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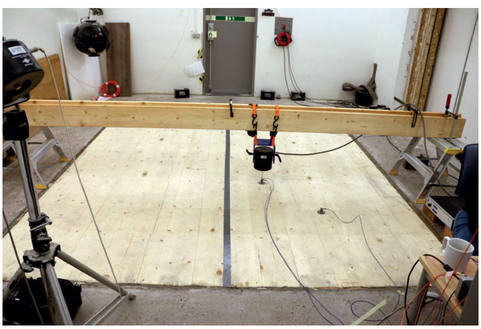
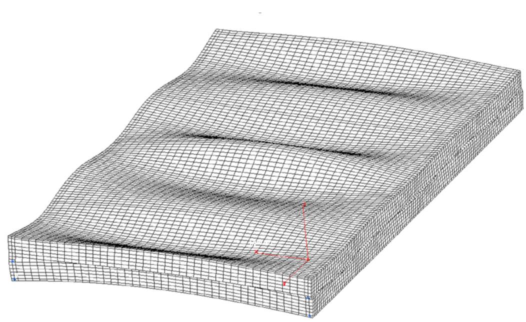
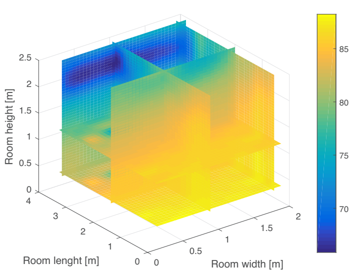
Jörgen Olsson

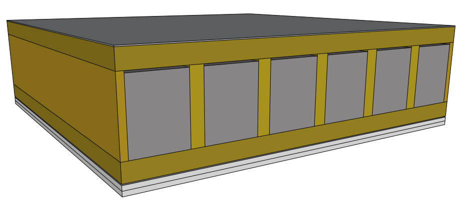
Low Frequency Impact Sound in Timber Buildings

– Transmission Measurements and Simulations









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Low Frequency Impact Sound in Timber Buildings – Transmission Measurements and Simulations
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Abstract

Olsson, Jörgen (2019). *Low Frequency Impact Sound in Timber Buildings – Transmission Measurements and Simulations,* Linnaeus University Dissertations No 364/2019, ISBN: 978-91-88898-99-9 (print), 978-91-89081-00-0 (pdf).

An increased share of multi-story buildings that have timber structures entail potential in terms of increased sustainability as well as human-friendly manufacturing and habitation. Timber buildings taller than two stories were prohibited in Europe until the 1990s due to fire regulations. In 1994, this prohibition was removed in Sweden. Thus, being a rather new sector, the multi-story timber building sector lags behind in maturity compared to the multi-story concrete sector.

The low frequency range down to 20 Hz has been shown to be important for the perception of the impact of sound in multi-story apartments with lightweight floors. This frequency range is lower than the one that has traditionally been measured according to standards and regulations. In small rooms, the measurement conditions tend to go from diffuse fields above 100 Hz to modal sound fields dominated by few resonances, below 100 Hz. These conditions lead to new challenges and to new possibilities for measurements and modelling.

In the present research, a frequency response functions (FRFs) strategy aimed to simplify simulations and correlations between the simulations and test results was used. Measurements made indicate that, in the low frequencies, the highest sound pressures occur at the floor level opposite the ceiling / floor that is excited. By having an iterative measurement strategy with several microphones and making measurements until a required standard error is achieved, it is possible to gain information about the statistical distribution of both the sound fields and floor insulation performance. It was also found that, depending on the excitation source, the FRF from an excitation point on the floor above to the sound pressure at a microphone position in the room below may differ. This indicates that non-linearities in sound transmissions are present. Thus, the excitation source used in a test should be similar in force levels and characteristics to the real excitation stemming, for instance, from a human foot fall to achieve reliable measurement results. The ISO rubber ball is an excitation source that is close to fulfilling this need. In order to obtain an FRF, the impact force must be known. A rig that enables the impact force from a rubber ball to be measured was developed and manufactured. The results show that the force spectra are the same up to about 55 Hz, regardless of the point impedances of the floors excited in the tests. Similar results have been found by others in tests with human excitations. This means that FRFs up to about 55Hz can be achieved without actually measuring the excitation force.

On the calculation side, finite element simulations based on FRFs may offer advantages. FRFs combined with the actual excitation force spectra of interest give the sound transmission. In higher frequencies, it is more important to extract the point mobilities of the floors and relate them to the excitation forces. By using an infinite shaft, sound transmission can be studied without involving reverberation time. The calculation methodology is used in the present research to evaluate different floor designs using FE models.

Keywords: Timber floors, FE-simulations, Light weight floors, Frequency response functions, Building acoustics.

Sammanfattning (in Swedish)

En ökad andel av flervåningshus med trästomme, av det totala beståndet av flervåningshus, medför potentiellt ökad hållbarhet i byggnaders livscykel samt ett användarvänligt byggande och boende. I Europa var träbyggnader högre än två våningar förbjudna fram till 1990-talet på grund av brandbestämmelser. I Sverige avlägsnades detta förbud 1994. Eftersom byggandet av flervåningshus med trästomme är en ganska ny sektor, ligger den efter i mognadsgrad jämfört med den del av byggbranschen som arbetar med flervåningshus i betong.

Lågfrekvensområdet ner till 20 Hz har visat sig vara viktigt för uppfattningen av och störning från ljud i lägenheter i flervåningshus med lätta, huvudsakligen träbaserade, bjälklag. Detta frekvensområde är lägre än det som traditionellt mäts i enlighet med standarder och förordningar. I små rum tenderar mätförhållandena att gå från diffusa ljudfält, över 100 Hz, till modala ljudfält som domineras av få resonanser, under 100 Hz. Dessa förutsättningar leder till såväl nya utmaningar som möjligheter inom mätning och modellering av stegljuds-transmission.

I det här avhandlingsarbetet användes frekvensresponsfunktioner (FRFer) med syftet att förenkla simuleringar samt korrelationer mellan simuleringar och testresultat. Mätningar som gjorts indikerar att i de låga frekvenserna uppstår det högsta ljudtrycket vid golvnivån i rummet under, mittemot golvet ovanför, där islagen görs. Genom att ha en iterativ mätstrategi med flera mikrofoner och genom att göra mätningar tills en förutbestämd standardmätosäkerhet erhålls, är det möjligt att få ut önskad precision och information om den statistiska fördelningen av både ljudfält och golvisoleringsprestanda. Det konstaterades också att beroende på excitationskällan kan FRF från en exciteringspunkt på golvet ovan till ljudtrycket vid en mikrofon i rummet nedan skilja sig åt. Detta indikerar att det finns icke-linjäriteter i ljudöverföringar. Således bör excitationskällan som används i ett test ge liknande i kraftnivåer och karaktär som den verkliga excitationen, till exempel som ett steg från en människa, för att ge pålitliga mätresultat. ISO-bollen är en exciteringskälla som är nära att tillgodose detta. För att kunna ta fram en FRF måste islagskraften vara känd. En rigg som gör det möjligt att mäta islagskraften från ISO-bollen utvecklades och tillverkades under avhandlingsarbetet. Resultaten visade att trots olikheter i punkt-impedanser för golven var kraftspektra ungefär lika upp till cirka 55 Hz. Liknande resultat har redovisats av andra forskare; då med exciteringar i form av steg från människor. Detta innebär att upp till cirka 55Hz, kan FRFer erhållas utan att mäta excitationskraften.

På beräkningssidan kan finta elementbaserade FRFer medföra fördelar. I kombination med exciteringarnas kraftspektra ger de ljudöverföringen. Vid högre frekvenser är det viktigt att mäta golvens punktmobiliteter och kombinera dem med excitationskrafterna. Genom att beräkningsmässigt använda ett oändligt långt schakt kan ljudöverföring studeras utan att efterklangstiden behöver involveras. Beräkningsmetodiken användes i avhandlingsarbetet för att utvärdera olika bjälklag och deras konstruktionsparametrar.

Nyckelord: Trägolv, FE simulering, Lättvikts golv, Frekvensresponsfunktioner, Byggakustik.

Acknowledgements

This work was carried out at the Department of Mechanical Engineering at Linnaeus University in Växjö, Sweden, and at the Department for Wood Building Technology at RISE Research Institutes of Sweden. The Ph.D. student work is mainly a part of the ProWood Industrial Graduate School supported by the KK Foundation, but it constitutes also parts of the projects Interreg Urban Tranquility, BioInnovation FBBB and Silent Timber Build. The measurement data evaluated in the first parts of the thesis came from the Interreg IV project Silent Spaces.

First of all, I would like to thank my supervisor Assistant Professor Andreas Linderholt for our good collaboration, his guidance, commitment, valuable knowledge and support throughout the research work. Secondly, I would like to express my gratitude to my co-supervisor, the late Professor Börje Nilsson, for his support and inspiring guidance. Unfortunately, he passed away during the early spring 2019.

I would also like to thank Marie Johansson, Kirsi Jarnerö, Karin Sandberg and all my other colleagues at RISE Wood Building Technology for their support, guidance and sharing of knowledge. A big thanks to Mats Almström at the Department of Mechanical Engineering at Linnaeus University for his efforts and skills in building things that we theorists cannot. I am grateful to those who ensure that our most important tools, the computers and software, work as well as possible, thanks Stefan Johansson and Micael Carlsson.

Further, I would like to express my appreciation for the collaboration and inspiring discussions with personnel at the Linnaeus University departments of Mechanical Engineering, Building Technology as well as Forestry and Wood Technology. I would also like to thank my other colleagues at RISE Built Environment and RISE Glass together with all the other Ph.D. students and the people within the ProWood organization. I think the companies involved in ProWood deserve respect for their willingness to contribute to building knowledge. A special thank you goes to my colleagues at RISE Sound and Vibration for the time we worked together; a valuable experience also for this research work. I hope for more collaboration in the future.

Furthermore, I would like to thank and express my gratitude to my parents, Eivor and Sven-Gustav and my brother Hans-Olof. I would also like to thank my friends for giving me something else to think of besides this project. Finally, I would like to sincerely thank my dear Isabel for all her support and understanding.

Jörgen Olsson
Växjö, October 2019

Appended papers

PAPER I OLSSON, J. and LINDERHOLT, A., Low-frequency impact sound pressure fields in small rooms within lightweight timber buildings - suggestions for simplified measurement procedures. Published in *Noise Control Engineering Journal,* Volume 66, Number 4, pp. 324 - 339, July-August 2018.

PAPER II OLSSON, J. and LINDERHOLT, A., Force to sound pressure frequency response measurements using a modified tapping machine on timber floor structures. Published in *Engineering Structures*, Volume 196, p. 109343, 1 October 2019.

PAPER III OLSSON, J. and LINDERHOLT, A., Measurements of low frequency impact sound frequency response functions and vibrational properties of light weight timber floors utilizing the ISO rubber ball. Submitted to *Applied Acoustics*, July, 2019.

PAPER IV OLSSON, J., LINDERHOLT, A. and Nilsson, B., *Impact evaluation of a thin hybrid wood based joist floor*. Proceedings of the International Conference on Noise and Vibration Engineering (ISMA 2016). Presented at ISMA 2016 in Leuven, Belgium in September 2016.

PAPER V OLSSON, J. and LINDERHOLT, A., Low frequency impact sound of timber floors: An FE based study of conceptual designs. Submitted to *Building Acoustics*, October 2019.

Jörgen Olsson's contribution to the appended papers

PAPER I — Olsson outlined and planned the measurements, supported by Linderholt. Olsson conducted most of the measurements with help from Linu Kuttikal Joseph. The data extraction and most of the calculations were done by Olsson. The analysis and the writing of the paper were done by Olsson together with Linderholt.

PAPER II — The idea was drawn up and discussed iteratively between Linderholt and Olsson. The measurements were made by Linderholt and Olsson. The data extraction and calculations were done mainly by Olsson. The analysis and the writing of the paper were done by Olsson together with Linderholt.

PAPER III — The idea was drawn up and discussed iteratively between Olsson and Linderholt. The measurements were made by Olsson and Linderholt. The data extraction and calculations were done mainly by Olsson. The writing of the paper was done by Olsson together with Linderholt.

PAPER IV — Olsson presented the idea and concept of the paper. Olsson designed and made finite element models of the floors. Nilsson extracted the necessary theory for the analytical calculation of sound radiation from the floors to the room / duct. Linderholt prepared the Matlab function for calculating modal-based transfer functions and the analytical sound radiation function from ceiling to duct. Olsson did the FE simulations of natural frequencies, merged the Matlab functions and did the calculations. The writing of the paper was mainly done by Olsson and Linderholt. Nilsson reviewed the paper.

PAPER V — This paper is an extension and a variation of Paper IV. The complete FE strategy and method were suggested by Linderholt. The floors were designed and FE-modelled by Olsson. The MSC Nastran simulations were done by Linderholt. The data extraction and analysis of the results were mainly done by Olsson. The writing of the paper was done by Olsson together with Linderholt.

