Journal of Sound and Vibration (1999) **227**(2), 259–269
Article No. jsvi.1999.2246, available online at http://www.idealibrary.com on IDEAL®

THE FREQUENCY RESPONSE OF DIFFERENTLY MACHINED WOODEN BOARDS

J. KOPAČ AND S. ŠALI

Faculty of Mechanical Engineering, University of Ljubljana, Aškerčeva 6, 1000 *Ljubljana, Slovenia*

(*Received* 28 *September* 1998, *and in final form* 15 *March* 1999)

The changes in frequency response function (FRF) of a guitar top plate due to different machining processes used to finish the guitar tops were analyzed. The way of machining the tops had a measurable but hardly explainable effect on their FRF because of not well-separated frequency peaks. The additional measurements of FRF of differently machined square-shaped plates showed more clearly the influence of the machining process on the acoustic properties of a wooden board. Therefore, a general conclusion is that the machining process influences the acoustic properties of a wooden board. The reason for this influence lies in the different surface shapes of differently machined boards.

1. INTRODUCTION

The task of a luthier is to achieve good sound, even if wooden parts of an instrument have to be modified. These modifications have to be made because the properties of wood vary from one sample to another [1]. For example, additional cutting of the bass bar on the sound board of a violin may lead to a better sound, but the procedure depends on the luthier's ear [2]. The problem originates in the unpredictability of cutting. Thus, the tone of the finished instrument is sometimes quite different from the desired tone. This implies that, in addition to the wood quality and design features, better production technology should be used to improve the tone of the finished instrument. In this paper, a method for measuring the acoustic properties of guitar tops as well as the continuation of the acoustic tests with square-shaped boards [3] is presented.

2. METHODS AND RESULTS

2.1. DEFINITIONS

The variable whose effect on the FRF was studied was the cutting process applied in top plate production (sanding or planing). Only two large braces were glued on each top plate. All braces and top plates were made of spruce (*Picea abies* Karst.), which was seasoned for 30 years: radially cut boards had been sawn from

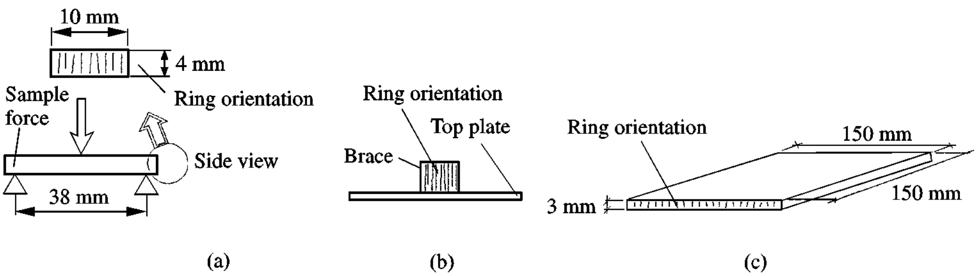


Figure 1. Ring orientation; (a) for samples in a bending test; (b) for braces; (c) for square-shaped specimens.

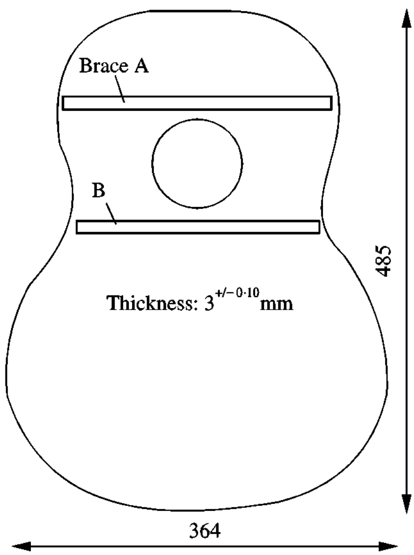


Figure 2. Top plate. Brace A, $10 \times 14 \times 258$ mm; brace B, $10 \times 18 \times 233$ mm

