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EFFECTS OF ANYSOTROPY OF WOOD ON MODAL ANALYSIS OF VIOLIN PLATES

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Abstract: *Numerous studies concerning the acoustical behavior of violins refer to the similarities and differences in the tonal qualities produced by old and new instruments, depending on various factors. Among these factors the acoustical characteristics of wood species are criteria for the selection of the most appropriated raw material for violin making. The main wood species used in violin construction are spruce (Picea abies) and maple (Acer pseudoplatanus). In this paper, the violin plates were modeled with finite elements analysis based on three hypotheses – the material is isotropic; the material is transverse isotropic; the material is orthotropic, was used for modal analysis of violin plates. The plates were considered flat and having uniform thickness of 3 mm. The patterns of vibration modes are affected by wood species and structure. Spruce plate shows nodal lines aligned to L anisotropic direction of wood, corresponding with the direction of the fibres. Maple plates show nodal lines aligned to R anisotropic direction of wood and to the direction of medullary rays in the LR plane.*
Keywords: *violin plate; resonance wood, simulation, anisotropy.*

1. INTRODUCTION

It is generally accepted that spruce is used for the top plate of the violin and curly maple for the back of the violin plate [1 – 4]. Spruce (Picea abies) for the top plate, called also spruce resonance wood or tone wood, and curly maple (Acer platanooides) of typical anatomical structure, for the back plate, have been traditionally used for violins since the Baroque era. Acoustical and mechanical parameters of these species have been widely presented in reference books [2, 3] and articles [5 – 8]. Violin plates should be quarter sawn, in the LR anisotropic plane of wood. However, some famous violins made for example by Guarneri, have the back plate in curly maple, flat sawn - in the LT anisotropic plane of wood [9 – 10]. [11 – 12] investigated the vibration modes of freely supported and edge-constrained top and back plates according to shape, arching, thickness graduation, the effect of coupling to the ribs, etc. The most recent state of the art reference on violin acoustics was published in 2014 by [13]. In these articles no reference is made to the amount of the violin plate's vibration surface. The aim of this article is to use FEA to study the effect of wood species on vibration modes of violin plates. Furthermore, FEA supports a simplified model for the computation of vibrating areas of plates.

2. MATHEMATICAL FORMULATION

The elastic properties of solids can be defined by generalized Hook's law relating the volume average of stress $[\sigma_{ij}]$ to volume average of strains $[\varepsilon_{ij}]$ by the elastic constants $[C_{ij}]$ in the form [9, 10, 14]:

$$[\sigma_{ij}] = [C_{ijkl}] * [\varepsilon_{kl}] \quad (1)$$

Or

$$[\varepsilon_{kl}] = [S_{ijkl}] * [\sigma_{ij}] \quad (2)$$

Where $[C_{ijkl}]$ is tensor of elastic stiffness; $[S_{ijkl}]$ – tensor of elastic compliances and i, j, k or l correspond to 1, 2, 3 or 4 directions. For solids of different symmetries such as isotropic, transverse isotropic or orthotropic, the stiffness matrix can be turned into a compliance matrix.

The simplest elastic symmetry is that of an isotropic solid with only two independent constants, λ and μ which are known as Lamé coefficients.

$$\mu = \frac{E}{2(1+\nu)} \tag{3}$$

$$\lambda = \frac{E*\nu}{(1+\nu)(1-2\nu)} \tag{4}$$

$$K = \frac{E}{3(1-2\nu)} = \lambda + \frac{2}{3}\mu \tag{5}$$

Where E is Young's modulus, μ is the shear modulus, ν – the Poisson's ratio and K is the bulk modulus.
For isotropic solid, the terms of stiffness matrix are:

