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LIGHTWEIGHT FLOOR: STEP BY STEP COMPARISONS BETWEEN MEASURED AND SIMULATED QUANTITIES

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This paper is the result of collaborative work between European members of the Silent Timber Build project. In this work, step by step comparisons are undertaken in regard to measured and simulated mobilities as well as single number values (impact and airborne) for timber joist floors. Such an approach was proposed in order to dissociate, at low frequency, acoustical phenomena and purely mechanical concerns. Thus, aim is to be able to build a robust mechanical model of a complex floor and then to address the whole vibro-acoustic problem. As a first step, narrow band experimental information is used to assess a FEM model of the load bearing structure. Then, the complexity of the system is increased by successively adding layers on top (floating floor) and below (suspended ceiling). Similar comparisons are performed for each step.
Keywords: timber joist floor, mobilities, impact sound, airborne sound

1. Introduction

Lightweight systems in building constructions are complex assemblies combining the intrinsic properties of each component to meet various constraints: mechanical durability, thermal insulation, sound insulation etc. Figure 1 gives an example of typical elements of wooden joist floors which can be structural transmission paths inherent to most timber based designs.

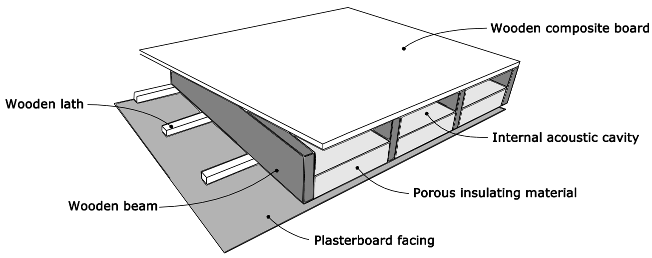

Figure 1 : Typical wood based lightweight system

The finite element method is attractive for modelling such systems due to its ability to solve boundary value problems within geometries of arbitrary complexity. However, in many cases, the main drawback of such an approach lies in a necessary modelling effort for every new geometry considered. A prerequisite to use FE models in the industry as a regular tool for building system design would consequently be to drastically reduce modelling efforts. Fortunately, typical building systems display rectangular geometries and components, periodic patterns and can be decomposed in regular layers. A high-level description of the geometry can be constructed in terms of parameters such as "layer thickness", "number of joists" etc. Then, such a high-level description can be interpreted by a dedicated FE code for the direct generation of the models and meshes. This was achieved and described in [1] and the obtained computational code is used in the following application.

2. Computational vibro-acoustic model

Typical wood-based systems can be decomposed into three classes of physical components: elastic solids that might be slender (facing panel, shear panel, etc.) or not (primary or secondary wood frames), poroelastic domains able to dissipate acoustic energy through thermal and viscous losses (mostly fibrous insulation materials) and acoustic fluid domains (air cavities). Classical linear theories [2], [3], dedicated to each class, can be used in order to build a coupled vibro-acoustic problem. The main difficulties for the creation of a mean relevant, vibro-acoustic model lie in the definition of material properties, assembly conditions and boundary conditions. Indeed, building materials are quite diverse and their precise elastic properties are rarely investigated apart from static resistance problems. Moreover, lightweight components and their assembly often are flexible. For example, when considering a board screwed onto a beam, point line or surface constraints can be applied depending on the assembly strength of the assembly connection. Thus, the ability of the finite element method to handle high complexity leads to many possible considerations. As soon how to model such details can be answered, the construction of the numerical model is straightforward.

2.1 Airborne sound insulation

Under laboratory conditions, airborne sound insulation is measured between a source room and the receiving room, separated by the evaluated system (see Fig. 2). According to the standard procedure [4], different loudspeaker positions have to be used and averaged for the generation of a steady sound pressure field in the source room.

