PREDICTION OF THE DYNAMIC PROPERTIES OF A CROSS LAMINATED TIMBER PLATE FROM AN INVESTIGATION OF THE EIGENMODES USING A SCANNING LASER VIBROMETER

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1 INTRODUCTION

Cross Laminated Timber (CLT) panels of multiple layers are used in Swiss building constructions. There are a variety of dynamic and static methods to determine the elastic constants of a plate e.g. time of flight measurements to determine longitudinal wavespeed [1], determining the elastic constants by optimisation of a Finite Element Method (FEM), direct measurement (*in situ)* of the bending wavespeed [2], and static measurement using a four point bending stiffness method [3]. Previous studies have shown that for cross laminated timber, determining the stiffness of whole panels is preferable to strip samples because the bending stiffness of CLT varies within a single panel, particularly panels with large lamellas, local non-homogeneities and defects, and the panel stiffness can also depend on the strength grading method of the raw material. Therefore strip sample testing requires a number of narrow strips (e.g. 5 or 6 100mm strips) or wider (e.g. 300mm) strips [3]. The advantage of the first three methods described above is that they can be performed on a complete CLT plate. The direct measurement of bending wavespeed can also be done *in situ*. In previous work [1] the first two methods were compared, and it was shown that time of flight measurements are not suited to multilayer anisotropic panels. This paper takes a closer look at the second and third methods; that is the determining the elastic constants by optimisation of a FEM and direct measurement of the bending wavespeed. A dynamic assessment of strip samples of a nominally identical panel is described in [4]. The aim is to estimate the elastic constants of CLT plates in order to provide more detailed input data to predict the sound insulation.

2 CLT PLATE PROPERTIES

Cross laminated timber is assembled from crosswise layers of lamellas into whole panels. The panel dimensions were 4.2 m x 2.9 m and consisted of three glued layers of small strips of visually graded Norwegian Spruce of density 438 kgm-^{2}. The total mass of the panel was 455 kg and the thickness of the sample was 80 mm, with the surface layers being built up of wood strips, 15 mm thick by 27 mm wide and the middle layer, being built up of wood strips, 50 mm thick by 27 mm wide. The raw material was of strength class C24 (CEN EN 338 (2003)), with elastic moduli in the parallel and perpendicular directions of $E^{\parallel}$=12.0x10^{9} Nm^{-2}, $E^{\perp}$=4.0x10^{8} Nm^{-2} respectively.

3 DETERMINING THE ELASTIC CONSTANTS BY OPTIMISATION OF A FEM

The FEM model was constructed using ABAQUS/CAE 6.14-2 software. In a conventional shell element the elastic properties of an orthotropic plate are fully described by four elastic constants E_x, E_y, ν_{xy}, and G_{xy}. ABAQUS also requires the shear moduli G_{xz} and G_{yz} to model transverse shear deformation through the thickness of a conventional shell. Displacements through the thickness can be calculated before or during the analysis. However these shear moduli G_{xz} and G_{yz} can be assumed to have little effect on the first few modes and were taken from nominally identical panels that were previously measured using bending tests [3]. Poisson's ratio ν_{yx} can be calculated from $\nu_{yx}=(E_y/E_x)\nu_{xy}$. A 25 x 25 mesh of rectangular S4R elements was compared with triangular STRI3 elements which enforce the Kirchhoff constraint. Based on calculation of the bending wavespeed there were more than six elements per wavelength up to ~1000Hz.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} 1/E_x & -\nu_{xy}/E_x & 0 \\ -\nu_{xy}/E_x & 1/E_y & 0 \\ 0 & 0 & 1/G_{xy} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{Bmatrix} \qquad [\mathrm{Nm^{-2}}] \qquad (1)$$

The CLT panel was suspended from a crane in the laboratory and scanned with a laser vibrometer (Polytec PSV-400 scanning head) to measure orthogonal velocity with an average of 5 measurements at each of 13 x 9 positions. There were more than six grid points per wavelength up to ~100Hz. Two shaker positions were used to excite the panels using a pink noise signal.

A simple iterative process [1] was used to estimate E_x, E_y and G_{xy} in FEM for the three lowest measured eigenfrequencies: f_{11}=8.2Hz, f_{20}=12.5Hz, and f_{02}=41.9Hz (to within 0.1Hz). The 50 lowest eigenfrequencies are plotted in Figure 2 for thick and thin plate models. For the first 11 frequencies (<100Hz) the thin plate model is adequate. Above the 35th mode the (>266 Hz) the thin plate model fails due to shear locking, further refinement of the mesh does not extend this upper bound.

