Experimental setup for acoustic and mechanical characterisation of lightweight building elements

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Summary

This study investigates the vibro-acoustic characterisation of lightweight building elements within an alternative test rig, developed in order to speed up the measurement procedure by reducing the size of the samples. Not so many years ago, building construction concerned mostly heavyweight and massive structures, although nowadays lightweight elements are often preferred. Sustainability and energy efficiency requirements have contributed in the development and widespread use of new building technologies and materials. In order to properly design acoustically comfortable buildings by using recently developed prediction models, such as the EN 12354 standards for example, it is necessary to accurately characterise each partition. A reliable characterisation of the elastic and damping properties of a certain building element is fundamental to determine the input data necessary for the simulation of its acoustics performance. This aspect is mostly underestimated and the designers often uniquely rely on database values. However, in order to apply simulation models to complex structures, such as inhomogeneous and non-isotropic elements or multilayer systems, it is indispensable to adopt homogenisation techniques and characterise the elastic and damping properties within the entire frequency range of interest. In this paper an experimental setup to investigate elastic and acoustic properties of lightweight panels, which are usually required as input data in prediction models, is presented. The test rig has been developed for a rectangular element with a surface area of approximately one square metre. One inhomogeneous plywood panel has been characterised, by measuring the plate response due to a mechanical excitation with accelerometers. The structures' elastic properties have been experimentally evaluated, by means of a wave correlation technique. Moreover the radiation efficiency of the mechanically excited plate has been determined. The obtained results are presented, and the advantages and drawbacks of this approach are discussed.

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1. Introduction

Lightweight elements are increasingly preferred to traditional heavyweight solutions in several fields. This is the case also in the building construction industry, since lightweight solutions often allow to reduce construction time and costs. Moreover, environmental concerns are becoming of primary importance within the industry and lightweight timber structures represent an eco-friendly building technology. However, lightweight building elements have received criticisms related to their acoustic performance, usually negligible with traditional massive structures, which can have a significant influence on indoor acoustic comfort. Several prediction models have been developed in the last decade to investigate sound transmission in buildings. In fact nowadays several tools to predict sound transmission in buildings can be found: either the computationally expensive finite element FE and boundary element BE software [1], or simplified algorithms, based for example on wave-based approaches, such as transfer matrix method TMM [2], or on the statistical energy analysis SEA [3]. Moreover, the EN 12354 [4] has been recently updated and refined in order to be suitable also to investigate lightweight structures, as described in ref. [5]. In order to obtain accurate numerical results it is of fundamental importance to have reliable input data to

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Table I. Geometric characteristics of the investigated plywood panel

L_x [m]	L_y [m]	h [m]	ρ [kg/m^3]
0.980	1.180	0.015	400

characterise the investigated elements, such as their elastic and damping properties, or certain acoustic descriptors like for example sound radiation efficiency. Reliable data can be obtained experimentally from vibro-acoustic measurements of the investigated element. Laboratory measurements on building partitions are usually performed on large samples, with a surface area of approximately 10 m^2, in order to fit the test window of the sound transmission facility, as described in the standard 10140-5 [6]. Such a large geometry is convenient in order to determine the sound insulation provided by the element, although it represents a disadvantage when other acoustic or mechanical descriptors have to be experimentally determined. A test rig has been designed to characterise the dynamic behaviour of lightweight building partitions on a small sample, which allows to reduce both the measurement time and the costs for the samples supply. From the measured dynamic response the vibrational mode shapes of the structure, its elastic properties and its radiation efficiency were determined. The obtained results were validated with the experimental data evaluated by using other traditional experimental approaches. The test rig used for the vibro-acoustic analysis, the experimental procedures and the investigated structure are presented in section 2.1. The determination of the structural elastic properties and the sound radiation efficiency are described in sections 2.2 and 2.3 respectively. Finally the results are shown, discussed and validated in section 3.