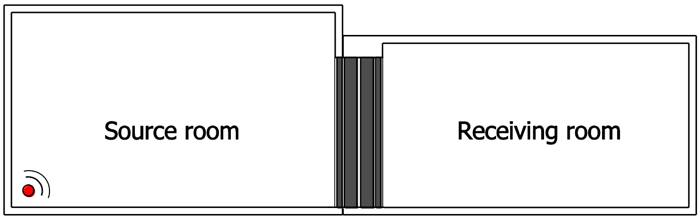

Figure 2 : Laboratory setup for airborne sound insulation

In order to evaluate the sound reduction index R, Sabine theory is classically used to link the sound power and the spatial average of the quadratic pressure field. Then, the sound reduction index reduces to

$$R = L_S - L_R + 10 \log_{10} \frac{S}{A}, \tag{1}$$

where L_S, L_R, S and A respectively denote the sound pressure level in the source and receiving room, system surface and equivalent acoustic absorption area. Hereinafter, the pressure field and the sound pressure level in the source room as well as the sound pressure level in the receiving room are evaluated in using a basis of analytical solutions of the Helmoltz equation in a rigid parallelepiped room. Such an approach was introduced in [1]

2.2 Impact sound insulation

Under laboratory conditions, the evaluated system is placed over a receiving room (see Fig. 3). A standard tapping machine is used for the generation of a quasi-steady mechanical excitation in various positions. Then, microphones in the receiving room provide the spatial sampling of the sound pressure field in the receiving room used to evaluate the spatial and time average of the quadratic pressure field. Consequently, a methodology similar to the one presented in for airborne sound insulation is used for the evaluation of the sound pressure level in the receiving room. In the following, we put the emphasis on the structural excitation, being a sequence of impacts resulting from the standard tapping machine [5], [6].

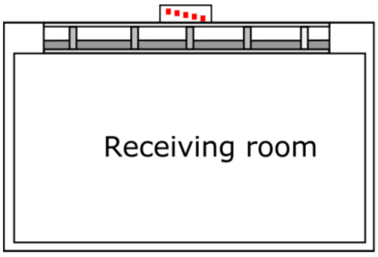

Figure 3 : Laboratory setup for impact sound insulation

2.2.1 Tapping machine model

According to [6], [7], the tapping machine consists of five equally spaced hammers with a mass $M_h = 0.5$ kg hitting the structure along a 40 cm line after a free fall from a height of $h_f = 4$ cm. Each hammer strikes the floor with the velocity $v_0 = \sqrt{2gh_f} = 0.886$ m/s (standard acceleration due to gravity $= 9.81$ m. s^{-2}) with a given time period $T = 0.5$ s and with a time shift from the previous one of $\Delta T = 0.1$ s. The force time signal $f_h(t)$ resulting from a periodic impact of hammer h can be expanded using a complex Fourier series such that :

$$f_h(t) = \sum_{n=-\infty}^{+\infty} f_n^h \, e^{i\frac{2\pi n}{T}t}, \tag{6}$$

where $\omega_n = \frac{2\pi n}{T}$. Thus, it can be noted that the force frequency spectrum takes discrete values for each hammer every 2 Hz. Moreover, low frequency bounds for f_n^h can be derived in using maximal and minimal momentum variations [7]. Fourier coefficients f_n^h are then comprised between $2M_h v_0/T$ and $M_h v_0/T$ respectively for an impact with rebound and an impact without rebound.

Let $F(\omega; \boldsymbol{x})$ be the whole external excitation field associated with the five hammers, we then have:

$$F(\omega; \boldsymbol{x}) = \sum_{h=1}^{5} \hat{f}_h(\omega) \; e^{-i\omega h \Delta T} \; \delta(\boldsymbol{x} - \boldsymbol{x}_h) \, . \tag{10}$$

where $\boldsymbol{x}_h$ denotes the impact position of hammer h.

It can be noted that every 10 Hz, the complex exponentials $e^{-i\omega h\Delta T}$ are in phase such that applied force and injected powers are then maximal. According to this model, the excitation spectrum takes discrete values every 2 Hz, with maximal values every 10 Hz. At very low frequencies, it can be understood that first structural resonances can either coincide with or occur between force maxima, then drastically modifying the measured performance.

