Improved low-frequency performance of cross-laminated timber floor panels by informed material selection

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Abstract
The paper demonstrates improved structural low-frequency vibroacoustic performance of cross-laminated timber (CLT) floor panels by informed selection of the wood material. The use of wood species and strength classes that are not traditionally assigned to CLT panels was investigated in order to study their influence on dynamic characteristics and vibroacoustic response metrics. The potential of each of the orthotropic material properties to alternate the vibration response was examined to determine the governing parameters of the low-frequency vibroacoustic performance. The effects on transfer mobility response functions, and eigenfrequencies and mode shapes were used for a rigorous performance study of the panels. It was found that using laminations with stiffness properties typical for hardwoods ash, beech, and birch can significantly improve the performance of a CLT floor panel, and they outperform laminations of typical softwood strength classes.

1. Introduction
Timber construction has gained momentum in recent years, partly due to an increased focus on environmentally friendly materials, sustainability, and climate change. Today, timber from mainly softwoods, such as spruce, is used for structural purposes in buildings. Climate change alters the growth conditions in our forests and with climate change comes also predicted increasing risks for forest fires and severe storms. These factors affect both the size of our forests and the stands of different species within them. Species that are fast growing and more resistant to climate change need to be studied in order to facilitate a continuous supply of raw material of sufficient quality for structural utilization.

A contributing factor to the increased use of timber for construction in recent years is the introduction of the composite engineered wood product referred to as cross-laminated timber (CLT), which was introduced in the 1990s [1]. CLT elements are composed of several layers of individual laminations that are adhesively bonded, see Fig. 1 for a sketch of an element comprising five layers. The layered composition of cross-wise oriented laminations allows the possibility of production of large panels with relatively high stiffness and load-bearing capacity in relation to their weight.

When using CLT panels as parts of floor systems, the advantage of having considerable bending stiffness in two directions at out-of-plane loading is utilized.

Early and important research contributions regarding use of CLT in wood-based construction include the doctoral thesis of Schickhofer from 1994 [2]. Significant developments have since then been made in many areas relating for example to production and structural design methods for assessing the load-bearing capacity and stiffness of CLT panels and joints between panels, see e.g. [3] for an overview and [4] for recent developments. Currently, CLT is produced mainly from softwood species such as spruce and fir. Attention has however recently also been drawn to utilization of hardwood species for CLT, thanks to benefits regarding availability and favorable mechanical properties in general and regarding rolling shear in particular. Studies considering beech [5], fast-growing poplar [6] and several other hardwood species [7] are found in the literature.

Vibroacoustic floor response can be induced by different types of sources, for example: footfalls from people walking; indoor vibrating machinery; and external rail traffic. The sources of vibration can lead to excessive noise and/or vibration levels, which can cause annoyance and discomfort for inhabitants, and malfunction of sensitive equipment. Residents and people working in multi-story timber buildings can perceive vibrations and structure-borne noise as annoying even though the buildings fulfill the build-
The natural variation of the mechanical properties of wood can furthermore be considerable, even within the same species. The stiffness properties and the relations between the stiffness properties for different directions of loading, with respect to the fiber direction, differ considerably between different wood species. The natural variation of the mechanical properties of wood can furthermore be considerable, even within the same species. The response to static as well as dynamic loads may thus differ considerably between wood species and between different timber laminations from the same species [19–23].

In the present paper, we demonstrate the possibilities to improve the structural low-frequency (<100 Hz) vibroacoustic performance of CLT floor panels by studying the effects of using various soft- and hardwood species, and strength classes. Because the variation in properties between various wood species is significant, the response of CLT panels using such species are expected to vary considerably. Our research hypothesis is that the vibroacoustic performance of CLT floor panels can be appreciably improved by using hardwood species and/or softwoods species of high strength classes. It should be noted that the present paper is limited to investigating material selection for the floor panels, while not considering other design choices regarding, for example, supporting conditions and surface coatings. We believe that knowledge regarding the material selection as an isolated design choice is valuable for holistic assessments in product development. The potential of each of the orthotropic material properties to alternate the response was investigated to study, in detail, the governing parameters of the performance, as well as the potential of using various wood species and strength classes. The effects on transfer mobility response functions, and eigenfrequencies and mode shapes were used for a rigorous performance study of the panels.

Herein lies the novelty of the paper: the presentation of a rigorous investigation regarding the effect of material selection on the low-frequency vibroacoustic performance of CLT panels.

The rest of the paper is structured as follows. Section 2 introduces the methods and the theory used in the presented research. Section 3 presents the employed engineering application of the CLT floor panels used in the paper. In Section 4, the potential of each of the orthotropic material parameters to affect the panels’ dynamic behavior is presented. Section 5 discusses a case study concerning the effects of different strength classes on the vibroacoustic performance. The possible improvements in performance of using different nonstandard hard- and softwood species is presented in Section 6. Section 7 provides a discussion on the results and their implications and applications. In Section 8, the conclusions are given.

2. Low-frequency vibroacoustics of cross-laminated timber floor panels

The choice of the material models, numerical methods, and analyses performed in the present study are based on the assumption of linearity because low-amplitude structural vibrations induced by, for example: human walking, vibrating machinery, or external traffic are considered. Consequently, linear elastic material response is assumed. The analyses are carried out in the frequency domain.

For a three-dimensional continuum representation, wood is commonly modeled as an orthotropic material with material principal directions defined by the longitudinal (L), radial (R) and tangential (T) directions. For most wood species, the stiffness in the longitudinal direction (parallel to the grain), E_L, is far greater than the stiffnesses in the radial and the tangential directions (perpendicular to the grain), E_R and E_T, respectively. The orientation of the principal directions may be assumed to follow the annual growth rings of the tree and can then be defined by a cylindrical coordinate system. However, in practical engineering calculations, it is often infeasible to model the timber at this level of detail and simplifying assumptions by use of orthotropy with rectilinear orientation of the material principal directions or transversal isotropy are commonly adopted.

