ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF ARTS AND SOCIAL SCIENCES

TOP PLATE VIBRATION ANALYSIS OF THE KANUN INSTRUMENT

Ph.D. THESIS by

Cem ÖMEROĞLU

Department of Music

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ SOSYAL BİLİMLER ENSTİTÜSÜ

KANUN GÖĞÜS TAHTASININ TİTREŞİM ANALİZİ

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KASIM 2020

Cem ÖMEROĞLU, a Ph.D. student of ITU Graduate School of Arts and Social Sciences student ID 409032001, successfully defended the thesis entitled "TOP PLATE VIBRATION ANALYSIS OF THE KANUN INSTRUMENT", which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

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Date of Submission : 20.11.2020
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To my spouse and children,

FOREWORD

I would like to dedicate this study to the memory of our teacher and my advisor Şehvar Beşiroğlu with whom I started to work together on this subject.

I would like to express my appreciation to my advisor Prof. Dr. Can Karadoğan and the jury members Doç Dr. Zeki Yağız Bayraktaroğlu and Doç. Dr. Yahya Burak Tamer for leading me through this study.

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And finally, I would like to thank to my family for their great support throughout this work.

November 2020

Cem ÖMEROĞLU
(Sound Engineer)

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ABBREVIATIONS

App : Appendix
FFT : Fast Fourier Transform
FRF : Frequency Response Function
FEM : Finite Elements Method
PC : Personal Computer
3D : Three Dimensional
RMS : Root mean square
ADSR : Attack, Decay, Sustain, Release
K. E : Kinetic energy
P. E : Potential energy
SWL : Sound Power Level
SIL : Sound Intensity Level
SPL : Sound Pressure Level
dB : Decibel
GFRP : Glass Fiber Reinforced Plastic
CFRP : Carbon Fiber Reinforced Plastic

SYMBOLS

a : Acceleration
A : Area
C : Speed of sound
d : Density
E : Young modulus
F : Force
G : Shear modulus
g : Gravitational acceleration
Hz : Hertz
J : Joule
K : Spring constant
l : Displacement
l : Length
M : Molecular mass of the gas
MPa : Mega Pascals
m : mass
N : Newton
P : Pressure
R : Adiabatic gas constant
s : Second
T : Period
T : Absolute temperature
T : Tension per unit length
t : Temperature
V : Volume
µ : Poisson's ratio
γ : Adiabatic gas constant
λ : Wavelength
Q : Quality factor

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TOP PLATE VIBRATION ANALYSIS OF THE KANUN INSTRUMENT

SUMMARY

This study focuses on the top plate vibration analysis of the Kanun instrument. The study aims to identify and predict the top plate's natural frequencies by using a 3D physical model. By this way; the frequency spectrum of the instrument is going to be analyzed and evaluated with the model results of two top plates made of different woods called spruce and plane tree along with alternative materials such as metal and composites.

The method starts with an experimental study to measure the resonant frequencies of the top plates with a hammer test experiment. Top plates are hammered and the impact responses are recorded to a PC software via an accelerometer. Several individual points on the plates are researched in order to store those data for the further stage of the study; physical model verification.

The experimental study stands for a reference point in order to confirm the performance and efficiency of the model and see if the results are close enough to the measured experiment results.

By physically modeling the different top plates and defining their characteristic material properties in the computer environment, the results were computed with a PC software in free modes of vibration for the 3D physical model of the plates.

After the verification is completed by comparing the free modes of vibrational behavior, the model is approved to be used safely for further computations. The fixed modes of vibration will be studied and the results will be evaluated within the frequency range of the instrument.

The fixed mode results comparison of two woods for the natural frequencies; plane tree has much more overtones (22) placed in the actual sounding frequency range of the instrument than the spruce (18). This may be evaluated as that the plane tree has much more potential in terms of loudness and sound radiation through the top plate.

Twelve different individual pitches are pronounced for plane tree as natural frequencies for the fixed boundary conditions. This number is counted nine in spruce.

Even a cm change in any dimension of the geometry effects the natural frequency results of the top plates. Therefore, the sound's harmonic content and radiation intensity in terms of frequency and the corresponding amplitudes may change in the perceived sound field. Dimensions determine the wavelengths. Therefore, the natural frequency may change with the change of dimensions as the speed of sound remains constant.

After handling several top plates during the manufacturing process, it seemed that although all plates have the same geometry and identical dimensions, the weights are variable in practice. Briefly; the density and the relative humidity of the wood strongly effect the natural frequency. Sound velocity is related with the Young Module

along with the density in solid materials. So when the density parameter is decreased, the natural frequencies tend to go higher inversely.

Plane and spruce woods resonate around different parts of the frequency spectrum. In this case, again the geometries are almost identical. In addition to density, Young Modulus, Shear Modulus and Poisson's ratio determine resonant frequency differences.

Alternative top plate materials instead of woods are also studied via computer physical modeling. Aluminum 3003-H18 as a metal, GFRP and CFRP Toray as composite materials are presented at this stage. All of the materials are compared in terms of plate thickness, number of harmonics within the frequency range of the instrument along with the material properties.

Form and geometry practices present alternatives via drilling a single hole and a pair of three holes. As a result, GFRP Toray top plate with a single hole produces maximum number of harmonics (25) within the frequency range of the instrument.

Formulas, algorithms and further modifications of the finite elements method model parameters can also be used as a specific data source for digital sound synthesis subject; such as physical modeling.

KANUN GÖĞÜS TAHTASININ TİTREŞİM ANALİZİ

ÖZET

Bu çalışma kanun enstrümanının göğüs tahtasının titreşim analizine yoğunlaşmıştır. Çalışmanın amacı göğüs tahtasının doğal titreşim frekanslarını tam anlamıyla çalışan ve geçerliliği yine bu çalışmanın içinde ispatlanmış bir üç boyutlu fiziksel model yardımıyla tanımlamak ve tasarım sürecinde öngörebilmektir. Böylelikle ladin ve çınar ağaçları ile birlikte metal ve kompozit malzemeler için de enstrümanın göğüs tahtasının frekans spektrumu model sonuçları ile birlikte analiz edilip değerlendirilecektir.

Yöntem, başlangıç olarak çekiç testi olarak adlandırılan darbe deneyi aracılığı ile doğal titreşim frekanslarını ölçmeye dayalı olan deneysel çalışmayı kullanmıştır. Göğüs tahtaları çekiç ile darbelenmiş ve darbe tepkileri bir ivmeölçer aracılığı ile bilgisayar yazılımına kaydedilmiştir. Plakalar üstünde ayrık ve birden fazla sayıda nokta araştırılmış ve bu bilgiler sonraki aşama olan üç boyutlu fiziksel modellemeyi doğrulama aşamasında kullanılmak üzere ayrılmış ve saklanmıştır.

Böylece deneysel çalışma, fiziksel modellemenin doğru ve tutarlı bir şekilde çalıştığını sağlamak ve gerçek ortam şartlarına mümkün olduğunca yakınlık sağladığını göstermek amacı ile bir referans noktası olarak kullanılmıştır.

Farklı göğüs tahtaları fiziksel olarak modellenerek malzeme karakteristik bilgileri tanımlanmış, sonuçlar bilgisayar yazılımı ile üç boyutlu fiziksel modelin serbest titreşim modlarına göre ilgili yazılımla hesaplanmıştır.

Model hesaplamaları ile deney sonuçları birbirleri ile kesiştikten ve serbest titreşim modlarına göre sağlama yapıldıktan sonra, fiziksel modellemenin sonraki çalışmalarda güvenli bir şekilde kullanılabilirliği onaylanmıştır. Sonrasında ise her iki farklı ağaç için sabitlenmiş titreşim modları çalışılmış ve sonuçlar aşağıdaki şekilde değerlendirilmiştir;

İki ağacın sabitlenmiş titreşim modları karşılaştırıldığında; çınar ağacının (22) ladine (18) göre enstrümanın frekans aralığında daha fazla doğuşkan içeriğine sahip olduğu gözlemlenmiştir. Bu sonuç çınar ağacının göğüs tahtasının gürlük ve ses yayılımı anlamında ladine göre daha fazla potansiyele sahip olduğunu açıklayabilir.

Geometriyi oluşturan plaka boyutlarında yapılacak cm bazında bir değişiklik dahi doğal titreşim frekans sonuçlarını etkilemektedir. Bundan dolayı; sesin doğuşkan içeriği ve yayılım şiddetinin genliklere bağlı olarak ses alanı içerisinde değişmesi beklenebilir. Boyutlar dalga boylarını belirlemektedir. Bundan dolayı, ses hızı sabitken boyutlar değiştiğinde doğal titreşim frekansında değişiklik beklenebilir.

Üretim aşamasında çeşitli göğüs tahtalarını incelerken tüm plakalar hemen hemen aynı geometriye ve ölçülere sahip olsalar dahi ağırlıkları dolayısıyla da yoğunluklarındaki değişkenlik gözlemlenmiştir. Kısaca; ağaçların yoğunluğu ve bağıl nem oranı doğal titreşim frekansını kuvvetli bir şekilde etkilemektedir. Katılarda ses hızı Young Modülü ve yoğunluğa bağlıdır. Bu şekilde sadece yoğunluk parametresi düşerse, doğal titreşim frekanslarının tam aksine arttığı gözlemlenmiştir.

Ladin ve çınar ağaçlarının göğüs tahtaları frekans spektrumu içinde değişik bölgelerde rezonansa girmektedir. Bu durumda yine geometrilerdeki benzerliğe vurgu yapılabilir. Yoğunluğa ek olarak Young Modülü, Sertlik Modülü ve Poisson's oranları bu doğal titreşim frekanslarını hep birlikte belirlerler.

Ek olarak, göğüs tahtası için ağaçlara alternatif olabilecek farklı malzemelerin incelenmesi de fiziksel modelleme yoluyla çalışılmıştır. Metal olarak Al 3003-H18, kompozit malzeme olarak da GFRP ve CFRP Toray malzemeleri bu aşamada sunulmuştur. Tüm malzemeler; plaka kalınlığı, enstrüman frekans sahasına düşen doğal frekans sayısı açısından karşılaştırılmış ve ek olarak malzeme özellikleri ile belirtilmiştir.

Biçim ve geometri çalışmaları ise göğüs tahtası üzerinde tek delik ve üç delik olmak üzere alternatif olacak şekilde çalışılmış ve sunulmuştur. Sonuç olarak, göğüs tahtasında tek delikli GFRP Toray malzemesi ile doğal frekanslar için enstrüman frekans sahasında maksimum sayıda (25) harmonik elde edilmiştir.

Modelde kullanılan sonlu elemanlar yöntemine ilişkin parametrelerden; formüller, algoritmalar ve sonraki aşamalarda yapılabilecek değişikliklerin, sayısal ses işleme ve sentezleme konusunda fiziksel modelleme araçları olarak da kullanılması beklenebilir.

1. INTRODUCTION

Sound can be defined as an acoustical energy which propagates via waves in a common medium such as the air flow around and between the perceiver and the source. Sound is an individual way of perception in terms of psychoacoustics. Music uses sound and silence as organization tools which is called a musical composition. The way how people would perceive that musical composition as sound is a subject of psychoacoustics.