2.2.2 Numerical evaluation of Fourier coefficient

The evaluation of Fourier coefficients f_n^h through Eq. (8) requires adequate sampling of the time signal $f_h(t)$. Within the context of lightweight building systems, and in particular for joisted floors, high spatial variations of input mobility can be observed. Thus, variations of impact forces can be expected to depend on the impact position. In the low frequency range, such variations belong to a 6 dB wide interval resulting from maximal and minimal theoretical momentum variation ($2M_h v_0/T$ and $M_h v_0/T$). Hereinafter, a finite element based method is undertaken for the sampling of the impact force time signal $f_h(t)$ at the impact position. Such an approach was introduced in [1].

3. Measurements

To provide real floor data for the verification of the calculation models, measurements in the laboratory on different floors were measured and provided by the project partners. In real situations, the sound transmission of a floor can be very complicated, as flanking transmission can have an influence on the transmission. Therefore, and because of the difficulty to perform measurements in real buildings, where all necessary information to simulate the floor is available, it was decided to perform measurements in the laboratory. Here the construction can be controlled much better and it is possible to perform more detailed measurements at a higher precision then in the field. The floor constructions were provided and installed in the laboratory by the German industry project partner Bauer Holzbau, and the aim was to use “standard” and “optimized” constructions, which are commonly used in Germany.

3.1 Description of the laboratory

The described measurements were conducted in the laboratory p8 of the Fraunhofer IBP in Stuttgart. The laboratory is made to test wooden floor constructions. It consists of concrete walls and floors and offers a frame, where a lightweight floor can be installed. All walls are equipped with lightweight linings with resonance frequency of approximately 60 to 80 Hz, reducing the flanking transmission in the frequency bands for standard testing from 100 to 5000 Hz. A sectional drawing of the laboratory is shown in figure 4. The room sizes are 4.78 m x 3.78 m x 3.82 m for the sending room and 4.78 m x 3.78 m x 2.67 m for the receiving room. All measurements were conducted on the basis of EN ISO 10140-4 [8].

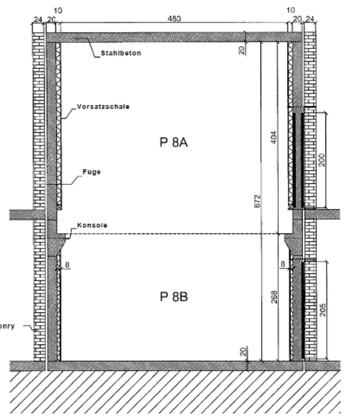

Figure 4 : Sectional view of the laboratory p8 of the IBP. The wooden floor construction was installed on the console, separating the laboratory into two rooms.

The measurements were performed with stationary microphones. The number of microphone positions in the receiving room was 6, in the sending room the number was 2. The number of loudspeaker positions in the sending room was 2. This leads to 12 independent measurements in the receiving room and 4 independent measurements in the sending room. The reverberation time was measured by the method of stationary signal suddenly turned off. In the receiving room, the measurement of the reverberation time was executed at 4 independent microphone positions and two different loudspeaker positions, giving a total of 8 independent measurements that were averaged.

3.2 Floor constructions

The load bearing structure consists of wooden beams with a layer of OSB panels on top. A multi-layered system is added on top, made up of a wood wool/cement board, mineral wool, drainage plate, OSB and cement screed. Finally, a ceiling closes the system and mineral wool fills the air cavity. Detailed description of the floors STB02 WG2 "*Validation of prediction tools and constructions*" that will be available on the project website [9].

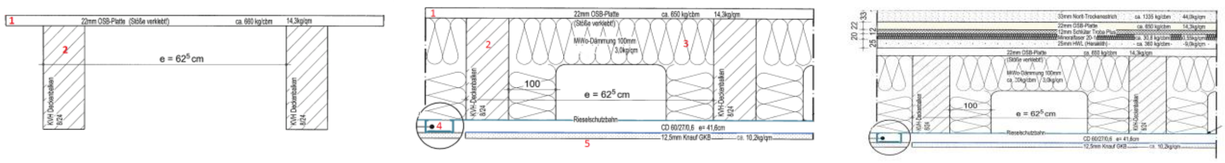

Figure 5 : Overview of measured floor constructions

Thus, the system complexity is increased step by step (see Fig. 5), from the raw load bearing structure to a complex system including floating floor and suspended ceiling. Indeed, few information is available for the direct construction of a numerical model for such a complex system. The knowledge of material properties, component connections and boundary conditions is incomplete and subjected to conjectures. Then, a step by step modelling process with experimental comparison is of great help to discriminate the influence of hypothesis and approximations.

