Dynamic and Acoustic Properties of a Joisted Floor

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Abstract: Lightweight structures find more and more applications in both vehicle and ship industries. To meet a growing demand, a variety of different types of joisted panels have been developed during the last few decades. One of the problems to deal with is the assessment of the acoustic performances of such panels once they are already mounted in their final place. In this case, it can be of importance to find a way to characterise their dynamic and acoustic properties, such as bending stiffness, internal losses and sound reduction index through non-destructive testing. A method for a quick determination of the bending stiffness of a lightweight joisted floor is presented. On the basis of the apparent bending stiffness and of the losses, it is possible to predict the sound reduction index of the panel in a fairly simple way. The results obtained from the mobility tests have been compared to the measurements carried out according to the ISO standard procedure.

1 INTRODUCTION

The expression "joisted floor" refers to a structure with a multi-layer plate bonded to joists placed at one side of the floor. This type of floor combines low weight with high strength. However the acoustic properties can be very poor, thus severely restricting the use of such lightweight elements. It is therefore essential to optimize the acoustic performances through predictions. In case of sandwich-like structures, some of the basic dynamic and acoustic parameters can be determined by means of simple tests using a beam element cut from the assembly (Nilsson and Nilsson, 2002) and (Nilsson and Liu, 2016). Some frequency response measurements can determine a number of natural frequencies of the beam. Based on these results the apparent bending stiffness can easily be determined through least square method applied to the experimental points. The apparent bending stiffness of a composite structure at one natural frequency is equal to the bending stiffness of a simple Euler beam having the same length, boundary conditions and weight as the considered sandwich structure at the same frequency. Obviously, it is not always possible to cut beams from an already mounted structure. (Roelens et al., 1997) and (Nightingale et al., 2004) have tested different measurement techniques on building components for the determination of their stiffness properties *in situ*. Although these methods seem to be quite complicated and a number of them cannot be adopted for some types of building components.

In the following sections, a method is presented through which the material parameters can be determined from simple point mobility measurements on a plate element. In particular, this method has been applied to a kind of panel which cannot be strictly defined as sandwich, since it is made up of several layers and some thick joists attached to one side of the assembly. It will be shown that the point mobility technique allows to estimate the apparent bending stiffness also in this case, thus taking into account the real boundary conditions of the mounted structure.

The sound reduction index results obtained from point mobility measurements will be compared to those found after the tests carried out according to the existing ISO standards.

2 BENDING STIFFNESS DERIVED FROM POINT MOBILITY MEASUREMENTS

If an harmonic force $F = F_0 \exp(i\omega t)$ is injected at a specific point of a system, it will move with a certain velocity v. In the point where the excitation is given, a point mobility function Y can be defined as the ratio between the Fourier transform of the velocity signal and the Fourier transform of the force signal measured at the same position:

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$$Y(\omega) = \hat{v}(\omega) / \hat{F}(\omega) \tag{1}$$

The behaviour of a finite vibrating structure can be predicted from that of infinite ones. The bending waves induced by a point force in an infinite plate can spread indefinitely in any direction. If a finite plate is considered, the bending waves reach the borders of the plate, and then are reflected back. If a force is acting at a specific point of the plate, the resulting velocity will be mainly determined by the plate dimensions, mass per unit area, bending stiffness and boundary conditions, thus the point mobility will depend on position and frequency. However, an averaging of the real part of the point mobility carried out over space and frequency for a finite structure is in the mid- and high-frequency region identical to the real part of the point mobility of an infinite structure made of the same material and having the same thickness:

$$\mathrm{Re}\langle \overline{Y}(\omega) \rangle = \mathrm{Re}(Y_\infty(\omega)) \tag{2}$$

Consequently, the power input, introduced into a panel having finite dimensions by a force acting randomly in time and space, can be determined as if the panel had infinite dimensions and were excited by a point force having a power spectral density equivalent to the sum of the power spectral densities of all the point sources operating on the finite structure. This statement is valid because in the mid-high frequency range the modal density is significant, thus making the mobility independent of the structure extension. This means that the theory cannot be applied without taking into account corrections for the first few modes in the low frequency range. It has been be shown (Fahy and Mohammed, 1992) that, in order to extend this consideration to the low frequency range, at least 5 modes have to be included within each frequency band of interest to have a fair accuracy. As concerns the space average, the mobility has to be measured over a large number of points, and the points have to be randomly distributed over the panel surface, in order to obtain a good representation of the dynamic properties of the panel. If these conditions are fulfilled, the mobility of a finite panel can be determined using the formulation of the mobility for an equivalent infinite structure (Nilsson and Liu, 2016). In this case, the frequency average of the mobility can be written as:

$$\mathrm{Re}\langle \overline{Y} \rangle = \frac{1}{8\sqrt{D_p \mu''}} \tag{3}$$

where D_p and μ'' are the bending stiffness per unit width and the mass per unit area of the panel, respectively. The bending stiffness per unit width of the panel at the central frequency of each frequency band is obtained as:

$$D_p = \frac{1}{64\mu''\left[\mathrm{Re}\langle \overline{Y} \rangle\right]^2} \tag{4}$$

For modes (m,n) having $m = 0$ or $n = 0$, it can be shown that (Nilsson and Liu, 2016):

$$\mathrm{Re}\langle \overline{Y} \rangle = \frac{1}{4\sqrt{D_p \mu''}} \tag{5}$$

Therefore, for the first natural frequencies corresponding to such mode types, the measured bending stiffness should be multiplied by a factor 4.