$$C_{11} = C_{22} = C_{33} = \lambda + 2\mu \tag{6}$$

$$C_{12} = C_{23} = C_{13} = \lambda \tag{7}$$

$$C_{44} = C_{55} = C_{66} = \mu \tag{8}$$

The anisotropy depends on the internal structure of material and perceived as the variation in material response with direction of the applied stress. The complex elastic symmetry of an orthotropic solid lies when constants are influenced by three mutually perpendicular planes of elastic symmetry. The corresponding stiffness matrix contains nine independent constants: six diagonal terms (C_{11}, C_{22}, C_{33}, C_{44}, C_{55},C_{66}) and three off diagonal terms (C_{12}, C_{23}, C_{13}). For transverse isotropy, the material may possess an axis of symmetry. In this case, the corresponding stiffness matrix contains five independent constants: three diagonal terms (C_{11}, C_{22}, C_{55}) and two off diagonal terms (C_{12}, C_{13}). It can be shown that the transverse isotropy is a particular case of an orthotropic solid. For monoclinic material, 21 independent terms, and $C_{ij} = C_{ji}$:

$$\begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} & C_{15} & C_{16} \\ C_{21} & C_{22} & C_{23} & C_{24} & C_{25} & C_{26} \\ C_{31} & C_{32} & C_{33} & C_{34} & C_{35} & C_{36} \\ C_{41} & C_{42} & C_{43} & C_{44} & C_{45} & C_{46} \\ C_{51} & C_{52} & C_{53} & C_{54} & C_{55} & C_{56} \\ C_{61} & C_{62} & C_{63} & C_{64} & C_{65} & C_{66} \end{bmatrix} \tag{9}$$

Orthotropic material: three symmetry axes, three symmetry planes and nine independent terms of the stiffness matrix:

$$\begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ C_{21} & C_{22} & C_{23} & 0 & 0 & 0 \\ C_{31} & C_{32} & C_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{66} \end{bmatrix} \tag{10}$$

Transverse isotropic material:

$$\begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ C_{12} & C_{22} & C_{23} & 0 & 0 & 0 \\ C_{13} & C_{23} & C_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{44} & 0 \\ 0 & 0 & 0 & 0 & 0 & (C_{11} - C_{12})/2 \end{bmatrix} \tag{11}$$

Isotropic material: two independent constants:

$$\begin{bmatrix} C_{11} & C_{12} & C_{12} & 0 & 0 & 0 \\ C_{12} & C_{11} & C_{12} & 0 & 0 & 0 \\ C_{12} & C_{12} & C_{11} & 0 & 0 & 0 \\ 0 & 0 & 0 & (C_{11} - C_{12})/2 & 0 & 0 \\ 0 & 0 & 0 & 0 & (C_{11} - C_{12})/2 & 0 \\ 0 & 0 & 0 & 0 & 0 & (C_{11} - C_{12})/2 \end{bmatrix} \tag{12}$$

In conclusions, the number of constants for various types of anisotropic materials is 21 for monoclinic materials, 13 for triclinic materials, 9 for orthotropic materials, 5 for hexagonal or transversely isotropic materials and 2 for isotropic materials.
The terms of the compliance matrix are given by relation:

$$\begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ S_{21} & S_{22} & S_{23} & 0 & 0 & 0 \\ S_{31} & S_{32} & S_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & S_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & S_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & S_{66} \end{bmatrix} \tag{13}$$

Where: S_{11}, S_{22}, S_{33} are relate an extensional stress to an extensional strain, both in the same direction. For solid wood, this relation gives the Young's moduli E_L, E_R, E_T. (L – longitudinal direction along fibres, R radial direction, T - tangential direction) (Figure 1); S_{12}, S_{13}, S_{23} – relate an extensional strain to a perpendicular extensional stress. In this way, the six Poisson's ratios can be calculated; S_{44}, S_{55}, S_{66} – relate a shear strain to a shear stress in the same plane and are the inverse of the terms C_{44}, C_{55}, C_{66}, corresponding to planes 23, 13 and 12. The relationship between the stiffness terms and the compliance terms for orthotropic solid are:

$$C_{11} = \frac{S_{22} * S_{33} - (S_{23})^2}{S} \tag{14a}$$

$$C_{22} = \frac{S_{11} * S_{33} - (S_{13})^2}{S} \tag{14b}$$

$$C_{33} = \frac{S_{22} * S_{11} - (S_{12})^2}{S} \tag{14c}$$

$$C_{12} = \frac{S_{21} * S_{33} - S_{23} * S_{31}}{S} \tag{14d}$$

$$C_{13} = \frac{S_{31} * S_{22} - S_{21} * S_{32}}{S} \tag{14e}$$

$$C_{23} = \frac{S_{31} * S_{12} - S_{11} * S_{32}}{S} \tag{14f}$$

$$C_{44} = \frac{1}{S_{44}} \tag{14g}$$

$$C_{55} = \frac{1}{S_{55}} \tag{14h}$$

$$C_{66} = \frac{1}{S_{66}} \tag{14i}$$

Where:

$$S = S_{11} * S_{22} * S_{33} + 2S_{12} * S_{23} * S_{31} - S_{11} * (S_{23})^2 - S_{22} * (S_{13})^2 - S_{33} * (S_{12})^2 \tag{15}$$

In case of wood, when the axes are labelled L, R and T for wood species, engineering constants are related to the compliances in the following form:

$$\begin{bmatrix} \frac{1}{E_L} & -\frac{\nu_{LR}}{E_R} & -\frac{\nu_{LT}}{E_T} & 0 & 0 & 0 \\ -\frac{\nu_{RL}}{E_L} & \frac{1}{E_R} & -\frac{\nu_{RT}}{E_T} & 0 & 0 & 0 \\ -\frac{\nu_{TL}}{E_L} & -\frac{\nu_{TR}}{E_R} & \frac{1}{E_T} & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{G_{23}} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{G_{13}} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1}{G_{12}} \end{bmatrix} = [S] \tag{16}$$

In case of wood, the stress and strain states of wood pieces are represented by stresses tensor and strains tensor, respectively [9, 15, 16]:

$$T_\sigma = \begin{pmatrix} \sigma_L & \tau_{RL} & \tau_{TL} \\ \tau_{RL} & \sigma_R & \tau_{RT} \\ \tau_{TL} & \tau_{TR} & \sigma_T \end{pmatrix}; \quad T_\varepsilon = \begin{pmatrix} \varepsilon_L & \frac{1}{2}\gamma_{LR} & \frac{1}{2}\gamma_{LT} \\ \frac{1}{2}\gamma_{LR} & \varepsilon_R & \frac{1}{2}\gamma_{RT} \\ \frac{1}{2}\gamma_{TL} & \frac{1}{2}\gamma_{TR} & \varepsilon_T \end{pmatrix} \tag{17}$$

where σ_L, σ_R și σ_T are normal stresses on longitudinal (L), radial (R) and tangential (T) direction; τ_{LR}, τ_{RT} și τ_{LT} – tangential stresses in planes LR, RT și LT; ε_L, ε_R și ε_T – strains; and γ_{LR}, γ_{RT} și γ_{LT} – shearing strain. Using the tensor of elasticity modulus E and the tensor of Poisson coefficients, results:

$$\begin{cases} \varepsilon_L = \frac{1}{E_L}(\sigma_L - \nu_{LR}\sigma_R - \nu_{LT}\sigma_T);\ \gamma_{TR} = \frac{\tau_{TR}}{G_{TR}}; \\ \varepsilon_L = \frac{1}{E_R}(\sigma_R - \nu_{RL}\sigma_L - \nu_{RT}\sigma_T);\ \gamma_{RL} = \frac{\tau_{RL}}{G_{RL}}; \\ \varepsilon_L = \frac{1}{E_T}(\sigma_T - \nu_{TL}\sigma_L - \nu_{TR}\sigma_R);\ \gamma_{LT} = \frac{\tau_{LT}}{G_{TL}}; \end{cases} \tag{18}$$

where ν_{LR}, ν_{LT}, ν_{RL}... are coefficients of transverse contraction (first index reprezents the direction of transverse contraction and the second, the direction of the stress which produces the elongation). From energetic reason, between the coefficients of transverse contraction ν and elasticity modulus, the relations (3) are:

$$E_L\nu_{RL} = E_R\nu_{LR};\ \ E_L\nu_{LR} = E_T\nu_{TL};\ \ E_R\nu_{TR} = E_T\nu_{RT} \tag{19}$$