the same log. At a 10% moisture content, the wood density of 20 samples (200 mm × 10 mm × 10 mm) was 451 kg/m^3 (standard deviation: 7 kg/m^3). The modulus of elasticity was measured by the static bending test used for metal testing. Bending of the 20 samples is shown schematically in Figure 1(a). It is assumed that the variation of the modulus of elasticity does not depend on the way of testing, but only on the anisotropy and non-homogeneity of samples. Because the static bending test for metals was used, the measured average modulus of elasticity is not reliable. Thus, it is reasonable to define a quantity which expresses relatively a variation of the modulus of elasticity with respect to its mean value. This quantity is denoted as the ratio of *standard deviation/mean value* of the modulus of elasticity and it was approximately 7% for the tested wood. A sketch of a top is shown in Figure 2, where A and B indicate the two variable braces. The cross-sections of braces A and B, were rectangular. All braces were machined by planing. Figure 1(b) shows the ring orientation of the glued braces which was always perpendicular to

Table 1

The characteristics of planing and belt sanding

Planing	Belt sanding
Diameter of cutting tool: 117 mm	Coating of the contact wheel: rubber, 40 Shore
Number/material of cutting edges: 3/HSS	Number of transversal oscillations of belt: 60/min
Cutting speed: 37 m/s	Abrasive material: garnet, grain size 60
Sharpness/rake angle: 45°/30°	Speed of the belt: 21 m/s
Feeding speed: 7 m/min	Feeding speed: 7 m/min
Depth of cut: 2 mm on each side (2 passes)	Depth of sanding: 1 mm on each side (four passes)

the plate. Table 1 shows the characteristics of the applied cutting processes. The final products made of raw boards (1600 mm × 180 mm × 11 mm) were as follows: top plates (see Figure 2); square-shaped specimens (150 mm × 150 mm × 3 mm) with equal grain and ring orientation [see Figure 1(c)].

The planes of cutting and feeding speed were always parallel to the grain orientation. The technological parameters of each cutting process were chosen to attain good surface quality of top plates and square-shaped specimens, respectively.

2.2. COMPARISON OF SANDED AND PLANED TOP PLATES

In order to set the experiments with the sanded and planed guitar tops in the context of similar investigations, tests with free-supported top plates were done. The top plates were suspended on two elastic nylon lines and their FRF was obtained by measuring the excitation and response signals simultaneously. The excitation of the top plates was carried out with the device shown in Figure 3(a). The main problem in testing relatively light top plates was to ensure an impulse excitation with no additional pulses after the first one. It is evident from Figure 3(a) that the problem was solved with a relatively small mass of the weight and the accelerometer on the one hand, and with low stiffness of the contacting surfaces (foam rubber F1) on the other hand [5]. The accelerometer was pressed into the weight made of relatively elastic material. Therefore, the weight and the accelerometer are considered as one part. The slide bumped into the weight-with-accelerometer, and then this impulsively excited the top plate indirectly over the foam rubber F1, which resulted in only one pulse. Foam rubber F2 also provided only one pulse since it prevented additional impacts between the slide and the weight-with-accelerometer after the first impact. Because of small area (15 × 15 × 5 mm) and mass of the foam rubber F1 its influence on board vibration is presumably negligible. During FRF measurement (0·256 s) the top was not in the contact with the weight-with-accelerometer. Namely, the foam rubber F1 provided a rebound of the weight-with-accelerometer after the impact into the top. The 0·65 g

