Analysis of Wood Peculiarities by Resonant Vibration Method

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Abstract

Most wood products meet dynamic loads. Evaluation of mechanical properties of wood in dynamic mode allows forecasting wood assortment behavior in dynamic loads zone. The article discusses modulus of elasticity research methodology and equipment for leafy and conifer (oak and spruce) rectangular wood plates (oak - 340×340×14 mm and 400×300×14 mm, spruce - 300×280×30 mm and 300×265×30 mm), glued-up panels (oak - 500×500×30 mm and 700×700×30 mm, spruce - 400×400×14 mm) and scantlings (oak - 670×60×30 mm, spruce - 400×40×14 mm).

Cross-resonant vibrations method was employed for wood assortment analysis. Resonant frequencies and bend forms (modes) were evaluated. It was illustrated that through different along and across mechanical properties of wood fiber vibration of such products to the above-mentioned directions are parallel to vibration of beam-formed body.

Applying theoretical calculations for beam-formed body, oak wood modulus of elasticity was estimated across (E_p= 900 – 2000 MPa) and along (E_l = 8000 – 14000 MPa) fiber. For spruce wood, respectively, (E_p= 140 – 500 MPa) and (E_l = 7500 – 14000 MPa).

Suggested methodology and equipment can be successfully applied to analysis and quality evaluation (assortment, hunt for defects, etc.) of other products, analogical to natural wood in terms of fiber direction.

Key words: wood plate, glued-up panel, scantlings, resonance vibrations, modulus of elasticity

Introduction

Mechanical properties of wood depend on its structure (Wagenführ 2000, 2004). In most cases this determines wood application area. Though not only selection of appropriate wood type is important for production of various products, but cut of material of a particular part of the log as well. Due to different along and across mechanical properties of wood fiber in most cases of defects it is replaced by glued wood. For this reason glued-up panels and layered wood are produced. Products made of such wood are more durable, but less sensitive to temperature and moisture changes.

Another important factor affecting wood products is load. Dynamic loads affect most products next to static loads as well. Such loads often affect building constructions, parts of ships, planes and musical instruments, etc.

Quality of wood assortment is evaluated by testing it statically and dynamically. During dynamic tests assortment is periodically loaded. Therefore evaluation of mechanical properties in such conditions allows forecasting wood assortment behavior in dynamic loads zone (Molin *et al*. 1984, Molin *et al*. 1988). In the course of such tests wood modulus of elasticity and coefficient of damping is usually evaluated. For this purpose free and forced assortment vibrations are employed. In comparison to static tests, the advantage of these tests is that tested assortment is not fragmented and it is not necessary to prepare specimen of a specific form for testing. Thus, dynamic tests can give more precise evaluation of modulus of elasticity and occurring defects of the assortment as well.

Tests of small-diameter logs, floor systems and grained flags by estimating assortment modulus of elasticity in free across vibrations method are well known (Wang *et al.* 2002, Cai *et al.* 2002, Hunt *et al.* 2004). This method is applicable to beam-formed bodies, when assortment properties are expressed to one (in this case along) direction. Free vibration method is not precise, since the frequency of such vibration depends on wood coefficient of damping. Thus, assortment striking excites vibration of higher modes, too. Therefore this method is not applicable to products of other form than beam. In this case vibration shall be excited to both along and across directions of fiber.

For evaluation of polymer mechanical properties coercive resonant vibration method is employed (Malkin 1983). In this case specimen form must also match beam form.

Modulus of elasticity is often determinable by vibration of sound frequencies. The velocity of spread of mechanical vibrations in the material depends on the modulus of elasticity and density of the material (Долацис 1985, Wang *et al.* 2002). Velocity of spread of sound in different directions of grain is different for anisotropic structure of wood. Modulus of elasticity of wood likewise depends on the direction of grain. (Wagenführ 2000).

Wood is orthotropic and layered body. Most of its products (boards, glued-up panels, etc.), subject to their dimensions, may have a form of not only beam, but of another well-known body (*e.g.*, plate). Their mechanical properties in along and across directions are different, just like properties of natural deciduous and coniferous wood. Therefore, such wood products can also be applied resonant vibration test method, considering modulus of elasticity of a product in appropriate fiber direction.