The CLT panels were modeled using conventional shell elements and the layered composition was accounted for by the use of composite lamination theory. The technique of using shell elements with lamination theory for CLT panel response predictions has been experimentally verified in for example [12,14]. The composite layup consisted of layers with linear elastic and orthotropic material behavior. Homogeneous orientation of the material principal directions according to a rectilinear coordinate system was used for each layer. The longitudinal direction was oriented in the length direction of the laminations, the tangential direction in the width direction of the laminations, and the radial direction in the thickness directions of the laminations (and of the CLT panel). The material was further assumed to be homogeneous in
the sense that knots and other growth defects were disregarded in the analyses. Using this shell modeling approach, the adhesive bonding between the laminations is implicitly assumed to be infinitely stiff. Full interaction between adjacent laminations within the same layer is also assumed. The numerical models considers only the properties of the CLT panel itself. Additional mass, stiffness and damping from for example, floating screeds and/or fills are hence not considered.

2.1. Numerical modeling

The FE mesh consists of shell elements with six degrees of freedom (DoF) per node (three displacements and three rotations) using Reissner–Mindlin theory with bi-quadratic Lagrange interpolation and reduced integration. The mesh density was selected as to ensure sufficient resolution for wave propagation and mode shapes up to 100 Hz. An FE model of a linear dynamic system results in the standard equation of motion in the time domain as

$$\mathbf{M}\ddot{\mathbf{u}}(t) + \mathbf{C}\dot{\mathbf{u}}(t) + \mathbf{K}\mathbf{u}(t) = \mathbf{f}(t),$$

where \(\mathbf{M}\), \(\mathbf{C}\), and \(\mathbf{K}\) denote the mass, damping, and stiffness matrices, respectively, whereas \(\mathbf{u}(t)\) and \(\mathbf{f}(t)\) denote the displacement and force vectors, respectively, as functions of time, \(t\). Because of the use of FE modeling, the mass and stiffness matrices are constructed based on the discretization of the structure. The damping matrix, however, is normally constructed on a system level by using measured values of the specific structure or empirical damping values; see the last paragraph in the section for details regarding the damping method as:

$$\mathbf{C} = \rho \mathbf{M} + \beta \mathbf{K},$$

where \(\rho\) is the density of the material, \(\mathbf{M}\) the mass matrix, \(\mathbf{K}\) the stiffness matrix, and \(\beta\) the damping ratio.

For a model with \(N\) DoFs, the displacement \(\mathbf{u}(t)\) can be expanded in terms of a set of orthogonal eigenmodes, \(\Phi_j\), where \(j = 1, 2, \ldots, N\), as \(\mathbf{u}(t) = \sum_j \xi_j(t)\Phi_j\), where \(\xi_j(t)\) is the eigenfunction of model \(j\) containing the corresponding generalized nodal coordinates. The undamped eigenmodes are identified by solving the following eigenvalue problem—which is obtained by combining the expanded displacements with the previously introduced equation of motion, and assuming \(\mathbf{C} = 0\) and \(\mathbf{f}(t) = 0\)—as:

$$\mathbf{K}\xi_j(t) = \omega_j^2\Phi_j\xi_j(t).$$

Harmonic loading is assumed, and the steady-state response is considered. The load and the corresponding displacements are expressed in terms of complex vectors as \(\mathbf{f}(t) = e^{j\omega t}\mathbf{f}\), \(\mathbf{u}(t) = e^{j\omega t}\mathbf{u}\), where \(\omega\) is the angular frequency and \(\mathbf{f}\) and \(\mathbf{u}\) denote the complex load and displacement amplitudes, respectively, and \(i\) denotes the unit imaginary number. The assumption is introduced in the equation of motion in the time domain to yield the equation of motion in the frequency domain for the direct steady-state solution as

$$\mathbf{D}(\omega)\mathbf{u} = \mathbf{f},$$

where \(\mathbf{D}(\omega) = -\omega^2\mathbf{M} + j\omega\mathbf{C} + \mathbf{K}\)

is the dynamic stiffness matrix. The frequency range considered in the analysis is 0.25–100 Hz. The frequency increments are set uniformly to 0.25 Hz in order to accurately describe the resonance peaks.

In the selection of the damping model, both Rayleigh and complex stiffness (a.k.a. structural damping) models were evaluated. It was seen that the choice had a negligible effect on the response as evaluated by the prediction metrics (cf. Section 2.2). Rayleigh damping was chosen here because it has been proven to be a suitable method for determining damping ratios in timber structures [23]. Hence, the damping matrix \(\mathbf{C}\) is established by the Rayleigh damping method as: \(\mathbf{C} = \alpha\mathbf{M} + \beta\mathbf{K}\). The European design code for timber structures, Eurocode 5 [24], is currently under revision and damping ratios of 2.5%–4% have been suggested to be used for CLT floors [25]. This range of damping ratios are also found in the Austrian national specifications [26] to the current version of Eurocode 5 and found in several design handbooks for CLT structures, see e.g. [27,28]. The coefficients used for the mass proportionality \(\alpha\) and stiffness proportionality \(\beta\), respectively, are 

$$\alpha = 3.21 \text{ and } \beta = 9.52 \cdot 10^{-3},$$

which corresponds to damping ratios of 2.5% ± 0.75% for the resonance peaks in the considered frequency interval.

2.2. Evaluation metrics

In the investigation, different metrics were used for studying the structural vibroacoustic transmission and response.

2.2.1. Normalized relative frequency difference

The normalized relative frequency difference (NRFD) was used to compare eigenfrequencies from different models to each other. It is defined, in percent, for the \(i\)th mode shape as

$$\text{NRFD} = \frac{|\omega_{\text{A}} - \omega_{\text{B}}|}{\omega_{\text{A}}} \times 100\%$$

where \(\omega_{\text{A}}\) is the ith eigenfrequency of model A and \(\omega_{\text{B}}\) is the ith eigenfrequency of model B.