Acoustical music instruments were the main source of sound for a long time in the history of music. Today we can say that all sounds that we hear around can be a suitable tool for various genres of music; like music concrete or electronic music. This is accomplished by the significant effects of several factors like technological developments in instrument materials, the change in acoustic and electronic music instruments design along with manufacturing methods.

Technological developments have also considerable effect on the distribution of the music. Digital audio technology has taken the lead from analog tape recording devices since the beginning of 1980's which can be given as an example of technological progression in music distribution which has high and spectacular effects on economic and musical results. As the music changes with technology, music industry changes with technological developments as well as many other related parameters which can be the subjects of musicology or anthropology research.

As emphasized before, technological developments open a gate to non-traditional sounds which can be considered as musical tools like; electronic synthesizers. Also sound of the nature which is recorded via tape or digitally is used as a musical instrument in compositions. The sounding frequency spectrum of the music keeps changing.

Technology makes a dynamic contribution in recording and producing music. Recording and producing process of acoustical music instruments can be seen as a trigger for researching and analyzing instruments in detail along with instrument design and manufacturing aspects. For example; today a sound recording engineer is aware of the fact that the microphone placement method for a woodwind instrument may not work for a brass instrument. As the studies on acoustics and psychoacoustics subjects evolve, the knowledge to analyze every possible component of the sound that radiates form the instruments gradually extends. The effect on this kind of technological progression in music and acoustical musical instruments is the starting point of this study.

Acoustical music instruments are one of the sound sources of the music which can also be described as cultural symbols reflecting the characteristics of the geography and society in which they have been performed through the years. The varieties in shape, dimension, geometry and material can be considered as basic examples.

Traditional Turkish music instruments are strongly related with the geographically surrounding cultures spreading from middle Asia to Africa, Balkans and Anatolia. This cultural diversity emphasizes the importance of the conservation and progression of these traditional music instruments. This goal can be achieved by analyzing the instruments with suitable methods along with the progression of new designed and manufactured instruments.

The study on sound and vibration goes back to Greek philosopher Pythagoras whom studied the string vibration on a system called monochord. This study can be considered as the first step of a scientific research on the perceived sound of a string. The mathematical relationship between the perception of sound and behavior of the string, the articulation of the music performance was identified. (Hutchins, 1982)

The string division concept introducing harmonic spectrum was maintained in the East by Farabi and Safuyiddin. A study of the well-known scientist (Galileo, G., 1542-1642) introduced the wavelength concept which defined the relationship between the sounding frequency and the string length. The study of Galileo also introduced the interval concept on which the music theory will stay on later.

Chladni (1756-1842) set a way to visualize the vibration on surfaces and introduced nodal and anti-nodal points referring minimum and maximum displacement respectively, due to vibration. Eventually, due to the frequency component of the source, vibration was seen as visual patterns with the help of various materials spreading on the surface like salt. Today, this method of vibration analysis is widely used in many instrument manufacturing ateliers.

Some musical and physical aspects of the sound have changed during time like the reference pitch A which was tuned to 425 Hz in the 18th century and was tuned up to 440 Hz in 19th century. One of the main results of this change was the increase in the perceived loudness of the stringed instruments. This feature was found as a supportive parameter for more balanced music loudness which ensembles in big concert halls.

The violin luthiers studied the neck length, neck angle and the bridge height were increased in order to compensate the tension change and timbre changes on the instruments. As a result, the frequency range of the instrument is widened (Hutchins, 1983). Eventually, this background works on custom manufacturing and interactive research with the composers and the performers as the instrument is widely used in different genres of music like classical, folk, jazz, rock, pop, including heavy metal. Felix Savart (1791-1841) made the first detailed acoustical studies about the modern violin.

The Baroque type guitar body is also widened and the ribs are placed as supportive wooden parts as a sequence called fan bracing. The study by (Torres, A., (1817-1892) played a leading role in the progressive studies of the modern classical guitar.

Helmholtz (1821-1894) studied music, physics, physiology and psychology through a multi-disciplinary vision resulting profound presentations on the subject of the perceived sound. Rayleigh (1845-1919) combined experimental acoustics with the theory (Rossing, 2007).

Turkish music instrument studies concerning the subject were made on the instruments like; Kanun, Oud, Baglama, Tanbur and so on. As most of the studies throughout this work were listed in the literature view and the references, one of the motivation aspect is to expand these progressive studies with the research involving Kanun top plate's vibrational behavior.

1.1 Purpose of Thesis

The purpose of this study is to contribute in musical instrument design and manufacturing process by using computer modeling effectively. Experimental studies which are considered to be the subject of design and manufacturing processes, will be associated with software modeling by using the results of Finite Element Method (FEM) analysis. And finally, the aim is to present that computer modeling results can be used as a meaningful tool within the process of musical instrument design and manufacturing.

The study is focused on the top plate vibration analysis of the Kanun instrument. The aim is to identify and predict the vibrational behavior of the instrument's top plate by using a 3D physical model. By doing so; the frequency spectrum of the instrument's top plate will be analyzed, evaluated and compared with two top plates of different woods called spruce and plane tree.

Another aspect is to identify and present the parameters which effect the vibrational characteristics of the Kanun top plate like; orthotropic wood material properties, geometry and acoustical concepts with a comparative study by using the spruce and the plane tree samples. Tools and methods would be presented for the Kanun instrument top plate in terms of design and manufacturing process.

Another aspect of the study is to present alternative materials for the top plate with various form and geometry in suitable conditions. Aluminum as a metal, GFRP and CFRP as composite materials will be analyzed via computer modeling as an alternative for woods.

1.2 The Method

The method can be defined as a comparative study within the scope of experimental and modeling processes. In order to initiate an experimental study which measures the resonant frequencies of the top plates with a roving hammer test experiment. Top plates are hammered and the impact responses are recorded on a PC software via an accelerometer. Several individual points on the plates are researched and the data is stored and kept for the further stage of the study; the physical model verification.

The experimental study stands for a reference point in order to confirm that the model is efficiently working and close enough to the result of the experiment.

By physically modeling of different top plates and defining their characteristic material properties in the computer environment, the results are computed with a PC software in free modes of vibration for the 3D physical model of the plates for the next stage.

After the verification is done via comparing the free modes of vibrational behavior, the model is approved in order to be used safely for further computations. In the next chapter, the fixed modes of vibration behavior is studied and the results are evaluated.

1.3 Literature Review

The effect of the Oud instrument's top plate vibrations on timbre is researched with the study as presented by (Değirmenli, 2015). Magnets are used to modulate the vibration characteristics of the top plate. The analysis processes are repeated while the location of the magnets on the top plate were changing. Relationship between the top plate's vibrational properties and the instrument's sound spectrum beside envelope are studied with the data obtained during the research. As a conclusion; it is stated that the natural vibrational modes of the stringed instruments strongly affect the timbre properties.

Top plate tuning method is offered within the graduate thesis by (Değirmenli, 2014) mentioning top plate vibrational properties control with the acoustical measurement techniques during manufacturing. The sound and vibration analysis systems are presented which could be easily found at the instruments designer's studio or atelier. Top plate tuning process using these analysis systems were presented. Wood material mechanical property measurement studies are also presented. As a conclusion, it is stated that the oud top plate's natural vibrational modes are affecting the instrument's timbre via the duration of the sound in free conditions.

A study by (Perry, 2014) indicates mainly the guitar and other stringed instruments sound formation characteristics research. The radiation efficiency related to mechanical vibrations of the instrument is studied and all the acoustic and vibration measurement methods are presented in detail. The effect of the frequency and the vibrational patterns are studied. It is stated that The nodal points are together visualizing the vibrational patterns which determines the formation of sound more strongly than the frequency.

Violin's bridge mobility is studied with the experiments by (Elie et all, 2013) briefly focused on the motion of the bridge and relationship between the instrument's body vibration formation efficiency with the force coming from the strings. The results are associated with the sound quality and the performance comfort of the instrument. Additionally, by changing one of the structural characteristics of the violin like the sound post and muffling the bridge; sound measurements are presented. Impact hammer and an accelerometer and a microphone are used during the study.

A graduate thesis by (Lu, 2013) presented a FEM analysis and experimental modal analysis of the violin top plate and stated a comparison between them. The modal patterns are seemed to match as a result. Additionally, the same measurements are done with the composite material made top plates. It is stated that if a composite material could be used instead of spruce for the top plate component of the instrument. As a conclusion; it is stated that a composite material presents very likely results with the spruce and it can be used as an alternative material for spruce.

Another study with a different instrument is by (Mansour and Scavone, 2012) which presents the modal analysis of the Persian Citar. Also various kinds of measurement equipment are used and advantages and disadvantages of these tools are studied. The impulses exciting through the bridge are obtained by an impact hammer and a shaker, the measurements are obtained by an accelerometer, accelerometer with laser and a microphone. All these measurement methods are listed as; measured frequencies, noise floor, budget range, extra mass addition to the system and the placement of these tools during the measurement process are studied. Finally, this study has a recommendation part which suggests to use sensors for the measurement of the natural vibrational modes.

Kanun and Tanbur were together analyzed acoustically in a Ph.D. thesis by (Gökbudak, 2011) which studied the frequency response of the instrument and the spectrum and octave band analysis. The envelope of the sound and the transient part of the waveform are also analyzed. Sound radiation property and sound pressure level are measured. Finally, the sound radiating from the instrument is analyzed in detail.

Top plates with different values of width are researched in relation with the sound produced by the instrument using acoustical analysis methods is presented with a study by (Yılmaz, Belenli, 2011). The envelope, and the harmonic spectrum of the sound are researched. Finally, the resulting data is evaluated and an ideal value of a width recommendation of the top plate is presented.

Violin's sound propagation characteristic is analyzed with a study by (Curtin, 2009) which studied mainly the frequency response of the instrument. Frequency response function is analyzed in detail using tools like an impact hammer, microphone and associated equipment and software. It is mostly a test method presentation rather than a comparison between various violin instruments.

Traditional Greek string instrument Cretan lyre's top plate vibration analysis is presented with a study by (Bakarezos et all, 2006) which uses electronic speckle pattern interferometry for the experimental method to indicate frequency patterns. FEM is used to analyze the frequency patterns. Both methods are compared and evaluated. Therefore, it is offered that any prediction about vibration characteristic of the instrument can be made via analysis techniques which is FEM in this case.

A graduate thesis by (Erdiş, 2006) studied acoustical properties of Baglama top plates. Fast Fourier Transform is used for the harmonic spectrum analysis. Extra supportive wood components called ribs that are traditionally used in guitar and oud are added to instrument body. Additionally, a sound hole is added as opposed to manufacturing traditional style. As a result, it is stated that the new designed instrument is louder and have much more upper harmonics than the traditional ones.

A study by (Schleske,2002) presents structural and acoustical analysis of violins made by important luthiers including Antonia Stradivari (1712). Arching, top plate width, exterior contour is studied in terms of design analysis. On the other hand; sound velocity, density, damping ratio and wood's microscopic structure are studied in terms of material analysis. The effect of the varnish on wood's acoustical quality is researched and the work is completed with modal analysis. This study has also another important aspect. It presents the necessity of researching older designs of acoustical instruments.