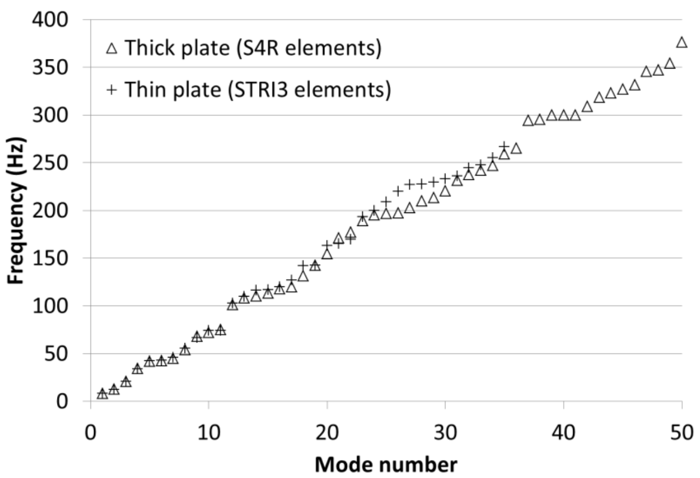

Figure 2: Eigenfrequencies of the thin and thick plate FEM models.

4 POISSON'S RATIO

Literature on Poisson's ratio for spruce in tension is extensive – see survey by Kohlhauser & Hellmich [5]. Figures 1 (a) and (b) show the alignment of the lamellas in the CLT with respect to the grain. Strength properties of the wood that were assumed to be the most significant here are the longitudinal (L) direction (aligned with the grain) and tangential (T) to the growth rings (Figure 1 (c)). Hence Poisson's ratios used for calculation were set to the mean of the values for Spruce in the L and T directions, ν_{LT}=0.486 and ν_{TL}=0.026 (first subscript denotes the passive direction).

Correct determination of Poisson's ratio for the whole panel is crucial to obtain good agreement with the higher modes [1]. One approach is to calculate Poisson's ratio of an equivalent single layered shell from the individual layers as carried out by Jones & Klein [6]. Calculating the equivalent strain energies of the layers which are symmetric about the middle surface the equivalent Poisson's ratio is defined as

$$\nu_{L_{eff}} = \frac{\sum_{i=1}^{n} \nu_{L_i} E_{L_i} \left(I_{0_i} + A_i (d_i)^2\right)}{\sum_{i=1}^{n} E_{L_i} \left(I_{0_i} + A_i (d_i)^2\right)} \qquad [\,\text{-}\,] \qquad (2)$$

where the layers (i) are numbered from the middle surface and the thickness of the first layer is one half the thickness of the middle layer. The layers are assumed to be thin, isotropic and perfectly bonded. Using this method the Poisson's ratios were found to be ν_{xy} =0.03, ν_{yx}=0.07. The relation $\nu_{yx} \approx (E_y/E_x)\,\nu_{xy}$ holds approximately for these values.

(a) **(b)** **(c)**

Figure 1: (a) Top and (b) side view of the CLT panel (c) L and T directions in a tree section.

5 DIRECT MEASUREMENT OF THE BENDING WAVESPEED

Direct measurement was based on Roelens *et al* [2] using an impulse hammer and two accelerometers. The phase difference between the two accelerometers was recorded using a Photon II. The distance between the source position (hammer) and the first receiver position was based on an estimate of the bending wavelengths. The optimum is to be out of the nearfield but sufficiently far from the edges to capture the direct field; this distance was chosen to be 0.33m although frequencies <315-400 Hz in the horizontal direction and <500-630 Hz in the vertical direction may have a near field part. Phase matching of the accelerometers was verified by placing them on top of each other. The optimum distance for the accelerometers was 0.04m; wavelengths shorter than this distance would show a >2π phase shift, however this means the upper bound for the data is well above the frequency range of interest. Data was collected up to 4000Hz. Measurements were made in the horizontal and vertical directions parallel to the plate edges with 20 measurements averaged at each point. Four measurement positions were used. The signal was windowed using a short flat topped steep exponential window (damping factor 20). Optimal window length was found to be 14 samples wide (0.683ms) with an additional pre-trigger of 10 samples.