2.2.2. Modal assurance criterion

The modal assurance criterion (MAC) [31] is used here for comparing mode shapes obtained from different models. The criterion provides a measure of the consistency between two mode shape vectors, were a MAC value of 1 is obtained in the case of two identical vectors. The MAC value for the \(i\)th mode shape of model A, \(\Phi_A\), and the \(j\)th mode shape of model B, \(\Phi_B\), is defined as

$$\text{MAC} = \frac{|\Phi_A^T \Phi_B|}{|\Phi_A^T \Phi_A| |\Phi_B^T \Phi_B|}.\quad (2)$$

In the MAC evaluations performed in the paper, all nodal degrees of freedom of a FE model are included. When comparing two different sets of modes to each other, this is referred to as cross-MAC and presented in matrix color plots in the paper. The color scale indicates the MAC value, between 0 and 1, for each pair of modes.

2.2.3. Mobility response functions

The transfer functions in the frequency domain can be expressed as frequency response functions (FRFs):

$$\mathbf{u} = (\mathbf{D}(\omega))^{-1}\mathbf{f} = \mathbf{H}(\omega)\mathbf{f},$$

where the frequency response matrix \(\mathbf{H}(\omega)\) contains these FRFs. FRFs expressed by vibrational velocity as a function of a unit load are known as mobility functions. Note that, within the frequency domain, the velocity is given by \(\mathbf{v} = j\omega\mathbf{u} = j\omega\mathbf{H}(\omega)\mathbf{f}\). The term—transfer mobility—is used here for loading and evaluation in separate locations. The velocity, or mobility, is used in this study because: velocity relates to kinetic energy and human perception to vibrations is constant for velocities above \(8\) Hz [17], which is also valid for the Vibration Criteria (VC) curves [32] that are often used for sensitive equipment. Note, however, that because the response
is assumed to be excited by harmonic loading, it is straightforward to change between displacement, velocity, and acceleration in the frequency domain. Therefore, it is also straightforward to switch between compliance, mobility, and acceleration. In this paper, the magnitude (or modulus) of the complex velocity amplitude, |v|, is presented as plots of mobility response functions.

2.2.4. Response metric

A root mean square (RMS) value of a transfer mobility was used in order to calculate a scalar value, indicating the structural vibroacoustic response level of a floor panel. The RMS value in the frequency domain is calculated according to Parseval’s theorem as:

$$\langle m_{li}\rangle_{\text{RMS}} = \sqrt{\frac{1}{J-1} \sum_{j=1}^{J} (m_{li}(\omega_j))^2}, \quad (4)$$

where $j$ is the frequency step in the steady-state analysis and $J$ is the total number of frequency steps. Further, $m_{li}(\omega)$ is the absolute value of the transfer mobility, where $l$ and $i$ refer to the spatial points $x_l$ and $x_i$. These will, in the present analysis, be chosen as nodal points. Hence, the values of $m_{li}(\omega)$ can be found as components of $j\omega|H(\omega)|$. It is noted that only vertical loads and velocity responses are considered. Reciprocity implies symmetry of $H(\omega)$ and therefore $m_{li}(\omega) = m_{il}(\omega)$.

3. Engineering application

In order to demonstrate the effects of using various wood materials on the vibroacoustic performance of CLT floor panels, four different floor designs which are typically used for long spans were selected (see Table 1). For long-span CLT floors, the dynamic response is commonly the governing aspect for the structural design [3].

The following setup was used as the reference floor in the investigations. The length of the floor is $l = 7$ m and the width is $w = 2.4$ m; see Fig. 2. The CLT panel is composed of seven layers, each with a thickness of 40 mm in the out-of-plane ($z$) direction, giving a total thickness of $h = 280$ mm. The laminations in the two outer layers on each side of the cross-section center were oriented in the same direction and parallel to the $x$-direction as shown in Fig. 3(left). This type of layup, with two outer layers oriented in the same direction, is commonly used for long-span CLT floors. The displacement boundary conditions (BCs) were set as a simply-supported single-span floor panel, which spans 7 m. The displacement BCs were applied in the mid cross-section (along the $z$-axis) of the panel. The two sides parallel to the $y$-axis (see Fig. 2) have displacement BCs for the $y$- and $z$-directions prescribed to zero. The panel is free to move in the length direction (along the $x$-axis), except at a single node to prevent rigid body motion. The two sides parallel to the $x$-axis are thus free.

To increase the general validity of the conclusions in the study, a total of four different setups in terms of layup, geometry, and BCs were studied, see Table 1 and Fig. 3. The results in Sections 4–6 are related to the reference setup. The results for Setups A–C are presented and discussed in relation to the reference setup in Section 7.1.

The layers were assigned material parameters according to Table 2, and these parameters were consistently used as a reference wood material. The values for the stiffness parameters $E_l, E_r, G_{LT}, G_{LR}$, and the mass density $\rho$ were taken from values given in the European standard EN 338 [29] for softwood strength class C24. The standard EN 338 defines a system of strength classes for structural timber to be used for design of load-bearing structures according to design codes, such as Eurocode 5 [24]. The standard gives, for example, mean values of stiffness parameters and densities for different softwood and hardwood strength classes. Because the rolling shear stiffness, $G_{LT}$, and Poisson’s ratios are not specified in EN 338, these values are instead based on mean values as reported in the comprehensive compilation of stiffness parameters for various spruce species given in [33]. Note that $E_r$ is omitted here due to the shell–plate theory assumption.

The mode shapes and eigenfrequencies of the reference floor panel are shown in Fig. 4, where the modes are named after the number of “bays” in the $y$- and $x$-directions, respectively. A total of eight modes occur below 100 Hz; the first mode occurs at 11.9 Hz.

The mechanical load is applied in a location where all modes within the frequency range of interest are excited, namely: location 1 (L1), which is located close to one of the corners; see Fig. 2. Thus, with solely one loading position, a full description of the vibroacoustic structural response is obtained. The vertical velocity response is evaluated at location 2 (L2), which is situated at the opposite corner of L1.