Substituting these parameters, we get the equations of stresses :

$$\begin{cases} \sigma_L = \frac{(1-\nu_{RT}\nu_{TR})E_L\varepsilon_L + (\nu_{LR}+\nu_{LT}\nu_{TR})E_R\varepsilon_R + (\nu_{LT}+\nu_{LR}\nu_{RT})E_T\varepsilon_T}{1-\nu_{RT}\nu_{TR}-\nu_{LR}(\nu_{RL}+\nu_{RT}\nu_{TL})-\nu_{LT}(\nu_{TL}+\nu_{RL}\nu_{TR})}; \\ \sigma_R = \frac{(\nu_{LR}+\nu_{LT}\nu_{TR})E_L\varepsilon_L + (1-\nu_{LT}\nu_{TL})E_R\varepsilon_R + (\nu_{RT}+\nu_{LT}\nu_{RL})E_T\varepsilon_T}{1-\nu_{RT}\nu_{TR}-\nu_{LR}(\nu_{RL}+\nu_{RT}\nu_{TL})-\nu_{LT}(\nu_{TL}+\nu_{RL}\nu_{TR})}; \\ \sigma_T = \frac{(\nu_{TL}+\nu_{RL}\nu_{TR})E_L\varepsilon_L + (\nu_{TR}+\nu_{LR}\nu_{TL})E_R\varepsilon_R + (1-\nu_{RL}\nu_{LR})E_T\varepsilon_T}{1-\nu_{RT}\nu_{TR}-\nu_{LR}(\nu_{RL}+\nu_{RT}\nu_{TL})-\nu_{LT}(\nu_{TL}+\nu_{RL}\nu_{TR})}. \\ \tau_{TR} = G_{TR}\gamma_{TR};\ \tau_{RL} = G_{RL}\gamma_{RL};\ \tau_{LT} = G_{LT}\gamma_{LT}. \end{cases} \tag{20}$$

In accordance with theory of elasticity presented above, three hypotheses – the material is isotropic; the material is transverse isotropic; the material is orthotropic where used to study the effect of wood anisotropy on violin plates made of spruce and maple, using finite elements analysis.

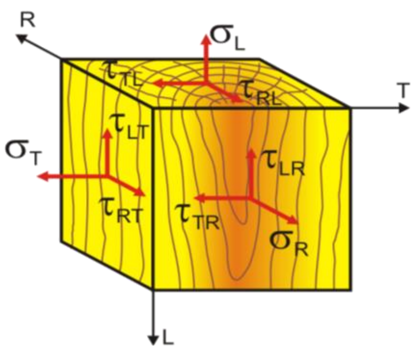


Figure 1. The main planes and axis of elastic symmetry of wood

3. MATERIALS AND METHODS

In accordance with theory of elasticity presented above, three hypotheses – the material is isotropic; the material is transverse isotropic; the material is orthotropic where used to study the effect of wood anisotropy on violin plates made of spruce and maple, using finite elements analysis. Wood mechanical characteristics and typical plate masses chosen to simulate the modes of vibration of plates are given in Table 1. The plates have no f – holes and are flat with a constant thickness of 3 mm. The finite element analysis of violin plates was performed by using ABAQUS software. The violin plate was meshed with a quadratic shell element (9495 finite elements), with eight nodes located in corners (total 28664 nodes). Such an element is based on the theory of small and medium thick plates and can be used in the analysis of large deformations. The plates were edge – constrained as shown in Figure 1a and meshed violin plate (Figure 2b).

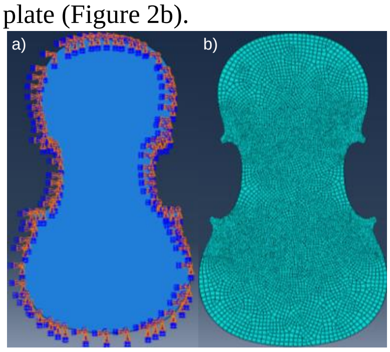


Figure 2. Violin plate edge – constrained, in pre-processing step. Legend: a) boundary condition; b) meshed structure.

