VIOLIN TAILPIECE DYNAMICS: DESIGN AND FUNCTION

University of Victoria

DEPARTMENT OF MECHANICAL ENGINEERING

Author(s): Oliver Pirquet

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Abstract (Executive Summary)

This report shows the ability of a violin tailpiece design to tune its resonant frequencies in order to impact the emitted sound of the violin. Here the modal pattern associated with a particular resonance of the tailpiece and its ability to move this resonance between frequencies of 617 and 731 Hz is shown. A slightly shorter tailpiece of the same design was then mounted on a violin. The ability of the tailpiece to impact an identified resonance of the violin was demonstrated by tuning the tailpiece to match the resonance of the violin. The magnitude of the accelerance recorded at a peak resonance on the violin bridge was reduced by 20 dB from its original value demonstrating the ability of the tailpiece design to affect violin resonances.

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1. INTRODUCTION

The Violin tailpiece has been known by instrument makers for hundreds of years to affect the sound of the instrument. The reason for this, it is postulated here, is that the tailpiece acts as a sprung mass damper working to absorb the driving force of the instrument preferentially at its resonant frequencies. The tailpiece then can be used to affect unwanted audible frequencies of the violin. The most notable case in the instrument world where an unwanted frequency interaction occurs is the “wolf note” (described by [1]) whereby a strongly emitting resonance of the violin body interacts with a note that the musician is trying to play. The result is a beat frequency that moves around the desired note but does not allow it to be played. A tailpiece has been created that has the ability to change the moment of inertia about its center by moving a mass along its length which it is thought will allow the tuning of at least one of its resonant mode shapes. The creation of a tuneable tailpiece that would work as a filter to tune out or to shift unwanted resonances of the instrument would provide the maker and the musician with a way to control undesirable emitance from the violin system. Here the ability of this tailpiece to be tuned and further how it affects the output of the violin body will be assessed. Modal analysis (of the kind performed in [2]) will be used to understand how the tailpiece is moving at frequencies of interest without the influence of the instrument. Tests will then be done using frequency response functions (FRFs) to assess the effect the tailpiece has on the instrument via measurements taken at the bridge of the violin. The present study will be focusing on 0-1000 Hz frequency range in its view of the frequency response of both the bridge and the tailpieces.

2. GLOSSARIES

- *Violin family instrument*- includes cello's, violins, bases, and viola
- *Violin anatomy* – (see figure 1 below)

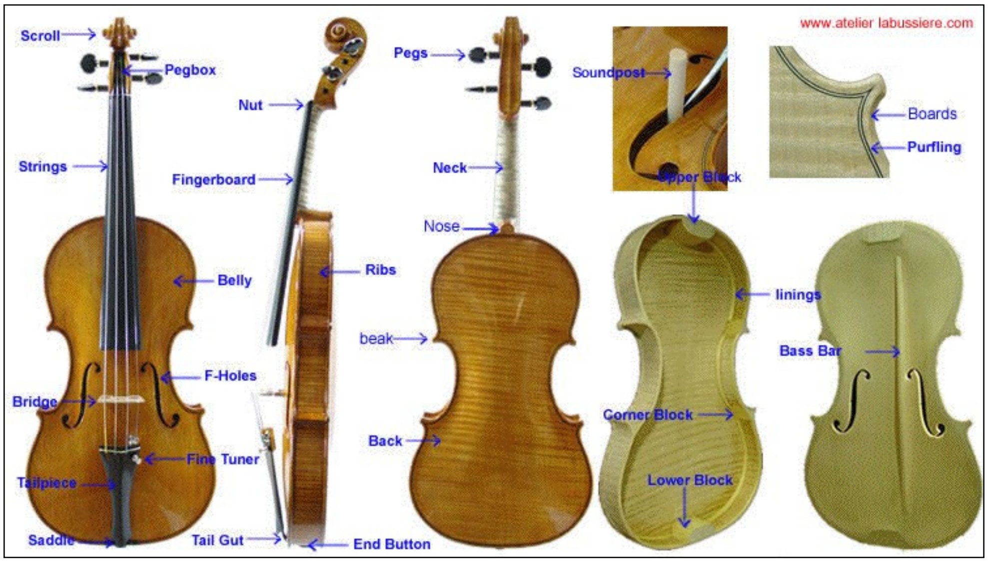

Figure 1 diagram of violin anatomy

- *French pattern tailpiece* –Rounded body (fig 2)

Figure 2 french pattern tailpiece http://www.harmonie.net/us/catalogue/a-cordier.html

- *Tail chord* – Chord that runs from the bottom of the tailpiece to the end button
- *After-length* – Length of string between the bridge and the break point on the tailpiece
- *Node* – location of zero displacement in a vibrating body.
- R_b - Resonance recorded at the bridge of the instrument with its peak between 500 and 700 Hz see 4.3.
- R_t - Resonance identified in the tailpiece design tested here between 500 and 800 Hz exhibiting a characteristic rocking motion see 4.1.

3. Materials and Methods

3.1. Tailpiece design

The tailpiece design used in this study was created by Ted White of Arbutus fittings. The design shown in figure 3 is an example of a French style tailpiece with a channel cut in the underside to allow the sliding of a weight which has the effect of changing the mass distribution of the tailpiece about the axis drawn in figures 3, 5, and 6. Figures 5 and 6 show the tailpiece lengthwise dimensions that will be referenced in the report. Variations in the mass position in the channel on the underside of the tailpiece will be described as percentages of the channel length (47 and 46 mm for the 112 and 108 mm designs respectively) 0% being right at the axis of rotation and 100 % being at the end of the slot or 47/46 mm down the channel towards the end of the tailpiece.