4. Governing material properties in vibroacoustic performance

In order to investigate the governing material properties and their effect on the structural vibroacoustic performance of the floor panel, one parameter at a time was varied while the others were kept constant. The intervals used for the material properties are presented in Table 3. These values are based on data in the European standard EN338 [29] for different strength classes and data for different hard- and softwood species found in the literature [33,34]. The presented limits refer to low and high mean values, respectively, for the different parameters. For the sensitivity study, the interval for each parameter is divided into five discrete values, equally spaced between the respective lower and upper limits.

To present the influence of the investigated parameters, MAC and NRFD values, mobility transfer functions and RMS values of these are assessed as measures.

4.1. Eigenfrequencies and mode shapes

The nine varied material parameters, with five tested values per parameter, result in 45 (5 values × 9 parameters) comparisons of modes against the modes in the reference model. The influence of the Poisson’s ratios ($\nu_{LT}, \nu_{LR}, \nu_{TR}$) on the response were insignificant, which is in line with statements in [35]. Therefore, they are hereafter omitted from the sensitivity analysis results. Variation in the mass density, $\rho$, the Young’s modulus in the longitudinal direction, $E_L$, or shear modulus, $G_{LT}$, give no shift in the order of modes. Variation in the remaining three parameters, $E_r, G_{LT}$ and

<p>| Table 1 |</p>
<table>
<thead>
<tr>
<th>Setup</th>
<th>$l$ (m) × $w$ (m)</th>
<th>BCs</th>
<th>$h$ (mm)</th>
<th>$N_{\text{modes}}$</th>
<th>Layup (mm)</th>
<th>Parallel two outer layers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Setup: Ref.</td>
<td>7×2.4</td>
<td>SFSF</td>
<td>280</td>
<td>7</td>
<td>7 × 40</td>
<td>Yes</td>
</tr>
<tr>
<td>Setup A</td>
<td>5×2.4</td>
<td>SFSF</td>
<td>180</td>
<td>5</td>
<td>40–30–40–30–40</td>
<td>No</td>
</tr>
<tr>
<td>Setup B</td>
<td>7×2.4</td>
<td>SSSS</td>
<td>280</td>
<td>7</td>
<td>7 × 40</td>
<td>Yes</td>
</tr>
<tr>
<td>Setup C</td>
<td>5×2.4</td>
<td>SSSS</td>
<td>180</td>
<td>5</td>
<td>40–30–40–30–40</td>
<td>No</td>
</tr>
</tbody>
</table>
result in one or a few mode shifts. The shifts occur for closely spaced modes. Moreover, as seen in Fig. 5, the MAC values as compared to the reference model are higher than 0.9 for all tested parameters, except for the highest mode when using the highest parameter value of $E_L$ (1500 MPa). This gives a slightly different mode (2,3) compared to that of the reference wood material, cf. Fig. 4h and 6. The MAC value in comparing these modes is 0.51.

The NRFD values were calculated using strength class C24 with material parameters according to Table 2 as reference. In calculating the NRFDs, the same modes are compared. This means that the shift in the order of modes, as seen in Fig. 5, has been accounted for. The NRFDs for the eight modes (which occur below 100 Hz) and the five tested values for each parameter are shown in Fig. 7.

As seen in Fig. 7a, the influence of the mass density on the eigenfrequencies can be significant. It can also be seen that—as expected—the NRFD is constant for all the modes. In sub-figure (b), the effect of $E_L$ is shown. The modulus has a strong influence on mode (1,1), cf. Fig. 4a, and it is also seen that the effect on mode (1,2)—cf. Fig. 4b—is less significant. This behavior is expected because the two outer layers on each side of the floor panel are oriented with the longitudinal direction parallel to the $x$-direction shown in Fig. 2 and because of the shape of the modes. This behavior is also notable for modes (2,1) and (2,2), cf. Fig. 4c and d. As the shape of the modes becomes more complex for higher modes, this behavior is not evident for those higher modes. The Young’s modulus in the tangential direction, $E_T$, has an insignificant effect on most of the modes (cf. Fig. 7c). For the modes (1,3) and (2,3), however, considerable effects can be noted. As seen in Fig. 7d, $G_{RT}$ significantly affects the frequency of the modes with two or more “bays” in the $y$-direction; especially for mode (1,2). It was seen that $E_L$ markedly affects the modes with one bay in the $y$-direction, such as modes (1,1), (2,1), and (3,1). Note that this behavior becomes less visible for higher modes. The shear modulus $G_{LR}$ has an insignificant influence on the eigenfrequencies (cf. Fig. 7e), with a maximum NRFD of only 2%. Here it can be pointed out that the effect of the Poisson’s ratios is even lower. The influence of the so-called rolling shear modulus, $G_{RT}$, is shown in Fig. 7f. It has a significant impact on the frequency of all modes below 100 Hz. Already for the first bending mode (1,1), the influence is considerable. In general, the effect increases for higher modes, reaching NRFDs of up to 40%.

### 4.2. Mobility response functions

The effect on transfer mobilities of the various material parameters are shown in Fig. 8. The load is applied at location L1, and the transfer mobility $m_{12}(\omega)$ is evaluated at location L2. The frequency shift in eigenfrequencies caused by the variation in the different parameters, presented as NRFDs in Fig. 7, can be observed in Fig. 8 as shifts in the velocity resonance peaks.
The effects on transfer mobilities by varying the mass density are seen in Fig. 8a. A significant reduction in the amplitude of the transfer mobility resonance peaks can be seen when increasing the mass density. For the Young’s modulus in the longitudinal direction, $E_L$, large shifts in the frequency of the resonance peaks shown in Fig. 8b can be observed, while their amplitudes are not affected considerably. As indicated in Fig. 7c, changes of $E_T$ only affect the higher modes and correspondingly only affect resonance peaks in the transfer mobility above approximately 90 Hz; see Fig. 8c. Thus, the Young’s modulus in the tangential direction, $E_T$, only affects a narrow frequency band. Similarly to the Young’s modulus in the longitudinal direction, $E_L$, the shear modulus $G_{LT}$ affects the frequencies of the resonance peaks while it does not have any appreciable influence on the amplitudes in the transfer mobility spectra. The shear modulus $G_{LR}$ has an insignificant impact on the transfer mobility amplitudes, and on the frequency of the peaks. It is repeated here that Poisson’s ratios have an even smaller effect on the performance of the floor panel. The influence of the rolling shear modulus, $G_{RT}$, is substantial, as shown in Fig. 8f. Both the amplitudes of the transfer mobility peaks and their frequencies are significantly affected.

