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Letter to the Editor

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

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1. Introduction

Non-destructive techniques of control and/or evaluation have been widely used to characterize any sort of material for a long time. Wood materials do not escape this tendency and these techniques tend to replace most traditional methods (visual classifications and static tests) and to become one of the key issues for an optimized wood chain, from the forest to the building (or furniture, or packaging) industry. Most of the methods developed are based on simple physical principles (vibration analysis, ultrasonic measurement, drilling apparatus), which offer numerous advantages:

- adaptation to all the dimensions of products,
- fast tests and immediate answers,
- low price investment in scientific material,
- appropriate methods for all the range of wood materials (logs, beams, panels), etc.

This paper presents a non-destructive apparatus, called VibraPann,[1] especially developed for wood-based panels. The originality of this work is the use of some particular properties of plates vibrations, and more exactly free vibrations, to build a simple bending device particularly appropriate to an on-line procedure for the control of a fwide range of wood-based panels.

The experimental results show that VibraPann measurements can be used to estimate four mechanical properties that are commonly used as quality control parameters in industrial applications: the bending stiffness and strength of wood-based panels along their material axis. The bending stiffnesses are directly calculated from vibration measurements while the mechanical strengths are obtained using correlations. In each case, VibraPann estimations, carried on whole

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[1] Brevet n° 0101162, déposé le 29 janvier 2001: Procédé d'évaluation et de contrôle non destructif de la qualité de panneaux. Inventeurs : Stéphanie BOS-CASAGRANDE/XYLOMECA, Frédéric BOS/LRBB, Jean-François DUMAIL/XYLOMECA.

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panels, are correlated with the similar average properties calculated, according to standard tests, from several small specimens.

2. Theoretical aspects of free vibrations

The pertinence of vibration analysis in order to evaluate and control mechanical properties has been known for a long time. The measurement of natural frequencies can be used for the estimation of stiffness properties and also for the localization of defects in structures [1,2]. An interesting literature survey of fundamental equations and theory is given by Leissa [3]. The flexural vibrations of an orthotropic rectangular plate are governed by differential equations. A complete formulation of the equations of motion, including shear deformations and rotary inertia, can be obtained considering the formulations of Mindlin [4] and Timoshenko [5] established, respectively, for isotropic plates and orthotropic beams:

$$\begin{aligned} &D_{xx}\frac{\partial^4 w}{\partial x^4} + 2(v_{yx}D_{yy} + 2D_k)\frac{\partial^4 w}{\partial x^2 \partial y^2} + D_{yy}\frac{\partial^4 w}{\partial y^4} \\ &\quad - \rho\left(\frac{h^3}{12} + \frac{1}{\beta' G_{xy}}\right)\left(D_{xx}\frac{\partial^4 w}{\partial x^2 \partial t^2} + D_{yy}\frac{\partial^4 w}{\partial y^2 \partial t^2}\right) \\ &\quad + \frac{h^3}{12}\frac{\rho^2}{\beta' G_{xy}}\frac{\partial^4 w}{\partial t^4} = -\rho h\frac{\partial^2 w}{\partial t^2}, \end{aligned} \tag{1}$$

where E_x and E_y are Young's moduli, G_{xy} is the in-plane shear modulus, v_{xy} and v_{yx} are the Poisson ratios, h is the thickness of the plate, ρ is the mass density, β' is a constant similar to the β factor used for orthotropic beams and

$$D_{xx} = \frac{E_{xx}h^3}{12(1 - v_{yx}v_{xy})}, \quad D_{yy} = \frac{E_{yy}h^3}{12(1 - v_{yx}v_{xy})}, \quad D_k = \frac{G_{xy}h^3}{12}. \tag{2}$$

For a thin rectangular orthotropic plate several simplifications can be made: in-plane inertia forces and rotary inertia are neglected [6]. Under such conditions, Eq. (1) can be simply written as

$$D_{xx}\frac{\partial^4 w}{\partial x^4} + 2(v_{yx}D_{yy} + 2D_k)\frac{\partial^4 w}{\partial x^2 \partial y^2} + D_{yy}\frac{\partial^4 w}{\partial y^4} = -\rho h\frac{\partial^2 w}{\partial t^2}. \tag{3}$$