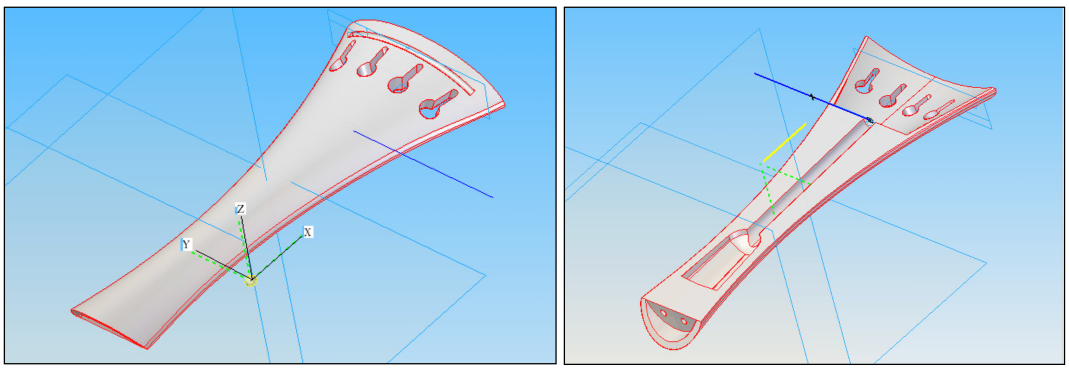

Figure 3 Tailpiece top (left) and underside (right) showing axis of rotation

3.2. Support for the Tailpiece

The tailpiece will be supported using the following parameters;

- Tail-chord length - 5.66 mm (112 mm design) and 9.66 (108 mm design)
- After length - 50 mm (constant for both)
- String type - D'ddario Helicore 4/4 medium tension strings. String tension was standardized by tuning to standard violin string notes (see figure 4).

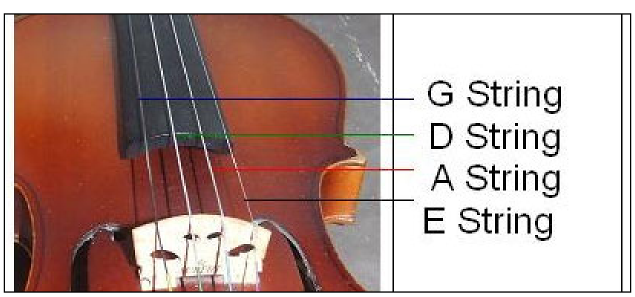

Figure 4 Notes of the violin strings

3.3. TAILPIECES USED IN EXPERIMENT

3.3.1. 112 MM TAILPIECE

For the preliminary mode shape characterization, the 112 mm tailpiece was used on the dead rig with the support parameters laid out in 3.2.
An assumption is made that the modal shapes exhibited by the 112 mm tailpiece are analogous to those observed in the 108 mm tailpiece.

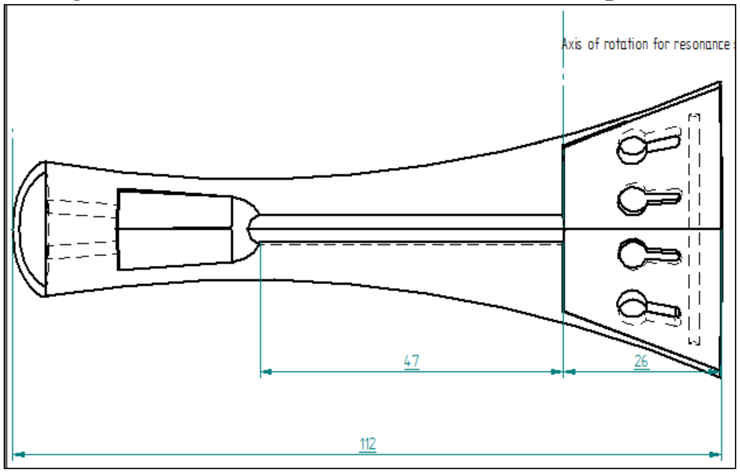

Figure 5 112 mm tailpiece design

3.3.2. 108 MM TAILPIECE

The 108 mm tailpiece design was used to test the ability of the tailpiece resonance to affect a resonance of the Violin. The set up on the violin was identical to the set up of the 112 mm tailpiece except for a change in the Tail gut length of 4mm due to the reduced length of the tailpiece. The after length was kept at a constant 50 mm.

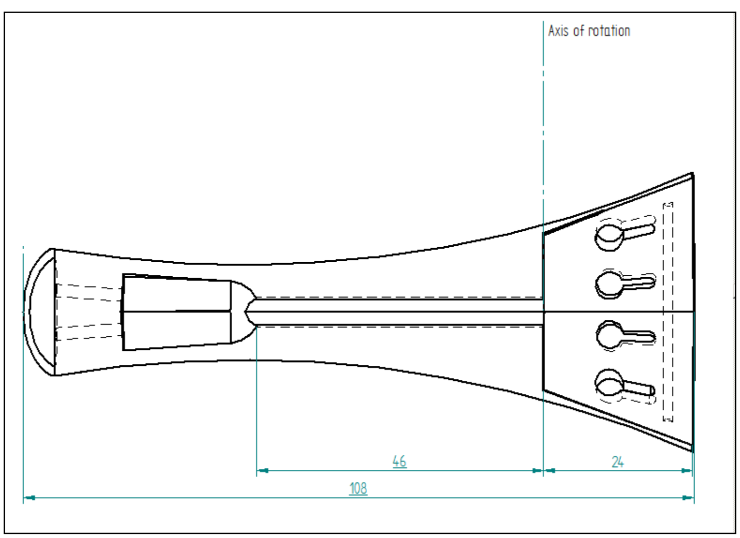

Figure 6 108 mm tailpiece design

3.4. Modal Analysis

3.4.1. Dead rig

In order to remove the tailpiece from the effects of the instrument a "dead rig" (fig 7) mimicking the instrument geometry (see appendix) was built from laminated birch plywood. The choice of material was in order to provide a structure with dimensional stability and increased mass relative to the instrument. The dead rigs were then mounted on marine engine compartment insulation to isolate the rig from the testing table.

