Radiation Efficiency Of Cross Laminated Timber Panels By Finite Element Modelling

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1 Introduction

Tall wood buildings are gaining an increasing interest in North America due to the recent development of mass timber panels and construction techniques. Cross laminated timber (CLT) is one of the most popular mass timber products, which consists of an odd number of orthogonal layers of dimensional lumber planks glued together by a structural adhesive. The cross-laminated layered structure in CLT makes it a dimensional stable engineered wood panel, but also an orthotropic plate in terms of its elastic properties. Its relative high strength/stiffness to density ratio is beneficial for structural seismic resilience, however, also causes challenges in meeting sound insulation requirements. Both the airborne and impact sound insulation performance of CLT panels have been tested to be low [1].

The sound insulation of a building element depends on its dynamic response to the actual excitation of either acoustic field or mechanical force and the efficiency as a sound radiator given the actual response pattern [2]. As an orthotropic material, the sound radiation behavior of CLT plates is not well understood. This study utilized a finite element (FE) modelling approach using ABAQUS FE software to investigate the sound radiation efficiency (RE) of typical Canadian CLT panels with experimental elastic constants.

2 Method

2.1 Theory

The RE of a given vibrating plate in a fluid is defined by:

$$\sigma = \frac{\bar{W}}{\rho_o c_o S\langle\overline{v^2}\rangle} \text{ or equivalently } \sigma = \frac{\langle\bar{I}\rangle}{\rho_o c_o\langle\overline{v^2}\rangle} \qquad (1)$$

where $\bar{W}$ the time-average radiated sound power, ρ_o and c_o are the fluid's density and sound speed, S is the plate surface area, and $\langle\overline{v^2}\rangle$ is the spatially averaged (temporal) mean-square velocity component normal to the plate surface. The term $\langle\bar{I}\rangle$ is the spatially averaged time-average sound intensity component normal to the vibrating plate.

2.2 Finite element modelling

To investigate the RE of a vibrating plate, we considered a plate with specific boundary conditions (BCs) at its middle surface situated above a semi-infinite (rigid baffle) acoustic medium (air) under forced vibration by harmonic loads. The plate was modeled as thick shell (4-node elements full integration) being coupled with the semi-infinite acoustic medium through solid-acoustic interaction. The acoustic medium contained three regions: a cubic region of fine mesh right below the plate (fine mesh assured accurate output), a spherical-segment region of coarse mesh (this region directed waves to the virtual extension of the medium), and the virtual extension of the medium to the infinity. The first two regions were modeled by tetrahedral acoustic elements (4-node) and the virtual region by acoustic infinite elements. Fig. 1 demonstrates the configuration of the FE model. The model was analysed under steady-state harmonic conditions. The loads were harmonic point loads at certain locations, and with the frequencies in the range of 10-1000Hz with a 3-Hz step. Loads with different locations were separately modeled and their resultant RE values were averaged at each frequency.

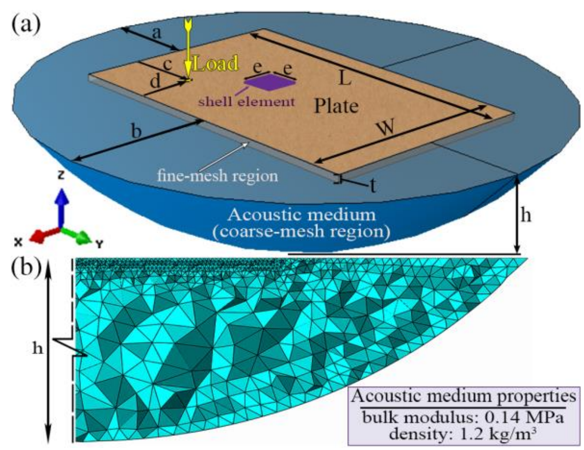

Figure 1: (a) the configuration of the FE model, (b) typical element size transition in the models.

The direct outputs of the modeling software were the complex-valued acoustic intensity vectors of the elements (at their centroid) below the plate, and the complex-valued nodal velocity vectors of the plate. The real part of the normal (to the plate) component of the intensity vector is equal to the time-average intensity ($\bar{I}$) as in Eq. 1. The half of the squared magnitude of the complex normal component of the velocity vector equals to $\overline{v^2}$ in Eq. 1.

The FE element size and the size of the explicitly modeled acoustic medium determine the accuracy of the FE model. We chose the largest element size of each medium equal to 0.1 of the smallest wavelength (λ) of propagating waves in the medium; but for the coarse-mesh region, it contained element size transition from 0.1λ to 0.3λ (Fig. 1b). Mesh sensitivity analyses on the element and acoustic

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medium sizes were performed and the accuracy of the results, for the purpose of model verification, was assessed by comparing the results with an analytical model regarding RE of an isotropic aluminum plate [3]. Subsequently, CLT panels were modeled as orthotropic plates coupled with the acoustic medium. The dimensions, material properties, and load cases of the plate are listed in Table 1.