Aim of the work is to evaluate modulus of elasticity along and across fiber directions of natural rectangular plate form leafy and conifer wood and analogous (in terms of fiber direction) glued wood products, applying theoretical beam-formed body methodology.

Materials and methods

As it was mentioned, mechanical properties of different bodies, having the shape of plates and beams, are estimated using resonance vibrations (Timoshenko 1985). Analysis of the vibrations of such bodies indicates, that their material is solid and isotropic.

The frequencies of transverse vibration of unfastened rectangular plate are calculated as follows:

$$f_1 = \pi^2 \sqrt{\frac{D}{\rho h}} \left(\frac{m^2}{l_1^2} + \frac{n^2}{l_2^2} \right) \quad (1)$$

In the case of a square plate:

$$f_1 = \frac{\pi^2}{l^2} \sqrt{\frac{D}{\rho h}} \left(m^2 + n^2 \right) \quad (2)$$

where: $D = \frac{Eh^3}{12(1-\nu^2)}$

E – MOE (Jung's module); ν - Poison's ratio; ρ - density of the material; h – thickness of the plate; l_1, l_2 – length and width of the plate, m, n = 1, 2, 3 ....

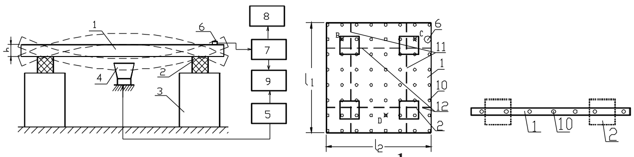
The frequencies of beam type bodies are calculated as follows:

$$f_2 = \frac{A}{2\pi l^2} \sqrt{\frac{EI}{\rho S}} \quad (3)$$

here: E – MOE (Jung's module); I – inertia moment of beam cross-section; S - cross-sectional area; l – length of beam; ρ - density of the material; A – coefficient, characterizing fastening method of beam ends and modes.

Thus, after ascertaining resonance frequency of transverse vibrations and other parameters of plate or beam, it is possible to estimate the dynamic MOE of the material.

A special stand was constructed for the study (Fig.1):



a b c

Figure 1. Stand (a) for the study of wooden articles and allocation scheme of vibration measurement points in the case of a plate (b) and beam (c): 1 – wood assortment and its bend character; 2 – elastic elements; 3 – massive supports; 4 – acoustic vibrator; 5 – generator of electric vibrations; 6 – sensor; 7 – measuring device; 8 – oscilloscope; 9 – phasometer; 10 – fastening points of the sensor; 11, 12 – knotted vibration lines of the first bend form (mode) along and across the fibre, respectively; B, C, D – characteristic vibration measurement points

Studied plate 1 is freely placed on four elastic elements 2. While studying a beam, it is placed on two elastic elements (Fig. 1, c). These elements are made of foam rubber (120×120×100 mm) and fixed on massive supports 3. An acoustic vibrator 4, which is regulated by the generator of electric signals 5, is used to induce resonance vibrations of the studied assortment 1. These vibrations are fixed by sensor 6, attached to studied object 1. In changing the frequency of generator 5, resonance vibrations of the studied object are induced and measured with measuring device 7. The form of vibrations is observed on the screen of oscilloscope 8. To ascertain bend direction of the studied assortment, the phase of vibrations is measured by phasometer 9. For this purpose the phasometer 9 receives signals from the measuring device 7 and the generator 5.

As far as the studied plate (beam) is freely placed on elastic elements 2, this case corresponds to extreme conditions of an unfastened plate (Timoshenko 1985).

Vibrating by each resonance frequency, the plate or beam bends in a certain mode. It is determined by recording the amplitude and phase of vibrations at all measurement points 10.

Results and discussion

Experiments were carried out with wooden plates sawn from oak and spruce boards and with one – layer glued-up panels.

Plates were sawn from randomly selected on both sides planed oak and spruce boards. The direction of fibre in them is parallel to the edge of the plate (Fig.2).