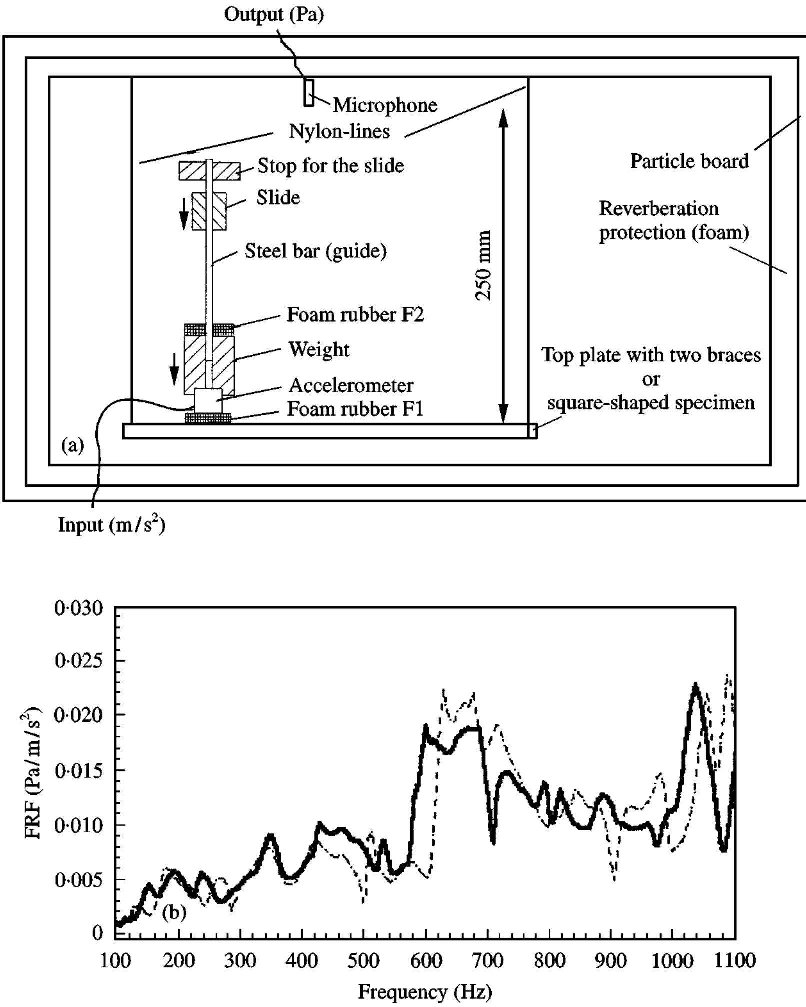


Figure 3. Testing of freely supported top plates (square-shaped specimens): (a) Experimental set-up; (b) Frequency response plots of top plates. planed sanded.

heavy accelerometer was actually used instead of a force transducer which means that the units of the input signal are m/s^2, but because of the constant mass of the weight-with-accelerometer these units are proportional to Newton units. The position of the pulse excitation was on the top plate symmetry line and 170 mm from the centre of the hole towards the bottom rim. The response of the top plate was measured by a condenser microphone, so the response signal was measured in Pa.

In this experiment the responses of eight top plates (four planed, four sanded) were measured. In order to obtain statistically significant results, three different braces A with equal shape were exchanged for each equally machined top plate. For each one of these three braces A, three braces B were exchanged. For each group of equally machined top plates this resulted in 36 ($4 \times 3 \times 3$) variations of changing of braces A and B. Figure 3(b) shows the resulting average FRF for the planed and sanded top plates with two braces.

2.3. TESTING OF SQUARE-SHAPED SPECIMENS

In order to confirm the different behaviour of the variously machined top plates, square-shaped specimens made of spruce (see Section 2.1) were selected to represent a typical portion of the guitar sound board. An experiment, where the square-shaped specimens were suspended from two elastic nylon lines, was performed. The procedure of testing was exactly the same as in testing the top plates which was described in Section 2.2. The position of the accelerometer's impact was in the centre of the specimen. Figure 4 shows the first peak of the average FRF for 20 planed and 20 sanded specimens.

A hypothesis was made that the average values for damping of the first peak, amplitude of the first peak and position of the first peak (Hz) of the two groups of differently machined specimens are significantly different at the statistical significance level of 0·05 [6]. The t- and F-tests were used for statistical calculations, where it was assumed that the individual values for damping, amplitude and position of the first peak (extracted from the individual FRF plots [5]) are distributed according to the Gaussian distribution.

The experiments with square-shaped specimens prove that the cutting process influences their acoustic properties in general: the average value for position (Hz) of the first peak in FRF plot is significantly higher for sanded specimens; no significant differences in the average value for amplitude (gain) of the first peak in FRF plot occurred for differently machined specimens; and the average value for damping of the first peak in FRF plot is significantly higher for sanded specimens.

2.4. DISCUSSION

Because of not well-separated frequency peaks, the average frequency responses of top plates [Figure 3(b)] are difficult to explain. In general, the sanded top plates have weaker response in the frequency range 100–500 Hz, especially for the first frequency peak, which indicates worse acoustic properties in comparison to the planed top plates. In contrast, the acoustic response in the range 500–1000 Hz was more intensive for the sanded top plates, which indicates better acoustic properties in comparison to the planed top plates.

