In This Issue:

- Thickness Graduation Mapping: Methods & Goals
- Bowing with a Glass Bow: An Update
- Finite Element Analysis of a Violin Corpus
- Violin Mode Relationships in Free Plates, After Attachment to the Ribs and in the Finished Instruments
- Selected Extended Abstracts from the International Symposium on Musical Acoustics (ISMA) 2001, Perugia, Italy

Catgut Acoustical Society

To increase and diffuse the knowledge of musical acoustics and to promote construction of fine stringed instruments

Vol. 4, No. 4 (Series II) November 2001
In this issue we feature articles that should further our understanding and appreciation of bow/string interactions, the inner workings of the violin and the classical guitar, plate tuning, graduation systems, and tools for acoustical measurements. In addition we introduce some new features.

We begin with a description of an ambitious project to compile, map, and statistically analyze thickness graduation data. Next, we have an exceptionally clear and thoughtful discussion of the physical dynamics of bowing a violin string by Robert T. Schumacher. This subject is more complicated than it is often portrayed, and Schumacher helps to clear up some commonly held misconceptions about what happens when bow touches string. Instrument makers should take interest in a major new work that represents several years of research by Oliver Rodgers and Pamela Anderson. These authors have used finite element models to measure the effects of slight changes in arching height, wood thickness, bridge character, soundpost location, bassbar height, and other factors. Results show that modifying certain factors may have much more influence on tonal quality than others. For plate tuning enthusiasts, Robert Wilkins describes a method for using the shaker table to take plate tuning one step further. In addition, we are pleased to have the opportunity to publish a sampling of extended abstracts from the International Symposium on Musical Acoustics (ISMA), held in Perugia, Italy in Sept. 2001. These papers describe “cutting-edge” research on important subjects.

We introduce a new section entitled “Questions and Answers” to help provide feedback to readers’ questions. This issue’s question is concerned with finding specific sources for synthetic materials for musical instruments. In addition, we introduce features on interesting articles in other journals, and interesting internet pages. The internet has become a major resource for makers, researchers, and enthusiasts, and we thank Kelvin Scott for compiling some web links that may be of interest to our readers.

We are continually striving to improve the journal. Please send us your comments and suggestions.

Good reading!

Jeffrey S. Loen

The CAS Journal is published twice a year by the Catgut Acoustical Society, Inc. a non-profit organization which aims to increase and diffuse knowledge of musical acoustics and to promote the construction of fine stringed instruments.

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55 Park Street
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Phone: (973) 744-0371
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E-mail: catgutas@msn.com
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The Catgut Acoustical Society is known for fostering pioneer research in musical acoustics and the application of these principles to the making of fine stringed instruments. To fulfill its mission, the Society supports publications, meetings for researchers and makers, musical compositions, lectures, and concerts.

The Catgut Acoustical Society Journal (ISSN 0882-2212) is published semi-annually by the Catgut Acoustical Society, Inc., 55 Park Street, Montclair, New Jersey 07042. Neither the Society nor the Journal's editorial staff is responsible for facts and opinions expressed in articles or other materials contained in the Journal.

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Pamela J. Anderson is a violin maker who restores and repairs instruments at Wm. Moennig & Son in Philadelphia, PA. She is a graduate of Bryn Mawr College with a BA in Philosophy, studied cello with Francis de Pasquale of the Philadelphia Orchestra, and in 1978 graduated from the International School of Violin Making in Cremona, Italy. During the past several years she has collaborated with Oliver Rodgers on acoustical experiments. Ms. Anderson is on the Catgut Acoustical Society Board of Trustees.

Jeff Loen is an earth scientist who also makes and repairs violins. Since earning his Ph.D. from Colorado State University in 1990 he has worked mainly as a mapping professional for State and Federal agencies, using geographic information systems (GIS) to display and analyze data about natural resources and geologic hazards. He also serves as Editor of the CAS Journal.

Oliver E. Rodgers is a mechanical engineer and violin player who upon retirement became fascinated with the intricacies of making and technically understanding violins. He has been a Catgut Acoustical Society Board member for many years and is a frequent contributor to the CAS Journal. The analytical work of the present paper was done using the computer system of the Mechanical Engineering Department at the University of Delaware during the time when the senior author had access to their facilities.

Bob Schumacher is a retired professor of physics at Carnegie Mellon University. Prior to his conversion to musical acoustics research in the 1970's he did research on various condensed matter projects using the tools of magnetic resonance. His musical acoustics interests have been mainly on the oscillations of musical instruments with emphasis on stringed instruments. In the work described in his "update" he is using his knowledge of the bowed string to investigate problems in the fundamentals of friction. He hopes at some point to return to questions more directly related to the violin family.

Robert A. Wilkins is a retired academic with a background in electrical mechanics, philosophy of science, and education. A long time first violinist in two non-professional orchestras, he became actively interested in violin making in 1985 out of a desire to make a good violin for himself, a good cello for his wife, Cathryn, and to satisfy his desire to understand the physics of the violin. To date he has made 25 violins, six cellos, and one viola using the Hutchins bi-tri octave plate tuning method.
Dr. Herman Medwin Receives ASA Gold Medal

In June, 2001, CAS member Dr. Herman Medwin was awarded the Gold Medal at the Acoustical Society of America meeting in Chicago, Ill. Each spring the Gold Medal is presented for contributions to acoustics. The first Gold Medal was presented in 1954, and biennially until 1981. It is now an annual award.

Hank Medwin was born in Springfield, Massachusetts. In 1941 he received his B.S. in Physics from Worcester Polytechnic Institute, and he later received a doctorate in acoustics from the University of California at Los Angeles. In the 1950's he began working at the Naval Postgraduate School in Monterey, California. His research interests there included the scattering of sound from rough surfaces, and acoustical oceanography. His publications include contributions on the subjects of the sounds made by falling raindrops, studies of bubbles near the sea surface, and the acoustics of auditoriums. Hank's work included studies based both on field measurements at sea, and experiments in laboratories. He was a co-author with Clarence Clay of the popular 1977 text "Acoustical Oceanography", and more recently (1997) "Fundamentals of Acoustical Oceanography". Medwin previously was awarded the ASA's Silver Medal in Acoustical Oceanography, and an honorary doctorate from Worcester Polytechnic Institute. Hank has had a lifelong interest in playing the violin, and has performed as a violinist with the UCLA Symphony Orchestra and the Los Angeles Civic Orchestra.

We applaud Hank Medwin for his many contributions to acoustics, and congratulate him upon receiving this important award.

NVFA Begins Newsletter Series

The New Violin Family Association, Inc. has published a newsletter entitled "The Violin Octet". It features information about publications, highlights about scientific investigations, memoriams, articles about the various members of the Octet by players, scholars, luthiers, and composers, a list of future performances, and information about recordings. We wish the NVFA the best of luck with their new endeavor.

The Violin Octet newsletter is published semiannually by the New Violin Family Association, Inc., 112 Essex Avenue, Montclair, NJ 07042.

Dr. Paul Krikorian

Dr. Paul Krikorian, physician and musician, died on Aug. 8, 2001, at his home in Harding Township, New Jersey. He was 79. His long career intertwined electrical engineering, medicine, and music. His early training was in electrical engineering, and he worked as a department chief at Western Electric. During World War II, despite a deferment, he enlisted in the Navy, and for two years he operated a radio station in the South Pacific that featured classical music, especially that of his favorite composers, Mendelssohn and Rachmaninoff. After the war, Krikorian studied pre-med at St. Peter's College in Jersey City, earned his medical degree from New York University, and opened his medical practice in 1953. His knowledge of electrical engineering, combined with his medical skills, allowed him to develop and patent a device for heart patients that increased blood flow to the heart. In his free time, he played the violin in several local orchestras in New Jersey and also sang in church choirs. Dr. Krikorian had been a member of the CAS since 1975.

CASJ Vol.4, No. 4 (Series II). November 2001
LETTER TO THE EDITOR

Comments on a Paper Entitled “Acoustic Radiation from Bowed Violins”
by Lily Wang and Courtney Burroughs

by Oliver Rodgers

We already know that our ears and brains cannot provide us with the details of the composition of a sound—only its timbre. Hence it was with great interest that I read the summary of Lily Wang’s thesis project [1]. This work produced the first detailed experimental results that measure and display the spacial distribution of sounds when played with a bow from the open G string up to beyond the 9th harmonic. I have known from my reading over the years that the sound distribution around a violin was very complicated, but the figures in this paper forced me to realize both how complicated the distributions can be in the higher harmonics and how impossible it is to describe in an absolute sense how a violin sounds. Results of this paper deserve the careful attention of violin makers.

Experiments were done in an acoustic space. Bowing was done using a continuous horse hair belt driven by an electric motor. The belt bowed open D and A strings and data were recorded on the spacial distribution of sound energy. Intensity fields are presented in a series of illustrations that make it possible for makers to develop some concept of how sounds of various harmonics are distributed. These show how difficult it is to develop a simplified model in our heads to use as we modify instruments to improve the sound quality. The paper forms the following conclusions:

- Most of the sound radiation is generated by the top plate, especially in the C bout region on the sound post side. The sound radiated from the top is stronger than that from the back. However, above 600 Hz the source of sound becomes more localized on the top plate, and the spacial distribution becomes more and more uneven as pitch increases.

- For tones up to about 600 Hz (one and a half octaves above open G string pitch) the spacial distribution of sound energy is the same as that of a point source (uniform in all directions and diminishing with distance from the apparent source, the center of the plate).

- The spatial distribution of sound becomes more complicated as frequencies increase, and each frequency (harmonic) has its own pattern. There may be beams of sound. At higher pitches, there are spaces full of sound and others with little sound energy, the patterns of which change drastically with very small changes in frequency.

- The sound pattern in space of any given note (pitch) is the same, regardless of how it is generated in a particular violin. For example, the unique spacial distribution of the sound field of the 6th harmonic of the A string matches the measured spacial distribution of the 9th harmonic of the D string which is very close in pitch (roughly 2640 Hz.).

The paper offers the following implications for makers and those trying to learn by recording and/or listening to violin sounds:

- The process of sound radiation is so intricate and detailed that it is impossible to duplicate the sounds of a violin being played in recordings.

- The space in which an instrument is played is important in determining sound patterns. What we hear is dependent on where we sit in a room.

- The best we can hope for is to record well enough to permit A-B comparison tests. To help produce usable input data, makers should find players willing to try out instruments to help evaluate the effects of changes.

- The top plate deserves a maker’s primary attention. Don’t spend much time on the back (especially the lower back) except as it might influence how the middle of the instrument sounds (through the sound post).

REFERENCE

Thickn ess Graduation Mapping: Methods and Goals

Jeffrey S. Loen
Kenmore, WA USA
Email: Casjeditor@aol.com

Most makers of violin family instruments hope to emulate the work of famous makers of the past, such as Amati, Stradivari, and del Gesu. For most this is an elusive, perhaps unattainable goal, and many efforts invariably fall short of the mark. This is partly because much of our information about fine old violins is little more than mythology. We lack data on old master instruments, so our questions are answered by opinions and speculations rather than by scientific observation and reason. A handful of modern makers who have had considerable success at reaching the “unattainable” goal share one characteristic—they have studied details of actual classic instruments in a scholarly and often scientific way. Other equally skilled makers lacking such knowledge can make fine quality violins, although it is unlikely that they perform mechanically or acoustically like old master instruments. Likewise, acoustical scientists and other researchers who wish to develop useful results for makers should, if possible, base at least some of their measurements, computer models, and experiments on actual classic instruments [1]. Valuable work along these lines has already been published in CASJ [2,3,4,5,6].

One of the most important measurements on classic instruments is thickness of top and back plates (see ISMA abstract by Loen, this volume). These subtle variations in thickness (along with arching and other factors) influence how plates twist, flex, and cause movements that we finally perceive as sound. Many arrays of thickness data points appear in books, posters, and technical drawings, and an initial database of thickness graduation data was compiled based on published sources. Data from museum files and other unpublished sources have been added and currently the database includes more than 230 plates of violin, viola, and cello.

Once collected, thickness measurements are mapped using a geographic information system (GIS), which allows colored contour maps to be made, plus other types of analysis. Contour mapping of violin graduations is not a new idea [7], although the use of GIS provides a systematic method that is fast, accurate, and reproducible. Besides generating useful visual displays (fig. 1, 2), statistical tests can be applied to the database, currently comprising more than 17,000 measurements, in order to derive general relationships and trends based on large numbers of plates. Please note that the use of statistical methods on large numbers of data points minimizes the effects of

Figure 1. Contour graduation maps of a violin made in 1650 by Nicolò Amati, produced using a GIS system (natural neighbor interpolation; contour interval 0.25 mm). Data collected by A.T. King and G. Frisch using hacklinger gauge (instrument is viewed from outside). Access to instrument courtesy of Division of Cultural History, Smithsonian National Museum of American History. Note high variability and marked asymmetry of top and pronounced concentric pattern and extreme thin spots (min 1.3 mm) on back.
spurious data caused by regraduation, shrinkage, patching, repairs, data collection errors, etc.

Objectives of the project are to develop an analytical and documentation database to help violin makers, restorers, curators, and acoustical scientists answer a variety of questions about fine instruments. Such questions might include the following:

- What are visible differences in graduation between good and bad sounding instruments (fig. 1,2)?
- What are the most common distributions of thickness used, depending on maker, time period, and location?
- What sorts of changes in graduation schemes occur during the working lives of important makers?
- What were the makers trying to achieve structurally when they carved their plates, and how does this affect the formation of eigenmodes?
- Are there “ideal” graduation patterns for tops and backs, or do fine violins display a variety of patterns?
- How do thickness distributions relate to pin holes on backs of Italian violins?
- Do positions of centers of back thickness vary according to maker, and how does this influence tone?
- Can we use statistics to discriminate one maker’s plates from those of another?
- Are highly arched plates graduated differently than plates having low arching?
- Are one-piece (slab or quarter) backs graduated differently than two-piece backs?

GIS can be used, for example, to compute “average thickness” maps, based on dozens of fine instruments, to serve as general guides for plate graduation. In addition, the problem of plate asymmetry can be addressed by comparing thickness on right and left sides of plates. The area, perimeter, volume, and modeled weight of plates can be compared. Results like these should help modern makers to produce results slightly closer to those of famous makers they wish to emulate. In addition, the database can be used as a documentation tool for restorers, historians, and curators, potentially to help identify unattributed or stolen instruments and forgeries.

We seek the collaboration of those, especially museums and collectors, who would allow instruments to be measured and added to the database (if desired, data can be kept confidential). Top and back plates of an instruments can be safely measured without removing strings, in about one hour. Data contributors receive complimentary color graduation maps, plus other benefits. More information is available online at: http://members.aol.com/viograds. We are interested in feedback and suggestions. Please email viograds@Aol.com, or send your comments to the CAS office, 55 Park St., Montclair, NJ 07042 USA; email catgutas@msn.com.

REFERENCES


Figure 2. Contour graduation maps of poor-sounding German Maggini copy (ca. 1920), produced using a GIS system (natural neighbor interpolation; contour interval 0.25 mm). Measured by author using caliper (instrument is viewed from inside). Note extreme thickness (max 5.8 mm) of top and lack of concentric pattern on back.
Bowing With a Glass Bow: An Update

R. T. Schumacher
Department of Physics
Carnegie Mellon University
Pittsburgh, PA, 15213 USA
E-mail: rts@andrew.cmu.edu

INTRODUCTION

In the November, 1996 issue of the CASJ Steve Garoff and I published an article [1] describing our observation of the wear track that is produced on the “bow” - a rosinied glass rod - as it is drawn over a violin string. Along with optical microscope and atomic force microscope (AFM) illustrations, we included signals from a force transducer at the bridge. The only reaction I received to the article occurred some years later when a violin maker mentioned that article to me in a rather wondering and skeptical tone, not aware that I was one of the authors. I took no offense at his skepticism since the relevance to violin acoustics, and certainly to violin making, reasonably seemed remote to him. I was not much help at that time, since possible applications of the simple observations in [1] were not obvious to me, either.

Since then, that seed has developed into a research tool for the investigation of friction in “autonomous oscillators” - the technical term for something that oscillates in response to a time-independent excitation. Our current application of the subsequent developments is strongly oriented towards tribological (i.e., friction) research. However, there may be some applications of interest to the stringed instrument research community, and I am taking this opportunity to bring the readers of the CASJ up to date subsequent work and to speculate on what might be learned that could be of value to violin acoustics.

First, however, I should acknowledge that the seeds for the current research were planted in my mind by Jim Woodhouse and the thesis of his student Jonathan Smith [2]. Smith and Woodhouse [3] measured the frictional characteristics of an autonomous oscillator that is simpler than a bowed string, namely, a single mass on a spring, but excited precisely in the way our glass bow excited a string. The problem of rather directly recording the friction force at the point of excitation and measuring the mass's response is easier for a single mass, as compared to a distributed mass (i.e., a string), and the velocity of the mass at the point of excitation is easier to measure. That is not to say that implementation of the idea in the laboratory does not require considerable ingenuity and careful design, and that is one thing that made it worth a Ph.D. thesis.

But their results really justified the effort, since it showed that the standard friction “function” that had been in use for at least since 1974 for the simulation of bowed string motion was incorrect in at least one important detail. The friction curve, that is, the function $F(r)$, where $F$ is the frictional force and $r$ the velocity of the mass, had an open loop. Figure 1 shows the two superimposed. Remember, though, that the Smith/Woodhouse result is for a single, localized mass, not the distributed mass of a string. The question remained, then, does

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**Figure 1.** From Smith/Woodhouse [2]. The solid curve is experimental data for the friction driven mass on a spring system. The dashed line is a double exponential fit to steady state friction force data, described in [2]. Note: The inadequacy of the dashed line for the dynamical system is the point of this figure.
the same type of frictional behavior govern the bowed string? Although it always seemed to me that the most probable answer was "yes," it is not wise to replace the old friction curve used in the simulations with a new one that came from measurements on a different system. After all, the old one was based on the quasi-static measurements of Lazarus [4]. But the implication of Smith/Woodhouse [3] is that a determination of a frictional force as a function of velocity, done by carefully measuring the force on a body sliding on something at one constant velocity, then at another, etc., is not relevant to describe the frictional force in an autonomous dynamical system, whether a mass on a spring, a string under tension, or squeaking brakes. That is the principle that motivates our current use of the bowed string as a model system to study friction.

The step that made the appearance of the wear track potentially more useful was the realization that if both ends of the string were terminated by transducers, the force on the string at the bowing point as well as the velocity of the string at that point could be reconstructed from the forces on the ends of the string, as recorded by those transducers. The method has much in common with techniques used in medicine to reconstruct images of internal organs and their functions in procedures that use ultrasonics, X-rays, positron emission by appropriate radioactive nuclei, and nuclear magnetic resonance. But the application to the bowed string is enormously simpler. I reported the application to the bowed string for the first time at the Edinburgh conference in 1997 [5]. Jim Woodhouse provided the essential step beyond the methods reported at that conference in 1999. On a visit to Pittsburgh, he noticed that the terminations on my experimental apparatus were sufficiently rigid that one could ignore losses of waves reflecting from them. If one also ignored the sound radiation produced by its movement through air, the only phenomenon that keeps the real string from being the textbook "ideal" string is the change of the wave shape on the string because of dispersion caused by bending stiffness of the string. Since Woodhouse had already written a paper in which the change of shape of an initial pulse caused by dispersion was calculated [6], he quickly realized the advantage in simplifying the methods I had been using to do the reconstruction. The details were published in the summer of 2000 [7].

RESULTS SO FAR

Now I will provide a brief survey of what we have learned with this new tool. Figure 2a shows two periods from a typical reconstruction, with the velocity of the string and the force on the string superimposed. We see the usual stick-slip phenomenon except that the stick portion does not remain exactly at the bowing velocity of 0.2 m/s. Note also the fluctuations in f(t), the frictional force, during sticking. These fluctuations show up even more dramatically when one plots each point in the force curve versus the corresponding point on the velocity curve, figure 2b. Since the digitizing rate of the apparatus is 128 kHz, there are 194 points per period if the frequency of the E string used is 660 Hz. All points for the two periods of figure 2a are plotted in figure 2b. The width of the vertical mass of points around 0.2 m/s shows the extent of the velocity fluctuations during sticking. The peak-to-peak height of the frictional force during sticking is a major result that we have used to draw some conclusions from the work.

The slipping loop in figure 2b, similar to that observed by Smith and Woodhouse [3], is typical, but there are variations in the openness of the loop, and even occasionally in the sense of circulation around the loop, that are a function of the normal force on the glass rod on the string and the thickness of the layer of rosin on the glass rod. Since

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**Figure 2a.** Two periods of string velocity and friction force at bowing point, for an E string at about 660 Hz. At the sampling rate of 128 kHz there are about 200 samples per period. The horizontal axis is time measured in samples.