For certain test configurations (all edges free FFFF or all edges clamped CCCC), Eq. (3) has no closed-form solutions: the differential equation of motion should be solved numerically. Under free boundary conditions, FFFF, the Rayleigh–Ritz method gives a good estimation of the resonant frequencies $f(i,j)$ for an orthotropic plate:

$$f(i,j) = \frac{1}{2\pi}\sqrt{\frac{1}{\rho h}}\sqrt{D_{xx}\frac{\alpha_1(i,j)}{L^4} + D_{yy}\frac{\alpha_2(i,j)}{l^4} + 2D_{xy}\frac{\alpha_3(i,j)}{L^2 l^2} + 4D_k\frac{\alpha_4(i,j)}{L^2 l^2}}, \tag{4}$$

where L, l and h are, respectively, the length, the width and the thickness of the panel. (i,j) identifies the vibrating modes and represent the number of half-wave lengths in each principal directions of the panel. The global mode shape of a plate results in the superposition of different

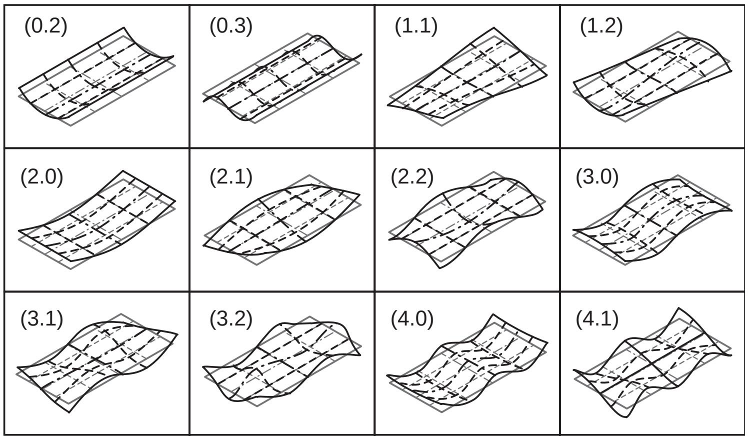

Fig. 1. Several mode shapes of a thin plate. (i,j) represent the number of half-wave lengths in each principal directions of the panel. The global mode shape of a plate results in the superposition of different mode shapes corresponding to the excited modes.

Table 1
Values of the coefficient α_k for different vibrating modes

i	j	α_1	α_2	α_3	α_4
Vibrating modes					
1	1	0	0	0	144
0	**2**	**0**	**500.6**	**0**	**0**
0	3,4,5,…	0	Y^4	0	0
2	**0**	**500.6**	**0**	**0**	**0**
3,4,5,…	0	X^4	0	0	0

$X = (i - 0.5)\pi$ and $Y = (j - 0.5)\pi$. Bold numerals correspond to the bending modes.

modes shapes corresponding to the excited modes. Several mode shapes are presented in Fig. 1 and the coefficients α_k are given in Table 1.

3. Using vibrations for NDE of bending properties

For an industrial point of view, the bending moduli along the material axis ($E_{11} = E_0$ and $E_{22} = E_{90}$) constitute some important properties of a wood-based panel that are commonly used as quality control parameters. Using a modal analysis, their determination only require the identification of the resonance frequencies of the two vibrating modes (2,0) and (0,2) (Fig. 2). Effectively, according to Eq. (4) and Table 1, the following applies

$$f(2,0) = \frac{1}{2\pi}\sqrt{\frac{1}{\rho h}}\sqrt{D_{xx}\frac{500.6}{L^4}}, \quad f(0,2) = \frac{1}{2\pi}\sqrt{\frac{1}{\rho h}}\sqrt{D_{yy}\frac{500.6}{l^4}} \qquad (5a,b)$$

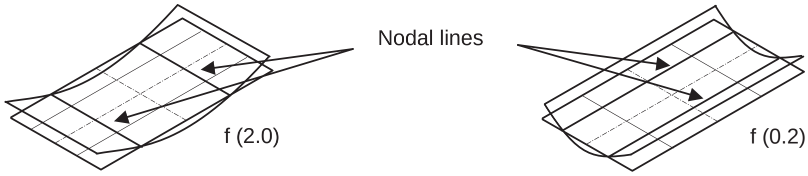

Fig. 2. Mode shapes and nodal lines corresponding to the vibrating modes (2,0) and (0,2).