2. Experimental Measurements

2.1. Test rig and experimental setup

The used test rig is constituted by a steel frame, in which a small rectangular sample, with a surface area of approximately 1 m^2, can be mounted clamping all four boundaries. The experimental investigation presented in this paper refers to a 15 mm thick plywood board made of 5 thin plies, whose physical and geometric properties are given in Table I. The adjustable height at which the sample can be suspended from the floor allows to place an electro-dynamic shaker underneath it, representing the exciting source. The shaker, connected to the board by means of a stinger which terminates with a small metal plate glued to the panel's surface, was driven by an exponential sine sweep signal generated from 30 Hz to 6000 Hz. The shaker was encapsulated within a wooden box, internally lined with porous material in order to reduce the sound field generated by the shaker, which could influence the measurements. The dynamic response was measured in several positions over the sample's surface by using PCB 352C22 miniature accelerometers. The acquired signals were then convolved with the inverse filter of the sine sweep to obtain the impulse responses IR of the structure [7]. In order to guarantee the same relative phase relationship between the signals acquired by non-simultaneous measurements, the IRs obtained by the the accelerometers placed on the panel surface were aligned in time with a reference IR obtained from an accelerometer placed as reference on the shaker's stinger. The complex acceleration spectra can be obtained from the IRs by means of a fast Fourier transform algorithm. The entire measurement procedure, form the signal generation to the acquisition and the processing of the measured responses, was controlled by means of in-house implemented software developed in the LabVIEW environment.

2.2. Determination of the elastic properties

The elastic properties of a structure can be experimentally evaluated by means of several different techniques based either on modal-based methods [8], or on wave propagation analysis, performed both within the audible frequency range [9, 10] and in the ultrasound domain [11]. Besides, the dispersion relation in the wavenumber domain can be obtained by applying a spatial Fourier transform to the complex spectra of the vibrational field measured on a certain number of points on the sample surface [12]. Otherwise, with the same purpose it is also possible to apply wave correlation methods [13]. One of the most commonly used approaches is known as inhomogeneous wave correlation method IWC [14]. This method was applied in order to evaluate the elastic properties of the investigated plywood panel. The vibrational field was measured on 21 points equally spaced at $\Delta x = 1$ cm apart, along different directions in line with the excitation position. For each angular frequency ω the complex spectra $\tilde{w}(\omega, x)$ is correlated to an inhomogeneous propagating wave $\tilde{o}(x, k_r, k_i) = -\mathrm{i}(k_r + \mathrm{i}k_i)x$, by the function:

$$\mathfrak{I}(k_r, k_i) = \frac{\left|\sum_{i=1}^{N} \tilde{w}(\omega, x_i)\, \tilde{o}(x_i, k_r, k_i)\right|}{\sqrt{\sum_{i=1}^{N} |\tilde{w}(\omega, x_i)|^2 \sum_{i=1}^{N} |\tilde{o}(x, k_r, k_i)|^2}} \qquad (1)$$

The structural wavenumber is determined by maximising the function $\mathfrak{I}(k_r, k_i)$ given in Eq. 1. The real part k_r of the complex wavenumber is strictly associated to the elastic properties of the panel, while its imaginary part k_i is proportional to the structural damping. Being interested in the panel elastic characteristics, rather then in its damping, the imaginary part has been neglected: $k_i = 0$, even though it could be determined by using an iterative process as suggested in [15]. The tested structures is not homogeneous, but constituted by a number of thin wood plies.

For this reason the structural wavenumbers were evaluated along five different directions in order to investigate the panel anisotropy.

2.3. Radiation efficiency evaluation

Radiation efficiency σ is a well know acoustic descriptor used to define the capability of a structure to convert vibrational energy into sound waves propagating in the surrounding fluid. It provides useful information about the investigated element and is required in many computational models as input data. It is defined as:

$$\sigma = \frac{W}{\rho_0 c_0 S \left\langle v^2 \right\rangle} \tag{2}$$

where $\left\langle v^2 \right\rangle$ is the mean square vibration velocity of the panel, S its surface area in square metres and ρ_0 and c_0 represent the fluid density and speed of sound respectively. The experimental evaluation of the radiation efficiency can be affected by a large uncertainty, mostly due to the fact that the radiated sound power W is a not directly measurable but needs to be derived from other measurable quantities. In building acoustics the sound power radiated by a vibrating surface is generally determined from the measured sound pressure by assuming a perfectly diffuse sound field. This is a quick experimental procedure, even though it is not very reliable at low frequencies, where the sound field is unlikely to be perfectly diffused, but is governed by the room modal response.