### 4.3. Summary

In order to summarize the sensitivity analysis regarding the influence of the various parameters on eigenfrequencies and veloc-
ity response, the RMS value of the transfer mobility and the mean of the NRFDs for the eight modes are shown in Fig. 9. Both metrics are normalized with respect to the respective values of the reference case (strength class C24). It should be noted that the NRFDs for all eight modes below 100 Hz are either increased or decreased when varying one parameter at a time in the sensitivity study. Therefore, the sign of the individual NRFD values are kept in the calculation of the mean value, as opposed to using the absolute values.

Starting from the left-hand side of the figure, it is seen that the mass density, $\rho$, has an appreciable influence on both the response amplitudes and the NRFDs. The stiffness parameter $E_L$ does not affect the overall level of the transfer mobility, but has a significant effect on the eigenfrequencies, which may have a considerable effect on the vibroacoustic response caused by, for example, walking loads due to the matching of the peaks in the walking load spectrum and the transfer mobility spectrum. The stiffness param-

![Fig. 5. The cross-MAC values, as compared to the reference model, for the five values (val. 1 – val. 5) of each of the three parameters ($E_T, G_{LT}, G_{RT}$) affecting the mode shapes in the sensitivity analysis.](image)

![Fig. 6. Mode (2, 3) calculated using C24 parameters, except $E_T=1500$ MPa.](image)

![Fig. 7. The effect on eigenfrequencies for the five values of the various tested material parameters in the sensitivity analyses. The legend is valid for all subplots. Blue/squares: first value; red/diamonds: second value; yellow/upward-triangle: third value; purple/downward-triangle: fourth value; green/hexagram: fifth value. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)](image)
eter $E_T$, as stated earlier, only affects a few modes located in the highest frequencies in the frequency range of interest. However, if such high loading frequencies are considered, the impact may be significant. The shear modulus $G_{LT}$ does not affect the level of the transfer mobility, but can have a considerable effect on the frequency of certain modes, which may be of significant interest for loads with narrow banded frequency content. It is evident, as seen in the figure, that the shear modulus $G_{LR}$ neither affects the transfer mobility nor the NRFDs to any appreciable extent. However, the effect is still greater than for the Poisson’s ratios. The rolling shear modulus, $G_{RT}$, strongly affects both the transfer mobility amplitudes and the frequency of the peaks, which in addition is shown as high NRFDs. The tested interval for $G_{RT}$ is rather wide: The lowest value (20 MPa) differs from the highest value (450 MPa) by a factor of 22.5, see Table 3. No other parameter interval comes near this variation in the sensitivity analysis (the rolling shear modulus is further discussed in Section 7). The set intervals reflect realistic variations of material properties between different wood species.

5. Case study: strength classes

In this section, the effect of the CLT layer properties according to various softwood strength classes on eigenfrequencies, mode shapes and transfer mobility response functions are presented. The considered strength classes and their material parameters are given in Table 4. These parameters are based on values given in the European standard EN 338 [29] and in [33]. The values for the stiffness parameters $E_L$, $E_T$, $G_{LT}$, and the mass density $\rho$, are taken from EN 338 [29]. As no values are given in EN 338 for the rolling shear modulus, $G_{RT}$, and Poisson’s ratios, these values are instead based on mean values reported in [33]. Since it was concluded from the sensitivity analysis (see Section 4) that the Poisson’s ratios have negligible influence on the results of interest, these parameters were assigned the same values for all considered strength classes.

The tested interval for $G_{RT}$ is rather wide: The lowest value (20 MPa) differs from the highest value (450 MPa) by a factor of 22.5, see Table 3. No other parameter interval comes near this variation in the sensitivity analysis (the rolling shear modulus is further discussed in Section 7). The set intervals reflect realistic variations of material properties between different wood species.
studied CLT panel. In order to account for its effect, a relation between the rolling shear moduli, $G_{RT}$, and the shear moduli $G_{LT}$ and $G_{LR}$ was determined based on mean values of the 22 different types of spruce given in [33]. First, the mean values of $G_{LT}$, $G_{LR}$, and $G_{RT}$, respectively, were determined. Because $G_{LT}$ and $G_{LR}$ are assumed to be equal in EN 338, an average was used here, resulting in $G_{LT} = G_{LR} = 692$ MPa. The mean of the rolling shear moduli for the 22 different types of spruce is determined to $G_{RT} = 49$ MPa. The relation between the determined moduli becomes: $G_{LT} = G_{RT} = 692/49 = 14$. This factor is used for each of the various strength classes in order to estimate their rolling shear moduli as given in Table 4.

### 5.1. Eigenfrequencies and mode shapes

The effects on the mode shapes and eigenfrequencies for the various strength classes are shown in Figs. 10 and 11, respectively.

The number of modes occurring below 100 Hz does not vary, all strength classes provide eight modes below 100 Hz. The NRFD values depend on both the change in the stiffness parameters as well as in the mass density, because the eigenfrequencies are proportional to the relation between the stiffness matrix and the mass matrix. Whether the NRFD is negative (lower eigenfrequency) or positive (higher eigenfrequency) depends on whether the stiffness matrix or the mass matrix has the larger impact on the eigenfrequencies. In Fig. 10, it is shown that the lowest strength classes (C14 and C18)—which have lower stiffness and mass density than the reference class C24—give lower eigenfrequencies, i.e. negative NRFDs, for all considered mode shapes with respect to the reference strength class C24. Along the same lines, but not as expected, the grading C30—which has higher stiffness (and higher mass density) than C24—also gives lower eigenfrequencies than C24, thus providing negative NRFD values. The highest considered strength class—C40—provides positive NRFDs for all modes. Using various strength classes for the laminations in the CLT floor panel can shift the eigenfrequencies between approximately −13% (when using C14 instead of C24) to about +6% (when using C40 over C24).