4. Comparison of predicted and experimental values

Modelling details such as mechanical parameters can be found within report STB02 WG1 "*Modelling prerequisites – FEM/SEA Impact and Airborne Sound*". Some of the mechanical parameters, such as mass density or dynamic stiffness for floor underlayers were provided by product sheets. Elastic parameters however were mostly estimated from existing knowledge of materials. Boards are modelled as slender solids in using first order shear deformation theory. Wooden beams are modelled as three-dimensional elastic solids. Mineral wool as a filling material was modelled as an equivalent fluid with limp frame [3]. Mineral wool as an underlayer was modelled as an elastic solid whith elastic properties were estimated from dynamic stiffness data. For the boundary conditions, lateral displacements (but no rotations) of the board layers are blocked. The wooden beams were modelled simply supported on a rigid support. Boards are punctually and rigidly tied to wooden frame.

Due to the high sensitivity of predicted results to external excitation parameters (in particular the source position) the computation of airborne sound insulation is performed by using 1000 source positions and the computation of impact sound levels with 100 tapping machine positions. Uniform probability distribution are used for the generation of source positions. Then, 98% confidence regions can be evaluated by the quantile method and are displayed in red color within following figures.

4.1 Raw floor

For the raw floor, shown in figure 5 left, the results are shown in Fig. 6.

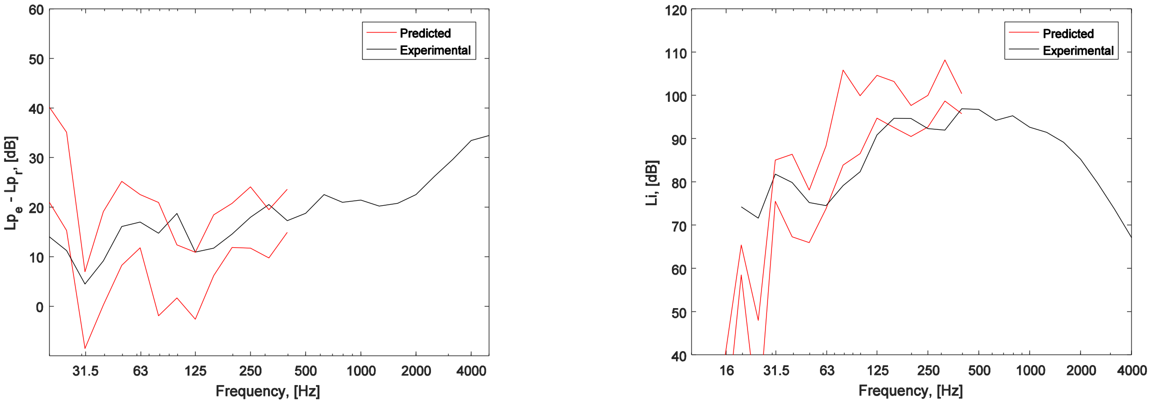

Figure 6 : Comparison of measured and computed airborne sound level difference (left) and impact sound level (right) for the raw floor

Below 50 Hz, the numerical prediction underestimates the actual impact sound level. A decrease of predicted sound pressure level is expected in the 25 Hz third octave band that corresponds to a minimum of magnitude for the overall tapping machine impact force. More, the five impact hammers spectra are in phase at the frequency and 10 Hz and multiples, consequently resulting in maxima of injected power at 20 and 30 Hz. However such a low level either means that 1) this model is not good enough for the description of the mechanical excitation resulting from the standard tapping machine or there is some inaccuracy in 2) the floor model or 3) the room model at the low frequencies. From 31.5 Hz upwards the model gives results close to measured values, from 63 Hz upwards the lowest bound is in good agreement to the measured values.