6 RESULTS AND ANALYSIS

Optimised values for elastic constants were found to be ν_{xy}=0.03, E_x=3.13x10^9 Nm^{-2}, E_y=8.48x10^9 Nm^{-2}, and G_{xy}=6.20x10^8 Nm^{-2}. With these values FEM gives good agreement (<1.7%) with the first 11 measured eigenfrequencies. The bending wavenumber was calculated from the phase difference between the accelerometers using eqn 8 from Roelens *et al* [2]. The choice of window length is a trade-off between accuracy at low frequencies and noise at high frequencies. A comparison of the results with thick and thin plate models are shown in Figure 2. In the horizontal direction reasonable agreement is obtained with a thin plate model up to 2500Hz, the data points above 3150Hz may be showing a transition to a thick plate. In the vertical direction reasonable agreement is obtained with a thick plate model up to 2500Hz, however the goodness of fit depends on accurate characterization of the Poisson's ratio. It is possible that the data point at 3150Hz may already show a >2π phase difference between the accelerometer positions however the data accuracy at this frequency due to the width of the window is already poor.

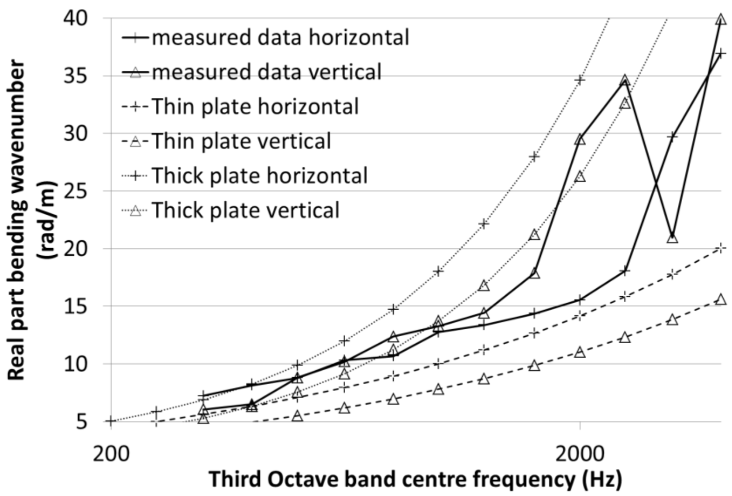

Figure 2: Directly measured bending wavenumber in the horizontal and vertical directions compared with thick and thin plate models.

7 CONCLUSIONS AND FURTHER WORK

Poisson's ratio (ν_{xy} =0.03) for CLT was calculated from the values in tension provided in the literature. However during bending one face of the panel is subjected to tension and the other compression therefore a more accurate calculation might take into account the different Poisson's ratios for compression or tension [7]. Moisture content of the panel may also be an influencing factor [8]. Further work is also required at high frequencies (>266Hz) where the plate may be modelled using a thick plate theory. Optimisation with FEM gave elastic moduli values for CLT which resulted in good agreement with the first 11 measured eigenfrequencies. Direct measurement of the bending wave velocity on a full panel was carried out at higher frequencies (315-4000Hz). This and the data obtained in the FEM indicated that thick plate bending theory may be required above 266Hz. Although in the horizontal direction, reasonable agreement is obtained with thin plate theory up to 2500Hz. The goodness-of-fit to thick plate calculated bending wavespeeds and modal shapes depends on an accurate Poisson's ratio.

8 REFERENCES

1. Churchill, C., Hopkins, C., Prediction of the dynamic properties of a cross laminated timber plate from an investigation of the eigenmodes using a scanning laser vibrometer, DAGA (2013)
2. Roelens, I., Nuytten, F., Bosmans, I., Vermeir, G., In situ measurements of the stiffness properties of building components, Applied Acoustics, Vol. 52, No 3/4, (1997), 289-309.
3. Steiger, R., Gülzow A, Czaderski, C., Howald, M. T., Niemz, P. Comparison of bending stiffness of cross-laminated solid timber derived by modal analysis of full panels and by bending tests of strip-shaped specimens, Eur. J. Wood Prod. 70, (2012), 141-153.
4. Van Damme B., Schoenwald S., Blanco M A., Zemp A., Limitations to the use of homogenised material parameters of cross laminated timber plates for vibration and sound transmission modelling, ICSV22 (2015)
5. Kohlhauser C., Hellmich C., Determination of Poisson ratios in isotropic, transversely isotropic, and orthotropic materials by means of combined ultrasonic-mechanical testing of normal stiffnesses: Application to metals and wood, European Journal of Mechanics A/Solids 33 (2012) pp82-98
6. Jones R. M., Klein S., Equivalence between single layered and certain multi-layered shells, AIAA Journal Vol 6, No 12, (1968) pp2295-2300
7. Ozyhar O., Hering S., Niemz P., Moisture-dependant orthotropic tension-compression asymmetry of wood, Holzforschung Vol 67 (4) (2013) pp395-404
8. Ozyhar O., Hering S., Sanabria S. J., Niemz P., Determining moisture-dependant elastic characteristics of beech wood by means of ultrasonic waves, Wood Science and Technology Vol 47 (2) (2013) pp329-341