**Figure 2b.** The two periods of 2a plotted, frictional force vs. bow velocity. Circles around data points for decreasing velocities aid the experimenter in determining the sense of circulation during slip, as marked on the figure. The "o" unconnected by lines locates the origin (0,0).
the amplifiers following the termination transducers are AC coupled, the DC component of the force is not measured. The lack of knowledge of the average force over a bow stroke is a serious one that will be remedied by the next addition to the experimental apparatus, but we can still draw some conclusions by making reasonable assumptions about what that force might be.

We are now in a position to shed light - experimentally confirmed light at that - on the question of how the bow maintains the slip-stick oscillation of the string. How is the energy supplied, where does it go, and what are the roles of the slip and the stick portion of each period?

Misinformation about bowed string dynamics is easy to come by, and part of it is a consequence of trying to understand the interaction of the dynamics and the frictional force from the type of friction function superimposed on the Smith/Woodhouse data in Figure 1. For example, I found the following quotation on the web:

"With high static friction, the bow tends to stick to the string ... and for a while it drags the string along with it. Meanwhile, the kink in the string travels along the string and reflects at the fixed end. When the kink returns to meet the contact point, the tension in the string now acts to pull it off the bow. Under appropriate bowing conditions (that are not easy to learn), it breaks free of the bow and then slides past it easily with very little friction, thanks to the rosin."

The above paragraph is misleading in the italicized portions (added) of the quotation. The best way to understand the slip-stick using the Helmholtz kinematics of the ideal string (a single slip and single stick per period) is to start from the statement that on a perfect string - no dispersion - with absolutely rigid terminations, all disturbances are periodic with period 2L/c, where L is the string length and c is the velocity of a wave on the string (modes of all frequencies have the same velocity by virtue of the no dispersion assumption of Helmholtz motion). The Helmholtz motion will maintain itself with no force needed once the motion is started. If energy losses from bridge motion and stirring up of the air by the string are small, then the force of the bow on the string just supplies the needed small corrections to the decaying motion to keep the string's vibration amplitude constant and to preserve the shape of the oscillating string throughout its period. As I mentioned before, the data shown (fig. 2) do not include the (unmeasured) DC, that is, time independent, components of the force of the bow on the string. Since the string is a linear system, one can simply superimpose the consequence of a time-independent force onto the consequences of the time-dependent force. The former is exactly the same as if you grasped the string at the bowing point and pulled it aside. The Helmholtz motion is then superimposed on this new average displacement of the string. The steady force has no role in maintaining the Helmholtz motion, which is now motion around the new string shape - a shape with a permanent kink at the bowing point. The magnitude of the kink for our experiment, incidentally, is about a string diameter, or a quarter of a millimeter, for the largest normal force we use, using an estimate of the average frictional force, F_a, that we think is reasonable. However, stay tuned. The static kink plays an important role in understanding the energy budget, as we will shortly see.

The other two sentences of the Web-based statement also fall in line with the facts of figure 2 and all other data that we have collected. The tension of the string does not act to initiate slipping. From the point of view enunciated above, the string does that by itself, and if the normal force of the bow on the string is not too large, the bow does not impede the stick to slip transition. Also, the notion that the slipping part of the motion occurs "with very little friction" is belied by the facts. The friction during the "loop" part of figure 2b is not smaller than the static friction forces. If ignoring the DC force bothers you, just imagine the F(v) curve moved rigidly upward by as much as you want - for example, so that the minimum force during sticking is zero - and see that the sliding friction is hardly small. In fact, in fig. 2b, and for many of

Figure 3a. Energy transferred into the string for the two periods of fig. 2a. The large nearly sawtooth waveform is, except for small wiggles, the displacement of the string at the bowing point multiplied by the assumed average friction force. The small wiggles on the sawtooth are the contribution of the product of the fluctuations of force and velocity during sticking, as shown on the curve of input energy with only AC reconstructed data. The net downward slope of about 2 micro Joules/period is probably caused by inaccuracies in reconstruction.
the experimental conditions of rosin layer thickness and normal force of bow on string, the largest frictional force occurs for fairly substantial string velocities, usually about -0.2 m/s. That means it is slipping relative to the bow at 0.4 m/s for the bow speed of 0.2 m/s. But the string’s position has changed by at most 10 micrometers by the time the relative velocity has gotten that large. That distance is roughly the width of a typical sticking scar on the wear track, so the string has not really slipped out of the sticking scar region when the maximum force is reached.

However, the DC force does play a role aside from maintaining the permanent kink. Let us consider the energy provided to the string. During a sampling time interval \( \Delta t \) (7.8 microseconds at our digitizing rate) the energy supplied to the string is

\[
\Delta E = \langle f(t) \rangle + \langle F_J \rangle v(t) \Delta t = f(t)v(t) \Delta t + F_J \Delta y(t)
\]

where \( \Delta y(t) \) is the change in the string’s displacement between \( t \) and \( t+\Delta t \). To get a graph of the energy as a function of time, one simply cumulatively adds the increments \( \Delta E \) (Note: the procedure is a pretty good approximation to an integral for the relatively high sampling rate). The result is shown in figure 3a, where I have assumed \( F_J \) to be half the peak-to-peak force during the sticking scribble of figure 2b. We see that the force on the string gives the string quite a lot of energy during the sticking interval, but it is almost all given back during slipping. The only part not given back is the result of the cumulative sum of the first term, \( f(t)v(t) \Delta t \). The second term is zero at the end of each period because \( y(t)=y(t+T) \), where \( T \) is the period. That is, if the motion is periodic, the string returns to its displacement a period later. The first term is non-zero at the end of each period because it gives the energy the string requires to make up for losses during a period. That energy is small. Small compared to what? For our experiment it is about 2% of the maximum energy of the string. We believe that 2% of that energy is too small for us to measure reliably because of the inaccuracies of the reconstruction from a variety of causes. But we really aren’t interested in it anyway.

What is more interesting is the energy supplied to the bow by the string. A very similar expression to eq. 1 applies there:

\[
\Delta E_b = -f(t) + F_J(v(t) - v_J) \Delta t
\]

The initial minus sign after the equals sign is from Newton’s third law: the force on the bow by the string is the negative of the force on the string by the bow. Subtracting \( v_J \) just moves the calculation to the rest frame of the bow. From the perspective of the bow, the string moves by, slipping and sticking as it goes, with an average velocity of \( v_J \). As figure 3b shows, the energy given to the string during sticking is almost zero, while the energy to the bow is supplied almost entirely during the slipping interval. However, from the point of view of the rosin on the rod, the small energy supplied during the long sticking interval is supplied to a small region of the rod, so the density of energy input to that region, the energy per length, might quite possibly produce a significant temperature rise at the bowing point during sticking. If so, that would affect the maximum frictional force the rosin can sustain while sticking, which in turn governs the maximum normal force that one can use and still sustain Helmholtz motion. These issues are now under investigation.

The picture of the string’s motion and the forces that keep it in periodic Helmholtz motion can then be restated from the bow’s perspective. During sticking not much happens; the string just rides along, jiggling a little bit (one can just integrate the velocity in the bow’s reference frame during sticking to see how much it goes back and forth), until the string’s existing kinematics causes the Helmholtz corner to pass by, and then the string scrapes on the bow at very high speed for a very short interval until the corner, having completed its round trip to the bridge, passes by again. Almost all the serious damage done by the string to the bow, if “damage” is measured by energy deposited onto the rod, is done during slipping. That can hardly be described as
sliding "past it easily with very little friction, thanks to the rosin."

One question remains. We have found that the motion of the string is roughly the same, both as the normal force of the rod on the string increases, and as the thickness of the rosin layer on the rod increases. Both increases result in an increase in average frictional force. The rising portion of the energy input to the string in figure 3a increases to greater heights as the frictional force increases. But the relatively constant kinematics of the string raises the question, where does the energy released to do damage to the wear track go during sticking? One would expect that the energy of the string would not change substantially for bow strokes with different normal force that preserve Helmholtz motion, since the kinematics of that motion does not change substantially as a function of normal force.

The resolution of the mystery appears at this writing to lie in that innocuous little kink. At the suggestion of Jim Woodhouse, I calculated the potential energy stored in the string for Helmholtz motion as a function of kink size. The result is that the maximum of potential energy of the string, which always occurs at the end of the sticking interval, increased by a factor of about 2.5 over the range of kink sizes produced in our experiments, and that increase matched reasonably well the increase in stored energy measured in our experiments.

Some of the more recent results described above were reported by myself and Steve Garoff at a meeting of the 75th American Chemical Society Colloid and Surface Science Symposium hosted by Carnegie Mellon University in June, 2001.

POSSIBLE APPLICATIONS TO THE ACOUSTICS OF REAL VIOLINS
This part of my note is highly speculative, and for practical reasons many of my speculations cannot be very vigorously pursued in my lab at this time. The applications of glass bow research that I can foresee now are in two categories: research on properties of commercial rosins, and research on properties of bows.

Rosin Properties
The material I have referred to above, somewhat carelessly, as rosin, is actually, as far as we can determine, abietic acid. It is a major constituent of stringed instrument rosin, which comes with various impurities that help distinguish bass rosin from cello rosin from violin rosin. And of course there is substantial overlap in all three categories. The only research I have done on the differences has been on the softening temperatures. I have found that the beginning of a slow distortion in the shapes of chips of various rosins under the force of gravity varies from 45°C for the example of bass rosin I examined, though the 50's for cello and some violin rosins, to as high as 65°C for a couple of examples of violin rosin. Since the constituents of the material are not known, we settled for our first set of experiments on a commercial grade of abietic acid from a chemical supply company. We found that the organic solvent xylene dissolves all but a small quantity of the solid material supplied, and we use, and "define" as abietic acid, that supernatant, with which we coat the rods. That procedure allows us standardize the material that we are using in our experiments.

"Misinformation about bowed string dynamics is easy to come by..."

One expects that a systematic survey of available commercial musical instrument rosins would reveal much of interest about the differences of available rosins. The problem that would have to be overcome is to insure that the properties of the thin layer of rosin on the string are the same as the properties of the bulk material. Clearly use of any solvent that does not completely dissolve the rosin will not adequately represent the material that is transferred to the bow. We have not investigated the range of common solvents available beyond ethanol and xylene, neither of which completely dissolves all rosins on the market.

Bow Properties
One would like to know how a bow responds to the forces on it. The most illusive of these is the force of the oscillating string on the bow. One approach would be a general characterization of bow response, either by applying a sinusoidal force one at a time to carefully selected places on the bow at all interesting frequencies (see reference 7 for a very early try at that), or to find a way to apply an impulsive force at those selected places. The response of the bow could be measured with accelerometers attached at a variety of places. Some thought and experimentation help find the best places.

Another approach would be to bow a string, secured and instrumented as we have done, and reconstruct the force on the string at the bowing point. Various precautions would have to be instituted. The bow hair should be bundled in order to excite the string over as small a region as possible (our excitation region with the glass bow is about 0.1 mm). Two differences between our experiments and ones with a real bow are important. The bow, unlike our glass rod, has a dynamic reaction to the forces applied to it. Our glass rods are securely fastened to the traveling sled. Transverse motions are severely constrained, and longitudinal motions along the rod are both small in amplitude and are very likely damped by the thin teflon tape that separates the rod from its semicircular slot in the aluminum carrier. As far as we can tell, our glass rod has "no dynamics".

The dynamic reaction of the bow is just what we want to look at. Instrumenting the bow with one or more small mass accelerometers would still be necessary. The frictional properties of rosined hair would be lost in the dynamical properties of the bow as it reacts to the force applied by the string, but the force of the string on the bow would be known, nevertheless, from the reconstruction. And in one bow stroke the reaction of the bow, via the accelerometers, could be recorded as a function of time, to be compared to the force that created that reaction, as determined by the reconstruction.
trivial or cheap. In addition to the two channels sampling at 128 kHz - it has to be that high to catch the dispersion wiggles [6] - there have to be additional channels, one for each accelerometer used on the bow, and they have all to be accurately synchronized. Each of our channels has its own analog to digital converter; they are not multiplexed, as is common with multichannel instrumentation. The reason is that one needs accurate timing information about the response of the accelerometers to a force at some distance away that occurred earlier than the response was recorded. I have to admit that without having tried it, I cannot possibly have foreseen all the difficulties involved. But I can think of no other way to excite a bow the way a bow is excited "in vivo", so to speak, and to do so with a known force.

ACKNOWLEDGMENTS
The "we" used frequently above refers, as I hope is clear from the text as well as the references, to Jim Woodhouse and Steve Garoff, both of whom I consult with almost daily. In addition, I have had valuable help in running the experiments from Chris Evans, and undergraduate Mechanical Engineering student at Carnegie Mellon University. I look forward to further participation from CMU undergraduates in the future.

REFERENCES
[8] Schumacher, R. T., 1975, Some aspects of the bow: Catgut Acoust. Soc. Newsletter No. 24, p. 5-8 [Note: This paper is reproduced as paper #33 in the Hutchins/Benade collection].
ABSTRACT
Part 1: A finite element calculation program is used to compute all mechanical vibrating configurations of a violin corpus up to a frequency of 2200 Hz (approximately one and one-half octaves above the open E string). Changes are then made to the violin and the effects on vibrating configurations and frequencies are calculated. Results suggest that top arching and top plate thickness in the vicinity of the f-holes are of primary importance in tone production. We also examine the role of wood stiffness across the grain with regard to frequencies in the nasal sound region just above 1000 Hz.

Part 2: Experiments conducted as a result of the Part 1 calculations confirm that
1. Few mechanical vibrating modes influence violin sound.
2. The f-hole region and details of bassbar design strongly influence tone.
3. Rib and top plate graduations in sensitive areas affect high frequency overtones and carrying power.

We stress the importance to violin makers of reopening an instrument to experiment with final thickness as a way of producing instruments with superior tonal characteristics.

INTRODUCTION
What happens to violin tone when we change a thickness or a contour of an instrument? Many violin makers have asked themselves this kind of question, but have come up with few answers. The complex design of the violin as well as problems with analyzing sound in repeatable experimental conditions have created nearly insurmountable obstacles to answering our questions.

However, with the development of computers and appropriate software we can now use computer analyses as tools to help us in the pursuit of clues, since the computer can calculate precisely the vibration modes, frequencies and deflection patterns of violin components, and can help us determine the effects of changes in a single dimension or variable.

This paper provides foundational information for a study of future "what if" questions. The data and input base is intended to be a beginning; the first part of this paper is written primarily for those who might be interested in continuing this work. After delineating what we have done and how comprehensive the computation system is, it describes just where we stopped.

The second part of this paper describes the magnificent complexity of the vibrating structure of the violin and the effect of design changes on those vibrating patterns. Makers can find clues here to guide their work.

The authors have been absorbed for years with the problem of determining the effects of the kinds of physical changes, which are available to makers as they design, fabricate, and adjust stringed instruments of the violin family. A recent previous paper advocates an "engineering approach", a method of making simplified partial analyses of a technical problem, which is sometimes all that is possible for a working violin maker [1]. This paper describes the results of one such investigation - finite element analyses of the corpus of a violin as changes were made in some design and physical characteristics. An overriding curiosity of any violin maker, amateur or professional, is to understand more fully what happens to the vibrating characteristics of an instrument as he/she removes wood from various locations on the instrument. The incentive to do this work was to discover whether any clues to guide a maker could come from such a mathematical analysis. The results of the work apply directly only to the vibration in a vacuum of a violin corpus consisting of the violin, less neck, fingerboard, and strings (a theoretical and impractical situation). However, since the only variables available to a maker are the dimensions and mechanical properties of the materials used it was hoped that some clues from this work might be of help.

BACKGROUND
Exploration of how a violin vibrates has been a preoccupation of makers and scientists for many years. A good summary of key early work is contained in two
compilations of papers in violin acoustics [2] and [3]. This reveals that technical investigators over the last 150 years have always been ready to use the latest developments in instrumentation to try experimentally, one more time, to identify the important modes of vibration of a violin. Analytical attempts with any specificity have only been possible since the development of large computers and finite element analysis. The earliest finite element model (FEM) analyses of the violin as a mechanical system, i.e. as though the instrument were vibrating in a vacuum, were made in the 1980's by Knott [4] and by Roberts [5]. A more sophisticated FEM analysis of a violin corpus was done by Moral and others [6]. The analysis is well documented but it includes information on only a few modes of vibration.

An early experimental study by Molin and Jansson [7] using a TV speckle interferometer technique and exciting the violin from the bridge explored the vibrating configurations of all acoustic modes of four violins up to 2000 Hz. Detailed descriptions of the deflection patterns of the modes up to 1000 Hz were provided. The configurations of the higher modes were said to be regular and overlapping.

While the work mentioned above has been important, most of it has been done to demonstrate methods, techniques, or configurations. The work reported here, in contrast, has been done to provide information about the effects of a number of the possible mechanical changes on the mechanical vibrating configurations over a wide range of frequencies and to illustrate how some of the results have been helpful to makers in practical situations. It represents a conscious choice to study the mechanical behavior of only the motion of the plates of the corpus in a vacuum in the hope that some insights might be revealed that would be of assistance to makers.

The senior author in the past has used finite element analysis to study in detail the mechanical modes of vibration of violin free plates and to develop information useful to makers on where the appropriate areas are for wood removal as they try to tune free plates to specified frequency targets [8]. In recent years he has had access to ABAQUS, a sophisticated finite element analysis system, and a large computer at the University of Delaware. This has made possible a study in similar detail of changes in the vibration modes of a violin corpus vibrating in a vacuum when changes are made to the mechanical system.

At the suggestion of one of the reviewers we provide the following overview and comparison of previous FEM studies of the corpus of the violin. The first attempt at applying FEM to an assembled violin was made by G.A. Knott [4], an engineering pilot in the U.S. Navy, who used the exercise to help learn how to run an early FEM program on a Cray supercomputer. He studied free plates, the corpus, and the entire assembled violin including its strings! He presented deflection diagrams for the first 9 modes of a corpus - up to 969 Hz (the thesis contains the diagrams). He also presented deflection diagrams for the first body modes of an assembled violin. He used NASTRAN and PATRAN FEM software and modeled the plates using shell elements.

At the same time G.W. Roberts [5] carried out a similar broad FEM investigation of the violin as part of his graduate work at the University of Wales. He also used NASTRAN FEM software. His elements were sophisticated thick shell quadratic elements, and he makes the point that these elements crudely model the rapid changes of slope near the ribs and the side blocks. He presented deflection contour diagrams for the first 14 modes of the corpus, up to 1128 Hz.

Recently Bretos, Santamaria, and Moral [6] carried out a similar FEM study using more modern computer methods and ABAQUS software. The nature of the finite elements used is unstated. The investigation related free plate vibrating characteristics with those of the assembled corpus. Six examples of deflection patterns of the corpus are presented, occurring between 411 and 1070 Hz.

In all these efforts, the values used for wood properties were close to those used by the authors (see Table 1) except that the ratio of modulus of elasticity along the grain and across the grain for maple was unusually high in the Knott study.

For all of the above studies except for the Roberts study the number of modes in the frequency region up to 1000 Hz and their general deflection configurations are in agreement with the authors' results. In the Roberts study there were more modes reported, and not all the configurations match, possibly due to the unusual finite element employed and the coarse element structure. In contrast to the present effort, these studies did not attempt to determine what the variations of dimensions or physical properties had on the computed output.

Part I of the present paper describes finite element specifications of the design of the violin corpus, which was the base case for calculations. It also describes the assumptions and simplifications incorporated in those specifications. It presents the deflected positions of those modes consisting primarily of deflections out of the plane of the top and back plates and describes a smaller number of modes in this frequency range, which consist primarily of in-plane deflections. It then summarizes the changes in vibration patterns when each of selected design parameters was varied. The sequence of calculations was not planned in detail and the frequency range was expanded as work continued. The work stopped when the senior author's tenure at the University of Delaware ended.

It should be emphasized that the analytical work in Part I provides information on the vibration behavior of a

Table 1. Wood material properties (Mega Pascals).

<table>
<thead>
<tr>
<th></th>
<th>Maple</th>
<th>Spruce</th>
</tr>
</thead>
<tbody>
<tr>
<td>E_{xx}</td>
<td>2000</td>
<td>700</td>
</tr>
<tr>
<td>E_{yy}</td>
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<td>0.02</td>
</tr>
<tr>
<td>\mu_{xz}</td>
<td>0.47</td>
<td>0.47</td>
</tr>
<tr>
<td>\mu_{yz}</td>
<td>0.48</td>
<td>0.42</td>
</tr>
<tr>
<td>Density (g/cc)</td>
<td>0.7</td>
<td>0.4</td>
</tr>
</tbody>
</table>
violin corpus as though it were operating in a vacuum. A basic assumption is made that the mechanical system and the fluid (air) system surrounding a vibrating violin are loosely enough connected so that a study of the mechanical system will reveal one or more strong modes that will dictate the air motion and thus the acoustical response at the frequencies of those modes. This assumption must be tested by an examination of the results of the calculations and by comparisons with experimental results. Of course, there are important additional vibration modes in which the air system plays the predominant role, and mechanical motions play a smaller part in determining the acoustical response as well as others that are truly complex.