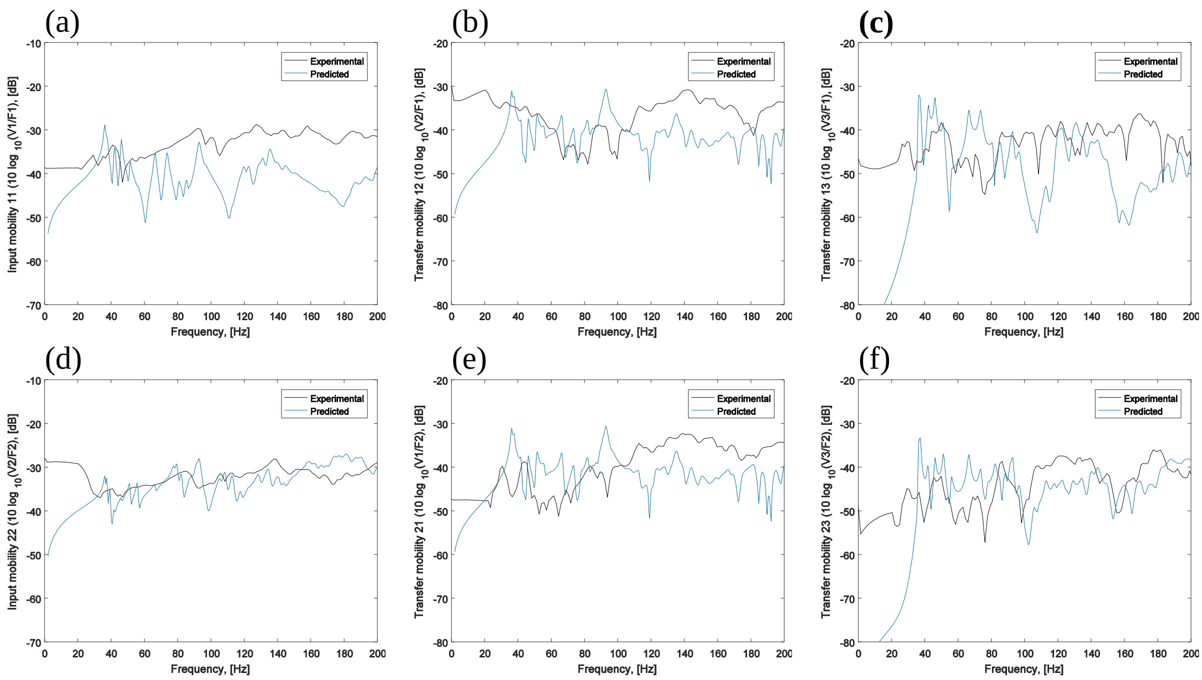

Figure 7 : Comparison of measured and computed mobilities for various points over the floor

In Figure 7, detailed results of input and transfer mobilities were provided for points 1, 2 and 3, where point 2 is between beams and point 1 and 3 over different. The numerical model underestimates the input mobility on top of beams (a) but gives a rather accurate level for the point between beams (d). This means that, despite having considered point connections between boards and beams, the local board stiffness might be overestimated. The most interesting output from those measurements is perhaps the high modal density and damping that can be observed in this frequency range. Below 50 Hz, it can be seen that the first global modes are highly damped/little responding. Thus, it can be assumed that structural connections are not stiff enough to yield an actual global modal behaviour. In fact, the simple raw floor doesn't display a strongly marked modal behaviour even in the lowest frequency range. Increasing complexity of the construction will mostly add mass and damping thus definitely settling the system in a "medium frequency like" behaviour.

4.2 Raw floor and ceiling

For the raw floor plus suspended ceiling, shown in figure 5 middle, the results are shown in Fig. 8. From 31.5 Hz upwards, the increased airborne and impact sound insulation due to the ceiling is consistent between predicted and experimental values. However, bellows 31.5 Hz discrepancies are still observed and even increased regarding the impact sound level. Figures 6 and 8 might indicate that the model is overall underestimating the actual performance above 31.5 Hz as experimental values are respectively closer to the higher bounds for airborne sound insulation and lower bounds for impact sound insulation. Below 31.5 Hz quite the opposite is observed.

