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POSSIBILITY OF MODELLING THE RECEIVING ROOM IN FEM CALCULATIONS

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ABSTRACT

Timber partition floors have problems with their impact sound insulation (ISI) at low frequencies under 100 Hz. The Finnish Government has sought to increase construction from timber in recent years, which causes ISI to play an important role in the structural design process of timber floors. Literature indicates the receiving room plays a significant role in the verifiable impact sound insulation of a timber joist floor. There are very few analytical calculation methods for ISI of timber floors, and their solutions are only valid for specific cases. The most common assumptions in analytical equations are about the diffusivity of the sound field and completely rigid room boundaries. This study revisits the theory of a modal sound field in a receiving room and coupling between a simply supported floor and the sound field of a receiving room. The possibility of modelling the receiving room in FEM calculations of ISI is discussed.

Keywords: *impact sound insulation, timber joist floor, modal sound field*

1. INTRODUCTION

In recent decades, timber has become more popular as a building material due to its environmental benefits. Timber floors however have problems with their impact sound insulation at low frequencies. [1–4]

It has long been acknowledged that the sound field of a room affects the verifiable impact sound insulation (ISI) of intermediate floors especially in the low frequencies where the sound field is modal [1,5–11]. Most calculation methods, however, assume a diffuse sound field and completely rigid boundaries causing innate uncertainty [7–8,11–14].

This article is an extract from the Master's thesis of Lahdensivu [15], where the effect of the room size to the ISI was examined, and focuses on the coupling between a plate and the sound field of a receiving room. The purpose of this study is to show how the coupling between the plate and the room affects the sound pressure field in the receiving room.

2. LITERATURE REVIEW

2.1 Sound field of a receiving room

The sound field of a room is divided by the Schroeder limiting frequency into diffuse and modal sound fields [16], [17]. In 1954 Schroeder defined a minimum requirement for a diffuse sound field to have a modal overlap factor of $M = 10$ [18]. In the 1962 article [19] the modal overlap factor was revised to $M = 3$ based on other studies. In a diffuse sound field, it is assumed that the energy density is equal everywhere in the room i.e. sound can propagate from and to any direction with the same probability and reflect to any direction with equal probability. A diffuse sound field can be studied purely statistically, making calculations simpler. [16–17]

The modal overlap factor is not an adequate indicator to assume a diffuse sound field on all frequency bands in a room. A diffuse sound field cannot be assumed in small, enclosed spaces, in rooms where the longest dimension is clearly greater in comparison to the shortest dimension, in rooms with inadequate amount of scattering and diffusing elements, in rooms with many absorbing surfaces with uneven distribution as well as in very large spaces where the sound field acts locally. [16–17]

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In a modal sound field, the amplitude of the sound pressure doesn't vary with time. Room modes are found on frequencies where at least one room dimension is a multiple of the half wavelength of the mode. There are three types of room modes: axial, tangential and oblique. [20]
In a room with perfectly rigid and reflective boundaries, room modes can be calculated using Eqn. (1).

$$f_{n_x n_y n_z} = \frac{c}{2}\left[\left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_x}{l_x}\right)^2 + \left(\frac{n_x}{l_x}\right)^2\right] \quad (1)$$

Where c = 343 m/s, n_x, n_y and n_z are the number of half wavelengths and l_x, l_y and l_z are the dimensions of the room. Eqn. (1) gives a reasonable estimate for the room modes in most rectangular rooms. Lightweight walls, however, cannot be considered rigid [20].
In a room where the boundaries are not completely rigid the eigenvalues of the wavefunction are complex [16–17,20]. Kuttruff [20] has proposed a simplified way to calculate the sound pressure in a receiving position with non-reflective surfaces. However, solving the system of equations requires numerical methods for iteration.
Kuttruff's equation utilizes the surface impedance of a wall, which is dependent on frequency and the material properties of the boundary. A lightweight gypsum board wall can be considered to perform as a plate resonator. The impedance of a plate resonator can be calculated for example using Eqn. (2) [20].

$$Z = r_s + \mathrm{j}\left(\omega m - \frac{\rho c^2}{\omega d}\right) \quad (2)$$

Where r_s is the flow resistivity of the absorbing material in the cavity, j = $(-1)^{1/2}$, ω is angular frequency, m is surface mass of the plate, ρ = 1,205 kg/m^3, c = 343 m/s, and d is the total thickness of the air cavity.
The absorption of the plate resonator is very localized to its natural frequency. Calculating the absorption coefficient from the plate resonator impedance does not correspond to a measured absorption coefficient of a lightweight wall. The conversion also loses the information about the complex part of the impedance which corresponds to the losses at the boundary.