Therefore, a more comprehensive study was performed with experimentation with square-shaped specimens which shows more clearly the influence of the machining process on the acoustic properties of the plates. Damping of the first peak in FRF plot is lower in the case of planing and the frequency of the first resonance peak in FRF of the planed specimens is lower than that for the sanded specimens. Recent measurements of acoustic response of square-shaped specimens clamped at four edges resulted in similar conclusions [3]: the specimens were vibrated by impacting them with a small wooden ball and the resulting oscillations were measured by an accelerometer mounted on the board. In these experiments, (i) the logarithmic decrement of damping of the planed specimens was lower than that of the sanded specimens, and (ii) the position of the first frequency peak in the power spectrum of the sanded specimens was significantly higher.

The most probable reason for the different behaviour of the differently machined tops and square-shaped specimens is to be sought in the differences in their surface layers [3]. It is well known that sanded boards have torn fibres whereas planed boards have chopped fibres. Tearing of fibres damages the integrity of the surface much more than chopping. The different shape of surface layers results in either one or both of the following consequences.

1. *Different modulus of elasticity* (E) *of boards.* It is logical that the planed boards will suffer smaller deformation in static bending test than the sanded boards. The reason lies in the less damaged surface layers of the planed boards. This is more true because the differences in thickness of differently machined boards were negligible. Because of the orthotropy of wood, three moduli of elasticity are relevant for its mechanical (acoustic) properties: E_L, E_R and E_T, where subscripts L, R and T indicate longitudinal, radial and tangential direction in wood respectively [7]. It is reasonable to say that because of different strength properties of the two surface layers and the middle layer of the wooden board, E is in fact an average modulus. Thus, moduli $\bar{E}_L$, $\bar{E}_R$ and $\bar{E}_T$ indicate mechanical properties of differently machined thin boards made of wood.

2. *Different density* (ρ) *of boards.* It is possible that the density of surface layers of the planed boards is slightly smaller in comparison to the sanded boards. Because of sharp cutting edges of the cutting tool in planing, the wood tissue was presumably cut ideally and without considerable pressures perpendicular to the surface [4]. In contrast, compression of wood tissue is more likely to happen in belt-sanding because of the action of the pressure bar. This presumably resulted in viscous deformation of wood and in increase of local density. As in the case of three moduli E, an average density of the board ($\bar{\rho}$) is considered in this analysis. Indeed measurements of weight did not show significant differences in average density of differently machined boards, but due to wood non-homogeneity this does not exclude the above hypothesis. Therefore, in the following analysis, two situations (for different and equal average density of differently machined boards) are commented.

According to Kollmann and Côté [8], wooden resonant boards of various musical instruments should translate as much as possible the input energy into sound radiation. More precisely, the ratio $\sqrt{((E/\rho)/\rho)}$ should be as high as possible and the ratio $\sqrt{E\rho}$ should be as low as possible. These two ratios indicate radiation damping ϑ and sound wave resistance Z, respectively. Thus, it is presumed that low damping and high amplitude of the first peak in FRF plot are desirable features of the tested boards. If so, for the square-shaped specimens planing is better than sanding.

According to the much higher stiffness of spruce in the longitudinal direction than that in the radial direction [7], one can assume that the analyzed resonance peak in the FRF plot is a result of the ith ($i = 1, 2, 3, \dots$) board mode, which is bending along the grain (in the longitudinal direction of the wood tissue). To explain the significant differences in acoustic response of differently machined wooden boards, one can conditionally use the following expression, which relates the ith mode frequency with mechanical properties of a homogeneous square plate (free-supported) [9]:

$$f_i = C_i t \sqrt{E/\rho(1 - v^2) l^4}. \tag{1}$$

Here C_i depends on i, t is the board thickness, v the Poisson's ratio and l the length (width) of the specimen. Out of six Poisson's ratios only two are relevant for the presumed modal behaviour of the board. These are $v_{LR} = 0{\cdot}37$ (longitudinal–radial plane) and $v_{LT} = 0{\cdot}42$ (longitudinal–tangential plane) [7]. By definition, Poisson's ratio is a ratio of passive to active deformation; thus in bending along the grain v_{LR} indicates the ratio of passive deformation in radial direction to active deformation in longitudinal direction. By analogy, v_{LT} indicates the ratio of passive deformation in tangential direction to active deformation in longitudinal direction. Thus, Poisson's ratio from expression (1) is presumably dependent on these two single ratios whose absolute values differ by only about 12%. For this reason, one can use $\bar{v}$ which indicates the mean value of the two single Poisson's ratios instead of v [see expression (1)] in the following analysis:

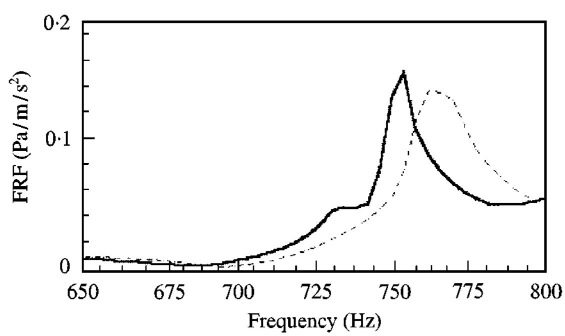
$$f_i = D \sqrt{\bar{E}_L/\bar{\rho}(1 - \bar{v}^2)}. \tag{2}$$

Here D denotes the shape of the specimen and the constant C_i. The experiments showed that the ith mode frequency is significantly higher by approximately 1·5% for the sanded specimens (see Figure 4). Considering this and the assumption that D is the same for the planed and sanded specimen, one can write

$$f_{is} \approx 1{\cdot}015 f_{ip} \Rightarrow \sqrt{\bar{E}_{Ls}/\bar{\rho}_s(1 - \bar{v}_s^2)} \approx 1{\cdot}015 \sqrt{\bar{E}_{Lp}/\bar{\rho}_p(1 - \bar{v}_p^2)}, \tag{3}$$

where subscripts s and p stand for sanding and planing respectively. This expression can also be written as

$$\frac{(1 - \bar{v}_p^2)}{(1 - \bar{v}_s^2)} \approx \frac{\bar{E}_{Lp}\bar{\rho}_s}{\bar{E}_{Ls}\bar{\rho}_p} \times 1{\cdot}015^2 \tag{4}$$



Characteristics of the first peak	SANDED Mean	SANDED Std.dev.	PLANED Mean	PLANED Std.dev
Position (Hz)	762	12	750	12
Amplitude (Pa/m/s^2)	0.137	0.06	0.15	0.04
Damping	0.047	0.017	0.039	0.004

Figure 4. The average FRF for the sanded and planed specimens (first peak in FRF, ith mode) planed; sanded.

Considering the above assumptions that $\bar{E}_{Lp} > \bar{E}_{Ls}$ and $\bar{\rho}_p \leqslant \bar{\rho}_s$, one sees that the value of ϑ is more favourable for the planed specimens:

$$\frac{\vartheta_p}{\vartheta_s} = \frac{\sqrt{\bar{E}_{Lp}/\bar{\rho}_p}/\bar{\rho}_p}{\sqrt{\bar{E}_{Ls}/\bar{\rho}_s}/\bar{\rho}_s} > 1, \tag{5}$$

whereas the ratio of Z for planed and sanded specimens could be greater, smaller or equal to one. In addition, from expression (4) one can see that

$$1 - \bar{\nu}_p^2 > 1 - \bar{\nu}_s^2 \Rightarrow \bar{\nu}_s > \bar{\nu}_p. \tag{6}$$

The differences in $\bar{E}_L$ and $\bar{\rho}$, and slightly higher $\bar{\nu}$ of the sanded boards relatively to the planed boards can explain the differences in their acoustic response. $\bar{\nu}$ depends on ν_{LR} and ν_{LT} (see above) whose characteristics are as follows.

