is the stiffness matrix. The displacement $u(t)$ and the load vector $f(t)$ are $N \times 1$ vector. $N$ is the number of degrees of freedom.

### 3.3 Modal analysis in multiple degrees-of-freedom systems

The following chapter will go through the basic solution procedures used in order to solve MDOF systems.

The solution of equation (13) would require $N$ equations with $N$ unknowns. The *mode-superposition method* introduced in the present section makes it possible, by use of normal modes of the undamped system, to transform the set of coupled equation to a set of uncoupled equations.

Obtaining the frequencies and modes by solving the eigenvalue problem bellow is the first step of mode-superposition solution.

$$[K - \omega^2 M] \phi_r = 0 \quad (14)$$

Equation (14) gives the *eigenpairs* $(\omega^2_r, \phi_r)$ with $r = 1, 2, \ldots N$.

Thanks to the *modal masses*, $M_r$, and *modal stiffnesses*, $K_r$, the modes $\phi_r$ are normalized,

$$M_r = \phi_r^T M \phi_r, \quad K_r = \phi_r^T K \phi_r = \omega^2_r M_r \quad (15)$$
In order to satisfy the orthogonality property\(^{1}\) of the natural modes and to orthogonolize the modes representing the repeated frequencies, the following orthogonality equations must be satisfied for all \( r \neq s \),
\[
\phi_r^T M \phi_s = \phi_r^T K \phi_s = 0
\]  
(16)

Finally the modal matrix can compile all modes in,
\[
\Phi = [\phi_1 \ \phi_2 \ \ldots \ \phi_N]
\]  
(17)

Henceforward, both matrices \( M \) and \( K \) can be rewritten as,
\[
M = \Phi^T M \Phi = \text{diag}(M_r),
\]  
(18)

\[
K = \Phi^T K \Phi = \text{diag}(K_r) = \text{diag}(\omega_r^2 M_r)
\]

The final and key step of the mode-superposition method consist in introducing the coordinate transformation,
\[
\mathbf{u}(t) = \Phi \mathbf{\eta}(t) = \sum_{r=1}^{N} \phi_r \eta_r(t)
\]  
(19)

where \( \eta_r(t) \) refers to principal coordinates in equation (13) in order to derive the equation of motion in principal coordinates, i.e.,
\[
M \ddot{\mathbf{\eta}} + \mathbf{C} \dot{\mathbf{\eta}} + \mathbf{K} \mathbf{\eta} = \mathbf{f}(t)
\]  
(20)

where \( M = \Phi^T M \Phi \) is the modal mass matrix: \( \text{diag}(M_r) \),
\( \mathbf{C} = \Phi^T \mathbf{C} \Phi \) is the modal damping matrix,
\( K = \Phi^T K \Phi \) is the modal stiffness matrix: \( \text{diag}(\omega_r^2 M_r) \)
and \( \mathbf{f}(t) = \Phi^T \mathbf{p}(t) \) is the modal force vector.

Equation (20) needs to be changed in a set of \( N \) uncoupled modal equations of motion. In order to do that, the matrix \( \mathbf{C} \), which is non diagonal, has to be diagonalized in a diagonal generalized damping matrix. Raleigh damping and modal damping are two common procedures to define it. However, the most recognized one is the modal damping.

This uncoupled modal damping, satisfying the orthogonality property, can be written in the equation bellow to wit,
\[
\mathbf{C} = \Phi^T \mathbf{C} \Phi = \text{diag}(C_r) = \text{diag}(2\xi_r \omega_r M_r)
\]  
(21)

---

\(^{1}\) The orthogonality property is the most important property of natural modes. Further information can be found in (Craig & Kurdila, 2006).
Next equation (20) can be reduced to the following set of \( N \) uncoupled modal equations of motion:

\[
M_r \ddot{\eta}_r + 2M_r \dot{\xi}_r \omega_r \eta_r + \omega_r^2 M_r \eta_r = f_r(t) \quad r = 1, 2, \ldots, N \quad (22)
\]

or in mass normalized form,

\[
\ddot{\eta}_r + 2 \xi_r \omega_r \dot{\eta}_r + \omega_r^2 \eta_r = \frac{1}{M_r} f_r(t) = \frac{1}{M_r} \Phi_r^T \mathbf{p}(t), \quad r = 1, 2, \ldots, N \quad (23)
\]

If the excitation is harmonic, i.e. considering \( \mathbf{p}(t) = \mathbf{P} \cos \Omega t \), equation (23) can be written,

\[
\ddot{\eta}_r + 2 \xi_r \omega_r \dot{\eta}_r + \omega_r^2 \eta_r = \frac{1}{M_r} F_r \cos \Omega t \quad (24)
\]

where \( F_r = \Phi_r^T \mathbf{P} \).

Equation (24) is solved by the complex frequency response technique,

\[
\ddot{\eta}_r + 2 \xi_r \omega_r \dot{\eta}_r + \omega_r^2 \eta_r = \frac{1}{M_r} F_r e^{i \Omega t} \quad (25)
\]

Next, the steady-state solution \( \bar{\eta}_r \) can be formulated in the following form,

\[
\bar{\eta}_r = H_{\eta r/F_r}(\Omega)F_r e^{i \Omega t} \quad (26)
\]

where \( H_{\eta r/F_r}(\Omega) \) is the so-called complex frequency-response function in principal coordinates, formulated so,

\[
H_{\eta r/F_r}(\Omega)F_r = \frac{1}{K_r} e^{i \Omega t} \left( \frac{1}{1 - r_r^2} + i \cdot (2 \cdot \xi_r \cdot r_r) \right) \quad (27)
\]

where \( r_r = \frac{\Omega}{\omega_r} \) represents the modal frequency ratio.

Following the reasoning, both modal response \( \eta_r(t) \) and phase angle \( \alpha_r \) can be given by,

\[
\eta_r = \frac{F_r/K_r}{\sqrt{(1 - r_r^2)^2 + (2 \cdot \xi_r \cdot r_r)^2}} \cos (\Omega t - \alpha_r) \quad (28)
\]

with

\[
\tan \alpha_r(f) = \frac{2 \xi_r \cdot r_r}{1 - r_r^2} \quad (29)
\]

Equation (19) can now be written in the complex form,

\[
\bar{\mathbf{u}}(t) = \Phi \bar{\eta}_r(t) = \sum_{r=1}^{N} \phi_r \bar{\eta}_r(t) \quad (30)
\]

where \( \eta_r(t) \) refers to principal coordinates.
The next step is combining equations (26), (27) and (30) to finally get,

\[ \mathbf{u}(t) = \sum_{r=1}^{N} \phi_r \phi_r^T P \frac{1}{K_r} \frac{1}{(1 - r_r^2) + i \cdot (2 \cdot \xi_r \cdot r_r)} e^{i\omega t} \quad (31) \]

The unit harmonic excitation at \( p_j \) giving a response at coordinate \( u_i \) is given by the complex frequency-response function (FRF) in generalized coordinates \( H_{ij}(\Omega) \), to wit;

\[ H_{ij}(\Omega) = H_{u_i/p_j}(\Omega) = \sum_{r=1}^{N} \phi_{ir} \phi_{jr} \frac{1}{K_r} \frac{1}{(1 - r_r^2) + i \cdot (2 \cdot \xi_r \cdot r_r)} \quad (32) \]

where \( \phi_{ir} \) is the element in row \( i \) and column \( r \) of the modal matrix \( \Phi \).

Both real part and imaginary are not introduced in this theoretical section because it is beyond of the scope of what will be really used in this thesis. More advanced developing surely can be found in literature.

