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Discussion

Comments on "On-line non-destructive evaluation and control of wood-based panels by vibration analysis"

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The authors are to be congratulated for their interesting paper [1], where they present a non-destructive apparatus (Vibra Pann), which has been specially developed for testing wood-based panels. It is shown that, from an industrial viewpoint, the setup yields approximate but useful results for the bending stiffness and strength of wood-based panels along their material axes when considered as orthotropic materials.

On the other hand, the writers feel that certain technical–scientific aspects of their paper must be clarified for a better understanding of their industrial paper.

Eq. (1) in their paper is valid for orthotropic plates when the plate executes small-amplitude, transverse vibrations and shear and rotary inertia effects are taken into account. In-plane inertia forces are seldom accounted for when using the classical linear theory of thin plates. Hence Eq. (3) in Ref. [1] constitutes the reduced version of Eq. (1) when one disregards shear and rotary inertia (certainly in-plane inertia forces do not come into play when using the proposed mathematical model).

When using vibrations for the NDE of bending properties, the authors write down the frequencies corresponding to the two vibrating modes, defined by them as (2,0) and (0,2) and shown in their Fig. 2, which clearly indicates "beam" vibrational modes since they depict the nodal lines as straight lines. Even in the case of an isotropic plate, the nodal lines are curved segments—as shown in Figs. 1 and 2 of the present discussion.

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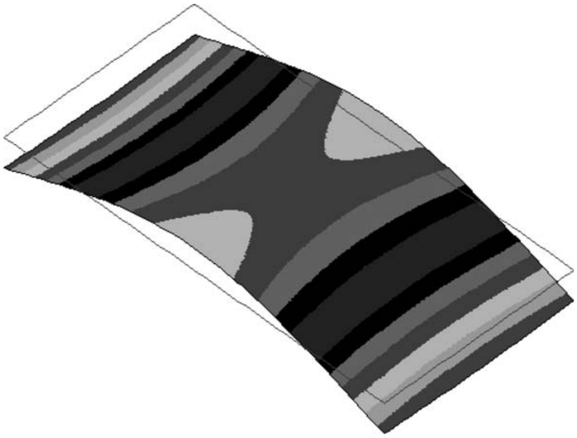

Fig. 1. Mode (2,0) in the case of an isotropic plate of aspect ratio equal to 2 (obtained using Ref. [2]).

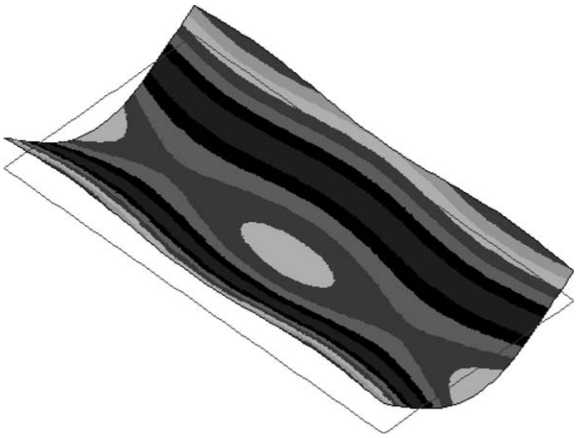

Fig. 2. Mode (0,2) in the case of an isotropic plate of aspect ratio equal to 2 (obtained using Ref. [2]).

On the other hand Eq. (5a) of Ref. [1] reads

$$f(2,0) = \frac{1}{2\pi}\sqrt{\frac{1}{\rho h}}\sqrt{D_{xx}\,\frac{500.6}{L^4}},$$

which can conveniently be expressed in the form

$$f(2,0) = \frac{1}{2\pi}\sqrt{\frac{1}{\rho h}}\sqrt{D_{xx}}\,\frac{22.37}{L^2}. \tag{1}$$

It must be recalled that the factor (22.37) is the exact frequency coefficient corresponding to the first non-trivial mode of a free–free beam, as rightly suggested by the authors. Similar considerations are valid with respect to Eq. (5b).

In conclusion, Eq. (5) could have been the starting point of the analysis without referring to the partial differential Eqs. (1) and (3) of Ref. [1], simply explaining that for the modes (2,0) and (0,2) the plates are replaced by "beams" taking into account a bi-axial state of stress (which is equivalent to say that it is assumed that the plate executes "cylindrical bending" in the x-direction for the (2,0) mode and afterwards in the y-direction. Certainly, in the case of the completely free rectangular plate, the curvature effect in two orthogonal directions is always present and the actual frequencies of the modes (2,0) and (0,2) will be, in general, lower than those predicted by Eq. (5).

With regard to the "nodal points" defined by the authors as $\pm\frac{1}{4}L$, $\pm\frac{1}{4}l$ one can see in Figs. 1 and 2 in the case of an isotropic plate ($L/l = 2$) that the distances to the free edges, measured along the adjacent edges, are $\pm 0.209L$ and $\pm 0.194l$, respectively, but it must be kept in mind that the nodal lines are curved segments. These parameters change as a function of L/l and Poisson's ratio (for the present determinations, it has been taken equal to 0.30). In the case of orthotropic plates, the two parameters above mentioned will be functions of all the elastic moduli.

For the isotropic plate depicted in Figs. 1 and 2, the frequency coefficients ($\Omega = \sqrt{(\rho h/D)}\omega L^2$) are $\Omega_{20} = 21.45$ and $\Omega_{02} = 87.84$. If one refers Ω_{02} to the dimension "l", one will have 21.96. It must also be recalled that for a vibrating free–free beam the distances of the nodal points to the edges are $0.22L$.

It must be emphasized that the writers recognize the significant importance and merit of the authors' contribution from an industrial viewpoint. However, from a technical–scientific viewpoint, the clarifications made by the writers are mandatory, especially if one considers that the prestigious Journal of Sound and Vibration reaches a wide audience, and keeping in mind that young researchers need rigorous thinking when developing their work.

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[2] ALGOR Professional Mech VE. Linear Stress and Dynamics Reference Division, Pittsburgh, PA, 2001.