Another study by (Schleske, 2002) studied the timbre results of the violin instrument's vibrational behavior. Sound propagation is studied in terms of angles. In every 30 degrees Schleske repeated his measurements to complete 360-degree cycle. Finally, an average of the results is taken into account to present the resonance profiles. Impact hammer and microphone is used to get FRF for every discrete angle measurement. Another important aspect of this study is to emphasize the relationship between the perception of the sound in terms of psychoacoustics and the physical vibration acoustical measurements which makes this study a leader of the following ones.

Low budget acoustical studies and measurement alternatives were studied by Morset (2001). These methods were compared in terms of advantages and frequency measurement ranges were indicated. Averaging and normalization process that is applied on the measurements were explained in detail. It is important that it offers luthiers atelier to a new set of tools with an economical alternative to study and design with.

Another Turkish music instrument Tanbur is analyzed in the study by (Erkut et all., 1999). First the frequency response functions are obtained by impact actions applied to instrument's bridge. Body vibration's damping characteristics are also presented. Instrument's fret offset values are indicated during the performance. This method is achieved with the application of a magnetic pitch up and an accelerometer. Tanbur; selected instrument of the study is important in terms of the following researches on Turkish music acoustical instruments.

A graduate thesis by (Taçoğlu, 1997) studied the Tanbur top plate's vibrational characteristics with various kinds of woods and plate width. The effect of these parameters were analyzed. Every plate is applied on the instrument and FFT are obtained according to instrument performances. Harmonic spectrum changes are indicated related to different top plate material properties and width. Chladni patterns are also used to present the modal patterns of the top plates. This study is one of the first examples of a presentation of the vibrational behavior of the Turkish music stringed instruments.

A study by (Wright,1996) presents audition tests with the synthesized guitar sounds that contains different vibrational properties. It indicates how the perceived sound would be affected by the vibrational modes of the instrument.

Another guitar study is presented by (Rossing,1988) which offers sound propagation characteristics of the vibrating sound sources. Average power spectra and 1/3 octave band analysis were the related study subjects in terms of psychoacoustics. It is stated that the attack and the decay times of the sound is related to timbre information. Therefore, envelope followed harmonic spectrum to define timbre properties of the perceived sound.

Frequency response as a result of the guitar vibration behavior effect on sound timbre information is presented in a study by (Caldersmith and Johnson, 1980). Chromatically performed two low and middle quality guitars compared via fundamental frequency, overtones and decay times. Secondly the force applied on the bridge is changed and compared measurements finally stated that vibrational behavior is important on instrument’s perceived sound quality.

1.4 Hypothesis

The main purpose of the study is to present computer modeling as a useful tool in traditional acoustical music instrument design and manufacturing. In the study, the question concerning the effectiveness of a computer physical model on traditional acoustic instrument design and manufacturing process will be answered.

Additionally, the prediction concerning top plate’s material effecting musical instrument’s playing range by using computer physical modeling will be another aspect of the study. Two identical samples; spruce and plane tree will be compared along with the metal and composite material alternatives.

Another question concerning the integration of acoustical aspects and the analysis results with design and manufacturing alternatives which were formed traditionally and intuitively will be answered.

The main objective of the study is to identify the natural frequency harmonic series of the top plates. Roving hammer test is used for this topic with an experimental study.

Another aspect of the study is to indicate that the natural frequencies may be predicted via physical modeling. Software analysis is studied for this purpose using FEM.

Comparative study of the experiment and the model described above leads to the verification of the physical model along with experiment results. This is another important stage of the topic for the following issues of the study.

Presenting the fixed modes of vibrational behavior of the top plates with a comparative study of the two woods will be another aspect of the study. Identification of the structural conditions of sounding instruments in the most similar condition to real world conditions during the modeling process makes it possible to get meaningful results to work with. This subject is presented through the study with the comparison of the spruce and the plane tree is to associate the vibration behavior of the top plates and the frequency response of the instrument.

The question of predicting the top plate performance via computer modeling during the design stage can be presented as another aspect of the study. Using top plate materials instead of woods will be studied and the results will be given as comparisons with several alternatives.

Finally, predicting the results of form and geometry variations on the top plate via physical modeling would be the final stage of the study.

Hopefully, all these alternatives would be a guide for instrument makers and designers at the end.

2. MUSICAL ACOUSTICS OF THE KANUN

In this chapter, first the acoustic and the psychoacoustic terms related to the study are briefly described. It is presumed to be a reference point for the terms that are used in the discussion topics within the following chapters of the study. Then the Kanun instrument is presented which briefly describes the musical acoustics of the instrument and its constructional structure within the framework of the manufacturing process.

2.1 Acoustic and Psychoacoustic Concepts within the Framework of the Study

Music is an expression in a common medium such as air flowing between us and the source of sound, additionally it is an individual way of perception in terms of psychoacoustics. The timbre is defined as the sound character, the understanding of which instrument is playing when a sound is heard. In acoustics, it is defined as a function of musical harmonics and their amplitudes.

So while hearing; the frequency and the amplitude components of the sound is transmitted as a function of time forming a unique sound information (such as one particular Kanun instrument) within our hearing system as a simplified definition of timbre perception. On the other hand, these parameters can also be analyzed right there at the sound source. Sound is transmitted via waves through the medium in which it presents at a time, such as solids, liquids and gases like air. A brief information about the sound wave properties, acoustic and psychoacoustic terms within the frame work of the study is given below.

2.1.1 Sound wave characteristic parameters

The acoustical parameters related with the study subject are listed and explained below which may present a sound wave together in a given time. We may use simple harmonic motion to describe any repeating cycle like the sound wave only in one axis motion conditions.

The acoustical parameters related with the study subject are listed and explained below which may present a sound wave together in a given time. We may use simple harmonic motion to describe any repeating cycle like the sound wave only in one axis motion conditions.

Most of the sound basic phenomenon may be described via understanding the simple harmonic motion which converts the idea of a linear motion of an oscillating air molecule (in this case it is indicated as m in Figure 2.1) to a function of time with its displacement component due to a driven external force of trigger.

A mathematical model like simple harmonic motion can be used to describe several kinds of vibration systems, such as the oscillation of a spring, simple pendulum or a molecular vibration.

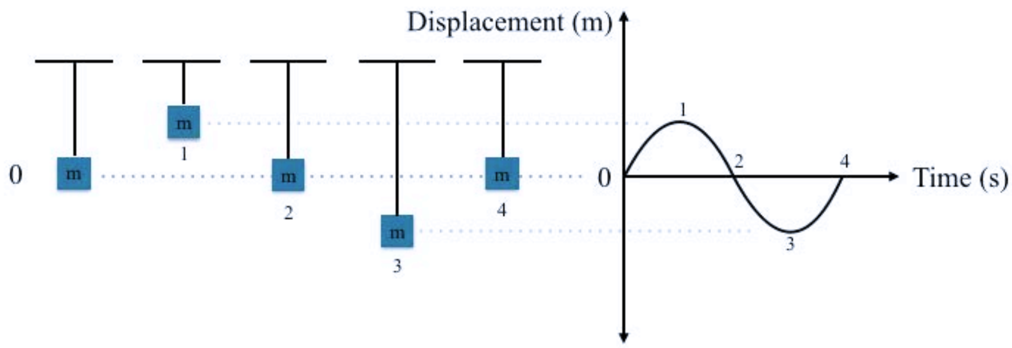

Figure 2.1: Spring and mass model; simple harmonic motion and sine wave.

Figure 2.1 can be taken as a mass spring model to describe simple harmonic motion. Air molecule is indicated as m and the interactive force between the air molecules are indicated with the spring attached to boundary. At point 0, there will be no motion of the air molecule as well as the instant times at point 2 and 4. As a result of the periodic motion, a sine wave can be expressed over time. At point 1 spring is compressed to its maximum displacement position as well as point 3. Their displacement absolute value is the same, however we use a minus sign for point 3 to indicate that the mass is below its equilibrium position.

The Hooke's Law; displacement, stiffness and restoring force parameters are given as follows:

$$F = -k\,.\,l \tag{2.1}$$

F: Restoring force, which always acts in a direction to restore m to its equilibrium position.

k: Spring constant, stiffness of the spring

l : Displacement from the equilibrium position

Newton's second law of motion is given as follows:

$$F = m.a = m.g \tag{2.2}$$

F: Force

m: mass

a: acceleration

g: gravitational acceleration

$$K = \frac{m\,.g}{l} \tag{2.3}$$

Equation (2.3) is a combination of equation (2.1) and (2.2). Spring constant, displacement, mass and the acceleration of the medium are interactively related for periodic motions.

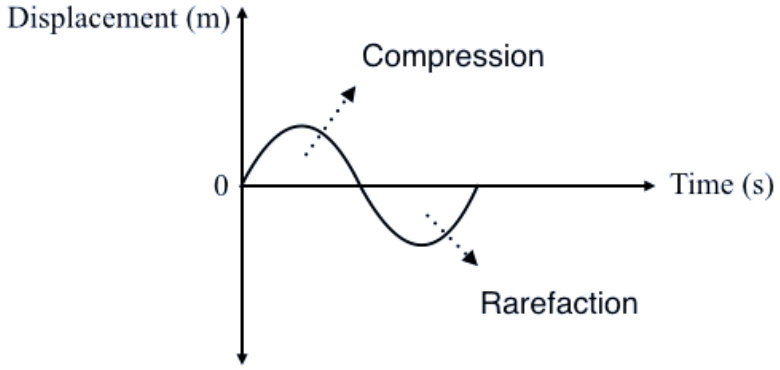

Figure 2.2: Pressure changes over time in a sound wave.

These fluctuations shown as successive compression and rarefaction regions of the air pressure regions in Figure 2.2 are transferred from one molecule to another, so the sound energy will be transmitted through the medium via waves.

Velocity would be the first sound wave characteristic parameter to be considered. It can be defined as the distance travelled in a unit time and may depend on the medium characteristics in which the sound wave propagates. This medium may be gas, liquid or a solid. Additionally, as the speed is independent of the direction of the waves, the way which the sound propagates differ in two forms like longitudinal and transverse waves. These terms are introducing the direction parameter into the subject, which let us use the term velocity instead of the speed.

In longitudinal or pressure waves; air molecule particle displacement direction is parallel to the velocity of the propagation. Sound waves travelling in air can be given as an example of this kind of compression waves. A listener standing in front of a musical instrument may perceive longitudinal waves with his/her ears.

Figure 2.3: Longitudinal wave presentation. (Nave, 2017)

In transverse or shear waves the motion of the medium particles are perpendicular to the direction of the wave motion. Waves travelling on a string of a musical instrument that is attached on both ends presenting a transverse wave situation.