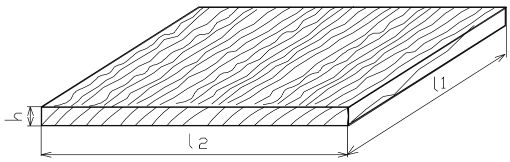


Figure 2. Scheme of plates: here l_1 – length of the plate, l_2 – width, h – thickness

Two types of oak solid wood plates were sawn – square (340×340×14 mm, 3 units) and rectangular (400×300×14 mm, 3 units). Spruce solid wood plates were 300×280×30 mm, 5 units and 300×265×30 mm, 5 units. Oak plates were conditionally marked $P_{1.1} – P_{1.3}$ and $P_{2.1} – P_{2.3}$. The density of the studied oak plates was 685 – 750 kg/m^3, the moisture content – 11 – 13 %, spruce plates – respectively, density was 380 – 500 kg/m^3, the moisture content – 10 – 12 %. Studies were carried out in laboratory conditions (t = 17 – 20°C, relative humidity – 65 – 70 %). Vibrations were measured within 20 – 2000 Hz range. To increase measuring sensitivity, the amplitude of vibration acceleration was recorded.

Initial measurement data were obtained recording vibrations in characteristic (freely chosen) points B, C, D (Fig.1, b) of the plate. In this way, amplitude-frequency characteristics of plates were obtained and resonance frequencies were ascertained.

Vibrating by each resonance frequency, the plate bends in a certain mode. Especially characteristic are the first bend modes of plates across and along the fibre. They are characterized by two knotted vibration lines (in these lines vibration amplitude equals zero (Fig.1, b). Higher and more complicated modes of plates were recorded as well. It was found, that in the case of oak square plates the number of resonance frequencies within measurement range fluctuates from 11 to 14, while the values of frequencies in vibration measurement points differ by about 1 – 4 %. Resonance frequencies of oak square plates are presented in Table 1.

The first modes of these plates across and along the fibre are shown in Fig.3. a, b. Positive and negative values of vibration amplitude A characterize bend direction of certain plate zones during one vibration half-period ("+" corresponds bend in one direction, "- " - in the opposite direction). During the other vibration half-period signs of the amplitude are changed *vice versa* (Fig.1 a).

Table 1. Resonance frequencies of oak square plates

Note of plate	Resonance frequency, Hz						
P1.1	170	218	396	512	587	661	801
	898	1135	1278	1382	1488	1716	1850
P1.2	149	181	355	437	517	575	675
	744	834	1030	1250	1345	1590	1805
P1.3	181	381	493	737	854	1054	1165
	1249	1305	1689	1868			

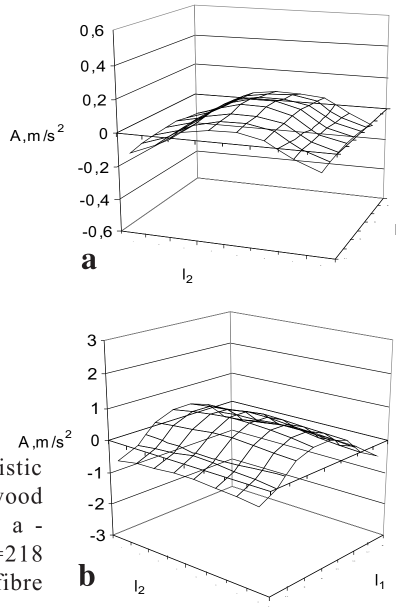


Figure 3. Characteristic modes of oak solid wood square plate (P1.1): a - across the fibre (f=218 Hz); b – along the fibre (f=512 Hz)

While studying square plates, different character of their bend was recorded. Under resonance frequency f=218 Hz (Fig. 3 a) panel ($P_{1.1}$) apparently bends across the fibre. However, at the same time was registered an insignificant (with by about 10 times less amplitude) plate bend in a perpendicular direction, *i.e.* along the fibre. Analogous situation was obtained also under resonance frequency f=512 Hz (Fig. 3, b). In this case plate visibly bends along the fibre and insignificantly – across the fibre.

The mode of the studied plate in each direction of the fibre is analogous to the mode of a beam shape body. Thus, MOE of a plate across (f=218Hz) and along the fibre (f=512 Hz) was calculated applying the theory of a beam shape body (equation 3).

Vibrating in other frequencies, modes of oak plate are much more complicated. Figure 4 presents plate bend forms vibrating at 396 and 661 Hz.