3.4 Frequency-response function

3.4.1 Frequency-Response of motion for SDOF Systems

Equation (13) of chapter 3.2.3 is re-introduced for the purpose of this chapter,

\[ M \ddot{u} + C \dot{u} + K u = p_0 e^{i\omega t} \quad (33) \]

where \( M \) is the \( N \times N \) mass matrix, \( C \) is the \( N \times N \) viscous damping matrix and \( K \) is the stiffness matrix. The displacement \( u(t) \) and the load vector \( p_0 e^{i\omega t} \) are \( N \times 1 \) vector.

The steady-state complex displacement response has the form,

\[ u(t) = U \cdot e^{i\omega t} \quad (34) \]

or written in detailed form below,

\[ u(t) = \frac{p_0/k}{(1 - r^2) + i \cdot (2 \cdot \xi \cdot r)} e^{i\omega t} \quad (35) \]

with the corresponding receptance FRF given by,

\[ H_{u/p}(f) = \frac{1/k}{(1 - r^2) + i \cdot (2 \cdot \xi \cdot r)} \quad (36) \]

where \( r = f / f_n \).
Next the complex velocity and acceleration can respectively be deduced from equation (34),

\[ v(t) = i\omega U e^{i\omega t} \quad \text{and} \quad a(t) = -\omega^2 U e^{i\omega t} \]  

(37)

Finally, the velocity output per unit force input - mobility FRF- is,

\[ H_{v/p}(f) = i\omega \frac{1/k}{(1 - r^2) + i \cdot (2 \cdot \xi \cdot r)} \]  

(38)

and the acceleration output per unit force input - accelerance FRF- is

\[ H_{a/p}(f) = -\omega^2 \frac{1/k}{(1 - r^2) + i \cdot (2 \cdot \xi \cdot r)} \]  

(39)

3.4.2 Frequency-Response of MDOF Systems

By reason of the fact that the damping of the system is modelled by modal damping the development of the FRF here below is only based on real normal mode. Starting from the equation of receptance FRF introduced earlier in the theory,

\[ H_{ij}(f) = H_{ui/pj}(f) = \sum_{r=1}^{N} \frac{\phi_i \phi_j r}{K_r} \frac{1}{(1 - r^2) \cdot (1 - r^2) + i \cdot (2 \cdot \xi \cdot r)} \]  

(40)

the magnitude is given by,

\[ |H_{ij}(f)| = \left| H_{ui/pj}(f) \right| = \sum_{r=1}^{N} \left| \frac{\phi_i \phi_j r}{K_r} \right| \frac{1}{\sqrt{(1 - r^2)^2 + (2 \cdot \xi \cdot r \cdot r)^2}} \]  

(41)

and the phase angle \( \alpha_r \) given by,

\[ \tan \alpha_r(f) = \frac{2 \cdot \xi \cdot r \cdot r}{1 - r^2} \]  

(42)

Again, as derived in equation (7) and (8), mobility FRF can be obtained for MDOF systems including modal damping,

\[ H_{v_i/p_j}(f) = i\omega \sum_{r=1}^{N} \frac{\phi_i \phi_j r}{K_r} \frac{1}{(1 - r^2) \cdot (1 - r^2) + i \cdot (2 \cdot \xi \cdot r \cdot r)} \]  

(43)

or in the same manner the acceleration output per unit force input-acceleration FRF- is,

\[ H_{a_i/p_j}(f) = -\omega^2 \sum_{r=1}^{N} \frac{\phi_i \phi_j r}{K_r} \frac{1}{(1 - r^2) \cdot (1 - r^2) + i \cdot (2 \cdot \xi \cdot r \cdot r)} \]  

(44)
3.4.3 Experimental Modal Analysis

Analyzing data from dynamic experiments is called experimental modal analysis, EMA (Brandt, 2011). EMA is based on the mode superposition principle of MDOF models of structure (see section 3.3). By use of vibrational testing and accelerometers it provides the frequency response functions (also abbreviated: FRFs) from which the modal properties, i.e. modes shapes, damping factors, natural frequencies, are deducted.

To get good modal properties there are some steps and choices that needs to be considered (Brandt, 2011). The first thing to consider is how the structure should be placed, e.g. suspended to have free-free boundary conditions. Other important things to consider are which excitation signal to use (see section 3.5) and where to place the accelerometers to be able to catch as many mode shapes as possible. Since the number of accelerometers represent the number of degree of freedoms it is important to make sure that they are not placed in node lines; lines where there are no motion for one or more mode shapes.

When all important choices have been made it is a good idea to do preliminary tests to see that the excitation signal gives results with good quality (Brandt, 2011). Further preliminary tests that should be checked are linearity and reciprocity. Linearity can be checked by applying two different excitation force levels and study the FRFs for one point on the structure. If the structure is linear the FRFs should be equal. Reciprocity needs to be checked when using two or more excitations sources. When comparing the FRF for one excitation point when loading at the other to the vice versa case, reciprocity can be estimated. If the FRFs are equal it means that the excitation is correct.

3.4.4 Acquisition system

A data acquisition system is designed to perform some measurements from a physical setup and transform those sampling signals into digital values. Thanks to a host computer running adequate controlling software the data can be further manipulated and analyzed.

The traditional components of data acquisition systems comprehend (Goswami, 2015):

- Sensors measuring physical variables and converting them into electrical signals.
- Signal conditioning circuitry converting those electric signals into digital values.
- Analog-to-digital converters, converting sensor signals to digital values.
A common acquisition system is the LMS Test.Lab 14A system distributed by Siemens, see Figure 5. It combines a high-speed multi-channel data acquisition system to which accelerometers are connected. The testing and analytical tools are integrated and make it possible to analyze the structure while measuring the prototype. The response of a structure submitted to vibrations can be evaluated by the Modal Analysis Tool of the LMS system. It is used to measure the FRF in some points of the physical model, evaluate the mode shapes and calculate the synthetized FRF of the whole structure (LMS, 2014).

![Figure 5: Lms system, from (LMS, 2014)](image)

3.5 Excitation signal source

There are a lot of different excitation methods that can be used when studying vibrations. Two of these, mentioned by Negreira (2013), are the ISO tapping machine, a standardization method commonly used in Sweden, and the rubber tie, the standard method used in Japan. Further methods mentioned by Negreira (2013) are shaker, Japanese ball and impact hammer. These are non-standardized sources that are preferred by many researchers. Homb (2006) found that the tapping machine, which is a discrete type of excitation, could not fully mimic a foot-fall.

When all important choices have been made it is a good idea to do preliminary tests to see that the excitation signal gives results with good quality (Brandt, 2011). Further preliminary tests that should be checked are linearity and reciprocity. Linearity can be checked by applying two different excitation force levels and study the FRFs for one point on the structure. If the structure is linear the FRFs should be equal. Reciprocity needs to be checked when using to or more excitations sources. When comparing the FRF for one excitation point when loading at the other to the vice versa case
reciprocity can be estimated. If the FRFs are equal it means that the excitation is correct.

3.6 Compression test

Compressions test, among others, can be used to evaluate material stiffness. The purpose of it is to apply an increasing static load and measure the displacement; this gives a deflection curve from which the stiffness deducted by measuring the slope of the curve. The curve should be limited between two points where the reactional behavior of the measured material is linear (COMSOL, 2014).

The force-displacement relationship can be expressed by:

\[ F(u) = F_0 + k(F_0, u_0)(u - u_0) \]  \hspace{1cm} (45)

where, \( F_0 \) is the force acting on the material deformed by a distance of \( u_0 \).

When a small force, \( \Delta F \), is applied to deform the body by an infinitesimally distance, \( \Delta u \), then the ratio of those will give the stiffness at the operating point given by the state variables \( F_0 \) and \( u_0 \). The linear stiffness can be mathematically expressed by equation

\[ k(F_0, u_0) = \lim_{\Delta u \to 0} \frac{\Delta F}{\Delta u} = \frac{\partial F_0}{\partial u_0} \]  \hspace{1cm} (46)

where \( \Delta F \) is the applied force in [N] and \( \Delta u \) is the displacement in [m].