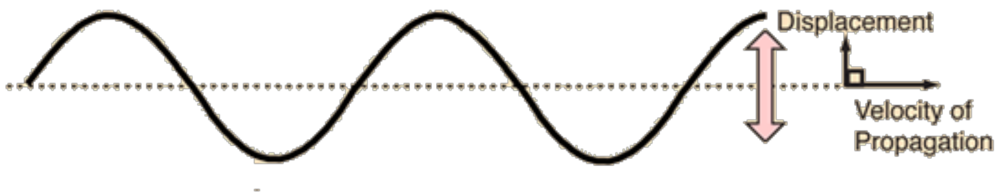

Figure 2.4: Transverse wave presentation. (Nave, 2017)

Generally, all mechanical wave's velocity could be expressed with equation (2.4), (Serway, Beichner, 2000).

$$C = \sqrt{\frac{Eleasticity\ property}{Inertia\ property}} \tag{2.4}$$

The velocity of the sound in air could be expressed with equation (2.5), (Howard, Angus, 2001)

$$C = \sqrt{\frac{E}{d}} \tag{2.5}$$

C: The velocity of the sound (m/s)

E: Young modulus of the gas (N / m^2)

d: Density of the gas (kg / m^3)

$$E = \gamma \,.P_{gas} \tag{2.6}$$

γ: Adiabatic gas constant

P: Pressure of the gas (N / m^2)

As indicated in equation (2.5) and (2.6); the velocity of the sound in ideal gases are related to adiabatic gas constant, the absolute pressure of the gas and the density of the gas.

$$d_{gas} = \frac{m}{V} = \frac{P\,.M}{R\,.T} \tag{2.7}$$

According to the ideal gas law, the density formula may form like equation (2.7) for the air.

m: The mass of the gas (kg)

M: Molecular mass of the gas (kg/mole)

R: The gas constant (8,314 J / Kmole)

T: Absolute temperature (Kelvin)

Thus, the sound velocity is given as follows:

$$d_{gas} = \sqrt{\frac{\gamma\,.R\,.T}{M}} \tag{2.8}$$

According to Oxygen, Nitrogen, Argon and other components of the gas combination, the sound velocity is dependent on temperature.

Sound velocity in air is expressed as follows:

$$C \cong 331{,}3 + 0{,}6.t \tag{2.9}$$

C: The velocity of sound (m/s)

t: Temperature of the air (^{0}C)

Because the disturbance moves so fast that there is no time for heat to transfer from the compressions or rarefactions, the adiabatic gas law equation is used (Howard, Angus, 2001)

One could also note about the subject that Isaac Newton first calculated the velocity of sound around 298 m/s regarding the heat transfer as a loss. (Newton, I., 1687) Then after, this error was rectified by Laplace in 1816. He was the first pointed out that the sound velocity in air depends on the heat capacity ratio also known as adiabatic constant or index.

Sound velocity depends on temperature and the composition of the consisting materials in air as can be seen from equation (2.8) and (2.9). However, the velocity of the sound is related to Young Modulus and the density of the environment in solids in one dimensional conditions as expressed in equation (2.2). Every other parameters of the sound waves are time dependent so the speed of the sound is interactively determining the frequency, period and the wavelength of the vibration.

An increase in air temperature may count in addition to velocity of sound as can be seen in equation (2.1). However, the velocity of sound in solids such as the wood part of the acoustic instruments may be evaluated in a different way rather than equation (2.2). As will be described later in Chapter 3; the orthotropic nature of the wood may present more than one dimensional motion for the wave velocity because of the transverse or so called shear waves.

In solids, Young modulus, denoted by E, is a measurement of the resistance to any change in solid material's length. It is a one dimensional consideration.

Shear modulus or modulus of rigidity is the ratio of shear stress to the shear strain. It is two dimensional because it is concerned with the deformation of a solid when it experiences a force parallel to one of its surfaces while its opposite side face experiences an opposite force.

Bulk Modulus is the measure of resistance to any change of the entire volume in solids and liquids. It is a three dimensional property. (Serway, Beichner, 2000)

The velocity of sound waves in solids is given as follows:

$$C = \sqrt{\frac{E}{d}} \tag{2.10}$$

Equation (2.10) can be used for one dimensional circumstances, indicating the speed of sound in stiff materials; like metal long rods.

C: Sound velocity (m/s)

E: Young Modulus of the material (MPa)

d: Density of the material (kg / m^3)

The velocity of sound waves on strings is given as follows:

$$C = \sqrt{\frac{T}{\mu}} \tag{2.11}$$

C: Sound velocity (m/s)

T: Tension on the string (N)

μ: mass for the unit length (kg/m)

Period is the duration of a one complete cycle in a vibrational motion. Regarding Figure (2.1); 0 and 4 is the same point which can be called as the start or end of a circular motion.

Unlike the nature of the sound spectrum like rapid cycles per unit time, period was studied on larger scales due to solar system and planet orbits. In early 17. Century, German astronomer Kepler introduced his third law which is called the Period's Law. It is indicated that the orbit period of a planet around the sun is proportional (by 3/2 exponential factor) with the distance between the planet and the sun.

$$T = \frac{1}{f} \tag{2.12}$$

T: Period (s)

f: frequency (Hz)

As indicated in equation (2.12), period and frequency are inversely proportional.

Frequency is the number of oscillations per unit time which is a second. The frequency of the simple harmonic motion is given as follows:

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{m}} \tag{2.13}$$

f: Frequency (Hz)

K: Spring constant (N/m)

m: Mass (kg)

According to equation (2.13), one may note that to double the frequency; the mass may be reduced to one-fourth of its size or the spring constant has to be made four times bigger (Rossing, 1993)

Frequency is the rate of vibration. It is the number of oscillations in a unit time which is one second. One count of vibration in one second is called 1 Hertz (Hz) as an example. Human ear is sensitive between 20 and 20000 number of oscillations in a unit time although most of the acoustic music instrument fundamental frequencies reach up around 4000 Hz. (Zeren, 2007)

In addition to the mass spring system, pendulum, a spring of air and a Helmholtz resonator are given as examples of systems that vibrate in simple harmonic motion.

The frequency of a pendulum with relatively a small angle compared to its length is given as follows:

$$f = \frac{1}{2\pi}\sqrt{\frac{g}{l}} \tag{2.14}$$

A spring of air frequency is given as follows:

$$f = \frac{1}{2\pi}\sqrt{\frac{\gamma.P.A}{m.l}} \tag{2.15}$$

A Helmholtz resonator frequency is given as follows:

$$f = \frac{C}{2\pi}\sqrt{\frac{A}{V.l}} \tag{2.16}$$

Wave length is the distance occupied in meters within one complete cycle for the given frequency.

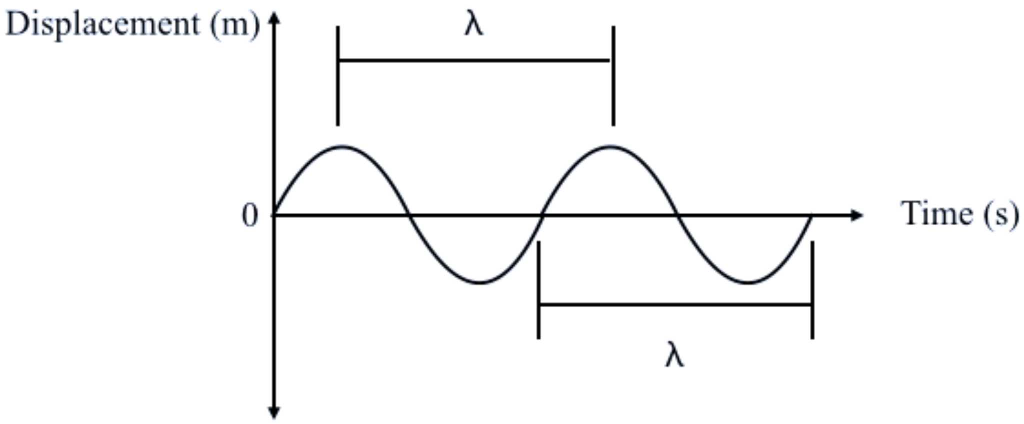

Figure 2.5: Wavelength of a periodic motion.

The distance between two node or antinode points would be the same for a specific constant frequency. As Figure 2.5 denotes; wavelength keeps constant if the vibration motion is periodic.

Wave length and the frequency are inversely related to each other. As equation (2.17) indicates the relation, if the sound speed is constant in given conditions then frequency and wavelength are inversely proportional.

$$C = f \,.\, \lambda \tag{2.17}$$

C: Sound velocity (m/s)

f: Frequency (Hz)

λ: Wave length (m)

Amplitude defines the energy level of the moving air molecule. It can be expressed as a distance taken in the acoustic environment as meter or pressure change according to a reference point as Pascals or it may be transferred in electronic circuits as Volts. Greater amplitudes are achieved with greater initial forces like musical dynamics such as plucking or hammering a string.

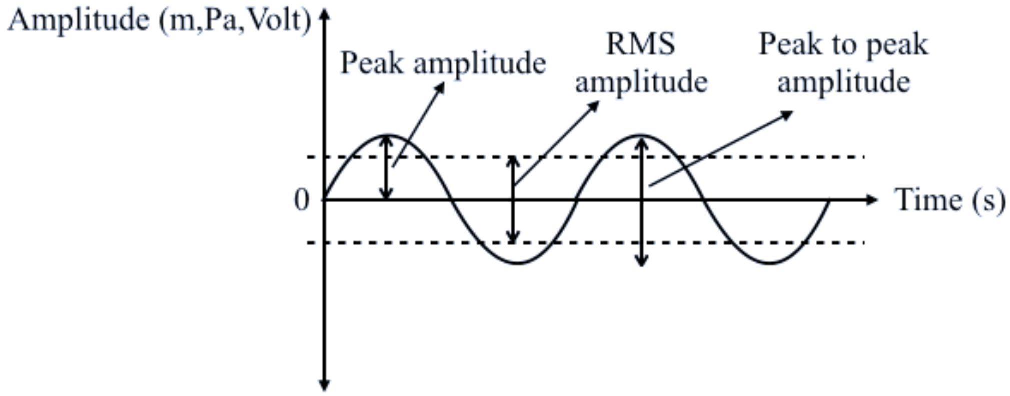

Figure 2.6: Different expressions of sound amplitude values.

As we can see from equation (2.13); the frequency is independent of the amplitude property of the sound. The same frequency may be perceived as the unison pitch may vary in terms of musical amplitude; like piano or forte. We may consider a violin duo playing a 440 Hz (pitch A) at a time, while one is playing piano (a soft touch of impact to strings) and the other one is playing the same as forte (a hard hit on strings).

Envelope is the change of amplitude over time. It has four characteristic stages to define sound like attack, decay, sustain and release. Attack is the time for the amplitude to reach its maximum value starting from zero. Decay is the time for the amplitude to take its optimum value. Sustain is the duration of that optimal stage of the amplitude values. Finally, release is the last stage for the amplitude to get back to its initial state of zero value presenting no motion at that given time.

