Reduction of Low-frequency Vibrations of Wooden Floors by Tuned Mass Dampers

H. Reichelt[1], U. Schanda[2], A. Rabold[2]

[1] *Getzner Werkstoffe GmbH, 6706 Bürs, Austria Email: hendrik.reichelt@getzner.com*
[2] *University of Applied Sciences Rosenheim, Germany, Email: schanda@fh-rosenheim.de*

Introduction

Over the last decade a steady increase of public and industrial buildings in the wood building construction sector has occurred. Often these buildings are built with wide spanned floors. These floors tend to induce unpleasant low-frequency (5-100Hz) vibrations. These low-frequency vibrations affect the inhabitants in two ways: namely through vibrational sensitivity of course, and also as noise.

Regulations

Therefore, the building code DIN 1052:2004-08 dealing with the structural engineering of wooden buildings, requires a limitation of deflection of

$$w_{G,\,inst} + \psi_2 \cdot w_{Q,\,inst} < 6\text{ mm}.$$

The reason for this limitation is to keep the 1st Eigenfrequency of the floor above 7.2 Hz in order to avoid inducing vibrations caused by people walking on the floor. But this limitation only approximates a description of the dynamic behaviour of the floor and is then of course very conservative in terms of its impact on cost efficient dimensions of floors.

If this requirement cannot be fulfilled, the standard allows the possibility of special analyses. These analyses are not specified in the DIN 1052:2004-08 but in a further comment (in § 9.2 Table 7) [1].

One criteria deals with the velocity of floor vibrations due to a heel-drop with an impact of 55 Ns and a contact duration of 50 ms:

$$v < 6 \cdot b^{(f_1 \cdot \zeta - 1)}$$

where v is the velocity, b the subsidiary width, f_1 the 1st Eigenfrequency and ζ the damping ratio of the floor.

The second criteria concerns the limitation of acceleration to a level below 0.1 m/s² that should guarantee the comfort of the inhabitants.

The acoustic standard DIN 4109:1989-11 deals only with frequencies above 100 Hz. That is why in wooden buildings, the requirements in this standard on the sound reduction index are often satisfied, yet the inhabitants still complain about an unpleasant rumble due to impact sound.

The spectrum adaptation value term $C_{I,\,50\text{-}2500}$ according to DIN EN ISO 717-2, provides some details regarding the transmission of sound in buildings down to 50 Hz, but it is not yet a compulsory quantity in Germany.

The Investigation

Besides the conventional constructions it is feasible to implement supplemental assemblies to reduce vibrations in the frequency rage between 5 and 100 Hz. Due to the strongly modal behaviour of light floors a significant reduction of the vibrations and sound radiation can be achieved. One possible method for reducing vibrations of wide spanned constructions is through the use of Tuned Mass Dampers (TMDs).