Alternatively, it is possible to derive the radiated sound power from sound intensity measurements. The average sound intensity is determined by scanning the vibrating surface with either a p-p probe (which measures the sound pressure with two calibrated microphones) or a p-u probe (which measures the sound pressure and particle velocity). This is a widely used approach which provides reliable results when the sound intensity probes are used in an anechoic environment. However, its accuracy is affected by a reverberant sound field, especially at low frequencies. The radiated sound power can also be determined by means of hybrid methods from the measured vibration velocity and numeric computation. From the vibration velocity $v\left(\omega, r_S\right)$ measured over a number of points r_S distributed over the panel surface, it is possible, by applying the Rayleigh integral [17, 16], to compute the sound pressure $p\left(\omega, r\right)$ radiated at the position r in the semi-infinite fluid domain:

$$p\left(\omega, r\right) = \frac{\mathrm{i}\omega\rho_0}{2\pi} \iint_S v\left(\omega, r_S\right) \frac{\mathrm{e}^{-\mathrm{i}kR}}{R} \mathrm{d}S \tag{3}$$

where R represents the distance between the position r_S on the panel surface and the position r in the acoustic domain and k_0 is the acoustic wavenumber. Form the sound pressure evaluated at each measured position over the panel surface $p\left(\omega, r_S\right)$ and the complex conjugate of the vibration velocity v^*, it is possible to determine the active sound intensity radiated by the plate as:

$$I\left(\omega, r_S\right) = \frac{1}{2}\mathrm{Re}\left\{p\left(\omega, r_S\right) v^*\left(\omega, r_S\right)\right\} \tag{4}$$

the radiated sound power can thus be computed by integrating the sound intensity over the panel surface:

$$W\left(\omega\right) = \iint_S I\left(\omega, r_S\right) \mathrm{d}S \tag{5}$$

An alternative approach, known as discrete calculation method (DCM) [18], computes the radiated sound power from the complex vibration velocity, measured on a grid of N points evenly spaced over the plate's surface, and the numerically computed self Z_{ii} and mutual radiation impedance Z_{ij}:

$$W\left(\omega\right) = \sum_{i=1}^{N} \left[\mathrm{Re}\left\{Z_{ii}\right\} \left|v_i\right|^2 + \sum_{j=1}^{N} \mathrm{Re}\left\{Z_{ii} v_i v_j^*\right\} \right] \tag{6}$$

Both approaches assume a baffled panel radiating in a semi-infinite space, neglecting thus the room's influence. In order to evaluate the plywood panel's radiation efficiency, the vibration velocity was measured on the investigated panel surface over a grid of 120 points evenly spaced 10 cm apart.

Table II. Geometric characteristics of the investigated plywood panel

	Kirchhoff's interpolation
$E_{\theta=0}$ [Pa]	1.67E+09
$E_{\theta=\pi/8}$ [Pa]	1.40E+09
$E_{\theta=\pi/4}$ [Pa]	1.18E+09
$E_{\theta=3\pi/8}$ [Pa]	1.27E+09
$E_{\theta=\pi/2}$ [Pa]	1.37E+09

3. Results and discussion

3.1. Structural wavenumbers and elastic properties

The elastic properties of the investigated panel were determined from the experimental structural wavenumbers measured along different directions: i.e. 0, $\pi/8$, $\pi/4$, $3\pi/8$ and $\pi/2$. The experimental wavenumbers obtained by maximising the function given in Eq. 1 were fitted using Kirchhoff's dispersion relation for thin plates plates, as shown in Figure 1. The fitting algorithm provides the estimated elastic modulus associated to the propagation direction, as reported in Table II. Like other timber layered structures, the investigated panel exhibits a slightly anisotropic behaviour, with the greatest elastic modulus along the direction parallel to the external layer fibres orientation: $\theta = 0$, while along the diagonal directions smaller elastic moduli were found.