Figure 7 Violin dead rig set up

3.4.2. Testing method

The procedure for modal analysis is done by;

- Laying out the geometry of the tailpiece and specifying measurement locations using the "mode shape" software (fig. 8).
- Mounting the accelerometer (fig. 8) in the appropriate plane. The accelerometer was mounted using wax provided by the manufacturer of the device and when measurements were taken at the bridge of the violin super glue was used to assure a solid connection.
- Recording the complex (magnitude and phase) frequency response functions (FRF'S) of the objects response to applied impulse signals in the form of hammer strikes at the measurement locations. The FRFs are taken as the average of 5 hammer strikes in order to reduce errors in measurement.

The modal hammer used in these experiments was a PCB 4.8 g hammer, model 086E80, and the vibrations were recorded using a Dytran uniaxial 0.6 g piezoelectric accelerometer model 3225E. In order to standardize the hammer strikes a hammer rig (fig. 9) was constructed providing a 6 degree of freedom tool for orienting the hammer head to the 3 principle directions on the tailpiece. The Software for acquisition and post processing of this data has been created and donated free of charge by George Stoppani.

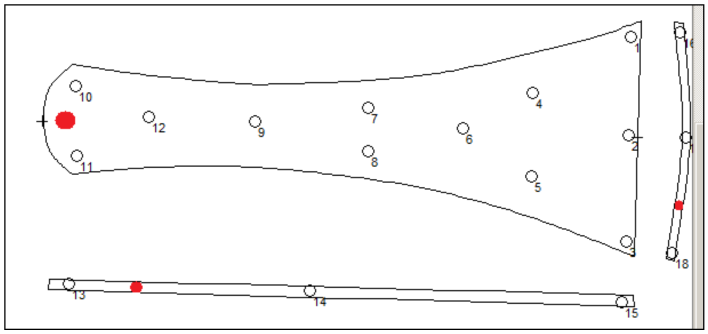

Figure 8 hammer and accelerometer locations for test tailpiece

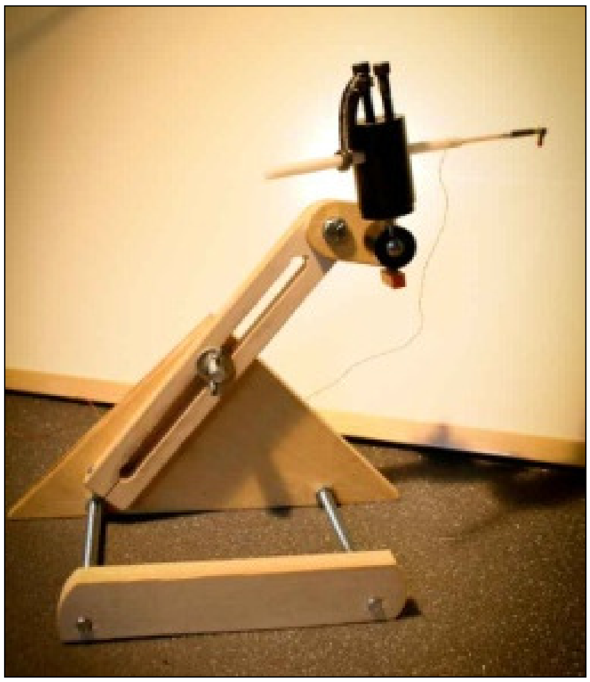

Figure 9 Hammer rig

3.4.3. POST PROCESSING

In order to determine the resonant peaks of the object being analyzed, the program "mode fit" provides a way in which to locate probable resonances and to fit curves to the acceleration response for each point. The fits were then saved and visualized in "mode shape" to view the patterns of movement at a particular frequency. Figure 10 shows an example of a fitted resonant peak. The black lines are the raw data and the red lines are the fitted functions describing the behavior of the FRF's within the specified window. Note that each line represents one measurement location.

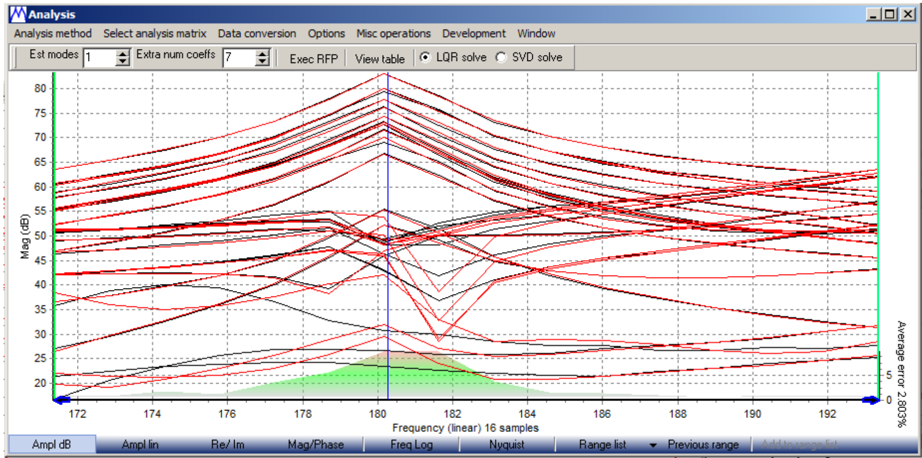

Figure 10 Mode fitting at around 180 Hz (violin tailpiece number 4)

The program Mode Shape was used to allow the motion at resonant frequencies to be visualized. Modal images presented in this paper will be given as normalized two dimensional color plots. Figure x can be interpreted by noticing that the black line is a position of zero displacement and the blue is into the page and the purple is out of the page. The displacement of the front and side are minimal. In the right of figure x can be seen an oblique angle to view the departure from the equilibrium position shown as an outline.

