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## **THE FREQUENCY RESPONSE OF DIFFERENTLY MACHINED WOODEN BOARDS**

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Summary:

The changes in frequency response function (FRF) of a guitar top plate due to different machining processes used to finish the guitar tops were analysed. 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.

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 the 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 the same log. At a 10% moisture content, the wood density of 20 samples (200 mm × 10 mm × 10 mm) was 451 kg/m<sup>3</sup> (standard deviation: 7 kg/m<sup>3</sup>). 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). We 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 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:

- top plates (see Figure 2), and
- 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 second) 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 heavy accelerometer was actually used instead of a force transducer which means that the units of the input signal are m/s<sup>2</sup>, 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 response of eight top plates (four planed, four sanded) was 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 is 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.
- 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 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 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 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 resonant peak in FRF of the planed specimens is lower in comparison to 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 in comparison to 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 the more true because the differences in thickness of differently machined boards were negligible. Because of 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  $\overline{E}_L$ ,  $\overline{E}_R$  and  $\overline{E}_T$  indicate mechanical properties of differently machined thin boards made of wood.

2. **Different density ( $\rho$ ) 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 ( $\overline{\rho}$ ) 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 following analysis two situations (for different and equal average density of differently machined boards) are commented.

According to Kollmann & Côté [8], wooden resonant boards of various musical instruments should translate as much as possible of the input energy into sound radiation. More precisely, the ratio  $\sqrt{((E/\rho)/\rho)}$  should be as high as possible and the ratio  $\sqrt{E \cdot \rho}$  should be as low as possible.

These two ratios indicate radiation damping  $\mathcal{G}$  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 much higher stiffness of spruce in longitudinal direction in comparison to that in radial direction [7], we can assume that the analysed resonant peak in FRF plot is a result of  $i$ -th ( $i=1, 2, 3 \dots$ ) board mode, which is bending along the grain (in longitudinal direction of wood tissue). To explain the significant differences in acoustic response of differently machined wooden boards we can conditionally use expression (1), which relates the  $i$ -th mode frequency with mechanical properties of a homogeneous square plate (free-supported) [9]:

$$f_i = C_i \cdot t \cdot \sqrt{\frac{E}{\rho \cdot (1 - \nu^2)} \cdot l^4}, \quad (1)$$

where  $C_i$  depends on  $i$ ,  $t$  is board thickness,  $\nu$  Poisson's ratio and  $l$  length (width) of the specimen. Out of 6 Poisson's ratios only two are relevant for the presumed modal behaviour of the board. These are  $\nu_{LR} = 0.37$  (longitudinal-radial plane) and  $\nu_{LT} = 0.42$  (longitudinal-tangential plane) [7]. By definition Poisson's ratio is a ratio of passive to active deformation, thus in bending along the grain  $\nu_{LR}$  indicates the ratio of passive deformation in radial direction to active deformation in longitudinal direction. By analogy,  $\nu_{LT}$  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 only by about 12 %. For this reason we can use  $\overline{\nu}$  which indicates the mean value of the two single Poisson's ratios instead of  $\nu$  (see expression (1)) in the following analysis:

$$f_i = D \cdot \sqrt{\frac{\overline{E}_L}{\overline{\rho} \cdot (1 - \overline{\nu}^2)}}, \quad (2)$$

where  $D$  denotes the shape of the specimen and the constant  $C_i$ . The experiments showed that the  $i$ -th 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, we can write:

$$f_{is} \approx 1.015 \cdot f_{ip} \Rightarrow \sqrt{\frac{\bar{E}_{Ls}}{\bar{\rho}_s \cdot (1 - \bar{v}_s^2)}} \approx 1.015 \cdot \sqrt{\frac{\bar{E}_{Lp}}{\bar{\rho}_p \cdot (1 - \bar{v}_p^2)}}, \quad (3)$$

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

$$\frac{(1 - \bar{v}_p^2)}{(1 - \bar{v}_s^2)} \approx \frac{\bar{E}_{Lp} \cdot \bar{\rho}_s}{\bar{E}_{Ls} \cdot \bar{\rho}_p} \cdot 1.015^2. \quad (4)$$