The influence of the mass of the accelerometer used during the mobility measurements has to be taken into account, especially for lightweight structures. As discussed in (Nilsson and Liu, 2016), the dynamic response of the structure and its modal behaviour, can be influenced by the added mass of the transducer $\Delta\mu$. The measured point mobility should then be modified according to the correlation:

$$Y_{measured} = Y / (1 + Y \cdot \Delta\mu \cdot i\omega) \tag{6}$$

In the low frequency region the effect of the mass $\Delta\mu$ is negligible. For higher frequencies, the denominator increases and the magnitude of the measured point mobility can decrease significantly.

This aspect must be taken into account in case of lightweight structures when the mobility is measured by using an impact hammer and an accelerometer, while an impedance head is less sensitive.

2.1 Sound Reduction Index

The response of a structure excited by an external sound field can be predicted fairly accurately once the apparent bending stiffness of the structure is known (Backström and Nilsson, 2007).

The derivation of the sound reduction index for homogeneous panels as a function of the bending stiffness of the plate and other parameters is discussed in (Cremer, 1942). The expressions can be used for sandwich structures once some adjustments are made. For this reason it is useful to introduce the critical frequency f_c. This particular parameter is the frequency for which the wavenumber in air is equal to the wavenumber of the flexural waves on the plate. The frequency f_c is given by

$$f_c = (c/2\pi) \cdot k_{plate}^2 / k_{air} = (c^2/2\pi) \cdot \sqrt{\mu''/D_p} \tag{7}$$

where c is the speed of sound in air. For a thin homogeneous panel, the bending stiffness D_p is not frequency dependent and f_c is a constant.

The transmission coefficient $\tau(\varphi)$ at the angle of incidence φ is given by

$$\tau(\varphi)= \\ =\left\{\left[1+\frac{\mu''\omega}{2\rho c}\cos\varphi\cdot\left(\frac{f}{f_c}\right)^2(\sin\varphi)^4\eta\right]^2+ \right. \\ \left. +\left[\frac{\mu''\omega}{2\rho c}\cos\varphi\cdot\left\{\left(\frac{f}{f_c}\right)^2(\sin\varphi)^4-1\right\}\right]^2\right\}^{-1} \quad (8)$$

where ρc is the impedance of air, equal to 415 kg/(m^{2}s) and f_c is the critical frequency satisfying equation 7. As previously stated, for a single leaf panel D_p is constant. However, for a multylayered structure the bending stiffness is frequency dependent. The sound reduction index of a sandwich structure can be derived by replacing f_c in equation 8 with a the parameter f'_c:

$$f'_c=\frac{c}{2\pi}\sqrt{\frac{\mu''}{D_p(f)}} \quad (9)$$

The sound reduction index R is defined as $10\cdot\log(1/\tau_d)$, where τ_d is the sound transmission coefficient for diffuse incidence:

$$\tau_d=2\int_0^{\pi/2}\tau(\varphi)\cos\varphi\sin\varphi\cdot d\varphi \quad (10)$$

The sound reduction index for the plates described hereafter was predicted according to equations 8-10 and compared to measurements carried out according to the procedures described in the international standards.

2.2 Specimen under Test

The investigated specimen is made of two main parts: a floor, made of different layers, and the joists (Figure 1).

Starting from the upper layer, the thicknesses are: 20 mm, 5 mm, 2 x 10mm, 8 mm, 140 mm, 22 mm. The joists are 120 mm x 160 mm wooden studs. The overall thickness of the floor is 215 mm.

The dimensions of the surface are 3.31 m x 3.38 m. Figure 2 shows a layout of the floor, together with the spacing between the ribs. The lower wooden layer and the studs are kept together by a thin layer of glue.

The overall mass per unit area of the floor is 85.08 kg/m^2.

