FEM based prediction model for the impact sound level of floors

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Up to now the research and development in the field of building acoustics is based mainly on measurements. The consequence is that the development and optimization of a new building component is a very tedious and expensive task. A considerably reduction of these costs could be achieved, if the optimization relying on measurements would be replaced – at least to some extent – by a computational prediction model. Motivated by these aspects a method is presented for using finite element techniques to estimate the impact sound level from lightweight floors. The overall approach consists of the three-dimensional modeling of the structure and the excitation source (tapping machine), the subsequent modal- and spectral analyses and the computation of the radiated sound from the ceiling.

1 Introduction

Commonly used prediction models are based on measured data for the component parts under consideration. These models are very useful for the evidence of performance of known building components [1], but they are not suited for the development of new components. An alternative approach in this context is the application of the finite element method (FEM) to the propagation of the sound transmission as shown in this contribution. The paper shows first an overview of the approach and deals then more elaborated with the excitation force of the floor, the modeling of the damping mechanism, and the radiation effects. It closes with some examples for the application of the model to the impact sound propagation of lightweight floors.

2 Survey of computation

The overview of the computation model, given in Figure 1, could be divided in the following steps:

Excitation of the impact-sound

In the experimental rating of the impact sound level of a floor the tapping machine is most commonly used as the excitation source. For the computation model the excitation force has to be expressed with respect to the interaction between the hammer of the tapping machine and the floor surface at the driving point.

Modeling of the structure

The thin-walled lightweight floor, consisting of plates and beams is discretized with a fully three-dimensional approach, where anisotropic high-order solid finite elements are applied allowing different polynomial degrees for each direction of the element [2].

Modal analysis

In the modal analysis the eigenvectors of the problem are computed and used to decouple the system of differential equations. In the second step the differential equations are Fourier-transformed and solved in the frequency domain. Due to the fact that the impact sound level of lightweight floors is dominated by transmissions at very low frequencies, the computation can be restricted to a small number of eigenvalues.

Impact sound spectrum

The response of the structure to the excitation force spectrum with respect to the modal damping of the structure is computed in the frequency domain by summing up the result of each transformed differential equation. This computation is done for each excitation-position of the tapping machine and each considered radiation-point of the structure.

Radiation of the impact sound

For the assessment of the impact sound insulation of floors the normalized impact sound pressure level L_n is calculated from the sound pressure level in the receiving room. In the prediction model, this quantity is computed from the radiated impact sound spectrum in different ways.