Data of the Element

Length:	5 m
Width:	1.5 m
Thickness:	0.135 m
Mass:	580 kg
Mass per m²:	71 kg/m²

Grid of reading points

40 cm x 40 cm

Frequency range

6 Hz to 74 Hz

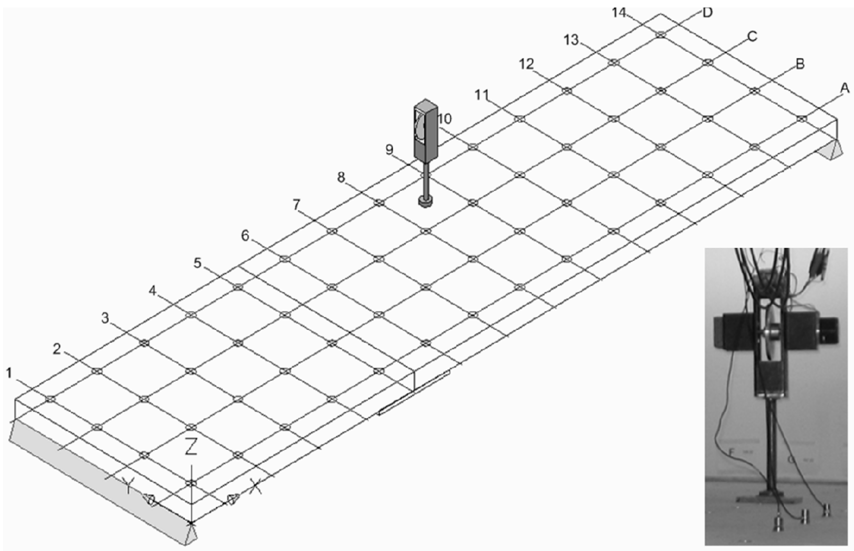

Figure 1: CLT-Element with 56 reading points excited by a vibrating machine

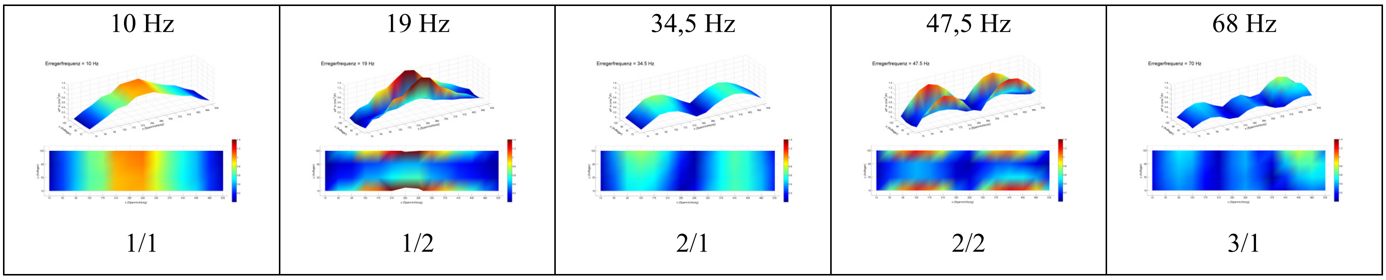

Figure 2: Mode shapes and the corresponding Eigenfrequencies of the CLT-Element

The author carried out investigations on the influence of tuned mass dampers on cross-laminated timber elements (CLT-Element) in 2008 at the University of Applied Sciences Rosenheim [2]. The CLT-Element (see Figure 1) was supported at two sides and had the following dimensions: (L; W; H; 5 m x 1.5 m x 0.135 m). To evaluate the Eigenfrequencies and the mode shapes 56 measure points were placed on top of the element on a grid of 40 cm x 40 cm. On each point the acceleration of the element, that was created by a vibrating machine that stimulated the element, was measured between 6 Hz and 74 Hz in increments of 0.5 Hz. This vibration inducing machine was designed and built at the University of Applied Sciences Rosenheim and presented at DAGA 2005 [3].

The CLT-Element shows five Eigenfrequencies in the range between 6 and 74 Hz with the following five Mode shapes. (see Figure 2)

Another way to show the modal behaviour of this element is to calculated the mean value of all accelerations measured on the 56 points for each frequency step from 6 to 74 Hz as shown in Figure 3.

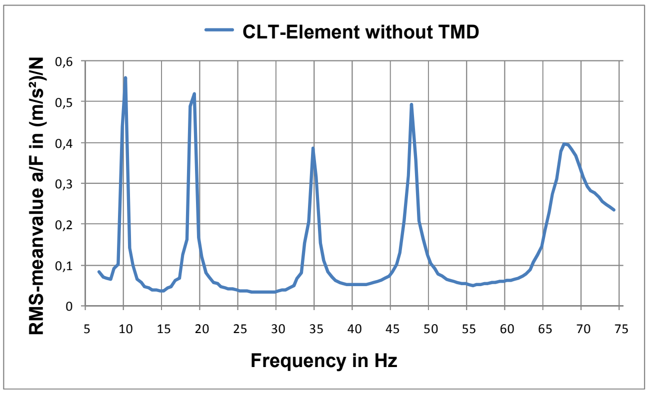

Figure 3: Eigenfrequencies of the CLT-Element

Tuned Mass Damper

Tuned mass dampers were designed for each mode shape and its (corresponding) Eigenfrequency. These were then placed on the CLT-Element at the points where the maximum acceleration occurred.

