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NUMERICAL ANALYSIS OF ACOUSTIC GUITARS SOUNDBOARDS WITH DIFFERENT FAN BRACINGS

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Abstract. *The present work is part of a larger project which aims the numerical analysis of musical instruments. In order to identify objective features that give rise to subjective quality and psychoacoustic concepts of the acoustic instruments, a numerical analysis is applied to the mechanical system represented by the classical acoustic guitar. Among the objectives is the identification and analysis of the influence of different materials, construction methods and some finishing details. In this work, the numerical analysis of vibration behavior of a handmade classical guitar was performed emphasizing the vibration modes of the soundboard regarding two different bracing patterns. With help of Finite Element Method, two soundboards with different fan bracing designs were analyzed. The dimensions of the soundboards (top plate) correspond to those of a Hauser classical guitar. The results show a significant variation on some of mode shapes and modal frequencies due to differences in soundboard stiffness.*

Keywords: *Classical Guitar, Fan Bracing, Modal Analysis, Numerical Modeling, Finite Elements*

1. INTRODUCTION

In the string instruments history, Antônio Torres (1817-1892) generally gets the credit for being the father of the modern classical guitar. His instruments incorporated incremental improvements over those of his predecessors and included design features that had later on been adopted by others luthiers (French, 2009), as seen in the projects of H. Hauser (1882-1952), renowned German luthier. Some Hauser's projects also acquired admirers worldwide, being among the most reproduced currently. One of these projects was chosen for this research.

Recently one observes a growing interest on the scientific study of the classical guitar (Richardson, 2010; Elejabarrieta, et al., 2000, 2002a, 2002b; French, 2009; etc.) as a vibration mechanical system. One of the reasons is the possibility of substitution of traditional wood species used in the construction of guitars, some of them obtained from rain forests and in extinction process as the Brazilian rosewood (*Dalbergia nigra*). There would be also economic reasons, especially with regard to the high cost of some traditional wood species. Other wood species could be used in the construction of high quality string instruments since they fulfill some characteristics that influence their acoustic behavior. It is then necessary to quantify the subjective "quality" of an acoustic guitar.

The traditional acoustic guitar has two main parts: the body and the neck. The body forms the guitar acoustic system that gives "color" to the sound through the selective transmission of harmonic frequencies generated by the guitar strings to the air in the environment around the instrument, transforming the kinetic energy into acoustic energy of surrounding air, which gives the perception of the produced sound. The parts of a guitar body are the soundboard or top plate, the back plate and the sides. Most of the sound is produced by the soundboard, which is forced by the oscillating tension of strings, and in a lesser extent by the back plate. The sides or ribs of the guitar act as fairly rigid supports for the top and back plates, so they contribute less to the radiated sound (Wright, 1996). Thus, as the most important part for the sound radiation, the soundboard is the study object of this work. The chosen material for soundboards construction must show resistance to withstand the stresses applied by the string and at the same time must be flexible to vibrate and transmit the sound. In addition, the thickness must be small enough to allow soundboard oscillations. Struts are added bellow the soundboard to increase resistance without increasing very much the mass (WRIGHT, 1996). Soundboard can be divided in three regions: upper bout, lower bout and waist (figure 1). The lower

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bout receives the fan bracing that also acts as reinforcement, but has equalization as its principal function, according some luthiers.

2. OBJECTIVES

This work inspects the effect of two different fan bracing patterns on the acoustic behavior of a soundboard. One develops a modal analysis with the help of Ansys® software (Mechanical APDL 14.0) based on Finite Elements Method. The modal analysis is a process by which the vibration of a structure can be described in terms of its natural frequencies, smoothing factor e modal shapes. These natural frequencies and their respective modal shapes are inherent to each kind of structure and depend on its inertia and stiffness; the frequency is represented by a real number measured in Hertz (Hz). The modal analysis provides the natural frequencies and modal shapes of vibration of a system associated to the structure movement under free vibration. Two fan bracing patterns were investigated: the Hauser's Project, above mentioned, from 1937, and the Project of the Brazilian luthier Maurício Barros, that follows the Hauser's Project to the soundboard plate and struts, but include a different fan bracing (sketches can be viewed in Fig. 1). The necessary data to the modeling was extracted from plan provided for Barros and from copy of Hauser's guitar plan drawn by luthier R. E. Bruné, in 2003. The wood was modeled as orthotropic material, which properties correspond to a general specie of spruce. Spruce is a tree of the genus *Picea* of coniferous evergreen trees of the family *Pinaceae*, found in the northern temperate and boreal regions, and it is traditionally used in the construction of classical guitar soundboards. Fig. 1 provides a bi-dimensional view of the two considered soundboards plans.