2.2 Modal coupling between a plate and a room

Multiple different calculation methods for the coupling between the sound field of a room and a structure have been made [7–8,13–14,21]. Most equations predicting the impact sound insulation of timber floors and coupling of the sound field of the room are based on assumptions of diffuse sound field and perfectly rigid room boundaries [7–8,13–14,21].

In 1967 Kihlman [21] suggested an equation to calculate the sound pressure level in the receiving room when the source room is excited with an omnidirectional loudspeaker. Neves e Sousa & Gibbs [7–8] modified Kihlman's [21] equation to research the modal coupling between the receiving room and a homogeneous concrete floor, when the floor is excited by a point force. The sound pressure level in the receiving room in frequency domain can be calculated using Eqn. (3).

$$p(x,y,z) = -\mathrm{j}\omega\rho \sum_{n_x,n_y,n_z=1}^{\infty} \frac{c^2(-1)^{n_z} C_{n_{x1}n_{y1}} \varphi_{n_x n_y n_z}(x,y,z)}{\left[\left(\omega_{n_x n_y n_z} + \mathrm{j}\delta\right)^2 - \omega^2\right] \Lambda_{n_x n_y n_z}} \quad (3)$$

where j = $(-1)^{1/2}$, ω is the angular frequency, ρ = 1,205 kg/m^3, n_x, n_y and n_z are the number of half wavelengths of the mode, c = 343 m/s, $C_{nx,ny}$ is the coupling factor, φ is the eigenmode function of the room, $\omega_{nx.ny.nz}$ is the angular eigenfrequency of the receiving room, $\delta = 6{,}9/T$ [21], [22], [23] and $\Lambda_{n_x n_y n_z} = \int \varphi_{n_x n_y n_z}(x,y,z)\,\mathrm{d}V = V/8$.
When the floor is excited by a point force, the coupling factor is

$$C_{n_x n_y} = \mathrm{j}\frac{4\omega F}{\pi^2 m} \sum_{n_{x1},n_{y1}=1}^{\infty} \left\{\frac{\varphi_{n_{x1}n_{y1}}(x_0,y_0)}{\omega_{n_{x1}n_{y1}}^2(1+\mathrm{j}\eta) - \omega^2} D\right\}$$
$$D = \frac{[(-1)^{n_{x1}+n_x} - 1][(-1)^{n_{y1}+n_y} - 1]}{n_{x1}n_{y1}\left[\left(\frac{n_x}{n_{x1}}\right)^2 - 1\right]\left[\left(\frac{n_y}{n_{y1}}\right)^2 - 1\right]} \quad (4)$$

Where j = $(-1)^{1/2}$, ω is the angular frequency, F is the point force, m is the surface mass of the floor, n_{x1} and n_{y1} are the number of half wavelengths of the plate mode, x_0 and y_0 are the coordinates of the point force, ω_{nx1ny1} is the angular eigenfrequency of the floor, $\eta = \eta_i + X/\sqrt{f}$ and n_x and n_y are the number of half wavelengths of the room mode.
According to literature, the modal sound field of the receiving room can have a significant effect on the sound pressure level and therefore on the impact sound insulation. The material properties of the plate affect the plate modes and the coupling between the plate and room modes. From the point of timber construction, it would be beneficial to examine the effect of modal coupling between the floor structure and the room sound field, since the low frequency range has been found to have problems in the ISI.

3. MATERIALS AND METHODS

The modal coupling of CLT and concrete slabs in different room configurations was studied to examine the differences in room response and modal coupling behaviour.