In Part 2, we explore possible implications of the calculated results in determining configurations that might produce sound. We expand on some of the findings of the finite element studies that may be of special interest to makers, as revealed by our continuing experimental work on test violins. We also suggest potential areas for investigation.

PART 1 - FINITE ELEMENT ANALYSIS

Description of the Finite Element Analysis

We chose to use a coarse mesh and simple elements, in order to simplify the work of modifying the details of the design for each run. The top and back were described by the same mesh patterns that had been used in the previous work on free plate vibration [8]. The element used for the plates is the thin plate version in which each element has a constant thickness and the principal axes for stiffness are in the plane of the element (bent plate configuration). The thickness distribution in the plates was determined so that the free plate frequencies conformed to the targets set by Carleen Hutchins. The bassbar and the blocks are described using solid elements. The linings are represented using triangular solid block elements. The ribs are also described by thin plate elements.

The same wood properties were specified that have been used in previous FEM studies of free plates (see table 1). Since wood stiffness properties are different in three directions, the stiff axis had to be specified. The top, blocks, and linings were given the stiffness properties of spruce. The stiff direction for the blocks was perpendicular to the plane of the plates. For the linings, the stiff direction was always tangential to the rib direction. Ribs and back were given the properties of maple. The stiff direction of the ribs was always tangent to the plane of the rib.

Figure 1 depicts the rib, lining, and block assembly and its mesh, as well as the element mesh for the front and back. The front and back are viewed from the top of the instrument. A simplified bridge was added later with elements of the thin plate type, thickness matching a classic tuned bridge, and cutouts chosen so that the calculated frequencies when the feet were fixed were substantially the same as those calculated for the standard tuned bridge in the earlier fundamental work on bridge vibration (see fig. 3). Later the bridge top was restrained at the points where the strings touch so that there would be no out-of-plane movement at those points. The soundpost was described for convenience as a mass-free spring element so stiff that it is effectively rigid at all frequencies of interest. Support for the corpus was provided by a system of soft springs so that the corpus was essentially floating. The first six modes calculated by the computer program are those of a rigid structure on soft springs and are ignored. In order to provide some idea of the restraint provided by the mass and stiffness of the neck-scroll-fingerboard assembly, one set of runs was made with fixed outside corners of the neck block. This corresponds to the assumption that the neck and fingerboard assembly is infinitely stiff and heavy (the modeling of the entire violin was beyond the scope of the project).

Results of an ABAQUS computer run include the determination of frequencies of mechanical resonances, which the program calculated to exist in the specified range of frequencies. For each resonance the location of the high points of the deflection patterns in three directions can be shown on a contour map. Deformation models of the corpus showing only its "edges" can also be displayed.

Since the purpose of the analysis was to determine gross effects of design changes, the description of the model could be crude and the range of the variables could be far beyond that which would produce accurate results. The calculations for many runs extended from the lowest frequencies of interest up to about 2200 Hz (C6sharp). It is well known from experiments that the lowest resonances of violins are modes of the air system. The lowest important mode caused by the mechanical system occurs above the open A string. This mode is called variously the "wolf tone", the "wood mode", "B1", or "T" by different investigators. It occurs at 590 Hz in the calculated FEM base case and is the third mode found by the FEM program. Configurations of the lower two modes (fig. 2a) are not further included in the analysis. Over the selected frequency range of slightly less than two octaves, 590 to 2200 Hz, the analysis identified about 36 modes - 10 in the lower octave and 26 more below 2200 Hz. The number of modes investigated in any one run varied from six to the full 36. Frequencies and deflection patterns varied somewhat as design changes were made.

INITIAL RESULTS

Plate Deflections Primarily Out of the Plane of Plates

Figures 2 (b,c,d) display 36 charts of peak deflections, nodal patterns, and some of the important nodal lines of every mechanical vibrating out-of-plane mode of the top and back plates of the corpus from 590 Hz up
Figure 2a. Nodal lines and peak deflections of resonances at 432 Hz and above for the base case. Maximum amplitude, whatever the direction, is denoted by +10. All other peaks are relative to the maximum amplitude. Where the amplitude is shown outside the figure it refers to the amplitude at the f-hole edge at the location shown by a nearby heavy dot. Conditions were as follows: standard bassbar contour, tuned free plates, only in-plane motion permitted in the bridge, standard soundpost location, and plate arching as per Sacconi [1]. Nodal lines are sketched in areas of major deflections. Where no nodal lines are shown the local deflections were difficult to deduce from the colored display. For each resonant frequency the numbers indicate the location and relative magnitude of plate deflection peaks. Note that three resonances have maximum amplitudes that are not on the top or back plate. Each plate is viewed from inside the instrument. Blank diagrams represent cases with either low deflections or else slowly changing deflection patterns in which nodal lines could not be identified.

Figure 2b. Nodal lines and peak deflections of resonances at 432 Hz and above for the base case (see fig. 2a for explanation).

The detailed motions of the bridge and the bassbar, at all calculated resonant frequencies from 590 Hz to as high as 2000 Hz in some cases, the maximum deflection occurred in a plate element perpendicular to the plane of the plate. Deflections of ribs in this frequency range were small (see section on rib deflections at higher frequencies in Part 2). These are unique frequencies of the mechanical system vibrating in a vacuum and do not necessarily represent sound-producing configurations of a violin corpus vibrating in air. The deflection numbers calculated by the computer and shown on the deflection patterns are relative. The largest deflection in any direction is displayed as +10. All
other deflection numbers are smaller and indicate the relative deflection at that location as compared to +10, the maximum deflection. Numbers located outside the violin shapes (fig. 2a-d) refer to deflections of the edge of the f-hole at the location identified by the dark spot.

It is significant that in this frequency range the maximum deflection occurs at the edges of the f-holes on the top plate in almost two-thirds of the modes. At lower frequencies they occur at the upper f-hole wings. At higher frequencies they occur on the side of the f-hole and occasionally at the lower wing. At the highest frequencies they are at the bottom curve of the f-hole. Since one would suspect from other instruments that sounds are more apt to be produced at these dislocations, we suspect that details of the f-hole region, especially the edges, are important in producing sound.

Weinreich [9] discusses these out-of-plane modes in his paper on "Directional Tone Color", makes crude estimates of their density in frequency, anticipates that they will have a uniform density with frequency and that some will be able to radiate more or less independently and even produce "beams" of sound radiation that will provide the directional tone color. He estimates the effect is most probable at frequencies below 2000 Hz. The information of Figure 2 (b-d) provides support for his estimates. The average frequency spacing between modes (fig. 2b-d) in the region between 1000 and 2000 Hz is 34 Hz. Weinreich's estimate of 44 Hz is in good agreement, especially since his analysis did not take into account possible additional modes due to the f-hole openings in the top plate. The calculations (fig. 2b-d) reveal no dominant out-of-plane motions in the ribs occurring below 2000 Hz. This agrees with our experimental nodal line results on several violins, which reveal such modes only above about 2500 Hz.

**Deflections Primarily in the Plane of the Top Plate**

It was possible with the available version of ABAQUS to display only the "edges" as defined by the computer both when the model is at rest and when it is at its maximum deflected position. With this display one could discover the deformation patterns of modes in which deflections are primarily in the x and y directions. The several twisting motions of the bridge starting at frequencies of about 1500 Hz were eliminated by specifying no motion out of the plane of the bridge at the points where the strings touch the top of the bridge. Figure 4 is a view from the neck end of the deformations of the top at 1336 Hz, a local mode involving the bridge and the bassbar and the lower part of the C bout on the soundpost side. Figure 5 is a view of the same mode from the top. A similar wagging of the bassbar occurs in other modes to conform to (or magnify) plate up-and-down motions (see, for example, the

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**Figure 2c.** Nodal lines and peak deflections of resonances at 432 Hz and above for the base case (see fig. 2a for explanation).
Figure 2d. Nodal lines and peak deflections of resonances at 432 Hz and above for the base case (see fig. 2a for explanation).

1544 Hz configuration; fig. 2c). In the 1854 Hz mode the entire corpus is also moving in its fundamental mode in the x-y plane, with the center section moving opposite to the upper and lower ends (see fig. 6). A similar, more complex x-y deformation pattern occurs at 1987 Hz. At 2050 Hz the center portion of the bridge moves perpendicular to the plane of the bridge (see fig. 7). Finally, sidewise motions of the bassbar in increasingly complex patterns were found at higher frequencies as a part of these deflection patterns.

VARIABLES
Time permitted the following changes to be made from the original base line runs:
- Rib thickness (3 changes) - 2.0, 1.0, 0.7 mm
- Linings omitted
- Wood stiffness variations:
  - ratio of cross grain stiffness to along-the-grain stiffness
  - ratio of shear stiffness to along-the-grain stiffness
- Bassbar variations:
  - height contour
  - shortened length
- Added weight to top of bridge, 2 variations
- Restraint of the outside corners of the neck end block
- F-hole placement - moving upper sound holes toward the center
- Major structural change - top arch height increased

Rib Thickness Changes and Linings Removed
Calculations were made with a rib thickness of 1.5 mm as the standard dimension. Early calculations of the influence of rib thickness changes, made before the plates were tuned, revealed the lack of uniform change in the frequency of various modes. Changes in rib thickness to 2.0 mm from 1.5 mm increased the frequency of five of the six lowest modes from 0.5% to 1.0%, and the frequency of the mode near 830 Hz increased 2.4%. Reducing rib thickness from 1.5 mm to 1.0 mm in this assembly reduced frequencies about 1.5% except for the mode near 830 Hz and the mode near 1200 Hz which went down 2.0% and 2.3%.

Figure 3. Comparison of conventional bridge shape (left) to simplified bridge used in calculations (right). Numbers indicate first two natural frequencies of both designs for in-plane vibration with rigid supports at the feet.
Later calculations on the final model, having tuned (free) plates, showed similar uneven frequency changes over the range 590 to 1400 Hz. Reducing the rib thickness from 1.5 to 1.0 mm reduced frequencies by 0.5% to 1.0%, except for one mode near 1200 Hz that decreased 2.4%. A further reduction of rib thickness to 0.7 mm resulted in a decrease of about 1.5% except for the mode near 830 Hz and the mode near 1200 Hz, which decreased 3.2%.

When linings were removed in the early calculations, the mode near 700 Hz went down twice as much as the others. In the final model, removing all linings decreased frequencies by 4%, except the frequency near 830 Hz, which decreased by 6.7%.

In all of the above runs, the general nodal line patterns were unchanged and the modes appeared in the same order. Thus, one can presume that the structure of blocks, ribs, and linings is quite stable and does not cause major changes in the mechanical vibrating properties of the corpus, at least at frequencies of less than 1400 Hz (F5).

**Changes in Wood Stiffness**

Earlier work on free plates had indicated that only three of the stiffness properties of wood are important in adjusting the vibrating frequencies of plates [10]. These are stiffness along the grain, cross grain stiffness, and shear stiffness in the plane defined by these directions. The most sensitive characteristic was revealed to be the ratio of the cross grain stiffness to that along the grain. Calculations were made of the first eighteen frequencies (up to about 1400 Hz) when the ratio of cross grain stiffness to along grain stiffness of the spruce (top and bassbar) was increased by 21%. All frequencies were increased by about 1.5% to 2.0% except for one at about 1150 Hz (corresponding to the 1120 Hz configuration; fig. 2b) that stood out with an increase of 3%. A similar calculation of the effect of a cross direction stiffness ratio reduction of 29% produced similar results, -2.5 to -3.0% except for a -6% decrease in the 1150 Hz mode. The 1120 Hz mode in the base run clearly is more strongly influenced by cross grain stiffness variations since all its nodal lines run along the grain.

Calculations of the effects of changing shear stiffness gave similar results. Only one mode, in this case at around 1270 Hz, seems to be affected more than others when the shear stiffness was changed, presumably because the deflection patterns of that mode were more sensitive to shear deformations.

**Bassbar Variations - Height Contour**

Two series of calculations were made to investigate the effect of the bassbar dimensions on the vibrating modes. In one set, the maximum peak height was reduced in several stages about 12 mm to 5 mm. It was found that a reduction of maximum height to 8 mm without any other height adjustments made small changes in the nodal line patterns of the corpus. When this bassbar was trimmed over its whole length to be in proportion to the 8 mm maximum height the first mode uniquely different from those of the full bassbar was at a frequency of 1250 Hz. The modes above 1250 Hz became progressively different until by 2000 Hz each mode was considerably different, both in frequency and in plate deflection patterns. When the bar height was reduced in proportion to the 5 mm maximum the first unique mode was again at a frequency just under 1250 Hz.
Most of the modes were unique above about 1750 Hz (A6). Occasionally the nodal line pattern in the top would be different but the back nodal pattern would be the same as that in a mode at a nearby frequency.

**Bassbar Variations - Length**

In a second set of calculations the length of the bassbar of full height was reduced proportionately to the size of the bout - by 19 mm in the upper bout and 25 mm in the lower bout. A comparison of nodal line patterns failed to reveal any significant differences at frequencies up to 1500 Hz.

**Soundpost Location**

Soundpost position is known by all makers to be a sensitive adjustment for changing the tone quality and for balancing the response of the strings. Therefore it was a surprise to find that relatively large adjustments to the top location of the soundpost made little difference in the resonant frequencies of the mechanical behavior of the corpus.

A finer local mesh was created in the top plate near the location where the soundpost contacts the top. Even so, the minimum 5 mm adjustment in the along-grain direction and the 2.8 mm adjustment across the grain was much larger than makers have found to be effective. Modes of vibration were calculated for five alternate locations of the top of the soundpost: +5 and +10 mm, each in line and moved 2.8 mm sideways. Many of the resonant frequencies were changed little. By far the largest frequency changes occurred in the frequency range of 900 to 1200 Hz. Moving the soundpost toward the G string made only four of the 42 frequencies, all in the neighborhood of 1200 Hz, change by as much as 1%. The frequencies at 1036 and 1118 Hz increased, and the frequencies at 1244 and 2087 Hz decreased. The same pattern of changes occurred when the same sideways shift was made when the soundpost was 10 mm farther from the bridge. Moving the soundpost farther from the bridge by 10 mm while holding the across location the same had somewhat similar effects. The mode at 930 Hz was shifted by -2%, and the mode at 891 Hz by -1.1%. Frequency changes in response to these fairly large position changes provide little insight into influence of the soundpost position (see, however, the soundpost experiment described in Part 2).

**Weight Added to the Top of the Bridge**

Two sets of runs were made with weight added to the top row of elements in the bridge. This was done by changing the density of the material in that row of elements. One set of runs was made with the density increased to three times that of maple, and another set was made with the density increased to ten times. The added weight was approximately 0.6 gm and 2.9 gm. For reference, a fitted bridge weighs about 3 gm and practice mutes weigh as much as three times the weight of a bridge.

When the top of the bridge was unrestrained its lowest vibrating mode as part of the whole structure in each case was bending out of its plane (flapping). These frequencies were 601, 384 and 221 Hz for the 1x, 3x and 10x bridge weights, respectively. The lowest frequencies at which the bridge deflected in a twisting mode were 1664, 1108 and 640 Hz, respectively. At all these modes, the corpus was essentially at rest. Limited calculations were made when the locations where the strings contacted the bridge top were prevented from moving out of plane - only as much as 1410 Hz for the 3x runs and as much as 1244 Hz for the 10x runs.

"It is obviously impossible to design and make violins 'by the numbers.'"

Deflection patterns of the corpus through the identified frequency range were quite similar for all three bridge, with few exceptions. These are: A mode was added at 708 Hz for the 3x case and 667 Hz for the 10x case, similar in each case but not related to a base case mode. Two unique modes were found in the 3x runs at 917 and 1165 Hz. A unique mode was found in the 10x runs at 936 Hz and another at 1197 Hz, which matched the 1165 mode in the 3x runs. Ten modes in this frequency range were alike in all three configurations at slightly different frequencies. It was surprising how similar the vibration patterns were in all three sets of runs.

One unusual feature was observed. The 1336 Hz rocking in-plane mode, previously described as the lowest frequency at which there is little deflection of the plates in the perpendicular direction, was found unchanged both in the base case and in the 3x runs. The calculation of the 10x modification stopped short of 1336 Hz so there is no information about the existence of the 1336 Hz anomaly from this run. The fact that this mode frequency did not change when weight was added to the bridge clearly indicates that the bridge is not the primary subunit of the corpus that is causing this unique 1336 Hz resonance.

**Restraint of the Outer Corners of the Neck End Block**

Utilizing the free-standing corpus as a way to model the violin is only a first step. A rough attempt to simulate the influence of the neck and fingerboard was made by stipulating that the outside corners of the neck block would be motionless, a crude exaggerated approximation of the effect of the additional mass and stiffness of these components. Runs were made both with the normal bridge weight and with the 3x weight added.

There was little correspondence between modes in the fixed-block runs at two different bridge weights and even less when compared with the base run. In only six modes did all three configurations have the same deflection patterns, and these were all between 735 and 1120 Hz (except one at 1469 Hz). There were only 11 deflection patterns common to the two added-weight cases out of 36 total modes found up to 2157 Hz. It is clear that restraint at the upper block exerts a strong effect on the mechanical vibrating deflection patterns. The mass and the inertia of the neck-fingerboard assembly are clearly important factors that need separate detailed further investigation.
F-Hole Placement - Moving Upper Sound Holes Closer to the Center
A simple change was made in the position of the f-holes. On the assumption that one critical dimension determining the vibrating configurations of the corpus was the separation of the upper f-hole circles, each f-hole was rotated so that the minimum dimension between them was reduced from 39.2 to 36.3 mm. The location of the centers of the lower holes was unchanged. A complete set of runs was made up to 2200 Hz.

There were few changes in the configurations of vibrating modes. Only 5 of the 23 calculated modes changed. These were two near 730 Hz, two near 1300 Hz, and one near 1700 Hz. Clearly, the shifting of each f-hole location by 1.45 mm made little influence on the modal shapes in this frequency range.

Major Structural Change - Top Arching Increased
One change was made in the arching height of the top. The arching dimensions of the base case were from a Stradivari violin, according to Sacconi [11]. It is well known to violin makers that there are noticeable differences in tonal character between violins having significant differences in arching height, so a set of runs was made on a model with arching 2 mm higher at the bridge location (with the increase blended proportionally less toward the edges in all directions). This approximated the differences between Sacconi's Stradivari arching [11] and that of a 1704 G.B. Rogeri violin depicted in a STRAD magazine poster [12].

There were substantial changes. Both the frequencies and the deflection patterns had changed to a significant extent. Some modes with only local motions had survived. There were still two modes in which the peak deflections occurred at both f-hole wings and in the same general frequency range - at 1126 and 1362 Hz. All of the “rogue” modes that occurred consistently in all the earlier calculations had disappeared. In their place were two modes at 1590 and 1606 Hz with obscure deflection patterns similar to those at 1336 Hz in the calculations of the Stradivari arch. The mode in which the bridge bulged out of plane in its midsection occurred at the same frequency (see fig. 7). All other modes could not be related in detail to deflection patterns perceived in earlier calculations. In the lower frequencies, there was a strong difference in deflection patterns at about 950 and 1250 Hz. At higher frequencies, there was no correlation.

This series of calculations illustrates the strong influence that changes in arching configurations have on vibrating configurations of the mechanical system. It is obviously impossible to design and make violins “by the numbers”.

Effect of the Above Changes on the In-Plane Deflections - the “Rogue” Modes
Three modes describing motions in the x and y directions, described and discussed earlier, consistently appeared and remained constant and existed during most of the modifications described above. The 1336 Hz frequency appeared no matter how variables of the plates were changed—f-hole, rib thickness, etc., including bridge weight. The only changes that caused the 1336 Hz mode to change were substantial changes in the height of the bassbar and changes in the values of the cross-grain stiffness of top plate wood. There was no 1336 Hz mode in the calculation with no bridge. Similar statements can be made about the 1987 Hz mode. It appears that the portion of the rib and block assembly that were unchanged are important elements in these modes.

“The arching pattern proved to be the most important design variable.”

Figure 7 is a side view of the bridge deformation at a mode at 2050 Hz. In this case, the four points of contact of the strings on the bridge are prevented from moving out of the plane of the bridge. It is obvious that most of the side deflection of the bridge is taking place in the region of the cutouts. This is the first mode in which the bridge is vibrating independently of the rest of the corpus. The first rocking mode of this bridge occurs at about 2200 Hz.