The mode of operation of a tuned mass damper can be described by a coupling of two damped mass-spring-systems as depicted in Figure 4.

Two degrees of freedom system

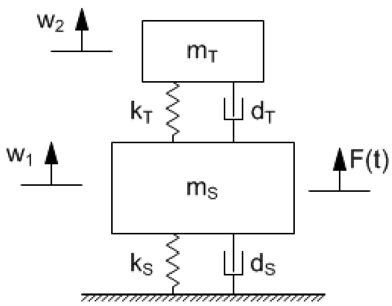

Figure 4: System description of a CLT- floor (primary system) with TMD

- m_S**:** effective mass of the primary system
- d_S**:** damping factor of the primary system
- k_S**:** spring rate of the primary system
- m_T**:** mass of the TMD
- d_T**:** damping factor of the TMD
- k_T**:** spring rate of the TMD
- w_1**:** displacement of the System
- w_2**:** displacement of the TMD mass
- ***F(t)*****:** harmonic load

$$\begin{bmatrix} m_S & 0 \\ 0 & m_T \end{bmatrix}\begin{bmatrix} \ddot{w}_1 \\ \ddot{w}_2 \end{bmatrix}+\begin{bmatrix} d_S+d_T & -d_T \\ -d_T & d_T \end{bmatrix}\begin{bmatrix} \dot{w}_1 \\ \dot{w}_2 \end{bmatrix}+\begin{bmatrix} k_S+k_T & -k_T \\ -k_T & k_T \end{bmatrix}\begin{bmatrix} w_1 \\ w_2 \end{bmatrix}=\begin{bmatrix} F(t) \\ 0 \end{bmatrix}$$

To ensure that the TMD is functioning in its optimal range, four different parameters have to be discussed [4].

- Damping rate of the primary system d_S

  The higher the damping rate of the system d_S is, the smaller the improvement due to the TMD becomes. Hint for engineering purposes:

  ζ should not be bigger than 0.05 where

$$\zeta = \frac{d_S}{2\sqrt{k_S m_S}}$$

- Mass ratio $\mu = m_T/m_S$

  The higher the mass of the TMD m_T is, the bigger the improvement due to the TMD becomes.

  Hint for engineering purposes:

  μ should be chosen between 0.03 and 0.07 %

- Optimum frequency f_T of the TMD

$$f_T = \frac{f_S}{1 + m_T / m_S} \quad \text{where}$$

$$f_T = \frac{1}{2\pi}\sqrt{k_T / m_T} \quad \text{and} \quad f_S = \frac{1}{2\pi}\sqrt{k_S / m_S}$$

Due to this criteria f_T is somewhat smaller than the frequency f_S of the primary system and dependent on the mass ratio μ

- Optimum damping ratio ζ_{opt} of the TMD

$$\zeta_{opt} = \sqrt{\frac{3(m_T / m_S)}{8 \cdot (1 + m_T / m_S)^3}}$$

This formula is only valid for a primary system without damping, but can also be used to provide a close approximation for a system with damping. This criterion does not have as great an impact on the efficiency of the TMD as the optimum frequency has.

Results

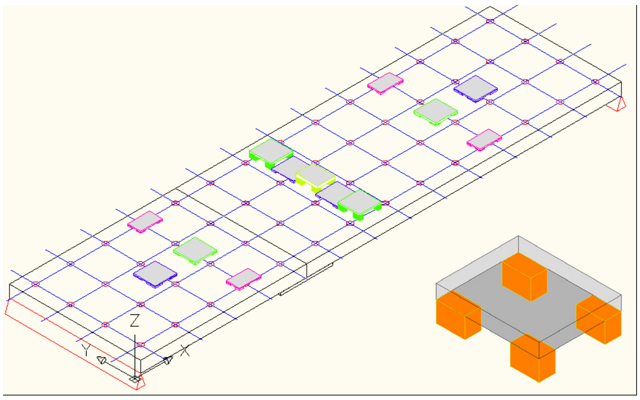

Figure 5: positions of the TMDS an the CLT-Element

For the investigation a total of twelve tuned mass dampers were used. These consisted of a steel plate, used for the mass of the TMD, placed upon four pieces of polyurethane foam (Sylomer®) that acted as a spring-damping-system. Five percent of the effective mass of the primary system was chosen for the mass of the TMDs. Calculations for the optimal Eigenfrequency of the tuned mass dampers were carried through by selecting for the right type and dimension of the polyurethane foam.