3.7 Finite Element modeling

The Finite Element Method is a powerful approximation which could be aligned with the so-called Assumed- Modes Method introduced in chapter 3.2.3.

The advantage of the FE method compared with the Assumed- Modes Method is that, thanks to technological progress in informatics, the integrals reflecting the deflected shape of the entire structure subjected to analysis are easily solved. Moreover the fact of using this different approach makes it easier for the analyst to choose the right set of integrals, also called deflections shapes, to model the system (Craig & Kurdila, 2006).

Practically, many of these shapes functions are assembled in order to model the desired structure (Ottosen & Petersson, 1992). In order to properly mimic the behavior of the studied structure, appropriate displacement compatibility rules are required on the nodes, joints between shape functions, and also proper boundary condition.
Nowadays FE modeling can be considered as the principal computational tool for structural analysis thanks to the evolution of technology (Brandt, 2011).

3.7.1 Mesh Convergence study

The mesh Mesh Convergence study is used to find the size of mesh and type of element that satisfactorily balances accuracy and computing resources while ensuring that the results are not altered.

The formal manual method commonly used to establish the mesh is the curve of critical result parameter, see Figure 6. In the frame of this thesis this result parameter will be the frequency analysis. At least three runs are required to obtain a relevant curve.

![Four points convergence curve](image)

*Figure 6: Mesh convergence curve.*

The last two points show that convergence is achieved at level of mesh refinement 7. At each run every elements of the model should normally be split but it is not necessary. The Saint Venant’s principle implies that a variation of the mesh in the model do not affect all elements of the mesh (NAFEMS, 2016). The tool implemented in Abaqus stemming from this principle is called the adaptivity meshing and refine the mesh only in the regions of interest. Unfortunately this option is not implemented for the frequency function that is why only the manual method is employed.

3.8 Evaluation

There are different methods used to evaluate results from experiments and FE-analysis. One method is to compare the mode shapes similarity using
modal assurance criterion, MAC (Brandt, 2011). MAC compares the correlation numerically between two vectors containing mode shapes and stores the result in a matrix. There are two types of MACs one called auto-MAC and one called cross-MAC.

Auto-MAC is calculated using mode shape vectors from the same set, e.g. experimental or analytical data, as

\[
MAC_{AUTO} = \frac{\left(\psi_r^T \cdot \psi_s\right)^2}{\psi_r^T \cdot \psi_r \cdot \psi_s^T \cdot \psi_s}
\]  

(47)

where \(\psi_r\) and \(\psi_s\) are mode shape vectors stemming from the same set.

To compare modes from two different sets, e.g. experiments and analysis or two different sets stemming from experiments or FE-analysis, the cross MAC can be calculated as:

\[
MAC_{CROSS} = \frac{\left(\psi_{Ar}^T \cdot \psi_{Bs}\right)^2}{\psi_{Ar}^T \cdot \psi_{Ar} \cdot \psi_{Bs}^T \cdot \psi_{Bs}}
\]  

(48)

where \(\psi_{Ar}\) and \(\psi_{Bs}\) are a mode shapes stemming from two different data sets.

If the MAC value between two modes is equal to or close to one then it is an indication that the modes are numerically equal and if the MAC is equal to or close to zero then modes are not equal (Allemang, 2003). In Figure 7 an example is shown of how the result from MAC calculations can be visualized.

![Figure 7: An example of a MAC-matrix, from (Allemang, 2003)](image)

The example shown above indicates that mode four and five are equal and that there are similarities between mode 12 and mode 13.
4 Method and implementation

This thesis is based on both experimental and analytical studies. Three different experiments were performed: microphone and analyzer, modal analysis and compression test. The analytical part was done with the Modal Analysis interface implemented in LMS and the FE modeling with Abaqus. Figure 8 gives an overview of those steps.

Figure 8: Thesis’ proceeding

The experimental steps are described here below and further detailed in the following chapters. Some pictures are given in the Appendix 1 in order to illustrate the laboratory work.

- Every beam were cut into the desired length and the density for the chipboard and gypsum were measured.

- The second step was to measure the axial modulus of elasticity for every beam and stud used in the physical model by use of a microphone and analyzer (see section 4.2).

- After this the structure was assembled.
The fourth step was the dynamic experiment on the assembled structure in different configurations using LMS Test Lab 14A, see Figure 9.

The final experimental step was to perform compression tests on the elastomers used in the set-up in order to verify the values given by the supplier.

The analytical analysis were done in parallel following those steps;

- Once the elasticity of the beams were given from experiment (step 2), the real properties of every beam could be implemented in the FE model (see Figure 10). The analysis of the natural frequencies and the mode shapes of the structure were performed in order to evaluate the positions of the accelerometers and shaker.

- After the dynamic experimental step in LMS was completed the FE model were relined and tuned in order to fit the modal data identified in the dynamic experiment.

Both auto-MACs and cross-MACs were used to evaluate the mode shapes stemming from both the experimental and the analytical approach.
4.1 Material

All materials used for the experiments were bought at a regular lumberyard in Sweden. No special care was taken when choosing the specimens to get a natural spread in the material. This disparity can be observed in Table 7 in chapter 5. The timber bought for the walls where of the quality C14 and the one for the floor where of the quality C24. More details about the material can be seen in Table 1.

Table 1: Material bought and used for the structure.

<table>
<thead>
<tr>
<th>Type</th>
<th>Length [mm]</th>
<th>Width [mm]</th>
<th>Height [mm]</th>
<th>Quality</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Beams</td>
<td>2500</td>
<td>45</td>
<td>70</td>
<td>C14</td>
<td>12</td>
</tr>
<tr>
<td>Beams</td>
<td>3300</td>
<td>45</td>
<td>145</td>
<td>C24</td>
<td>4</td>
</tr>
<tr>
<td>Chipboards</td>
<td>1800</td>
<td>600</td>
<td>24</td>
<td>-</td>
<td>3</td>
</tr>
<tr>
<td>Gypsumboards</td>
<td>2400</td>
<td>900</td>
<td>12.7</td>
<td>-</td>
<td>4</td>
</tr>
</tbody>
</table>

Wooden materials were stored during two weeks in the same environment as the tests were performed to ensure that the material was in equilibrium with the ambient climate. Both densities for the chipboard and the gypsumboard were calculated.
4.2 Microphone and analyzer

The axial modulus of elasticity of every beam was measured using a hammer impulse technique. A hammer blow was performed on the edge of every beam and the respective impulse response was recorded by a microphone (GRAS, type 26 CA). By use of the analyzer (DataPHYSICS DP700-60) the impulse response was transformed to the frequency domain with Fast Fourier transform software. Only the first axial mode was considered and the global axial dynamic modulus of elasticity $E_{dyn,1}$ is given by,

$$E_{dyn,1} = 4 \cdot \rho (f_{a,1} L^2)$$

(49)

where $\rho$ is the density [kg/m$^3$], $f_{a,1}$ is the axial resonance frequency [Hz] of the first axial mode of vibration and $L$ is the length [m] of the beam (Oscarsson, et al., 2014).

In order to gain objectivity in the results, different kind of blows were performed at different places. The aim of it was to check the influence of the intensity and the position of the blow on the material. Also the surface on which the tested beams were placed was also modified. It was observed that a beam placed on soft supports or hard supports gave the same results. Finally the protection of the microphone was taken away to see its influence. It was observed that without this protection, which gave a little dispersion of the signal, the curve given by the program was more pronounced. The last one was used to find out the axial modulus of elasticity of every beam given in Table 7 in chapter 5. In Figure 11 is a view of the set-up for the measurements given, other pictures are given in Appendix 1.

