JOURNAL of the CATGUT ACOUSTICAL SOCIETY

an international publication devoted to research in the theory, design, construction, and history of stringed instruments and to related areas of acoustical study.

Number 46 November 1986

Published Semiannually

ISSN: 0882-2212

A VIOLINMAKER'S PRACTICAL TEST OF WOOD PROPERTIES SUGGESTED FROM FEM-ANALYSIS OF AN ORTHOTROPIC SHELL

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ABSTRACT

An orthotropic shell element is introduced into a FEM-program. The program is used for numerical experiments, i.e., to calculate vibration modes of wooden plates of nonuniform thickness. Calculated results are shown to be in agreement with measured ones. As a result, parameters of the material, spruce, can be determined. A method to determine material parameters of anisotropic shells is thus proposed and a simple method for the violinmaker to test his material.

INTRODUCTION

The vibration properties of the violin plates determine the quality of a violin and they have been extensively studied - the relations between thin rectangular plates and free violin plates (Beldie, 1968), the plates glued to the ribs (Jansson, Molin & Sundin, 1970), the adjustment, "tuning" of the free violin top plates (Hutchins, 1980, 1981) and the relation between free plate properties and those of assembled instruments (Alonso Moral, 1984). The properties of the wood for a musical instrument are considered to be of great importance. But, as no practically applicable tests exist, no real quantitative evaluation has been made. The properties have been investigated many times. Usually, pairs of wood strips are used, one longitudinally and one transversally cut (Barducci & Pasqualini, 1975; Jansson, 1975, 1978; Haines, 1979, 1980). Such methods are, however, rather time-consuming. Furthermore, laboratory equipment is needed. Recently, it was tried and found that normal modes of vibration of the wooden "blank" can be recorded in a simple way, and the result can be used as a starting point for violin making. The results indicated that the normal modes may in a simple way reflect the major material properties of the wood and, thus, make it possible to test material properties in the luthier's workshop. In a first investigation, normal modes of blanks for back and top plates were mapped (Ek & Jansson, 1985). The present work is a condensed version of a previous, more extensive report (Molin, Tinnsten, Wiklund & Jansson, 1984). The information most relevant to the violin maker has been extracted. After its first completion, other related works have been reported (Caldersmith, 1984; McIntyre & Woodhouse, 1984, 1985, 1986).

The present work is our first step to study properties of free plates, i.e., to test a FE-model of the wood. As the results are positive, properties of free plates have been studied and a preliminary report has been published (Molin, Lindgren & Jansson, 1986).

1. Material and Acoustical Measurements

Two "blanks" each for violin tops and backs were made with measures close to chosen standards, see Fig. 1. The geometry and the masses of the blanks were carefully measured.

Two series of acoustical measurements were made (cf., Ek & Jansson, 1985). First, the input admittance was measured in a corner with the blanks hung in rubber bands. Thereby, frequency responses were obtained with peaks representing the different resonances. The first three peaks correspond to the eigenmodes investigated in this report. In some cases, the third peak of interest may be the fourth peak in frequency order (cf., Ek & Jansson, 1985).

Secondly, the nodal line patterns were measured by means of Chladni patterns. The plate is placed over a loudspeaker and is supported by wedges from plastic sponges (Hutchins, 1983). Small particles, for instance sawdust, are sprinkled over the plate. The loudspeaker gives a loud tone. The frequency is adjusted so that the sawdust "jumps" maximally. The sawdust will soon collect at nodal lines giving a picture of the vibrations of that particular eigenmode. If in wrong positions, the supports must be moved to nodal lines and the loudspeaker to an antinodal position.

For the present project the resonance frequencies, the widths (at -3 dB) of the resonance peaks, and the nodal patterns for the three lowest modes were noted for each of the four "blanks."