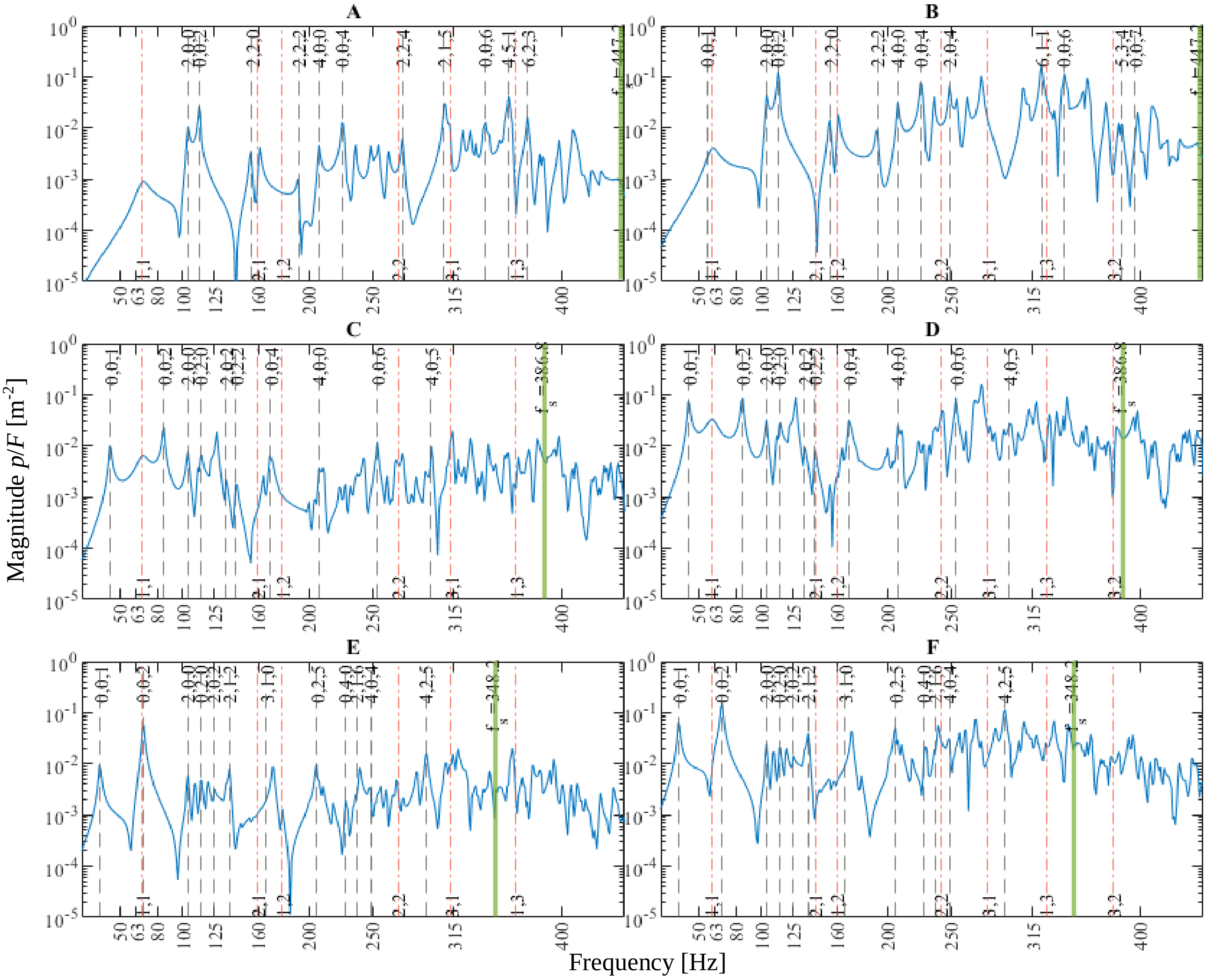

Figure 1. Magnitude of sound pressure in the middle of the receiving room caused by the vibration field of a floor. **A)** Concrete floor $V = 30$ m^3 **B)** CLT floor $V = 30$ m^3 **C)** Concrete floor $V = 40$ m^3 **D)** CLT floor $V = 40$ m^3 **E)** Concrete floor $V = 50$ m^3 **F)** CLT floor $V = 50$ m^3. The room modes are depicted with vertical dashed lines, the plate modes with vertical dotted dashed lines and the Schroeder limiting frequency with a solid vertical line. The corresponding room modes are written on the top of the graph and plate modes at the bottom.

The sound pressure in $V = 30$ m^3, $V = 40$ m^3 and $V = 50$ m^3 rooms were calculated utilizing Eqn. (3) at positions (0,7;0,7;0,7), (0;0;0) and (lx+0,5; ly+0,2; 1,5) from the centre of the room. A CLT and concrete plate with different dimensions were studied. The floors were excited with a 5 N point force from the middle of the floor. Material properties used in the calculations are presented in Tab. 1.