Figure 11: Set-up of the microphone and analyzer experiment
4.3 Assembling of the structure

As the structure, see Figure 12, has for purpose to mimic what is actually used in the industry, no extra efforts for accuracy were made regarding the dimensions of the structure. It means that the walls and the floor had an error up to half centimeter. Nevertheless extra glue and screws were used in order to have a stiff structure. Finally some tension and torsion were given in some of the beams to have walls and a roof rectangular enough. How the elastomers were placed in the junctions can be seen in blue in Figure 12.

![Figure 12: The structure without covering sheets](image)

4.4 Vibrational Experiment

The elastomers used in this thesis are blue SR 28 Sylomer® and red SR 220 Sylomer® delivered by Getzner Werkstoffe GmbH. Those were used within their static range given by Table 2.

<table>
<thead>
<tr>
<th>Elastomer type</th>
<th>Static range of use [N/mm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>SR28</td>
<td>0.028</td>
</tr>
<tr>
<td>SR220</td>
<td>0.22</td>
</tr>
</tbody>
</table>

In order to evaluate the general behavior on whole static range of the blue elastomers SR28 different levels of pre-load were subjected. The red elastomers were placed under the structure in the second configuration (see section 4.4.1).
4.4.1 Test setups

In this study, two strips of blue SR28 elastomers were used in the wall-floor junctions having an undeformed dimension of 80 x 25 x 1200 mm. The elastomers were placed on top of the walls, clamped by the weight of the floor, see Figure 13a. In the second configuration (see Figure 13b) both SR28 and SR220 elastomers were used; the SR28 placed as in the first configuration and the SR220 under the structure, as a piece with dimensions 40 x 25 x 100 mm in each corner. The reason to why the structure was separated from the concrete floor was to investigate how it influenced the eigenfrequencies and the damping ratios.

![Figure 13: Two configurations: a) standing on a heavy concrete floor (hereinafter called fixed structure) and b) standing on four SR220 elastomer (hereinafter called damped structure).](image)

Twenty accelerometers were placed on the floor and 28 on each wall. The accelerometers were uniaxial accelerometers of two different types. The decision to have more accelerometers on the walls was taken since studies have shown that the elastomers can increase the walls vibration out of the plan, see chapter 2.3.2., making responses of the walls particular interesting.

4.4.2 Dynamic test

An electromagnetic shaker was used to create a dynamic load on wall A and the floor of the structure, see Figure 14. The shaker was hanging from the roof in elastic ropes and was attached to the structure via a stinger. Pictures from the dynamic experiments can be seen in Appendix 1.
The excitation of the wall was done to represent the force component parallel to the floor that arises from walking.

A stepped sine signal going from 4 Hz to 250 Hz was applied with an increment of 0.5 Hz, see Table 3.

**Table 3: Amplitude of the load**

<table>
<thead>
<tr>
<th>Frequency range [Hz]</th>
<th>Load [N]</th>
<th>Frequency range [Hz]</th>
<th>Load [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>0.5</td>
<td>70</td>
<td>3.5</td>
</tr>
<tr>
<td>10</td>
<td>0.5</td>
<td>80</td>
<td>4</td>
</tr>
<tr>
<td>20</td>
<td>1</td>
<td>90</td>
<td>4.5</td>
</tr>
<tr>
<td>30</td>
<td>1.5</td>
<td>100</td>
<td>5</td>
</tr>
<tr>
<td>40</td>
<td>2</td>
<td>110</td>
<td>5.5</td>
</tr>
<tr>
<td>50</td>
<td>2.5</td>
<td>120</td>
<td>6</td>
</tr>
<tr>
<td>60</td>
<td>3</td>
<td>250</td>
<td>6.5</td>
</tr>
</tbody>
</table>

Thirty accelerometers were used, mounted with wax to the structure. Since the number of nodes was 76 several runs were needed in order to cover the entire structure.
Totally six runs were performed, see Table 4

Table 4: The preformed test runs

<table>
<thead>
<tr>
<th>Set</th>
<th>Shaker position</th>
<th>Boundary condition</th>
<th>Load Floor [kg]</th>
<th>Extra load [kg]</th>
<th>Total load [kg]</th>
<th>Per meter elastomer [kg/m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Floor</td>
<td>Concrete</td>
<td>76</td>
<td>0</td>
<td>76</td>
<td>31.7</td>
</tr>
<tr>
<td>2</td>
<td>Wall</td>
<td>Concrete</td>
<td>76</td>
<td>0</td>
<td>76</td>
<td>31.7</td>
</tr>
<tr>
<td>3</td>
<td>Floor</td>
<td>Concrete</td>
<td>76</td>
<td>86</td>
<td>162</td>
<td>67.5</td>
</tr>
<tr>
<td>4</td>
<td>Floor</td>
<td>Concrete</td>
<td>76</td>
<td>240</td>
<td>316</td>
<td>131.7</td>
</tr>
<tr>
<td>5</td>
<td>Floor</td>
<td>Elastomer</td>
<td>76</td>
<td>0</td>
<td>76</td>
<td>31.7</td>
</tr>
<tr>
<td>6</td>
<td>Wall</td>
<td>Elastomer</td>
<td>76</td>
<td>0</td>
<td>76</td>
<td>31.7</td>
</tr>
</tbody>
</table>

The first set measured was the structure placed directly on the concrete floor in the laboratory hall (see Figure 13a) with the floor itself being the only mass load (76 kg) on the elastomers (hereinafter called no load). The excitation was placed on the floor, see Figure 14. The fixed structure was also tested with the shaker placed sideways, exciting the wall.

Two different mass loads were then added on top of the structure (see Figure 15). The first added mass (hereinafter called mass load 1) had a total weight of roughly two times 86 kg, making the total load on the elastomers 162 kg, including the weight of the floor. The second mass tested (hereinafter called mass load 2) had a total weight of roughly two times 120 kg, making a total load on the elastomers 316 kg. The measurements were done having the shaker on the floor. These sets are called 3 and 4.

The final sets were done on the configuration with four elastomer pieces placed under the structure, one in each corner, separating the structure from the concrete floor (see Figure 13b). The measurements were done with both the shaker being placed on the floor, set 5, and on the wall, set 6.
4.4.3 Modal data

To identify the modal parameters, experimental modal analysis was done using the modal analysis tool in LMS Test Lab 14A.

Two preliminary tests were done; checking linearity and reciprocity (see section 3.4.3). The linearity of the structure was verified, both with and without added mass loads. It was done by comparing several points FRF on the structure when applying the load according to Table 3 with load having half the amplitude. Since the FRFs were equal the structure is linear according to Brandt (2011).

The reciprocity was tested by plotting the FRF of the force point on the floor while exciting the force point on the wall as well as the FRF of the point on the wall when exciting the point on the floor. The two FRFs are shown in Figure 16.
The red curve is the FRF for the point on the floor and the green curve is for the point on the wall. The similarity between the different FRFs ensures that the shaker was properly attached to the structure giving the same excitation.

To extract the mode shapes from the FRFs pole picking was done using PolyMAX, a sum of the FRFs from all points and a multivariate mode indicator function, MIF.

For the configurations that were measured both with the excitation of the floor and the wall the mode shapes were preliminary extracted separately. When comparing the synthesized FRF with the FRFs for each point, using the synthesis modal tool in LMS, it was clear that the correlation was much higher for the points on the floor when having the shaker exciting the floor than compared to the FRFs of the points on the walls. The same was found when having the shaker on the wall resulting in much higher correlation between the synthesized FRF and the FRFs for each point on the walls.
As a second step the FRFs from both sets were combined, using the previous extracted mode shapes as references. The same procedure were used for both the fixed and the damped configuration.

4.5 Compression test

The test was performed with a MTS 810 material testing system. Hydraulic Wedge Grip MTS 647 was coupled to it to grip the crossheads. The system is a completely integrated testing package design to measure the mechanical properties. The setup is seen in Figure 17.