Thereafter, test bars of specific dimensions were cut out of the blanks, as sketched in Fig. 1. This is a delicate business as it is vital and difficult to cut out representative samples. The frequency and the width of the first resonance of the "free-free" bar were measured in the following way. By means of a small loudspeaker and supporting plastic wedges at the nodal lines (22% from the ends), the bar was set into vibration, i.e., with principally the same system as for the plate. An optical sond registered the vibrations and gave an electrical signal proportional to the vibration amplitude. This signal was fed into the measurement system. Thus, the frequencies (a measure of the modulus of elasticity) and the resonance bandwidths (a measure of the internal losses) can be measured for bars cut along and across the growth direction of the wood.

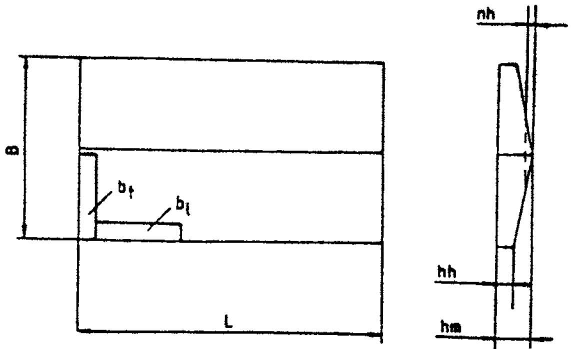

Fig. 1. Measures of the experimental blank (standard size, if different within brackets: Mattsson, 1984): L = 386 (385)mm, B = 215 mm, hh - 8.5 (7.5)mm, hm = 20 mm, nh = 0 mm, and mass = 570 g (spruce).

The measured properties were found to be representative by comparison with measurements of five top and five back blanks (Jansson, Hansen, & Hansen, 1984).

For the following numerical experiments one spruce blank was chosen as representative. After the FEM-calculations, a "too large" wooden blank was worked down in steps to measure the frequency for perturbations shifts corresponding to those of the numerical experiments.

2. FEM-Analysis of Orthotropic Shells

An orthotropic shell element allowing variable thickness was introduced in a FEM-program, called FEMP (Nilsson & Oldenburg, 1983).

An orthotropic material such as wood can be described as a stratified or transversely isotropic material in which a rotational symmetry exists within the plane of strata. The plane of the strata is fixed relative to a universal coordinate system relative to which also the nodes of the triangular shell element are defined with their x-,y-,z-coordinates. The shell thickness in each node is, however, given in the local element coordinate system, that is measured in right angle to the surface of the shell. This means that it resembles the situation when a violin top plate is carved out of a plate made out of orthotropic material, see Fig. 1. Plane stress conditions on the surface of the (thin) shell element are assumed. This assumption reduces the number of necessary material parameters to four:

- Young's modulus along and across the strata, E_l and E_t, respectively.
- The shear modulus across the strata, G_t, and
- Poisson's number ν_t for the contraction across the strata per unit extension along the strata.

That is, index l (longitudinally) is associated with the behavior in the plane of the strata and index t (transversally) with a direction normal to this.

It can always be questioned if plates of wood really can be described as an orthotropic material and, of course, this is an assumption that is not perfectly true but it is a promising simplification (Barducci & Pasqualini, 1975; Kollman, 1951). In this work, large scale variations in different parts of the tree were not taken into account. It was assumed that annual rings were flat and equidistant and that there were no inhomogeneous parts.

Calculations and experiments gave results that closely agree. Thus, the mentioned uncertainties seemed to be of minor importance.

3. Numerical Experiments on Wooden Plates

A FE-model of the plate, see Fig. 1, was made. The model consisted of 192 elements and was also tested with twice as many elements. The same material parameters as obtained from calculations on the bars, i.e., E_l = 14500 MPa, E_t = 1020 MPa, G_t = 1800 MPa, and ν_t = .02 were assumed. For simplicity, only the thickness was varied with a flat medium surface on the FE-model of the blank. Later control calculations proved that this simplification gave negligible influence compared to the shape of Fig. 1.

The first three normal modes of vibration were calculated for the "free-free" plate both in shapes and resonance frequencies. The results were compared to experimental ones, and it was found that the shape of the modes were already almost satisfactory but the frequencies differed, especially in the first mode (cf., Fig. 3).