Table 1: Properties of the models for verification of the FE models.

Model: 80mm CLT plate [3]
BCs1: all edges simply supported; BCs2: all edges fully clamped
$E_X = 10800MPa$, $E_Y = 734MPa$, $G_{XY} = 600MPa$, $G_{XZ} = 240MPa$, $G_{YZ} = 572MPa$, $\nu_{XY} = 0.02$, $\rho = 484\,kg/m^3$, $\mu = 0.06$
$W = 2900mm$, $L = 4200mm$, $a = 1220mm$, $b = 1870mm$, $t = 90mm$, $h = 1660mm$, $e \approx 30mm$
Load case 1: $c = 500mm$, $d = 800mm$
Load case 2: $c = 3600mm$, $d = 900mm$
E, G: Young's and shear moduli, ν: Poisson's ratio, ρ: density, μ: loss factor.

Following the verification of the results of the 80mm CLT plate, we modeled Canadian CLT plates (Table 2). The acoustic medium was scaled to fit the plates.

Table 2: Properties of the Canadian CLT plates' models.

Canadian CLT plates ($W = 3660mm$, $L = 2440mm$) BCs: all edges simply supported						
Model	E_X (MPa)	E_Y (MPa)	G_{XY} (MPa)	G_{XZ} (MPa)	G_{YZ} (MPa)	t (mm)
3ply	10393	711	518	240	300	105
5ply-1	9657	2293	542	110	360	131
5ply-2	9368	2201	543	200	350	175
7ply	9873	2488	338	300	400	220
$\nu_{XY} = 0.02$, $\rho = 520\,kg/m^3$, $\mu = 0.08$, $e \approx 30mm$; Load case 1: $c = 1830mm$, $d = 1220mm$; Load case 2: $c = 915mm$, $d = 610mm$						

3 Results and Discussion

3.1 Verification of CLT plate with literature

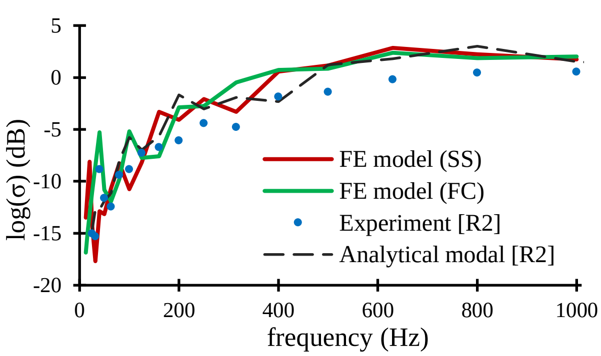

Figure 2: Comparison of the simulated CLT plate RE (σ) with the experimental and analytical values [4]. SS: simply supported, FC: fully clamped.

A general agreement can be found among the results in Fig. 2, but experimental values are always lower than modelled values with frequency higher than 200 Hz. The boundary conditions affect the RE values at certain frequencies.

3.2 Application to Canadian CLT plates

As shown in Fig.3, the RE values of different CLT plates varies under the critical frequencies, which is around 400 Hz for the 5 and 7-ply CLT plates. The 3-ply CLT plate has lower RE values than the others under the same frequency band with higher critical frequencies around 600 Hz. Due to the difference in layup and thickness, the elastic constants and area density varies among these CLT plates. The 3-ply CLT plate has less orthotropy than the other three plates. The other three plate have closer elastic constants and especially higher E_Y values than that of the 3-ply CLT. The effect of area density plays a major role in the difference among 5 and 7-ply CLT plates.

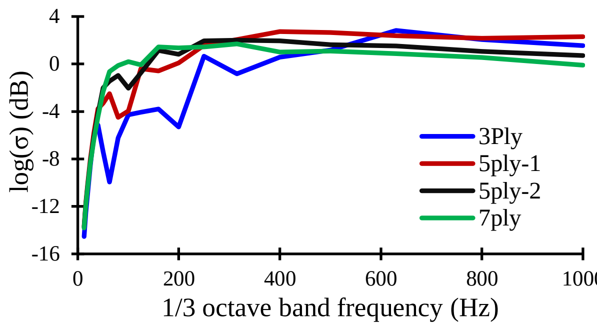

Figure 3: Simulated RE of Canadian CLT plates.

4 Conclusion

The proposed FEM approach is effective in evaluating the radiation efficiencies of different CLT panels. The elastic constants and boundary conditions are critical to the accuracy of the model. Further study will focus on the experimental verification and characterization of these effects.

Acknowledgments

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