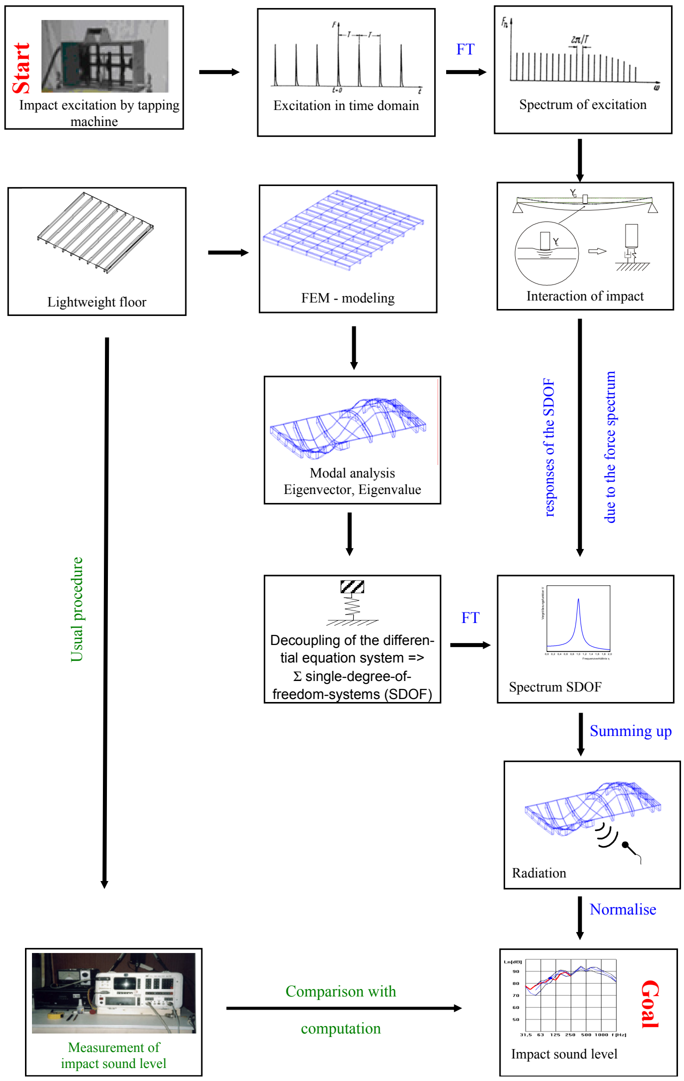

Fig. 1: Workflow of computation

3 Excitation of the impact-sound

The standard tapping machine according to ISO 140-06, consists of five steel cylinders (hammers) with the mass of $M_0 = 0.5$ kg per hammer. Each hammer hits the floor after a free fall of $h = 0.04$ m two times per second. Fig. 1 (top) shows the tapping machine and its excitation force in the time and frequency domain. Expressions for the force spectrum due to the impact of the hammer can be derived by the Fourier transformation of the force pulse in the time domain. The problem to solve in this context is the interaction of the hammer with the floor structure during the contact time, which shows a strong material dependency. Several approaches were made to solve this problem [3,4,5,6,7] starting by *Cremer* 1948 who expressed the force spectrum of the impact by the momentum of the hammer.

Based on the literature solutions the model presented here describes the impact of the hammer on the floor by a two degree of freedom system as shown in Fig. 2.

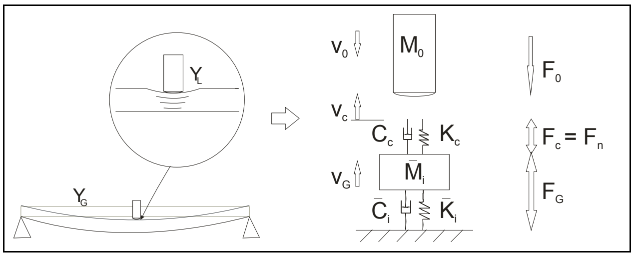

Fig. 2: Description of the excitation by the local and global parts of the admittance Y at the driving point

The upper part of the system, shown in Fig. 2 as the local admittance Y_L, represents the effects at the driving point of the hammer impact. The hammer shows a free, damped oscillation for half of a period (as the time of contact) with the velocity v_0 as initial condition. The lower part, representing the floor, shows a forced damped oscillation with the velocity v_i. It could be expressed as the global admittance $Y_{G,i}$ for each eigenmode of the floor.

Considering the frequency the equilibrium of forces and continuity of velocities in Fig. 2 yields:

$$F_0 = M_0 j\omega_n v_0 + F_n;\ \ v_c - v_G = F_n Y_L;\ \ F_n = \frac{v_G}{Y_G} + F_R \quad (1)$$

This could be rewritten for the time of contact ($v_0 = v_c$) as:

$$F_n = \underbrace{F_0 - F_n Y_L M_0 j\omega_n}_{F_{n,L}} - \underbrace{F_n Y_G M_0 j\omega_n}_{F_{n,G}} + \underbrace{F_R Y_G M_0 j\omega_n}_{F_{n,R}} \quad (2)$$

The time-dependency of the system refers to the local admittance and therefore to $F_{n,L}$ as the local part of the excitation spectrum. With the assumption that the influence of the global admittance to the contact time is small we can compute $F_{n,L}$ as given by *Vér* in [5] with additional respect to the local damping D_c:

$$F_{n,L} = \frac{1}{T}\int_0^T F(t)\, e^{-j\omega_n t}\, dt = \frac{1}{T}\int_0^{T_c} M_0 \dot{v}(t)\, e^{-j\omega_n t}\, dt \quad (3)$$

$$v(t) = \hat{v}_0 \cos(\omega_c t)\, e^{-D_c \omega_c t};\qquad \omega_c = \sqrt{K_c / M_0}$$