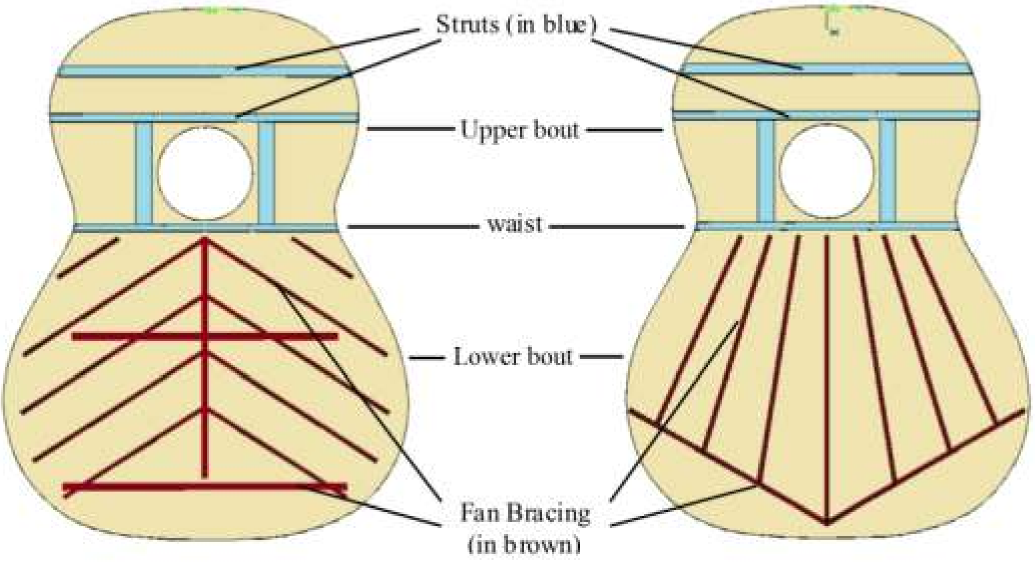


Figure 1 – Barros soundboard (left), Hauser soundboard (right) and their features.

3. METHODOLOGY

Two soundboards based on two different fan bracing designs were modeled in Ansys. It can be assumed that soundboard behaves as a linear system because the vibration amplitudes are sufficiently small (Wright, 1996). It was also assumed that there is no damping in the system. The geometry construction was the first step to perform the computational modeling in Ansys. For this construction, the dimensions of soundboard components have been derived from the Hauser and Barros project plans. The plate geometry was created by "keypoints" (with the coordinates of soundboard contour), "splines" (created using the keypoints) and areas (created from splines); the command EXTRUDE AREA was used to created the plate volume; a constant and uniform thickness of 0.0026m was considered in the plate for both soundboards. Struts and fan bracings were modeled as blocks, taking into account the dimensions and position measured in soundboard plans. The command GLUE was used to merge shared areas to represent the coupling between all soundboard components. In order to simulate a natural behavior, an orthotropic material was chosen. The values of material properties are shown in Tab. 1**.**

In a real instrument, struts and fan bracing parts are cut taking into account their fibers direction, but they are arranged on the plate not necessarily aligned with the fibers of soundboard plate. These factors influence greatly the structure behavior and, to include these features, Local Coordinate Systems – LCS were created to each direction, each to one different from the Global Coordinate System direction. These LCS's were related to each component accordingly to the desired behavior.

Table 1- Values to orthotropic material properties [Ref.: Parametrical 3D Structural Co-modelling of Stringed Instruments, Enrico Ravina].