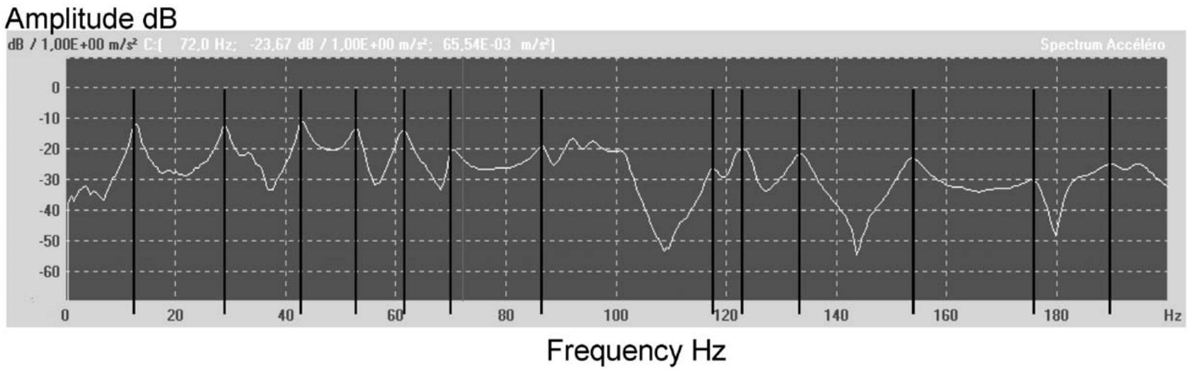

Fig. 3. Frequency spectrum obtained for an OSB panel under free vibrations (panel hangs up on rubber ties).

and finally,

$$E_0 = \frac{48\pi^2 \rho L^4}{500.6h^2}\frac{f^2(2,0)}{(1 - v_{xy}v_{yx})}, \quad E_{90} = \frac{48\pi^2 \rho l^4}{500.6h^2}\frac{f^2(0,2)}{(1 - v_{xy}v_{yx})}. \tag{6a, b}$$

If $v_{xy}v_{yx} \ll 1$, Eq. (6a) is similar to the one obtained for a beam under natural vibrations.

Several authors have used modal analysis to determine the mechanical behaviour of wooden plates. Hearmon [7,8] was certainly the first; Larson [9], Carfagni and Mannuci [10] and Schulte et al. [11] have continued more recently. Different approaches and boundary conditions have been studied (FFFF, CFFF, all edges simply supported SSSS, etc.) and all these works conclude that this powerful method gives good correlation between dynamic and static moduli. However, it seems that there is no suitable system today for on-line use and panel production is still essentially based on experience with little feedback on quality indicators. In fact, there are two main problems with vibration analysis techniques:

- first, the boundary conditions are often difficult to implement in an industrial context; and
- secondly, it is difficult to make systematic the identification of the interesting resonance frequencies in the frequency spectrum (Fig. 3 and Table 2) because their order can vary for each kind of panel.

Table 2
An example of the order of the resonance frequencies obtained for different wood panels tested under free vibrations (panel hangs up on rubber ties)

	$f(1,1)$	$\mathbf{f(2,0)}$	$f(2,1)$	$f(3,0)$	$f(3,1)$	$\mathbf{f(0,2)}$	$f(1,2)$	$f(4,0)$
Plywood, 18 mm	1	**2**	3	4	5	**6**	7	8
OSB, 15 mm	1	**2**	3	5	—	**4**	6	—
Particleboard, 16 mm	1	**2**	3	4	5	**6**	—	—
MDF, 19 mm	1	**2**	3	6	—	**4**	5	—

Bold numerals correspond to the bending modes.