The hydraulic lifts and locks allowed the manufactured 100x100 mm crosshead to be easily positioned. The same dimension as in the dynamic tests were used for the red elastomer, 40x100 mm surface. However crosshead could not be manufactured for the blue elastomers (1200x80 mm). Therefore the compression test was applied on a sample of 80x100 mm for the blue elastomer instead. Thanks to the shape factor given by the supplier those measurement could be extrapolated to the values used in the real setup and implemented in Abaqus.

Figure 17: Two compression tests, where a) is the compression test of SR 28 and b) is the compression test of SR 220.
4.6 Finite Element Modeling

4.6.1 Frequency analysis

A natural frequency analysis of the structure was required beforehand the vibrational testing and a convergence study were done to the model with frequency analysis as result parameter (see section 3.7.1). The FE model was first updated with the actual properties of every beam stemming from experiment 1 and the densities of the gypsum and the chipboard.

In order to calculate the natural frequencies and the corresponding mode shapes of the structure, Abaqus performs an eigenvalue extraction. Three methods can be used; the Lanczos method, the automatic multi-level sub structuring (AMS) and the subspace iteration. Those methods can be further theoretically explored by use of adequate literature as for example (Craig & Kurdila, 2006) but it is beyond the purpose of this section. After having run the three of them and followed the guideline of the manual of Abaqus the AMS eigensolver was found to be the fastest and the most adequate for a model with such a big amount of degrees of freedom as in this study.

The model described and represented in Figure 18a were used only for studying the natural frequencies. Instead of modeling elastomers, as what is the case in reality, a tie connection between both walls and the floor were used. The surface of contact shown in Figure 18b was used for the tie connection. The reason why elastomers were not model at this stage of the thesis is because the properties of the elastomers were not known perfectly yet.

![Figure 18a](image.png)  ![Figure 18b](image.png)

*Figure 18: In a) the general structure and in b) a zoom of the tie surface of contact in the junctions*

Different surfaces of contact were used between the floor and the walls to analyze the impact on the modes shapes. The result was that the mode shapes were similar if the surface of contact between the roof and both walls was a line or a surface.
An important result given by the frequency analysis was the number of modes in function of the element type used. The C3D8R element with an element size of 0.025 m, see Table 5, gave 300 modes for a range going from 1 Hz to 150 Hz which is far too many to be a correct result. The C3D20R was finally used for the simulations, this type element gave 17 modes from 1 Hz to 150 Hz.

### Table 5: Parameters of mesh convergence study

<table>
<thead>
<tr>
<th>Size of mesh [m]</th>
<th>Element types</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.025</td>
<td>C3D8R, 8-node linear brick elements</td>
</tr>
<tr>
<td>0.015</td>
<td>C3D10: A 10-node quadratic tetrahedron</td>
</tr>
<tr>
<td>0.010</td>
<td>C3D20R: A 20-node quadratic brick, reduced integration</td>
</tr>
</tbody>
</table>

The size element of 0.010 m took far too much computing resources therefore larger element sizes were tested. The results of the convergence study having why element size of 0.015 and 0.025 can be seen in Figure 19, which show that an element size of 0.025 m were enough.

<table>
<thead>
<tr>
<th>Index</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>8 nodes, 8 nodes, 8.53 Hz @ 34.8 Hz</td>
</tr>
<tr>
<td>2</td>
<td>10 nodes, 10 nodes, 8.53 Hz @ 34.8 Hz</td>
</tr>
<tr>
<td>3</td>
<td>20 nodes, 20 nodes, 8.53 Hz @ 34.8 Hz</td>
</tr>
</tbody>
</table>

After several run with those different parameters the most accurate solution was selected out of the convergence study. The combination of the C3D20R element type with an element size of 0.025 gave the best result without being overly demanding of computing resources. The use of adaptive meshing tool, which allows the program to refine the mesh at areas where it is needed, could unfortunately not be used within Abaqus for the frequency step analysis.

This previous investigation of the 3D Finite Element model had for aim to give enough understanding of how the real structure would behave. More specifically it gave an idea of the different modes shapes of the structure without elastomers placed in the junctions. Knowing this it was possible to select the positions of the accelerometers on the structure and the position of

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Sarah Stenberg & Adrien Vercruysse
the shaker in order to capture the maximum number of mode shapes. In Figure 20 the structure is represented with the positions of the accelerometers.

![Image](image.png)

*Figure 20: Accelerometer’s positions on the structure, marked with red dots.*

### 4.6.2 Frequency Response Function analysis

The second part of the analytical analysis was performed by use of two different structures in Abaqus; the first one is called “fixed structure” and the second one is called “damped structure” (for explanation see section 4.4.1). Those models went through a *Steady State Dynamics Analysis*.

**Fixed Structure**

The first set-up tested within the laboratory was the structure standing on the concrete floor directly. The only difference between this finite element model and the one used for the frequency analysis is the fact that elastomers were between the walls and the floor. The complete structure (walls and floor) was standing directly on the floor. Figure 21 illustrates the model.
A good way to model the elastomers in Abaqus was proved to be with springs and dashpots on a row (Negreira, 2013). This way of modelling gives a good stiffness in rotational directions rot-x and rot-y (Bolmsvik & Linderholt, 2015). Besides, the use of multipoint constraints MCP is an effective way to relate degree of freedom to one another within the elastomers. The purpose of the selected springs for the modeling is to transfer stiffness of the elastomers between the nodes of walls and the nodes of the floor. The two nodes attached to every spring have identical degree of freedom.

The length of the spring is the height of the elastomers. Six springs were used along each connection giving total twelve springs for both elastomers and a total of 48 spring stiffnesses (x, y, z and rot-z directions). In Figure 22 it is shown some details of the MCPs linked to the springs.
Figure 22: In a) detail of MCP on a wall and in b) a system loaded with time-varying load.

The second set-up tested was the same structure as above but with elastomers placed between the structure and the concrete floor, see Figure 23.

Figure 23: Damped model in Abaqus.
The combination of springs, dashpots and MCP was once more used for the modeling the elastomers under the structure.

Regarding the floor, or more precisely the concrete in the laboratory, the option of building small block of high density and stiffness was chosen to link the MCP to it for one side and set encastre boundary condition on the other side, see Figure 24.

![Figure 24: In a) MCP of the floor b) boundary conditions of the floor.](image)

Steady State Dynamics Analysis function was performed on both models but only used as reference points with what was giving the LMS system therefore no further explanation are given on that subject.

Tuning the analytical model is trying to add realistic parameters in order to fit the reality. It was found out that different parameters were important: the modulus of elasticity of the gypsum and of the chipboard. They might have a higher importance for the tuning of the FE model. Theses parameters were modified in order that the FE model returns similar mode shapes as the physical model.

In order to tune the model every parameter was modified separately to analyze its influence on the modes shapes given by the frequency function in Abaqus. The experiment of the compression unfortunately did not give excellent results but it gave a point of comparison with the values kindly sent by the supplier. Those values are given in Table 6

<table>
<thead>
<tr>
<th></th>
<th>Blue elastomer</th>
<th>Red elastomer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transl. stiffness</td>
<td></td>
<td></td>
</tr>
<tr>
<td>x direction</td>
<td>39200</td>
<td>705600</td>
</tr>
<tr>
<td>y direction</td>
<td>39200</td>
<td>705600</td>
</tr>
<tr>
<td>z direction</td>
<td>311376</td>
<td>3736480</td>
</tr>
<tr>
<td>Rot. stiffness</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rot z</td>
<td>4720</td>
<td>113280</td>
</tr>
</tbody>
</table>

Table 6: Stiffnesses implemented in Abaqus for blue elastomer and red elastomer
5. Results and analysis

5.1 Modulus of elasticity

The studs and beams used for the structure were measured before assembling using a hammer impact technique, to capture their modulus of elasticity. The axial modulus of elasticity for all beams and studs used in the walls and the floor are shown in Table 7. The name for each beam and stud listed in the table corresponds to the ones shown in Figure 12. Every beam that starts with A was used in wall A and every beam that starts with B was used in wall B.