At this point in our investigations, both time and energy ran out. Further experimental studies using these results will determine whether it is worthwhile to extend the work.

SUMMARY OF RESULTS OF FEM CALCULATIONS
The arching pattern proved to be the most important design variable. Increasing the arching of the top by 2 mm at the bridge and blending this increase in all directions substantially changes the frequencies and deflection configurations of most vibrating modes. Also, the thickness and edge patterns in the vicinity of the f-holes causes a large number of modes with peak deflections. Changes in material stiffness properties seemed to affect significantly only those modes which have deflections principally in the cross direction. Over the range studied there were only two modes affected, at 1150 and 1300 Hz, both in a frequency band in which acoustic tests indicate important harmonics [13]. Other obvious variables have only limited effects.

A variable that needs additional study is the restraining effect that the neck-fingerboard assembly imposes on the violin corpus by restricting the motion of the upper block. Calculated results show that complete immobilization of the corners of the outer surface of the upper block makes a substantial difference in the modal patterns. Thus, a heavy, stiff neck and a thick fingerboard will have a substantial effect on the mechanical vibration patterns of the violin. In addition, the strong influence of the bassbar on deflections of the top plate through the coupling of its out-of-plane motions suggests that more study is needed concerning bassbar stiffness and shape, especially with regard to out-of-plane bending.

It must be reemphasized that all of our calculations have been of the mechanical vibrations of the corpus structure as though it were vibrating in a vacuum. The relevance of this work to the actual sound produced by a violin derives from the assumption that
some sounds will be driven by the mechanical configurations of some of the natural frequencies of the corpus. Specific design changes have been studied to determine which changes have a significant effect on these configurations.

PART 2 - SOME RELATED INVESTIGATIONS RELATING TO THE FEM RESULTS

Usefulness of the Results Relating to the Amount and Frequencies of Sound Emitted

A fundamental question raised as a result of this FEM work is which, if any, of the vibration patterns of the wooden structure contribute to the sounds produced by a violin. It is well established that there are only about 25-35 modes that contribute to violin sound quality. For instance, in the 1970's Matthews and Kohut [14; see a short summary of their work in ref. 15] worked on how to synthesize a violin-like tone. They found that about 35 harmonics of more or less equal intensity and equally spaced harmonically, with damping enough so that the hollow between peaks were about 10 dB lower than the peaks, created a sound that many listeners could mistake as a violin sound.

There are also in the technical literature many harmonic analyses of tones produced by violins. Those produced by bowing in particular can be examined for clues. For several years we have been using the CONQT experimental method to record the harmonic components of all the sounds produced on a violin when a glissando is played. We also have been probing the vibrating surfaces of violins with a small microphone for the nodal line patterns when excited by a shaker at various frequencies. All of the above have provided useful information on the details of violin vibrations that have been discussed in previous papers [15,16,17,18].

Some reasoning is offered below to help develop criteria for plate mechanical vibration patterns that might produce sound. These criteria must include:

a. Vibrating areas on plates must not cancel each other out. For example, a positive deflection pattern next to a corresponding negative deflection pattern of plate vibration will disturb little air.

b. Deflection patterns that result in a change in volume of the corpus will produce sound by acting as an inflating balloon.

c. The movement of one particular area, even when most of the plate surface is fairly quiet, may cause the emission of sound (with a strong directional component).

d. In-phase deflection patterns of a plate or a major component may cause sound.

e. Directional sound can be produced by the flapping motion of the entire corpus, or some segment of it. These patterns are more likely to occur in completed instruments when the relatively large masses of the neck and fingerboard vibrate in opposition to the corpus.

f. Large motions of the f-hole wings and edges may produce sound by stimulating repetitive local air-flow patterns.

The application of the above criteria to the modes (fig. 2b-d) rules out most of the modes. One must conclude that the only modes likely to produce sound are those at 590, 960, 1120, 1374, 1409, 1888, 1912, and 2090 Hz.

Is this small number of modes—eight in almost two octaves—a reasonable number? A CONQT analysis of the sound produced by bowing (fig. 8) provides some clues on the number to be expected based on the experimental data. The sound peaks are from a CONQT test of a French 19th century factory violin that had been modified to have conventional top thickness and bassbar contour. This violin had good projection and a bright, strong sound in the upper register, presumably due to the strong peaks above 3000 Hz. The sound peaks in this instrument revealed by the CONQT test are, at the most, about eight or ten per octave. The shapes of the high peaks are spread out enough so that they signal appreciable damping in the wood and/or coupling of modes. Below the CONQT

Figure 8. Frequency distribution of mechanical frequencies compared to those shown by a typical experimental CONQT plot. Each peak represents a measured sound peak from a test of a French 19th century factory violin that had been modified to have conventional top thickness and bassbar contour. Each black tick mark on the line labeled “Mechanical vibrating frequencies” represents one of the frequencies displayed in fig. 2. Possible sound frequencies based on reasoned criteria are displayed as black tick marks on the bottom line.
plot are two sets of marks. The upper set indicates every calculated mechanical resonance frequency of the base FEM case (fig. 2b-d). The lower set contains marks only at the eight frequencies selected by the above reasoning process. Note that the spacing of these two sets of marks clearly does not match for this particular violin.

Unfortunately the nodal line technique, which searches for the deflection configuration of the plates, has proved to be unable to define experimentally the details of the deflection patterns of sound-producing modes above about 1200 Hz. However there were some small local areas in the plates in which the microphone response was strong at every frequency found in the CONQT test that has a strong audible response. These may be not only on the surface of the plates but also on the surfaces of the ribs, away from the blocks. The sound patterns are certainly directional at higher frequencies. There is no easy answer to the above question about sound produced by plate vibration.

**Soundpost Experiment**

Several years ago we performed a series of tests on soundpost placement, using an inexpensive violin, the plates of which had been tuned to Carleen Hutchins' free plate target frequencies. These results were reported in a CAS Journal article [13]. Three top soundpost positions were tested and CONQT sound analysis charts were taken at each step. On the final move, which was made to the position that sounded best to both of us, a new peak appeared in the charts at 932 Hz. We were able to describe the deflection patterns of the mode using the nodal line seeker (fig. 9). The full description is contained in the reference paper. Note that three quarters of the top is vibrating in phase. Thus, this experiment demonstrated one condition when a mechanical vibration will produce a sound, i.e. when most of the top plate is vibrating in phase with itself.

Detailed FEM computations of the effects of variations of soundpost position, as summarized earlier, tell a similar story. The greatest shift in frequencies occurred in the frequency range of 900 to 1200 Hz when the top position of the soundpost was moved. Some indication of the shifts in the vibration patterns and nodal lines is given by nodal line patterns for several soundpost positions in the frequency range near 966 Hz (fig. 10). The nodal line pattern from the base case is given in the top right hand corner for comparison. At that base line almost the entire top is vibrating in phase and most of the back is also in unison but in the opposite phase. The base case for the soundpost calculations is slightly different and the vibration pattern is also different. But notice that this pattern resembles closely the pattern achieved when the soundpost position is moved 10 mm away from the bridge. The shift in vibrating pattern has been quite drastic because of the move of the soundpost. Shifts of 2.8 mm in the other direction, on the other hand, produced little change in the nodal line patterns. Note that the peak amplitude remains at the upper f hole wing after all shifts. Similarly large shifts in vibrating patterns occurred at two or three resonances above and below this frequency when the soundpost was moved.

**Example of an F-hole Wing Configuration Producing Sound**

The 1120 Hz mode (fig. 2) in which both upper wings have maximum deflection is an interesting mode that produces sound. Nodal line probes on several instruments taken before the FEM work had produced evidence of sound at roughly that frequency.
in several instruments when the microphone probe was held directly over the two top wings during a nodal line experiment. It had then been assumed that this was caused by motion of the air system inside the corpus and breathing at the top wings of the f-holes. After the computer calculations had revealed the deflection configuration of the wings (fig. 2) at 1120 Hz it was possible to make a more detailed nodal line survey and to discover that the maximum response was directly over the wings and much less over the neighboring upper open holes. As a check, a CONQT set of runs was made when tiny weights were added on the top wings. The peak in the CONQT plots disappeared and there was a marked decrease in the sound quality. Evidently, the large amplitudes of f-hole wings must be regarded as strong potential sound producers at any frequency where this deflection pattern exists.

Additional Bassbar Experiments

An account of an early experiment of investigating the effect of adding weight locally to the top in line with the bassbar and later verifying the validity of this experiment by modifications of several violins is contained in the earlier companion paper [16].

Figure 5 is another view of the vibration pattern shown in Figure 4. This edge-on view shows the bridge deflected to the soundpost side. The bassbar is also rocking sideways with the highest amplitude in the lower bout. At some higher frequencies, similar sideways motion of the bassbar is found, usually linked to plate vibration patterns that encourage such a deformation. The inertia and twisting stiffness of the bassbar are additional variables in determining plate vibration patterns and amplitudes and must be considered by makers as they determine detailed adjustments in plate thickness. The basic design of the cross section of the bassbar will have an influence. For example a high narrow cross section should produce different plate vibrating patterns from those produced by a bar with a low wide cross section. Such information and reasoning gives credence to the experiments in bassbar cross section shape reported by Zaret [19]. Experiments on the effects of adding weights locally along the bassbar are now in progress.

Energy Absorbing Modes that Prevent Production of Sound

It is well known in mechanical engineering practice that a small resonant system can absorb energy and dampen vibrations of a structure to which it is attached if both have the same resonant frequency. There is a cello wolf damper on the market that is based on this principle. It is to be attached to the top plate near the maximum amplitude point. As has been described earlier the FEM calculations have revealed that the bassbar has frequencies of its own in important frequency regions in which it wags out of its plane, rotating at the joint between the bar and the top plate. Some of these modes are coupled to local deformations of the rib and block systems in the neighborhood of the C bouts. These frequencies can be easily varied by modifying the thickness of the bar, the taper in bar thickness, the contour of the bar height, as well as rib thicknesses or mass of corner blocks. The lowest of these frequencies occur at about 1500 Hz. Also, at higher frequencies there are modes at which different parts of the bassbar are moving in opposite phase. These modes will absorb energy and thus may mute some frequencies.

Clearly the finite element calculations have revealed additional variables; additional ways in which the corpus can vibrate, absorb energy, and therefore influence the sound produced by the instrument. These findings about bridge and bassbar vibration have important implications for instrument makers.

Investigation of Configurations of Higher Frequency Resonances

The FEM analytical approach has proven to be incapable of providing clues about the behavior of the violin at higher frequencies. Some experimental information is available from combinations of CONQT tests and nodal line probes made on two quite different violins.

One of the interesting resonances found on an 18th century English violin was a strong peak at about 3840 Hz (B7), a pitch close to that produced by a stopping the E string at the end of the fingerboard. Figure 11 is the A string CONQT run of this instrument. Note that this frequency has the highest response on the CONQT plot. It provided a bright strong overtone on many of the lower sounds of the instrument similar to the singer's formant. As an illustration of the complexity of the mechanical vibration patterns at this frequency, Figure 12 shows resonance patterns of the mechanical system of the FEM base case run at 3811 Hz, the closest resonance, computed for the top and back. Note that the peak amplitudes in the top are at the edges of the f-holes. This indicates once again the importance of the details of the f-hole region. When the nodal line microphone test system was tried on the above instrument at this frequency, it was impossible to sort out the complexity of the patterns. However, it was possible to scan the ribs and discover that there were strong active vibrating regions as indicated by the arrows. The sounds produced by these vibrations interacted in the space next to the top, and it was possible to use the microphone to detect intense sounds close to the instrument and other areas having no appreciable net sound. The plate vibrating pattern is beyond the ability of any maker to detect or to modify constructively. Even the rib pattern would be difficult.

The next lower similarly strong sound peak was found by the CONQT test on this instrument at about 2876 Hz. Figure 13 is a similar black and white display of the vibration patterns of the mechanical system of the FEM base case corpus at this frequency. There are about the same number of intense regions on the plates. However, the nodal line test picked up few vibrating regions on the ribs and those seem to be at the locations near the ends of the side blocks. Thus, the nodal line technique may still be useful to pick up locations where rib thickness and local variations of rib thickness may be important variables for defining the higher local frequency
responses of a violin. As an aside, it is worth mentioning that the cello will have quite different rib vibration responses from those on the violin, probably much lower in frequency relative to the other mechanical frequencies, because its ribs are proportionately much deeper and thinner than those on the violin.

Many of the better violins tested to date with the CONQT system do not have sharp peaked resonances at high frequencies but instead have a much smoother response profile. The authors have not yet identified modifications that strongly influence these resonances. This may be due to the wood and the varnish in these instruments providing higher damping values. The maker can influence the damping values by the choice of varnish, as discussed by Schleske [20], and thus control to some extent the "brightness" of the sound.

**CONCLUSIONS**

A. Every step in the final thickness adjustment process of every component of the violin influences the tuning of the natural frequencies of those components.

B. The mass of the neck and fingerboard will determine the degree to which the corpus is restrained from motion at the upper block. This will result in different patterns of mechanical resonance frequencies, especially in the lower octaves up to the first octave harmonic on the E string (1320 Hz).

C. The tuning of the bassbar is a poorly known but important area in need of experimental work by careful craftsmen. It can be used, presumably, both to deaden and to augment particular frequencies.

D. The ribs are sound producers in the high frequency region and are probably responsible for some of the brightness and projection qualities of good instruments. Therefore, the thickness of ribs and the details of linings, especially near the blocks, are probably important. This, too, is an unexplored area.

E. The f-hole wings are the most active area in many modes. Further experimental

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**Figure 11.** CONQT display of A string glissando run of an 18th Century English violin with a strong bright sound (left axis is time, seconds; bottom axis is frequency, Hz). The plot at the top is the peak amplitude occurring during the test at each frequency. Note that a peak at 3840 Hz occurs in almost every tone.

**Figure 12.** Resonance patterns showing deflections of back (left) and top (right) plates at 3811 Hz (viewed from inside instrument; based on ABAQUS display). Amplitudes are numbered using the same relative scale as in fig. 2 (a-d). Arrows indicate locations of prominent local sound peaks in ribs. Note that peak amplitudes in the top are at edges of f-holes.

**Figure 13.** Resonance patterns showing deflections of back (left) and top (right) plates at 2876 Hz (viewed from inside instrument; based on ABAQUS display). Amplitudes are numbered using the same relative scale as in fig. 2 (a-d). Arrows indicate locations of prominent local sound peaks in ribs.
investigations are needed of the influence of f-hole wing design on sound output.

Finally, the authors cannot imagine producing really fine instruments without tuning all of the components mentioned above, as well as making the traditional adjustments. We conclude that makers should expect to open up really fine violins for adjustment, as part of the finishing touches. A lifetime of restoring valuable instruments (for the most part restoring previous repair and modification work), coupled with the knowledge gained from various experiments and analyses described above, has drawn us to this conclusion. Consequently, the initial gluing of the top should be done in such a way that several internal adjustments can be done easily and securely.

REFERENCES

Violin Mode Relationships in Free Plates, After Attachment to the Ribs and in the Finished Instruments

Robert A. Wilkins
23 Polglass Way, Ardross 6153 Western Australia
E-mail: wilkinsc@iexpress.net.au

ABSTRACT
Vibrational modes of two violins are studied via matched free plates, with plates glued to the ribs, and finally in finished instruments. Mode changes and relationships are noted.

INTRODUCTION
In adapting my technique to use the Hutchins model of bimodal plate tuning [1], I have found that it generally produces good violins, violas and cellos but there are still some variations in the finished products that are unexplained and leave room for improvement. A further stage of mode research is needed to fill in the gap between the frequencies of free plates and their behavior when constrained in the corpus. The ideal methodology would be to utilize holographic interferometry but this is beyond the reach of most violin makers, so out of necessity only the humble shaker table was used but with surprisingly good results.

If we are to understand and control the process of violin making, we need to be able to trace the changes in mode frequencies and shapes throughout the various stages of assembly leading to the finished instrument. Will we be able to find relationships between free plate modes and the modes that can be traced when ribs are attached, neck attached, box closed, and soundpost, bridge and strings fitted? There are many variables to be considered with each one having implications for modification by the maker.

Is it possible to measure the effects, including interaction effects, of these variables in a cumulative way as they are added in the process of assembly? If so, the aim would be to acquire the knowledge necessary for making modifications at each stage towards a target set of mode relationships in the finished instrument. The ultimate demand on the maker is that he or she has measurable criteria that can be accomplished at each stage of making and assembly. It is recognized that this is a formidable task, but without this degree of detailed knowledge the uncontrolled changes that occur between free plate mode tuning and modes in the finished instrument will always be a serious problem for the maker.

So far the leap from free plate modes to assembled instrument is too great. The uncontrolled intervening variables can lead to unpredictable outcomes despite great care in accomplishing free plate criteria. That is not to detract from the importance of free plate tuning but to emphasize that other cumulative factors are also part of the puzzle that we need to have control over in order to predict the behavior of the finished instrument whether it be violin, viola or cello.

Transition from Free Plates to Plates Fixed to Ribs
Mode patterns for two violins were photographically recorded for free plates, for plates fixed to the ribs and again in the finished instruments. The attachment of ribs was not intended to establish the plate boundaries of an assembled violin but simply to see what happens to free plate vibrational modes and their free plate matching when the ribs are glued on. Each violin in this study was made using the outline and archings of the "Betts" Stradivarius [2]. The wood was European spruce and Bosnian maple seasoned for fifteen years. The ribs and free plates were first varnished, cured for 18 months, polished and the plates were then tuned with the aim of achieving bimodal matching or at least a match for modes #2. Mode shapes and frequencies were recorded for the range 0-1200 Hz using the well-known shaker table apparatus (See [1]).

Figure 1 shows photographs of the mode shapes and frequencies of top and back plates for violin No. 22 when the plates were free. Figure 2 shows the mode shapes and frequencies after the plates are independently glued to the ribs. Violin No. 21 produced almost identical mode shapes in all three conditions, i.e. free plate, with ribs attached and in the finished violin and is included as a check for consistency.

With the ribs attached in turn to each plate, some mode shape changes can be seen especially for mode #2. I found that free plate modes #1 and #2 for both violins were moved up the frequency range while mode #5 moved down (See Table 1). This was nearly true for the back plates with the exception that mode #1 moved down not up. The extent to which these modes moved was different in top and back such that
Wilkins - Violin Mode Relationships in Free Plates, after Attachment to the Ribs and in the Finished Instruments

Figure 1. Free plate mode patterns for violin No. 22.

Table 1. Change in violin mode frequencies from free plates to plates with ribs attached.

<table>
<thead>
<tr>
<th>No.21</th>
<th>TOP Hz</th>
<th>BACK Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODE</td>
<td>Free Plate</td>
<td>With Ribs</td>
</tr>
<tr>
<td>#1</td>
<td>89</td>
<td>96</td>
</tr>
<tr>
<td>#2</td>
<td>178</td>
<td>248</td>
</tr>
<tr>
<td>#5</td>
<td>340</td>
<td>302</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>No.22</th>
<th>TOP Hz</th>
<th>BACK Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>88</td>
<td>93</td>
</tr>
<tr>
<td>#2</td>
<td>174</td>
<td>263</td>
</tr>
<tr>
<td>#5</td>
<td>356</td>
<td>303</td>
</tr>
</tbody>
</table>

modes that matched in the free plates no longer matched in the fixed plates. However, in both violins the mode movements were similar (See Table 1).

This leads to the conclusion that a different sequence and spacing is occurring in both top and back plates in the transition from free plates to plates with ribs attached and that prior mode matches are now no longer present.

Another important observation was made while attempting to locate the center of each mode in order to read off its frequency. The frequency range in which excitation of the glitter occurred either side of its center was considerable with both violins. Regretfully the possible significance of this was realized too late to collect all the data from both violins, however, some data from No. 22 was collected and is shown in Table 2. Recording this spread of excitation can only be approximate because it dies away at the extremes and varies with speaker amplitude. The most accurate way to accomplish this is to take all frequency readings at constant speaker amplitude and check them on the way up and again on the way down in sweeps of the sine wave generator.

The wide spread of the observed frequencies raises the question of modal matching versus spacing in fixed plates. Is it this spread of excitation overlapping between top and back plates that is the desired feature of a good violin? Have we focused too much on the center of modes and ignored the extent to which their spread of excitation affects the creation of violin sound? After all, plate vibration in the antinodal areas is what transfers energy to the air and eventually to our ears as sound. This energy transfer from wood to air works best at the height of each antinode. Gaps between antinodes are gaps in energy transfer and are thus a problem for the creation of violin sound. However, some transfer occurs either side of center as evidenced by the movement of glitter as frequencies approach the center of each mode. Perhaps we need to start recording the observable spread of excitation either side of the modes in addition to the central
Table 2. Spread of observed glitter excitation with ribs separately attached to top and back plates.