List of symbols

B	Frequency bandwidth (Hz)
B'	Flexural rigidity of a plate (Nm)
E	Young's modulus (N/m^2)
H	General notation for a transfer function
$\mathbf{K}$	Stiffness matrix
$\boldsymbol{K}$	Modal stiffness matrix
L	Sound pressure level (dB, ref. 20µPa)
L'	Total length of the edges in the room (m)
L_x	Length in x direction (m)
L_y	Length in y direction (m)
L_z	Length in z direction (m)
$\mathbf{M}$	Mass matrix
$\boldsymbol{M}$	Modal mass matrix
N	Integer number, number of modes
N'	modal density (modes/Hz)
P	Excitation force (N) or sound pressure (Pa) amplitude
$\mathbf{P}$	Force vector
$\boldsymbol{P}$	Modal force vector
S	Surface area (m^2)
S'	Surface area of a room (m^2)
T	Reverberation time (s), 60 dB decrease, or Transmission coefficient (Pa/Pa).
$\mathbf{V}$	Viscous damping matrix
$\boldsymbol{V}$	Modal viscous damping matrix
V	Volume (m^3)
U	Response in a transfer function, here it may be structural motion such as displacement amplitude (m), velocity amplitude (m/s), acceleration amplitude (m/s^2) but also sound pressure (Pa), depending on the receiving sensor.
Y	Complex valued mobility; point mobility (m/Ns), transfer mobility (m/Ns)
Z	Complex valued impedance; specific acoustic impedance (Ns/m^3), mechanical impedance (Ns/m)
c	Speed of sound (m/s)
c_0	Speed of sound for air (m/s)
c_L	Longitudinal speed of sound in a solid structure (m/s)
f	Frequency (Hz)
f_n	Natural frequency (Hz)
f_s	Schroeder frequency (Hz)
i	Imaginary number, $i = \sqrt{-1}$

k Stiffness (N/m)
m Mass (kg)
m'' Surface Mass (kg/m^2)
n Integer, mode number
p Force (N), Pressure (Pa)
$\hat{p}$ Peak pressure (Pa)
r Ratio between excitation frequency and natural frequency
$\mathbf{u}$ Displacement vector
u Displacement (m)
$\dot{u}$ Velocity (m/s)
$\ddot{u}$ Acceleration (m/s^2)
v Viscous damping (Ns/m)
v_{cr} Critical viscous damping (Ns/m)
Ω Circular excitation frequency (radians/s)
$\mathbf{\Phi}$ Modal matrix
β Adiabatic compression modulus (N/m^2)
ζ Relative critical damping
$\boldsymbol{\eta}$ Modal coordinate vector
ρ Density (kg/m^3)
$\boldsymbol{\phi}_n$ Eigenvector / eigen mode shape
ω_n Circular natural frequency (radians/s)

Contents

1 Introduction

A small introduction is provided with the aim of describing the background and meaning of "impact sound in timber buildings" and to show how the subject fits into a wider context.

1.1 Background

Timber is a renewable material. An increased use of timber as a construction material is, due to its properties and role in nature, considered as a potential way to build a more sustainable society. Carbon dioxide is stored in wood, through photosynthesis and the carbon cycle, both as forest and in wood objects such as timber buildings. Extended use of timber in buildings may contribute to increased energy efficiency due to the resulting decrease in energy consumption during the life cycles compared to, for instance, concrete buildings (Gustavsson and Sathre, 2006, Sandanayake et al., 2018). The carbon storage capacity of wood and timber buildings may be seen as an opportunity to improve the carbon dioxide balance from the previous fossil fuel emissions caused by mankind, i.e. the well-known greenhouse effect (Mitchell, 1989). In countries such as Sweden, wood-based industries also help keep job opportunities in rural areas. In addition, timber buildings have properties that may increase safety in areas of seismic hazards (Ceccotti et al., 2013).

Due to a number of urban fires that occurred in the 19th century, fire legislation throughout Europe prohibited timber buildings taller than two stories. After technological development and improvements in fire protection, tall timber buildings were eventually considered to be safe. In Sweden, the fire legislation was revised in 1994, whereby timber buildings more than two stories high became permitted. Today, most West European countries have revised their legislation and allow five or more stories (Östman et al., 2010).

Examples of motivators for developing multi-story buildings are urbanization and higher prices for land in cities and also visions of densifying cities for increased walkability and effective transportation by bicycling, walking and public transport in order to enable a good life without, or with less need for, cars.

Due to the previously mentioned positive aspects of wood, there is an increasing interest in developing multi-story timber-based buildings. As of 2019, the tallest timber-based building, Mjøstårnet in Brumunddal, Norway, had reached a record of 18 stories and 85.4 meters in height (Abrahamsen, 2018), see Figure 1. Timber buildings of similar height are underway or planned, for instance, the Sara Kulturhus in Skellefteå, Sweden, which is planned to consist of 20 stories.

Figure 1. Mjøstårnet in Brumunddal, Norway.

High-rise timber buildings have been found to imply some technical challenges, not the least within the areas of structural dynamics and acoustics. Timber has a high strength-to-weight ratio and a rather high stiffness-to-weight ratio along its fiber direction. Due to the high strength, there will not be a need for big masses in the buildings. The same properties, i.e. low weight (in relation to strength), which are beneficial for seismic safety also lead to less desired effects when it comes to wind loadings. For medium to high-rise buildings, the dynamic properties of the buildings make them more wind sensitive if they are designed with a timber frame (Johansson et al., 2015), if not special measures for improving this are added. This issue could be resolved if a certain extent of other materials were allowed, such as concrete in stabilizing walls or elevator shafts, such as is used in the 18-story tall timber building Brock commons at the University of British Columbia, Canada, which was finished in 2017. Research with the aim to minimize the use of concrete and other materials that have more negative impacts on the environment, in favor of timber in high rise buildings, is ongoing (Johansson et al., 2016).

Disturbance from vibrations stemming from activities, such as walking, running, etc., is and has been a challenge (Jarnerö, 2014). This issue seems to be resolved by having a proper stiffness of the floor systems. At the time of writing, the Swedish national annex to the Eurocode allows a maximum deflection of 1.5 mm/kN for a point load; this limit is under examination. For instance, in Finland, the static deflection limit is 0.5 mm/kN on a floor.

This thesis concerns impact sound, mainly measurements and simulation techniques, within the low-frequency range for multi-story timber buildings. After revising the regulations, and thereby allowing multi-story timber buildings, it became eventually a well-known issue that footfall noise can be unsatisfactory despite that formal impact sound requirements have been fulfilled.

It has been shown that impact sound measurement data give low correlation to the satisfaction of footfall noise for the residents of multi-story timber buildings, compared to the equivalent in multi-story concrete buildings (Östman et al., 2008). This initiated, among others, the AkuLite research project in Sweden with the goal to find the causes of poor correlations. The research outcome is that the measuring frequency range 50 – 3150 Hz and the weighting method according to the standard of impact sound measurements at that time (ISO 140-7, 1998, ISO 717-2, 1996) were not sufficient for timber buildings. By extending the frequency range down to 20 Hz, it was shown that the impact sound correlation to subjective ratings could be improved significantly (Späh et al, 2013, Ljunggen et. al, 2014, Ljunggen et. al, 2017). This resulted in a revision of the Swedish sound class rating / requirement standard SS 25267 (2015), which now recommends including the extended range, down to 20 Hz, for the highest sound classes, A and B. The measurement method used for the SS 25267:2015 rating is ISO 16283-2 standard, which has replaced the ISO 140-7 (1998) standard. It should be noted that the 16283-2:2018 standard describes the measurement procedure for the frequency range 50 – 5000 Hz, i.e. the lowest frequency is a bit higher than 20 Hz.

Even though knowledge and methods are getting better concerning correlation of measurements to subjective perception in the low frequency range, there remain obstacles for the building industry regarding impact sound. Timber building companies commonly have concerns regarding design parameters and methods for how to achieve cost-effective solutions to impact sound within the low frequencies. Another aspect is that impact sound insulation in some building systems tends to have a rather large variation (Öqvist et al., 2012). This is an important aspect since a potentially wider distribution in sound insulation requires a greater margin than the average sound insulation, which also influences building costs.

Regarding product development, in many industries (notably the vehicle, aerospace and maritime) there are endeavors for increasing the share of simulations and decreasing the amount of testing in the development of new products. This is also reflected in the amount of simulation software companies that have appeared in recent last decades (Ansys Inc., Altair Engineering, ESI group, MSC software, Dassault Systémes, MathWorks, etc.). The main purpose is to decrease the need for prototypes since building and testing them tend to be expensive. This possibility / potential should also be valid for the building industry when it comes to developing new building systems and floor systems. Ideally, numerical simulations can save costs, speed up development and decrease the risk and effort of testing new radical designs.

Impact sound measurements are, according to the previously mentioned standards, mainly made with excitations using tapping machines (ISO 10140-3+A1, 2015, ISO 10140-5, 2010, ISO 16283-2, 2018). Although progress has been made by researchers to more accurately simulate the tapping machine (Rabold et al., 2010, Qian et al. 2019), thereby enabling numerical simulations of impact tests using that device, there remains a lack of implementation in the building industry.

Research related to various acoustic aspects of timber buildings has been extensive in recent years and has resulted in several Ph.D. theses. Worth mentioning are, for instance:

- Negreira (2016) has studied and simulated sound transmission performance and the parameters of timber buildings and components. Examples are elastomers and sound transmission in timber buildings with Finite Element analyses. Negreira has also conducted studies of the excitation forces of the tapping machine and the perception and annoyance of vibrations on floors.

- Öqvist (2017) has emphasized the statistical and precision aspects of the sound insulation of timber construction, the precision of measurement methods and also correlations of subjective perception of different objective descriptors. That research deals with the quality we have in the overall chain of sound insulation, from construction, to measurements, and to the correlation between subjective satisfaction and measurement results.

- Hagberg (2018) has been working with the management of acoustics in lightweight buildings and how to achieve a design process that makes a building fulfill requirements and user expectations. Four main points are addressed in the research: (1) sound insulation descriptors, (2) targets to strive for, (3) how to predict sound insulation and (4) the risk for acoustic failure during the erection of a building. This type of knowledge is valuable, especially for those already working in the building sector and who want to enter the timber building sector.

- Amiryarahmadi (2019) has developed a virtual design studio for low-frequency sound from walking in lightweight buildings. It contains a method for measuring forces caused by walking and a mathematical model for simulations of impact sound that a neighboring apartment could hear. The research also contains listening tests from a studio designed as an apartment but equipped with speakers in the ceiling. The results are interesting and show, for instance, deviations in the perception of impact characteristics, depending on the type of building (concrete or timber) the listener lives in.

1.2 Research questions and aim

The development of buildings involves important acoustical issues. The present development is, in practice, dominated by measurements of acoustic performance. Most of the methods used are based on the diffuse field theory, which requires a higher modal density than the one that exists in the modal range (the low-frequency region, where the modes are well separated). For a timber building, the problem area has been found to be in a lower frequency range than the classical standardized measurement methods were originally developed for. In the low-frequency (low modal overlap) range, deterministic methods, such as the Finite Element Method (FEM), are widely adapted and used in structural dynamics. Such methods have a low implementation rate in applied building acoustics. However, in acoustic research, there are numerous simulations of impact sound. For instance Bard et al.(2008) have made FE simulations of the structural sound attenuation of lightweight timber floors. Flodén et al. (2015) have simulated the sound transmission through cavities of lightweight timber floors. Brunskog and Hammer (2003) have treated sound transmission theoretically, using periodical lightweight floors. Rabold et al. (2008) have made accurate predictions of sound transmission and impact sound levels of a tapping machine excitation of lightweight floors using FE models. Sousa and Gibbs (2011) have developed a prediction model for the estimation of low-frequency impact sound for homogenous and floating floors, and they used FRFs for correlations. Sjökvist et al. (2008) have made a Fourier series model for vibrational response simulations of periodically stiffened light floors. Diaz-Cereceda et al. (2011) have derived and calculated impact sound transfer with analytical models of noise transmission through different structural connections. Hirakawa and Hopkins (2018) have made transient simulations of heavy impact sound with statistical energy analysis and FEM. However, there is a lack of endeavors to apply Frequency Response Functions (FRFs) in simulations as correlations with acoustical measurements in the low-frequency range, or for impact sound measurements in general, as an alternative to the current method with excitations using the ISO tapping machine. The overall question is if a measurement methodology that sets out from FRFs in combination with FE models could improve the quality of low-frequency impact sound compared to the methods used today.

Efforts to increase the number of multi-story buildings made of timber entail a need for more knowledge and better utilization of methods in structural dynamics and acoustics within the building industry. The research questions in this thesis are:

1. What is the nature of low-frequency impact sound distribution in lightweight timber buildings? This question is especially valid for small rooms where the modal range, in which eigenmodes are well

separated, becomes a dominating part of the measurement range. The objective is to measure the sound within the modal range and even the range below that correctly. This knowledge is important to improve measurement methods and standards. The potential influence of excitation characteristics is also a variable of interest.

2. A Frequency Response Functions (FRFs) strategy can provide a common basis for simulations, measurements and correlations between them. The question is whether or not aiming for FRFs is practically feasible in field measurements in buildings. To develop a methodology that can be accepted by and applied in the building industry is an important issue. Traditionally, there have been limited possibilities to make comparisons between impact sound measurement data and results from FE calculations. The reason is that, although the tapping machine used for impact sound measurements is well specified, it is difficult to predict its force spectra for different floors. This is an obstacle to validated simulations of impact sound performance. The hypothesis is that by moving towards an FRF approach for both calculations and measurements, especially at low frequencies, this obstacle for correlations between simulations and measurements can be removed.

3. Is the FEM, together with an FRF-based approach, a useful tool for calculating low-frequency impact sound in lightweight timber buildings? In classical mid- and high-frequency range acoustics, the FEM is not considered to be an efficient tool, and Statistical Energy Analyses are considered to be more computationally effective in these ranges. However, in lightweight timber buildings, the area of interest is the low-frequency, modal, range. The vision here is to be able to simulate the sound levels that persons in the room below experience from the impact of a heel on the timber floor above.