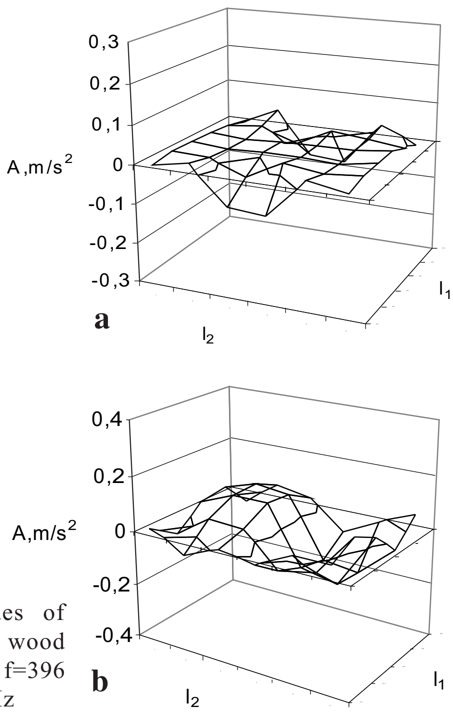


Figure 4. Modes of square solid wood plates (P1.1): a – f=396 Hz, b – f=661 Hz

Analogous results were obtained also studying plates P1.2 and P1.3, as well as rectangular plates P2.1 – P2.3. The values of characteristic plate resonance frequencies and MOE are presented in Table 2.

Table 2. Characteristic resonance frequencies and MOE of oak solid wood plates

Note of plate	Along the fibre		Across the fibre	
	Resonance frequency,Hz	MOE, MPa	Resonance frequency, Hz	MOE, MPa
P1.1	512	10974.7	218	1989.6
P1.2	517	12033.6	181	1474.9
P1.3	493	10804.8	181	1456.4
P2.1	318	8639.8	232	1499.5
P2.2	308(with defect)	7871.5	222	1320.1
P2.3	353	10558.1	242	1601.8

As the data in table 2 show, MOE of plates along the fibre fluctuates within 8640 – 12034 MPa range, while across the fibre within 1320 – 1990 MPa range. It is known, that oak wood MOE along the fibre is 8000 – 16000 MPa, while across the fibre – about 1000 - 2000 MPa (Wagenführ, R., 2000).

Analogically were investigated and spruce panels. Were obtained MOE of plates along the fibre fluctuates within 7500 – 10400 MPa range, while across the fibre within 150 – 350 MPa range. It is known, that spruce wood MOE along the fibre is 6600 – 17200 MPa, while across the fibre – about 450 MPa (Wagenführ 2000).

Investigation data of oak and spruce panels were processed statistically. Statistical rate of modulus of elasticity are presented in Table 3 and Figure 5.

Table 3. Statistical rates of modulus of elasticity of wood panels

Statistical rate	Modulus of elasticity, MPa			
	Oak, along the grain	Oak, across the grain	Spruce, along the grain	Spruce, across the grain
Average value	10147.1	1557.1	8580.8	189.9
Dispersion	2459128.1	53087.7	1049111.1	3086.4
Average square deflection	1568.2	230.4	1024.3	55.6
Variation coefficient,%	15.5	14.8	11.9	29.3

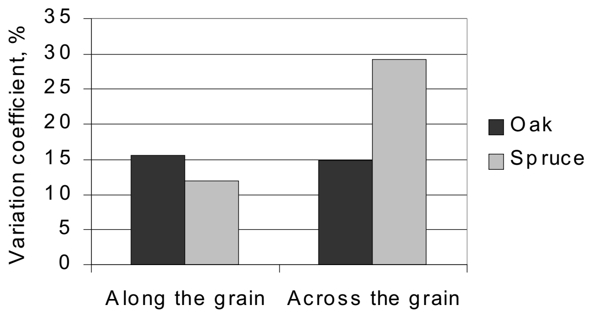


Figure 5. Distribution of variation coefficient of modulus of elasticity of panels in oak and spruce log

The obtained values of variation coefficient of modulus of elasticity shown, that wood from different places of log and from different logs characterizes separate elastic properties.

It can be seen that dispersion of modulus of elasticity of oak panels along and across grain is close (variation coefficient is 15.5 and 14.8 %, respectively). Dispersion of values of modulus of elasticity of spruce is slightly less, than that of oak (variation coefficient 11.9 %), but across grain – about 2 times more.