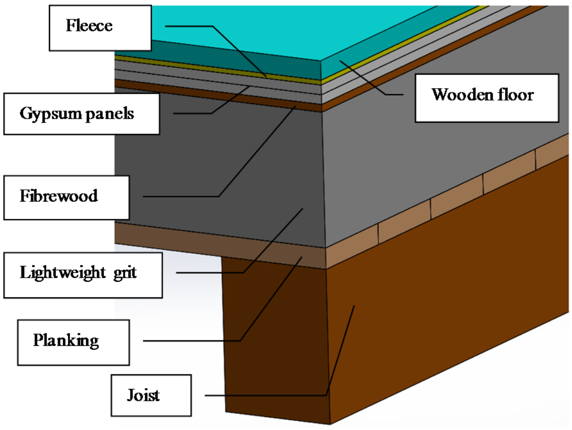

Figure 1: Layers of the joisted floor.

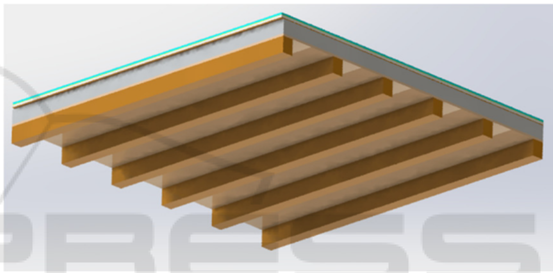

Figure 2: Layout of the ribs and main dimensions of the panel.

3 TEST METHODS AND MATERIALS

The floor is mounted between the ground floor and the second floor of a small building (Figure 3).

The lateral sides of the floor are sealed by placing a resilient mat plus high density foam to fill all the gaps between the wooden frame and the wall.

On the joisted side, 20 point mobility measurement positions were spread on the surface: 10 positions on the joists and 10 positions in between them.

A PCB Piezotronics accelerometer type 352C33 was attached to the floor and then the floor was his as close as possible to it by means of an impedance hammer PCB Piezotronics type 086C03, equipped with a nylon tip. The velocity and force signals coming from the transducers were acquired by an OROS 36 multi-channel system able to compute directly the real and imaginary parts of the mobility function. The frequency span of the acquisition was selected from 0 to 6.4 kHz, 1 Hz resolution.

Figure 3: Pictures of the floor used for the tests and of the building where it is mounted.

3.1 Point Mobility Measurements

The point mobility measurements were performed directly *in situ*.

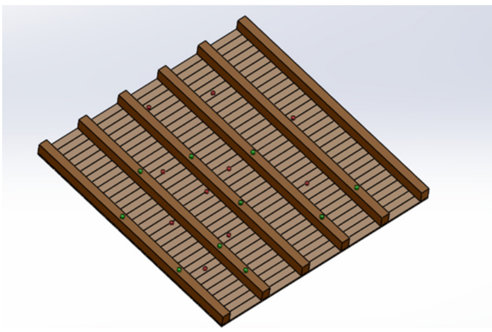

Figure 4: Measurement positions for point mobility.

Figure 4 shows how the measurement positions are distributed across the panel surface.

The post processing of the data was carried out by exporting the text data from the OROS NVGate software. Importing the real part of the mobility in an ad-hoc software, it was possible to compute the average mobility for the 20 measurement positions. Finally, the mobility value was computed synthesising the values into 1/3 octave bands, extended to have at least 5 modes inside the frequency span defined by each band. Once the average mobility is known, it is possible to compute the related bending stiffness and to use this value to determine the apparent bending stiffness through the least square method applied to a set of points f_n, D_n and to the following equation

$$\frac{A}{f}D_x^{3/2} - \frac{B}{f}D_x^{1/2} + D_x - C = 0 \tag{11}$$

which describes the general behaviour of the apparent bending stiffness D_x for a sandwich panel (Nilsson and Nilsson, 2002). As shown in (Piana, 2016), the general form of equation 11 can be used to describe the bending stiffness of different types of orthotropic panels, including ribbed structures.

Since the modal density in the low frequency range is low, there is some lack of points for computing the bending stiffness. For this reason a fictitious bending stiffness point D_0 has been introduced in order to “guide” the curve in the very low frequency region. The static bending stiffness D_0 can be computed once some geometrical and material parameters are known using the following equation:

$$D_0 = \frac{E_l h_c^2 h_l}{2} \tag{12}$$

where E_l is the Young’s modulus for one laminate, h_c is the core thickness and h_l is the thickness of one laminate.

4 RESULTS

4.1 Determination of the Loss Factor

Before starting with the computation of the sound reduction index of the panels, it is necessary to determine the losses. The determination of the losses was made through the evaluation of the structural reverberation time.

Each of the impulse responses recorded in the positions used for the measurement of the mobility was post processed to determine the decay in each octave band of interest. Then it was possible to compute the losses through the formula:

$$\eta = \frac{2.2}{f_0 T_r} \tag{13}$$

where f_0 is the central frequency of the octave band of interest in hertz, and T_r is the measured structural reverberation time in seconds for each frequency band. The decay for the different frequency bands was obtained by post processing the impulse response signals using Adobe Audition and the Aurora plugin. The resulting decays, computed for the frequency bands of interest, were further post processed in order to obtain the losses as a function of frequency (Figure 5).