The global part of the excitation force spectrum $F_{n,G}$ represents the reaction of the floor to the impact. It can be computed for each eigenmode φ_i by:

$$F_{n,G} = M_0 F_n j\omega_n \sum_{i=1}^{n_{eigen}} Y_{G,i}\ \varphi_i^2(x_0, y_0) \quad (4)$$

Thus Eq.(2) gives for the first impact of the hammer ($F_{n,R} = 0$):

$$F_n = \frac{F_{n,L}}{1 + M_0 j\omega_n \sum_{i=1}^{n_{eigen}} Y_{G,i}\ \varphi_i^2(x_0, y_0)} \quad (5)$$

Due to the fact that the measurement of the impact sound level is carried out in a steady state vibration of the floor, we have to respect a relative velocity between the dropping hammer and the vibrating floor, which is represented in Eq.(2) by the force $F_{n,R}$.

$$F_{n,R} = M_0 j\omega_n \sum_{i=1}^{n_{eigen}} v_{R,i} \quad (6)$$

The vibration velocity v_R results from the previous hammer –impacts. It is reduced due to the structural damping during the free oscillations between two impacts.

Comparison with measurements

There are two limiting conditions for the described system. The first case considers very stiff floors ($Y_G \to 0$) where just the upper part of the system in Fig. 2 is relevant. A comparison of measured and computed data for this case is shown in Fig. 3. For the measurement the force transducer was placed between a chip board and a stiff base and driven by a single hammer of the tapping machine.

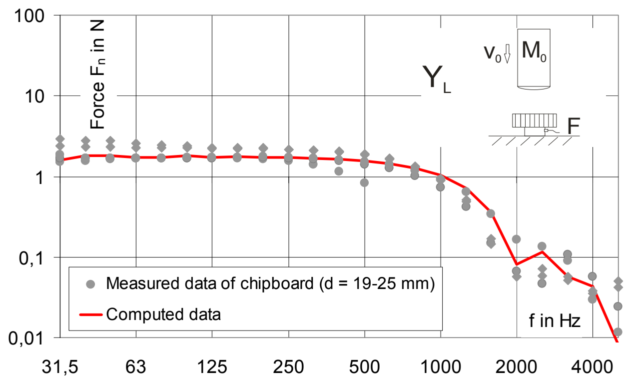

Fig. 3: Excitation force of the tapping machine (single hammer, f = 2 Hz) considering local effects at chipboards with a thickness of 19-25 mm on a stiff base.

The second case considers floors with very hard surfaces ($Y_L \to 0$) where just the lower part of the system in Fig. 2 is relevant. For this case the force transducer was placed on the floor and directly driven by the hammer of the tapping machine (metal – metal). The computation was carried out for the first eigenmode of the floor planking. The results of computation fit very well to the measurements.

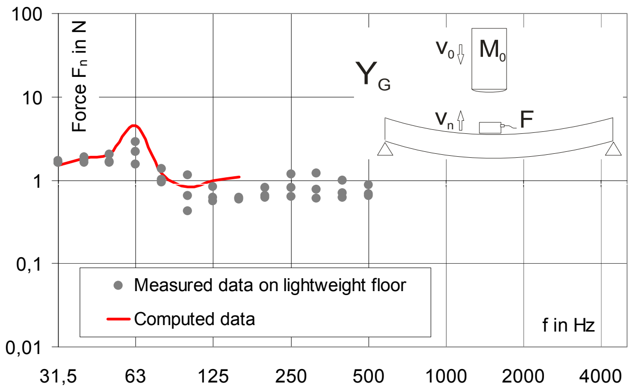

Fig. 4: Excitation force of the tapping machine (single hammer, f = 2 Hz) considering global effects of a lightweight floor for the first eigenmode of the planking at f_0 = 80 Hz (measured 63 – 100 Hz).