<table>
<thead>
<tr>
<th>Beam</th>
<th>Length [mm]</th>
<th>Weight [kg]</th>
<th>Density [kg/m³]</th>
<th>Axial modulus of elasticity [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>1161</td>
<td>1.9</td>
<td>529.5</td>
<td>13480</td>
</tr>
<tr>
<td>A2</td>
<td>1161</td>
<td>2.0</td>
<td>543.9</td>
<td>12870</td>
</tr>
<tr>
<td>A3</td>
<td>1161</td>
<td>1.7</td>
<td>474.0</td>
<td>12426</td>
</tr>
<tr>
<td>A4</td>
<td>1161</td>
<td>1.7</td>
<td>475.1</td>
<td>12309</td>
</tr>
<tr>
<td>A5</td>
<td>1349</td>
<td>2.0</td>
<td>479.3</td>
<td>13049</td>
</tr>
<tr>
<td>A6</td>
<td>1348</td>
<td>1.8</td>
<td>430.3</td>
<td>7456</td>
</tr>
<tr>
<td>B1</td>
<td>1160</td>
<td>2.0</td>
<td>559.0</td>
<td>12035</td>
</tr>
<tr>
<td>B2</td>
<td>1160</td>
<td>2.0</td>
<td>542.9</td>
<td>14897</td>
</tr>
<tr>
<td>B3</td>
<td>1160</td>
<td>1.6</td>
<td>436.2</td>
<td>9762</td>
</tr>
<tr>
<td>B4</td>
<td>1160</td>
<td>1.6</td>
<td>446.0</td>
<td>9932</td>
</tr>
<tr>
<td>B5</td>
<td>1348</td>
<td>2.1</td>
<td>491.1</td>
<td>11603</td>
</tr>
<tr>
<td>B6</td>
<td>1348</td>
<td>1.9</td>
<td>451.6</td>
<td>10672</td>
</tr>
<tr>
<td>C1</td>
<td>1200</td>
<td>3.9</td>
<td>498.6</td>
<td>10619</td>
</tr>
<tr>
<td>C2</td>
<td>1198</td>
<td>3.2</td>
<td>417.2</td>
<td>9908</td>
</tr>
<tr>
<td>C3</td>
<td>2072</td>
<td>6.4</td>
<td>477.1</td>
<td>11976</td>
</tr>
<tr>
<td>C4</td>
<td>2072</td>
<td>5.9</td>
<td>434.0</td>
<td>9909</td>
</tr>
<tr>
<td>C5</td>
<td>2071</td>
<td>6.7</td>
<td>508.1</td>
<td>12261</td>
</tr>
<tr>
<td>C6</td>
<td>1200</td>
<td>3.8</td>
<td>496.6</td>
<td>11442</td>
</tr>
<tr>
<td>C7</td>
<td>1198</td>
<td>4.1</td>
<td>535.0</td>
<td>13072</td>
</tr>
</tbody>
</table>

As shown, the modulus of elasticity varies a lot with values from 7456 to 14897 MPa. The modulus of elasticity for the beams and studs in wall A were in average higher (11932 MPa) than for the ones used in wall B (11484 MPa).

Figure 25 shows the density in function of the modulus of elasticity for the elements in the structure.
The pink dot represent the parts used as studs and the green dots parts used as top and bottom plates in the walls. The blue dots represent parts used as floor joists and the red dots parts used in the short ends of the floor.

5.2 Compression tests on elastomers

It is shown in Figure 26 and in Figure 27 the results of the compression tests on the red and blue elastomers.
Both graphs show that performing static test on those elastomers does not give a constant value as already experienced by other researchers (Bolmsvik & Linderholt, 2015). Even if the deflection curve of the red elastomer is much more constant, the difference between stiffnesses is too large therefore the supplier’s values were used in the FE model in this study.
5.3 Exciting different parts of the structure

The shaker were placed on the floor and in order to catch more modes on the walls the shaker was also placed sideways, exciting wall A. The eigenfrequencies and damping ratio found for the combined setups are shown in Table 8. To make sure that the mode shapes captured were different, an auto-MAC were extracted from LMS and can be seen in Appendix 2.

Table 8: Eigenfrequencies and damping ratio found when combining the different excitations

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Damping [%]</th>
<th>Excitation position</th>
<th>Frequency [Hz]</th>
<th>Damping [%]</th>
<th>Excitation position</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.7</td>
<td>2.3</td>
<td>Wall</td>
<td>74.8</td>
<td>0.9</td>
<td>Wall</td>
</tr>
<tr>
<td>20.7</td>
<td>12.5</td>
<td>Floor</td>
<td>86.3</td>
<td>2.3</td>
<td>Both</td>
</tr>
<tr>
<td>23.7</td>
<td>1.0</td>
<td>Wall</td>
<td>94.1</td>
<td>1.0</td>
<td>Wall</td>
</tr>
<tr>
<td>26.3</td>
<td>9.2</td>
<td>Floor</td>
<td>103.5</td>
<td>0.4</td>
<td>Wall</td>
</tr>
<tr>
<td>33.7</td>
<td>6.6</td>
<td>Wall</td>
<td>116.3</td>
<td>1.0</td>
<td>Floor</td>
</tr>
<tr>
<td>45.3</td>
<td>1.2</td>
<td>Both</td>
<td>123.9</td>
<td>0.7</td>
<td>Wall</td>
</tr>
<tr>
<td>60.1</td>
<td>1.1</td>
<td>Both</td>
<td>133.5</td>
<td>0.3</td>
<td>Wall</td>
</tr>
<tr>
<td>68.8</td>
<td>1.1</td>
<td>Wall</td>
<td>138.9</td>
<td>0.2</td>
<td>Wall</td>
</tr>
<tr>
<td>73.4</td>
<td>2.4</td>
<td>Both</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Adding a run with the shaker exciting the wall made it possible to capture more eigenfrequencies. The first mode shape captured when exciting the wall is a rigid body mode that occurred at 8.7 Hz, see Figure 28a. This mode shape was not captured when exciting the floor. The first mode shape captured when exciting the floor is also a rigid body mode which occurred at 20.7 Hz, see Figure 28b. The right column in Table 8 shows which part of the structure that was excited when capturing the mode shape.

![Figure 28: The first mode shapes where a) is the first captured when exciting the wall and b) is the first captured when exciting the floor.](image-url)
The modes at lower frequencies are in average higher damped than the higher eigenfrequencies, which is consistent with the discoveries made by researchers such as (Bolmsvik & Brandt, 2013). Furthermore, the damping ratios for the modes when exciting the floor were higher than the ones for the wall.

The bending modes for the walls were captured both when exciting the wall and the floor. For the setup with the shaker on the wall the first bending mode that was captured for wall A occurred at 68.7 Hz, see Figure 29. Corresponding mode shape for wall B occurred at 74.8 Hz. The fact that the mode shape occurred at a lower frequency for wall A was not an expected result since the modulus of elasticity in average was higher for wall A. As mentioned earlier (see section 3.1) the stiffness is one of the parameters that affects the eigenfrequency (Jarnerö, et al., 2015). A higher stiffness should result in a higher eigenfrequency (Craig & Kurdila, 2006). An explanation to the result may be that the connections were stiffer for wall B. This is a parameter that is hard to measure. Even though the assembling of the structure and thereby all connections were done in the same way and by the same person, there still may be a difference in the workmanship.