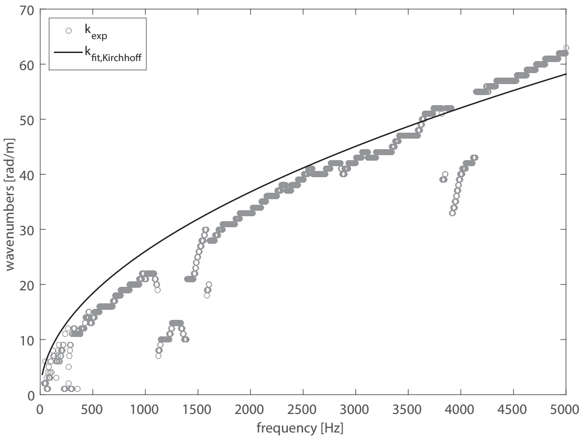

Figure 1. Mounting condition of the investigated CLT plates

3.2. Plate dynamic response and radiation efficiency

From the complex dynamic response, measured on a grid of 120 points, it is possible to visualise the mode shapes of the mechanically excited panel. Figure 2 shows, as an example, the plate's resonant modes identified between 30 Hz and 300 Hz. The experimental radiation efficiency, evaluated from the plate dynamic response by using the DCM approach [18], is given in Figure 3 in terms of radiation index $L_\sigma = 10\log\sigma$. In the low frequency range the radiation behaviour is governed by the plate modal response. The lower radiation mode, identified by the peak around 44 Hz, is associated with the first plate resonance. A rising number of structural modes contribute to sound radiation as the frequency increases. The critical condition, representing the frequency at which the structural wavenumber propagating along any possible azimuthal direction of the plate is equal to or smaller than the acoustic wavenumber propagating in the air, is found between 2000 Hz and 2500 Hz, where the curve of the radiation index exhibits a sharp peak. The critical condition has been consistently verified by comparing the structural wavenumbers k_{θ_i} along the five investigated directions with the acoustic wavenumber k_0. As shown in Figure 4, the first coincidence, associated with the direction $\theta = 0$, is found around 2000 Hz, while the last coincidence, which also represents the critical condition, falls around 2400 Hz and is associated with the propagation angle $\theta = \pi/4$.

3.3. Results validation

In this section, the results previously presented are compared with the experimental data obtained by standard techniques, in order to verify the reliability of described procedure. The radiation index $L_{\sigma,DCM}$ determined by means of the DCM method from the vibration velocity, measured on a grid of points, is compared in Figure 3 to the radiation index $L_{\sigma,S.Int.}$ obtained from sound intensity measurements, performed on the same structure using a p-p probe into the anechoic chamber at the Engineering Department of the University of Ferrara. It should be noticed that in this latter case it is not possible to use a sine sweep excitation signal, therefore the shaker was driven by a white noise instead. Also considering that two different excitation signals were used, a good agreement is shown between the two curves, even though the radiation index obtained from sound intensity measurement is noisier than the results determined by using the DCM approach; moreover, the peak associated with the critical condition is less emphasised, meaning that a lower radiated sound power was measured by using the sound intensity probe.

To further validate the proposed methodology, the same panel was also investigated into the sound transmission test facility for small elements of Adler Evo acoustic laboratories, in Turin, Italy. In this case it was possible to acoustically excite the panel. Two sources were used to generate the incident sound field within the reverberant emission room, driven with a white noise signal. The sound pressure exciting the plate was measured by means of two rotating microphones, while, on the semi-anechoic receiving side, the radiated sound intensity was measured by scanning the panel's surface with a p-p probe. The plate transmission loss TL was determined as:

$$TL = L_{p,1} - L_i - 6 \tag{7}$$

where $L_{p,1} = 20\log(p_1/p_0)$ is the sound pressure measured in the emitting room (dB re 20μPa) and L_i is the sound intensity level measured on the receiving side (dB re 10^{-12}W). Moreover, in order to compute the radiated sound power, the panel's dynamic response was measured over a grid of points equally spaced 5 cm apart, as described in the previous sections. In Figure 5 the radiated sound power level spectrum $L_{W,DCM}$, computed from the surface vibration velocity using the DCM approach, is compared with the sound power level $L_{W,S.Int.}$ determined from sound intensity measurements. A very good agreement is found between the spectra levels obtained from the two approaches within the entire frequency range.