The cross-MAC values for the various strength classes with respect to the reference strength class C24 are shown in Fig. 11. The diagonal values in the cross-MAC matrices are larger than 0.99999, showing that all different strength classes possess nearly identical mode shapes below 100 Hz without any shift in the order of modes.

### Table 4

Material parameters used for the various strength classes (C14–C40). Note that the properties for strength class C24 are repeated here (identical to Table 2). The left-out values, marked with a hyphen, are the same as for C24.

<table>
<thead>
<tr>
<th>Property</th>
<th>C14</th>
<th>C18</th>
<th>C24 (Ref.)</th>
<th>C30</th>
<th>C40</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_L$ (MPa)</td>
<td>7000</td>
<td>9000</td>
<td>11,000</td>
<td>12,000</td>
<td>14,000</td>
</tr>
<tr>
<td>$E_T$ (MPa)</td>
<td>230</td>
<td>300</td>
<td>370</td>
<td>400</td>
<td>470</td>
</tr>
<tr>
<td>$G_{LT}$ (MPa)</td>
<td>440</td>
<td>560</td>
<td>690</td>
<td>750</td>
<td>880</td>
</tr>
<tr>
<td>$G_{LR}$ (MPa)</td>
<td>440</td>
<td>560</td>
<td>690</td>
<td>750</td>
<td>880</td>
</tr>
<tr>
<td>$G_{RT}$ (MPa)</td>
<td>31</td>
<td>40</td>
<td>49</td>
<td>53</td>
<td>62</td>
</tr>
<tr>
<td>$\rho$ (kg/m³)</td>
<td>350</td>
<td>380</td>
<td>420</td>
<td>460</td>
<td>480</td>
</tr>
</tbody>
</table>

![Fig. 10](image-url) The effect on eigenfrequencies for the various strength classes. The legend is valid for both subplots. Blue/squares: C14; Red/diamonds: C18; Black/cross: C24 (Ref.); Purple/downward-triangle: C30; Green/hexagram: C40. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

![Fig. 11](image-url) The cross-MAC values for the various strength classes, as compared to C24.
5.2. Mobility response functions

The transfer mobility response function for the various strength classes are shown in Fig. 12. The transfer mobility \( m_{12}(\omega) \) (see Fig. 12) was calculated for loading in location L1 and response in location L2, see Fig. 2 for the locations L1 and L2. In Fig. 12, the frequency shift of the resonance peaks is observed. Significant changes in several peak amplitudes can also be seen, which is mostly a consequence of the change in mass density and the rolling shear modulus, as concluded from the sensitivity analysis. It can be observed that a lighter wood material results in higher amplitudes and vice versa. For the transfer mobility spectra, it is clear that all eight modes are excited.

As a measure of the overall response level, the RMS value is used. In Fig. 13, RMS values normalized to the RMS value of the reference case of C24 are seen. As seen in the figure, the influence of changing the strength class of the material is clear. Considerable benefits of using C30 over C24 are observed; the benefit being more pronounced, as expected, for C40. If using C14, the performance can be significantly worse, resulting in an increased RMS value of the mobility of 36%.

6. Case study: wood species

The influence of using laminations from different hard- and softwood species on eigenfrequencies, mode shapes, and transfer mobility response functions is presented here. The considered species and their respective material parameters are given in Table 5, where the given values are taken from [34].

6.1. Eigenfrequencies and mode shapes

The changes in eigenfrequencies for the various wood species are shown in Fig. 14. In calculating the NRFD values, strength class C24 with material parameters according to Table 2 is used as reference.

The number of modes occurring below 100 Hz varies. Strength class C24 and Scots pine have eight modes below 100 Hz. Beech and Douglas fir have seven modes below 100 Hz, whereas ash and birch only have six. As seen in Fig. 14, all considered wood species generally give higher eigenfrequencies, i.e. positive NRFD values with respect to the reference case of C24. The NRFD values are of significant magnitude, especially for ash, beech and birch—the frequency shift reaches about 20% for the latter half of the calculated modes with respect to the reference C24. By solely studying the fundamental mode (1,1), it is seen that beech gives a negative NRFD, while providing the highest positive NRFD for modes (3,1) and (3,2). This is because of the relatively low Young’s modulus in the longitudinal direction, \( E_L \), which strongly affects the first bending mode, combined with a high mass density, \( \rho \), tangential Young’s modulus, \( E_T \), and shear moduli \( G_{LT}, G_{GR}, \) and \( G_{RT} \).

The cross-MAC values for the various wood species with respect to the reference, C24, are shown in Fig. 15. A shift of modes (3,1) and (1,3) can be seen for ash and birch, respectively. It should be noted these modes differ less than 1 Hz in frequency. All tested wood species yield MAC values above 0.98 for all eight modes, i.e. the mode shapes are almost identical.

6.2. Mobility response functions

The effect on transfer mobility response functions for the studied wood species are shown in Fig. 16. The transfer mobility was calculated for loading at location L1 and response at location L2. The significant frequency-shifts of the resonance peaks are clearly seen in Fig. 16 as is the appreciable change in amplitude at the resonances of the transfer mobility. All tested wood species reduce the amplitude of the resonance peaks, as compared to C24.

As concluded from the sensitivity study presented in Section 4, it is the Young’s modulus in the tangential direction, \( E_T \), and especially the rolling shear modulus, \( G_{GR} \), that have the largest influence on the frequencies above about 75 Hz. As seen in Fig. 16, it is at these frequencies where the most marked deviations between the curves are present. Here, ash, beech, and birch show a significantly lower response than the other considered species.