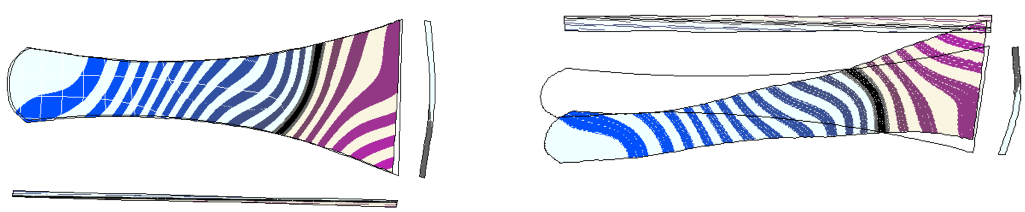

Figure 11 2D color plot of 112 tailpiece at 731 Hz

Fourier transforms representing the frequency domain of the recorded time functions displayed in this report will be graphed using the amplitude of the signal Z vs. the frequency at which the amplitude occurs. Phase information is useful for coordinating the motion of points in the modal visualizations but for the purposes of comparing individual FRF's this information is neglected. For real component data Re and imaginary data Im the Db amplitude of the acceleration Z is given by,

$$Z = 20\, log(Re^2 + Im^2)^{.5}$$

4. RESULTS/DISCUSSION

4.1. IDENTIFYING MODE SHAPE ON "DEAD RIG"

Mode shapes will be first identified using no mass in the slot. This will set the base line for further studies.The coordinate system can be seen in figure 3.

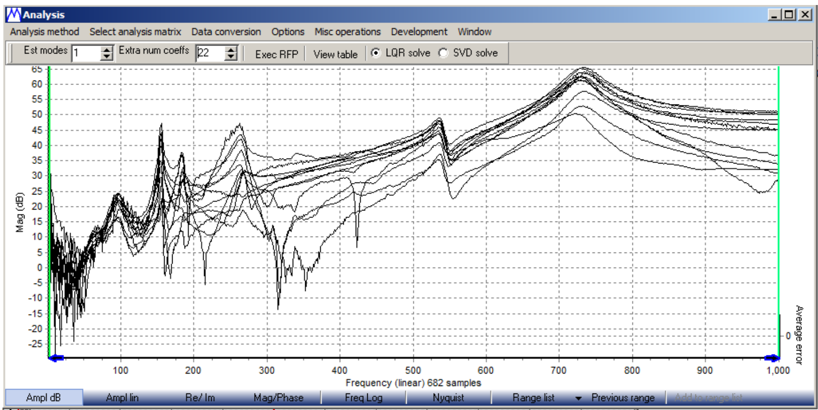

Figure 12 Resonant frequency at around 750 Hz

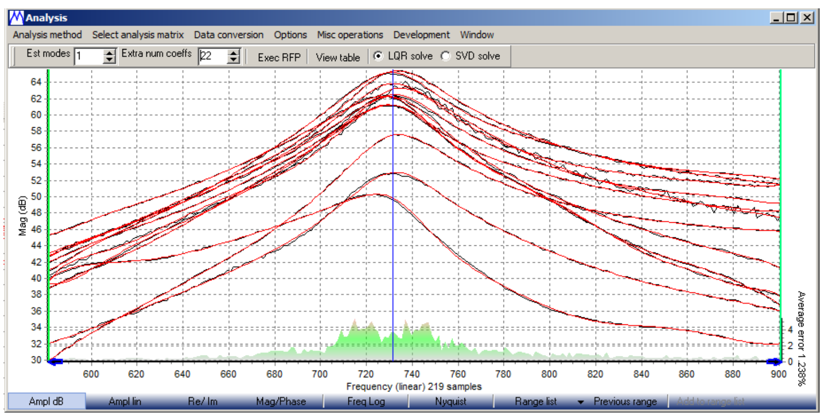

Figure 13 Mode fit for 731.58 Hz

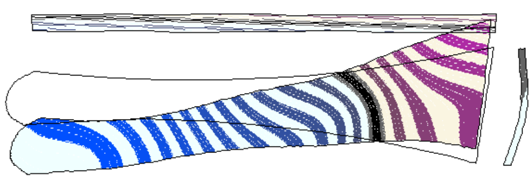

Figure 14 Mode shape for 112 tailpiece xy plane at 731 Hz

Figure 14 shows the mode shape for the resonance peak identified in figure 12. The task then is to attempt to use the mass in the slot to change the frequency response of the tailpiece at this mode shape. A design hypothesis is that at the 0 % position the mass is close to the node in the figure 13, thus the added mass will not appreciably add to the mass distribution about this axis of rotation. The more the mass travels down the slot towards the bottom of the tailpiece the greater the mass distribution about this axis will be. For mass moment of inertia I and angular velocity w, the kinetic energy of the tailpiece at the point of constant velocity in a cycle of motion (i.e. No acceleration) is given as the kinetic energy Ek at that point,

$$Ek = \frac{1}{2} I w^2$$

Because of the principle of superposition, this mode of vibration can be considered as a closed system, conservation of energy then should hold for a first approximation. So with an increase in mass distribution we must have a decrease in w which corresponds to a decrease in frequency. Meaning that at the point of zero acceleration the angular velocity of the tailpiece should be less for a larger mass distribution corresponding to a decrease in frequency of that mode of vibration.