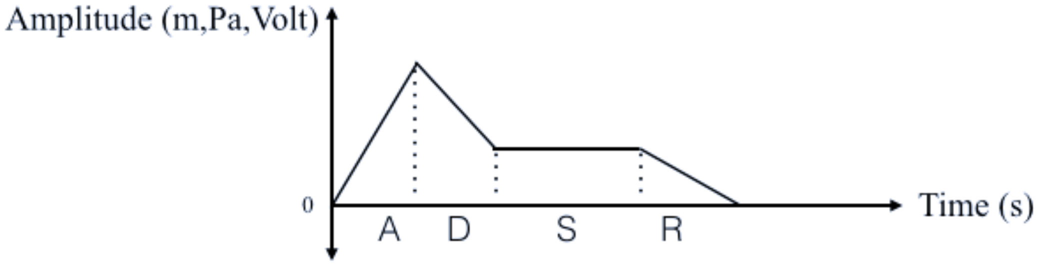

Figure 2.7: Envelope of the sound wave; presenting amplitude changes over time.

Harmonic spectrum would be another important topic while we keep presenting the wave characteristic parameters. Research on string vibration and therefore modal analysis dates back to Pythagoras; as it is an easy way to see and describe harmonics and overtones as a series of integer multiples of the fundamental frequency.

Fundamental frequency is the first harmonic. And the first overtone is called the second harmonic respectively.

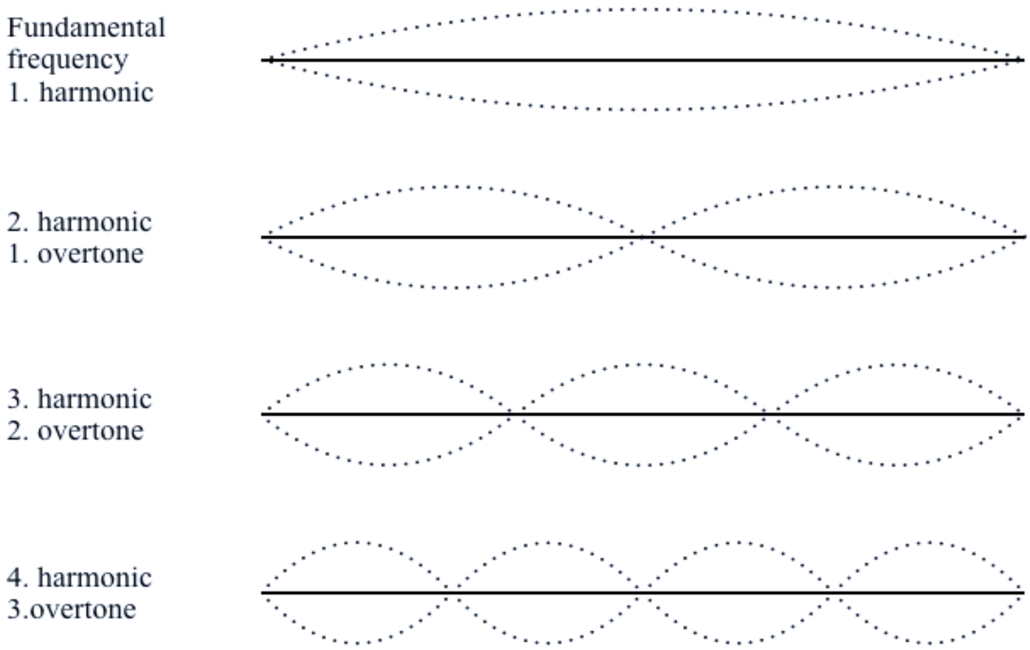

Figure 2.8: Fundamental frequency, harmonics and overtones on a string.

The first four harmonics of an attached string with both ends can be seen in Figure 2.8. One should notice that every adding multiple of the frequency is also dividing the string proportionally. For example; second harmonic is dividing the string in equal two regions and the third harmonic (second overtone) is dividing the string in three equal parts and so on.

Phase is the sound wave's angular location information within the time reference. Most of the practical issues are recognized about the subject when more than one wave is interacting in the sound field. Hearing with two ears may be a subject of this issue. Time arrival differences of the sound reaching two ears may cause constructive or deconstructive interferences, resulting amplitude changes because of the phase issue as an example.

$$\phi = f \, . \Delta \, t \, . 360 \tag{2.18}$$

Φ : phase difference (angle expressed with degree)

f: frequency (Hz)

Δ t: time difference (seconds)

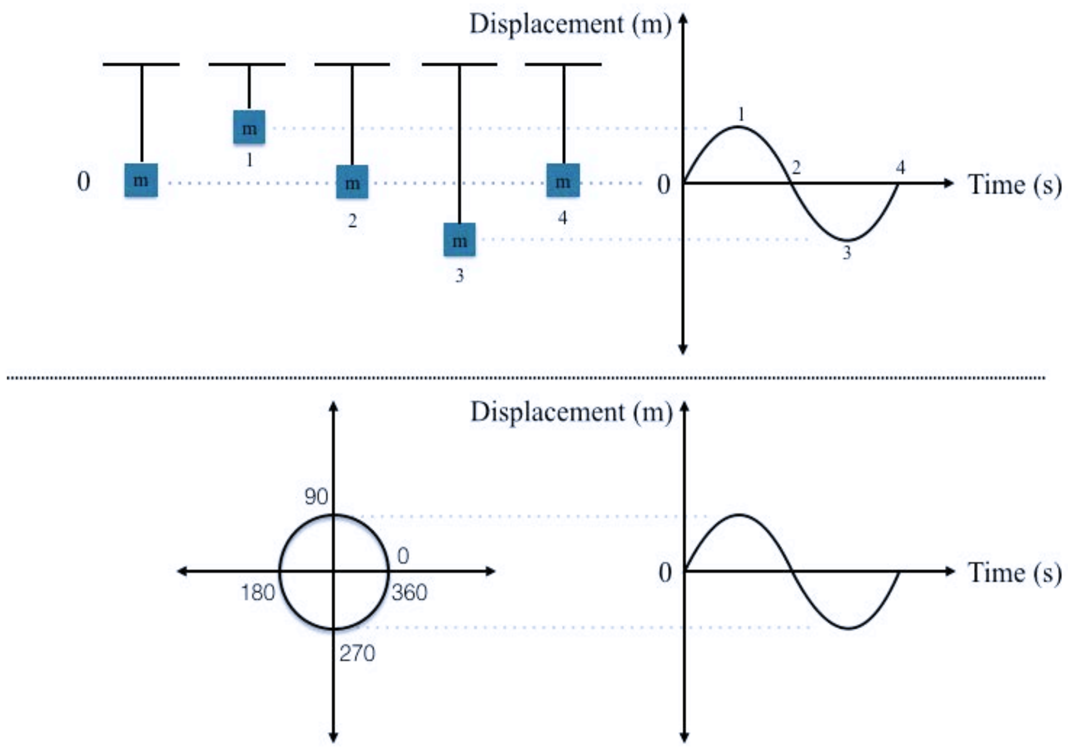

Figure 2.9: Sound wave phase presentation according to one periodic cycle.

The velocity and the pressure components of the sound wave are following each other with a 90^0 phase difference. So when the velocity component gets its maximum value then the pressure component takes its minimum value or vice versa.

2.1.2 Vibrating system properties

Energy, damping and resonance topics will be presented below in order to describe vibrating system properties. In acoustics, mechanical energy is mostly the main concern which is closely related to do work. Vibrating systems have mechanical energy; mechanical energy is carried by the moving molecules in a sound wave. Energy like work is measured in Joules. Sometimes a distinction is made between energy of motion called kinetic energy and stored energy, called potential energy (Rossing, 1990).

The formula for the kinetic energy of a moving mass is given as follows:

$$KE = \frac{1}{2}\, m.C^2 \tag{2.19}$$

KE: Kinetic energy (Joule, Nm)

m: mass (kg)

C: Velocity of the moving particle (m/s)

The formula for the potential energy of a moving mass is given as follows:

$$PE = \frac{1}{2}\, K.y^2 \tag{2.20}$$

PE: Potential energy (Joule, Nm)

K: Spring constant

y: the distance, stretched or compressed from the equilibrium position.

The vibrating system considered so far is has a single coordinate sufficient to describe its motion. In other words; it has one degree of freedom. Vibrating systems with two or more degrees of freedom have more than one mode of vibration. Different modes would generally have different frequencies. Adding a third mass to the system adds additional modes of vibration. These modes of vibration may be seen both in longitudinal and transverse types.

In mass-spring vibrating systems, each new mass added will end up with a new mode of longitudinal and transverse vibration. As the number of masses increases the system takes on a wave like appearance. In fact, a vibrating music instrument string can be thought of a mass-spring system with very large number of masses (Rossing, 1990).

Resonance would be an important topic in addition to vibrating system properties which were presented above. Every vibrating system has a natural frequency. A mass-spring system like Figure 2.1 would be given as an example if we think that the spring is attached to a crank. Assuming the crankshaft revolve at a frequency (f) which can be thought as an external force to the system and let the mass-spring system's natural frequency be (f_0). If f is varied slowly, the amplitude of the mass is observed to change, reaching its maximum amplitude when f is equal to f_0.

$$Q = \frac{f_0}{\Delta f} \tag{2.21}$$

Q: Quality factor; sharpness of the resonance

f_0: natural frequency of the system

Δf: line width

Q is defined as the sharpness of the resonance. Just as the maximum amplitude, Δf depends on the energy loss due to friction or damping. For a heavily damped system, Δf is large and the amplitude would be relatively small. Conversely, for a system with little loss; a sharp resonance with relatively small Δf and relatively large amplitude occurs (Rossing, 1990).

2.1.3 Psychoacoustic terms and their relationship with acoustic concepts

Pitch is defined as a property that makes the perception sound high or low; determines its position on a scale. For a pure tone or so called sine wave that have been discussed so far, the frequency determines the pitch. Despite the main determination is made by the frequency, the sound level may change pitch perception of a pure tone. In addition, the harmonic spectra and the duration also effects the pitch of complex sounds.

The pitch perception is highly related with the frequency component of the sound although the sound level and duration can make some changes as pointed out with the pure tones (Rossing, 1990).

Loudness is the perceived sound measurement parameter which is interacted with the sound power level, sound intensity level and finally sound pressure level both are measured with decibels (dB).

Sound power level (SWL) defines the emitted energy from the source of sound to all directions.

$$SWL = 10\log\frac{W_{actual}}{W_{ref}} \tag{2.22}$$

SWL : Sound power level (dB).

W_{ref} : 10^{-12} Watt

W_{act} : The actual sound power level (Watt)

Sound intensity level (SIL) is related within the region consisting between the sound source and the sound perceiver. The rule of inverse square law in free fields also counts on this sound intensity phenomenon. The sound intensity represents the flow of energy through a unit area (Howard, Angus, 2001).

$$SIL = 10 \log \frac{I_{actual}}{I_{ref}} \tag{2.23}$$

SIL : Sound intensity level (dB).

I_{ref} : 10^{-12} (Watt / m^2)

I_{act} : The actual sound intensity level (Watt / m^2)

Eventually the perceiver point of sound propagation is measured with sound pressure level (SPL).

The loudness is a subjective parameter which makes it a term of psychoacoustic. In general, it is related with the amplitude component of the sound wave (Rossing, 1992).