The purpose and aim of the research are to obtain tools and methods that simplifies computer simulations of low frequency vibroacoustic transmission and correlations to measurements of lightweight wooden floors with connecting rooms.

2 Low-frequency impact sound and acoustics

This thesis in purely technical. However, all the reasons for this research is due to that humans are exposed to impact sounds and may affected by it buildings. An introduction to human perception is provided in this chapter. Also an introduction to acoustic measurements is made since Paper I – III deals with measurements. Some fundamental information are presented concerning the nature of low frequency modal sound fields in small rooms. This since lightweight multi-storey buildings tend to have a larger proportion of impact sound transmission in the low frequency range, which means different measurement conditions, especially in small rooms, compared to classic heavier concrete floor buildings.

2.1 Human perception

Low-frequency sound in buildings involves the lowest hearing range that can be perceived by humans. The lower limit for human tonal hearing, i.e. the perceiving of a sine wave of sound pressure as a tone, is commonly considered to be around 20 Hz. However, research has shown that humans can perceive sound pressures below 20 Hz (Møller, and Pedersen, 2004).

Tonal hearing perception is defined in the ISO 226 standard (2003). The curves in this standard are refined from the classical hearing curves of Fletcher Munson (Fletcher and Munson, 1933), see Figure 2 showing the equal loudness perception levels throughout the human tonal hearing range. The A-weighting filter, commonly used in acoustics for rating of disturbances and noise, is based on the 40-phone curve of the ISO 226 standard. The ISO 226:2003 equal loudness curves show that the tonal hearing limits are higher for sound pressures at lower frequencies than the limits in the frequency range where humans have their best perception of low frequency sound pressure levels (1000 - 5000 Hz). After exceeding the hearing threshold, the equal loudness curves are denser in the 20-50 Hz range than in the 1000 Hz range.

The results from the AkuLite project (Ljunggren et al. 2015) show that the required weighting for a transient impact sound disturbance is not met by the classical A-weighting curve (IEC 61672, 2013) nor the classically used curve of reference values for impact sound weighting (ISO 717-2, 2013). A requirement for the tonal hearing tests, according to ISO 226, is that the duration of the tones are at least one second. If the duration is shorter, people will perceive the sound as less loud. Walking or running causes transient sounds, normally shorter than one second (Amiryarahmadi, 2019). Together, this indicates that the equal-loudness tonal curves of ISO 226:2003 are not very well suited for the transient character of impact sound.

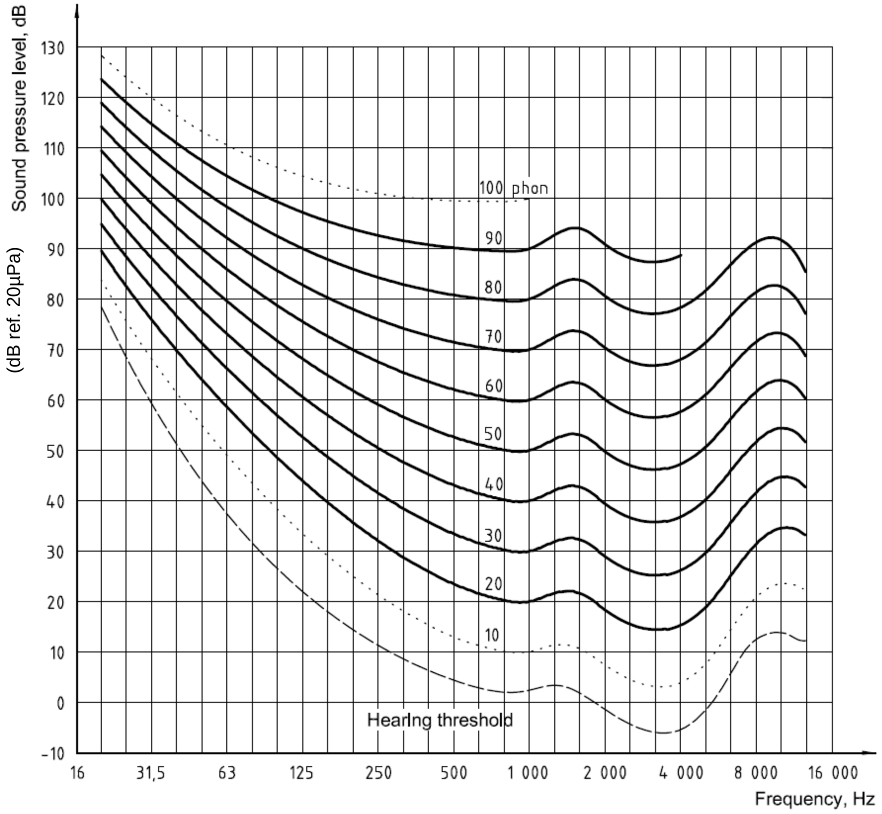

Figure 2. The ISO 226:2003 Equal-loudness curves, which are refined for the Fletcher Munson hearing curves. Equal loudness means a constant human subjective perception of the same loudness of pure continuous tones.

It should also be noted that the hearing perception of tones is different from the perception of random noise. Hearing curves that better correlate with short click sounds and bursts of random noise are defined in the ITU-R 468 recommendations, see Figure 3. The character of the sound, if it is tonal, transient or random, and which pressure level it has, affect the perception of the sound within the low-frequency range. Hearing curves for transient sounds, as counterparts to the ISO 226 equal-loudness or the ITU-R 468, have not yet been found. However, progress has been made in the weighting of excitation devices in relation to subjective perception in the low-frequency range (Ljunggren et al. 2017).

Another interesting aspect, since lightweight buildings are also more sensitive to vibrations caused by walking, is that persons exposed to vibration that correlates with the sound are significantly more annoyed than persons who are only exposed to the same noise level (Lee and Griffin, 2013). This phenomenon has been observed for people in housing exposed to railway noise.

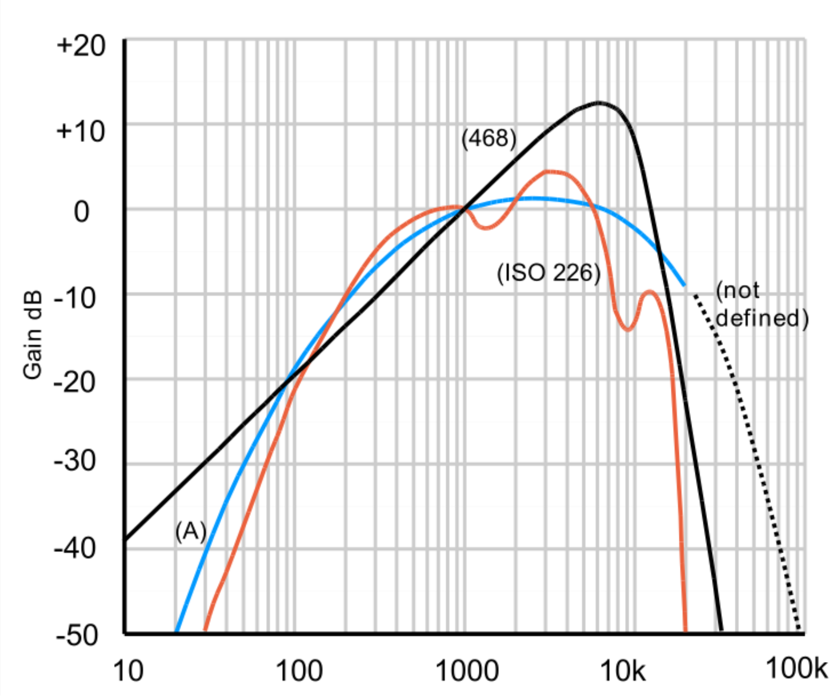

Figure 3. A comparison between the ITU-R (black), A-weighting (blue) and the inverse ISO 226 40-phone curve (red).

2.2 Measurements

The low-frequency range implies some differences in buildings compared to the more traditionally used range for impact sound (>100 Hz). The frequencies down to 20 Hz constitute a range with low modal density; this is especially true within small rooms. Traditional building acoustics concerning airborne sound insulation and impact sound are based on diffuse field theory. This is essentially a statistical approach that is assumed to give repeatable values by using a number of random measurement locations in a room with a sufficient modal overlap. In room acoustics, the diffuse field assumption implies the following methodology for measuring the sound pressure in a room:

- Measure the sound pressures in a sufficient number of locations within the room for a certain amount of time for each measurement. The purpose is to get a sufficiently low standard error of the total average value to fulfil a wished for or standardized accuracy in repeatability.

- Transform the measurement results into the frequency domain and present them integrated over a certain frequency range, such as an octave, or more commonly, a 1/3 octave band (commonly referred to as third octave). The desired value is the average of the energy from all the measurements. The expression for averaging sound pressure levels in decibels (dB) is:

  $$L = 10 \cdot lg\left(\frac{1}{N}\sum_{j=1}^{N} 10^{L_j/10}\right) \tag{1}$$

  where L_j, j=1, 2, …, N, denotes the sound pressure level at N different positions in the room and lg is the common, or decadic, logarithm.

- Measurements made too close to walls, floors and ceiling are not to be used in the calculation of a sound pressure level average. Hence, the measurements have to be conducted at certain distances, given by regulations. These constraints are due to the increase in sound pressure close to hard surfaces where the sound is reflected. Thus, the inclusion of measurements close to these objects renders in an increased standard deviation and thereby decreased accuracy in the average value.

- Determine the damping of the system by measuring the reverberation time in the room for each octave or 1/3 octave band. This is done in order to capture the influence of the absorption caused by, for instance, the furnishings in the room. The measured sound levels are transformed to the specified reference reverberation time (0.5 seconds) in order to obtain values suitable for comparison and requirements.

- Before conducting a measurement, make sure that the background noise is sufficiently low in order to not affect the overall test data. Correct the measured levels in the frequency bands (usually 1/3 octaves) where the background noise has influenced the results. Discard the test data that have too high measurement uncertainties to be able to be corrected.

The measurement standards are detailed and use refined techniques, but they are not described here. An example of this methodology is the standard for impact sound insulation measurements in laboratories, ISO 10140-3 (2010). The correction for background noise is not seen as specific for the diffuse field methodology. Steps towards taking the modal characteristics into account have, however, been taken, for instance in 16283-2 (C. Hopkins, and P. Turner, 2005). According to this standard, corner values should be taken into account for low-frequency measurements. This is because the highest sound pressure values usually occur in corners. This is a deviation from a pure diffuse field approach since the maximum of the values from fixed measurement points is used instead of only a statistical number of averages. This shows that diffuse field theory can be combined with a modal approach in order to obtain better quality in measurement results.

The most common device as the source of excitations for measurements of impact sound is the tapping machine (ISO 10140-5, 2010 ISO, 16283-2, 2018), see Figure 4. It has five metal hammers that weigh 0.5 kg each. Each hammer falls on the floor two times per second, i.e. ten impacts per second for all the hammers together. The distance between pairs of hammer centerlines is 100 mm, and the dropping height is 40 mm. The tapping machine has several benefits: it is statistically efficient with five excitation points for each measurement setup; it is also easy to operate; and it excites a wide frequency range. However, for excitation in the low-frequency range, the ISO rubber ball (commonly called the rubber ball and sometimes the Japanese impact ball), also shown in Figure 4, has become more of an alternative in measurement standards in recent years (ISO 10140-5, 2010, ISO 16283-2, 2018). An advantage of using the impact ball in field measurements is its higher signal-to-noise ratio (SNR) in the low-frequency range compared to the SNR for the ISO tapping machine (Homb, 2005, Olsson et al. 2012). Also, the rubber ball´s excitation characteristics are more similar to excitations made by a human footfall than those of the ISO tapping machine and other compared devices, see Figure 5 and Figure 6 (Homb, 2005, Jeon et al., 2006, Späh et al., 2013). The disadvantage is the impact ball´s lower SNR in the mid- to high-frequency range (>200 Hz), in which the ISO tapping machine performs better (Homb, 2005, Olsson et al. 2012).

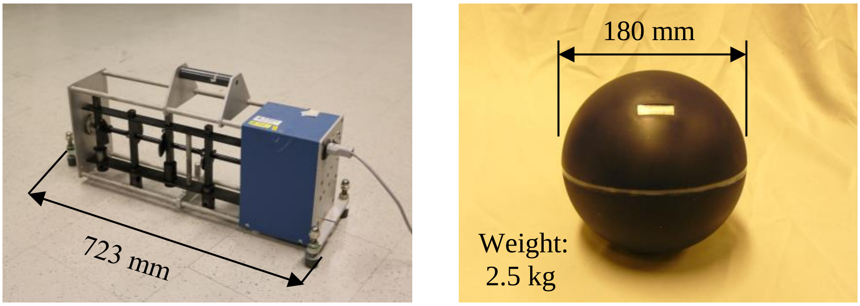

Figure 4. Photo on the left, the ISO tapping machine. On the right, the ISO rubber ball.