4.2. XY Plane Resonance Shifting Moving Mass 0-100 %

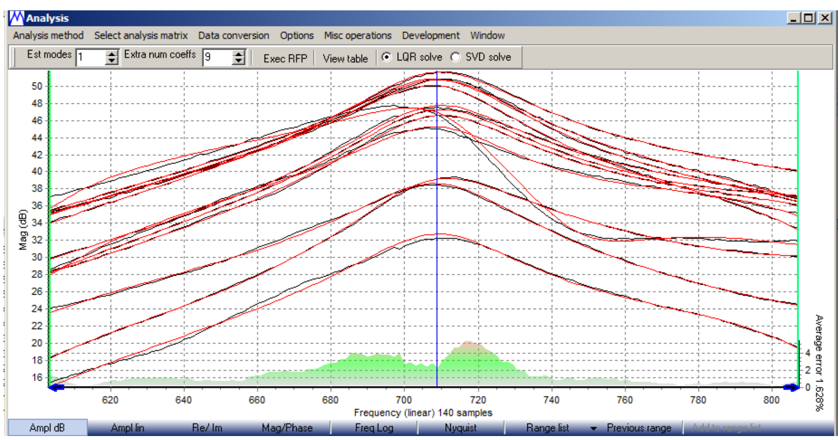

Figure 15 Resonance peak w/ 2.2 g weight at 0%

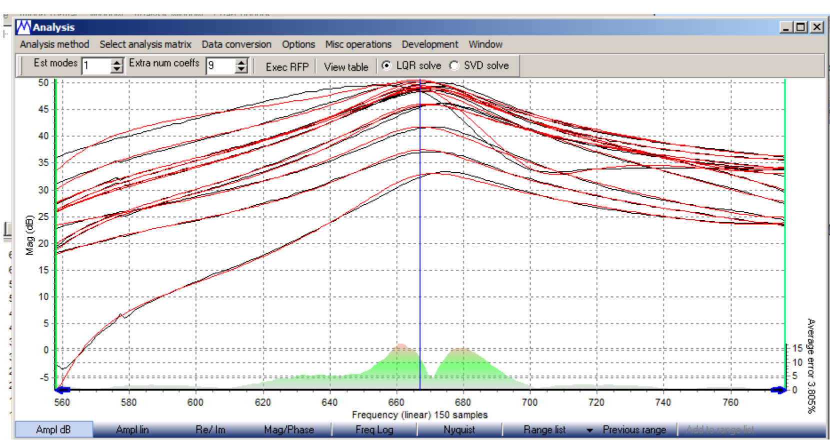

Figure 16 Resonance peak w/ 2.2 g weight 50%

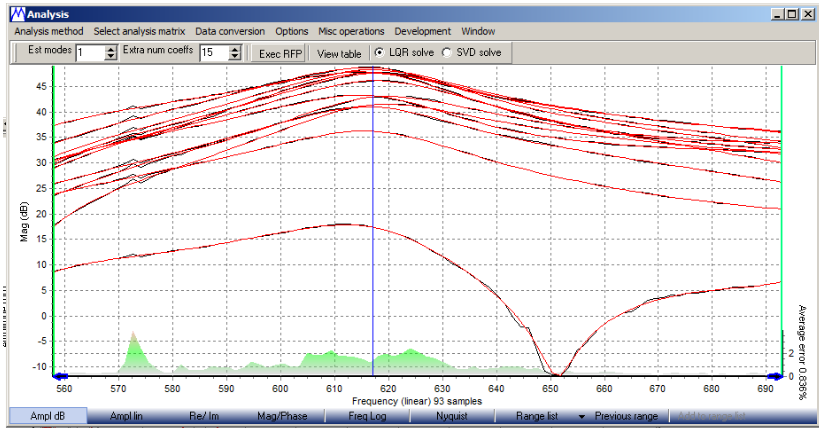

Figure 17 Resonance peak w/ 2.2 g weight 100%

Table 1 Change in frequency due to change in mass distance along channel

Test	Mass (g)	% of distance in channel	Frequency (Hz)
1	0	N/A	731
2	2.2 g	0	709
3	2.2 g	50	668
4	2.2 g	100	617

The change in frequency seen in these tests represents a shift of 113 Hz. At this frequency the change is audible in the musical sense as a change in pitch from just below F# to just below E#. It was predicted that the proximity of the mass at 0 % to the axis of rotation would not change the frequency appreciably. The change observed of 22 Hz could be due to the fact that the 0% location is not exactly at the axis of rotation. Also in the assumptions the geometry of the mass is not taken into account.

4.3. TESTING 108 MM TAILPIECE ON VIOLIN.

Tests were performed on a good quality violin (as determined by ted white). The Violin was supported using rubber bands at the end button and around the neck to isolate it from external influences. Both the strings and the after-lengths were damped using packing foam in order to reduce the influence of noise caused by string harmonics. Individual FRFs were taken in order to observe the location of the resonance peaks. It is assumed that the major resonance found in the xy plane data between 500 and 800 Hz is representative of the same mode shape found in 4.1 on the 112 mm tailpiece.

4.3.1. IDENTIFYING RESONANCE ON INSTRUMENT BRIDGE

From figure 17 the resonance peak at 550 and 700 Hz of the bridge FRF will be used to verify the ability of the tailpiece to influence the instruments resonance amplitude and position. This peak (outlined in red in fig18) is chosen because it is the largest resonance peak in the frequency range and it lies within the likely range of adjustability of the tailpiece.