Timbre is used to define tone color or tone quality. The American National Standards Institute defines it as; ''Timbre is that attribute of auditory sensation in terms of which a listener can judge two sounds similarly presented and having the same loudness and pitch as dissimilar.'' An explanatory note is added; 'Timbre depends primarily on the spectrum of the stimulus, but it also depends upon the waveform, the sound pressure, the frequency location of the spectrum and the temporal characteristics of the stimulus.''

Pratt and Doak (1976) have suggested an alternative definition like;'' Timbre is that attribute of auditory sensation whereby a listener can judge that two sounds are dissimilar using any criteria other than pitch, loudness and duration.

It seems impossible to construct a single subjective scale of timbre of the type used for loudness (sones) and pitch (mels) so the term timbre may be described as a multidimensional attribute of sound (Plomp,1970).

In discussing timbre, it is important to distinguish between the timbre of steady complex tones and those that include transients and variations in time which brings out the envelope parameter beside the harmonic spectrum in the context.

A research on the timbre of the steady tones was carried out by Helmholtz (1877). Helmholtz demonstrated that the sounds of most musical instruments including the vocal cords) consist of series of harmonics that determine the timbre. He also described a way in which the ear could comprehend timbre. On the basis of his experiments, he formulated the following general rules;

Simple tones, such as those of tuning forks and widely sopped organ pipes, have very soft, pleasant sound, free from roughness but dull at low frequencies.

Musical tones with a moderately loud series of harmonics up to the sixth (such as those produced by the piano, the French horn, the human voice) sound richer and more musical than simple tones, yet remain sweet and soft if the higher harmonics are absent.

Tones consisting of only odd harmonics (narrow stopped organ pipes, clarinet) sound hollow and if many harmonics are present, have a nasal character. When the fundamental frequency predominates, the quality of the tone is rich; when the fundamental is not sufficiently strong, the quality of the tone is poor.

Complex tones with strong harmonics above the sixth and the seventh are very distinct, but the quality of the tone is rough and cutting.

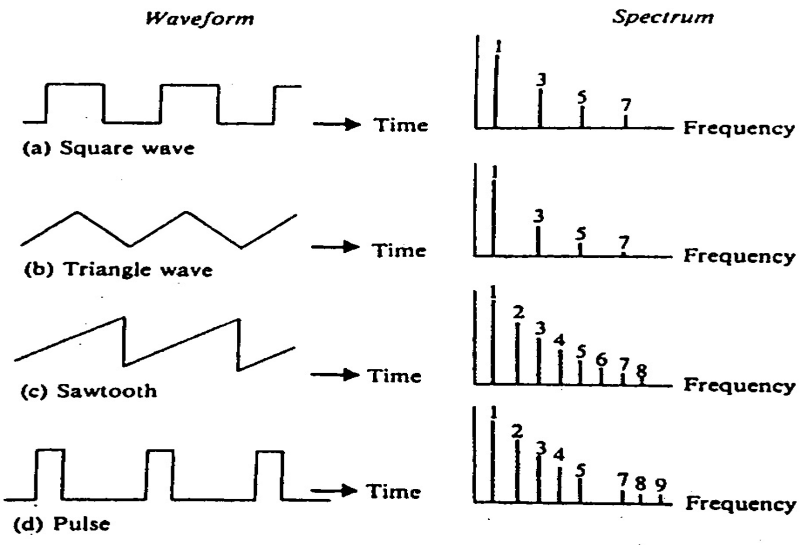

Figure 2.10: Basic waveforms and their harmonic spectrum (Rossing, 1990).

Spectra of four different waveforms are presented in Figure 2.10. Spectra can be defined as a specification of the strengths of the various harmonics.

The square wave is composed of only odd numbered harmonics with amplitudes in the ratio of 1/n. Therefore, if the fundamental frequency is f and the amplitude is A; the other overtones in the spectrum will have frequencies of (3f,5f,7f,9f …), and (A/3, A/5, A/7, A/9…)

The triangle wave has odd numbered harmonics with amplitudes in the ratio of $1/n^2$. Therefore, if the fundamental frequency is f and the amplitude is A; the other overtones in the spectrum will have frequencies of (3f,5f,7f,9f …), and (A/9, A/25, A/49, A/81…)

The sawtooth wave has both odd and even numbered harmonics with amplitudes in the ratio of 1/n, thus presents amplitude values like (A, A/2, A/3, A/4…)

The determination of the harmonic components of a periodic waveform is called Fourier analysis, after Joseph Fourier (1768-1830). His mathematical theorem states that;'' Any periodic vibration, however complicated, can be built up from a series of simple vibrations, whose frequencies are harmonics of a fundamental frequency, by choosing the proper amplitudes and phases of these harmonics''. Contrary, constructing a complex tone from its harmonics (the opposite of Fourier Analysis) is called Fourier synthesis. The terms spectrum analysis, harmonic analysis and sound analysis are sometimes used to describe Fourier analysis that is applied to sound.

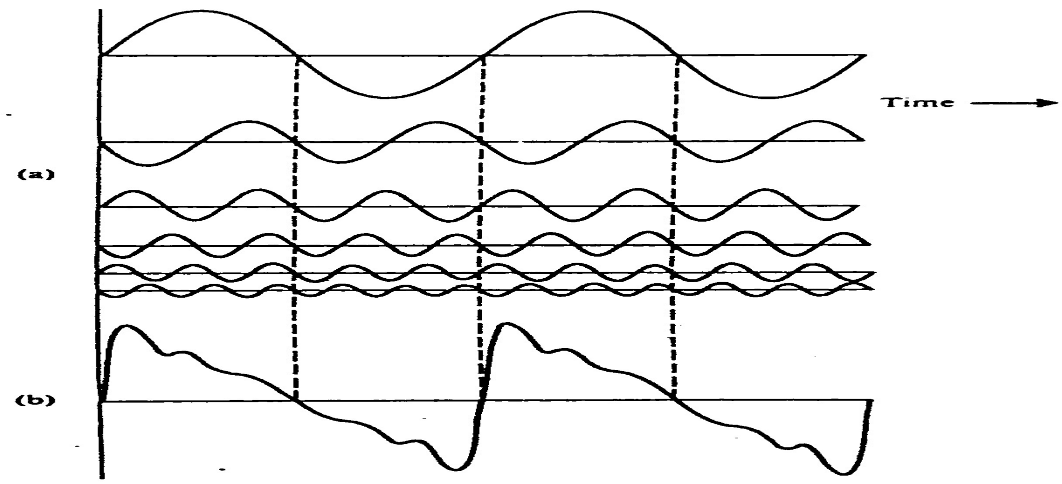

Figure 2.11: Fourier synthesis of a sawtooth wave (Rossing, 1990).

In Fig. 2.11 an illustration is presented how Fourier synthesis works. In a section, first six harmonics are shown individually. When combined in proper phase relationship, the first six harmonics summed and approximate to a sawtooth wave. Continuation of adding higher harmonics may give the resulting waveform more like a sawtooth shape than the figure.

Another aspect like a stimulus placement variable may end up with different harmonic series is presented in Figure 2.12.

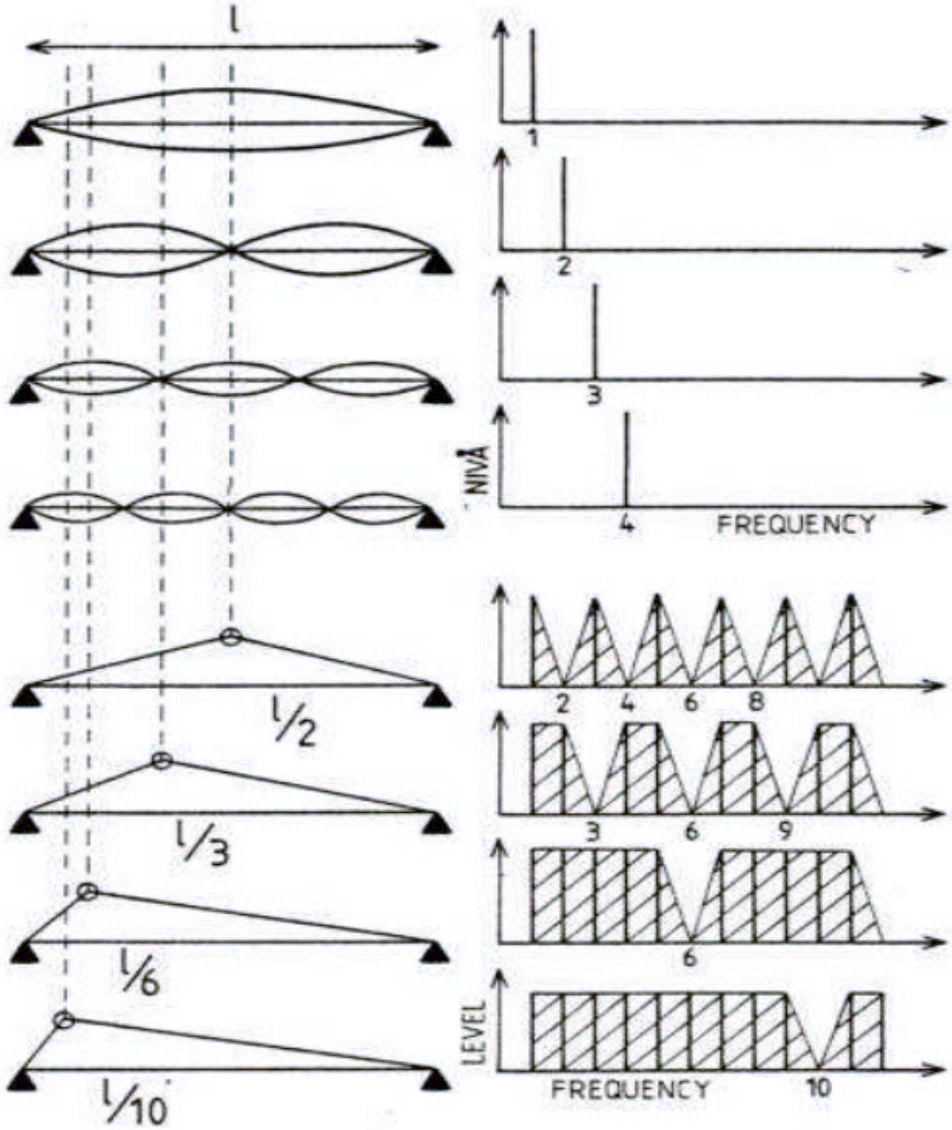

Figure 2.12: Harmonic series on a string according to stimulus placement (Jansson, 1983).

One should notice that; according to the proportion of the stimulus over the entire string, presents an absence like a notch filter of the related harmonic series in Fig. 2.12. Another parameter can be added to previous string vibration analysis, introducing musical instrument's constructional behavior; like top plate vibrations.