5.4 Using different mass loads on top of the structure

To investigate how mass loading affect the structure’s dynamic behavior, the assembly without pre-load were compared with two different added masses. The measurements were done using the configuration with the excitation on the floor. The extracted eigenfrequencies and damping ratios are listed in Table 9.
Table 9: Eigenfrequencies and damping ratios found when adding different mass loads on top of the structure.

<table>
<thead>
<tr>
<th></th>
<th>No load</th>
<th>Mass load 1</th>
<th>Mass load 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Frequency [Hz]</td>
<td>Damping [%]</td>
<td>Frequency [Hz]</td>
</tr>
<tr>
<td>20.9</td>
<td>12.3</td>
<td>18.8</td>
<td>8.1</td>
</tr>
<tr>
<td>26.1</td>
<td>9.3</td>
<td>35.5</td>
<td>1.4</td>
</tr>
<tr>
<td>33.5</td>
<td>3.1</td>
<td>44.2</td>
<td>0.4</td>
</tr>
<tr>
<td>47.2</td>
<td>7.3</td>
<td>56.4</td>
<td>1.1</td>
</tr>
<tr>
<td>71.0</td>
<td>0.8</td>
<td>72.1</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>73.2</strong></td>
<td>2.4</td>
<td><strong>77.1</strong></td>
<td>2.6</td>
</tr>
<tr>
<td><strong>86.1</strong></td>
<td><strong>2.4</strong></td>
<td><strong>84.4</strong></td>
<td><strong>1.4</strong></td>
</tr>
<tr>
<td>92.9</td>
<td>2.4</td>
<td>115.0</td>
<td>1.2</td>
</tr>
<tr>
<td>115.9</td>
<td>1.5</td>
<td>122.7</td>
<td>1.0</td>
</tr>
<tr>
<td>135.3</td>
<td>0.9</td>
<td>133.6</td>
<td>1.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>140.5</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

For the setup with no extra mass load added four mode shapes were identified below 50 Hz. This is one more than for the cases with added mass load. This is due to that the rigid body mode shapes of the floor were not captured when having extra mass load on top of the structure. The loads were placed close to the edges resting on the walls, therefore the added masses changes the boundary conditions making it more and more clamped for each mass loading. Clamped boundary condition does not permit rigid body modes. The mode shapes with much movement of the points not pressed by the masses, such as bending modes were easier to find.

The extra mass load had also an impact on the damping ratio for the mode shapes below 50 Hz; decreasing from a mean value of 8.0 % for the unloaded case to 3.3 % for the both cases with added mass load. This may be due to that none of the mode shapes with added mass load show much motion at edges with the elastomers. This is consistent with the discoveries made by Bolmsvik & Brandt, 2013 who found that the damping ratio were larger for the modshapes with much movements where the elastomers were placed. However Jarnerö et. al. (2015) registered, during in situ tests, that the damping increased when measuring on the same floor several times during different steps of the erection of a multistory building, consequently with added mass each time.

The different colors in Table 9 correspond to two mode shapes on the floor that was found in all sets, namely the second and third bending modes. The yellow mark corresponds to the second bending mode (see Figure 30a) and the green marks corresponds to the third bending mode, see Figure 30b.
To easily visualize this, the FRFs for two points on the floor were plotted. Figure 31 shows the FRF for a point in the middle of the floor. The vertical lined indicate the second bending mode for each setup. The red curve correspond to a mass load of the floor by itself (76 kg), the green curve correspond to mass load 1 (162 kg) and the blue curve correspond to mass load 2 (316 kg).

As shown in the FRFs the bending mode for the lightest mass load occurred at the lowest frequency closely followed by the second lightest. The mode shape for the heaviest mass load tested occurred at a much higher frequency.

Figure 32 shows the FRF for another point in the middle of the floor with the third bending mode for each setup marked. The red curve corresponds to a mass load of the floor by itself (76 kg), the green curve corresponds to
mass load 1 (162 kg) and the blue curve corresponds to mass load 2 (316 kg).

Figure 32: The third bending mode for the floor where the red curve is for no load, the green curve corresponds to mass load 1 and the blue to mass load 2.

The orders of the eigenfrequencies are almost the reverse from previous mode shape (see Figure 31). The setups with the added mass loads have a mode shape at lower frequencies than the setup with the weight of the floor as only mass load.

To examine if the placement of the mass load had an influence, the mass load was moved roughly 5 cm closer to the middle of the floor. The result shows that the FRFs for each point varied between the two different measurements, see Figure 33.

Figure 33: Comparison of FRFs for one point on the floor when placing the mass load differently.
The result indicates that where the mass load is placed on the floor structure changes the vibrational behavior of the structure.

5.5 Using different boundary conditions under the structure

The eigenfrequencies and damping ratio found for the measurements with the structure standing on elastomers and with shakers in two directions are shown in Table 10. To make sure that the mode shapes captured were different auto-MACs was extracted from LMS and can be seen in Appendix 2.

Table 10: Eigenfrequencies and damping ratio found when having the structure placed on elastomers.

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Damping [%]</th>
<th>Frequency [Hz]</th>
<th>Damping [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.3</td>
<td>4.6</td>
<td>74.6</td>
<td>2.5</td>
</tr>
<tr>
<td>13.6</td>
<td>5.9</td>
<td>76.7</td>
<td>0.6</td>
</tr>
<tr>
<td>16.3</td>
<td>7.2</td>
<td>84.5</td>
<td>0.8</td>
</tr>
<tr>
<td>20.5</td>
<td>5.4</td>
<td>86.8</td>
<td>2.4</td>
</tr>
<tr>
<td>23.9</td>
<td>3.3</td>
<td>90.8</td>
<td>1.2</td>
</tr>
<tr>
<td>27.9</td>
<td>1.7</td>
<td>94.3</td>
<td>3.3</td>
</tr>
<tr>
<td>33.7</td>
<td>6.4</td>
<td>98.4</td>
<td>1.1</td>
</tr>
<tr>
<td>39.7</td>
<td>4.8</td>
<td>109.2</td>
<td>0.7</td>
</tr>
<tr>
<td>57.6</td>
<td>2.3</td>
<td>121.2</td>
<td>0.5</td>
</tr>
<tr>
<td>62.1</td>
<td>0.8</td>
<td>129.1</td>
<td>0.9</td>
</tr>
<tr>
<td>71.3</td>
<td>0.8</td>
<td>137.3</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Having the structure separated from the concrete floor results in more mode shapes under 230 Hz than having the structure resting directly on the concrete floor. To compare the different mode shapes captured, for the two different setups, cross-MACs were extracted using LMS, see Figure 34.
The stacks in red and yellow are mode shapes that are similar between the different setups. Since the three parts of the structure (wall A, wall B and floor) have different properties it seems to be hard to capture the exact same mode shapes. There are always one or two parts that move differently. But when observing the visualization of the mode shapes it is possible to see that the movement of one or two of the parts is similar, even though the cross-MAC between the mode shapes is not equal to one.

The mode shapes in the cross-MAC that shows highest correlation are the ones with much movement on the floor. It seems thereby that the different boundary conditions have a larger influence on movements of the walls.

5.6 Finite element model

The results and analysis for the finite element model are divided into the two different configurations; fixed and damped structure.

5.6.1 Fixed structure

When comparing the mode shapes stemming from the experimental modal data with the mode shapes extracted in the finite element model, eight modes were identified as similar. The ones believed to be most equal is presented in Figure 35 and the rest can be seen in Appendix 3.
The first mode shape compared is a translational rigid body mode of the walls. In the mode extracted from LMS there is some extra movements of some of the points on wall A. This cannot be seen on either wall B nor in the FE-modal. The eigenfrequency is also close between the mode from LMS and the mode from the FE-modal.