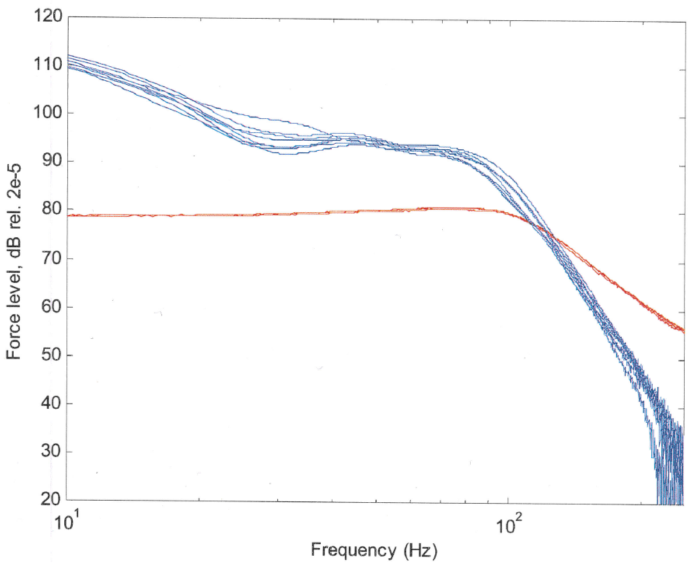

Figure 5. Impact forces as functions of frequency from Homb (2005). The blue curves represent repeated heel impacts, and the red curves are repeated impacts from one ISO tapping machine hammer.

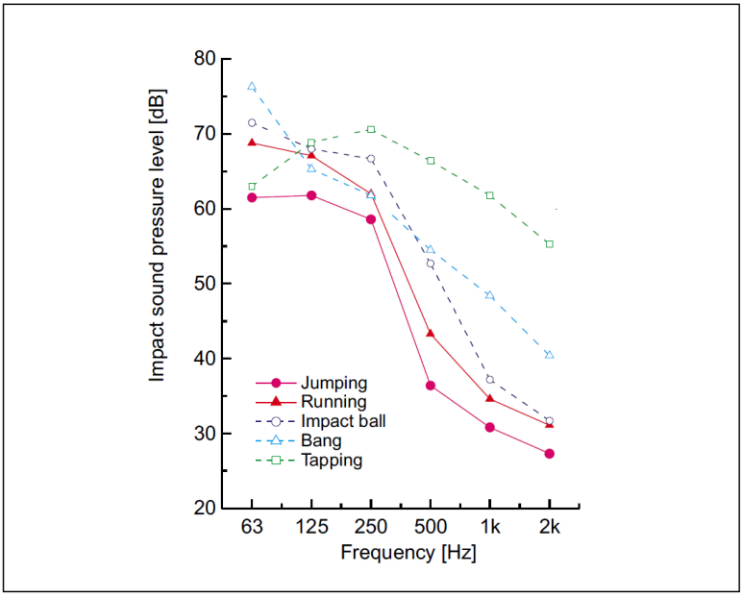

Figure 6. Frequency characteristics of real impact sounds generated by a 26-kg child and by standard impactors from Jeon et al. (2006).

2.3 Low-frequency sound fields in small rooms

The statistical diffuse field approach performs less efficiently in the low frequency range, the low-modal overlap range, since a few modes dominate the sound distribution. The sound in the room gets a character that is similar to these modes. Room modes, i.e. standing waves, have peaks and nodes; peak values always occur at the hard reflecting surfaces of the walls, see Figure 7. This is governed by the impedance differences between air and the walls.
The natural frequencies, f_n (Hz), of room modes (axial, oblique and tangential) for a rectangular room can be calculated using the following formula (Bodén et al., 2001),

$$f_n = \frac{c_0}{2}\sqrt{\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2}, \qquad (2)$$

where c_0 is the speed of sound in air (m/s), n_x, n_y and n_z are the order of the mode in the room and L_x, L_y, L_z, are the length, width and height respectively.

The exact limit between the modal and diffuse field ranges is a bit vague and sometimes still debated (Skålevik, 2011). A commonly stated limit to room acoustics is defined by Schroeder and Kutruff (1962) as;

$$f_s = 2000 \cdot \left(\frac{T}{V}\right)^{0.5} \tag{3}$$

in which f_s is the Schroeder frequency in Hz, T is the reverberation time, in seconds, for the sound to decay 60 dB, and V is the volume of the room in m^3. This limit corresponds to a threefold overlap, i.e. three modes per half power bandwidth for a frequency response function (FRF). The half power bandwidth is described in Figure 8. The half-power bandwidth can be calculated with the formula (Skålevik, 2011),

$$B = \frac{ln10^6}{2\pi T} \approx \frac{2.2}{T}. \tag{4}$$

It is necessary to be able to calculate the modal density in order to obtain information about the modal overlap. In diffuse fields, the modal density is defined as

$$N'(f) = \frac{dN}{df}, \tag{5}$$

where N is the mode number. A statistical modal density can be calculated using the formula (Bodén et al. 2001),

$$N'(f) = \frac{4\pi f^2 V}{c_0^3} + \frac{\pi f S'}{2c_0^2} + \frac{L'}{8c_0}, \tag{6}$$

where f is the band center frequency (Hz), V is the room volume (m^3), S' is the total surface area of the room (m^2) and L' is the total length of all the edges (m). By integrating this formula over a frequency range, the statistical number of modes within the range can be estimated. By taking the inverse of this function, the statistical frequency separation of the modes can be calculated. In the low-frequency range, this will be a coarse tool since the room mode distribution is discrete and not as smooth as this function indicates. Adding modes calculated using Equation (2) is more precise in this range.

Below the first room mode, the sound pressure in a room becomes more evenly distributed as the frequency gets lower and, thereby, further from the first eigenmode. The distribution of sound pressure then gets closer to a static

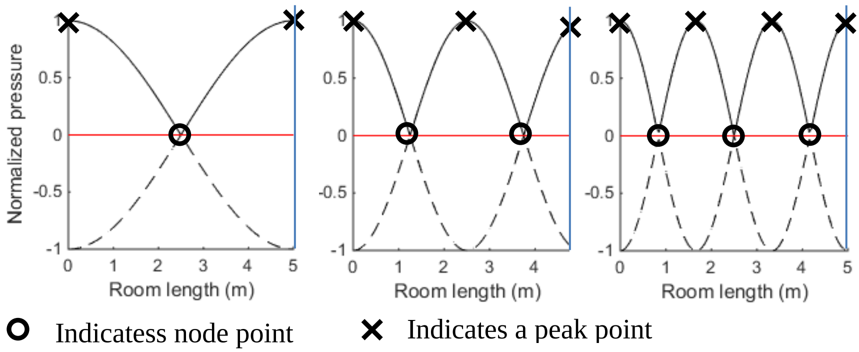

Figure 7. An example showing the pressure distribution for the first three modes between two opposing parallel, hard-surface walls in a room. The y-axis indicates sound pressure, and the x-axis indicates the distance throughout the room between the walls. The sound pressure is constant and in theory equal to zero at the nodes. The highest sound pressure levels occur at the peaks.

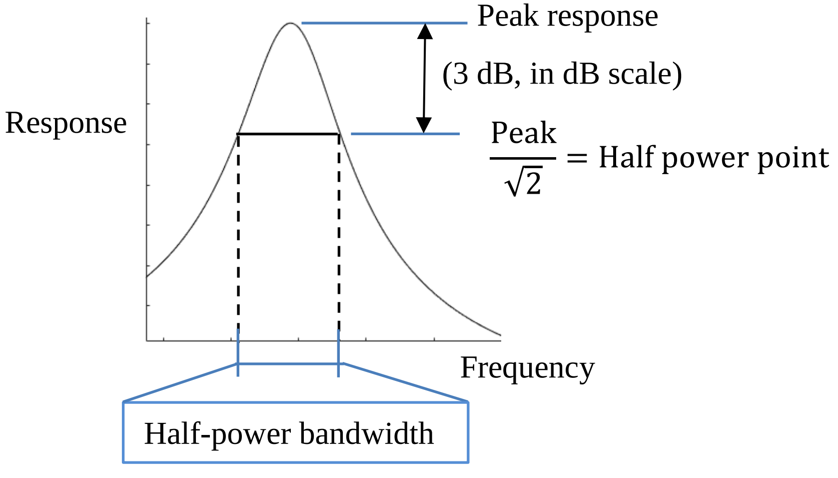

Figure 8. A frequency response function, i.e. a harmonic response per unit of a harmonic excitation as a function of frequency. Figure reproduced from Craig (1995) and Bodén et al. (2001). In acoustics, the response is commonly sound pressure (Pa), and the excitation for impact sound is a force (N). The peaks in the plot that define the half-power bandwidth are centered around a resonance. The relative frequency distance between the points below and above the resonance where the power is half the power of that at the resonance peak, defines the half-power bandwidth.

character, i.e. the pressure variation in the room becomes lower in this range (zero modes) compared to the variation within the modal range (1 – 3 modes per half-power bandwidth). Using Equation (2) for a small room, for example an office room, with 4 m in length, 2.25 m in width and 2.7 m in height gives the first mode at 41.7 Hz. There are few modes in the lowest frequency range in small rooms if 20 Hz is set as the lower limit for impact sound measurements. The highest sound pressure level may even occur below the first room mode (shown for instance in Paper I in this thesis). Also, in a room with the previous dimensions, the Schroeder frequency is around 284 Hz for a reverberation time of 0.5 s. A plot of the number of modes in the room is shown in Figure 9. The area of the room is about 9.2 m^2, i.e. normal size for single-occupancy office rooms and also a normal size for single-occupancy bedrooms in apartments. Skålevik (2011) debate that above the Schroeder frequency it is diffuse fields. However, lower than the Schroeder frequency there is no distinct limit between diffuse and modal sound fields. It may be defined as a cross over range. Skålevik mention 0.45 times the Schroder frequency as a possible limit where the modal range ends, based on Schorder´s own assumptions. For this small room it would imply around 128 Hz. When having increased noise in low frequencies this shows the significance of using a modal approach and having an understanding of the consequences of including a lower frequency range than the diffuse field approach is intended for.

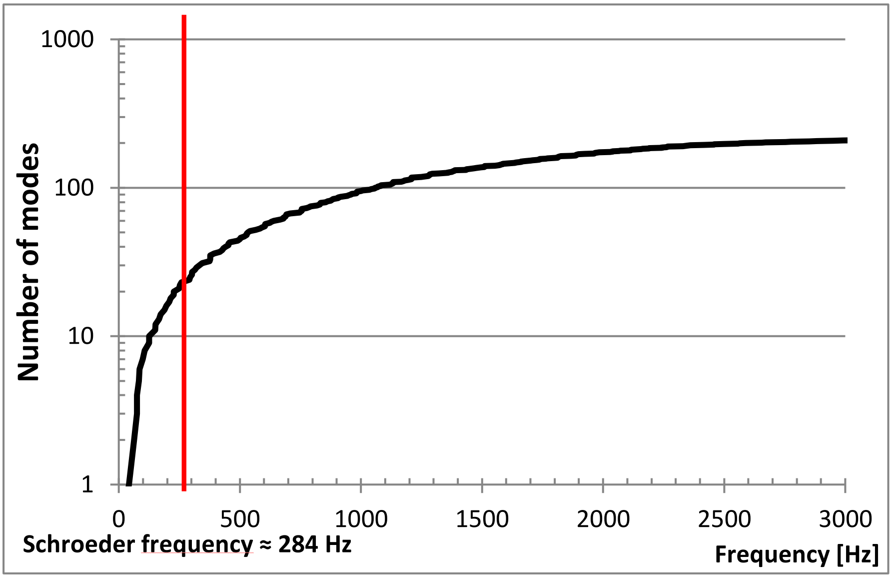

Figure 9. Number of room modes, according to Eq. (2), for a room with length 4 m, width 2.25 m and height 2.7 m. The red line shows the Schroeder frequency for the reverberation time 0.5 s.

3 Sound transmission of impact sound

Sound and vibration transmission is a vast area in technical applications. Some important physical principles for sound transmission related to impact sound in buildings are presented here.

Impact sound stems from interactions between different media or structures that transmit sound. The excitation is made by a force, e.g. the contact force between a foot and a floor. From the floor surface, vibrations are transmitted through different layers of materials, such as floor mats, gypsum boards, or wooden flooring with plastic foam sound insulation layers underneath, timber joist structures, possible air spaces and ceiling structures underneath. The vibration of the bottom of the floor causes the air to vibrate, and, subsequently, sound is transmitted. The transmission trough the floor is called direct transmission. The vibrations may also transmit from floors over to structural connections and down to walls that may radiate to adjacent rooms. This is called flanking transmission.

The only way sound can propagate in a gas, such as air, is through compression waves. Solid structures vary in the nature of sound transmissions; commonly, transmissions are made by bending waves, by plane compression waves and also by shear waves. Bending waves are common in floor structures, and this is also a wave type that is efficient in radiating sound into rooms.

Some of the incident sound pressure at the interface of two materials will be reflected and some will be transmitted into other materials. The impedance difference at the intersection of two materials governs sound transmission efficiency. Consider a plane wave from one elastic medium that meets another elastic medium where the wave is travelling perpendicular to the interface, see Figure 10. The elastic media can consist of gases, liquids or solid materials. The specific acoustic impedance, Z (Ns/m^3), of an elastic medium is

$$Z = \rho c, \tag{7}$$

where, c is the speed of sound (m/s), and ρ is the density of the material (kg/m^3). The speed of sound of a compression wave through a medium is;

$$c = \sqrt{\beta/\rho} \tag{8}$$

where β is the adiabatic compression modulus (N/m^2). Consequently, $Z = \sqrt{\beta\rho}$. There are usually tabular values for both the densities and speeds of sound to be used, but both properties can be measured fairly easily as well. The sound pressure, $\hat{p}_t$ (Pa), that is transmitted into a medium from a compression wave with the sound pressure, $\hat{p}_i$ (Pa), from an adjacent medium is described by the equation,

$$T = \frac{\hat{p}_t}{\hat{p}_i} = \frac{2\rho_2 c_2}{\rho_2 c_2 + \rho_1 c_1} \tag{9}$$

where T is the transmission coefficient.