As a measure of the overall response level, the RMS value is used. In Fig. 17, RMS values normalized to the RMS value of the reference case of C24 are shown. In the figure, it is clearly seen that the species ash, beech, and birch stand out as improving the performance significantly, with beech being the most beneficial. By using one these three species, a halving of the response is indicated. Noteworthy is also the species Douglas fir, which outperforms C40 as analyzed here. The use of Scots pine still results in a considerable lowering of the response levels as compared to C24.

7. Discussion and remarks

It has been found that the vibroacoustic structural response of CLT floor panels can be significantly reduced by using laminations with the properties of ash, beech or birch according to Table 5 instead of laminations with properties of C24 according to Table 2. The response level in terms of the mobility transfer functions, eval-
Evaluating as RMS values in the frequency range 0–100 Hz, may be lowered with approximately 50% as compared to using C24. The advantages with these three species in terms of vibroacoustic structural response are more profound at higher frequencies, around 75–100 Hz. In addition, the tested species Douglas fir and Scots pine were found to be beneficial to use in terms of vibroacoustic response for a broad frequency spectra. Here, the RMS value were reduced with approximately 30% as compared to using C24.

Table 5
The material parameters used for the different wood species.

<table>
<thead>
<tr>
<th>Property</th>
<th>Ash</th>
<th>Beech</th>
<th>Birch</th>
<th>Douglas fir</th>
<th>Scots pine</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_L$ (MPa)</td>
<td>15,800</td>
<td>13,700</td>
<td>16,300</td>
<td>16,400</td>
<td>16,300</td>
</tr>
<tr>
<td>$E_T$ (MPa)</td>
<td>800</td>
<td>1140</td>
<td>910</td>
<td>910</td>
<td>900</td>
</tr>
<tr>
<td>$G_L$ (MPa)</td>
<td>890</td>
<td>1060</td>
<td>1180</td>
<td>1180</td>
<td>1160</td>
</tr>
<tr>
<td>$G_T$ (MPa)</td>
<td>1340</td>
<td>1610</td>
<td>190</td>
<td>79</td>
<td>66</td>
</tr>
<tr>
<td>$v_{LT}$ (%)</td>
<td>0.51</td>
<td>0.51</td>
<td>0.43</td>
<td>0.37</td>
<td>0.51</td>
</tr>
<tr>
<td>$v_{LR}$ (%)</td>
<td>0.46</td>
<td>0.45</td>
<td>0.49</td>
<td>0.43</td>
<td>0.42</td>
</tr>
<tr>
<td>$v_{TR}$ (%)</td>
<td>0.36</td>
<td>0.36</td>
<td>0.38</td>
<td>0.40</td>
<td>0.31</td>
</tr>
<tr>
<td>$\rho$ (kg/m$^3$)</td>
<td>670</td>
<td>750</td>
<td>620</td>
<td>590</td>
<td>550</td>
</tr>
</tbody>
</table>

Fig. 14. The effect on eigenfrequencies for the various wood species. The legend is valid for both subplots. Black/cross: C24 (Ref.); Blue/squares: ash; Red/diamonds: beech; Yellow/upward-triangle: birch; Purple/downward-triangle: Douglas fir; Green/hexagram: Scots pine. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

Fig. 15. The cross-MAC values for the various wood species, as compared to C24.

Fig. 16. The effect on transfer mobility response functions for the various wood species. Black: C24 (Ref.); Blue: ash; Red: beech; Yellow: birch; Purple: Douglas fir; Green: Scots pine. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

Fig. 17. The effect on transfer mobility response functions for the various wood species, as compared to C24.
It was found that the rolling shear modulus, $G_{RT}$, has a strong influence on the eigenfrequencies throughout all calculated modes; in general, the effect increases with higher modes. It should be noted that $G_{RT}$ is difficult to measure with precision due to the variation in annual ring pattern alignment in the laminations, and that it is unknown how the variations in $G_{RT}$ over all layers in a CLT panel influence its dynamic response. The results of the sensitivity study presented in Section 4 relate to a wide range of values of the $G_{RT}$, between 20 and 450 MPa. The study of the influence of using various softwood and hardwood species presented in Section 6 include values of $G_{RT}$ between 66 and 460 MPa, reflecting the wide range of values of the rolling shear stiffness and the variability between different wood species. The values of $G_{RT}$ given in Table 5 agree in general well with values found in recent research publications. A compilation of previous test results for the rolling shear modulus is given in [5], with $G_{RT}$ between 380 and 470 MPa for beech and between 40 and 55 MPa for spruce. The test results on beech presented in [5] gave a mean value of 370 MPa. A comprehensive test program concerning rolling shear properties and considering investigations of six different European wood species for potential use as base material for CLT is further presented in [7]. The tests were carried out using a test configuration similar to the test configuration for longitudinal shear given in the European standards EN 408 [36]. Based on these tests, the following mean values of rolling shear modulus are proposed to be used for CLT applications: 100 MPa for Norway spruce, 150 MPa for pine, 120 MPa for poplar, 180 MPa for birch and 350 MPa for beech and ash.

Using the high strength class of C40 over C24 can also be beneficial; the RMS value was decreased with approximately 15%. In contrary, using C14 amplified the RMS of the structural response by approximately 35%.

The Young’s modulus in the tangential direction, $E_{T}$, has a high influence on a few modes. The Young’s modulus in the longitudinal direction, $E_{L}$, has a strong effect on the first and second bending modes; the effect is less significant for higher modes. A similar tendency was also found for the shear modulus, $G_{LT}$, where especially the first twisting mode is affected by the parameter. The influence of the mass density was found to be marked, and has the same effect for all modes.