Considering the above assumptions that  $\bar{E}_{Lp} > \bar{E}_{Ls}$  and  $\bar{\rho}_p \leq \bar{\rho}_s$ , we see that the value of  $\mathcal{G}$  is more favourable for the planed specimens:

$$\frac{\mathcal{G}_p}{\mathcal{G}_s} = \frac{\sqrt{\bar{E}_{Lp} / \bar{\rho}_p} / \bar{\rho}_p}{\sqrt{\bar{E}_{Ls} / \bar{\rho}_s} / \bar{\rho}_s} > 1, \quad (5)$$

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

$$1 - \bar{v}_p^2 > 1 - \bar{v}_s^2 \Rightarrow \bar{v}_s > \bar{v}_p. \quad (6)$$

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

- bending along the grain results in longitudinal and therefore in radial deformation of wood tissue, too. The ratio of both deformations is denoted by  $v_{LR}$ . 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  $\varepsilon_R$  to longitudinal deformation  $\varepsilon_L$  for the sanded in comparison to planed boards subject to tension in longitudinal direction. Therefore, in bending along the grain (longitudinal direction), the ratio of  $\varepsilon_R$  to  $\varepsilon_L$  is logically higher for the sanded boards:

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

- bending along the grain results in longitudinal and therefore in tangential deformation of wood tissue, too. The ratio of both deformations is denoted by  $v_{LT}$ . 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  $\varepsilon_T$  to deformation in longitudinal direction  $\varepsilon_L$  for all boards subject to tension in longitudinal direction. This also implies a situation in bending:

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

Expressions (7) and (8) show that indeed the  $\bar{v}$  (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 considerable higher  $\overline{E}_L$  in comparison to  $\overline{E}_R$ , it is not likely that the analysed frequency peak in FRF plot corresponds to the  $i$ -th mode of the board, which is bending across the grain (in radial direction of wood tissue) [7, 10]. Theoretically it is possible that the analysed  $i$ -th mode is a torsional mode. Torsional modes, however, depend mainly on shear moduli [8], and expression (1) is therefore not a useful starting point to explain the different acoustic response of the boards. According to Ono [10] only the shear modulus in longitudinal-tangential plane ( $G_{LT}$ ) is responsible for the torsional vibrations of a wooden plate. This is logical, because the shear modulus in longitudinal-radial direction is not relevant for torsional vibrations and is in 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 our thin wooden specimens denoted as an average modulus  $\overline{G}_{LT}$  similarly to the  $\overline{E}_L$ ,  $\overline{E}_R$ ,  $\overline{E}_T$  and  $\overline{\rho}$ . Higher speed of sound in wood, regardless of the direction of propagation, indicates better acoustic response [8], therefore we can say that higher propagation velocity of elastic torsional waves also indicate better acoustic response. Similarly as for  $\overline{E}_L$  and  $\overline{E}_R$  we can assume, that  $\overline{G}_{LT}$  is more favourable (higher) for the planed specimens because of higher strength of the fibers on both surface layers presenting a considerable portion of board's thickness. In addition, the active cross section of longitudinal-tangential plane (denoted as  $A_R$  in Figure 5) is presumably higher for the planed boards which implies a higher  $\overline{G}_{LT}$  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, we can say:

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

which agrees with better acoustic response of the planed specimens.

Equations (1) to (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 & Côté [8] the soundboards of top quality may have slightly higher sound wave resistance in comparison to the 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 high density of frequency peaks are 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 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.

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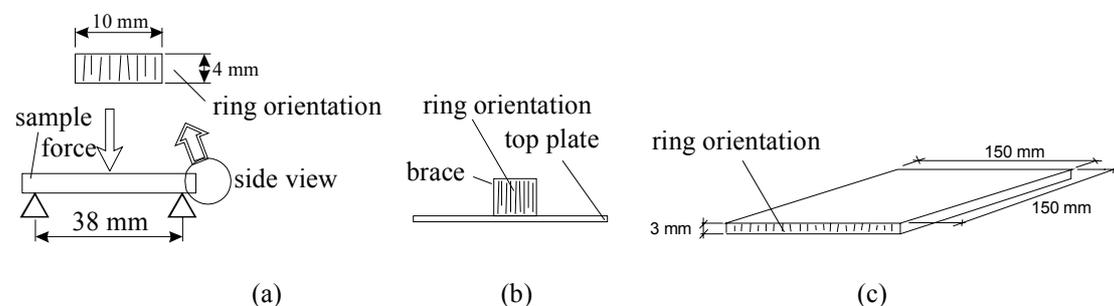


Figure 1: Ring orientation; (a) for samples in a bending test; (b) for braces; (c) for square-shaped specimens.