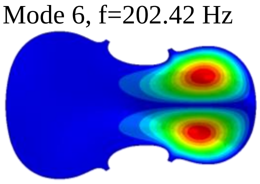
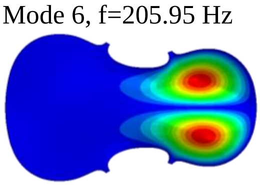
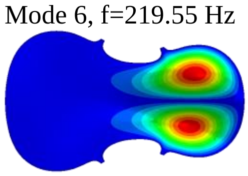
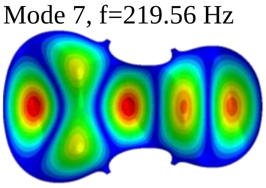
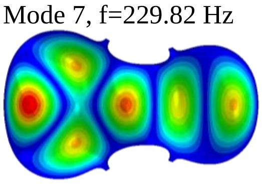
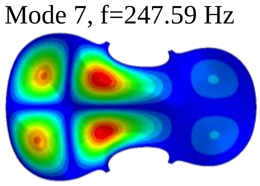
Table 1. Mechanical characteristics of wood (data from *[19] and ** [20]) Note: * In the hypothesis of transverse isotropic solids, we have ER = ET; GLR = GLT, and three Poisson ratios.

Elastic parameters	*Units*	*First hypothesis Isotropic solid*		*Second hypothesis* Transverse isotropic **		*Third hypothesis** Orthotropic*	
		Spruce	Maple	Spruce	Maple	Spruce	Maple
Density	kg/m^3	400	600	400	700	430	590
Young's moduli	MPa	15 000	10 000				
E_L	MPa			13000	10 000	13500	10 000
E_R	MPa			700	2 000	890	1 570
E_T	MPa			700*	2 000*	480	870
Shear moduli	MPa	840	700				
G_{RT}	MPa			60	720	32	290
G_{LT}	MPa			900	1600	500	1100
G_{LR}	MPa			900*	1600*	720	1222
Poisson ratios		0.37	0.37				
ν_{LR}				0.37	0.47	0.45	0.46
ν_{RL}						0.03	0.093
ν_{LT}				0.42	0.50	0.54	0.50
ν_{TL}						0.019	0.038
ν_{RT}				0.47	0.50	0.56	0.82
ν_{TR}						0.30	0.40

3. RESULTS AND DISCUSSIONS

The effect of wood anisotropy on vibration modes of violin plates made of spruce and maple, studied with FEA refers to the vibration of violin plates in three characteristic situations, namely the material has isotropic symmetry and has three elastic constants, the material has transverse isotropic symmetry and has five elastic constants and the material has orthotropic symmetry and has nine elastic constants.
As regards the orthotropic anisotropy the following ratios should be taken into consideration: ratio of Young's moduli EL /ER; EL/ET, ratio of shear moduli GLR / GRT; GLT / GRT and Poisson ratios [13]. As we shall see further, the displacement of these surfaces is illustrated for various frequencies, for spruce and maple. The displacement is indicated by the colour scale, ranging from blue – displacement zero, to red, maximum displacement, 1 mm. Table 2 and Table 3 show for spruce and maple, typical frequencies and mode shapes of several modes simulated for edge – pinned violin plates of uniform thickness of 3 mm, flat plates, without f - holes, modelled using three hypotheses for the material's anisotropy.
The eigenmodes are indicated by the color scale. As regards the vibration of spruce violin plate illustrated in Table 2, we note that wood anisotropy has no effect on the vibrations patterns for modes 1 to 6. In this table only mode 6 is illustrated. For mode 6 and isotropic, transverse isotropic and orthotropic anisotropies the frequencies are respectively 202.42 Hz, 205.95 Hz and 219.55 Hz. The vibrating surface of high amplitude (yellow and red) is very small.

Table 2. Spruce plates, fixed edge - vibration patterns in three hypotheses of elastic symmetry.