<table>
<thead>
<tr>
<th>No. 22</th>
<th>TOP HZ</th>
<th>BACK HZ</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Lowest Center</td>
<td>Highest</td>
</tr>
<tr>
<td>#1</td>
<td>88</td>
<td>93</td>
</tr>
<tr>
<td>#2</td>
<td>246</td>
<td>263</td>
</tr>
<tr>
<td>#5</td>
<td>275</td>
<td>303</td>
</tr>
<tr>
<td>#Un-numbered</td>
<td>460</td>
<td>493</td>
</tr>
</tbody>
</table>

frequencies so as to get a more complete picture of how violin plates behave when fixed to the ribs and in the final assembly.

Differences in frequency spread might be a critical variable in the search for better violins. For example, when you play a chromatic scale across the whole range of a violin, some strong and some weak tones will be found. The weak ones are a problem for the player. Three research questions arise from this phenomenon. (1) Can an optimum point be located when plate thinning where the glitter spread either side of the target modes #2 and #5 is activated over a wider range of frequencies? (2) Would such a point in a free plate produce an optimum spread of excitation in a plate with fixed boundaries? (4) Would such an optimum spread of glitter in the free plate modes #2 and #5 increase the spread of excitation at higher modes in the finished violin and effectively fill any sound gaps in the instrument's output? These questions could prove to be a fruitful area for further research.

Testing a Wooden Frame as a Research Substitue for Ribs

A lot of work is involved in gluing and ungluing plates to ribs so it makes sense to see if a removable frame can be used as a simple substitute for ribs. A lightweight wooden frame with clamps and nylon bolts was made so that each plate could be used instead of the ribs. The complete frame with clamps and nylon bolts was approximately the same weight as a set of ribs. This is in contrast to the frame used by Atwood [3] which was made deliberately heavy "...to lower the frequencies of modes...". With a heavy frame, Atwood had difficulty detecting complete mode patterns no doubt because they were dampened by the mass of the frame. With the light frame, many mode shapes were clearly visible.

When thinking about the relationship between ribs and plate margins one should be careful in referring to plate margins as "the hinge about which the plate moves as it vibrates" [3]. The analogy of a hinge suggests that the plate vibrates as a whole and flexes at the plate margins adjacent to the ribs. It follows from this that if the plate margins are thinned all the way around, the plate will vibrate more freely like a diaphragm. Could the hinge analogy be misleading?

The areas of greatest vibration are the antinodes. These vibrations are "hinged" along adjacent nodal lines where they are minimized and change phase. The widely different mode patterns indicate that vibration occurs in many different places often involving only small areas of the plate. Only occasionally do the plate margins become energized and then often only in certain places not as a whole. Schleske [4] gives some good examples and discussion of the way nodes and antinodes pass across plate margins and through the ribs to the opposing plate. Perhaps the practice of thinning plate margins, as described by Saunders and Hutchins [5], is more important if performed only in specific places. Such selective thinning may facilitate a "hinge" for the complex bending and twisting of the corpus in relation to where nodes and antinodes cross the edges rather than some simple notion of allowing whole plate vibration.

Using the lightweight frame it was discovered that many of the detectable modes, though not mode #5, were shifted up the frequency range (not down as Atwood thought) when the plate was clamped to the frame. These frequency placements were very sensitive to varying the weight of the frame and tightness of the clamps. This suggests support for Hutchins and Voskuil's [6] conclusions that the rib stiffness changes the B1 mode and that stiffening the ribs increases A1-B1 delta spacing and changes the playing characteristics of the violin.

Because mode frequencies undergo dramatic changes with variations in the mass and stiffness of the frame, the method of using a removable frame as a research tool is not recommended. Gluing the plates to the ribs offers more constant and reliable data. Research using this method with ribs and linings of different mass and stiffness can add to our knowledge about the effect of these variables on plate mode frequencies but only at that point in the assembly process. Further changes are to be expected when the opposing plate is glued on to the ribs.

In a further step, an additional frame was made to arch over the top plate in order to attach a bridge and sound post to the top plate while it was constrained around the edges by the lightweight frame. An adjusting screw could be tightened to increase or decrease the pressure of bridge and sound post on the plate. It was found that mode #5 glitter pattern was now impossible to detect while mode #2 was still clearly visible. This is reasonable when we consider that mode #5 vibrates powerfully from the center of the plate. Clamping the plate in the center by means of bridge and sound post damps mode #5 at its place of origin. Mode #2 on the other hand is not suppressed by bridge and sound post.

Observations Concerning Plates Glued to Ribs

My observations from testing both free plates and plates glued to the ribs from two
Wilkins - Violin Mode Relationships in Free Plates, after Attachment to the Ribs and in the Finished Instruments

violins are as follows:

1. Mode frequencies in both top and back free plates changed when the plates became separately fixed by the addition of ribs.

2. These frequency changes were inversely related with mode #2 increasing while mode #5 decreased.

3. Top and back plates that were matching in mode #2 when free were no longer matched when constrained at the edges by attachment in turn to the ribs.

4. The bridge and sound post condition suppressed mode #5 but not mode #2.

5. When plates were constrained at the edges, the mass and stiffness of the constraining device had a big influence on the frequency placement of the observed modes. The use of a substitute frame instead of ribs did not produce data resembling those of glued-on ribs or data that are reliable.

6. Research using modal plate tuning might be enhanced by recording the spread of plate excitation in addition to just the central frequencies, so that attempts can be made to correct dead spots in the sound spectrum and degrees of overlap can be studied.

**Modes in the Finished Instruments**

The most prominent modes in the finished instruments were found by using the shaker table again but this time substituting tea leaves in place of glitter. Tea leaves slide better over the convex outside surface of the violin than glitter which tends to adhere at first to the varnish and then jump off when the speaker amplitude is increased. If tea leaves are used and the speaker amplitude is carefully increased, the tea leaves will slowly move into the mode shapes without too many of them slipping to the lowest point near the purfling. Careful frequency adjustments of the sine wave generator were made throughout repeated trials in order to obtain the most accurate measures of mode frequencies. By this method, a wide variety of mode shapes were found up to just under 1200 Hz. Modes at higher frequencies could be detected but not their shapes.

Since the top and back plates are coupled through the ribs, it is not possible to trace the changes in the original free plate modes #2 and #5 with any certainty. However, it now becomes possible after finding a prominent mode in the top or back, to trace its corresponding mode in the opposite plate. These paired mode shapes and their frequencies can be seen in Figure 3. A more complete reading of these data are given in Table 4 where the most clearly identified frequencies are shown for the top and back in both violins 21 and 22. The frequencies, identified on the left as a, b, c and d and also shown in bold numerals in Table 3, represent strong paired modes. Other

![Figure 3: Mode patterns and frequencies in paired top and back for finished violin No. 22.](image-url)
Table 3. Mode frequencies for finished violins Nos. 21, 22, 11 and 12.

<table>
<thead>
<tr>
<th>No.21</th>
<th>No.22</th>
<th>No.11</th>
<th>No.12</th>
</tr>
</thead>
<tbody>
<tr>
<td>TOP Hz</td>
<td>BACK Hz</td>
<td>TOP Hz</td>
<td>BACK Hz</td>
</tr>
<tr>
<td>255*</td>
<td>255*</td>
<td>236*</td>
<td>235*</td>
</tr>
<tr>
<td>267</td>
<td>263</td>
<td>263</td>
<td>343</td>
</tr>
<tr>
<td>412</td>
<td>380</td>
<td>380</td>
<td>387</td>
</tr>
<tr>
<td>a 472</td>
<td>472</td>
<td>456</td>
<td>456</td>
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<tr>
<td>548</td>
<td>548</td>
<td>560</td>
<td>557</td>
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<tr>
<td>c 698</td>
<td>698</td>
<td>693</td>
<td>698</td>
</tr>
<tr>
<td>793</td>
<td>740</td>
<td>793</td>
<td></td>
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<tr>
<td>d 855</td>
<td>855</td>
<td>807</td>
<td>807</td>
</tr>
<tr>
<td>924</td>
<td>996</td>
<td>910</td>
<td>911</td>
</tr>
<tr>
<td>1103</td>
<td>1060</td>
<td>1070</td>
<td></td>
</tr>
</tbody>
</table>

clearly identified frequencies are also shown while some, which were very small in amplitude or overlapping the more prominent frequencies, were omitted.

The frequencies marked with an asterisk produced a strong movement of tea leaves away from the neck in the top and button in the back extending over the entire surface of the plate almost to the end block region. Further testing showed that this is not a plate mode but rather the strong vibration set in motion by the neck and fingerboard, i.e., the BO mode (Hutchins and Voskuil, 1993).

While Figure 3 shows the mode shapes and their frequencies, it should be remembered that the dynamic process of mode formation reveals more to the watching eye than can be seen in a static picture. It was observed that the tea leaves tended to move rapidly outwards from a point at the center of the antinodes often moving away from two or more antinodes to be massed at a nodal boundary between them.

Small movements of the frequency dial either side of the mode center caused the nodal boundary to shift back and forth slightly. This observed phenomenon is best explained by the sound waves shifting the points at which, on the one hand, their crests come exactly into phase to form the largest combined wave at an antinode and, on the other hand, their crests and troughs meet in opposing phase to form a node or point of zero vibration.

Larger shifts of the dial often caused the tea leaves to break away and the lines to disappear altogether. This is best explained by the interaction of passing sound waves moving away from the points of optimum antinodal combination where plate motion is concentrated, after which motion is dispersed weakly over the plate surface to undo both antinodes and nodes.

At points where nodal lines intersect with the edges of the plates, the lines tend to become wider and more diffused which is likely to be the result of damping by the mass and stiffness of the ribs. Antinodal areas that overlap the plate edges are most often found in the C-bouts where the tea leaves can be observed to bounce vigorously and travel along the purfling indentation towards the upper and lower bout edges and past the corner blocks. This phenomenon occurs more in the back than the top no doubt because the C-bouts of the top are partly isolated from the center of the plate by the f-holes.

In some instances the tea leaves do not form at a node but rather are deposited like foam at a shore line where the waves have lost the energy to drive them further. These deposits are distinguished from nodal lines by their diffuse appearance and by the fact that they can be pushed further away from their antinode by increasing the speaker amplitude. In this case there is no adjacent antinode to vibrate the tea leaves back to form a nodal boundary.

In the frequency spacings between modes where no overlapping modes exist, the tea leaves do not move indicating no vibration of the plate and hence there can be no consequent transfer of sound to the surrounding air. Luckily these dead spots in one plate are most often complimented by spread from a separate vibration in the opposing plate so as to produce fewer gaps in the overall spectrum of possible sound production.

In frequencies above 800 Hz, the antinodal areas tend to decrease in size and increase in number while their nodal lines no longer intersect the plate edges. This was also observed by Miller [7] and Schleske [4], both of whom noted that top and back plates had thus become decoupled. With the decoupling of the plates at about the top of the second octave in the violin, the vibrational behavior of the independent plates now becomes an important focus for studying, and possibly modifying, the playing characteristics of the violin in the higher positions on the E string at, and above, the third octave.

**Cross Check with Two Other Violins**

At this point it was decided to cross check these mode patterns and frequencies with two other finished violins made previously to the same Strad specifications. Both these violins had lower free plate frequencies than those above (see Table 4). These frequencies were measured on varnished and cured free plates several years ago at the time of their assembly. Experience has shown that plate frequencies tend to increase a few Hz during the first few years after making so the differences between these violins and violins 21 and 22 may not now be so large. The right hand side of Table 3 sets out the mode frequencies for finished violins Nos. 11 and 12. The neck/finger board
Table 4: Free plate frequencies for violins Nos 11 & 12.

<table>
<thead>
<tr>
<th>No.</th>
<th>TOP Hz</th>
<th>BACK Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>85</td>
<td>101</td>
</tr>
<tr>
<td>#2</td>
<td>166</td>
<td>165</td>
</tr>
<tr>
<td>#5</td>
<td>343</td>
<td>341</td>
</tr>
<tr>
<td>No.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>#1</td>
<td>86</td>
<td>98</td>
</tr>
<tr>
<td>#2</td>
<td>167</td>
<td>167</td>
</tr>
<tr>
<td>#5</td>
<td>334</td>
<td>335</td>
</tr>
</tbody>
</table>

Tentative conclusions are first that the mode shapes in top and back for violins 21 and 22 shown in Figure 3 and also for violins 11 and 12 (not shown) show the effect of coupling through the ribs for the first three modes descending down the figure having frequencies below 800 Hz. This coupling tends to occur most through the C-bouts and is evidenced by the observed movement of tea leaves away from the ribs and towards the edges and the absence of tea leaf deposits in the C-bouts along the purling. Tea leaves that slid into the C-bouts quickly vibrated along the purling past the corner blocks to the upper and lower bouts. Above 800 Hz, nodal lines tended to create bimodal islands on the plates with a more widely distributed build up of non-vibrating tea leaves near the edges including the C-bouts.

Second, it is possible to find some links between free plate frequencies and those in the finished violins. The data in Table 3 are derived from four violins whose free plates were matched for mode #2 and near matches for mode #5. The free plate mode #2 frequencies were higher in Violins 21 and 22 than they were for violins 11 and 12, which corresponds to the respective higher frequencies for the B1 modes shown at b in Table 3.

Looking again it can be seen that there were some divergences in the free plate octave relationships that might be an alternate focus to the one above. Violin 21 and 22 had average separations from the octave* by -10 Hz and +6.5 Hz respectively while violins 11 and 12 were only 0 Hz and +0.5 Hz. These separations might be significant in comparing the modes in the finished instruments. The relative position of b (B1 mode) in the frequency spectrum moves closer to A and further from C for violins 11 and 12 than for violins 21 and 22. Manipulating the absolute frequency values of bimodally matched free plates as well as the octave relationships between modes #2 and #5 might be a way of changing the frequency spacing of the B1 mode relative to the other modes.

Finally, one more test was performed. The centrally mounted rosewood shoulder rest was removed from violin no. 22 and mode shapes and frequencies were again recorded. The mode shapes remained essentially the same except that ring modes tended to extend and move more freely in the region near where the shoulder rest had been. At the same time the frequencies for the coupled modes shown at a, b, and c in Table 3 moved up by about 11 Hz. Frequencies shown at d in Table 3 all moved up by about 40 Hz. Many more frequencies were now visible from 1000-1500 Hz in the top and from 1000-2000 Hz in the back and those that had been seen previously now appeared to be much higher. The shoulder rest adds considerable mass and probably some stiffness to the lower end of the violin and this, it seems, has the effect of raising mode frequencies but not altering mode shapes or relationships at least for the coupled modes shown at a, b, and c.

CONCLUSIONS
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REFERENCES

Postscript
The two violins under study turned out to be a delight to play with rich G strings, even tones across all strings, good bright and strong E strings, and with easy to play harmonics.

Note*: To calculate the average separation from the octave, first in top and back double the value for mode #2 to get their respective octaves. Take the value of mode #5 from the octave and preserve the sign. If the signs are the same in top and back, add the values together keeping the sign then divide by 2. If the values of top and back have opposite signs, use the signs when adding together and preserve the resulting sign then divide by 2. In each case the answer is the average separation index.
Admittance at the Frets of a Classical Guitar

Ricardo R. Boullosa
Centro de Instrumentos, UNAM
Circuito Exterior, Ciudad Universitaria
Apdo. Postal 70-186, C. P. 04510
Mexico D. F., Mexico
rrb@aleph.cinstrum.unam.mx

ABSTRACT
Admittance measurements show that the energy input from the string towards the guitar body comes not only from the bridge in the soundboard, at some low-medium to high frequencies the energy comes also from the fretted end of the string. It is shown that the mechanical admittance is lower over a broad frequency range at frets located near the ‘foot and heel’ of the guitar, than that measured at other frets. The admittance is relatively high at frets located over the body of the guitar, some comments on the influence of these results on the tone of the guitar is made. Some other measurements are reported with respect to the mode shapes of the lowest resonant frequencies of the neck.

INTRODUCTION
In the following, measurements are described relating to the vibration of the soundboard and neck of the guitar that are part of a continuing study on the classical guitar [1]. In order to explore the energy input from the string at the resting points on the frets, the driving point admittance was measured at different points on the neck, and compared with that measured at the bridge. The transfer admittance was also measured in a mesh of points in the neck to extract some deflection shapes, exciting at one point. A brief description of the equipment used and parameters measured are given in each case.

MECHANICAL ADMITTANCE AT THE NECK AND BRIDGE
A brief description of the measurements follows. The excitation was made utilizing a conventional electrodynamic exciter (B&K 4809) with an impedance head attached (B&K8001). The guitar (code-named RRB in ref. [2]) was suspended horizontally from elastic supports in an anechoic chamber, these supports grabbed softly the edge of the lower bout and the lower part of the neck; an elastic band fixed the peg in order to give stability to the guitar.

In order to measure the input force and velocity, the mechanical driving point admittance (velocity/force) was measured at 18 frets and at the bridge, using the electromagnetic exciter. As this exciter adds stiffness and damping to the measurements, which should show up specially at higher frequencies, the bandwidth was limited, in some cases, to a value around 2 kHz. The excitation point was located at each fret next to the first string along the bar from the first fret, up to the eighteenth fret, and finally at the bridge. The signal used was white noise fed trough an audio amplifier. Figure 1 shows the magnitudes of the driving point admittance, at the nut (front graph), consecutive frets up to the 18th, and the bridge (last graph on the back). A trend behavior of higher driving point admittance values at a given fret at several frequencies can be seen, specially at some medium and high frequencies, as one moves towards the bridge. At around the twelfth fret, beginning at fret ten, the admittance drops significantly which would account for the wood block (foot and heel of the neck) that fixes the bar to the main body, and rises again with higher relative values, towards the bridge. The rising values of admittance in the region of the body of the guitar have been observed also in the case of the ud [3]. Figure 2 shows the real part (conductance) of the driving point admittance. The conductance at the 18 frets, at the nut and at the bridge, is shown in the same order as in Figure 1. These results suggest that the frets play, at certain frequencies, a more
active role in energy transmission than is generally accepted. As the conductance is proportional to the energy input from the string to the frets or the bridge (other contributions to the conductance come from material losses at the neck and bridge, air friction, etc.), it is strongly related to the decay time of the sound of the guitar. These effects are also found in the electric guitar, but in this case they are detrimental to the sustain of the tone, and are thus called 'dead spots' or points of high input mobility [4]. The body and neck of the electric guitar should not vibrate in order to avoid the dissipation of energy. The existence of these admittance peaks, at the frets indicates that the string's energy dissipates at these points (and some of this energy might be radiated by the body and neck as sound). It was shown in [5], that the neck-guitar system has several modes of vibration. These modes might be excited at the neck, if the string happens to have a frequency component around a modal frequency. The nearer the fret is to an antinode of a given mode the greater the excitation of that mode, and greater the effect on the string frequency (tuning) and decay rate.

The deleterious effects in frequency, and decay rate of the string, occur due to the fact that the admittance is different from zero at both, bridge (with higher influence), and frets. The imaginary part of the admittance (susceptance) is related to the inharmonicity caused by the displacement of the frequency of the string due to coupling to a resonance of the top plate of the guitar [6,7], or as these results indicate, of the neck-body system. The frequency can be lowered or raised with respect to the ideal string frequency (fundamental or harmonic) depending on whether the susceptance termination behaves as a spring or as mass respectively.

The conductance, real part of the admittance, is related to the energy input and thus also related to the energy that is removed from the string, either by radiation as sound, or internal dissipation (material loss). It is then related to the tone decay. This can be shown by considering the energy balance in the string and the guitar's structure. The average power dissipated is equated to the average power input.

**Figure 1.** Driving point mechanical admittance magnitude of guitar RRB. Excitation points were located next to the third string along the bar on the frets.