It may seem that the mass density, $\rho$, has a surprisingly high effect on the performance in relation to the stiffness parameters. One should then note that the single parameter $\rho$ determines all the entries of the mass matrix, while there are four moduli that may significantly affect the stiffness matrix: $E_{L}$, $E_{T}$, $G_{LT}$, and $G_{RT}$. Moreover, the variation in amplitudes of the resonance peaks depends on whether a displacement, velocity or acceleration output is considered. The results presented in this paper are based on mobilities, i.e. velocity output. Converting the mobility spectra to compliance (i.e. displacement) or acceleration (i.e. acceleration) would give lower and higher weight, respectively, to the response at higher frequencies.

The shortcomings with studying only eigenfrequencies and mode shapes in a structural vibroacoustic performance analysis are evident. If both the stiffness and the mass density are increased, the effects may be balanced out such that no influence on the eigenfrequencies nor mode shapes are seen. However, in such a case, the vibroacoustic response may still be decreased significantly. The strength class C30 is an example of this, in comparison to C24. The class C30 provides more or less identical eigenfrequencies and mode shapes to those of C24. However, the reduction in the overall response level, evaluated as RMS, is considerable (about 10%).

7.1. Effects of floor size, layup and supports

All results presented in Sections 4–6 relate to a specific floor panel geometry regarding span, width and lay-up as described in Section 3; see “Setup: Ref.” in Table 1. The CLT layup in terms of the number of layers, the individual layer thicknesses, and the layer orientations will influence the behavior at static and dynamic loading. To study the sensitivity of the results presented in the paper to variations in the setup, such as other dimensions, laminate layups, and BCs, three other setups were analyzed (cf. Table 1). The setup presented in Section 3 is used as reference here and is denoted “Setup Ref.” The other tested setups are referred to as Setup A–C. The different setups have the same width as the reference panel. The setups shown in Table 1 are repeated here:

- **Setup A**: The length of the CLT panel is $l = 5$ m and the width is $w = 2.4$ m; see Fig. 2. The panel is composed of five layers, with thicknesses 40–30–40–30–40 mm in the out-of-plane ($z$)-direction, giving a total thickness of 180 mm; cf. Fig. 3 (right). The laminations in the outermost layers on either side of the cross-section center were oriented in the same direction, and parallel to the $x$-direction. The displacement BCs are the same as for the reference floor setup.

- **Setup B**: The panel dimensions and laminate layup are the same as for the reference setup—cf. Fig. 3 (left)—whereas the displacement BCs are different. For Setup B, the panel is simply supported on all four sides.

- **Setup C**: The panel dimensions and laminate layup are the same as for Setup A (cf. Fig. 3 (right)), whereas the displacement BCs are different. For Setup C, the displacement BCs are the same as for Setup B.

In order to show the effects of the different setups, the RMS values of the transfer mobility were evaluated for loading in L1 and evaluation in L2; the results are presented in Fig. 18. Note that the results for the reference setup presented in the figure are those shown in Figs. 13 and 17, respectively. As shown in the figure, similar tendencies and trends when varying the material properties are observed for the various setups. Setup B and Setup C, which are supported on all four sides, show large differences in RMS values for strength class C14 as compared to Setup Ref. For Setup B and Setup C, the strength class C14 provides lower resonance frequencies than the other classes, which results in more resonance frequencies in the considered frequency range. It is clear that the results presented in Sections 4–6 are valid for other floor panel configurations as well.

7.2. Load spectra and receiver sensitivity

It should be noted that the predicted effects of various wood material parameters, strength classes, and species, depend on the
engineering case at hand, i.e. the frequency content of the load (e.g. walking load) and the sensitivity spectra of the receiver (e.g. human perception). It is repeated here that neither the load nor the predicted velocities were weighted to mimic certain engineering cases. Cases of narrow-banded load spectra of the source, such as footfalls, and/or narrow-banded sensitivity spectra of the receiver, such as sensitive equipment, may affect the results. Depending on the source and the receiver, different wood species may be the most beneficial one to use. As an example, the relatively high Young’s modulus in the tangential direction and rolling shear modulus of ash, beech, and birch are especially beneficial for higher frequencies; these species may therefore be advantageous to use for loads with significant energy content at, as well as a receiver sensitive to, those higher frequencies. On the other hand, the great advantages for vibration levels shown for the use of beech instead of C24 may be reduced if footfall loading is studied; the selection of beech may lower the first eigenfrequency to the frequency range where footfalls have significant energy content.

8. Conclusions

The paper demonstrates potential advantages for structural low-frequency vibroacoustic response of cross-laminated timber (CLT) floor panels using nonstandard wood types, which currently are used to a very limited extent. For structural applications involving long spans, the dynamic response often is the determining design factor. The governing orthotropic material parameters of the CLT panel response was investigated, along with two case studies that quantify the material-dependent performance: (i) the effects of using various softwood strength classes, and (ii) the effects of using nonstandard soft- and hardwood species.

The presented study indicates the following in the terms of CLT floor panel performance:

- the potentially significant benefits of using the hardwood species ash, beech, and/or birch
- the rolling shear modulus, \( G_{RT} \), as the most significant parameter
- potential advantages of using high strength classes such as C40
- a high possibility to move eigenfrequencies by changing the wood material
- a low possibility to affect the mode shapes or shift the order of modes by changing the wood material.

These insights highlight the need for adequate numerical analyses in order to make well-informed decisions in the design of CLT floor panels in terms of vibroacoustic response, such as the selection of wood material. For future work, it is suggested that detailed studies of the effects of the rolling shear modulus, and that specific engineering cases—because the expected performance being dependent on the frequency content of the load and the frequency susceptibility of the receiver—are investigated.

CRediT authorship contribution statement

Peter Persson: Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Writing - original draft, Writing - review & editing, Funding acquisition. Ola Flodén: Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Writing - original draft, Writing - review & editing. Henrik Danielsson: Conceptualization, Methodology, Validation, Formal analysis, Investigation, Writing - original draft, Writing - review & editing, Funding acquisition. Andrew Peplow: Methodology, Validation, Writing - original draft, Writing - review & editing. Lars Vabbersgaard Andersen: Methodology, Validation, Writing - original draft, Writing - review & editing, Funding acquisition.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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