4 Structural damping

The model of the damping behaviour is required in the spectral analysis to compute the amplitudes of the floor vibration. In building acoustics it is common to use the loss factor η for describing the damping of a structure. It is defined as the ratio of the energy loss E_V per period (2π) and the stored (reversible) energy E_R in a dynamic system.

$$\eta_{structure} = \frac{E_V}{2\pi E_R} \tag{7}$$

The structural damping of a lightweight floor consists of several parts as illustrated in Fig. 5.

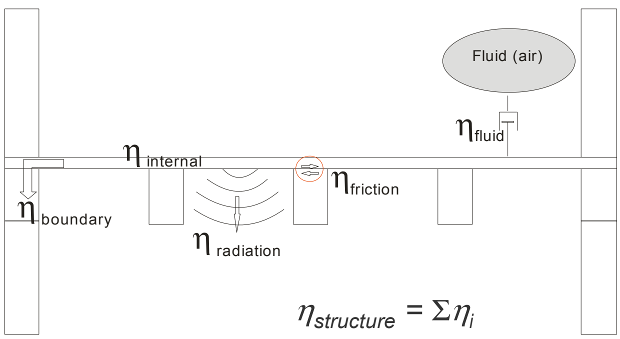

Fig. 5: Different parts of the loss factor

Internal losses $\eta_{internal}$

The internal losses are describing the material damping which can be measured experimentally for different materials. In case of a structure with different materials – as the lightweight floor in Fig. 5 - the internal loss factor for the whole structure is required. It can be computed as the ratio of the total strain energy of the different materials $E_{pot,l}$ multiplied by the internal loss factors $\eta_{internal,l}$ and the strain energy in the whole structure $E_{pot,ges}$ multiplied by the unknown internal loss factor $\eta_{internal,ges}$ (see [8,9]).

$$\eta_{internal,ges} = \sum_{l=1}^{n} \eta_{internal,l} \frac{E_{pot,l}}{E_{pot,ges}} \tag{8}$$

Boundary losses $\eta_{boundary}$

The boundary losses are related to the energy flow via the supports of the floor. In case of a well known test facility empirical data can be used. Otherwise they can be computed by Eq.(7). The energy losses E_V are given by the strain energy in the support and the flanking wall respectively. The stored Energy E_R is given by the strain energy in the floor.

Frictional losses $\eta_{friction}$

The consideration of the energy dissipation in frictional joints was carried out in a simplified model based on the frictional force $F_N\,\mu$ and the relative movement between the planking and the beams, see Fig. 5.

$$E_V = \int |F_N|\,\mu\,\xi'\,dx \tag{9}$$

Radiation losses $\eta_{radiation}$

The loss factor which considers the energy losses due to the radiated sound is given in [11] as a function of the radiation loss factor σ and the area related mass m' of the radiating surface.

$$\eta_{radiation} = \frac{2\rho_0 c_0 \sigma}{2\pi f m'} \tag{10}$$

Fluid damping η_{fluid}

An empirical approximation for the fluid damping by the air, depending on the mass m' of the radiating surface, is given by *Müller* [10] with:

$$\eta_{fluid} \approx \frac{0,00273}{m'\sqrt{f}} \tag{11}$$

Comparison with measurements

The loss factors of a lightweight floor from a computation are shown in Fig. 6. A comparison between the measured and the computed total loss factors are shown in Fig. 7.