The panel's experimental transmission loss TL, determined from eq. 7 is shown in Figure 6 both in narrow frequency and in one-third octave bands. The critical condition, identified by the dip around 2000 Hz is consistent with the findings obtained from sound radiation and wavenumber analysis.

4. CONCLUSIONS

In this paper, an alternative test rig to perform a vibro-acoustic analysis on a lightweight panel, possibly used in building construction, has been presented.

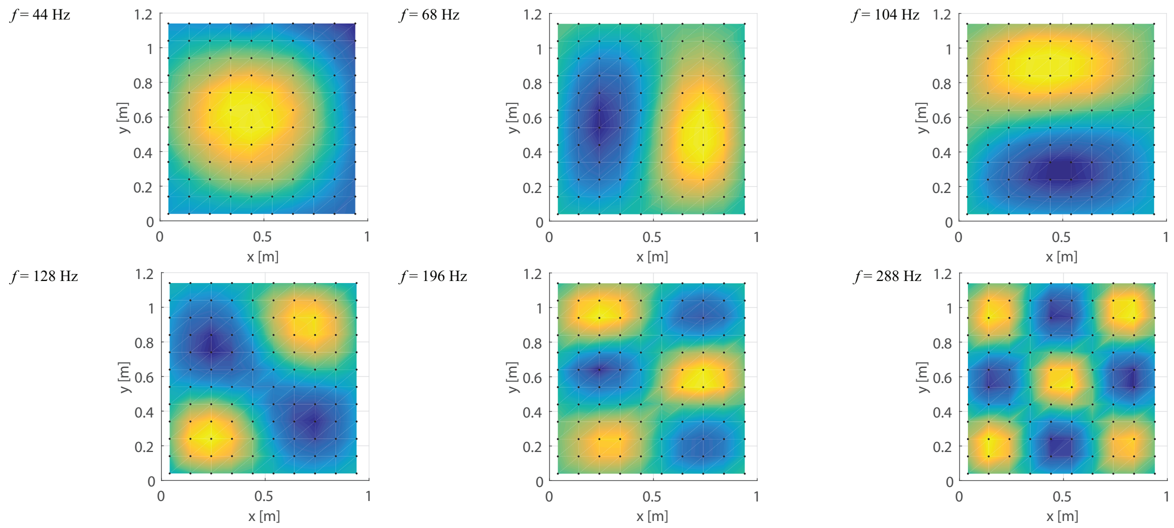

Figure 2. Experimental mode shapes of the investigated plate, identified between 30 Hz and 300 Hz

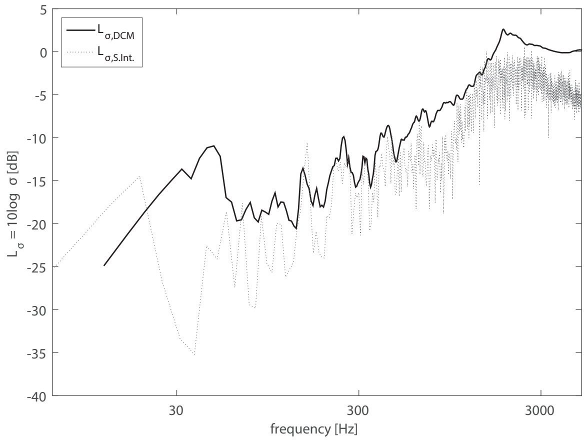

Figure 3. Experimental radiation index of the plywood panel: comparison DCM and sound intensity results.