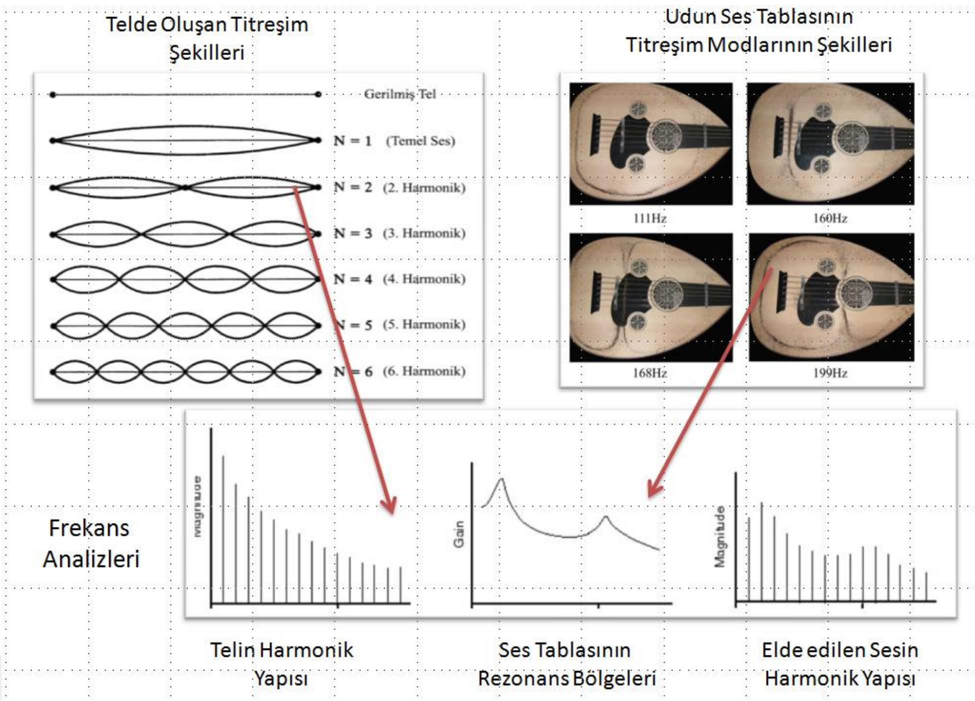

Figure 2.13: String and sound plate effect on harmonic spectrum (Değirmenli, 2015).

Figure 2.13 presents how the sounding instrument's harmonic series is effected by the medium on which the strings are attached. On the left above string harmonics are listed and the related harmonic structure of the string is given in left below. Right above Oud's top plate natural frequency nodal points are presented in the form of Chladni patterns. Middle below is showing the resonant frequencies of the top plate as peak points of the frequency spectrum. These points are also called natural or Eigen frequencies. Finally, right below presents the resulting sound's harmonic spectrum.

2.2 Kanun Instrument

As a plugged string instruments family member, Kanun has 22 to 27 plastic made strings with three strings on each course approximately providing three and a half musical octaves (Karadoğan, 2010). It has a half trapezoid shape with dimensions of 95 to 100 cm length, 38 to 40 cm width and 5 cm depth.

Figure 2.14: Kanun instrument view during performance.

Beside its delicate form, ornamented texture and its complex view, the instrument may be categorized as a simple acoustic model like; a trapezoid box shaped body and strings attached to it.

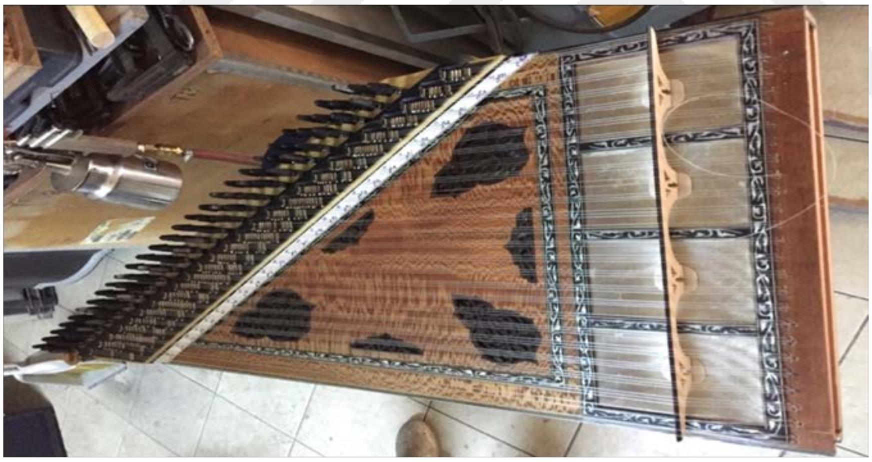

Figure 2.15: Kanun instrument top view.

String and body connectors are called bridges and these are the energy transmitters in the solid environment beside the air movement around the instrument. The box is formed with the main carcass and its covered with the back and top plates.

Figure 2.16: Kanun instrument body view with back and top plates.

Back plate is made of MDF and top plate is made of wood. Main carcass is also supported with various wood elements to increase reflection and therefore reverberation inside the instrument body. (Şaştım, F., 2019)

Figure 2.17: Kanun instrument skin covered bridge construction view.

Reverberant energy is also supported and radiated with the help of the composite material design, a membrane or skin is located beside the top plate. Reinforcing impact energy transmission and increasing reverberant sound is giving the instrument a unique tone of depth via radiation.

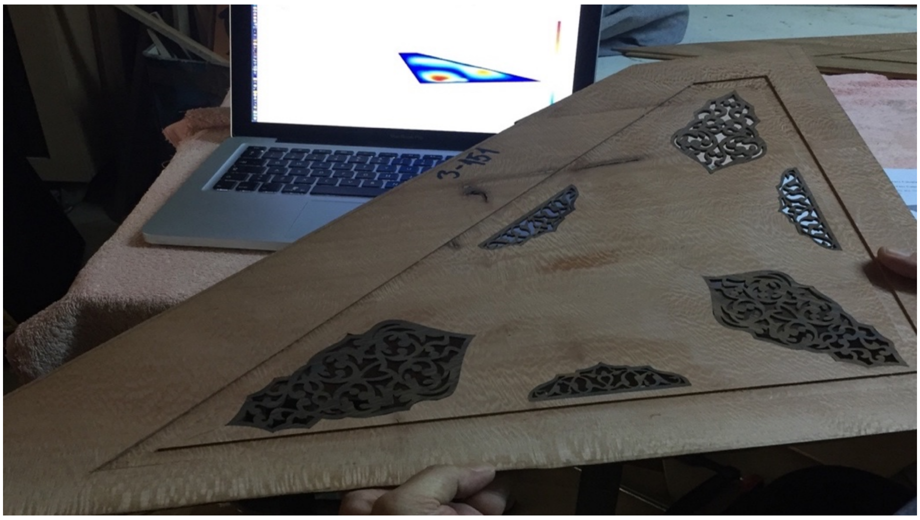

Figure 2.18: Kanun instrument top plate view as a Helmholtz resonator.

Top plate is drilled with few ornaments, making the air flows in and out of the body which makes the whole system a Helmholtz resonator. These air flow openings are presented in Figure 2.18.

Plastic made three string groups are also subject to sympathetic vibration due to mechanical coupling and sound energy radiation field beside their chorus effect on timbre. Other mechanical tuning parts and switches on the instrument makes it possible to fine-tune in cents for micro tunings.

Figure 2.19: Kanun instrument string groups near view.

As can be seen in Figure 2.19, Kanun instrument string groups are attached via small metal fine-tuning head machines; allowing to perform microtonal musical scales of Turkish music.

The tendency to pick up particularly the top plate as a study subject is highly related with the study as presented by (Lee et al., 2016) briefly mentioning the top plate as a potential in future research to increase the loudness of the instrument. According to Lee, (2016) ''The soundboard is the most important part of the good quality guitars. Research has proven that the top plate plays an increasingly important role compared to sound hole, back plate and the bridge at high frequencies''.

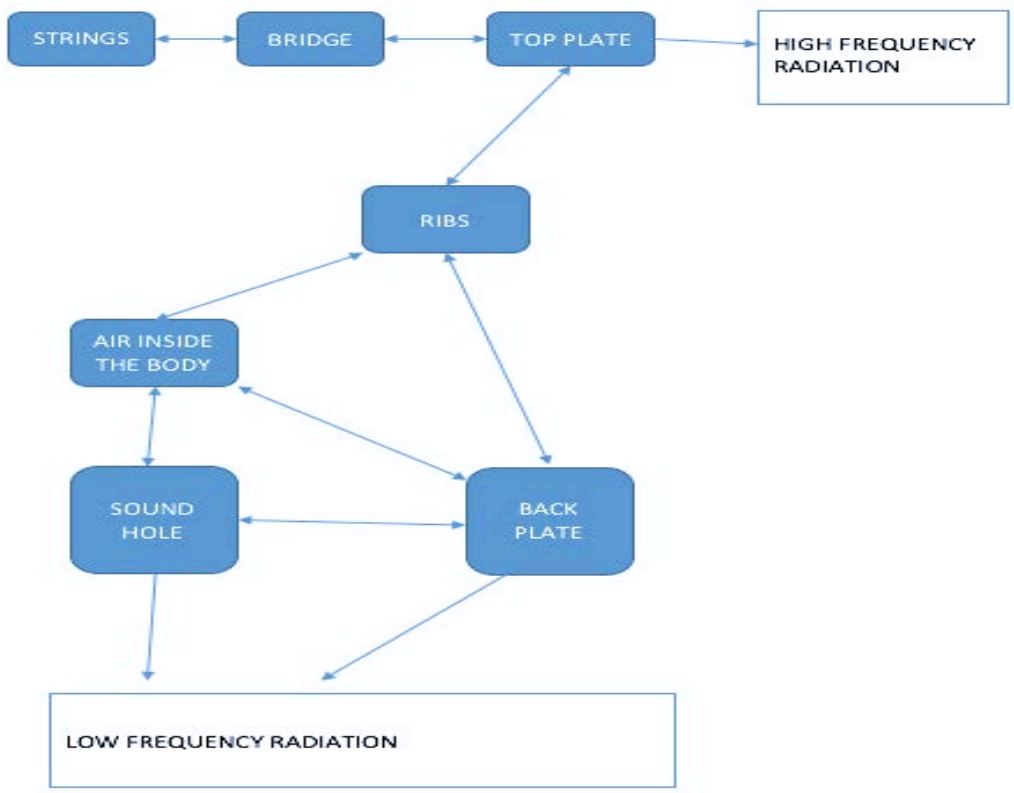

Figure 2.20: Simple schematic of a guitar sound radiation. (Rossing, 1992)

Although the shape, dimensions and form of the Kanun differs from the guitar, the model assembled by Figure 2.20 actually works for Kanun as a basis. We may add the membrane part of the body beside the top plate for a Kanun model. Sound hole area is also extended through the top plate area, opposing a one single unit for the guitar. Another difference is the three group string sections which also present sympathetic vibrations practically more than a guitar.