Parameter	Value (unit)
Elast. Mod. DirX (EX)	1.3e10 (Pa)
Elast. Mod DirY (EY)	8.9e8 (Pa)
Elast. Mod DirZ (EZ)	6.49e8 (Pa)
Poisson's Ratio XY (PRXY)	0.2
Poisson's Ratio YZ (PRYZ)	0.4
Poisson's Ratio XZ (PRXZ)	0.4
Rigidity Mod. XY (GXY)	1.015e9 (Pa)
Rigidity Mod. XY (GXY)	4.16e8 (Pa)
Rigidity Mod. XZ (GXZ)	7.15e8 (Pa)
Specific Mass (DENS)	400 (kg/m^3)

The element SOLID187 was chosen to build the whole mesh. It is a tetrahedral element, with 10 nodes (4 at the corners and 6 at the midpoints of the edges) and 3 degrees of freedom at each node. This element has a quadratic displacement behavior and is well suited to modeling irregular meshes and for the use with orthotropic and anisotropic materials (Feijó, 2007). Displacement restrictions have been applied in the total lateral area of the plate to represent the real restriction that soundboard receives. The SMART SIZE command was used at different element sizes and discretization levels. A Modal Analysis was performed with the option for the Block Lanczos Extraction Method, the standard method to Modal Analysis in Ansys, requesting the first 15 vibration modes and frequencies of the structures.

Mesh independent results were obtained at the level 3, being the largest element size equal to 0.004 m, for both soundboards, and the difference to a higher refinement being in order of hundredths. The mesh of Hauser Soundboard is shown in Fig. 2. The number of elements was 79.411 for Barros plan and 77.557 for Hauser's plan.

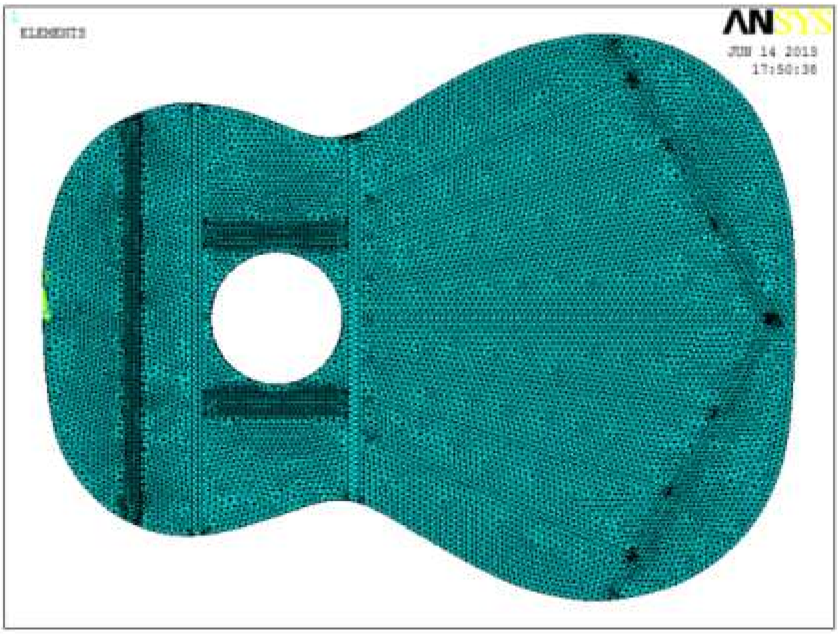
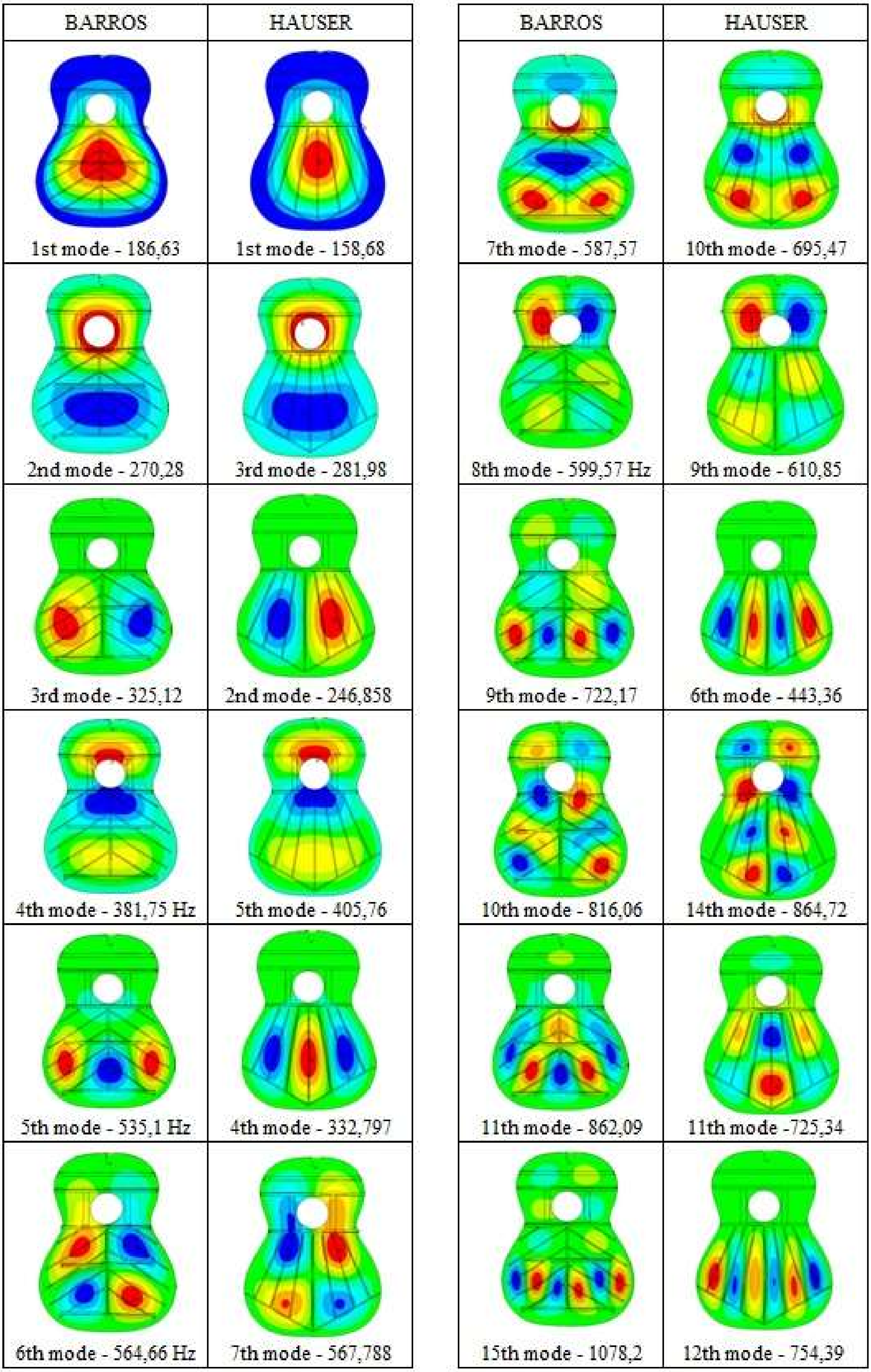