Mode	Isotropic symmetry	Transverse isotropic	Orthotropic symmetry
	Lower modes		
6 **Mode:** Identical **Frequency:** very near	Mode 6, f=202.42 Hz 	Mode 6, f=205.95 Hz 	Mode 6, f=219.55 Hz 
7 **Mode:** identical **Frequency:** very near	Mode 7, f=219.56 Hz 	Mode 7, f=229.82 Hz 	Mode 7, f=247.59 Hz 
8	Mode 8, f=224.71 Hz	Mode 8, f=230.65 Hz	Mode 8, f=248.18 Hz

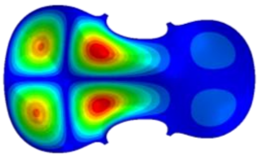
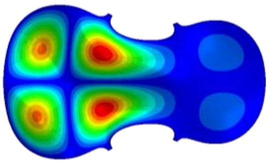
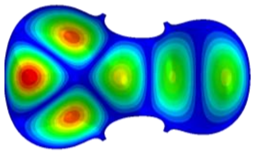
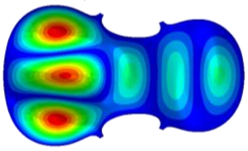
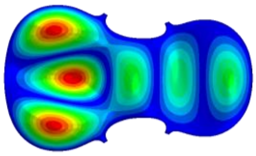
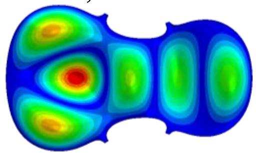
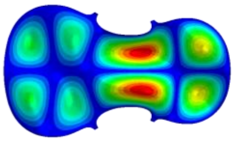
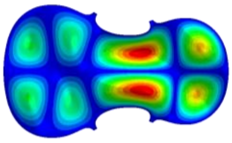
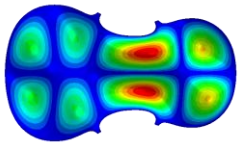
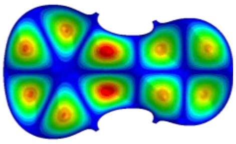
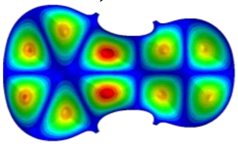
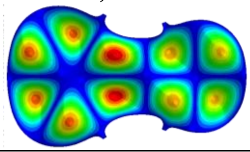
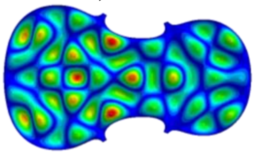
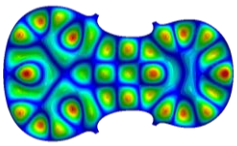
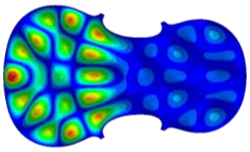
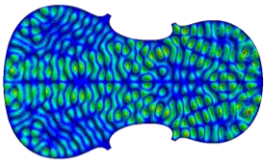
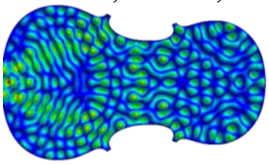
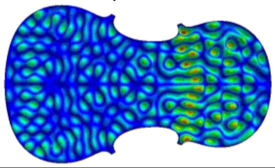
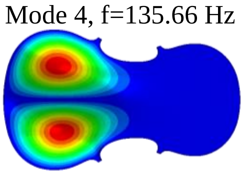
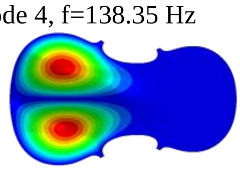
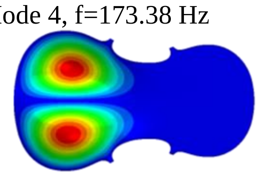
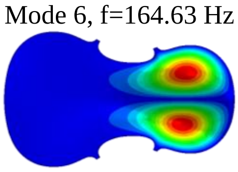
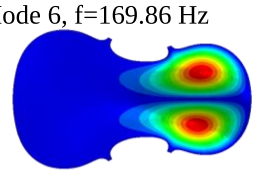
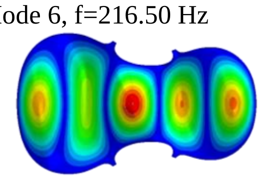
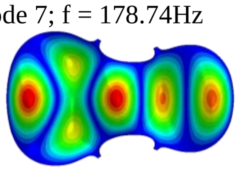
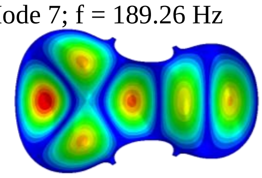