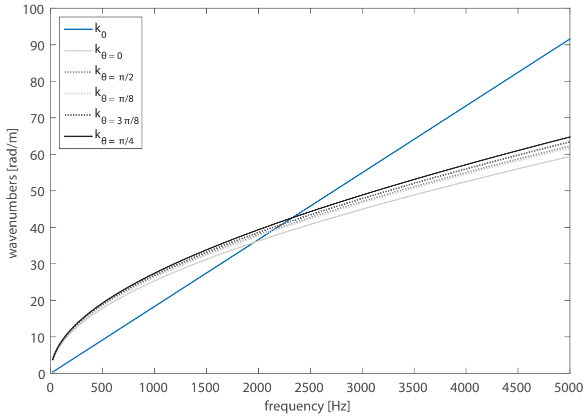

Figure 4. Comparison between the structural wavenumbers along different directions k_{θ_i} and the acoustic wavenumber k_0

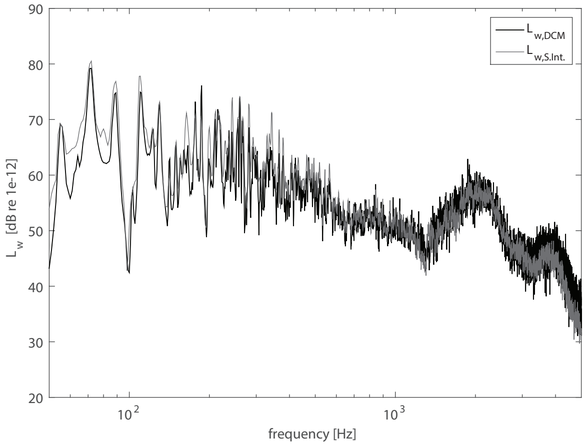

Figure 5. Sound power levels radiated by the plywood panel acoustically excited by a white noise signal determined from the experimental sound intensity and vibration velocity.

The investigated element, mounted in a steel frame clamping all four boundaries, was excited by a shaker driven with a sine sweep signal, while the panel's dynamic was determined by measuring the structural impulse responses, associated with several positions distributed on the panel's surface, with miniature accelerometers. The elastic properties of a plywood board were determined by applying a wave correlation method from the plate's response measured along a line of points aligned with the excitation position, for different directions. At the same time, the vibration velocity measured over a grid of points evenly distributed over the panel surface, was mapped in order to visualise the resonant mode shapes of the structure. Moreover, it was possible to determine the panel's radiation efficiency by using a hybrid method which combines experimental data and numerical calculation, obtaining results which were consistent with the radiation efficiency evaluated from sound intensity measurements. The coincidence region of the non-isotropic investigated structure identified from the structural wavenumbers is consistent with the one

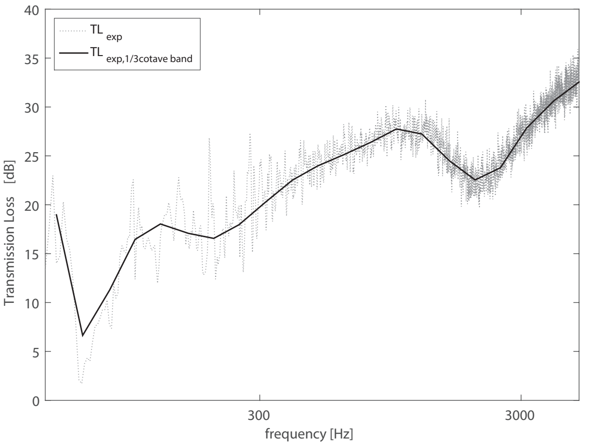

Figure 6. Sound transmission loss of the plywood board; comparison between experimental data (both in narrow band and one-third octave band) and results from transfer matrix method (one-third octave band).

highlighted by the radiation efficiency. In order to further validate the entire procedure, additional measurements of the radiated sound power with acoustic excitation and sound transmission were performed. Once again, no significant differences were shown between the radiated sound power derived from sound intensity measurement and the sound power computed from the experimental vibration velocity. Results of sound sound transmission loss measurement were consistent with the findings obtained by investigating the structural wavenumbers and sound radiation. It has been shown that it is possible to perform a vibro-acoustic characterisation of a lightweight partition on a small sample, inserted into a steel frame, obtaining reliable results and reducing time and costs. This setup can be useful to quickly characterise the structural elastic properties needed in sound radiation or sound transmission prediction models.

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