It is known, that variation coefficient of oak assortments, falling in one strength class, can be by 20 %, spruce assortments – by 15 % [EN 338].

Analogous studies were carried out with one layer glued-up oak and spruce panels. The panels of scantlings were glued up from randomly selected scantlings, joining their edges. The scantlings were sawn from dried oak and spruce wood and planed on all four sides. Some panels were glued from oak scantlings sized 550×74×32 mm, others – 750×74×32 mm. The panels were glued produced on PVA basis and corresponding D3 class adhesives, trimmed according to required dimensions and polished on both sides. The dimensions of the obtained panels were the following:

S1.1 and S1.2 – 500×500×30 mm, S2.1 and S2.2 – 700×700×30 mm (Fig.6). The density of panels fluctuates within 680 - 770 kg/m^3, while the moisture content within 10 – 13 % range. The dimensions of the spruce scantlings were 4004014 mm. The density of scantlings fluctuates within 420 - 500 kg/m^3, while the moisture content within 8 – 11 % range.

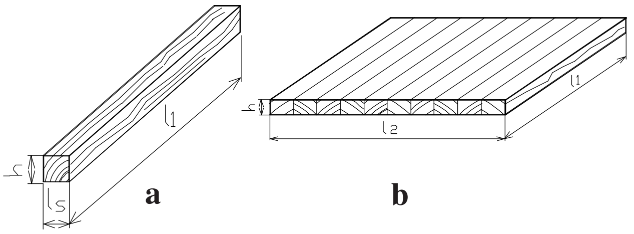


Figure 6. Scheme of scantling (a) and glued-up panel (b): l_1 – length of the scantling and panel, l_S,l_2 – width, h – thickness

The glued-up panel, similarly to a solid wood plate, during its vibration bends analogically in different modes. Many of these modes were complicated. However in all cases the characteristic modes along and across the fibre were recorded. Characteristic modes of one - layer oak glued-up panel (S1.1) across and along the fibre are shown in Figure 7.

It was found that bending of these glued-up panels across and along the fibre is also more characteristic of the bending of a beam type body. Although at the same time a very insignificant panel bend in a perpendicular direction was recorded. Therefore, MOE of glued-up panels as well as that of solid plates may be estimated depending on its mode by applying theoretical beam shape body calculations (equation 3). Results of the measurements and calculations are presented in Table 4.

Table 4. Characteristic resonance frequencies and MOE of glued-up panels

Note of panel	Along the fibre		Across the fibre	
	Resonance frequency, Hz	MOE, MPa	Resonance frequency, Hz	MOE, MPa
S1.1	524	12990	184	1602.3
S1.2	521	12091.5	173	1333.2
S2.1	265	12444.5	93	1532.7
S2.2	277	12081	77	933

As seen from the data presented in table 4, the values of MOE of oak glued–up panels along the fibre was 12080 – 13000 MPa, across the fibre – 930 – 1610 MPa.

Analogous results were received for spruce wood. Spruce beam panels, vibrating in the abovementioned frequency range, also bowed in various modes (difficult, in most cases). However, in cases of all panels typical along and across fiber modes were registered. After evaluating vibration frequencies of such modes calculated panel modulus of elasticity along and across fiber matched established modulus of elasticity values for spruce wood in respective fiber directions.

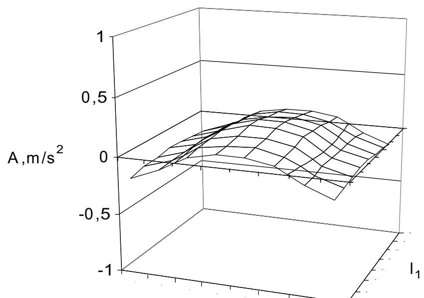
Values of modulus of elasticity of glued-up panels were processed statistically. The data are presented in Table 5.