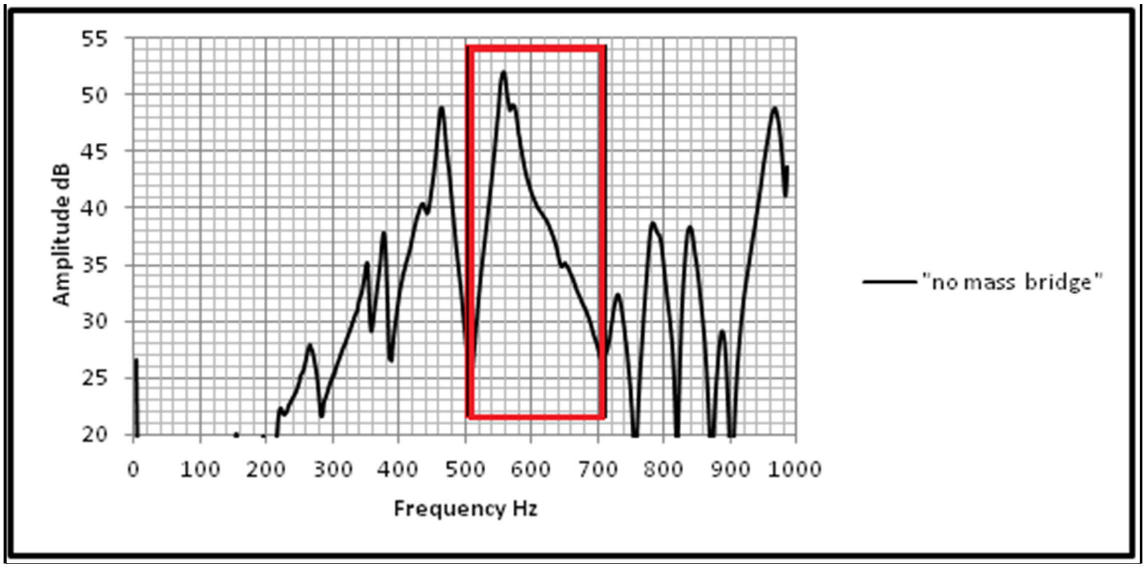

Figure 18 Bridge measurements on violin w/ no mass in tailpiece

4.3.2. TRACKING CHANGES MADE BY ADJUSTING MASS IN TAILPIECE

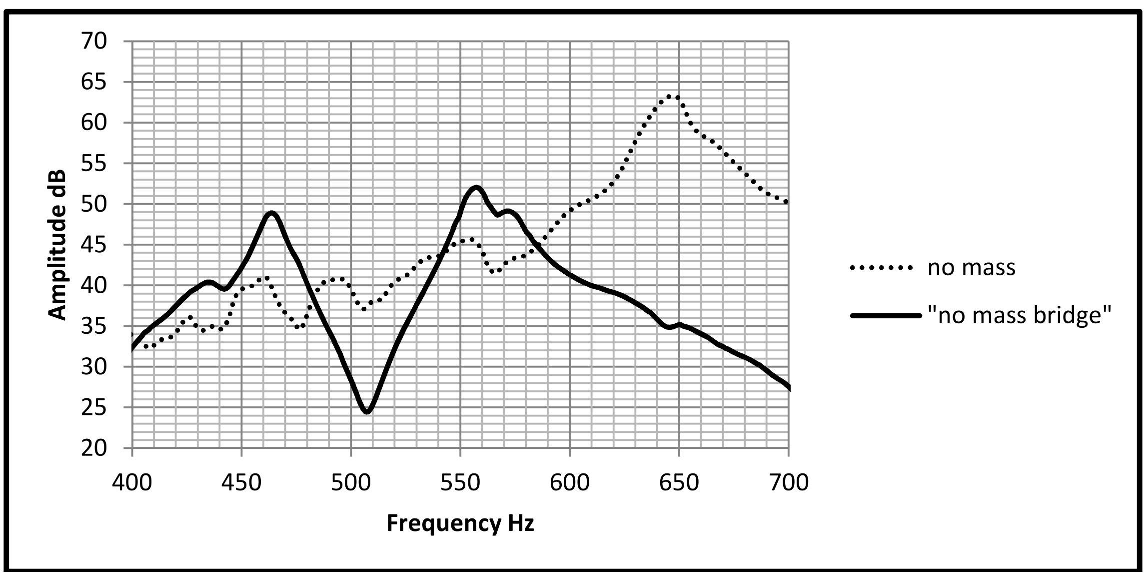

Figure 19 Tailpiece and bridge measurements w/ no mass in tailpiece

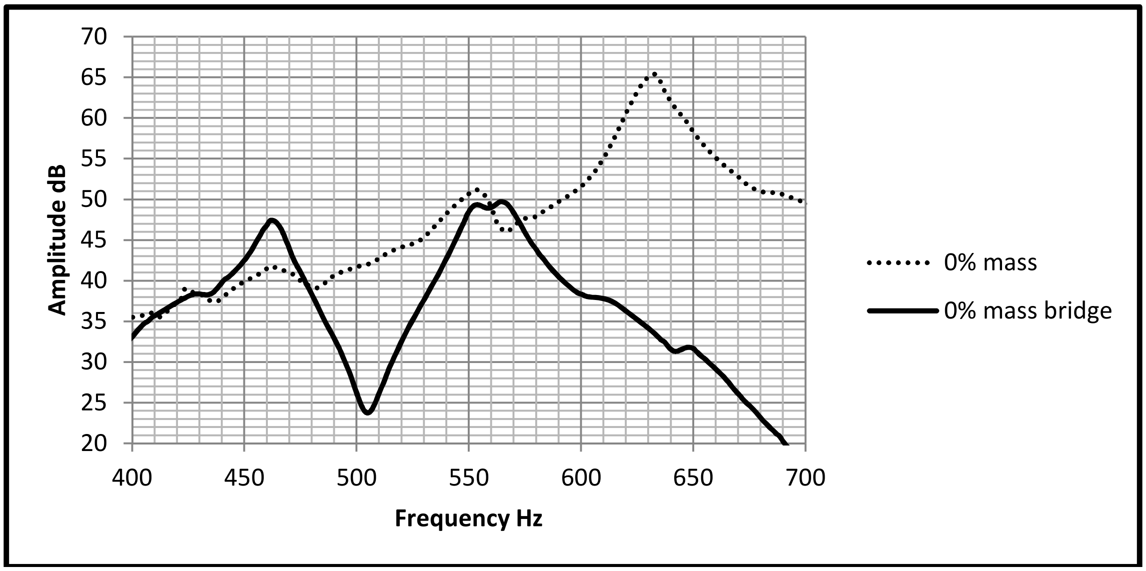

Figure 20 Tailpiece and bridge measurements w/ 2.5 g mass at 0%

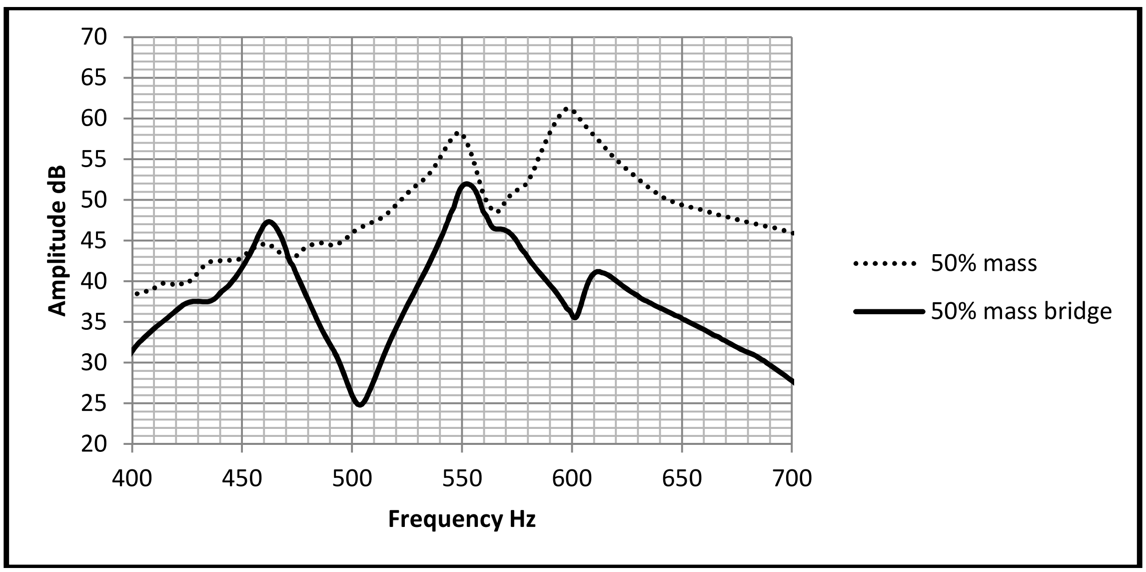

Figure 21 Tailpiece and bridge measurements w/ 2.5 g mass at 50%

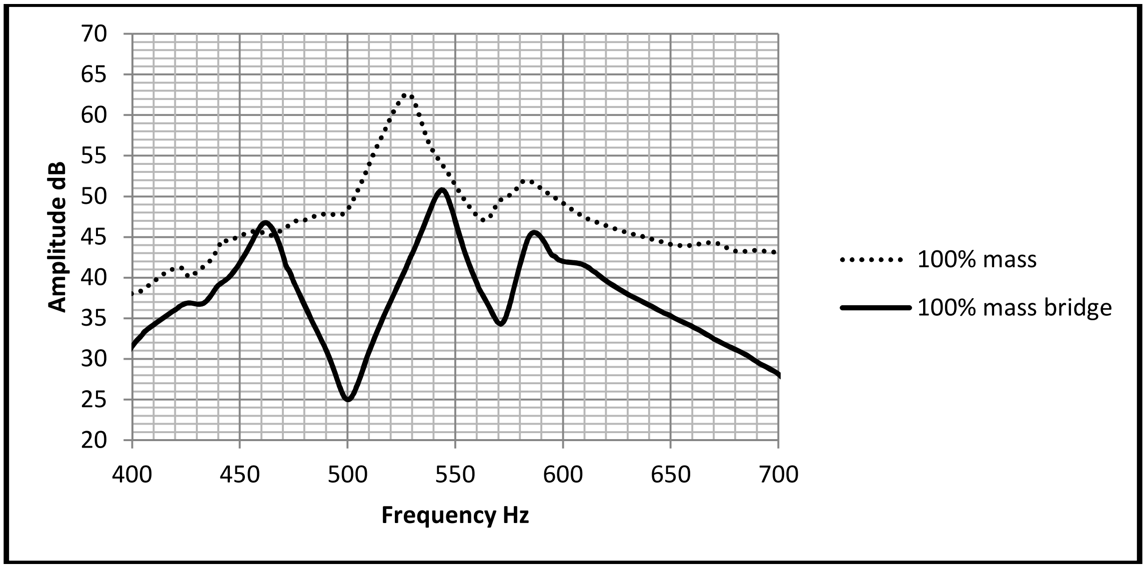

Figure 22 Tail piece and bridge measurements w/ 2.5 g mass at 100%