This basic principle of impedance is also valid at impact points and intersections of different parts of a floor system and its connections to walls. In reality, the transmission becomes more complex when sound waves are not perfectly perpendicular to the structure or to the fluid. The example shows, however, the important principle of the dominating transmission parameters.

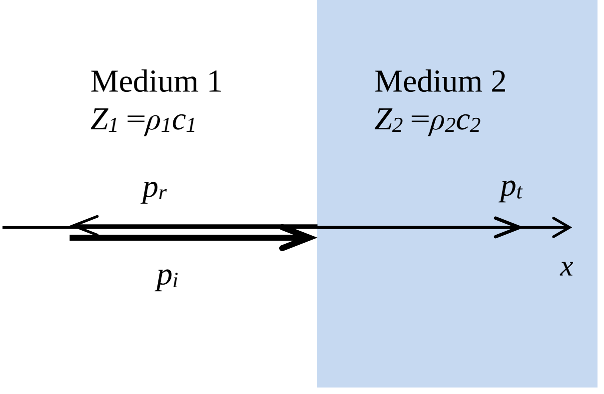

Figure 10. When a perpendicular planar harmonic wave reaches the interface of two media, some of the incident sound wave is reflected and some is transmitted into the other medium. The picture is reproduced from Bodén et al.(2001).

Besides the speed and size of the impact excitation source, the transmitted force into a structure depends on both the mechanical impedance at the interface of the impact source and the impedance of the excited structure. The velocity, $\dot{u}(\Omega)$, of a structure at the excitation point at the circular frequency Ω (radians/s) is described by

$$\dot{u}(\Omega) = Z^{-1}(\Omega)P(\Omega), \tag{10}$$

where $Z(\Omega)$ is the complex valued mechanical impedance (Ns/m), and $P(\Omega)$ is the force. A remark is that the impedance of a structure is commonly direction sensitive, i.e. the angle of the force affects the response of the structure. It is also common to use the inverse of the impedance, i.e mobility, in equations and in the results of vibrations and vibration transmissions. The previous equation in mobility is,

$$Y(\Omega) = \frac{\dot{u}(\Omega)}{P(\Omega)} = Z^{-1}(\Omega). \tag{11}$$

The mobility, Y, thus, has the unit m/(Ns). In finite element software, impedances / mobilities at or between dofs are easily calculated. In the event of an interface with two objects, as with floors and an excitation source with an impedance, it could be described as (Cremer et al., 2005).

$$\dot{u}_F(\Omega) = \frac{\dot{u}_E(\Omega)}{\left(Y_E(\Omega) + Y_F(\Omega)\right)} Y_F(\omega) \tag{12}$$

Where, $\dot{u}_F$ is the floor velocity, $\dot{u}_E$ is the free velocity of the excitation source, and Y_E is the mobility of the excitation source at the interface to the floor, which has the point mobility Y_F. This is a simplification of the excitations as free harmonic loads and is, of course, not the situation for real foot impacts. There is no harmonic-free vibration of a footfall before the impact, and there is no harmonic steady state excitation after the impact. However, it helps to understand the fundamental relations and importance of the different variables of vibration transmission when a moving object meets a stationary one.

4 Frequency response functions

The purpose of the chapter is to present the theory for modal-based receptances used in Paper IV and Paper V for the calculation of responses between excitation points and the discrete points that make up a sound-radiating ceiling. The chapter makes up prerequisite for the sound radiation theory and sound transmission model presented in Paper IV.

4.1 Fundamentals of modal analysis

The abbreviation FRF stands for Frequency Response Function, which is also known as a transfer function. A FRF is the relationship between a harmonic output (response) at dof i, U_i, and a harmonic input (excitation) at dof j, P_j, as a function of frequency. The general mathematical expression is written as

$$H_{ij}(\Omega) = \frac{U(\Omega)}{P_j(\Omega)} \tag{13}$$

in which H is the frequency response function. Transfer functions can be used in a wide range of applications. Within structural dynamics, the input (P) is commonly a vector of forces (N) as a function of the circular excitation frequency Ω (rad/s). Using accelerometers, the measured responses (U) form an acceleration vector (m/s^2), and using microphones, the response is a vector of sound pressures (Pa). Transfer functions can be presented in different integrations and as their inverses, depending on the matter to be analyzed. Common names for structural dynamic FRFs are presented in Table 1 and Table 2. FRFs consist of complex numbers that contain information of the phase angles and the magnitudes, having the physical unit of the response divided with the physical unit of the excitation.

Table 1. Commonly used response / excitation FRFs within structural dynamics.

Dimension	Displacement / Force	Velocity / Force	Acceleration / Force
Name	Admittance, Compliance, Receptance	Mobility	Accelerance, Inertance

Table 2. Commonly used excitation / response FRFs within structural dynamics.

Dimension	Force / Displacement	Force / Velocity	Force / Acceleration
Name	Dynamic Stiffness	Mechanical Impedance	Apparent Mass, Dynamic Mass

Models used in structural dynamics calculations are divided into two groups:

1. Continuous models
2. Discrete models

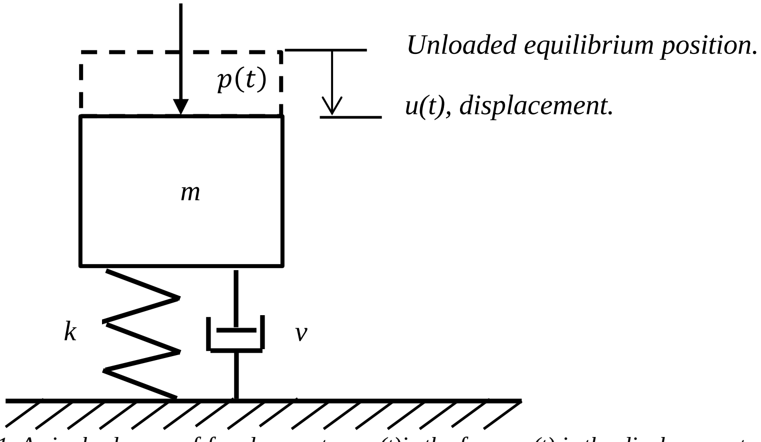

Figure 11. A single-degree-of-freedom system. p(t)is the force, u(t) is the displacement, m, k and v are the mass, stiffness and damping coefficient, respectively.

Discretized models are divided into single-degree-of-freedom (SDOF) and multiple-degree-of-freedom (MDOF) systems. An SDOF system describes the most fundamental dynamic system. It consists of a mass (kg), a spring with the stiffness k (N/m), possible a damper with the damping coefficient v, an excitation force, p (N), and a displacement coordinate u (m), see Figure 11. The most commonly used damping model is a viscous damping representation with the damping coefficient, v (Ns/m). The damping force is, then, proportional to the velocity of the mass.

For the SDOF system shown in Figure 11, the governing equation of motion becomes

$$m\ddot{u} + v\dot{u} + ku = p(t). \tag{14}$$

A harmonic force can be written as

$$\bar{p} = \bar{P}e^{i\Omega t} \tag{15}$$

where Ω is the circular excitation frequency (radians/s). For a linear system subjected to a harmonic excitation, with the circular frequency Ω, the response will also be a harmonic with the frequency Ω. Hence, the steady-state response is solved by assuming the harmonic solution

$$\bar{u} = \bar{U}e^{i\Omega t} \tag{16}$$

The SDOF response due to a harmonic excitation subsequently becomes

$$\bar{U}(\Omega) = \frac{\bar{P}(\Omega)}{k - m\Omega^2 + iv\Omega} \tag{17}$$

where $\bar{U}$ is the complex valued displacement. In the equation denominator, the circular frequency Ω that makes $k - m\Omega^2$ vanish is the undamped circular resonance frequency, or alternatively, the circular natural frequency of the undamped SDOF system. The natural frequency, f_n, in Hz, for an SDOF system is,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}. \tag{18}$$

The FRF, $\bar{H}(\Omega)$, from force to response for an SDOF system is achieved by dividing the displacement with the force,

$$\bar{H}^{rec}(\Omega) = \frac{\bar{U}}{\bar{P}} = \frac{1}{k - m\Omega^2 + iv\Omega} \tag{19}$$

This FRF is a complex valued receptance. Transformations to other derivatives of the displacement are made by multiplying by $i\Omega$ as many times as needed. Hence, the transformation from receptance to mobility becomes,

$$\overline{H}^{mob} = i\Omega\overline{H}^{rec}\ [\mathrm{m/(Ns)}], \tag{20}$$

A transformation to accelerance is done by yet another multiplication by $i\Omega$

$$\overline{H}^{acc} = -\Omega^2\overline{H}^{rec}\ [\mathrm{m/(N}s^2)], \tag{21}$$

4.2 Damping

In vibrations of undamped systems, energy is conserved and transformed between kinetic energy and potential energy. However, in almost all systems, there is a certain degree of energy that disappears from this transformation. Most often this is due to some type of friction or hysteresis within the material. The damping is important in FRF analyses since the maximum amplitudes, at resonances, are governed by damping. Commonly, the damping is modelled as viscous damping. This means that the damping force is linearly proportional to the velocity. This model is useful both for the fundamental understanding of dynamic systems and for real applications. Damping is often described as a relative damping factor in engineering; the ratio between the damping and the critical viscous damping is

$$\zeta = \frac{v}{v_{cr}}. \tag{22}$$

The critical damping for an SDOF system is

$$v_{cr} = 2\sqrt{km} = 2m\omega. \tag{23}$$

Hence, the relative viscous critical damping is

$$\zeta = \frac{v}{2m\omega} \tag{24}$$

4.3 Multiple-degree-of-freedom systems

Timber floors are usually complex designs that make SDOF models insufficiently representative for their dynamics. An MDOF approach is subsequently needed.

The Finite Element Method (FEM) is a numerical method used to solve differential equations by discretizing systems by user selected shape functions. In this thesis, FEM is used for calculations of transfer functions from one degree-of-freedom of the structure (for instance, a point impact force from a foot) to other degrees-of-freedom of the structure (for instance, the sound radiating ceiling underneath the floor of the foot impact).

4.3.1 Natural frequencies and mode shapes of undamped systems

The equation of motion for free decay of a linear undamped MDOF system, see Figure 12 for an example, is written as

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{0} \tag{25}$$

where **M** is the mass matrix, **K** is the stiffness matrix and **u** is the displacement vector.

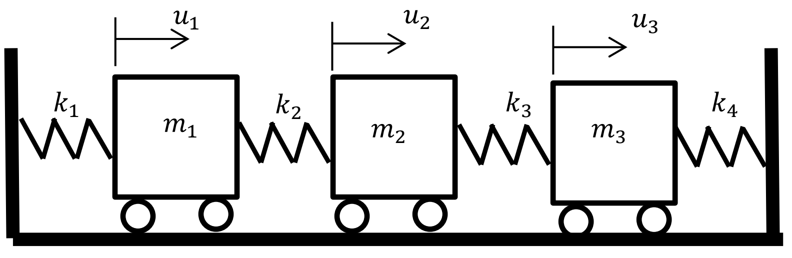

Figure 12. An example of an MDOF system.

The stiffness and mass matrices of the system are

$$\mathbf{K} = \begin{bmatrix} k_1 + k_2 & -k_2 & 0 \\ -k_2 & k_2 + k_3 & -k_3 \\ 0 & -k_3 & k_3 + k_4 \end{bmatrix}, \ \mathbf{M} = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}. \tag{26}$$

The ansatz for a solution is a harmonic displacement of the form

$$\mathbf{u}(t) = \boldsymbol{\phi}_n e^{i\omega_n t}. \tag{27}$$

By assuming this, the equation of motion can be written as

$$(\mathbf{K} - \omega_n^2 \mathbf{M})\boldsymbol{\phi}_n e^{i\omega_n t} = 0. \tag{28}$$

From this equation, a solution of ω_n^2 is such that it makes the determinant of the first part of the equation vanish

$$\det(\mathbf{K} - \omega_n^2\mathbf{M}) = 0. \quad (29)$$

The square root of a solution ω_n^2 is the n^{th} natural circular frequency (rad/s) of the MDOF system. Subsequently, the eigenvector ϕ_n associated with each natural circular frequency, ω_n , can be calculated by solving

$$(\mathbf{K} - \omega_n^2\mathbf{M})\boldsymbol{\phi}_n = 0. \quad (30)$$

The solved eigenvectors, $\boldsymbol{\phi}_n$, are dimensionless displacement vectors with arbitrary scaling but with determined relations between all the degrees-of-freedom. In structural dynamics, the eigenvectors are known as mode shapes. Mode shapes can be collected in a modal matrix

$$\boldsymbol{\Phi} = [\boldsymbol{\phi}_1\ \boldsymbol{\phi}_2 \ldots \boldsymbol{\phi}_N]. \quad (31)$$

4.3.2 The mode superposition method

The basic principle behind the mode superposition method is that a vibrational mode, or shape, can be expressed as a linear combination of a structure's eigenmodes, see Figure 13. Each natural frequency is associated with an eigenmode shape, i.e. a periodic motion that repeats itself with the period of time corresponding to the natural frequency. Each eigenmode is also associated with a specific damping factor and a modal mass.