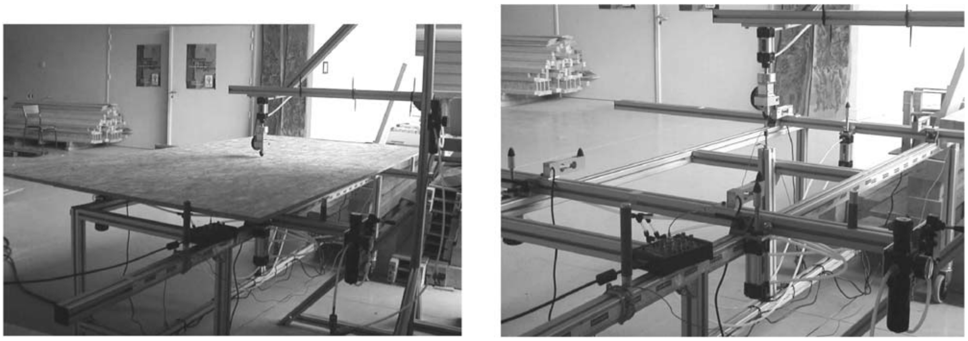

Fig. 4. Vibrating system: VibraPann.

4. Presentation of the VibraPann system

The VibraPann system (Fig. 4) has been developed, with the active collaboration of an industrial partner Plysorol, especially for on-line applications to production size panels. The first step of this study was to find a test able to favour the vibrating modes (0,2) and (2,0) and to reduce the expression of the other modes in order to place, for every kind of panel, the resonance frequencies of the two vibrating modes at the same place in the frequency spectrum (see Table 2).

An analysis of the modes shapes presented in Figs. 1 and 2, shows that the intersection of the nodal lines of the vibrating modes (0,2) and (2,0) produces four nodal points common to these modes placed at $x = \pm L/4$ and $y = \pm l/4$. So, very simply boundary conditions to implement in a testing system can be obtained using four pin supports positioned on these four nodal points. The vibrating modes (0,2) and (2,0) keep then free when another modes like (1,1), (3,0), (0,3) and (4,0) are attenuated (the four pin supports are not placed on nodal points for these vibrating modes).

The measurement is realized by an accelerometer, placed at the centre of the panel on the inferior face, which records the vibrating signal generated by a pneumatic tool. The impact is realized at the centre of the panel on the upper face. An appropriate spring keeps the accelerometer in position during the test without any perturbation of the panel vibrations. Under such conditions, the frequencies corresponding to the vibrating modes (0,2) and (2,0) are correctly recorded (the centre of the panel is a maximum displacement point for these modes) when the vibrating modes (1,2) and (2,1) are not detected. The two resonance frequencies that appear at the

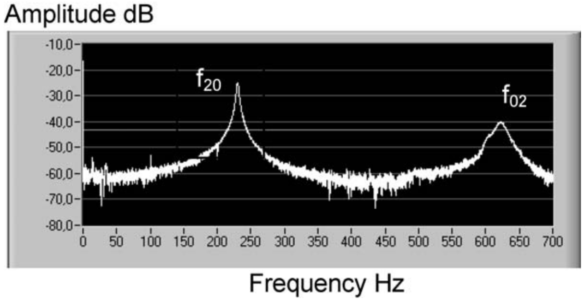

Fig. 5. Typical frequency spectrum obtained with VibraPann for any kind of panel.

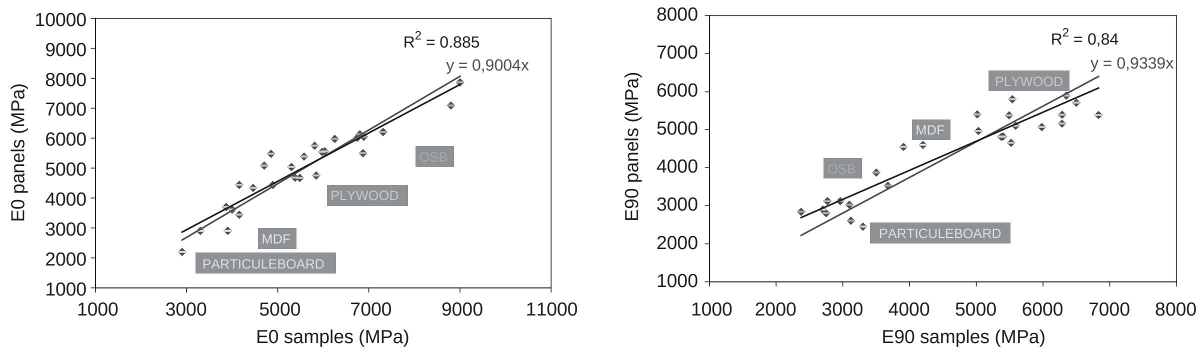

Fig. 6. Correlation between the dynamic modulus of different whole wood based panels and the average static modulus obtained on small samples according to EN 789.

beginning of the frequency spectrum are then always f_{20} and f_{02}, whatever the kind of panel (oriented strand board (OSB), plywood, medium density fibreboard (MDF) or particleboard) and its thickness. A typical spectrum is presented Fig. 5.