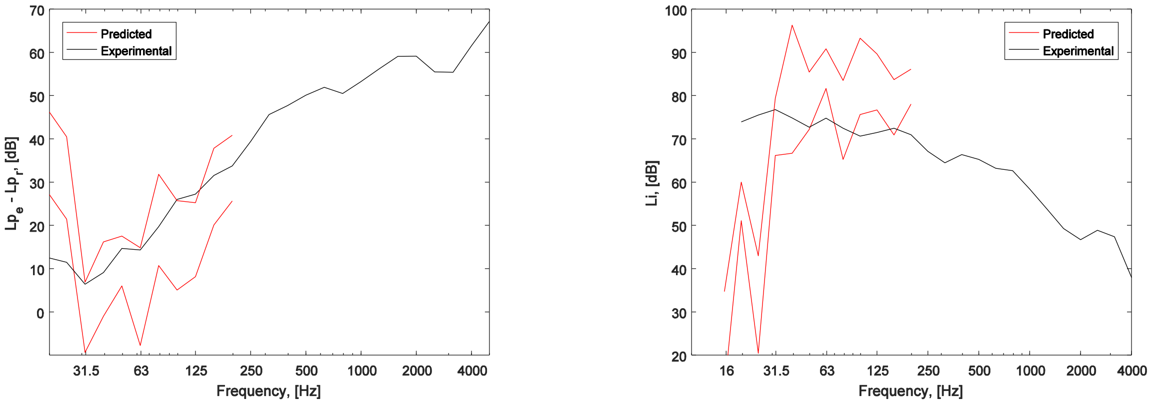

Figure 8 : Comparison of measured and computed airborne sound level difference (left) and impact sound level (right) for the raw floor plus suspended ceiling

4.3 Multilayer and ceiling

For the whole floor including suspended ceiling and a multi-layer treatment, shown in figure 5 right, the results are shown in Fig. 9. From 31.5 Hz upwards, the increased airborne and impact sound insulation is consistent between predicted and experimental values. Moreover, confidence regions look tighter than within previous computations. The fast decay of the impact sound pressure due to the addition of the mulit-layer system on top of the floor is well predicted

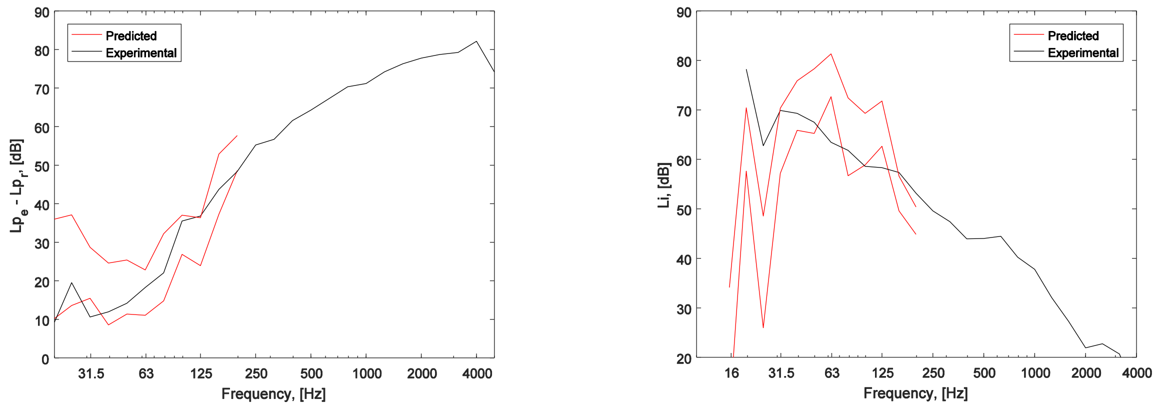

Figure 10 : Comparison of measured and computed airborne sound level difference (left) and impact sound level (right) for the whole floor

5. Conclusions

A step by step comparison of predicted as well as measured quantities was undertaken, below 200 Hz, for wood based systems of increasing complexity. Regarding parameters such as source positions, confidence regions were constructed. Consistent trends can be observed above 50 Hz for airborne as well as impact sound insulation. However, the numerical prediction systematically underestimates the impact sound level in the 20 and 25 Hz third octave bands.

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