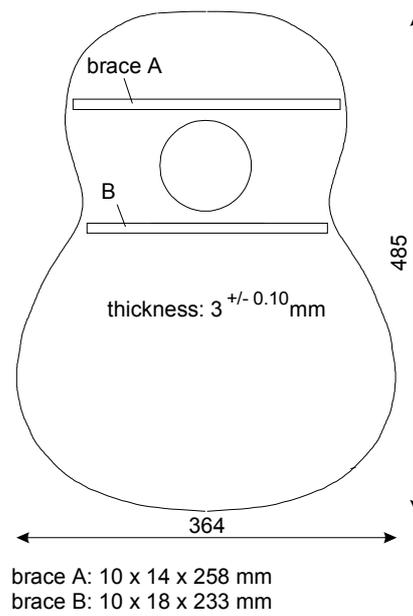


Figure 2: Top plate.

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 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 (4 passes)

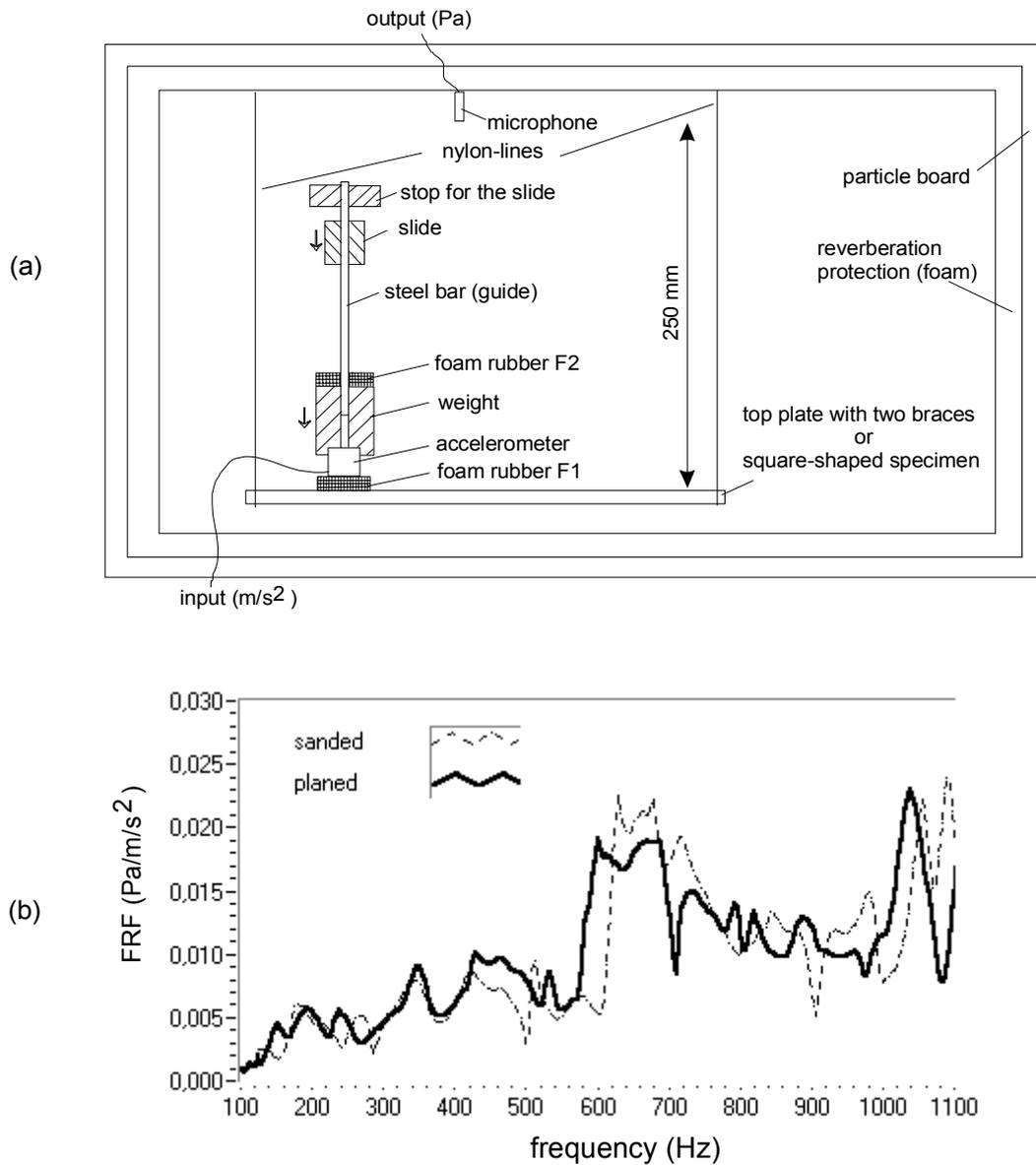


Figure 3: Testing of freely supported top plates (square-shaped specimens): (a) Experimental set-up; (b) Frequency response plots of top plates.