5. Experimental results and discussion

Approximately 260 wood-based panels with the same structural dimensions and various thicknesses ($10 \rightarrow 22$ mm) were tested during 1 year: eight OSB panels (dimensions $1250 \times 2500\,\mathrm{mm}^2$), 250 plywood panels (dimensions $1250 \times 2500\,\mathrm{mm}^2$), two particle boards (dimensions $1250 \times 2500\,\mathrm{mm}^2$) and one MDF panel (dimensions $1850 \times 3050\,\mathrm{mm}^2$). Most of them ($>230$) were tested at the production site.

All results presented are compared with the average mechanical characteristics obtained from small normalized specimens according to EN 789 (four points bending test) or EN 310 (three points bending test). A global survey of this experimental work is shown Fig. 6a and b (only a few results obtained for plywood panels are presented in this figure).

The totality of the results obtained for plywood panels are presented Fig. 7a and b. The good correlations obtained between dynamic and static properties ($R^2 > 0.8$) are similar to those

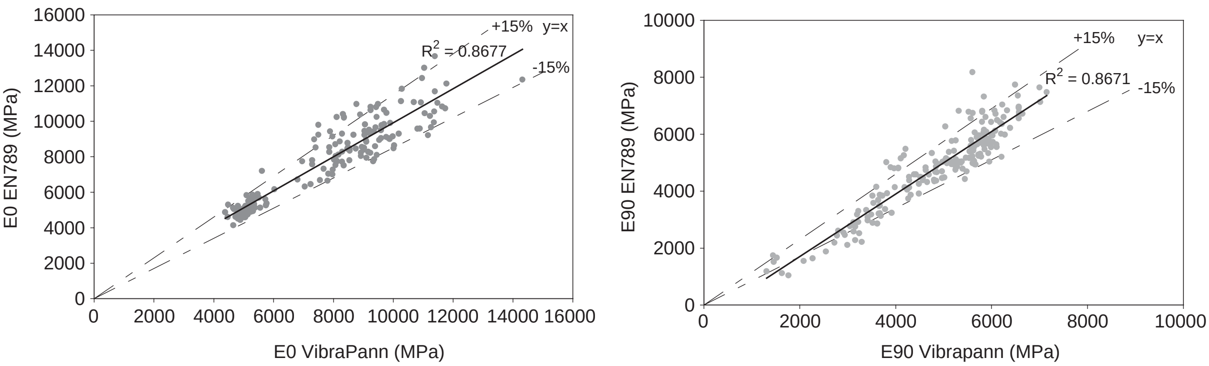

Fig. 7. Correlation between the dynamic modulus plywood panels and the average static modulus obtained on small samples according to EN 789.

presented in the literature. As usual, for wood-based materials, the dynamic moduli are higher than the static values: by approximately 10%. For such materials, often very fragmented and which contain a significant number of glued interfaces, an important part of this difference is generally explained by the different rates at which the stresses are applied in the dynamic and the static tests. The contributions of the viscoelastic behaviour of wood and especially of glue become influential during static tests for which the duration is often higher than 5 min in standard conditions.

Anyhow, these coefficient of correlations remain only qualitative indicators because an attempt is being made to compare very different results: the elastic properties of a plate and some average moduli more or less representative of the spatial variability within the panel and obtained from small samples which have a mechanical behaviour similar to a beam.

One plywood panel has been sampled more rigorously to illustrate the spatial variability within a panel. Four dimensions of specimens have been cut off and tested from this panel: $1200 \times 600\,\mathrm{mm}^2$ (four specimens tested), then $600 \times 300\,\mathrm{mm}^2$ (16 specimens tested), then $300 \times 150\,\mathrm{mm}^2$ (16 specimens tested) and finally $150 \times 75\,\mathrm{mm}^2$ (32 specimens tested). The procedure of sampling is shown Fig. 8.