How a structure is excited affects which eigenmodes that will dominate the vibrations. The location of the excitation, on the structure, affects which modes that can be engaged. The frequency or duration of the excitation force affects how the energy is distributed between the modes. In other words, the amplitude of the excitation directly affects the amplitude of the response.

In the low-frequency range, the idea is to calculate the eigenmodes in order to get a modal model that accurately represents the dynamic response of the structure being studied. The equation of motion for a viscously damped linear MDOF system is

$$\mathbf{M\ddot{u}} + \mathbf{V\dot{u}} + \mathbf{Ku} = \mathbf{p}(\mathbf{t}), \quad (32)$$

where **M** is the mass matrix, **V** is the viscous damping matrix, **K** is the stiffness matrix and **p(t)** is the excitation force.

a). A beam or a simplified floor, just before an excitation / a step.

b). A displacement shape of the beam / floor consisting of a summation of the structure´s eigenmode shapes

c). Eigenmode 1, $M_1, f_{1,}\boldsymbol{\phi}_1, \zeta_1$

d). Eigenmode 2, $M_2, f_{2,}\boldsymbol{\phi}_2, \zeta_2$

e). Eigenmode 3, $M_3, f_{3,}\boldsymbol{\phi}_3, \zeta_3$

f). Eigenmode N, $M_N, f_{N,}\boldsymbol{\phi}_N, \zeta_N$

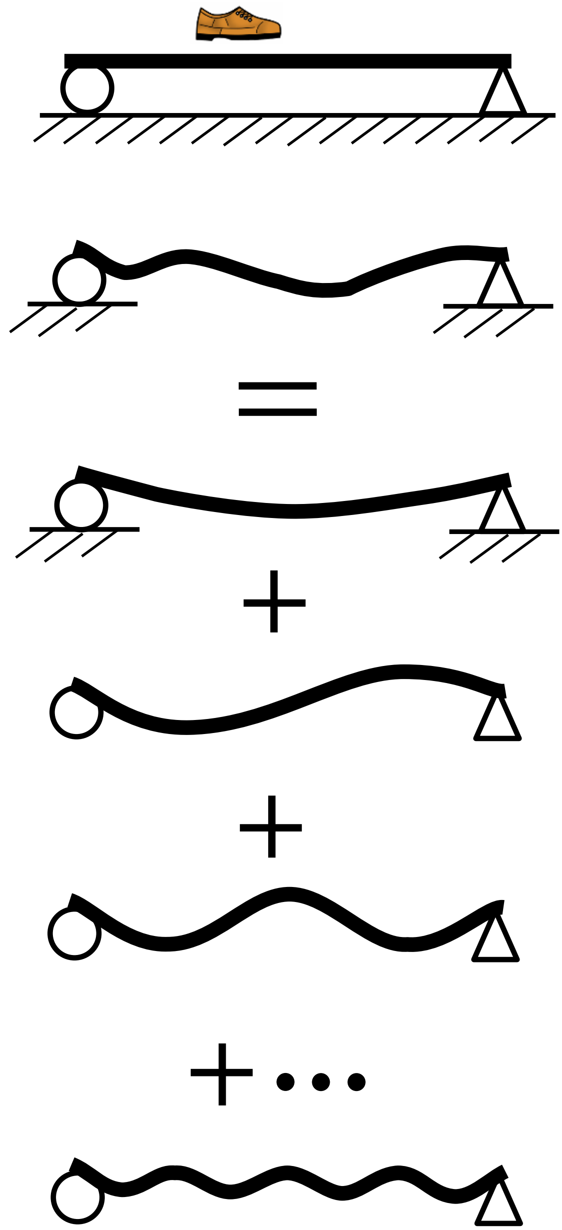

Figure 13. The principle of mode superposition.

The transformation

$$\mathbf{u} = \mathbf{\Phi\eta} \tag{33}$$

where $\mathbf{\Phi}$ is the modal matrix, and $\boldsymbol{\eta}$ is a vector containing the modal coordinates, $\boldsymbol{\eta} = [\eta_1, \eta_2, \eta_3, \dots, \eta_N]^T$is applied. Due to the mass orthogonality of the eigenmodes, $\mathbf{\Phi}^T\mathbf{M}\boldsymbol{\phi}$ and $\mathbf{\Phi}^T\mathbf{K\Phi}$ become diagonal. Thus, the strategy is to pre-multiply the terms in Equation (33) by $\boldsymbol{\phi}^T$, which renders

$$\mathbf{\Phi}^T\mathbf{M\Phi}\ddot{\boldsymbol{\eta}} + \mathbf{\Phi}^T\mathbf{V\Phi}\dot{\boldsymbol{\eta}} + \mathbf{\Phi}^T\mathbf{K\Phi}\boldsymbol{\eta} = \boldsymbol{\phi}^T\ \mathbf{p}(t)\mathbf{.} \tag{34}$$

For simplicity;

$$\boldsymbol{M} = \mathbf{\Phi}^T\mathbf{M\Phi} \tag{35}$$

is denoted the modal mass matrix. Correspondingly, the modal stiffness matrix is defined as

$$\boldsymbol{K} = \mathbf{\Phi}^T\mathbf{K\Phi}. \tag{36}$$

There is no reason for $\mathbf{\Phi}^T\mathbf{C\Phi}$ to become diagonal, but computationally it can be forced to be diagonal, and that can be a reasonable approximation for lightly damped systems. The modal damping matrix is defined to be

$$\boldsymbol{V} \quad = \mathbf{\Phi}^T\mathbf{V\Phi}. \tag{37}$$

Further, the modal force vector is,

$$\boldsymbol{P}(t) = \mathbf{\Phi}^T\mathbf{p}(t) \tag{38}$$

Hence, with the assumption above on the damping, the equation for motion becomes

$$\begin{bmatrix} M_1 & 0 & \cdots & 0 \\ 0 & M_2 & \cdots & 0 \\ \vdots & \vdots & \ddots & 0 \\ 0 & 0 & 0 & M_N \end{bmatrix} \begin{Bmatrix} \ddot{\eta}_1 \\ \ddot{\eta}_2 \\ \vdots \\ \ddot{\eta}_N \end{Bmatrix} + \begin{bmatrix} V_1 & 0 & \cdots & 0 \\ 0 & V_2 & \cdots & 0 \\ \vdots & \vdots & \ddots & 0 \\ 0 & 0 & 0 & V_N \end{bmatrix} \begin{Bmatrix} \dot{\eta}_1 \\ \dot{\eta}_2 \\ \vdots \\ \dot{\eta}_N \end{Bmatrix}$$
$$+ \begin{bmatrix} K_1 & 0 & \cdots & 0 \\ 0 & K_2 & \cdots & 0 \\ \vdots & \vdots & \ddots & 0 \\ 0 & 0 & 0 & K_N \end{bmatrix} \begin{Bmatrix} \eta_1 \\ \eta_2 \\ \vdots \\ \eta_N \end{Bmatrix} = \begin{Bmatrix} P_1(t) \\ P_2(t) \\ \vdots \\ P_N(t) \end{Bmatrix}, \tag{39}$$

or

$$\boldsymbol{M}\ddot{\boldsymbol{\eta}} + \boldsymbol{V}\dot{\boldsymbol{\eta}} + \boldsymbol{K}\boldsymbol{\eta} = \boldsymbol{P}(t). \tag{40}$$

The system now consists of *N* uncoupled generalized SDOF equations

$$M_n\ddot{\eta}_n + V_n\dot{\eta}_n + K_n\eta_n = P_n(t). \tag{41}$$

ϕ_n is commonly scaled to give M_n a specific value, e.g. normalized so that $M_n = 1$. To calculate the steady-state response caused by a harmonic load, at dof j, each decoupled equation of motion is solved:

$$M_n\ddot{\eta}_n + V_n\dot{\eta}_n + K_n\eta_n = \boldsymbol{\phi}_{j,n}\bar{\mathbf{p}}e^{i\Omega t} \tag{42}$$

The ansatz

$$\eta_n = \mathrm{\eta}_n \mathrm{e}^{i\Omega t}, \tag{43}$$

results in

$$[K_n - M_n\Omega^2 + iV_n\Omega]\mathrm{\eta}_n = \boldsymbol{\phi}_{j,n}\bar{\mathbf{p}}. \tag{44}$$

Hence,

$$\mathrm{\eta}_n = \frac{\boldsymbol{\phi}_{j,n}\bar{\mathbf{p}}}{K_n - M_n\Omega^2 + iV_n\Omega} \tag{45}$$

Subsequently, the physical response is

$$\mathbf{u}(t) = \sum_{n=1}^{N} \boldsymbol{\phi}_n \eta_n. \tag{46}$$

Solving Equation (32) for these conditions gives

$$\mathbf{u}(t) = \sum_{n=1}^{N} \left(\frac{\boldsymbol{\phi}_n \, \boldsymbol{\phi}_{j,n} \overline{\mathbf{P}}}{K_n} \right) \left[\frac{1}{(1 - r_n^2) + \mathrm{i}(2\zeta_n r_n)} \right] e^{i\Omega t}, \tag{47}$$

where r_n is the ratio between the excitation frequency and the n[th] natural frequency,

$$r_n = \frac{\Omega}{\omega_n}, \tag{48}$$

The complex frequency response function for a response at degree-of-freedom *i* due to a harmonic excitation at dof *j* becomes,

$$\bar{H}_{ij}(\Omega) = \sum_{n=1}^{N} \left(\frac{\phi_{i,n} \phi_{j,n}}{K_n} \right) \left[\frac{1}{(1 - r_n^2) + \mathrm{i}(2\zeta_n r_n)} \right]. \tag{49}$$

This transfer function calculation is applicable for FE models that represent structures from one dof to any other dof in the model.

5 Summary of the appended papers

A brief summary of the purpose and results of each paper is presented below.

Paper I

Title*: Low-frequency impact sound pressure fields in small rooms within lightweight timber buildings — suggestions for simplified measurement procedures.*

This paper is related to research question number one. This study had two purposes. The first was to study the nature of low-frequency impact sound pressure distributions in small rooms within two lightweight timber buildings. The second purpose was to use the measurement data from the sound pressure fields to evaluate the quality of simplified field measurements and to propose an improved method.

The measurements were performed in two office rooms, one in each building, with nearly identical dimensions (25 m^3 and 26 m^3), with different design solutions and sound insulation performances. The rooms were divided into three-dimensional grids with distances ranging from 25 cm to 30 cm. The excitations were done in the middle of the room above with an ISO rubber ball, and measurements of the sound pressure at the grid points were made in sequence. The ball drops showed high repeatability in the frequency range spanning from 20 to 50 Hz. The average standard deviation of the transmitted sound in third octaves was less than 0.5 dB for all repeated ball drops.

It was found that the highest sound pressure levels occurred in the lowest parts of the receiving rooms, especially in the corners, farthest away from the excitation point on the middle of the floor in the room above. The height of the room was the single-most important variable for the total sound pressure exposure. The highest impact sound levels occurred in the low-frequency range 20 – 50 Hz in both buildings. Measuring close to all four corners of the floor, i.e. at the surface farthest from the excited surface (the ceiling), is an efficient way to measure values similar to the 95th percentile sound pressure level. For the measured rooms, this was observed from 12.5 Hz up to around 160 Hz in third octave bands. In order to achieve average values, more extensive measurements would be needed. The domination of a few room modes at low frequencies gave a wide range of pressure distribution within the room. Tests were conducted with different samples from the grid points in order to find a method to achieve simplified field measurements. One such method is the NSP0.7m requirement. This means that none of the microphones are to be in the same x-, y- or z-planes and that the minimum distance between two microphones is not to be larger than or equal to 0.7 m. Different sets of five microphones were evaluated. The NSP0.7m method requires slightly fewer measurements than a random selection of microphone points in order to achieve a specified standard error.

Paper II

Title*: Force to sound pressure frequency response measurements using a modified tapping machine on timber floor structures.*

This paper is related to research question number two: Would an FRF measurement methodology be a feasible improvement in the strategy for simulations and measurements? This study concerned the use of frequency response functions (FRFs) for analyses and correlation to measurements of impact sound. Impact sound transmission FRFs can be calculated by using finite element software, and FE calculations are well-suited for the low-frequency range. FRFs may offer common ground for studies of correlations between measurements and analyses, which may benefit the share of simulations in acoustic development in the building industry.

In this study, the aim was to evaluate how well a modified tapping machine functions to realize impact sound insulation FRF measurements in practice. A successful outcome would imply that both FRF measurements and classical impact sound measurements could be performed with the same excitation device. Tests on real floor objects were conducted. The measurements were done in a timber building (the M building) at Linnaeus university in Växjö, Sweden, and in the RISE acoustics lab in Borås, Sweden, where two configurations of a cross-laminated timber (CLT) floor were tested. The CLT had a floating screed layer above the CLT in one test, and the CLT was plain, without any layer, in the other test. The aim was to use the excitation device to measure FRFs from impact points to microphone, and potentially also use accelerometer positions in the receiving room. The purpose of the accelerometers was to use the FRFs to obtain information about the transfer paths of impact sound and to visualize the vibrations of the ceiling, floor and walls in the receiving room. By having the exact same excitation point on the floor to the exact same response point in the room below, the different excitation devices were expected to give similar FRFs.