Bending along the grain results in longitudinal and therefore in radial deformation of wood tissue, too. The ratio of both deformations is denoted by ν_{LR}. Figure 5 shows the presumed active cross sections in three directions. It is evident that an active cross-section in radial direction A_R is larger for the planed specimens and the active cross-section in longitudinal direction A_L is the same for both groups of specimens. This implies a larger ratio of radial deformation ε_R to longitudinal deformation ε_L for the sanded in comparison to planned boards subject to tension in longitudinal direction. Therefore, in bending along the grain (longitudinal direction), the ratio of ε_R to ε_L is logically higher for the sanded boards:

$$(A_R/A_L)_s < (A_R/A_L)_p \Rightarrow |\nu_{LR}|_s = |\varepsilon_R/\varepsilon_L|_s > |\nu_{LR}|_p = |\varepsilon_R/\varepsilon_L|_p. \tag{7}$$

Bending along the grain results in longitudinal and therefore in tangential deformation of wood tissue, too. The ratio of both deformations is denoted by ν_{LT}. Figure 5 shows that the active cross-section in tangential direction A_T is the same for both groups of specimens. This and the equality of A_L's for all specimens implies an equality of the ratios of deformation in tangential direction ε_T to deformation in longitudinal direction ε_L for all boards subject to tension in longitudinal direction. This also implies a situation in bending:

$$(A_T/A_L)_s = (A_T/A_L)_p \Rightarrow |\nu_{LT}|_s = |\varepsilon_T/\varepsilon_L|_s = |\nu_{LT}|_p = |\varepsilon_T/\varepsilon_L|_p. \tag{8}$$

Expressions (7) and (8) show that indeed the $\bar{\nu}$ (mean value of the two analysed Poisson's ratios) is higher for the sanded boards, which together with the differences in $\bar{E}_L$ and $\bar{\rho}$ explains the differences in the acoustic response of differently machined boards.

According to the considerably higher $\bar{E}_L$ in comparison to $\bar{E}_R$, it is not likely that the analyzed frequency peak in the FRF plot corresponds to the ith mode of the board, which is bending across the grain (in the radial direction of the wood tissue) [7, 10]. Theoretically, it is possible that the analyzed ith mode is a torsional mode. Torsional modes, however, depend mainly on shear moduli [8], and expression (1)

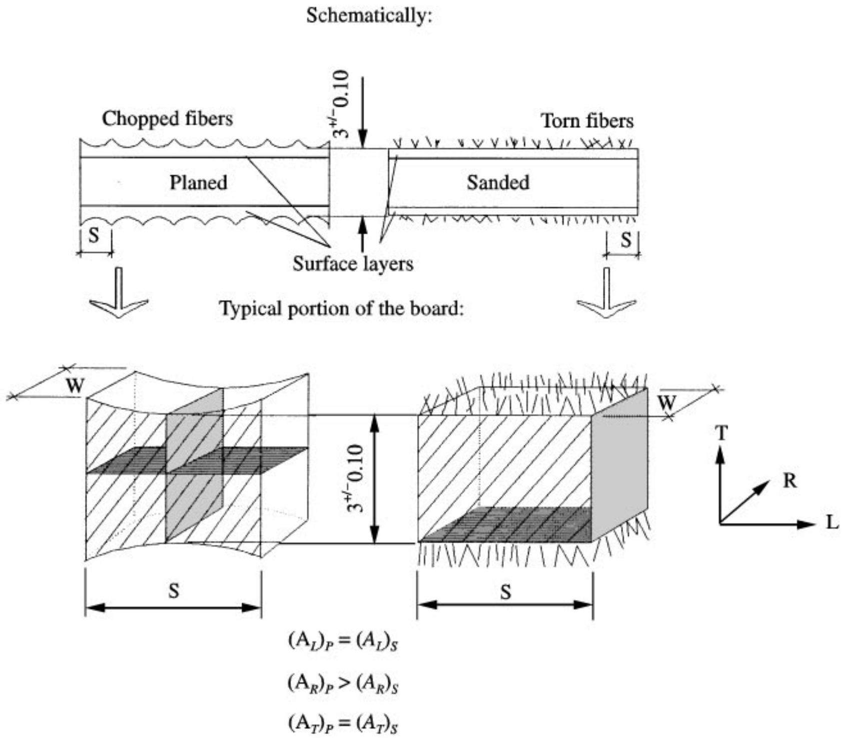


Figure 5. The differences in surface layers and active cross-sections for the sanded and planed specimens. □ A_L; ▨ A_R; ■ A_T.