Each specimen is tested with the VibraPann system (the impact and the accelerometer are adapted to the dimensions of each specimen and the range of the frequencies to identify). The representation of the spatial variability is built using Matlab from the resonance frequencies obtained for each specimen. For each level of sampling the difference in percentage between the resonance frequency of the specimen and the average resonance frequency of the population have been plotted. The calculation of the elastic moduli has not been realized because relations (6a) and (6b) are not valid for small specimens. Fig. 9 shows the variability of $f(0,2)$ but a similar representation is obtained for $f(2,0)$.

Considering the reliability of the VibraPann system, it seems that the initial deformation of the panels, which can be important for thin plywood panels, has low influence on the evaluation of the mechanical properties. In the same way, it is not necessary to place exactly the four pin supports on the four nodal points of the panel: an error of few centimetres is tolerated.

The mechanical strength of plates cannot be directly estimated by vibration analysis. However, the correlations between elastic properties and strength that are generally significant for wood

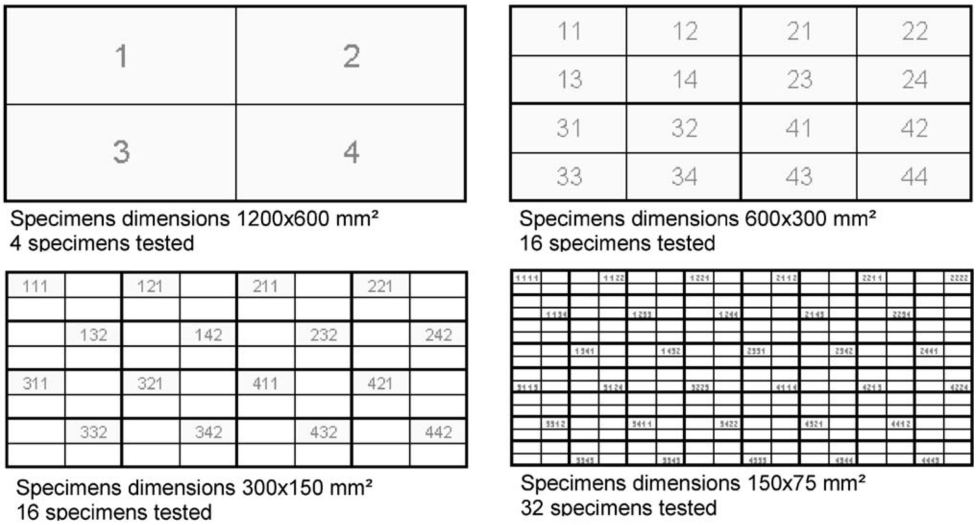

Fig. 8. Procedure of sampling used to illustrate the spatial variability within a panel.