The tapping machine itself does not measure forces at impact, and it is therefore normally not used to achieve force spectra and, consequently, not FRFs. Therefore, the tapping machine was modified to measure forces to achieve FRFs. Only one hammer, the one modified to give estimates of the forces, was active. The same point-to-point measurements were made with the modified tapping machine, an electrodynamic shaker and a modal hammer, in order to compare their quality as excitation devices for FRF measurements. An ISO rubber ball was also used, mainly for comparisons of the repeatability in transmissions from impact excitations, since the force measurement rig in Paper III had not been developed at the time of these measurements. The measurement results showed differences in the estimated FRFs. Measurements were made with the electrodynamic shaker with two different excitation forces (10 N and 30 N amplitudes, stepped sine). Comparisons showed that the maximum deviation relative to the FRFs from the 10 N excitation was about 13 % (1.06 dB), and this occurred in the M building at 18 Hz. For the CLT floors, the largest deviation was 10.6% (0.88 dB) at 22.4 Hz. The point impedance of the excitation points also had the largest deviations at the lowest frequencies. It was found that the largest narrowband deviation in the CLT measurements corresponded to 6 dB (105%) at around 15 Hz. An almost identical deviation pattern was observed at the same frequency when the same floor without CLT screed was measured, with a slightly lower peak of around 5.2 dB (81%). The largest normalized deviations in the FRFs for the electrodynamic shaker occurred at the lowest frequencies and decreased gradually at higher frequencies. A comparison of the different excitation devices from the excitation point to the microphone points showed that the largest deviation, normalized against 10 N shaker excitation, occurred for the tapping machine for the plain CLT-floor. There was a difference of 3 dB at the 31.5 Hz third octave band.

Paper III

Title: *Measurements of low frequency impact sound frequency response functions and vibrational properties of lightweight timber floors utilizing the ISO rubber ball.*

This paper was also related to the measurement side of research question number two: Would an FRF measurement methodology be feasible, from the practical point of conducting measurements? In order to extract FRFs, the impact force has to be known. In this study, a measurement rig was designed and manufactured for the force measurement since the ISO rubber ball is not easily equipped with a force gauge. The question of whether or not measuring point mobility by using an accelerometer beside the impact point could estimate the static stiffness of the floors together with the fundamental natural frequencies was also investigated. Stiffness and natural frequency are common design requirements of lightweight floors.

The measurement rig was tested on two different designs of lightweight timber floors, a ground concrete floor, parquet floating on EPS layers above a concrete floor and a simply-supported 25-mm thick steel plate with a span of 2.0 meter and width of 40 cm. Due to the large variations in the design of the test objects, the point mobilities covered a large span. The point mobilities also showed deviations between excitations with a modal hammer and a rubber ball. Point mobilities can be used to estimate the fundamental natural frequency of floors. The tests to extract the static stiffnesses of floors from the dynamic measurements were not successful. The precisions at the lowest frequency range seemed too insufficiently stable in the measurements to make reliable estimates. The measured force spectra were similar up to 55 Hz, regardless of the floor or surface tested. To obtain FRFs from ball drops higher in frequencies, the impact force has to be measured, for instance, by using a ball measurement rig.

Paper IV

Title: *Impact evaluation of a thin hybrid wood based joist floor*.

This paper is related to research question number three: Is the FEM, together with an FRF-based approach, a useful tool for calculating low-frequency impact sound in lightweight timber buildings?

The purpose of this paper was twofold. The first was to develop a numerical analysis procedure by combining FRFs from FE models with analytical formulas for sound emission and transmission from the ceiling and downwards within a room with four walls. The aim of applying this approach was to obtain a tool that calculates the relative impact sound between different floors in the low-frequency range. The second purpose was to benchmark a thin hybrid wood-based floor with similar thickness, surface weight and global bending stiffness as a concrete hollow core floor structure to study differences in sound transmissions. The question is relevant since it may be necessary to make thinner wood-based floors in high-rise buildings if wood remains competitive with concrete. The results showed that the direct transmissions of impact sound were similar around the first bending mode. As the frequency increased, the modes in the structures differed significantly. Below 100 Hz, the concrete floor had four modes while the hybrid floor had nine modes. The results showed that it is possible to have similar sound transmission properties around the first bending modes for a timber-hybrid floor and a hollow core concrete floor structure with similar thicknesses. At the first modes of the structure, information about surface weight and global bending stiffness is useful for the prediction of sound transmission properties, but they are not sufficient at higher modes.

Paper V

Title: *Low-frequency impact sound of timber floors: A finite-element model-based study of conceptual designs.*

The purpose of this paper is similar to the purpose of Paper IV and is related to research question number three. The concept of simulation is similar to the concept in the previous paper. However, in this study fluid elements were used instead of the analytical calculation in Paper IV. Infinite elements were positioned five meters below the ceiling of the floor in the models. Modal-based transfer functions were used to calculate FRFs from the excitation points above the floor to the evaluated nodes, which were at the intersection at 1.45 meter below the ceiling in the room below. With an infinite shaft, there will theoretically not be any reflections of sound in the receiving room, which was the assumption used in Paper IV. The calculations were used to make relative comparisons of the floors. No impact or reverberation needs to be simulated. The idea is that, for estimations of real sound insulation in buildings, comparisons have to be made against reference floors that have both simulated and measured performances.

In this study, a floor concept that was similar to the concept used in lightweight modular buildings in Sweden was modelled. It had an upper floor frame part, from the upper building module, and a lower ceiling part from the lower building module. Between these were elastic elements for sound and vibration insulation. A number of modifications were made in order to investigate the influence of different potential improvement measures on the sound transmissions. For instance, the weight of 50 mm of sand put between the parquet floor and the particle board underneath, or 50 mm of sand on the gypsum boards of the ceiling, or 25 mm of sand evenly distributed on both the floor part and the ceiling part. 50 mm of screed on the floor, single layers of cross-laminated timber consisting of softwood and birch and a few other variants were tested. Four different excitation points were used: one in the center of the floor above a joist and one between two joists, similar to the two excitation points about a quarter from the floor edges. Excitations above a joist and between joists were done due to the relatively large difference in point mobilities that floors with joists and periodical stiffeners may have. Two cases, one clamped and one with moment-free boundary conditions along the longitudinal direction of the ceiling frame, were used. The difference in results is interesting, since the boundary conditions in reality is likely in between these two cases.

It was found that, for the concept, the best improvement in FRFs in relation to the unchanged reference model in the 20 to 50 Hz range was to have sand on the particle board or the same amount of sand evenly distributed between the ceiling and the floor particle board. At higher frequencies, from 50 to 200 Hz, screed on the floor and 50 mm of sand on the gypsum boards of the ceiling gave the best improvements. It was also found that changing the CLT from spruce to birch improved impact sound by 3-4 dB, on average, in the frequency range from 20 to 50 Hz. However, in this study the birch seemed to perform worse in the 50 – 100 Hz range. The findings indicate that some models are less prone to vibrate due to forces on the chosen excitation points.

The point mobilities were also presented. The impact sound level also depend on the excitation force level. There is currently no such descriptor of relevant force excitations spectra in relations to point mobilities spectra. The CLT models had the lowest mobilities, which indicates that the impact forces may be larger for CLT models than for softer floors, for example, for tapping machine excitations. At the lower frequencies, this should be less of an issue. Research by others (Amiryarahmadi et al, 2016), has indicated that the force excitation levels from walking are largely unaffected in the low-frequency range. The measurements have not yet been verified with experiments.

6 Discussion and conclusions

The most important conclusions from the appended papers are summarized below.

Within the small rooms (about 25 m^3) in the two building structures, the largest sound pressure levels at low frequencies occurred in the corners of the floors, when the impact was in the middle of the room above. The height position was the single most important parameter for the variation of the sound pressure in the room. The ISO rubber ball drops showed high repeatability in the low-frequency range 20 - 50 Hz (the average standard deviation of the transmitted sounds in third octaves was less than 0.5 dB for all repeated ball drops).

For both building designs, the largest sound pressures occurred below 50 Hz, when the excitation was done with the impact ball. The results from one of the two buildings show that the highest sound pressure could occur even at a lower frequency than the frequency of the first room mode. The 95th percentile intervals for the variation of the sound pressure in the rooms were the largest among the first room modes. Although there was variation of the sound pressure around the first modes, it is efficient to measure the corners of a floor at the excitation from above in order to obtain the maximum values within the low-frequency range. In order to guarantee a certain low standard error of the mean value, many samples are needed. By measuring iteratively, with a number of microphones simultaneously using both different excitation points and different microphone positions, fast measurements can be obtained. By adding the results as the measurements progress, it is possible to stop the procedure at a precise standard error. By changing both the excitation positions at each set of microphone points and at each microphone position, the statistical sound pressure distribution in the receiving room and the floor insulation distribution could be obtained. With prescribed rules (NSP0.7m) for the selection of microphone points, such as the minimum distance between the microphones and no microphones on the same plane, fewer measurements are needed than for a random selection of microphone points.

Frequency response functions measurements between two rooms in a timber building, and through a CLT floor with a floating screed and no screed in a lab were measured. A modified tapping machine, an electrodynamic shaker a modal hammer and the ISO rubber ball were tested, and the frequency response functions from a force on the floor in the room above to sound in the room below and accelerations of radiating walls and the ceiling in the room were measured. It was found that, in the range of the peak levels, the FRFs indicate non-linear transmissions, depending on the excitation force characteristics. This implies that the excitation force and the frequency characteristics of an ideal excitation device, which measures force, should be similar to the impact of human foot excitation in the low-frequency range. The ISO rubber ball is an excitation that is the closest to fulfilling this criterion.

It was demonstrated that the measured FRFs of impact sound and structural response could be pedagogically visualized. This helps to evaluate responses in the low-frequency range and may also be helpful to identify transfer paths. Also, measuring the response of the radiating surfaces in the receiving room has potential in terms of omitting reverberation time compensations. In low-frequency measurements, it is difficult to compensate for reverberation time. Low modal density implies that smooth, single-slope decay curves, as in diffuse fields, will not occur. Also, short decay times in relation to filter bandwidth, the BT factor, is a more pronounced problem in low frequencies for where the filer is more likely to affect the decay curve. If FRF measurements are made on radiating surfaces, there should not be a need to compensate for the furnishings in the receiving room.

To enable measurements of impact forces that stem from the ISO rubber ball, a rig for measurements of forces and potential point mobilities was manufactured and evaluated. Impact force measurements were made using lightweight timber floors as well as on concrete floors. Within the frequency range up to around 55 Hz, the force spectrum of the impact ball was stable, regardless of the floor system measured. It appears to be possible to use a measured force spectrum for the ISO ball, together with impact sound measurements, to create accurate impact force-to-sound frequency response functions for different floor systems.

Modelling and calculations were done using the finite element method together with analytical formulas of sound radiation in a rectangular duct (room). The floor models had two purposes. The first purpose was to see if a wood-based floor could, through reasonable means, achieve a similar thickness, surface weight and global stiffness as a modern hollow-core pre-stressed concrete floor. The second purpose was to see if the same properties in surface weight and global stiffness would imply the same impact sound transmission properties in the low-frequency range. The study showed that it is possible to achieve almost the same surface weight, thickness and bending stiffness for the wood-based floor as for the concrete floor, with use of sand within the floor and not excessive use of steel sheets for added stiffness. The results also show that, around the first bending mode, the sound radiation for a given force becomes similar. However, despite that the global bending mode is the same, the higher order modes differ more due to different local mass and stiffness distributions. The conclusion is that, at the lowest frequencies, surface mass and stiffness may follow a rule of thumb for sound transmission properties but not for modes higher in frequency.

In the last paper, fluid elements and infinite elements were tested for simulation FRFs from impact points above a floor to a room intersection area 1.45 m below the floor. A number of variants of the floor were tested to investigate improvement measures. It was found that modelling floors and simulating them are feasible and fast compared to making prototypes. It is, however, important to have a parameter-controlled pre-processor in order to efficiently create and modify models. Impact sound level also depends on the excitation force level. The excitation forces may depend on the floor properties of the impact points. There is currently no such descriptor. For instance, the simulations with a screed layer showed low-point mobility, indicating that, although the screed has a low FRF transmission, the force from a tapping hammer excitation will increase and, thus, decrease the sound insulation performance compared to a softer floor surface. At lower frequencies, below 50 Hz, others (Amiryarahmadi, et al., 2016) have found that the force excitation levels from walking are largely unaffected. This means that evaluating the lowest frequencies may not require any calibration of the excitation measurement device to floor properties, i.e. point mobilities. Only the sound insulation FRFs, and no point mobilities, are needed to evaluate and compare different floor systems in simulations.

The measurement results indicate that some floor systems may be favored by smaller vibration levels due to the selection of excitation points. These types of simulations would benefit from the approach suggested in Paper I, with an iterative number of excitation and receiving points in order to obtain a statistically certain standard error of the frequency response functions.

Comments on the conclusions in relation to research question number one: What is the nature of low-frequency impact sound distribution in lightweight timber buildings? Paper I contributes to this knowledge, albeit for the measurement conditions and types of building in the tests, i.e. small rooms with lightweight timber floors and excitation from the middle of the room above.

Comments on the conclusions in relation to research question number two: Is measuring FRFs practically feasible in field measurements in buildings?

Papers I - III address this. It was found in Paper III that, in the lowest frequencies ranges up to 55 Hz, FRFs could be achieved with only drops of the ISO rubber ball and by measuring in the receiving room. This is practically feasible. However, the general issues with measuring a sound field room with low modal densities applies not only to FRF measurements. These issues may also apply to measurements with a tapping machine at low frequencies. A measurement method to handle these issues is described in Paper I. A measurement methodology for measuring the sound pressure fields in a receiving room should be practical and fast. The method described in Paper I needed five ball drops multiplied by five microphone positions. This means that the operator has to leave the room five times to move the microphones. This is rather similar to a tapping machine measurement (ISO 16283-2, 2018) where the excitation device is usually moved at least four or five times for each measurement.