Table 5. Statistical rates of modulus of elasticity of glued-up panels

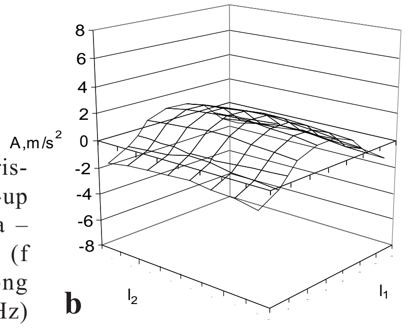
Statistical rate	Modulus of elasticity, MPa			
	Oak, along the grain	Oak, across the grain	Spruce, along the grain	Spruce, across the grain
Average value	12000.4	1238.2	12157.0	307.1
Dispersion	507807.4	92768.6	1540794.0	1702.9
Average square deflection	712.6	304.6	1241.3	41.3
Variation coefficient,%	5.9	24.6	10.2	13.4

It can be seen (Table 5) that estimated variation coefficient of modulus of elasticity is in many cases not insignificant.

It was found that variation coefficients of oak and spruce glued-up panels are major across grain (respectively 24.6 % and 13.4 %). Least dispersion of elastic properties is estimated in oak wood along grain. In this case variation coefficient of modulus of elasticity is 5.9 %. Variation coefficient of spruce modulus of elasticity in this grain direction is almost 2 times more.



a



b

Figure 7. Characteristic modes of glued-up oak panel (S1.1): a – across the fibre (f =184 Hz); b – along the fibre (f = 512 Hz)

Example

Analysing wood glued-up panel as a beam type body it was estimated, what influence panel width has on the value of MOE. In proof of it the mentioned method of resonance vibrations was applied to study individual scantlings and their MOE was ascertained. The scantlings were sawn from randomly chosen oak and spruce boards. Fibre direction in the scantlings is parallel to the edge, while annual rings are randomly distributed. The size of oak scantlings was 670×60×30 mm, density – 680 – 780 kg/m^3, while the moisture content varied within 10.5 – 13 % range, the size of spruce scantlings was 400×40×14 mm, density – 420 – 500 kg/m^3, while the moisture content varied within 8 – 11 % range. Fastening the sensor in different places along the length of a scantling (Fig.1, c), the mode of scantlings was estimated. Owing to their dimensions and fibre direction, as compared to the studied wooden plates and glued-up panels, scantlings are closer to isotropic beam shape bodies. It has been found, that the modes of scantlings are analogous to the modes of beam shape bodies. After ascertaining the frequency of the first mode of scantlings along the fibre, in accordance with the (3) expression, the values of scantling MOE along the fibre were estimated. It was found, that it fluctuates within 9100 – 13400 MPa range (oak scantlings) and 8600 – 14000 MPa (spruce scantlings). It can be seen, that in this case it also corresponds to the MOE values of statically bent oak and spruce wood. Later scantlings were glued among themselves joining their edges under laboratory conditions. Initially two scantlings were glued together. Having determined the frequency of the first mode of such a combination, we calculated MOE according to (3) expression. Later the third scantling was added and MOE of this combination was determined. This continued until a square panel was obtained. At the same time, mean MOE of the used scantlings was calculated in each case. For the study, three types of panels were prepared. They were glued from scantlings with close MOE values. MOE of oak scantlings in these panels was the following: in panel T_1a – 12016 – 10823MPa, in T_1b panel –10906 – 10377 MPa, in T_1c panel - 10186 - 9310MPa. The scantlings were glued using adhesives based on PVA pitch, corresponding to D2 class. The change of approximated MOE curves of a glued-up panel T_1a and average MOE of scantlings, depending on the number of scantlings (panel width), are shown in Figure 8. In this case each point of curve 1 represents an average MOE value of the glued scantlings, while respective points of curve 2 – MOE value of the obtained glued-up panel. The change of glued-up panel MOE and average MOE of the used oak and spruce scantlings, depending on the number of scantlings, is presented in Figure 8.

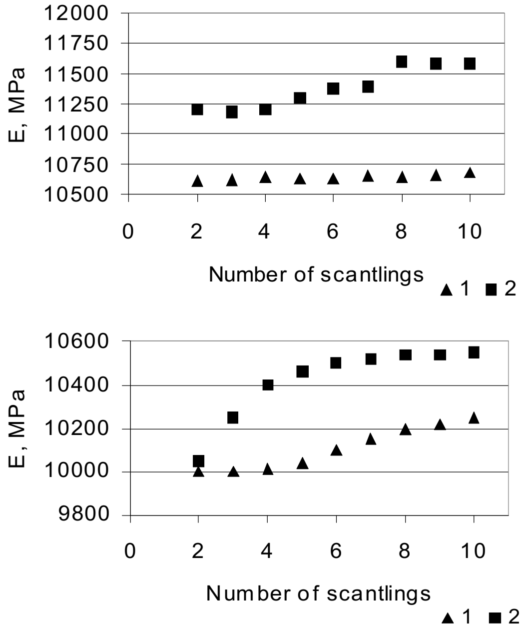


Figure 8. The change of oak (a) and spruce (b) glued-up panels MOE and average MOE of the used scantlings, depending on the number of scantlings in it (panel width): 1 – average MOE of glued scantlings; 2 – panel MOE