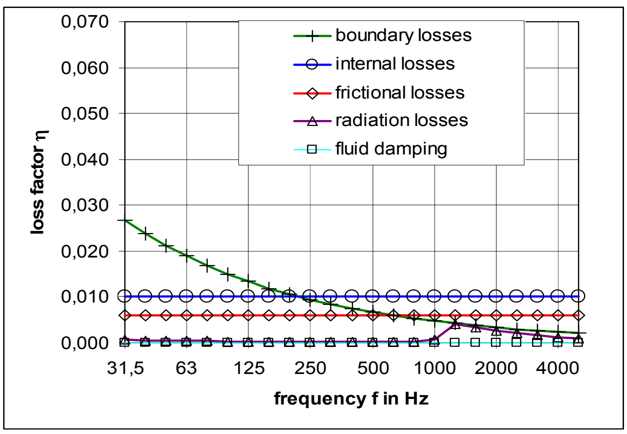

Fig. 6: Computed contributions to the structural damping

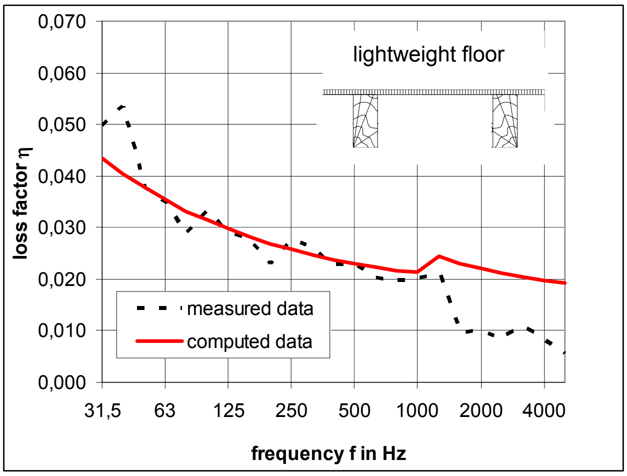

Fig. 7: Comparison between measured and computed loss factor

5 Radiation of impact sound

In the last step of the approach the radiation of the impact level is considered. The methods proposed in the literature which are dealing with the radiation can be divided in two groups. First the fully coupled methods, where beside the radiation of the structural vibration into the room also the influence of the resulting sound pressure (in the room) to the structure is considered. Second the weakly coupled methods which neglect this influence.

For the application presented here two methods of the weak coupled group are used. Therefore the structural vibration of the floor was computed in the first step. Then the known vibrating velocity v of the floor surface is used to compute the sound pressure in the room (with fluid F) as shown in Fig. 8.

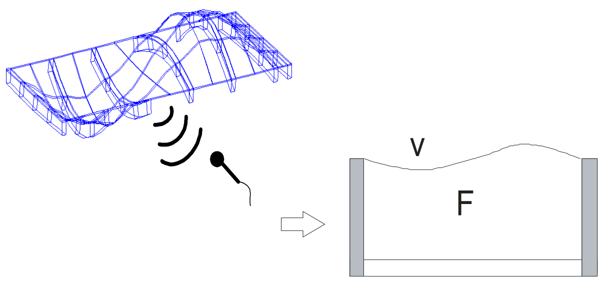

Fig. 8: Radiating sound of the vibrating structure into the room

Modal method

The first method is based on the Helmholtz equation and makes use of the room's modal solution. The unknown sound pressure p in any point (x,y,z) of the room can be computed by the known velocity v_n of the radiating surface and the Green elementary solution G_n of the room [12].

$$p_n(x,y,z) = j\omega_n \, \rho_F \int_S G_n v_n \; dS \tag{12}$$

The Green elementary solution is given by the admittance Y and the eigenmodes Φ of the room at the surface of the structure and at the point (x,y,z).

$$G_n = \sum_{i,k,l} \Phi_{i,k,l}(x,y,z) \; \Phi_{i,k,l}(S) \; Y_{i,k,l} \tag{13}$$

The eigenmodes Φ of the room are given by the dimensions of the room.

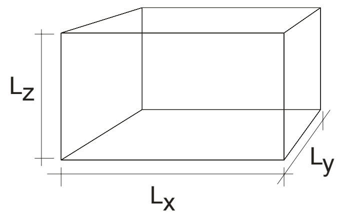

Fig. 9: Rectangular room with the dimensions L_x, L_y, L_z

$$\Phi_{i,k,l} = \cos\left(\frac{i\,\pi\,x}{L_x}\right)\cos\left(\frac{k\pi\,y}{L_y}\right)\cos\left(\frac{l\,\pi\,z}{L_z}\right) \tag{14}$$

Integral method

The second model discretizes the vibrating area of the floor - as shown in Fig. 10 – in sound radiating monopoles [13].