Issues concerning how to measure and compensate for reverberation time at low frequencies in small rooms have not been dealt with in this thesis. Low modal density and the third octave filter bandwidth in combination with short reverberation time remain a question mark in efforts to achieve high precision in low-frequency sound field measurements (Jacobsen, 1987, ISO 3382-1, 2009). There are advices for dealing with this issue, see (Hopkins and Turner, 2005). However, the precision of the current state of compensations is likely less than commonly above 100 Hz measurements.

Comments on the conclusions in relation to research question number three: Is FEM, together with an FRF-based approach, a useful tool for calculating low-frequency impact sound in lightweight timber buildings? Papers IV – V address this issue. It was found that modelling floors is feasible. Floor systems usually consist of simple geometrical forms, which are easy to make parametric modifications of. Simulations with modal-based transfer functions are practically feasible in the low-frequency range with today's computers. By applying a relative comparison of a simulated and a measured reference floor, reverberation time issues can be omitted, and an estimation can be made of the performance if a simulated floor was installed in the same building as the reference floor. Simulation accuracy comparisons with real measurements have, however, not been done yet. Also, it was found that timber buildings may be non-linear (Paper II), however, modal-based transfer functions require linear models. It is still advisable to strive for linear models due to their simplicity. The causes of and measures to avoid non-linearities and reduce quality deviations in sound transmission, and methods to ensure the quality of the damping data on floor systems and components are proposed as future work.

7 Future work

Some proposals for future work based on challenges discovered in this thesis are:

In order to obtain an FRF methodology for use at higher frequencies, floor properties have to be identified and how these properties relate to relevant excitation force spectra has to be investigated. Impact sound insulation depends on the transmission properties described by the FRFs together with the excitation force levels of the floor. For lower frequencies, it is seen that the walking force spectra is stable, regardless of floor properties (Amiryarahmadi et al, 2016). However, this does not seem to be valid at higher frequencies. This means that, in order to predict floor insulation properties at frequencies higher than 50 Hz in simulations, the floor properties (i.e. point mobilities) have to be related to force excitation spectra for relevant walking forces. There is no such descriptor today.

Measurements are needed in order to benchmark and gain knowledge about the precision of FRF simulations versus FRF measurements.

It was found in the measurements that timber buildings may be nonlinear. This is a dilemma. The FE simulations conducted in this thesis were linear. The causes of potential non-linearities and quality deviations should be investigated. Also, measurements and methods for modelling the damping of timber floors and components are of interest for FRF simulations. Potential simplification of mineral wool absorption to FE modeling of modal-based transfer functions could also improve FRF simulations.

In low-frequency measurements, it may be difficult to compensate for reverberation time, as mentioned in Discussion and conclusions. One step further in accuracy, would be to improve methods for measuring and compensating for short reverberation times in sound fields with low modal density.

References

Abrahamsen, R.B. (2018). Mjųstårnet-18 storey timber building completed, 24 Internationales Holzbau-Forum IHF, 2018. Garmisch-Partenkirchen, Germany.

Amiryarahmadi, N., Kropp, W., & Larsson, K. (2016). Identification of low-frequency forces induced by footsteps on lightweight floors. *Acta Acustica united with Acustica*, *102*(1), 45-57.

Amiryarahmadi, N. (2019). *A Virtual Design Studio for Low-Frequency Sound from Walking in Lightweight Buildings* (Doctoral dissertation).

Bard, D., Sonnerup, J., & Sandberg, G. (2008). A finite element solution of structure-borne sound attenuation for a lightweight timber floor. *Building Acoustics*, *15*(2), 137-151.

Bodén, H., Carlsson, U., Glav, R., Wallin, H. P., & Åbom, M. (2001). Sound and vibration. *Stockholm: Norstedts Tryckeri AB*.

Brunskog, J., & Hammer, P. (2003). Prediction model for the impact sound level of lightweight floors. *Acta acustica united with acustica*, *89*(2), 309-322

Ceccotti, A., Sandhaas, C., Okabe, M., Yasumura, M., Minowa, C., & Kawai, N. (2013). SOFIE project–3D shaking table test on a seven-storey full-scale cross-laminated timber building. *Earthquake Engineering & Structural Dynamics*, *42*(13), 2003-2021.

Churchill, C. E. (2018). *Direct and flanking transmission across timber concrete composite floors with cross laminated timber walls* (Doctoral dissertation, University of Liverpool).

Craig, R. R. (1995). *Structural dynamics: an introduction to computer methods*. Book, Society for Experimental Mechanics, Inc, 7 School St, Bethel, CT 06801, USA.

Cremer, L., Heckl, M., & Petersson, B. A. T. (2005). Structure-borne sound. Springer.

Díaz-Cereceda, C., Hetherington, J., Poblet-Puig, J., & Rodríguez-Ferran, A. (2011). A deterministic model of impact noise transmission through structural connections based on modal analysis. *Journal of Sound and Vibration*, *330*(12), 2801-2817.

Fletcher, H., & Munson, W. A. (1933). Loudness, its definition, measurement and calculation. *Bell System Technical Journal*, *12*(4), 377-430

Flodén, O., Negreira, J., Persson, K., & Sandberg, G. (2015). The effect of modelling acoustic media in cavities of lightweight buildings on the transmission of structural vibrations. *Engineering Structures*, *83*, 7-16.

Gustavsson, L., & Sathre, R. (2006). Variability in energy and carbon dioxide balances of wood and concrete building materials. *Building and Environment*, *41*(7), 940-951.

Hagberg, K. (2018). *Management of acoustics in lightweight structures* (Doctoral dissertation, Doctoral Thesis, Engineering Acoustics, LTH, Lund University, Lund, Sweden).

Hirakawa, S., & Hopkins, C. (2018). Prediction of heavy impact sounds using transient statistical energy analysis and finite ele-ment methods. *Proceedings of ICSV25, Hiroshima, Japan*.

Homb, A. (2005), *Low frequency sound and vibrations from impacts on timber floor constructions*, Ph.D. thesis, Department of Structural Engineering, Norwegian University of Science and Technology (NTNU), Trondheim, Norway.

Hopkins, C., & Turner, P. (2005). Field measurement of airborne sound insulation between rooms with non-diffuse sound fields at low frequencies. *Applied Acoustics*, *66*(12), 1339-1382.

Jacobsen, F. (1987). A note on acoustic decay measurements. *Journal of Sound and Vibration*, *115*(1), 163-170.

Johansson, M., Linderholt, A., Jarnerö, K., & Landel, P. (2016). Tall timber buildings-a preliminary study of wind-induced vibrations of a 22-storey building. In *World Conference on Timber Engineering, WCTE*

Johansson, M., Linderholt, A., Bolmsvik, Å., Jarnerö, K., Olsson, J., & Reynolds, T. (2015). Building higher with light-weight timber structures: the effect of wind induced vibrations. In *Internoise, 9-12 August, 2015, San Francisco*. Society of Experimental Mechanics (SEM).

Jik Lee, P., & Griffin, M. J. (2013). Combined effect of noise and vibration produced by high-speed trains on annoyance in buildings. *The Journal of the Acoustical Society of America*, *133*(4), 2126-2135.

Ljunggren, F., Simmons, C., & Hagberg, K. (2014). Correlation between sound insulation and occupants' perception–Proposal of alternative single number rating of impact sound. *Applied Acoustics*, *85*, 57-68

Ljunggren, F., Simmons, C., & Öqvist, R. (2017). Correlation between sound insulation and occupants' perception–Proposal of alternative single number rating of impact sound, part II. *Applied Acoustics*, *123*, 143-151.

IEC 61672:2013 (2013). Electroacoustic - Sound level meters, Electroacoustics Std.

ISO 140-7:1998 (1998). Acoustics – Measurements of sound insulation in buildings and of building elements. Part 7: Field measurements of impact sound insulation of floors. *International Organization for Standardization, Geneva, Switzerland.*

ISO 226: 2003 (2003). Acoustics–Normal equal-loudness-level contours International standard. *International Organization for Standardization, Geneva, Switzerland.*

ISO 3382-1. (2009). Acoustics—Measurement of room acoustic parameters—Part 1: Performance spaces. *International Organization for Standardization, Geneva, Switzerland.*

ISO 717-2:2013 (2013). Acoustics – Rating of sound insulation in buildings and of building elements. Part 2: Impact sound insulation. *International Organization for Standardization, Geneva, Switzerland.*

ISO 10140-2: 2010 (2010). Acoustics - Laboratory measurement of sound insulation of building elements. Measurement of airborne sound insulation. *International Organization for Standardization, Geneva, Switzerland.*

ISO 10140-3: 2010 + A1: 2015 (2015). Acoustics - Laboratory measurement of sound insulation of building elements. *Measurement of impact sound insulation. International Organization for Standardization, Geneva, Switzerland.*

ISO 10140-5 (2010). Acoustics-Laboratory measurement of sound insulation of building elements-Part 5: Requirements for test facilities and equipment. *International Organization for Standardization (ISO), Geneva, Switzerland.*

ISO 16283-2:2018 (2018). Acoustics — Field measurement of sound insulation in buildings and of building elements — Part 2: Impact sound insulation. International Organization for Standardization.

ITUR-R BS.468-4. Recommendation – Measurement of audio-frequency noise voltage level in sound broadcasting. https://www.itu.int/dms_pubrec/itu-r/rec/bs/R-REC-BS.468-4-198607-I!!PDF-E.pdf. *International Telecommunication Union (ITU)..*

Jarnerö, K. (2014). *Vibrations in timber floors: dynamic properties and human perception* (Doctoral dissertation, Linnaeus University Press)

Jeon, J. Y., Ryu, J. K., Jeong, J. H., & Tachibana, H. (2006). Review of the impact ball in evaluating floor impact sound. *Acta Acustica united with ACUSTICA*, *92*(5), 777-786

Mitchell, J. F. (1989). The “greenhouse” effect and climate change. *Reviews of Geophysics*, *27*(1), 115-139

Møller, H., & Pedersen, C. S. (2004). Hearing at low and infrasonic frequencies. *Noise and health*, *6*(23), 37-57.

Negreira, J. (2016). *Vibroacoustic performance of wooden buildings: Prediction and Perception*. Lund University.

Olsson, J., Sjökvist, L. G., & Jarnerö, K. (2012). Low frequency measurements of impact sound performance in light weight timber frame office buildings. In *EURONOISE 2012*. European Acoustics Association (EAA)

Qian, C., Ménard, S., Bard-Hagberg, D., Kouyoumji, J. L., & Negreira, J. (2019). Calibration of the ISO tapping machine for finite-element prediction tool on a wooden-base floor. *Building Acoustics*, 26(3), 157–167.

Rabold, A., Buchschmid, M., Düster, A., Müller, G., & Rank, E. (2010). Modelling the excitation force of a standard tapping machine on lightweight floor structures. *Building Acoustics*, *17*(3), 175-197.

Rabold, A., Duster, A., & Rank, E. (2008). FEM based prediction model for the impact sound level of floors. *Journal of the Acoustical Society of America*, *123*(5), 3356.

Sandanayake, M., Lokuge, W., Zhang, G., Setunge, S., & Thushar, Q. (2018). Greenhouse gas emissions during timber and concrete building construction—A scenario based comparative case study. *Sustainable cities and society*, *38*, 91-97.

Schroeder, M. R., & Kuttruff, K. H. (1962). On frequency response curves in rooms. Comparison of experimental, theoretical, and Monte Carlo results for the average frequency spacing between maxima. *The Journal of the Acoustical Society of America*, *34*(1), 76-80.

Sjökvist, L. G. (2008). *Structural sound transmission and attenuation in lightweight structures* (Doctoral dissertation, Chalmers tekniska högskola)

Skålevik, M. (2011, June). Schroeder frequency revisited. In *Proceedings of Forum Acusticum* (Vol. 2011).

Sousa, A. N., & Gibbs, B. M. (2011). Low frequency impact sound transmission in dwellings through homogeneous concrete floors and floating floors. *Applied Acoustics*, *72*(4), 177-189.

Späh, M., Hagberg, K., Bartlomé, O., Weber, L., Leistner, P., & Liebl, A. (2013). Subjective and objective evaluation of impact noise sources in wooden buildings. *Building Acoustics*, *20*(3), 193-213.

SS 25267:2015 (2015). Acoustics – Sound classification of spaces in buildings – Dwellings. Swedish Standard. Swedish Institute for Standards. Stockholm, Sweden.

Öqvist, R. (2017). *Measurement and perception of sound insulation from 20 Hz between dwellings* (Doctoral dissertation, Luleå tekniska universitet).

Öqvist, R., Ljunggren, F., & Ågren, A. (2012). On the uncertainty of building acoustic measurements–Case study of a cross-laminated timber construction. *Applied Acoustics*, *73*(9), 904-912.

Östman, B., Mikkola, E., Stein, R., Frangi, A., König, J., Dhima, D., T. Hakkarainen & Bregulla, J. (2010). Fire safety in timber buildings. *Technical guideline for Europe. SP Report 19.*

Östman, B., Jarnerö, K., Sjökvist, L. G., Larsson, K., & Tillberg, K. (2008). Acoustics in wooden buildings. State of the art 2008. SP - Technical Research Institute of Sweden, SP Report 2008:16