**Figure 2.** Real Mechanical Admittance at 18 frets and the bridge, last graph, which shows greater values of conductance. Guitar RRB.
The power flow into the neck at the fret is given by: \( P_f = F(t)^2 \cdot ReY \), where \( F(t) \) is the exciting point force of the string, \( ReY \) the real part of the admittance. The peak potential energy stored in the string is given [8], by:

\[
E^p_{\text{pot}} = \frac{1}{4} \left( \frac{T}{L} \right) A^2 \pi^2
\]

where \( T \), \( L \) and \( A \) are the tension, string length and amplitude, respectively. The loss factor is defined [9], as:

\[
\eta = \frac{W}{2 \pi E^p_{\text{pot}}}
\]

where \( h \) is the loss factor (ratio of energy lost per cycle and peak potential energy stored in the system), and \( P_{\text{dis}} = \frac{W}{T} \) (\( T = 1/\text{frequency} \)), is the average power dissipated, thus: \( W = P_{\text{dis}} \cdot T \), and:

\[
P_{\text{dis}} = E^p_{\text{pot}} \eta \omega.
\]

On the other hand, if the string is vibrating in one of its modes, \( Y(s, t) = A \sin(\omega t) \cdot \sin(\pi \delta) \), and the average quadratic force is: \( F^2(t) = T^2 A^2 k^2 \) where \( k \) is the string's wave number. Thus the power input is: \( P_i = (1/2) T^2 A^2 k^2 \). Equating this average power input and the average power dissipated (eq. 3), one has:

\[
\eta = \left( \frac{4}{\pi} \frac{m_L \cdot L \cdot f}{T} \right) \cdot \text{ReY}.
\]

where \( m_L \) is the string’s mass/length. The amplitude time constant (time for the string amplitude to attain 1/e of its initial value, or decay constant) is related to the loss factor [10], by: \( \tau = 1/(pfh) \) (s), thus the energy time constant is:

\[
\tau = \left( \frac{1}{8m_L \cdot L \cdot f^2} \right) \cdot \text{ReY} \text{ (s)},
\]

which agrees with that given in reference [11]. This expression can be put in terms of the tension of the string and its length, by considering the squared frequency of the string: \( f^2 = (1/4 \cdot L) / (T/m) \):

\[
\tau = \frac{1}{2(T/L) \cdot \text{ReY} \text{ (s)}}.
\]

A similar expression holds for the effect of the bridge and the both effects should add to the decay rate (which includes all factors, material losses and radiation). The relation between the decay rate and the \( \text{ReY} \), was investigated in [12], but no mathematical relation between them was given.

**MODES OF VIBRATION OF THE NECK**

The peaks and dips in the admittance is related to the existence of modes of vibration in the neck-guitar system, as was mentioned before. These modes should add to the effects of the top plate resonances in the decay and inharmonicity of the tones of the string. Shown in Figure 3, are two of the mode shapes of lowest frequency. The deflection shapes were obtained by measuring the transfer mechanical admittance in a mesh. Thirty points distributed in groups of three along the bar, every 5 cm, were used. The velocity was measured with a laser velocimeter (B&K 8323), the excitation (white noise) was applied at a point in the first fret next to the first string. The mode shapes were obtained by picking the values of conductance at each frequency at all points; several shapes were obtained, but only two are displayed here. The first mode shape seems to belong to a torsional mode at 140 Hz, the second shape shown seems to be a combined torsion and out-of-plane shape. There seems to be a lower frequency mode around 90 Hz, but was not extracted in this set of measurements, which would agree with that reported in [5]. The real admittance graphs, when seen in closer detail show several groups of very close frequencies, the peak at 278 Hz seemed to have two close frequencies, but the resolution bandwidth used here (4 Hz) precluded their discrimination. The frequency at 278 Hz seems to agree with the modal frequency found in [5]. The clear discrimination of torsional and out-of-plane modes needs further work.

**Figure 3.** Operational deflection shapes of two of the lower resonant frequencies of the neck. Guitar: Nova.
CONCLUSIONS
The fretted end of the string in a classical guitar is not, as it is generally considered, a rigid termination. The admittance measured at the frets is significant when it is compared with that at the bridge, at certain frequencies. The admittance response functions show a dip (as a function of frequency and position) at the region around the twelfth fret, which corresponds to the existence of the “foot and heel” in the guitar. At some medium and high frequencies, the vibratory energy input from the string to the guitar, comes not only from the bridge input but also from the frets. The admittance peaks are due mainly to the modes of vibration of the neck-guitar system. The admittance at the frets influences the inharmonicity and decay of the tones and is an additional effect to that taking place at the bridge-top plate of the guitar. Depending on the position of the fret along the neck, and the frequency of the fundamental of the shortened string, the string could excite a given mode if it happens to have a frequency component around the modal frequency.

The nearer the fret is to an antinode, the greater the excitation of that mode.

REFERENCES
VIBRATIONAL DYNAMICS OF THE RESONANCE BOX OF THE GUITAR: FINITE ELEMENT METHOD AND MODAL ANALYSIS

M.J. Elejabarrieta¹, A. Ezcurra², C. Santamaría
Departamento de Física Aplicada II, Universidad del País Vasco, Apdo.644-48080 Bilbao, Spain

Present address:
¹ Mondragon Unibertsitatea, 20500 Mondragon (Spain)
² Departamento de Física, Universidad Pública de Navarra, 31006 Pamplona (Spain)

ABSTRACT
A numerical model has been designed for a guitar resonance box implementing the Finite Element method. The vibrational behavior of the guitar box has been studied starting on its main components, assembling them and allowing the soundboard - back coupling via the inside air. The model allows one to study the influence of each component on the whole box, and the contribution of the modes of the components (wooden box and its parts, and air), to the coupled modes. The results of the numerical model have been compared with the experimental modal analysis of a real guitar box.

INTRODUCTION
The guitar is a complex mechanical system because its dynamic behavior is determined by the interaction of several components. In particular two main distinguishable parts form the guitar box: the wood structure and the inside fluid. This work is devoted to the determination of the dynamic behavior of the guitar box by means of numerical calculations and experimental measurements.

We present the calculated low-frequency modes and natural frequencies of the resonance box (top plate, back side, ribs, edges and blocks) starting on the dynamics of the fundamental components: top and back plates. The boundary conditions have been chosen in order to make clear the interaction soundboard - back through the internal air exclusively. It must be noted that the numerical model corresponds in detail to a real guitar box, as designed and constructed by a skilled craftsman, and the numerical results are compared to the experimental modal analysis.

NUMERICAL MODEL
In this work we attempted to achieve a numerical model of the guitar box with a high degree of accuracy both in geometrical design and in material parameters. With this idea in mind, an expert craftsman built a real box simultaneously to the development of the numerical model. A partial model was previously used to study the soundboard dynamics along its construction process [1]. It should thus allow us to study the effects of several parameters on the behavior of the soundboard, both alone and forming part of the box.

The numerical model and application of the finite element analysis were accomplished using the ABAQUS (Hibbit, Karlson & Sorensen) software implemented on an Alpha 2100 workstation, in the case of the wooden box and the inner air individually. The numerical model evolved following the real manufacturing process designed by the craftsman, modifying its geometric characteristics and adding the internal structures, and became quantitatively more complex. Figure 1 shows two images of half of the mesh defined for the resonance box of the guitar. As can be appreciated the system of internal struts and the bridge are included. It must be noted that the thickness of the top plate varied from point to point. The resonance box contained 15946 nodes and 3132 elements and the numerical model guarantees the response up to 600 Hz. Regarding the materials they were decided by the luthier in the constructed box, and

Figure 1. Two longitudinal sections of the finite element mesh; the geometrical design and the added structures can be seen.
their material constants applied here too. The top plate was made of Canadian Cedar; the back plate, the bridge and the ribs were made of Indian rosewood. The bars and rods were made of spruce. The material parameters used for the numerical model can be found in [2]. No viscous effects were taken into account and so the modes for both the independent domains and the coupled system were normal.

The mesh for the air inside the cavity (see Figure 2) consists of 10788 nodes and 8474 elements and the element size guarantees the validity of the results up to 1 kHz. The material parameters corresponding to the air were defined under standard atmospheric conditions; no viscous effects were taken into account. The fluid was considered as being confined in a rigid cavity with an orifice, the sound hole, with additional boundary conditions. These conditions should guarantee the continuity of the fluid and should take into account the radiation of sound through the sound hole. The concept of “correction length” on the neck was used, and the length was determined by means of a preliminary finite element analysis [3].

The fluid - structure coupling is a complex phenomenon, regarding the numerical model, the analysis and the interpretation of the results. The coupling takes place in the interface of the two domains through the imposed boundary conditions. These domains describe different physical situations but none of them can be analyzed without taking into account another’s influence. In our case the interaction has been limited to small amplitude movements. The coupled numerical model and application of the finite element analysis were accomplished using the SYSNOISE software by means of the modal coupling method.

In order to study the soundboard - back plate coupling via the inner air and to compare the results with the experimental ones additional boundary conditions were imposed. The ribs were considered as completely clamped, to avoid their displacement. These conditions were applied to the real system by fixing the ribs by means of polyurethane foam to a metallic mould [4].

Thus, the dynamical behavior of the resonance box of the guitar in the low-frequency range has been calculated through the vibration modes of the structure and of the fluid inside the cavity, with the ribs fixed. On these uncoupled modes the modal coupling method has been applied to calculate the vibration patterns and natural frequencies of the coupled modes of the box. The interface is the air surface in contact with the inner surface of the box. Both surfaces are similar, but nodes and elements are not totally coincident, so the displacement of the structural nodes corresponding to the vibration modes of the air has been interpolated on the nodes of the interface. Although several approximations are implied in the numerical method the results are in fair agreement with the experimental measurements and allow us to analyze them.

RESULTS AND DISCUSSION

The Empty Box and the Air Inside the Cavity
The starting point of this work is the behavior of the main components of the box,
soundboard and back plate, individually, under fixed boundary conditions; that is, allowing the perimeter to have zero displacement and free slope, the so-called simply supported boundary conditions. Both components were finished in this stage with the complete strut system (bridge, transverse bars and fan struts in the case of the soundboard, and longitudinal and transverse bars in the case of the back). The patterns are conditioned by the presence of the main struts.

When the box is assembled (top and back are joined through the ribs and blocks), the upper block affects modes presenting vibration over the sound hole. Vibration appears only in one part (soundboard or back) and not in both simultaneously. This result is obvious: if the ribs are fixed and the cavity is empty there is no way to couple these two parts, and the modes are limited to one of them. Patterns are shown in Figure 3. Concerning the natural frequencies the only significant changes appear in the modes affected by the assembly of the box. In that case the modes are more rigid since their frequencies increase in comparison with the preceding results. The most sensitive frequencies to the assembly are those of the patterns of the back (1,2)b and (1,3)b, with a relative increase of the 15% with respect to the preceding stage. In the case of the top plate, the sensitive modes to the assembly, (1,2)s and (3,1)s, increase their frequencies less than 7%.

According to this it is possible to predict the behavior of the resonance box by observing the dynamics of the independent components in fixed boundary conditions. The only affected modes are those presenting vibration in the zone of the upper block. This piece hinders the movement in its zone, and increases the frequencies of the affected modes. The six lowest modes and frequencies corresponding to the air inside the rigid cavity are presented in Figure 4, and are named in the usual mode, starting from A0.

**The Guitar Box**

In this paragraph the dynamics of the complete guitar box, that is the coupled modes of the wood structure together with the inside air, are presented. The air contained in the box allows the connection between both structural components. In this way the coupling among the soundboard, the air and the back gives rise to the modes of the box. Figure 5a shows the calculated coupled modes; they are indexed by the TB capital letters, referring to the top - back coupling, followed by the mode number starting on the lowest frequency.

As can be seen the air inside the cavity substantially affects the vibration patterns, thus it causes the soundboard and the back move together at certain frequencies. Some comments can be made on this Figure. First of all they show that the influence of the air is greater at lower frequency. Comparing this figure to Figure 3 shows that the natural frequencies of the corresponding structural modes have decreased, that is, the fluid acts as an added mass. Moreover the interaction between structural and acoustic modes depends on the vibrational character of the structural modes. For example some modes appear only in one component, top or back; that is the case of the transversal flexural modes.

**THE MODEL AND THE EXPERIMENTAL RESULTS**

As has been explained above the model corresponds to a detailed hand crafted guitar box. This guitar box was studied by means of the modal analysis method along its constructive stages. In particular the vibrational behavior of the box was determined by imposing similar boundary conditions of this finite element model. Figure 5b shows the experimental modal patterns and frequencies. As can be seen by comparing Figure 5a to Figure 5b, the proposed finite element model predict with a high degree of accurateness the dynamics of the guitar box in the low frequency range, both qualitative and quantitatively. Some comments can be made about the coupled modes.

TB (I): This is the fundamental mode of the resonance box. In this pattern the soundboard presents an antinodal zone at the position of the bridge. The upper zone of the soundboard remains motionless due to the presence of the transverse bars and the upper block. Also
in this case, vibration is limited to the lower part of the back but in this case the amplitude of vibration is three times smaller than in the soundboard in the calculated mode; the experimental results only indicate that the vibration amplitude is higher in the soundboard. Figure 5 shows that the top and back plates vibrate in phase opposition; this leads to considerable changes in the inner volume. The frequency is reduced by 40% with respect to the empty box, pointing to the added mass effect of the fluid interacting with the structure.

TB (2): The second vibration mode of the resonance box, named as T' (1,1) or first top plate resonance by other authors. The shape of this mode is similar to the TB (1) mode but in this case soundboard and back vibrate in phase, and so the involved volume changes in the inner air are smaller. Figure 5 shows that the maximum displacement of soundboard and back are equal and situated in the lower part of the box.

TB (3): In this mode the back remains motionless: as has been explained before, the internal bars of the back hinder the vibration of its transversal flexural patterns and thus the contribution to radiation due to the back will be negligible. No changes were found on comparing TB (3) with (2,1) s, either in pattern or in frequency, and therefore no soundboard - back coupling is present in this mode, so it can be analyzed as a soundboard mode.

TB (4): Its pattern is similar to TB (1) mode, involving large volume changes, so TB (4) is an efficient mode of radiation too. This mode could not be experimentally determined, probably due to the proximity of TB (3).

TB (5): The soundboard and the back vibrate in phase opposition, with a longitudinal flexion character. The maximum vibration amplitude occurs around the sound hole in both plates. The lower part of the box, below the bridge, displays a weak movement that is more visible in the soundboard. The experimental measurements indicate that the vibration amplitude is larger for the back than for the soundboard; this fact is not so clear in the calculated mode. This mode, like TB (1) and TB (2), is a strong radiator and together with the TB (6) mode covers most of the fundamental notes and most of the second partials of the lowest notes of the instrument.

TB (6): This pattern presents a character of longitudinal flexion, just like TB (5), but in this case the soundboard and back vibrate in phase, so the air volume changes are smaller. Figure 5 shows that the maximum amplitude happens at the soundboard, around the sound hole. Unlike mode TB (5), this mode presents a notable movement around the bridge and will therefore be easily excited by the strings. Modes TB (5) and TB (6) form a symmetrical-antisymmetrical pair, as do TB (1) and TB (2). Both modes vibrate with a longitudinal flexion character with two antinodal zones in the lower and middle zones of each component. The presence of vibration all around the sound hole indicates that both modes will be efficient in sound radiation.

TB (7): In this mode the soundboard remains motionless, just as did the back in TB (4). Both TB (4) and TB (7) have a transversal flexural character. Since this mode appears only in the back without coupling with the soundboard, acting on the strings will not excite it and so it does not contribute to the sound radiation and quality of the guitar.

TB (8): This mode has a longitudinal flexural character in the back, presenting an antinodal region between the transverse bars. The soundboard remains nearly motionless, except for a slight vibration in the lower zone, not visible in the experimental results. Due to the slight

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**Figure 5.**

a. Calculated coupled vibration modes and natural frequencies of the resonance box of the guitar with the ribs fixed. The amplitude values cannot be compared one each other because they are independently normalized.

b. Modes of vibration and natural frequencies of the resonance box of the guitar with the fixed ribs as obtained from experimental measurements analyzed by the Modal Analysis technique.
vibration of the lower part of the soundboard, acting on the strings will excite this mode although the strongest vibration will occur on the back.

**CONCLUSIONS**

The Finite Element method has been applied to the resonance box of a guitar with the purpose of analyzing its vibrational behavior. The model has been progressively developed, starting with the soundboard and back, then the assembled box and the inside air separately and finally the whole box, that is, the wood structure and the air together. In this way the mode evolution can be tracked, establishing the influence of each component on the final box. Comparing the modal patterns and frequencies with the Modal Analysis results corresponding to a real guitar box has proved the goodness of the model. The newest feature of the work is the successful development of the model to calculate and analyze the box - air coupling. In this sense we can conclude that:

- Taking the air into account is essential to describe the vibrational modes of the guitar box.
- The soundboard - back coupling via the inside air is decisive for the two lowest modes.
- Modes presenting longitudinal flexural character in both soundboard and back of the empty box are capable of coupling.
- Modes presenting transverse flexural character either in the soundboard or in the back of the empty box produce non-coupled modes, that is, only one component vibrates.
- The Helmholtz resonance participates whenever soundboard and back couple.
- The acoustical and the structural modes couple when their vibrational characters (longitudinal or transversal) coincide.
- The influence of the inside air is decisive for the natural frequencies of the lowest modes of the box.

**REFERENCES**


THICKNESS GRADUATION SYSTEMS OF VIOLIN FAMILY INSTRUMENTS: PRELIMINARY STATISTICS AND CONCLUSIONS

Jeffrey S. Loen
18725 60th Ave NE
Kenmore, WA 98028 USA
Email: viograd@aol.com

"It is in the adjustment and regulation of these thicknesses that the true talent of the fiddle-maker asserts itself, or is conspicuous by its absence." Ed Heron-Allen

ABSTRACT
Thickness data on 232 plates of violins, violas, and cellos constructed by 44 important makers have been systematically compiled using a geographic information system. Descriptive statistics, plus analysis of contour maps of plate thickness indicate that these instruments have highly variable graduations (often with extremely thin tops), and highly asymmetrical graduation patterns. These plates are quite unlike what is portrayed in modern violin making books.

Contour maps are used to define a membrane-like uniform system, a bull's-eye-like concentric system, and a backbone-like longitudinal system. Instruments made in Cremona, Italy from 1604 to 1745 by the Amatis, Guarneris, Stradivari, and others display, with few exceptions, uniformly graduated tops and concentrically graduated backs. In contrast, the concentric system is used in both tops and backs made in Brescia, Italy prior to 1623 by da Salo, Maggini, and others.

INTRODUCTION
Thickness distributions of top and back plates of violin family instruments have important implications for understanding both the working methods of the finest makers and the acoustics of the finest instruments. Plate graduations can have a major influence on timbre, clarity, and volume of an instrument's voice, and the literature of violin making contains many testaments to the importance of attaining optimum thickness graduations for a particular model, arching, and wood quality. Little has been done with thickness measurements on classic instruments since the work of Antonio Bagatella [1], Cozio di Salabue [2], Eugen Vitachek [3], and the Hills [4,5]. This paper gives preliminary results of an on-going compilation and statistical evaluation of currently more than 15,700 graduation measurements of 232 plates representing six celli, 20 violas, and 98 violins by 44 leading Italian, Austrian, German, French, Hungarian and American makers who worked between 1564 and 1988.

DATA COMPILATION, PROCESSING, AND ANALYSIS
Graduation data (plus other descriptive data) were compiled from published sources, museum files and collections, and private collections. The database was constructed using a geographic information system, which is a computer program capable of storing, retrieving, analyzing, and displaying data distributed on an X-Y coordinate system. Thickness data (generally maps showing point measurements within plate outlines) were digitally scanned and the images were registered with regard to X-Y coordinates. Locations of point measurements on scanned images were screen digitized, and data values associated with point locations were typed into an associated database.

Table 1. Summary statistics of thickness data for violin, viola, and cello.

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Number compiled Inst. Plates</th>
<th>Total data points</th>
<th>Ave. number measurements per plate</th>
<th>Ave. plate length (mm)</th>
<th>Ave. arching height (mm)</th>
<th>Back mean</th>
<th>Ave. Thickness Back min</th>
<th>Ave. Thickness Top min</th>
<th>Ave. Thickness Top max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Violin</td>
<td>98</td>
<td>12985</td>
<td>71</td>
<td>352.5</td>
<td>15.2</td>
<td>15.8</td>
<td>2.10 3.06 44.73</td>
<td>2.02 2.69 3.57</td>
<td></td>
</tr>
<tr>
<td>Viola</td>
<td>20</td>
<td>2183</td>
<td>59</td>
<td>423.0</td>
<td>18.8</td>
<td>19.1</td>
<td>2.10 2.90 3.80</td>
<td>2.10 3.30 4.80</td>
<td></td>
</tr>
<tr>
<td>Cello</td>
<td>6</td>
<td>574</td>
<td>51</td>
<td>731.5</td>
<td>27.6</td>
<td>26.2</td>
<td>2.74 4.72 7.82</td>
<td>2.73 3.71 4.95</td>
<td></td>
</tr>
<tr>
<td>Totals</td>
<td>124</td>
<td>15744</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

CASJ Vol.4, No. 4 (Series II), November 2001
RESULTS
Graduation patterns of violin family instruments in the database generally are asymmetrical and the precision of the graduation is less than might be expected from the work of accomplished makers according to modern violin making standards. Consequently, many contour maps show a scattering of thinner and thicker areas roughly 20-40 mm or more in diameter. Data for many classic violin plates include very low thickness values (as low as 1.2 mm), especially in marginal areas of top plates. Sixty-six Italian violins show a range of 1.20 to 4.50 mm (average 2.66 mm) for tops and 1.20 to 6.50 mm (average 3.09 mm) for backs.