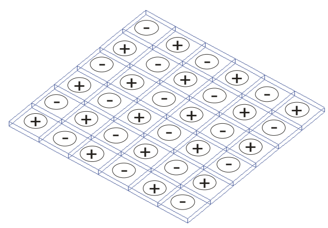

Fig. 10: Illustration of the integral solution

The radiated sound power P is computed from the direct radiated sound power of each monopole P_i and the interaction between the monopoles $P_{i,l}$.

$$P = \sum_{i=1}^{N} P_i + \sum_{i=1}^{N}\sum_{l=1}^{N} P_{i,l} \tag{15}$$

P_i is given by the impedance $\rho_F\, c_F$ and the wavenumber k of the fluid (air), the discretized surface element ΔS and the vibration velocity v. $P_{i,l}$ is additionally given by the distance $d_{i,l}$ and the phase α between the monopoles.

$$P_i = \frac{\rho_F c_F}{2\pi} k^2 (\Delta S_i v_i)^2 \tag{16}$$

$$P_{i,l} = \frac{\rho_F c_F}{2\pi} k^2 (\Delta S_i v_i)(\Delta S_l v_l) \frac{\sin kd_{i,l}}{kd_{i,l}} \cos(\alpha_{i,l}) \tag{17}$$

6 Examples

As a first validation of the method some comparisons between the measured and the computed impact sound level of lightweight floors are shown. The first example in Fig. 11 indicates a simple lightweight floor consisting of timber beams and chipboard planking. The diagram displays the measured data of similar floors in different test facilities and the computed data with the integral and the modal radiation method. Fig. 12 shows this comparison for a laminated timber floor with a floating floor screed on mineral wool and crushed stones as loading of the floor. The agreement of data from computation is good and lies within the well known scatter in acoustic measurement results of these test elements.

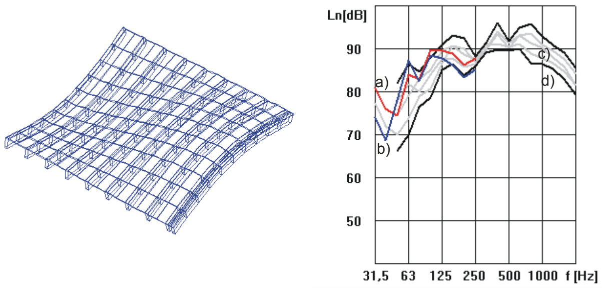

Fig. 11: Comparison measurement – computation for a simple lightweight floor. a) computation using an integral method for the radiation b) computation using a modal method for the radiation c) measured data of similar floors in different test facilities d) mean value of measured data ± 2σ

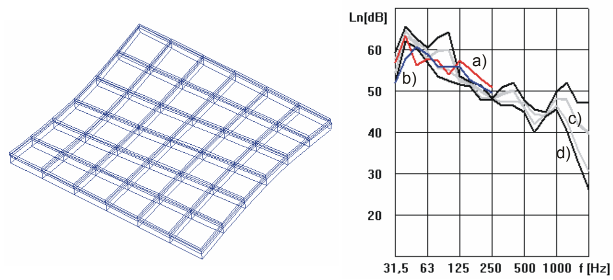

Fig. 12: Comparison measurement – computation for a laminated timber floor with a floating floor screed on mineral wool and crushed stones as loading.
a) computation using an integral method for the radiation b) computation using a modal method for the radiation c) measured data of similar floors in different test facilities d) mean value of measured data ± 2σ

7 Conclusion

A method is presented to estimate the impact sound level from lightweight floors by using finite element techniques. The approach consisting of the three-dimensional modeling of the structure, the modeling of the excitation source (tapping machine), the subsequent modal and spectral analyses and the computation of the radiated sound. It shows a good accordance between measured and computed data of the impact sound level.

Acknowledgments

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