To improve the knowledge of how different parameters affect the modes of vibration, two series of numerical experiments were made. In the first series either of E_l, E_t, and G_t was changed 10% or ν_t 50% separately while all other parameters were held at the original value. All geometrical factors were held constant. The results are presented in Fig. 2 and Table I. From the figure it is clear that the first mode depends strongly upon G_t, the second upon E_l, and the third upon E_t. For an isotropic plate the resonance frequencies are proportional to the square root of the modulus of elasticity which means that a 10% change in modulus results in a 5% change in frequency. In Fig. 2, on the right hand side, it can be seen that in

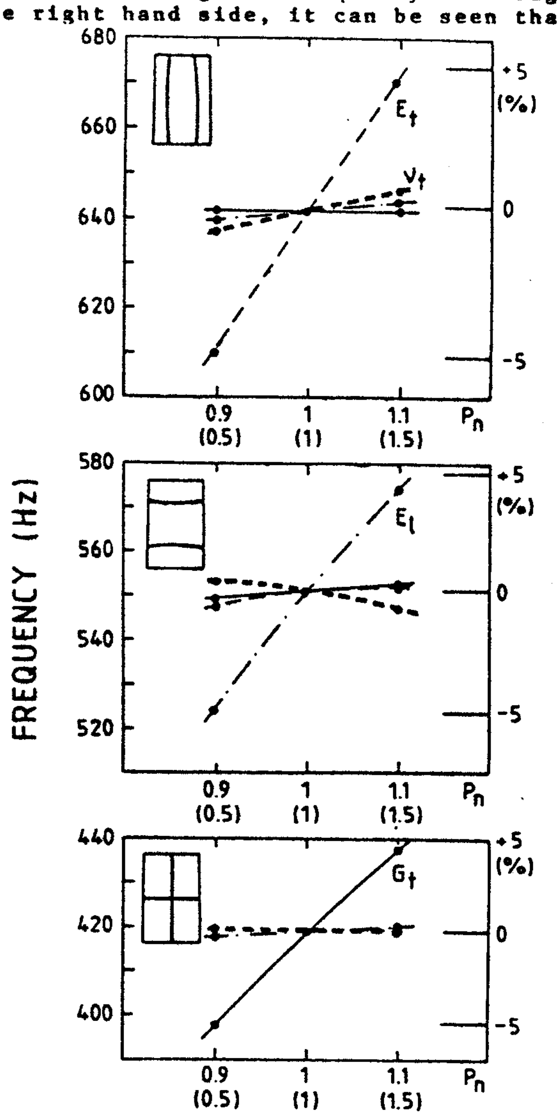

Fig. 2. Numerical experiments with material parameters: calculated frequency shifts of the three normal modes for 10% shifts in E_l, E_t, and G_t, and 50% shifts in ν_t, respectively. Relative frequency shifts in % are marked along the right hand vertical scale and the normalized material parameters P_n along the horizontal axis.

Table I: Measured and calculated frequencies in Hz for one blank.

Normal case:	Mode no 1	Mode no 2	Mode no 3
Measured	272	562	631
First calculation	419	551	642
E_l -10%/+10%	418/420	525/574	640/644
E_t -10%/+10%	419/419	548/552	611/672
G_t -10%/+10%	398/438	549/552	642/642
Poisson's number -50%/+50%	418/420	525/574	640/644
Length -10%/+10%	468/379	630/459	688/639
Width -10%/+10%	464/381	557/518	788/558
Edge thickness -10%/+10%	416/423	541/562	642/644
Center planed down 2 mm	409	539	628
Normal case:			
Final calculation	277	533	644

this anisotropic case the dependence for a 10% change is almost 5% but this is now only true for G_t and mode one, E_l and mode 2, and E_t and mode 3. This justifies fairly large extrapolations for each mode and its corresponding parameter. Thus, if the resonance frequencies are known from the experiments, E_l, E_t, and G_t can easily be determined by interpolation using a standard size blank.