Other findings are evident based on visual examination of contour maps. For example, certain combinations of systems in top and back plates seem to characterize various makers and schools of making. The most common pair (72%) for pre-1762 Old Master makers is the uniform system for the top plate combined with the concentric system for the back plate. The second most common pair (12%) is concentric system for top plate combined with concentric system for back plate. The concentric top/concentric back combination is found in most (66%) Brescian instruments, whereas uniform top/concentric back is common in most (81%) Cremonese instruments.

DISCUSSION
Graduation Systems of Classic Makers
The literature contains many generalizations about plate thickness and graduation systems of primary Italian makers, some of which are confirmed based on the present study, but many of which could not be confirmed. For some reason there seems to be a history of authors presenting their own notions of idealized classic graduation patterns, rather than actual data.

The Hills [4] describe a gradual diminishing of top center thickness from da Salo to Mazzini, to the Amatis, and finally to Stradivari. This may correspond to a change from a concentric system in da Salo and Mazzini's top plates to the uniform system used by the Amatis and the later Cremonese makers, as documented by the present study. Uniform tops appear as early as 1628 in instruments of Nicolo Amati and the Brothers Amati, based on the present study. However, this conflicts with Bagatella [1] who, based on an examination of instruments by the Brothers Amati, postulated a concentric graduation system for both tops and backs. It is worth noting...
that Vitachek, who measured many important instruments in Russian collections between 1898 and 1926, was unable to confirm Bagatella's ideas [3], and Yankovskii [6] states that Bagatella "ridiculously simplified the distribution of thickness".

Statements by Cozio di Salabue [2] that the Amatis adjusted their top thickness in three sizes, the thickest of which was presumably a concentric circle in the center of the C-bouts, could not be confirmed. Nine tops by Antonio, Hieronymus, and Nicolo Amati are all classified in the uniform system (although one top by Andrea Amati in 1574 is concentric). Salabue also remarks that Stradivari made tops in "three thicknesses" (presumably concentric system) during the 1600's, and then changed to a uniform system in the 1700's. However this also could not be verified because nine pre-1700 Stradivari tops in the database are all classified in the uniform system. Data on more plates by these makers may help to resolve such questions.

In some cases modern authors have apparently presented their own ideas about perfect graduation patterns as that of famous makers. Certainly Sacconi’s fig. 66 [7] represents no single Stradivari violin. In an example from a violin making book, Doerr [8] presents a graduation contour map of the "King Joseph" Guarneri del Gesu of 1737 in which the top is much thicker in the center than is indicated by actual data measured on the same violin [9]. Doerr likely increased the original thicknesses to fit his particular taste, since he states on the map sheet that he used "license" in creating his graduation drawing. Likewise, it is unknown which "Guarneri" is portrayed as being perfectly symmetrical and concentric by Sandvik [10, fig. 56], because the map bears little resemblance to 27 Guarneri top plates in the database. Other authors of violin making books [11,12] depict idealized graduation maps that are not attributed to a particular classic maker, although these maps imply that graduation patterns should be symmetrical and even, which is unlike old master plates in the database.

The Question of Modifications and Regraduation

An important consideration is that the shapes of plates have changed because of the passage of time and actions of individuals who have performed repairs, regradation, and patching. It is likely that some plates have been modified. However, thickness distributions and graduation patterns of three violins (2 Stradivari copies, 1 Guarneri copy) by French luther J.B. Vuillaume, are similar to those of many instruments by Stradivari and Guarneri in the current database. Vuillaume had access to many original instruments in during the middle 1800's, and his impeccable copies may provide "snapshots in time". However, regardless of modifications, I propose that from an acoustical and historical point of view it is worth documenting plate configurations of famous instruments that have satisfied musical demands of the world's leading players.

CONCLUSIONS

The use of contour maps to evaluate top and back plates of classic violin family instruments leads to the tentative conclusion that most European old masters carved plates that were highly variable and asymmetrical, which is quite unlike modern violin making practice. Moreover, the gross structure of plates, as reflected in the use of uniform, concentric, and longitudinal graduation systems, varies depending on era and locality. These conclusions are preliminary and more data clearly are needed. It is hoped that interested readers will donate data and make instruments available for measurement, which will allow more comprehensive analyses to be published later.

REFERENCES

A LOW-COST PC-BASED TOOL FOR VIOLIN ACOUSTICS MEASUREMENTS

Lars Henrik Morset
Group of Technical Optics
Institute of Physics
Norwegian University of Science and Technology (NTNU)
NO - 7491 Trondheim, Norway
E-mail: morset@winmls.com

ABSTRACT
We present a PC-based technique for measuring important acoustical properties of the violin. The technique can be used with standard low-cost equipment to make a flexible tool for violinmakers. The problem of proper excitation by suitable transducers is discussed. We demonstrate how the tool can be used to estimate the radiation power and to detect the A1 mode of the violin.

INTRODUCTION
The violinmaker would benefit from performing some of the measurements that can be found in the violin acoustics literature. This requires inexpensive instrumentation that can be used to perform measurements in the workshop. Many violinmakers already have the basis of this instrumentation, which is a PC equipped with a sound card and a “multimedia” microphone. In this paper we demonstrate how the software WinMLS® turns a PC into a tool for measuring violin acoustics (a free time-limited evaluation version can be downloaded from www.winmls.com).

Using a PC-based tool the measurement is performed fast and the stored data can be analyzed and visualized in the most suitable way.

MEASUREMENT SOFTWARE
The software WinMLS is designed for general-purpose sound and vibration measurements and therefore has a wide range of applications. The software can easily be customized for measurement of the violin as all settings can be saved in setup-files. The data processing consists of the standard fast Fourier transform (FFT), including window functions and different types of frequency domain smoothing. Time domain and frequency domain may be viewed simultaneously, so we may for example change the windowing of the impulse response and immediately see how it affects the frequency response. The data and the plots can be saved in several formats. This and possibilities for measurement calibration makes it easy to exchange, share and compare results with other violinmakers. The results can be displayed and charted in all usual ways, which is useful for presentations and publications, e.g. all figures in this article have been made by WinMLS. There is cursor readout, several zooming possibilities and no limitation in the number of curves that can be plotted on top of each other.

TRANSUDCERS
The usual way of finding the linear properties of a system such as a violin is to excite the system with a known sound pressure level and then measured and processed. Different types of excitation signals may be used as long as the excitation signal contains enough power at all frequencies that are of interest. Mechanical transducers placed on the violin will influence the measured frequency range of the instrument. In some cases software may correct for this influence of the transducer(s), but it is still preferred to use transducer(s) that are not loading the violin.

Methods for exciting the violin
The most intuitive method of exciting the violin is bowing the strings. We might then determine the frequency response by recording the chromatic scale played by a violinist. A major problem here is to play each tone at equal force and velocity, as the violinist often tends to compensate weaker tones. Also, only the response at the partials of the played tones will be determined. If we play a glissando at constant speed using constant bow force on the G-string all frequencies above 196 Hz are excited. However, playing such a glissando is not simple, which makes it hard to get reliable results. It should be noted that the mounting of the violin is also a very important factor, and small changes will affect the measurements.

The traditional tapping method seems to be easier to control. A small hammer or the finger can be used to tap the violin. The resulting short pulse excitation will in most cases generate sufficient power over the desired frequency range. Our experiments also indicate that not even a hard hit causes enough non-linearity to have a negative effect on the measurement. Figure 1 indicates the
Figure 1. Three measurements of a Stradivarius model violin tapped using the fingernail at the G-string side of the bridge normal to the top plate. The response was measured using a microphone ~7 cm in front of the right f-hole.

Table 1. Comparison of excitation signals.

<table>
<thead>
<tr>
<th>Excitation signal</th>
<th>Repeatability</th>
<th>Price</th>
<th>Frequency range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Played glissando</td>
<td>Fair</td>
<td>Zero</td>
<td>&gt;196 Hz</td>
</tr>
<tr>
<td>Finger tapping</td>
<td>Good</td>
<td>Zero</td>
<td>200 - 3000 Hz</td>
</tr>
<tr>
<td>Impulse Hammer</td>
<td>Very good</td>
<td>High</td>
<td>20 - 20000 Hz</td>
</tr>
<tr>
<td>Magnet and coil</td>
<td>Very good</td>
<td>Low</td>
<td>50 - 8000 Hz</td>
</tr>
</tbody>
</table>

typical reproducibility of a finger tapping measurement. Exciting with a hammer mounted in a pendulum will give even better reproducibility since the hammer hits in exactly the same position and in the same direction.

A loudspeaker can also be used to excite the violin. However, this excitation is not suitable for sound pressure measurements as the loudspeaker also radiates directly to the microphone and it is difficult to get a sufficient cancellation of the direct sound. None of the excitation methods discussed above normalize the force applied to the bridge. This normalization is not in all cases necessary, for example when detecting resonance frequencies, but it is needed in order to find the correct level and also the shape of the frequency response since the spectrum of the excitation signal is not constant. An impulse hammer, which has a small built-in transducer for measuring the force, may be used [1], but this is a rather expensive transducer.

From numerous experiments we have found that the simplest (and cheapest) excitation for high quality measurements is to attach a small magnet with wax to the G-string side of the bridge and excite it using a coil. The output signal from the sound card is fed to the coil via an amplifier (e.g. a HiFi amplifier). This “exciter” is calibrated, i.e. the frequency response is normalized to an constant force. A so-called MLS is used as excitation signal because of its robustness to background noise. A comparison of this “coil-magnet” MLS-method and the impulse hammer method showed good agreement [2]. Table 1 summarizes this discussion.

Measuring vibration and sound pressure

Vibration can be measured either as displacement, velocity or acceleration. Using integration/differentiation any of these quantities can be found from the other. Note that displacement decays 40 dB per decade and velocity 20 dB per decade compared to acceleration, therefore transducers measuring displacement usually do not give sufficient noise immunity at high frequencies. A mechanical transducer has to be attached onto the violin body, while an optical transducer has the advantage that it is not in contact and thus will not add extra weight to the violin. We have not yet found a low-cost optical transducer that we are satisfied with, but it is probably only a matter of time before it will be available. As for sound pressure measurement, even inexpensive microphones are acceptable.

APPLICATION 1: ESTIMATION OF NORMALIZED RADIATION POWER

For all measurements mentioned in the application examples, the violin was mounted with pieces of rubber at the bottom part of the body and the upper part at the neck. The magnet-coil system described earlier in this paper was used to excite the violin on the G-string side of the bridge.

If we normalize the measured radiation power with respect to the excitation force squared we find the acoustic radiation for a given force input. The normalized radiation power is therefore an important parameter for characterizing a violin. For the reference normalized radiation power measurement, the violin radiation was measured at a distance of 1.05 meters and a resolution of 22.5 degrees over a sphere in an anechoic
chamber. Using this method we computed the normalized radiated power [3]. We thereafter investigated if the normalized radiation power could be estimated with fewer measurements and without using an anechoic chamber. We first measured the input mobility at the bridge. The real part of the input mobility, \( \text{Re}[Y] \), is proportional to the power transferred from the bridge into the violin. However, due to mechanical losses in the violin and violin mounting, only a part of the input power will be radiated.

In figure 2 and figure 3 examples comparing two violins are shown.

Usually the literature on violin acoustics [1] presents the magnitude of the mobility, \( |\text{Abs}[Y]| \), and not the real part, \( \text{Re}[Y] \), as we use here. By comparison, we found that \( \text{Re}[Y] \) gives a better estimate of the radiated power than \( |\text{Abs}[Y]| \). However, \( \text{Re}[Y] \) may be more difficult to compute since it depends strongly on the accuracy of the phase. We solved this by performing a reference calibration measurement of the “coil-magnet” system using a B&K force transducer. The Thibout violin shown in Figure 2 has a wolf tone at C (525.6 Hz). The peak of \( \text{Re}[Y] \) was found exactly at this frequency. At the wolf tone, the mobility of the bridge is so large that the bridge simply moves too much for the string to maintain a standing wave.

Above we have used \( \text{Re}[Y] \) to estimate the radiated power. But \( \text{Re}[Y] \) will not give an accurate estimate since the internal mechanical losses cannot be found. Also, devices for measuring velocity at the bridge may be costly and not so easily available as microphones. We therefore investigated if simplified near-field measurements of sound pressure could be used to give an estimate of the radiated power. Such measurements can be done with a single microphone. Near the violin the direct sound is dominant and the influence from the room can be neglected. 16 near field measurements were performed and averaged. The measurement positions were equally spaced with a resolution of 22.5 degrees around a circle 15 cm from the center of the violin. The thick curve in Figure 4 represents the reference normalized radiation power measurement. The thin lower curve, representing the

**Figure 2.** Normalized radiation power (thick curve) and real part of input admittance (thin curve) for a conventionally built violin (Thibout, 1843).

**Figure 3.** Normalized radiation power (thick curve) and real part of input admittance (thin curve) for an unconventionally built violin (Hagstro, 1996).

**Figure 4.** Normalized radiation power and averaged near field measurements on a Stradivarius copy. Thick curve – reference radiation power measurement. Lower thin curve – average over 16 microphone positions. Upper thin curve – average over 4 microphone positions.
average of all 16 near field measurements, shows a good agreement with the reference normalized radiation power curve below 3 kHz. The thin upper curve, representing an average of only 4 measurements, also shows a good agreement. Note that the absolute levels of the estimation measurements are not correct, but shifted to simplify the visual comparison. The near field measurements are performed along a circle, while the far field measurements are performed over an entire sphere. Using only the measurement at a circle in the far field for comparison gives very similar results. The radiated power may be measured accurately by measuring and integrating the outgoing intensity vector in the near field around the violin, but even the integrated pressure, which is measured in this study, gives, as is shown experimentally, an estimate of radiated power. (Radiated power is proportional with sound pressure squared at distances to the violin larger than the linear dimensions.)

**APPLICATION 2: DETECTING THE A1 MODE**

There exist several methods for detecting violin modes as described in the literature [4]. We will show an example of detecting the A1 mode using our tool. The excitation and violin mounting were as described above. We determined the A1 resonance by measuring the sound pressure inside the violin and a low-cost microphone small enough to slip through the f-holes was used for this purpose.

Figure 5 shows measurement results for the microphone inside (placed in the middle of the lower circle) and outside (placed 1 cm from left f-hole) the violin. We found that the detection was improved if the nearby resonances were damped e.g. by holding the thumb near the left bridge foot and the other fingers at the back plate. The thickest curve, representing the inside-microphone with the damping applied, shows a peak at 455 Hz, which is the A1 resonance. The thick curve represents the microphone placed just outside the right f-hole and with no damping applied. The thin curve, representing the inside-microphone with no damping applied has a weak indication of the same peak, but other modes nearby makes the 455 Hz peak less obvious. Note that the damping may shift the resonance frequency somewhat.

**CONCLUSIONS**

A low-cost tool based on a PC and standard instrumentation has been presented for performing vibrational and acoustical measurements on violins in a workshop. Modes can be detected and measuring the real part of input mobility or averaging of near-field measurements seems to provide estimates of the normalized radiation power. Further work will be to optimize the near-field method and to verify the results using more measurement objects.

**ACKNOWLEDGMENTS**

The author wishes to thank professors U. P. Svenson, A. Krokstad, O. J. Løkberg, violinist A. Larsen, violinmaker E. Grimstad and Hagstro Fioliner.

**REFERENCES**

**Impact of String Stiffness on Digital Waveguide Models of Bowed Strings**

Stefania Serafin, Julius O. Smith III  
CCRMA, Department of Music  
Stanford University  
Stanford, CA  
serafin@ccrma.stanford.edu, jos@ccrma.stanford.edu

**ABSTRACT**
We propose a digital waveguide model of a bowed string instrument that accounts for string stiffness. We show how dispersion due to string stiffness can be accurately modeled and how it improves the quality of the synthesis.

**INTRODUCTION**
Physical models of musical instruments have achieved a level of understanding that allows to implement in real time most of the phenomena that appear in real instruments. Bowed string instruments have been also deeply studied starting from the results obtained twenty years ago by McIntyre, Schumacher and Woodhouse in [4].

In this paper we propose an approach to model stiffness in virtual bowed string instruments. As described in [1], stiffness rounds the sharp corners that are a characteristic of the ideal Helmholtz motion. This significantly affects the quality of the sound synthesis. We therefore believe than an accurate model of a bowed string instruments needs to take into account the stiffness of strings.

**A WAVEGUIDE MODEL OF A BOWED STRING**
We developed a digital waveguide model of a bowed string as and extension of the model proposed in [9]. Transversal and torsional waves propagation in the strings is modeled using fractional delay lines, and losses at the bridge and along the fingerboard are modeled using lowpass filters.

The friction curve representing the bow-string interaction is modeled using a decaying hyperbola. This enables to solve analytically in real-time the coupling between the bow and the string, and at the same time to obtain the same waveforms as the one measured in real instruments, as shown in [7]. Recent results in [8] show that a more complex behavior happens at the bow point, but we choose to keep the bow-string interaction of this form in order to have a simulation that runs in real time also in a personal computer. The overall structure of this basic model is shown in figure 1.

The bow excites the string in a single point, from which torsional and transversal waves propagate toward the nut and the bridge respectively. They are reflected at the extremities, where lowpass filters model losses at the bridge and at the nut for both transversal and torsional waves.

Then the waves are reflected and meet again at the bow point, where solving the coupling between the bow and the string creates the new waves that propagate toward the extremities.

As described in [7], when a proper combination of the parameters that drive the bow, i.e. the bow force, bow velocity and bow position is chosen, this model allows to obtain the Helmholtz motion, which is the well-known motion of a bowed string that every player tries to achieve.

**IMPROVING THE MODEL**
The model described in the previous section can be improved in order to account for

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**Figure 1.** Block diagram structure of a simplified physical model of a bowed string.
other phenomena that appear in real strings. One of these improvements consists of accounting for string stiffness. Usually in instruments like the violin the role of stiffness is almost negligible, so the corresponding models do not account for it. However we noticed that accounting for stiffness improves the quality of the model, especially in instruments like the cello with a higher stiffness coefficient, as we will show below.

**MODELING THE STIFFNESS OF THE STRING**

Let's first consider a lossless stiff string of length $L$, mass $m$ submitted to a strength $T$. The transverse displacement of the string is given by (see [5]):

$$\frac{1}{\rho} \frac{\partial^2 y(x,t)}{\partial t^2} - \frac{\partial^2 y(x,t)}{\partial x^2} = \frac{1}{\rho} \frac{\partial^2 y(x,t)}{\partial t^2} \frac{\partial^2 y(x,t)}{\partial x^2} \quad [1]$$

where $c = (T / \rho)^{1/2}$ is the velocity of transversal waves in case of no stiffness, $\rho = m / L$, and $k = (T / EI)^{1/2}$ represents the effect of stiffness, $E$ is the Young modulus, $I$ the inertia. Partial solutions of this equation are plane waves of the form

$$y(x,t) = Ae^{i(\omega t + kx)} \quad [2]$$

in case the dispersion relation

$$\omega = c_k \sqrt{1 + (k / \kappa)^2} \quad [3]$$

is satisfied, where $\omega$ is the angular frequency.

Since the string is attached at both ends, $k$, which is the wave character, can take only the discrete values $k_n = n\pi / L$, so the frequency of the wave vibrations is discretized as ([2]):

$$\omega_n = n\omega_c \sqrt{1 + Bn^2} \quad [4]$$

for $n=1,2,...$ where $B = (\pi / KL)^2$ denotes the inharmonicity factor of the string and $\omega_0 = \pi c / L$ is the fundamental frequency in case of no stiffness.

The relationship in equation 4 states that the modes of transversal vibration are not in harmonic frequency ratio, but are shifted upwards. An example of this phenomenon is shown in figure 2, where a string with fundamental frequency $f_0 = 147$ Hz and inharmonicity factor $B = 4e - 4 Nm^2$ is considered. Note how, according to equation 4, the shift increases at higher frequencies.

**DESIGNING ALLPASS FILTERS FOR DISPERSION SIMULATION**

In order to account for dispersion in the digital waveguide model of a bowed string, we choose a numerical filter made of a delay line $q^{\Delta t}$, and a $n$-order stable all-pass filter.

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**Figure 2.** Shift of partials for a string with $f_0 = 147$ Hz, $B = 4e - 4 Nm^2$

Horizontal axis: partial number, Vertical axis: frequency (Hz).

**Figure 3.** Frequency error (in cent) after the allpass filter approximation, for a cello D string, with $B = 4e - 4 Nm^2$.