As seen from Figure 8, independently of the number of scantlings in panels, the law of panel MOE change corresponds to the law of the average MOE change in the used scantlings. Analogous results were obtained also for other glued-up panels. It was found, that in all cases MOE of a glued panel is slightly higher (3 – 10 %) than the average MOE of comprising it scantlings.

The presented study results show, that the method of resonance vibrations may be applied to estimate MOE of wood products of the mentioned rectangular plate shape. Due to different mechanical properties along and across the fibre, vibrations of such products in these directions are analogous to those of a beam shape body. Therefore, calculation method of a beam shape body may be applied to estimate MOE. This method may be applied also to other products, analogous to solid wood by fibre direction.

The provided method and equipment may be successfully used for the investigation of such assortments and for the estimation of their quality – sorting, detection of defects, *etc.*

Conclusions

According to the results of studies, the following conclusions can be drawn:

1. It has been proven, that studying plate shape leaf and coniferous trees assortments across and along

the fibre, the method of resonance vibrations may be applied.

2. It was ascertained, that due to different wood mechanical properties along and across the fibre for wooden boards, glued-up panels, scantlings and other analogous by the direction of fibre articles, theoretical MOE calculations of isotropic beam type bodies may be applied.

3. The obtained MOE values of wooden articles along the fibre (oak - E_l= 8000 – 14000 MPa, spruce - E_l= 8000 – 14000 MPa) and across it (oak - E_p= 900 – 2000 MPa, spruce E_p= 150 – 500 MPa) correspond to MOE values of statically bent wooden articles.

4. It is shown, that width (number of scantlings) change of glued-up panels practically has no influence on the value of MOE – under maximal change of panel width, MOE on average changes within the range of 5 %, i. e. the oscillations of panel are analogically like a beam shape body.

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АНАЛИЗ ОСОБЕННОСТЕЙ ДРЕВЕСИНЫ МЕТОДОМ РЕЗОНАНСНЫХ КОЛЕБАНИЙ

Й. Воболис и Д. Албректас

Резюме

Многие деревянные изделия испытывают динамические нагрузки. Испытания механических свойств древесины в динамическом режиме дают возможность прогнозировать поведение изделий из древесины в условиях динамических нагрузок. В статъе представлена методика и оборудование для исследования модуля упругости древесины лиственных и хвойных пород (дуба и ели). Исследованы различные сортименты – прямоугольные пластины (дуба - 340Ч340Ч14 мм и 400Ч300Ч14 мм, ели - 300Ч280Ч30 мм и 300Ч265Ч30 мм), столярные плиты (дуба - 500Ч500Ч30 мм и 700Ч700Ч30 мм, ели - 400Ч400Ч14 мм) и бруски (дуба - 670Ч60Ч30 мм, ели - 400Ч40Ч14 мм).

Для исследования деревянных изделий использован метод поперечных резонансных колебаний. Установлены резонансные частоты и формы изгиба (моды) изделий. Показано, что, благодаря различным механическим свойствам древесины вдоль и поперек волокон, колебания образцов по этим направлениям волокон аналогичны колебаниям объекта, имеющего форму стержня.

Используя теоретические расчеты колебаний стержня, рассчитан модуль упругости древесины обеих пород. Установлено, что модуль упругости сортиментов из дуба вдоль волокна составляет $E_в$ = 8000 – 14000 МПа, поперек – $E_п$= 900 – 2000 МПа, а ели $E_в$ = 7500 – 14000 МПа и $E_п$= 140 – 500 МПа, соответственно.

Указанная методика и оборудование могут быть успешно использованы для исследования и других изделий, аналогичных натуральной древесине по направлению волокон.

Ключевые слова: деревянная пластина, столярная плита, брусок, резонансные колебания, модуль упругости