Figure 2. Mesh of Hauser soundboard.

4. RESULTS AND DISCUSSIONS

Vibration modes are presented in Fig. 3 and Fig. 4. In Fig. 3, modal shapes found to be similar for both bracing patterns are shown. Barros soundboard vibration modes were positioned in first column in ascending order of frequency and Hauser soundboard vibration modes, in the second column, according to the similarity presented. In Fig. 4 one presents the modal shapes without similarity among the two studied cases. The obtained natural frequencies are shown in the Tab. 2 in ascending order.



BARROS	HAUSER	BARROS	HAUSER
1st mode - 186,63	1st mode - 158,68	7th mode - 587,57	10th mode - 695,47
2nd mode - 270,28	3rd mode - 281,98	8th mode - 599,57 Hz	9th mode - 610,85
3rd mode - 325,12	2nd mode - 246,858	9th mode - 722,17	6th mode - 443,36
4th mode - 381,75 Hz	5th mode - 405,76	10th mode - 816,06	14th mode - 864,72
5th mode - 535,1 Hz	4th mode - 332,797	11th mode - 862,09	11th mode -725,34
6th mode - 564,66 Hz	7th mode - 567,788	15th mode - 1078,2	12th mode - 754,39

Figure 3. Similar Mode Shapes of Barros and Hauser soundboards and respective frequencies in Hertz(Hz).