Examples of modal shapes correcponding to the parameter changes in Fig. 2 are shown in Figs. 3 and 4. The first three modes of vibration are shown, and it is obvious that the first mode is best described by torsion and the second and third as bending about orthogonal axes. This fact also explains the strong dependence of G_t, E_l, and E_t, respectively. This is also independently discussed (Caldersmith, 1984; McIntyre & Woodhouse, 1984,1985;1986) and confirms our results. The nodal lines from the experiments with real plates are indicated in Fig. 3. The nodal lines from the calculations are somewhat straighter than the experimental ones for modes no. 2 and no. 3. But as can be seen in Fig. 2, a certain (small) amount of influence is given by, for instance, E_t and G_t in mode no. 2, that is, the dependence on E_l is not total.

The shape of the first mode was little affected by changing parameters. This justifies rather large extrapolations to change the frequency by changing G_t, cf., Fig. 2.

The shapes of modes no. 2 and no. 3 were somewhat more sensitive to changes in material parameters. An increase of E_l, a decrease of E_t, and an increase of ν_t gave nodal lines for modes no. 2 and no. 3 that were slightly more curved, i.e., closer to the experimental ones. The shifts in modal shapes are similar and are well represented by those resulting from shifts in E_l, cf., Fig. 4. A higher numerical value for E_l and E_t raises the frequency of modes no. 2 and no.3, while a higher value of ν_t gives a decrease in frequency for mode no.2 and an increase for mode no.3. This last effect may arise from the fact that ν_t couples the deformation in orthogonal directions to each other. A

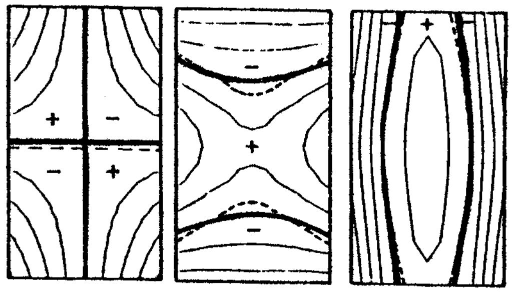

Fig. 3. Numerical experiments: vibrations of the first three normal modes with the initial values for the material parameters. Thick lines represent calculated nodal lines and broken lines measured. Thin lines represent equidistant (displacement) amplitude lines. Plus and minus signs mark the phase of the vibrations. Calculated frequencies are 419, 551, and 642Hz, and measured 272, 562, and 631Hz respectively.

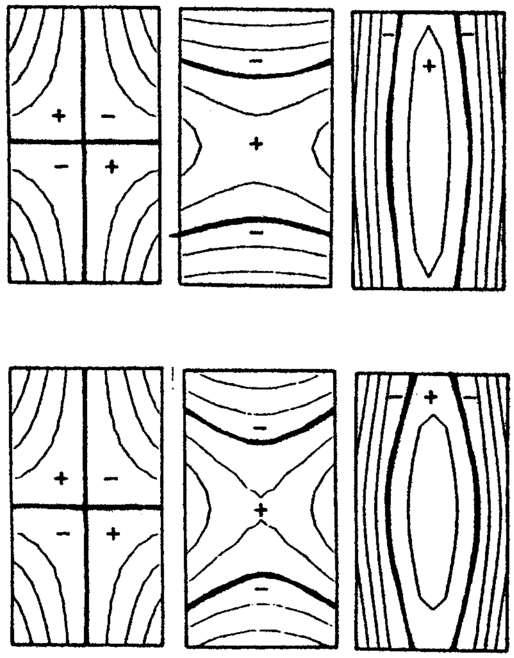

Fig. 4. Numerical experiments - as Fig. 3 but with E_l decreased/increased 10%. Resulting frequencies are 418/420, 525/574, and 640/644Hz respectively.

close examination of the shape of the modes of vibration revealed that if ν_t is increased by 50% then the nodal lines were more curved, in better correspondence with the experimental ones. This indicates that a too small value for ν_t was used in the calculations. The relation between Poisson's numbers in different directions for wood is given (Kollman, 1951) by

$$E_t \cdot \nu_l = E_l \cdot \nu_t$$

The reference also supports our belief that we are using a too small value of ν_t.