The second mode compared is the second bending mode of the floor. The visualization of the mode shape in the FE-modal show a smoother bending curve since the modal has more degrees of freedom than the experimental modal. The same can be seen on the last compared mode shape in Figure 35.
To evaluate the result shown in the visualization a cross-MAC was calculated with the eight mode shapes shown in Appendix 3. The result can be seen in Figure 36.

![Cross-MAC between the mode shapes stemming from LMS and the ones from Abaqus, for the fixed structure.](image)

The cross-MAC does not show as good correlation as expected when comparing the visualization of the mode shapes. As explained earlier it was hard to capture clear modes in LMS. When one of the parts of the structure had a mode shape, the other parts also showed some movements without believed to have an eigenfrequency. This affects the cross-MAC since the FE-modal had more mode shapes with only one part moving at the time.

Apart from this, the mode shapes believed to have the best correlation (see Figure 35) also showed the highest MAC-values in the cross-MAC.

5.6.2 Damped structure

For the damped structure nine modes were considered to show similar movements when comparing the visualization of the mode shapes. Some of these mode shapes were extracted from higher frequency range, up to 220 Hz, compared with the modes for the fixed structure. The ones believed to be most equal is presented in Figure 37, the rest can be seen in Appendix 3.
Figure 37: Mode shapes showing similar motion in the damped configuration.
The mode shapes picked to the comparison are similar to the ones that showed the highest correlation in the fixed structure between the modes from LMS and FE-modal. The difference is the fourth mode shape, which was not used in the comparison for the fixed configuration. The reason is that an attempt was made to only use the eigenmodes in the low frequency range for the fixed configuration.

To evaluate the result shown in the visualization a cross-MAC was calculated with the eight mode shapes shown in Appendix 3. The result can be seen in Figure 38.

![Cross-MAC between the mode shapes stemming from LMS and the ones from Abaqus, for the damped structure.](image)

The cross-MAC shows similar result as the fixed structure even if the frequencies at which modes from Abaqus could be fitted with modes stemming from LMS were generally higher.
6. Discussion

The first experiment (microphone measurement) could have been performed by a 3D scanner in order to obtain the modulus of elasticity in three directions for every beam of the structure. This would have allowed creating a more realistic FE model. It is most likely that the mode shapes stemming from the FE model would have been closer to the ones stemming from the vibrational testing.

6.1 Method discussion

The experimental conditions of the vibrational testing were not optimum for several reasons and the following effects may had an influence on the results:

- Moving the accelerometers and the use of wax made it hard to place the accelerometers exactly at the same position.

- Some of the vibrations given by the shaker could have travelled through the concrete floor and influence the measuring in an uncontrolled way. Since there are no obvious differences between the damped and fixed case, that uncontrolled effect is not believed to have a large influence.

- A larger amount of accelerometers for such a structure combined with several other excitation points would have covered many other directions and positions. Such a setup would have certainly captured more modes shapes and thus given more details about the implication of the elastomers within wooden junctions.

The results of this thesis show the weaknesses of the FE model. Several factors are believed to be able to improve the FE model:

- Even if the elastomers are used in their static range of use, they still have a nonlinear behavior. Another way of modeling them would have been to use connectors in order to dive deeper in nonlinear elasticity in Abaqus. This way of working would have the advantage to add more realistic effects to the model but the counter part is that the material testing in rubber-like materials would have been huge to back up the use of those functions. Moreover the elastomers have a geometric nonlinearity which is evaluated by a shape factor stemming from dynamic analysis given by the supplier.

- Also, the screws and the way they were compressed and glued within every part of the structure are hardly possible to model. A possible improvement would be to model the screws in steel and tie degree of
freedom on the points of the screws instead of using a tie connection over the whole surface.

- Finally, it seems that trying to calibrate a FE-model of that complexity is too ambitious. A step approach would probably have given better results. Starting with modal analysis on a structure only composed with one floor and one wall clamped in a steel frame would probably be easier to model in Finite Element. Indeed, the extra steel elements will give mode shapes at higher frequency than the one studied. Also, material properties of steel are much easier to implement because of its isotropic. After a first run of test on this floor and wall, the gypsum can be added on the wall. Finally the third run would be performed with chipboard. This way of working allows to fit the FE-model with less parameters at a time.

6.2 Result discussion

When exciting the wall, more mode shapes were captured within the examined frequency range. This can be explained by the fact that the floor is stiffer than the walls. Another explanation may be that the number of mode shapes captured cannot exceed the number of accelerometers used on the structure. Since more accelerometers were used on the walls than on the floor this should give possibility to capture more modes.

When exciting the floor both mode shapes of the floor and the walls were captured. However the results showed that all mode shapes of the walls were not captured with this configuration. To be able to excite those modes a load in the same direction was needed. The damping ratio is higher in the low frequency which makes the peak picking in LMS more difficult and thereby the mode must be well excited to be easier to find.

Adding excitation in the direction parallel to the floor gives more information of the dynamic behavior of the structure. When only exciting the floor some of the lower eigenfrequencies will probably be missed. This may be an explanation to the problem with structure-borne sound. Even though tests in situ may show that a structure has a good resistance against low frequency impact sound, this may not be the whole truth. If not both directions are considered some low frequency modes may be missed.

The cross-MACs calculated between the data stemming from LMS and from FE model shows that it did not matter how the structure was placed. When comparing the cross-MACs between the fixed and damped structures it is clear that having the structure on elastomers does not give a better correlation between the modes from FE model and from LMS. This was not an expected result. Having the structure more free was believed to give a better correlation between the two sets of data since the boundary conditions
then are closer to the free-free boundary condition, which was believed to be easier to model in Abaqus.
7. Conclusions

Building wooden structures is an increasing market. Still there are issues to solve concerning sound transmission. This thesis focuses on the connections of wooden structures. In the connections elastomers are often used to decrease the sound transmission.

The effect of adding masses on top of the structure close to the short ends of the floor changes the boundary conditions, making it more and more clamped. In this study the effect of the damping is studied and the results shows decreasing damping ratio since there is not as much movement where the elastomers are placed as when the floor is freer to move.

Today calculations on sound transmission in wooden structures are not possible. The impact sound transmission is steered by the amount of vibration distribution. To obtain a calculation prediction tool the transmission of vibration distribution has to be possible to model. There are a lot of different parameters that influence vibrational measurements, for example how much preload placed on the elastomers and where the load is placed. This makes the task to find an analytical way to predict the vibrational behavior at low frequencies of a structure hard. When comparing the experimental result with the analytical result it is clear that more data are needed to make the FE model more realistic. In that sense the second aim of this thesis is not entirely achieved, but the analytical analysis gives a good direction to further research in modeling.
References


Appendixes

Appendix 1: Pictures from the different experiments
Appendix 2: Auto-MACs extracted from LMS
Appendix 3: Mode shapes compared between LMS and FE-modal
APPENDIX 1: Pictures from the different experiments

Preparation of the beams

Experiment microphone and analyzer
Assembling the structure
Dynamic experiment
APPENDIX 2: Auto-MACs extracted from LMS

Structure placed on the concrete floor (fixed structure)
Structure placed on the elastomers (damped structure)
APPENDIX 3: Mode shapes compared between LMS and FE-modal

Fixed structure

23.7 Hz  
33.7 Hz  
45.8 Hz

29.4 Hz  
30.7 Hz  
53.8 Hz
68.8 Hz
63.4 Hz
73.4 Hz
95.6 Hz
74.8 Hz
110.4 Hz
86.3 Hz
120.8 Hz
94.1 Hz
147.2 Hz
116.3 Hz
167.8 Hz
Damped structure

57.6 Hz

42.3 Hz

62.1 Hz

42.6 Hz

74.6 Hz

94.7 Hz
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153.5 Hz

178.3 Hz

230.0 Hz

234.7 Hz
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