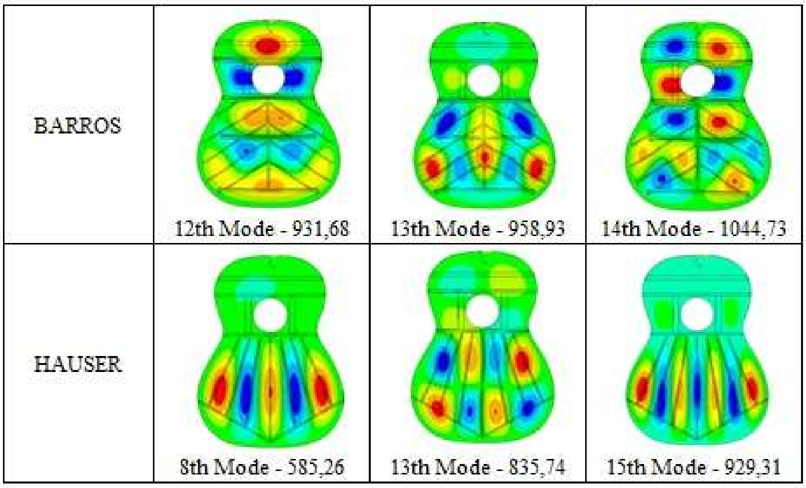


Figure 4. Mode shapes with less similarities between Barros and Hauser soundboards.

Table 2. Natural Frequencies to soundboards of Barros and Hauser in Hertz (Hz).

BARROS	HAUSER
186.63	158.68
270.28	246.86
325.12	281.98
381.75	332.80
535.10	405.76
564.66	443.36
587.57	567.79
599.57	585.26
722.17	610.85
816.09	695.47
862.09	725.34
931.68	754.39
958.93	835.74
1044.7	864.73
1078.2	929.31

The first natural frequencies obtained in these work for the two fan bracing patterns are in the range of those found by Richardson (2010), who experimentally investigated the first vibration modes of a Torres guitar using holographic interferometry as analysis tool and provided results about soundboard frequencies taking into account the resonance box. Elejabarrieta, et al. (2002b) also simulated an acoustic guitar based on a Torres plan, being the lowest natural frequency obtained for the soundboard by this researcher equal to 139 Hz. Although these two analyses differ from present work on dimensions, materials properties and details of fan bracing patterns, the obtained modal shapes and natural frequencies are similar and typical for a classical acoustic guitar. According Elejabarrieta, et al. (2002b), when one considers the air contained in the box, as in the Richardson's research, the natural frequencies decrease in comparison with those of the structural modes because the air acts as an added mass in the box. Results found by Richardson are shown in Fig. 5 and results found by Elejabarrieta, et al. can be viewed in Fig. 6.

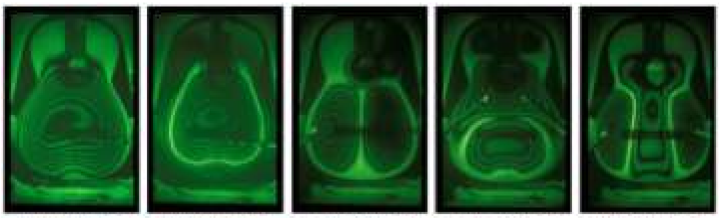
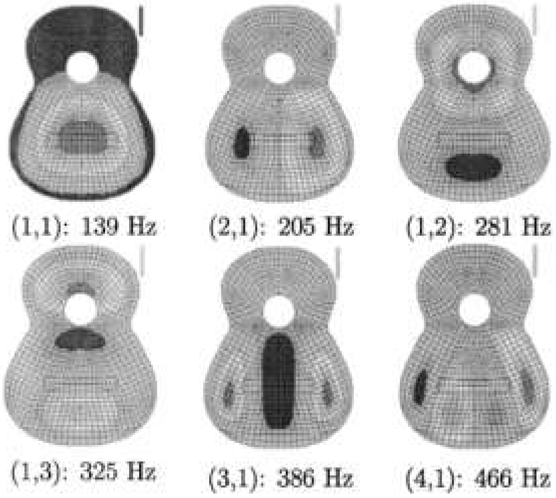

(a) 103 Hz (b) 215 Hz (c) 268 Hz (d) 436 Hz (f) 553 Hz

Figure 5. Modes of a conventional Torres guitar found by Richardson (2010), using holographic interferometry.