is therefore not a useful starting point to explain the different acoustic responses of the boards. According to Ono [10] only the shear modulus in the longitudinal–tangential plane (G_{LT}) is responsible for the torsional vibrations of a wooden plate. This is logical, because the shear modulus in the longitudinal–radial direction is not relevant for torsional vibrations and is in the radial–tangential plane relatively small in comparison to the other two moduli. In addition, the propagation velocity of elastic torsional waves is $\sqrt{G/\rho}$ [8] and depends mostly on the shear modulus G_{LT} (see above) which can be for the present thin wooden specimens denoted as an average modulus $\bar{G}_{LT}$ similarly to the $\bar{E}_L$, $\bar{E}_R$, $\bar{E}_T$ and $\bar{\rho}$. The higher speed of sound in wood, regardless of the direction of propagation, indicates better acoustic response [8]; therefore one can say that the higher propagation velocity of elastic torsional waves also indicate better acoustic response. Similarly as for $\bar{E}_L$ and $\bar{E}_R$ one can assume that $\bar{G}_{LT}$ is more favourable (higher) for the planed specimens because of the higher strength of the fibers on both surface layers presenting a considerable portion of board's thickness. In addition, the active cross-section of the longitudinal–tangential plane (denoted as A_R in Figure 5) is presumably higher for the planed boards which implies a higher $\bar{G}_{LT}$ in tests of shear strength of the board in the corresponding plane.

Considering also the assumption about smaller or equal average density of the planed specimens, one can say

$$(\sqrt{\bar{G}_{LT}/\bar{\rho}})_p > (\sqrt{\bar{G}_{LT}/\bar{\rho}})_s, \tag{9}$$

which agrees with better acoustic response of the planed specimens.

Equations (1)–(9) can explain the relations between the machining process, mechanical properties of the wood and its frequency response. Even if the average densities for the sanded and planed specimens are considered to be equal, equation (5) is still fulfilled. In this case, the radiation damping is still more favourable for the planed specimens and sound wave resistance is definitely slightly higher for this group of specimens. The latter means a decrease in acoustic response, but according to Kollmann and Côté [8] soundboards of top quality may have slightly higher sound wave resistance than do soundboards of average quality.

3. CONCLUSION

There is no doubt that the machining process influences the acoustic properties of a wooden board in general. However, this influence depends on the board shape and probably also on the mode of the board support [3]. Because of not well-separated frequency peaks, FRF plots of braced top plates are difficult to explain. The probable reason for the high density of frequency peaks were the two braces on the top plates which increased the stiffness of the plate in the radial direction. This resulted in increase in number of modes (frequency peaks), which depend on radial stiffness.

Experiments with square-shaped specimens showed more clearly the superiority of planing in comparison to sanding which agrees with the conclusions from the recent tests with the same specimens but clamped at all four edges [3]. One of the probable reasons for this influence is to be sought in the differences between surface structures of sound boards that are machined in different ways. It is likely that these differences result in different moduli of elasticity, Poisson's ratios, shear moduli and probably also the average density of differently machined wooden boards. This suggests that a comprehensive analysis of acoustic properties of an instrument should include the machining process as a variable, not a constant.

REFERENCES

1. G. Tsoumis 1991 *Science and Technology of Wood*. New York: Van Nostrand Reinhold Company.
2. S. Arakelian 1978 *Die Geige*. Frankfurt am Mein: Das Musikinstrument.
3. S. Šali and J. Kopač 1998 *Catgut Acoustical Society Journal* **3**, 15–21. The influence of different machining processes on the acoustic properties of wooden resonant boards.
4. C. M. Hutchins 1983 *Catgut Acoustical Society Journal* **1**, 49. Experimental evidence that free plate mode tuning is primarily a method of controlling stiffness in the top and back of violins (abstract).
5. D. J. Ewins 1984 *Modal Testing: Theory and Practice*. Letchworth: Research Studies Press Ltd.

6. G. K. Kanji 1993. 100 *Statistical Tests*. London: SAGE Publications Ltd.
7. J. Bodig and B. Jayne 1982 *Mechanics of Wood and Wood Composites*. New York: Van Nostrand Reinhold Company.
8. F. Kollmann and A. Côté 1968 *Principles of Wood Science and Technology*, *Part* 1: *Solid Wood*. Berlin: Springer.
9. C. M. Harris 1988 *Shock Vibration Handbook*. New York: McGraw-Hill Book Company, third edition.
10. T. Ono 1989 *Concise Encyclopedia of Wood & Wood-based Materials*. Oxford: Pergamon Press.