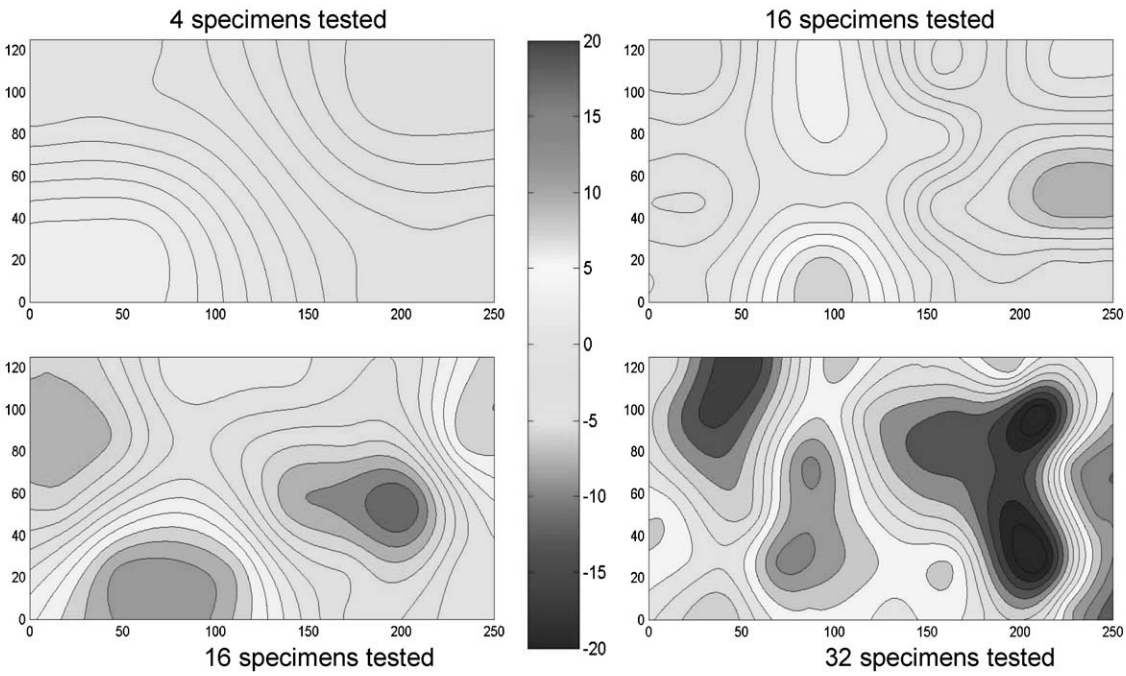

Fig. 9. An example of the spatial variability within a panel with 4 different levels of observation. In each case the scale represents the difference in percentage between the resonance frequency of the specimen and the average resonance frequency of the population.

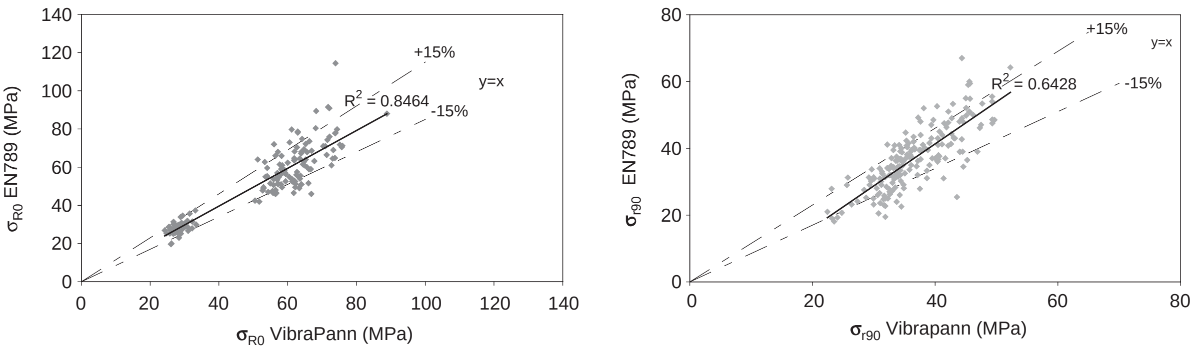

Fig. 10. Predictive model of strength for plywood panels built from vibration tests carried on whole panels. Correlation with static tests (EN 310).

products allows one to build predictive models of strength based on simple or multiple correlations. Of course, these results remain only indicative but give a good estimation of the mechanical strength especially for wood-based panels which are relatively homogeneous at the scale of the panel.

Fig. 10 shows two predictive models of plywood strength built from dynamic moduli (10 panels were used to build each model, the other presented results in Fig. 7 are obtained from predicted values). The predictions of the strength along the long axis of the panels σ_{R0} ($R^2 > 0.8$) seem to be better than the short axis ones σ_{R90} ($R^2 > 0.6$). This result can be explained considering the tightness of the transversal strength distribution.

6. Conclusion

The good correlation, related in the literature, between the static and dynamic properties resulting from vibration analysis are confirmed. However more significant results emerged from this work. Vibration analysis can be really used in an industrial context since the boundary conditions are simple to implement and results interpretation is fast and evident to provide a good feedback on quality indicators. The VibraPann system have been developed especially for on-line control of whole panels with regards to those conditions and most of the results presented (>200 panels) have been carried out on different production sites during a few months. Four mechanical properties are given by the VibraPann system to provide a good feedback on the production quality: the elastic moduli along the main axis are directly measured and the corresponding mechanical strength are estimated from a model based on simple correlations.

Further works will focus on the estimation of the in-plane properties (stiffness and strength) and cohesion indicator.

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