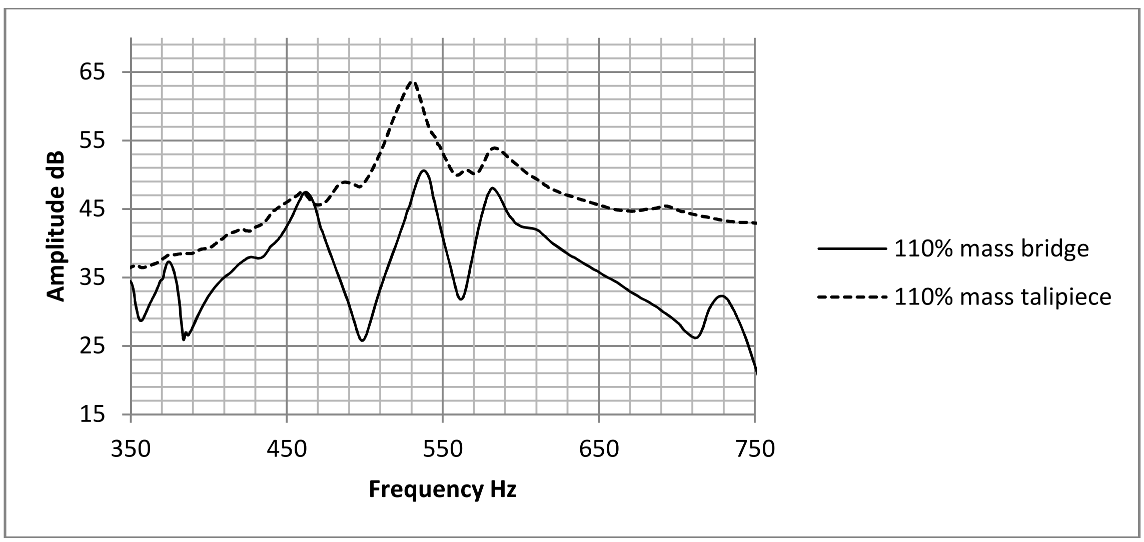

Figure 23 Tail piece and bridge measurements w/ 2.5 g mass at 110%

Table 2 Peak data for resonance shifting tests

	Bridge peak 1			Bridge peak 2			Tailpiece peak 1			Tailpiece peak 2		
	Freq (Hz)	Amp(dB)	Q (quality)	Freq (Hz)	Amp (dB)	Q (quality)	Freq. (Hz)	Amp (dB)	Q (quality)	Freq. (Hz)	Amp (dB)	Q (quality)
no mass	560	52	2.63	n/a	n/a	n/a	643	63	1.28	n/a	n/a	n/a
0% mass	560	50	2.43	n/a	n/a	n/a	630	65	1.3	555	51	1.03
50% mass	551	52	0.96	610	41	1.05	596	61	1.56	550	58	1.06
100% mass	543	51	1.07	585	46	0.62	586	52	2.16	526	62	1.46
110% mass	539	51	1	580	48	0.87	583	54	2.14	529	63	1.36