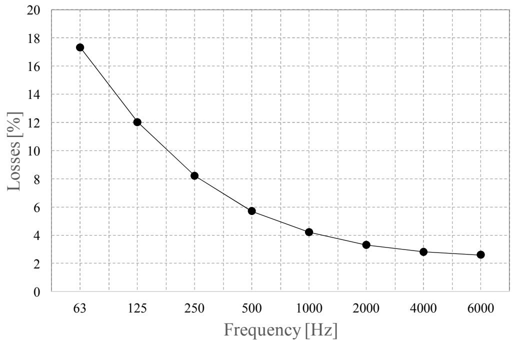

Figure 5: Measured losses of the joisted floor.

4.2 Sound Reduction Index from Mobility Measurements

Once the losses have been determined and the apparent bending stiffness has been derived from the mobility measurements, it is possible to compute the sound reduction index of the panel. The computation is carried out according to the theory described in the previous sections, and in particular by using equation 4.

Starting from the dimensions of the panel, its mass for unit area and the mobility measurements, the bending stiffness is computed in the frequency range 1 Hz – 6400 Hz. Figure 6 shows the bending stiffness points, derived from the point mobility measurements, and the apparent bending stiffness function, obtained by fitting equation 11 to the experimental data.

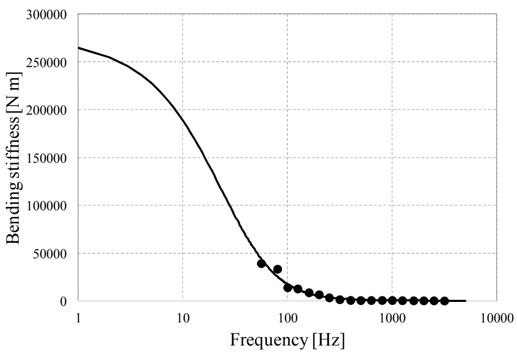

Figure 6: Measured bending stiffness (points) and apparent bending stiffness (solid line).

Once the bending stiffness is computed, the critical frequency f_c can be easily determined and the sound reduction index can be computed.

4.3 Measurement of the Sound Reduction Index to ISO 10140-2

After performing the mobility measurements, the joisted floor was tested according to the procedure described in ISO 10140-2 standard in order to determine its sound reduction index of the structure. Two source positions were used so to have a good average of the sound field and ten sound pressure level measurements were performed for both the source and the receiving rooms. The difference between the average sound pressure level of the source room (L_{SR}) and the average sound pressure level of the receiving room (L_{RR}) was weighted for the size of the partition S and the sound absorption area of the receiving room A_{RR} to compute the sound reduction index according to the following formula:

$$R = L_{SR} - L_{RR} + 10\log\frac{S}{A_{RR}} \tag{15}$$

Figure 7 shows the comparison between the sound reduction index measured using the procedure given by the standard and the reduction index resulting from the mobility measurements, which displays a fairly good agreement. The dotted line represents the sound reduction index computed according to the mass law for a floor having the same mass per unit area of the one used for the experimental tests. It can be noted that the mass law brings to a sound reduction index which is at least 10 dB higher than the real one for each frequency band of interest.

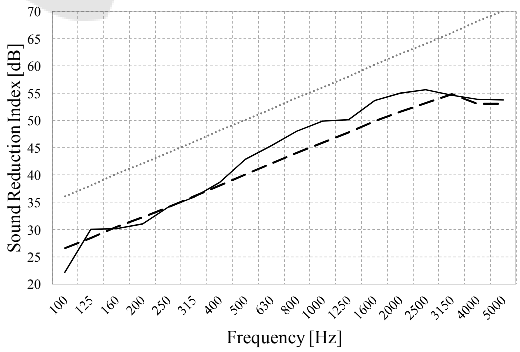

Figure 7: Measured (solid) vs predicted (dashed) sound reduction index.

5 CONCLUSIONS

A simple technique for the prediction of the sound reduction index of lightweight joisted floors has been proposed and tested. The technique has already been adopted for predicting the dynamic and acoustic properties of sandwich and honeycomb panels, but seldom on lightweight, strongly orthotropic structures. The method is based on *in-situ* point mobility measurements performed on a sufficiently high number of experimental points randomly distributed over the specimen surface, which, once space- and frequency-averaged, allow to compute the apparent frequency-dependent bending stiffness of the structure. This technique is easier to implement if compared to other methods found in the literature, and it allows to better take into account the anisotropic behaviour of the specimens. The predicted sound reduction index has been compared to measurements performed according to the relevant international standard, showing encouraging results. Further experimental tests will allow to validate the applicability of the method to other types of structures, such as double-wall panels.

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