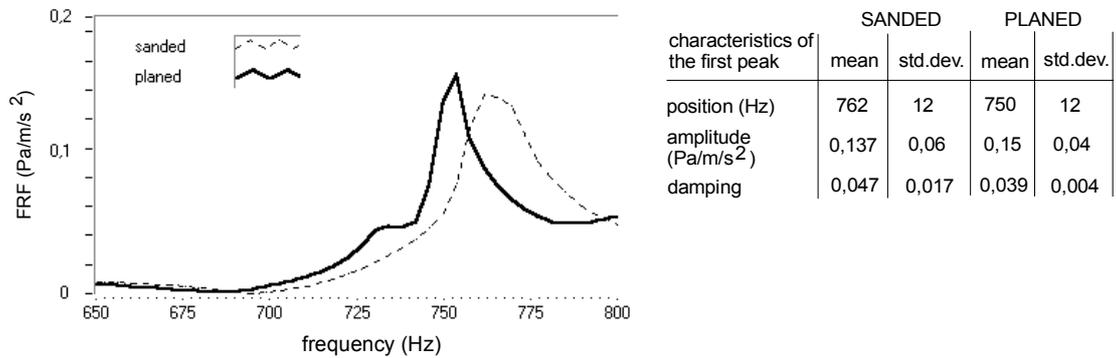


Figure 4: The average FRF for the sanded and planed specimens (first peak in FRF,  $i$ -th mode).

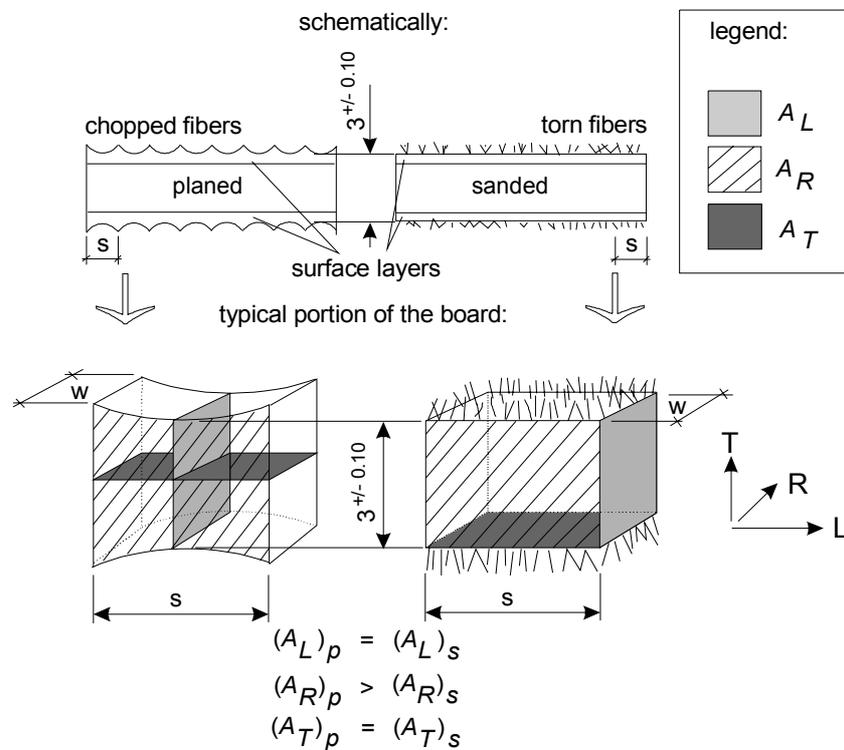


Figure 5: The differences in surface layers and active cross-sections for the sanded and planed specimens.