In the second series of numerical experiments the length L, the width B, and the thickness of the plate hh at the edge were changed with 10%, and the middle (nh) was planed down 2 mm (i.e., 10% of hm), see Fig. 1, one at a time keeping all other parameters in-

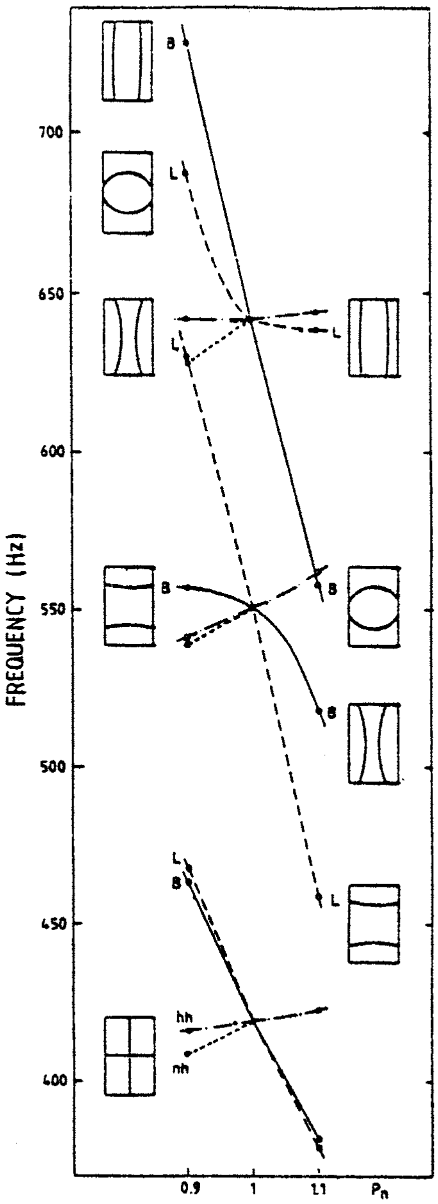

Fig. 5. Numerical experiments with geometrical parameters: calculated frequency shifts for 10% shifts in L, B, and hh, and nh increased to 2 mm (reduced thickness). P_n along the horizontal axis marks normalized geometrical parameters.

cluding the material ones at their nominal values. In Fig. 5 the first three modes are shown and the influence that each geometrical factor has on the frequency. Examples of corresponding mode shapes are shown in Fig. 6. The frequency shifts were accurately verified in experiments made after the FEM-calculations.

An increase of the plate edge thickness increases the resonance frequencies and a cutting down nh decreases them slightly but less than increasing B and L. Increasing B and L will decrease the resonance frequencies. The frequency decrease is especially large for the second mode (for L) and for the third mode (for B). The first mode is affected about the same amount in frequency by both L and B.

Changes in length L and width B can dramatically change modes no. 2 and no. 3 while the shape of mode no. 1 was almost not affected at all. The changes in mode shapes for the changes

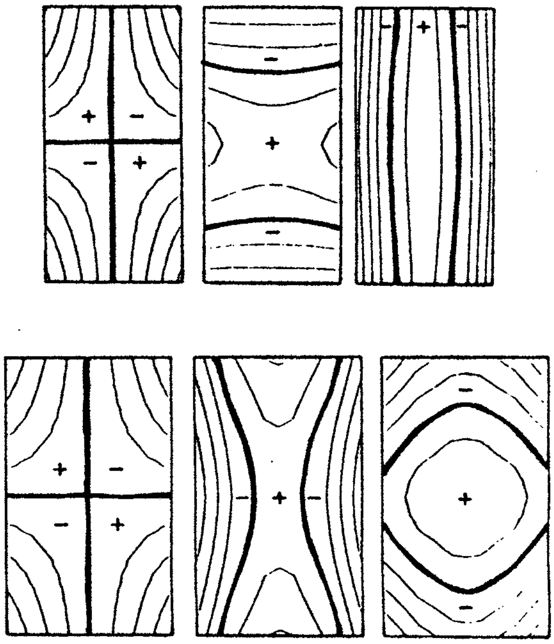

Fig. 6. Numerical experiments - as Fig. 3 but B decreased/increased 10%. Resulting frequencies are 468/379, 630/459, and 688/639Hz respectively.