Table 1. Material properties.

Material	ρ [kg/m^3]	E [MPa]	ν [-]	η [-]
Concrete 200 mm [24]	2 500	33 000	0,2	0,015
CLT 200 mm [25]	420	4 539*	0,4	0,015

* Effective value $E_{eff} = \sqrt{E_x E_y}$

4. RESULTS

The magnitude of the sound pressure in the middle of the receiving room for concrete and CLT floors in 3 different room volumes are presented in Fig 1. The highest peaks in the magnitude of the sound pressure are on room modes f_{001} and f_{002}. The shape of the magnitude of the sound pressure is similar between the concrete and CLT slabs in same volume rooms but the magnitude is higher for the CLT slabs. The results are similar with other calculation positions and floor dimensions.

5. DISCUSSION

Neves e Sousa & Gibbs [7–8] found that the 1st vertical room mode f_{001} is the most significant for the ISI of homogeneous concrete floors. However, the calculations suggest that the 2nd vertical room mode f_{002} is significant as well. The peaks in the magnitude in Fig 1. correspond to a weaker sound insulation of the structure. The higher calculated magnitude of sound pressure for CLT floors also points to a weaker sound insulation in comparison to concrete slabs.

The problem with analytical models is the assumptions and simplifications done to derive the wanted equations. With other calculation methods, for example FEM calculations, complex boundary conditions can be considered. However, defining the boundary conditions can be difficult.

In FEM models the studied floor often radiates into an infinite space. Lietzén et al. [26] modelled the ISI of a full timber mock up floor that was also measured in a laboratory setting (V = 56 m^3). The simulated and measured results are in Fig. 2.

The simulated result differs from the measured result especially around 63 and 80 Hz octave bands. The first vertical room mode of the laboratory is f_{001} = 38 Hz and the second room mode f_{002} = 86 Hz. The second room mode could explain the difference in the low frequencies in the results shown in Fig. 2. Modelling the whole receiving room in a FEM software could lead to better agreement with the measured result on low frequencies.

To model the room and study the coupling further, the boundary conditions need to be determined with an adequate precision. Recent studies [27–32] utilising FEM modelling of the room sound field are transient or determine a single value impedance or absorption coefficient for all the surfaces of the room.

The boundary conditions that can be given for the room are impedance, sound hard and sound soft boundaries. Both sound hard boundary and sound soft boundary do not correspond to room sound fields in timber buildings.

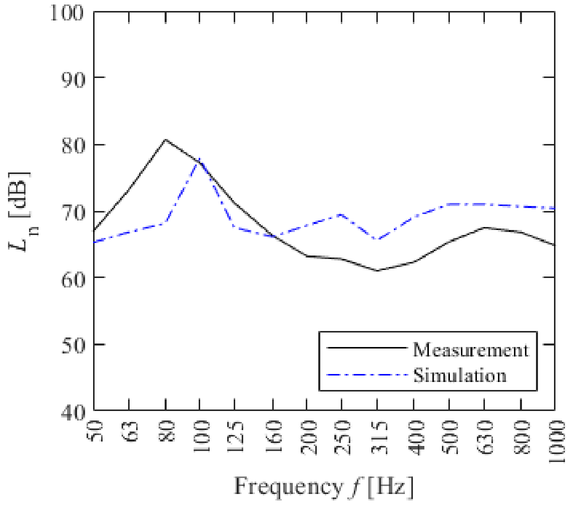

Figure 2. Measurement and simulation result of a timber mock-up floor [26].

Calculating the absorption coefficient from the plate resonator impedance does not correlate to the reverberation time in a room. The boundary condition can also be the absorption coefficient. This however only gives real value results though the reverberation time of the room would be closer to reality. Using the absorption coefficient also loses the information about the phase change.

6. CONCLUSIONS

The 1st and 2nd vertical room modes f_{001} and f_{002} cause the highest peaks in the sound pressure in the room causing the sound insulation of a plate to be weaker. Therefore, modelling the room could improve the agreement between measured and simulated ISI results on low frequencies. However, methods for defining non-rigid boundary conditions are limited to single number values of impedance or absorption coefficient. Other methods for defining non rigid boundary conditions should be investigated further.

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