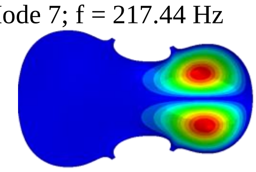
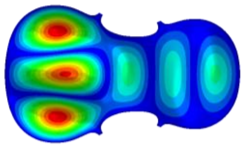
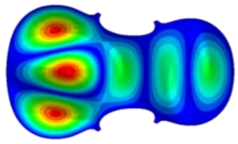
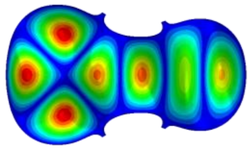
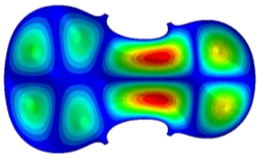
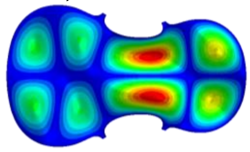
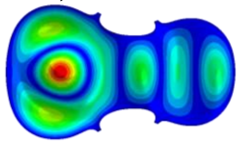
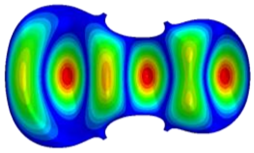
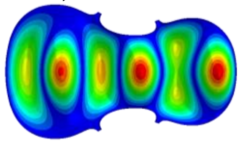
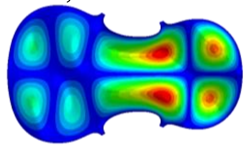
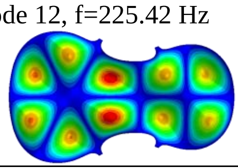
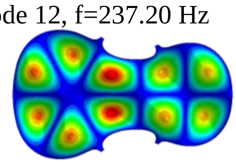
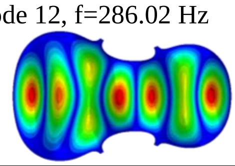
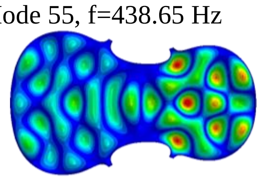
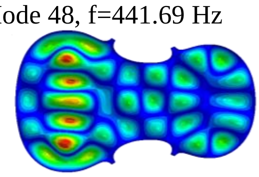
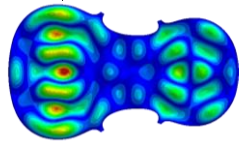
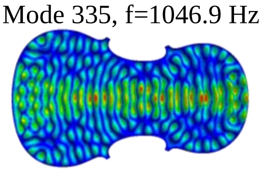
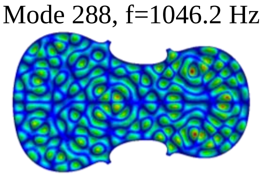
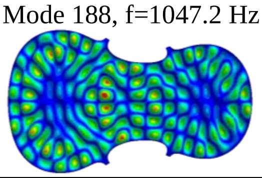
Mode: identical **Frequency:** very near			
9	Mode 9, f=232.74 Hz	Mode 9, f=239.71 Hz	Mode 9, f=257.02 Hz
Mode: Identical **Frequency:** very near			
10	Mode 10, f=256.79 Hz	Mode 10, f=267.25 Hz	Mode 10, f=285.93 Hz
Mode: Identical **Frequency:** very near			
12	Mode 12, f=276.84 Hz	Mode 12, f=288.70 Hz	Mode 12, f=310.52 Hz
Mode: Identical **Frequency:** Different			
	Superior modes		
49 – 66	Mode 66, f = 588.15Hz	Mode 56, f = 591.70 Hz	Mode 49, f = 595.16 Hz
Mode: different **Frequency:** very near			
358 – 508	Mode 508, f = 1568.4 Hz	Mode 423, f = 1568,2 Hz	Mode 358, f = 1568.3 Hz
Mode: different **Frequency:** Identical			