in B are shown in Fig. 6. The same changes but reversed were found for the changes in L. The changes in mode shapes were negligible for the changes in hh and nh. A decrease of length L and an increase of width B(B < L) interchanged the order of modes no. 2 and no. 3, and the third modes were not simple one-axial bending but "ring-modes" with nodal lines forming an almost closed path within the plate. An increase in L and a decrease in B gave an even more pronounced behavior with orthogonal and "straight" nodal lines in modes no. 2 and no. 3 and kept of the order of appearance.

This strong influence of geometry shows that it is essential to choose a specific geometry if it is wanted to determine material parameters from vibration modes in a plate. The geometry of our plate, see Fig. 1, seems to be a reasonable choice. Geometrical measures can also be determined in quite an accurate way.

Care must be taken not to extrapolate the curves in Fig. 5 too much and it should be remembered that the calculations are true only for values close to the nominal ones. However, since a 5% increase in B or L corresponds to a roughly -10% change in G_t for the first mode, it is clear that the geometrical dimensions of the plate must be known quite accurately to allow us to draw conclusions about material parameters.

4. Measurement of Shear Modulus

The shear modulus used in the calculations turned out to be suspiciously high. Comparisons with published data (Haines, 1979; Beldie, 1969) confirmed this. Therefore, the frequency of the first torsional mode of the ("longitudinal") test bar was measured in the following way.

The bar was clamped vertically in a vice and loaded at its free end by two horizontal parallel bars made out of brass. At the end of one of the bars, the accelerometer with magnet was fastened and the driving coil was placed just outside it. The same measurements as for the input admittance curves were made. The loading and the length of the test bar were varied. As the shear modulus is closely the same for "longitudinal radial" and "longitudinal tangential", it is justified to calculate the shear modulus from these measurements.

Using the simplest possible theory assuming pure torsion in a rectangular bar, the shear modulus is calculated as if the material is isotropic:

$$G = \frac{LJ_0}{K} (2\pi f)^2$$

where

G is the shear modulus
L is the length of the torsion bar
J_0 is the moment of inertia clamped at the bar end.

$$K = 0.77 \quad \frac{ab^3}{3} \;;\; \frac{a}{b} = 2.81$$

a,b are the two widths of the bar.

Measurements with three different L:s and three different J_0:s gave

$$G = 750 \pm 100 \text{ MPa}$$

This value will be used as a value for G_t in the following calculations. It agrees with the extrapolated value from Fig. 2 and mode no. 1.

5. Final FEM-modeling

The experience and results shown gave a new set of material parameters for the spruce plate: E_l = 15500 MPa, E_t = 1020 MPa, G_t= 750 MPa, ν_t = 0.02. The mode shapes stayed as in Fig. 3, i.e., were close to those measured. The frequencies were shifted to be quite close to the experimental ones and can of course be brought even closer. One such change would be to try ν_t = .03.

6. A Practical Test for the violinmaker

The physical and the numerical experiments provide the fundamentals for a practical and simple test.

It was shown that the position of the nodal lines of the first mode is not affected by varying the different parameters. Thus, by

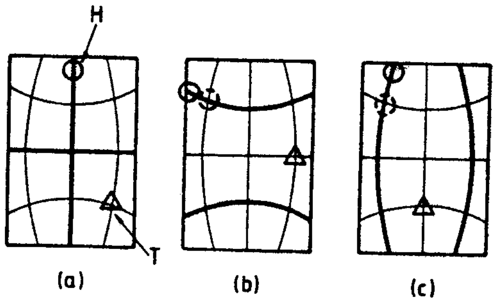

Fig. 7. Tap tone tests with wooden blanks of standard measures (cf., Fig. 1): optimum holding points (circles) and tapping points (triangles) and safe initial holding points for modes no. 2 and no. 3 (circle with a broken line). Thick lines mark nodal lines of the normal mode investigated: (a) mode no. 1, (b) mode no. 2, and (c) mode no. 3.