One TMD was used to reduce the acceleration of the CLT-elements 1st mode shape, two for the 2nd and 3rd and four for the 4th and for the 5th mode shape (see Figure 5).

Figure 6 shows the mean value of the acceleration for each frequency step for the CLT-element without TMDs (blue line) and with all the TMDs (red line). Thus, a significant reduction in the acceleration of the CLT-element can be identified.

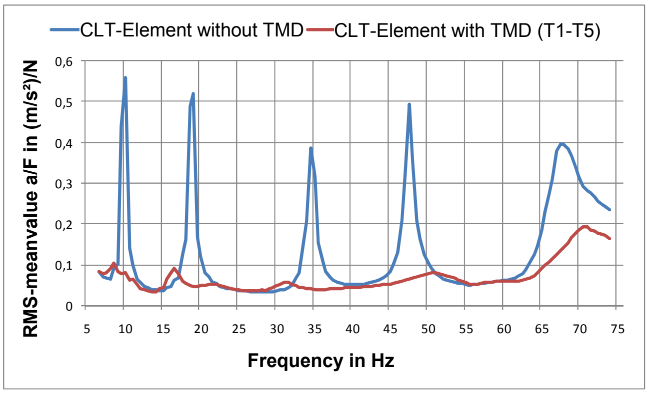

Figure 6: Behaviour of the CLT-Element without and with TMD

An improvement of 17 dB for the 1st, 21 dB for the 2nd, 20 db for the 3rd, 18 dB for the 4th and still 10 dB for the 5th mode shape was measured (see Figure 7)

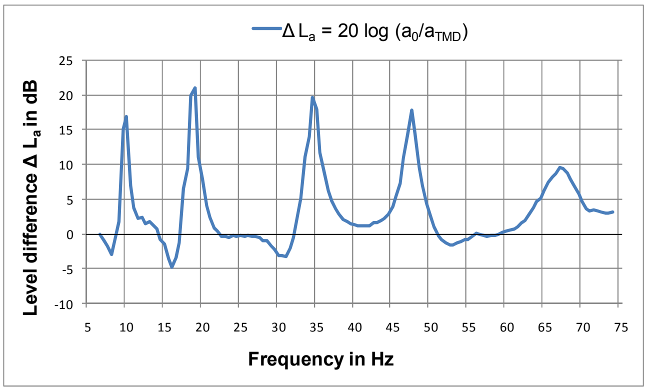

Figure 7: Improvement of the behaviour of the CLT-Element due to the use of TMDs

The results show that the application of TMDs is effective in reducing the vibrations of wooden floors. Dimensioning the TMDs requires the knowledge of the Eigenfrequencies of the floor assembly.

With the ongoing development of simulation of the building constructions with the finite element method it will be possible to determine the Eigenfrequencies and mode shapes of floor assemblies. Therein lies the potential for implementing TMD usage in prefabricated floors.

References

[1] Blaß, H., Ehbeck, J., Kreuzinger, H., & Steck, G. (2004). Erläuterungen zu DIN 1052: 2004-08. München: DGfH.

[2] Reichelt H. (2008). Schwingungsreduzierung bei Holzdecken durch Schwingungstilger. Masterarbeit Hochschule Rosenheim: Rosenheim

[3] Schanda, U., Rogge, K., Meistring, P., & Mühlberger, R. (2005). Prototyp eines Schwingungserregers für tieffrequente Schwingungen von Holzdecken. München: DAGA'05

[4] Petersen, C. (2001). Schwingungsdämpfer im Ingenieurbau. München: Maurer Söhne.