As the Frequency of the tailpiece resonance (shown as a dashed line in figures 19 through 23) shifts lower in frequency and gets closer to the resonant peak described in 4.3.1 (henceforth referred to as R_b) it responds by splitting and reducing its quality and amplitude. The resonance of the tailpiece (R_t) responds also by splitting as it comes closer to the R_b. A measurable effect of the R_t coming close to R_b is that its original signal magnitude at 560Hz of 52 dB has a maximum reduction at the "110% mass" case of 20 dB representing a decrease of an order of magnitude. The splitting of R_b creates 2 peaks (in the bridge FRF response) that lie either side of the original (560 Hz) at 537 and 582 Hz (seen in fig 22). The magnitude of the split peaks is close to the original dB level but a change in the quality of the two peaks of R_b at the 110% mass position has diminished from 2.63(no mass condition) to 1 and .87 for the lower and higher frequency peaks respectively. This change in quality represents an increase in damping for each peak relative to the first.

The observation that both resonance peaks show splitting at their point of overlap implies that there could be a "sharing" of excitation force, in that for a finite sum of energy at the frequency of overlap both peaks are attempting to use the energy excite available at that frequency to excite their preferred modal force distributions causing both peaks to diminish. Investigation into the energy distribution around the frequencies of overlap may reveal the mechanism behind this peak splitting phenomenon.

5. CONCLUSIONS/ RECOMMENDATIONS

The investigation into the modal response of the tailpiece at the R_t resonance has provided justification for the potential of the tailpiece design to shift the frequency of resonance (R_t) by moving the weight down the channel on the underside of the tailpiece (shown in 3.3). Because of the modal shape observed in 4.1 where the tailpiece teeters about the axis of rotation shown in figure 13 (black line) and in figures 3, 5 and 6 the movement of this weight in the tailpiece channel adds to the mass moment of inertia of the tailpiece about this axis of rotation (node) and thereby, it is postulated, lowers the frequency of excitation as (described in 4.1). The ability of the 112 mm tailpiece to shift the frequency of this mode shape through a range of frequencies between 617-731 Hz has been demonstrated in 4.1 by moving the mass through its range of motion in the channel and performing modal analysis with no mass and with a 2.2 g mass at 0% 50% and 100% of distance along channel length. This data corroborates the postulate of design function in that the frequency lowers with greater mass distribution about the axis of rotation. The ability of the 108 mm tailpiece design to affect the response of the violin was shown in 4.3 where the identified resonance peak of the violin bridge (R_b) was shown to be disrupted by the presence of the R_t resonance. The disruption caused a maximum change in amplitude about the original R_b frequency (560Hz) of 20 dB and a splitting of the original peak causing the R_b resonance to appear as two resonant peaks either side of the of the original 560 Hz at 537 and 582 Hz observable in figure 23. Both the resonant peak of the Tailpiece and the resonant peak of the bridge motion were seen to split. It is noted that looking into the energy distribution between the two peaks at their overlap could reveal understanding of the mechanism behind this peak splitting phenomenon.

The tailpiece appears to function as is designed. Further investigations into the modal response of the tailpiece designs and how they can be tuned could be carried out using FEA for preliminary design analysis. FEA would provide a diagnostic tool for assessing the function of the tailpiece without having to make one first. Its use could potentially save resources and time. Currently the results from this experiment are being used to develop FEA models that can serve as test rigs for further investigation. It is recommended that this be the focus of further studies, helping to expedite the design process.

6. REFERENCES

[1] O. Ina´ cio, J. Antunes, M.C.M. Wright (2008). "Computational modeling of string–body interaction for the violin family and simulation of wolf notes" *Journal of Sound and Vibration* (N° 310): 260–286
[2] *1. Stough, B.* (1996), "The lower tailpiece resonances". *CASJ Catgut Acoustical Society Journal* 3 (N°1 series II): 17-25.
[3] Henry A. Strobel, "Useful Measurements For Violin Makers, a reference for shop use" *Has Oregon* (July 1988)

7. APENDIX

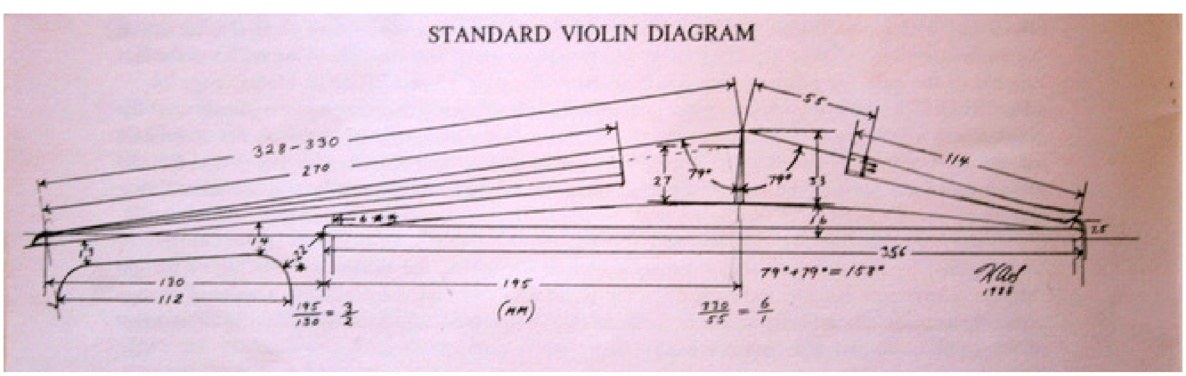

Figure 24 Dimensions used in Violin "dead rig" From "*Useful Measurements for Violin Makers*", Henry A. Strobel