holding and tapping as in Fig. 7a, the first normal mode is set into vibration and the corresponding eigentone can be heard. The four points where the nodal lines of the second and third modes intersect, that is, at 26% of the width and 23% of the length, were largely independent of geometry and material properties. By holding at a point where the nodal lines intersect and tapping at optimum points as marked in Figs. 7b and 7c, the eigentones of the second and third modes can be heard. The two tones can be further separated by suppressing one mode at a time by seeking the point where the nodal line of the normal mode studied cuts the plate edge. The marked holding and tapping positions also suppress higher modes that can interfere (cf., Ek & Jansson, 1985).

The geometrical variations modeled were large. In practise they can and should be kept small, i.e., less than 1 mm for L and B and 0.5 mm for hh and hm. Thereby, the uncertainties in tap tone frequencies will be less than 2%.

The measured material parameters are for G_t, E_l, and E_t equal to 840 MPa ± 26%, 15000 MPa ± 14%, 760 MPa ± 34%, respectively (Haines, 1979). The corresponding resonance frequency shifts are small (≤ 1%) except for the shifts of the single main parameter of each mode.

The geometrical measures are close to a suggested standard for blanks. The blank with minimum hh corresponds almost exactly (Mattsson, 1984). From recorded frequencies of the three tap tones, the material parameters are obtained as a constant multiplied by the frequency squared and by the mass. The constants become for G_t, E_l, and E_t equal to 17, 89 and 4.4 kPa /kg·Hz^2, respectively. These constants are calculated from numerical results, shown in Table I. The corresponding tap tones from a later investigation (Jansson et al., 1984) were 294 ± 23Hz (tone D), 544 ± 35Hz (tone C-sharp), and 648 ± 37Hz (tone E), respectively.

It is thus suggested that the violin maker carefully plans his blanks to the given dimensions, measures the mass (the weight), and records the tap tones. The mass and the tap tones can be noted for later references, and can also be used to calculate the elasticity parameters.

7. Conclusions

In this paper it is shown that FEM-calculations can be brought in close agreement with experimentally obtained values for a shell made of an orthotropic material such as wood. This result can be used in several ways. First, it can be used to determine material parameters of a plate with a reasonable choice of geometry. The first three vibration modes depend strongly on the shear modulus, the "longitudinal" elasticity modulus, and the "transversal" elasticity modulus, respectively. The measurement of frequencies and mode shapes is simple to do and it can be proposed as a method for the modern industry to use, especially as the measurements of the three main parameters can be made with a plate in one setup, instead of in two or more steps when test bars are used. In many applications it can be used as a nondestructive test method of blanks, which later are adjusted into preselected dynamic or static properties. It may be especially useful when using a complex material where the dynamic properties may differ from the static ones, and the finished item is to be used with dynamic loading. It represents a simple nondestructive way to test wood properties of blanks to music instruments such as violins where the traditional way with test bars is difficult to carry out with high accuracy.

In conclusion we find that the FEMP-calculations have given more answers than we sought in the first step. In a just finished investigation we have calculated the influence of material, parameter, thickness, thickness distribution, and arch height (Molin, Lindgren & Jansson, 1986).

Acknowledgments

This paper is based on ideas of Erik Jansson and Nils-Erik Molin. The FEM-modeling and calculation were made by Mats Tinnsten and Ulf Wiklund for a Master's Degree in Mechanical Engineering at the Luleå University of Technology. Their diploma work was carried out under the supervision of Nils-Erik Molin and Lars-Gunnar Nilsson. The acoustical measurements and other experiments were done by Jansson. Nils-Erik Molin and Erik Jansson have written the present report.

We gratefully acknowledge the fruitful cooperation with Stig-Roland Åström, Brännberg, Saw Mill, Boden.

The work was supported by a grant from the Swedish Natural Science Research Council.

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