Table 3. Maple plates- vibration patterns in three hypotheses of elastic symmetry

Mode	Isotropic symmetry	Transverse isotropic	Orthotropic symmetry
		Lower modes	
4	Mode 4, f=135.66 Hz	Mode 4, f=138.35 Hz	Mode 4, f=173.38 Hz
Mode identical **Frequency** Near			
6	Mode 6, f=164.63 Hz	Mode 6, f=169.86 Hz	Mode 6, f=216.50 Hz
Mode identical **Frequency** Different			
7	Mode 7; f = 178.74Hz	Mode 7; f = 189.26 Hz	Mode 7; f = 217.44 Hz
Mode identical **Frequency** Different			
9	Mode 9; f=189.49 Hz	Mode 9; f=197.58 Hz	Mode 9; f=248.31 Hz

Mode identical **Frequency** Different			
10 **Mode** identical **Frequency** Different	Mode 10; f=208.99 Hz 	Mode 10; f=218.92 Hz 	Mode 10; f=257.72 Hz 
11 **Mode** identical **Frequency** Different	Mode 11, f=210.48 Hz 	Mode 11, f=224.05 Hz 	Mode 11, f=274.93 Hz 
12 **Mode** identical **Frequency** Different	Mode 12, f=225.42 Hz 	Mode 12, f=237.20 Hz 	Mode 12, f=286.02 Hz 
		Superior modes	
48-58 **Mode** different **Frequency** Different	Mode 55, f=438.65 Hz 	Mode 48, f=441.69 Hz 	Mode 58, f=593.89 Hz 
188 -335 **Mode** different **Frequency** Identical	Mode 335, f=1046.9 Hz 	Mode 288, f=1046.2 Hz 	Mode 188, f=1047.2 Hz 

Mode 7, f=247.59 Hz, show identical patterns for isotropic and transverse isotropy and are different from orthotropic symmetry for which the upper bout, (in blue) is not vibrating. Mode 8 – the patterns are different for the three cases. The orthotropic plate vibrates (red and yellow) mostly on the wider lower bout, at f = 248.18 Hz. Mode 9 – the patterns are different for the three elastic symmetries, but have some similarities - the central part of the lower bout vibrates mostly. Mode 10, f = 285.9 Hz, the center bout vibrates identically for the plates in the three cases of anisotropy. At superior modes, above 588 Hz, the patterns are different for each case of anisotropy. However, in the case of the orthotropic plate, mode 49, at f = 595.16 Hz the upper bout does not vibrate. At frequencies higher than 1500 Hz, the vibrating surfaces of small amplitude are distributed equally on the plate surface. No large amplitudes (red color) were observed. For mode 4 and the isotropic, transverse isotropic and orthotropic anisotropies the frequencies are respectively 135.66 Hz, 138.35 Hz and 173.38 Hz. The vibrating surface (yellow and red) is in the lower bout. Mode 6 shows identical patterns for isotropic and transverse isotropy and is different from orthotropic symmetry for which the central bout vibrates mostly. Mode 7 – the patterns are very similar for isotropic and transverse isotropic cases and different for the orthotropic case. If we compare the modal frequencies of vibration patterns of plates made of spruce and maple at modes ranging from 5 to 12, and for the three cases of anisotropy, one can note that for all anisotropy cases the modal frequency is higher for spruce plate than for maple plate. Furthermore, in the case of isotropy the difference between the modal frequency of spruce plate and maple plate is between about 13% and 28%. In the case of transverse isotropy, the difference between the modal frequency of spruce and maple is between 17.52% and 18.08 % for modes 6, 10 and 12. In the case of orthotropic symmetry the differences are a maximum of 7.88 % for mode 12 and 1.38 % for the first mode. Therefore, it can be noted that theoretically, for orthotropic plates, at the same mode, the modal frequencies of plates made of spruce and maple are very close.

3. CONCLUSION

The paper presents the finite element analysis of violin plates made from spruce and maple, using three hypotheses in pre-processing step related to elastic properties of materials. With the variations of wood anisotropy and modal frequencies, the plate made of maple is more susceptible to vibration than the plate made of spruce. This finding seems to justify once more, the traditional combination of these two species for violin making. The frequency range of analysis was between 65 Hz and 2637 Hz. With the variations of wood anisotropy and modal frequencies, the plate made of maple is more susceptible to vibration than the plate made of spruce. This finding seems to justify once more, the traditional combination of these two species for violin making.

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