**Figure 4.** Block diagram of a digital waveguide model of a bowed string including the allpass filters for stiffness simulation.
Figure 5. String velocity at the bridge in the case of a perfect flexible string. Horizontal axis: time in samples, vertical axis: amplitude.

Figure 6. String velocity at the bridge in the case of a stiff string. Horizontal axis: time in samples, vertical axis: amplitude.

Figure 7. String velocity at the bridge after a steady-state motion is achieved. Top: stiff string, bottom: flexible string.

\[ H(q) = \frac{q^s P(q')/P(q)}{P(q) = p_0 + \cdots + p_n q^{n'} + q^s} \]

and \(s\) and \(n\) are appropriately chosen.

As proposed in [3], we want to minimize the \(\infty\)-norm of a particular frequency weighting of the error between the internal loop phase and its approximation by the filter cascade:

\[ \delta_0 = \min_{A \rightarrow \infty} \| W_D(\Omega) [\varphi_D(\Omega) - (\varphi_D(\Omega) + r(\Omega))] \|_\infty \]

where \(\varphi_D(\Omega)\) is the phase of, \(H(\omega)\) and \(W_D(\Omega)\) is the frequency weighting \(W_D(\Omega)\) is zero outside the frequency range, i.e. \([\Omega_1, \Omega_N]\). From an acoustical point of view, it is important to have a frequency weighting that approximates the way the auditory system perceives the difference between original and simulated phase dispersion.

In order to restrict the approximation to the first few thousand hertz, Rocchesso and Scalcon in [6] propose to use frequency warping ([10]). In our implementation, we choose a weighting function that stresses the accuracy at low frequencies.

**SIMULATION RESULTS**

We consider a cello D string (147 Hz), with stiffness coefficient \(B = 4e - 4 Nm^2\), which is a value suggested in [11]. Figure 3 shows the results obtained approximating the phase dispersion using three sixth order allpass filters in cascade. Note how the error is below two cents in the frequency range [0.4200 Hz], which means that the approximation is correct for the first 28 partials of the cello D string.

**APPLICATION TO THE BOWED STRING MODEL**

We inserted the estimated filters in the string loop as shown in figure 4. The bridge filters for losses and then by the dispersion filters first filter the transversal waves propagating toward the bridge. Note how, since the string is a linear system, inverting the three blocks corresponding to the propagation of the waves in the bridge side of the string would not change the results of the simulation.

**Influence of Stiffness in Virtual Bowed Strings**

We run our digital waveguide model using the same cello D strings as before, with bending stiffness \(B = 4e - 4 Nm^2\). The string, starting from rest, is excited by a constant bow velocity \(v_s = 0.05 m/s\), a bow force of \(f_b = 0.2 N\) and a normalized bow position of 0.1, where 0 represents the bridge while 1 represents the nut. Figure 5 shows the result of this simulation where the string velocity at the bridge in the case of the flexible string has been captured for about 20 periods from the attack. Figure 6 shows the simulation with the same parameters but in the case of the stiff string. Notice how a regular Helmholtz motion is achieved earlier in the case of the flexible string.

The visible difference on the attack waveform is significantly audible.
Accounting for stiffness improves the quality of the synthesis even when the input parameters are stationary over time. The waveform on the top of figure 7 shows the velocity of a string with stiffness coefficient $B = 4e - 4 N m^2$, while the waveform on the bottom shows a string with no stiffness. Note how the Helmholtz corner is sharp in the waveform on the bottom while it is rounded in the one on top. This difference has again a significant impact on the quality of the sound synthesis.

CONCLUSIONS

In this paper we proposed a digital filter design method that can be used to model stiffness in virtual bowed string instruments. This method allows obtaining good approximations for rather high stiffness values, which makes it suitable to be used also to model inharmonicity in instruments where stiffness is considerably noticeable like, for example, the piano.

REFERENCES

QUESTION AND ANSWER FORUM

We invite questions ranging from acoustics to violin making to any other subject appearing in the CAS Journal. Questions will be fielded to authorities in the subject, and answers promptly returned by email. A selection of the question and answer threads that are of broad interest will appear in the Journal. Please submit your questions by email to caseditor@aol.com. Alternatively, letters, phone messages, and faxes can be addressed to the CAS Office, 55 Park Street, Montclair, NJ. 07042 USA, (973) 744-0371 (phone); (973) 744-0375 (fax); Catgutar@msn.com.

Information Please: Foam for Composite Soundboards

I was inspired by Charles Besnainou's recent note regarding use of composite materials in musical instruments (reference to CASJ vol. 4, November 2000, p. 9) to begin exploring the possibility of making a composite soundboard for a harpsichord. I have now read most of the references from that article (the Douan thesis is apparently not available to the general public) and I have been able to locate sources for most of the supplies and equipment mentioned in the article. However, I am having trouble finding a source for suitable foam core material. A large number of different foam compositions come up on a web search, and most of them appear not to be readily available as thin sheets. More detailed information, from Charles or from other readers, regarding suitable foam compositions would be very helpful, even if only as key words for a more focused search. —James Bunch, Los Alamos, N.M.

Charles Besnainou replies:
I have already tested many different foams (available in 1, 2, 3 mm thickness) such as: DIVIYICELL: info@diabgroup.com, HEREX/KAPEX/AIREX: www.alusuisse.com, and ROHACELL: www.roehm.com.

Joseph Curtin replies:
I've worked with Charles Besnainou developing violins and violas using composite materials. Though I have little advice about specific foams, other than trying the ones he suggests you might also try such low-density woods as Balsa. Though of course this brings in some of the unpredictability of wood, supply seems plentiful and in a considerable range of densities. From my limited experience, the acoustical properties seem good. It can be laminated in a whole variety of configurations to form very light and stable plywoods that are then reinforced with graphite. As a way of getting started with Charles, I gave him a 3" by 8" strip of good spruce and good maple and asked him to make up similarly dimensioned samples using graphite in combination with various types of foam. Tapping each with a knuckle was a good way of comparing, as a first rough estimate, their acoustical properties. I chose the combination that came closest to the wood I liked, and proceeded from there. Best of luck!

Editor's note:
You might also check suppliers for making lightweight aircraft. Composite materials have been used for aircraft for many years, and the materials are all available for home use. An excellent source is Aircraft Spruce & Specialty Company, 1-877-4-SPRUCE, International: +909-372-9555, www.aircraft-spruce.com. Their 600-page catalog (free in USA; $8 Canada; $15 international) is a treasure trove of sitka spruce, balsa wood, carbon graphite, epoxies, vacuum bagging supplies, vacuum pumps, books on composite sandwich construction methods, and even a composite materials practice kit. They sell several types of foam including divinycell and Last-a-foam (although the thinnest sheets listed are 5 mm). They also teach 2-day workshops in composite construction (www.sportair.com). Seems like composite harpsichord & fiddle makers could learn a lot from these folks!
BOOK REVIEW

The Dance Master’s Kit (La Pochette du Maître a Danser) by Claude Lebet

Jeff Loen

This lovely, well-crafted book on the smallest (length, 32-55 cm) of bowed instruments, known as “kits” or “pochettes”, removes much of the mystery from instruments wrongly considered by some to be mere curios, toys, or child-size violins. Claude Lebet shows us what museum curators have long known, that kits are masterpieces of lutherie, as well as being intriguing acoustical paradoxes that have some interesting implications for instrument scaling theory and the New Violin Family.

During the 17th and 18th centuries, wealthy patrons commissioned kits from leading violin makers for use by the most esteemed dance instructors, often for royal dance ensembles. Many of the instruments are highly ornamental, but Lebet points out that kits are more than just ornaments. Basically they were working tools, with the advantage that they easily slipped into the large pockets of the bulky coats of the Baroque era.

Kits are a direct descendent of the ancient rebec. They evolved into two main types, boat-shaped and violin-shaped kits. Most boat-shaped kits were carved from a single piece of hardwood (often flamed maple), and then a spruce or cypress top and Baroque-style fingerboard was added. Many kits of both types sport carved heads instead of scrolls, although scrolls and shield designs are common. In contrast, most violin-shaped kits were apparently made the same way small violins were made (often using an interior mold, with corner and end blocks), except that exceptionally long necks were installed. Highly ornamented kits were made from exotic woods (ebony, walnut) and precious materials (ivory, tortoiseshell, gold, silver, mother-of-pearl). Photographs and diagrams are presented for the “Clapisson” kit (1717), one of several kits made by Antonio Stradivari, which appears something like a violin viewed in a thin-man mirror at an amusement park. In addition to kits, the book provides photos and descriptions of rare surviving bows (by Tubbs, Dodd, and others), made in the Baroque style out of snakewood, teak, or pernambuco, with ivory, boxwood, or ebony frogs. In a chapter devoted to the interesting story of 19th Century forgeries made at a time when kits were in great demand, the author names specific kits in famous museums that he believes are, in fact, fakes.

The strongest aspects of the book are the historical treatment, particularly of the changing role of dance (and consequently, kits) in European culture during the Renaissance and Baroque eras, and the outstanding illustrations that include many interesting paintings and drawings from the 17th and 18th Centuries. Something conspicuously lacking in the book, however, is a discussion of acoustical attributes of kits. The author doubts that kits were tuned an octave above the violin (as had been suggested by other authorities), like the treble violin in the New Violin Family. It is doubtful that kits were tuned this way because they are invariably strung with four gut strings, the highest of which could not attain the pitch of a high E (1320 Hz) without breaking. Most kits have string lengths of 22-27 cm (similar to 1/4 to 1/2 size violins), although Stradivari’s 1717 kit has a string length of about 33 cm, which suggests that it could have been tuned as a full-size violin. Some acoustical compromises were obviously made in favor of portability, although it seems possible that kits sound better than one might expect for instruments with rather short string length and small air volume. For one thing, it is implied throughout the book (especially in contemporary illustrations) that kits projected well enough to play them in groups of people, outdoors, and in noisy surroundings. Furthermore, kits were played mostly by accomplished professional musicians who presumably distained a shrill, weak tone.

Makers interested in building kits will find photographs of dozens of kits of a wide range of design and quality (plus a few interior views), basic dimensions, scale drawings of four different models, a sketch of an assortment of bridges, and a drawing of bass bar and soundpost locations. A comprehensive list of hundreds of kits in museum collections throughout the world is included, as well as a good bibliography.

PUBLICATION INFORMATION
ACOUSTICAL SCIENCE AND TECHNOLOGY
Eiichi Obataya, Yoshitaka Ohno, Misato Norimoto, and Bunichiro Tomita, 2001, Effects of oriental lacquer (urushi) coating on the vibrational properties of wood used for the soundboards of musical instruments: vol. 22, no. 1, p. 27-34.


AMERICAN LUTHERIE


Paul Schuback, 2000, An American in Mirecourt, Violin making as learned from Rene Morizot: no. 63, Fall 2000, p. 20-33

John Monteleone, 2000, Designing the archtop guitar for sound: no. 62, Summer 2000, p. 6-17

Dana Bourgeois, 2000, Still voicing, Still dreaming: no. 61, Spring 2000, p. 28-33.

David Riggs, 2000, Clark “Neo-Irish” harp: no. 61, Spring 2000, p. 34-37.

JOURNAL OF THE ACOUSTICAL SOCIETY OF AMERICA

INSTRUMENTENBAU-ZEITSCHRIFT

MICHIGAN VIOLINMAKERS ASSOCIATION NEWSLETTER

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STRINGS

VIOLIN SOCIETY OF AMERICA JOURNAL

VIOLIN MAKERS ASSOCIATION OF ARIZONA INTERNATIONAL JOURNAL
Jeff Loen, 2001, Removing the mystery from violin graduations - A progress report on the “viograds project”: vol. 43, no. 4, April 2001, p. 6-10.

INTERESTING WEB PAGES

by Kelvin W. M. Scott
e-mail: kelvins@umich.edu
web site: www.ksviolins.com

Note: The following list of web links, compiled by Kelvin Scott, is for our member’s general information. Please note that the Catgut Acoustical Society does not necessarily endorse the concepts and products mentioned on these pages. Also, it is likely that some internet addresses will change and some links may cease to function because the internet is a dynamic, evolving medium. We hope to place a permanent links page on the Catgut Acoustical Society web page (www.marymi.edu/~cas). Please look there for additional information and a more current list of links. – Editor

ACOUSTICS:
http://www-ccrma.stanford.edu/marl/ The home page of The Musical Acoustics Research Library (MARL), a collection of independent archives and libraries in the field of musical acoustics research.
http://www.marymt.edu/~cas/research/articles/modetune/ “Mode Tuning for the Violin Maker” by Carleen M. Hutchins and Duane Voskuil.
http://www-ccrma.stanford.edu/links/ An interesting list of links on musical acoustics, assembled by the Center for Computer Research in Music and Acoustics at Stanford University.
http://www.ciarm.ing.unibo.it/ The home page for the Interuniversity Center of Acoustics and Musical Research.

THE FOLLOWING THREE PAGES ON VIOLIN ACOUSTICS ARE PROVIDED BY THE UNIVERSITY OF NEW SOUTH WALES DEPARTMENT OF MUSICAL ACOUSTICS:

SOFTWARE FOR ACOUSTICAL STUDIES:
General Software
http://www.hitsquad.com/smm/ General sound programs for Mac and PC, including spectrum analyzers, oscilloscopes, metronomes, ear training, signal generators and sound editors; mainly shareware and freeware. A great place to start.

Signal Analysis Tools for the PC
http://www.syntrillium.com/cooledit/ A site providing demos of Cooledit by Syntrillium Software.

Signal Analysis Tools for the Macintosh
http://www.channld.com/software.html A site providing demos of Mac the Scope 4.0 by Channel D Corporation.

MUSEUM & LIBRARY COLLECTIONS
http://www.ued.edu/smm/ Shrine to Music Museum in Vermillion, South Dakota. A marvelous collection of stringed instruments including many Cremonese masterworks. This website includes a comprehensive index of all instrument in the collection.
http://www.vam.ac.uk/ The musical instrument collection of the Victoria & Albert Museum includes a range of stringed instruments, including one Stradivari.
http://www.ashmolean.ox.ac.uk/ Ashmolean Museum. Home to the Hill Collection and Stradivari's "Messiah."
http://www.si.edu/resource/faq/nmah/strad.htm Smithsonian Museum. This collection includes the 1701 "Servais" and the Herbert R. Axelrod Stradivarius Quartet of ornamented instruments.
http://www.metmuseum.org/collections/department.asp?dep=18 The musical instrument collection of the Metropolitan Museum of Art, including one Stradivari.
http://www.oberlin.edu/library/colldev/SCP/goodkind.html Home page for The Herbert K. Goodkind Collection at Oberlin College Library, containing an excellent collection of books relating to the violin family of instruments.

VIOLIN FAMILY DISCUSSION BOARDS
http://presto.maestronet.com/discussion/discuss_frame.cfm Maestronet's Discussion Boards. Maestronet offers web surfers four specialized bulletin boards, providing a lively environment in which to discuss violin making, violin performance, violin politics, and much more.
http://www.pa-roots.com/cgi-bin/message/sheila.cgi? Sheila's Violin Corner BB. A friendly place to share ideas on a wide range of educational topics relating to the violin family.
http://www.cello.org/bulletinfooter.htm Internet Cello Society BB. A bulletin board devoted to the Cello.
http://www.stringsmagazine.com/ubb/cgi/forumdisplay.cgi?action=topics&number=1&SUBMIT=Go Strings Talk BB A general discussion group in which a wide variety of topics regarding violin family are explored.
http://www.forumboard.net/1761/ The Soundpost Online. A discussion forum devoted to the examination of topics relating to violin making, restoration, repair, and appraisal.

MISCELLANY ON THE VIOLIN AND VIOLIN MAKING
http://www.moesandmoes.com/docs/strings.html How to Look For & Evaluate The Work of Today's Instrument Makers
http://www.hometown.aol.com/viograds/index.htm An interesting collection of contour maps depicting graduation patterns of classical instruments
http://www.tarisio.com/ This web site for Tarisio auctions and the archive of instrument photographs it contains is a valuable resource for those interested in viewing high-resolution images of fine violins.
http://www.vsa.to/information/journal/hargrave.htm An article on classical Cremonese edgework by Roger Hargrave.
http://www.msen.com/~violins/news.html A collection of interesting articles by luthier Joseph Curtin, touching upon a range of subjects from experimental violin making, to violin setup, to violin acoustics.

THE FOLLOWING FIVE LINKS ARE SAMPLE ARTICLES FROM THE SOUTHERN CALIFORNIA ASSOCIATION OF VIOLIN MAKERS BULLETIN
http://www.scavm.com/Fulton.htm Two Articles on Finishing Instruments by Bill Fulton.
http://www.medimaging.com/uncon/uncon.html Information on the use of CT scanning technology for the sake of creating non-invasive images of instruments.
http://www.music.ed.ac.uk/euchmi/cimcim/iwd.html#iwdc A list of 664 technical drawings of musical instruments, including a number of stringed instruments.
Interesting Web Pages

GENERAL ORGANIZATIONS OF INTEREST TO MAKERS AND PLAYERS

Internet-based Organizations

http://www.maestronet.com A commercial site offering a range of resources for players and makers, including price histories, registry of missing instruments, sheet music and more.


http://www.das-streichinstrument.de/links/ "Das Streichinstruments," a site containing links to numerous on-line articles regarding the violin family.

http://www.violink.com Violink. A commercial site that maintains a wide array of useful links and resources.

http://www.vanzandtviolins.com/ Violinmaker David T. Van Zandt's Links Page. This venerable set of links is still one the best in cyberspace.

http://www.cello.org/index.htm Internet Cello Society


Other Organizations


http://www.afvbm.com/ American Federation of Violin and Bow Makers.


http://www.lutesoc.co.uk/ The Lute Society.


http://www.hfiaa.org/ Hardanger Fiddle Association of America.

http://www.vdgs.demon.co.uk/ Viola da Gamba Society (United Kingdom).


International Musicological Society 17th International Congress (IMS 2002)

IMS 2002 will take place at the Monsignor Sencie Institute of the Catholic University in Leuven, Belgium, from 1 to 7 August 2002. The Congress will offer symposia on eight themes, as explained in detail on the IMS website (http://www.ims-online.ch/) and on flyers available on request from the Secretary General of the IMS (fax 41-1-923-1027, e-mail imsba@swissonline.ch):

1. Hearing - Performing - Writing
2. The Dynamics of Change in Music
3. Who Owns Music?
4. Musica Belgica
5. Musical Migrations
6. Form and Invention
7. Instruments of Music: From Archeology to New Technologies
8. Sources

Please consult the web page or contact the Chair of the program committee for more information:
Prof. Barbara Haggh: IMS 2002
Clarice Smith Performing Arts Center
School of Music, Room 3110-C
University of Maryland
College Park, Maryland 20742 USA
Fax: (1) 301-314-9504

First Announcement
International Symposium of Musical Acoustics Mexico City

Theme: “Musical Acoustics and an Interactive Musical Instruments Museum”

ISMA Mexico City will be held from 9th – 13th December, 2002, in Mexico City. The symposium will have as its main theme: the relevance of musical acoustics in an interactive museum of musical instruments. The thrust of the symposium is creating an interest in musical acoustics among young and old generations by having audiences visiting museums of musical instruments learn the principles of musical acoustics in an attractive and interactive manner; and by assisting them in understanding the functioning of musical instruments. This will bring together musical acousticians, musical museologists and museographers, curators and directors of museums of musical instruments; musicologists; biologists doing research on the acoustical properties of different woods, neuroscientists, and educators. The symposium will also cover the classical topics of musical acoustics meetings, such as the acoustics of stringed, wind and percussion instruments, as well as the acoustics of the human voice, musical psychoacoustics, musical reproduction, electronic music, and room acoustics.

This will be a satellite symposium of the 144th Meeting of the Acoustical Society of America, the 3rd Iberoamerican Congress of Acoustics and the 9th Mexican Congress of Acoustics, which will jointly meet from 13th November to 6th December, 2002, in Cancún, Mexico.

Mexico City lies on the crossroads of the Americas. It was founded by the Aztecs in 1325 and later became the capital of New Spain and Independent Mexico. It is one of the outstanding cultural centers of Latin America. This city is situated in south central Mexico and is one of the largest metropolitan areas in the world, with over 20 million people. Mexico City is connected by air with the main cities of Europe, North America, South America, the Orient and the Pacific, as well as the main tourist resorts of Mexico. The temperature in December ranges between 8°-20°C, and the weather is dry and sunny.

Languages: The official languages of this symposium will be English and Spanish.

Transportation: For those persons attending the Cancún meeting there will be a special air connection to Mexico City.

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ISMA Mexico City
Escuela Nacional de Música
Universidad Nacional Autónoma de México
Xicoténcatl 126
Del Carmen, Coyoacán
04100 México, D.F.
(525) 651-5187 (tel)
(525) 680-3746 (fax)
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