(1,1): 139 Hz (2,1): 205 Hz (1,2): 281 Hz

(1,3): 325 Hz (3,1): 386 Hz (4,1): 466 Hz

Figure 6. Modes of vibration and natural frequencies of the soundboard in hinged boundary conditions found by Elejabarrieta, et al. (2002b).

Ansys provides, besides frequencies and modal shapes, values to total mass (in kilograms), center of mass coordinates (in meters) and moments of inertia about origin and about center of mass (in kg.m²). The Tab. 3 shows these results.

Table 3. Total mass, center of mass and moments of inertia about origin and about the center of mass.

BARROS	HAUSER
Total Mass = 0.17510	Total Mass = 0.16764
Center of Mass	Center of Mass
Xc = 0.25529	Xc = 0.25092
Yc = 0.24685e-03	Yc = 0.27458e-03
Zc = -0.81357e-04	Zc = -0.29351e-03
Mom. of Inertia about Origin	Mom. of Inertia about Origin
Ixx = 0.1296e-02	Ixx = 0.1256e-02
Iyy = 0.1446e-01	Iyy = 0.1350e-01
Izz = 0.1575e-01	Izz = 0.1475e-01
Ixy = -0.2303e-04	Ixy = -0.2401e-04
Iyz = 0.1097e-06	Iyz = 0.9759e-07
Izx = 0.1359e-04	Izx = 0.2545e-04
Mom. of Inertia about Center of Mass	Mom. of Inertia about Center of Mass
Ixx = 0.1296e-02	Ixx = 0.1256e-02
Iyy = 0.3044e-02	Iyy = 0.2941e-02
Izz = 0.4337e-02	Izz = 0.4194e-02
Ixy = -0.1199e-04	Ixy = -0.1246e-04
Iyz = 0.1062e-06	Iyz = 0.8408e-07
Izx = 0.9952e-05	Izx = 0.1311e-04

The total mass presented by the both soundboard are near from that founded by Elejabarrieta, et al. (2000). As shown in Tab. 3, Barros soundboard presented a bigger mass, with a difference of approximately 7 g. The changes in the center of mass position and moments of inertia are small and can be neglected. Taking into account the ascending order of frequencies, to the first 15 vibration modes of the structures, Barros soundboard presented the highest frequencies. The differences among them are in a range of 14,31 Hz to 179,97 Hz. From the ninth vibration mode, the differences stay over 110 Hz. Taking into account the mode shapes, both Barros and Hauser designs show similarity in 12 of them (Fig. 3). When one compares these similar modes, Hauser Soundboard shows highest frequencies in 6 of them and Barros Soundboard in the 6 remaining modes. On the other hand, Barros soundboard presents the highest frequencies for all the modes without clear similarity (Fig. 4). On Hauser Soundboard, some vibration modes are more characteristically located in lower bout. The first vibration mode that corresponds to fundamental structure mode present one antinode in lower bout; to this mode, Barros soundboard vibrates at frequency 18% higher.

5. CONCLUSIONS

The finite element method was successfully applied to modal analysis of two different acoustic guitar soundboard designs. The differences in two fan bracing patterns gave rise to significant differences in natural frequencies and in shape modes. Quantitative and qualitative results are in the range and are similar to those obtained by Richardson (2010), experimentally, and the simulation results of Elejabarrieta, et al. (2002b). Natural frequencies of Barros Soundboard design were generally higher than Hauser design. One reason for that could be a higher stiffness of Barros design, which shows additional struts in lower bout. These additional struts are not as thick as the struts in upper bout, but they are clearly more rigid than the bracing in this region of soundboard. When one considers the similar mode shapes presented, it can be seen that Hauser soundboard presents frequencies slightly higher for some vibration modes and the Barros soundboard presents higher frequencies for the remaining modes; but in general, the highest frequencies showed by Barros soundboard are more significant. It can be pointed out that some vibration modes did not present very clear similarities. Here, on Hauser Soundboard, it can be viewed a concentration of vibrations in lower bout. These general changes on shape modes due to fan design gives rise to questions about how each fan bracing pattern may affect the forced response, when the soundboards are under the stress imposed by the strings. The herein presented results must be followed by further analyses in order to link the natural acoustic responses of a soundboard to the musical performance of the instrument. This shall be the object of further research.

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