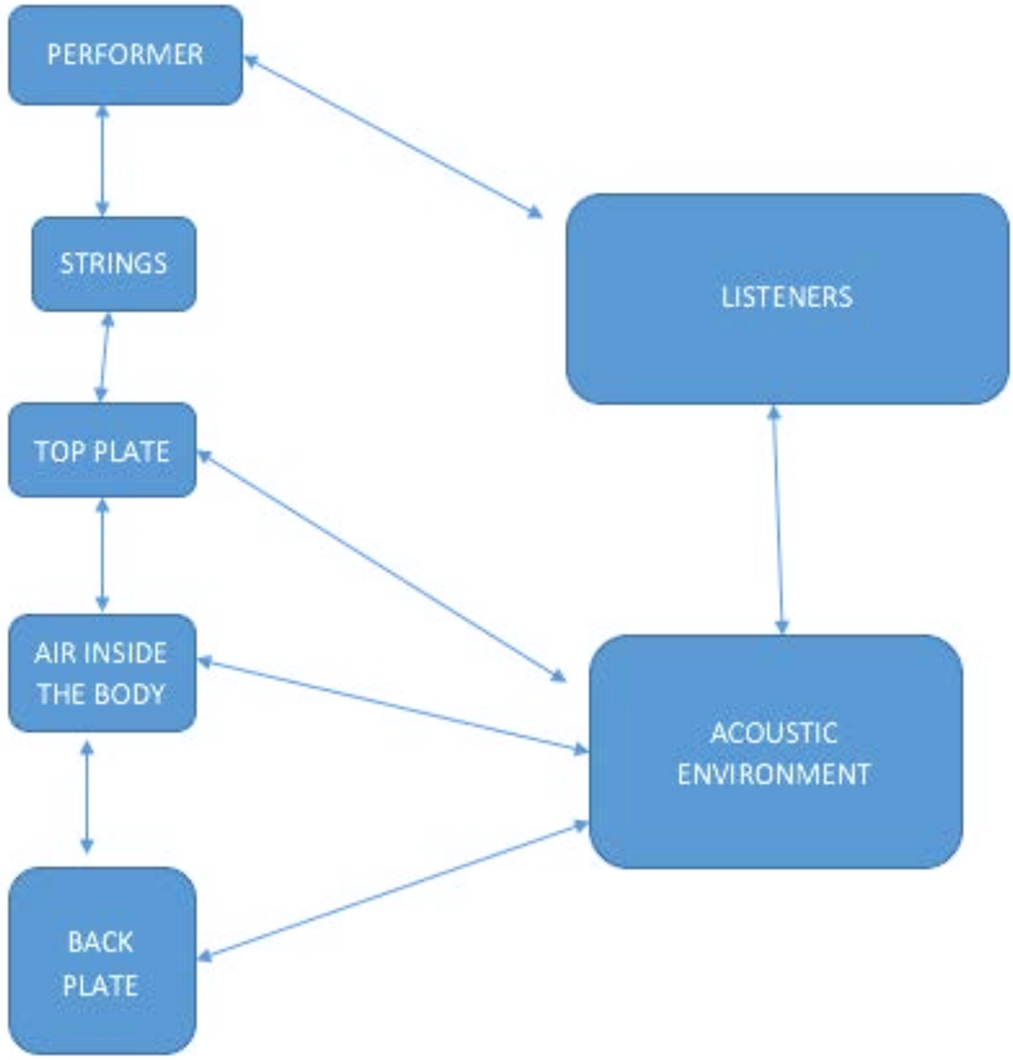

Figure 2.21: Schematic of a guitar sound. (Wright, 1996)

All these different parts of the construction interacting each other within the acoustic environment during the performance. It would be ideal to remind that this study focuses only the top plate component of these sound schematics above in (Figure 2.20 and Figure 2.21).

Behavior of soundboards depending on sympathetic vibrations is the main concern of this study. Since sound radiation depends on air displacement, radiation from a vibrating string is greatly enhanced by the sympathetic vibration of a soundboard. Violins, guitars, cellos, lutes and other stringed instruments as well as piano and harpsichord depend almost completely on sympathetic vibrations of the wood sounding box for radiation of their sound. Most of the sound radiation in these instruments comes from sympathetic vibrations of the top plate which is driven by the vibrating strings through the bridge. The top plate has many resonances of its own distributed throughout the playing range, and these resonances, to a large part, determine the quality of the instrument (Rossing, 1992).

Since the instrument's quality can be taken as timbre, the main focus of the study will be picking up the top plates and analyze their effect in instrument's playing frequency range comparatively.

3. EXPERIMENTAL STUDY

In this chapter a comparative experimental study of top plate resonant frequencies will be assembled. The method is a roving hammer impact's frequency measurement through the predefined points of the spruce and the plane tree top plates. The tools for the measurement and the test results are also presented for both.

3.1 Roving Hammer Test

The experimental study at this starting stage of the work is called a roving hammer test. Briefly, the top plates are impacted with a hammer. Then the pressure and velocity information is captured with an accelerometer and transmitted to a computer to analyze. These velocity and pressure changes leads to a displacement while vibrating. The displacement graphs or the amplitude changes may be seen in the computer modeling and analysis chapter.

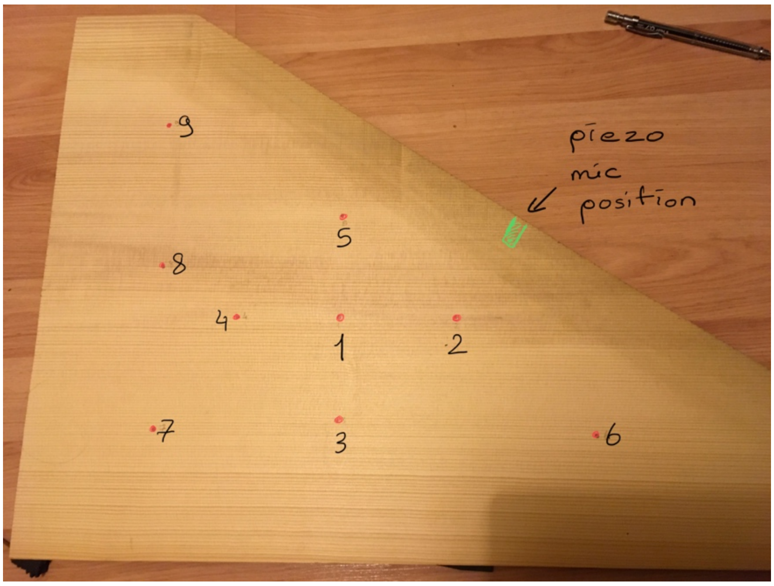

Figure 3.1: Hammer test impact points and the accelerometer position for spruce.

Experiments are performed on both plates, as spruce can be seen above in Figure 2.1; the numbers indicate the hammer impact positions on the top plate. The hammer is Dytran Dynapulse with a built in Dytran 5800b3 force transducer. A steel tip was used to get proper excitation in the range of interest, which was from 20 Hz to 1500 Hz, a range that starts from the human audible extreme low frequency to Kanun's highest pitches of fundamental frequencies. The plates are freely suspended and therefore extremely lightly damped. The green highlighted region in the figure is covered with an accelerometer. The plates were drilled tiny enough to be hanged from their corners. The plates were suspended from their corners using soft rubber bands. Thus, the damping effects due to boundary conditions were kept small enough to be neglected.

The output signal was measured using a small light-weight DJB Instruments Accelerometer Type A/123/e. The force and vibration signals were analyzed using the 01 Db multichannel analysis system type Db4. Post processing of the data was carried out using the NetdB software. Using a signal analyzer, the resonant frequency of the plates could be monitored.

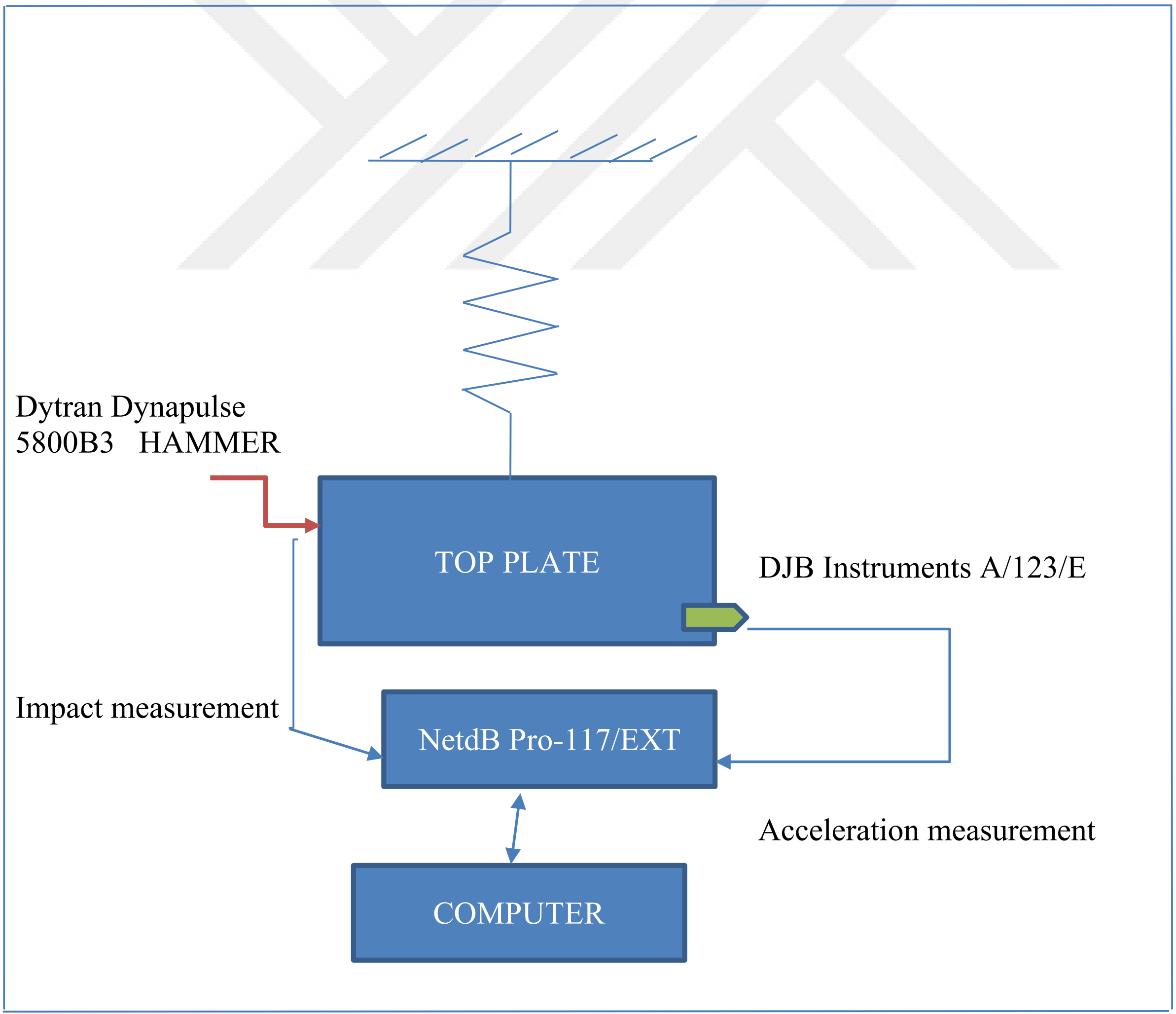

Figure 3.2: Schematic model description of the hammer test experiment.

Appendices A, B and C are covered with the results of the hammer test experiments with spruce plane tree and spruce and plane tree together respectively.; as frequency response functions (FRF's) obtained from the measurements.

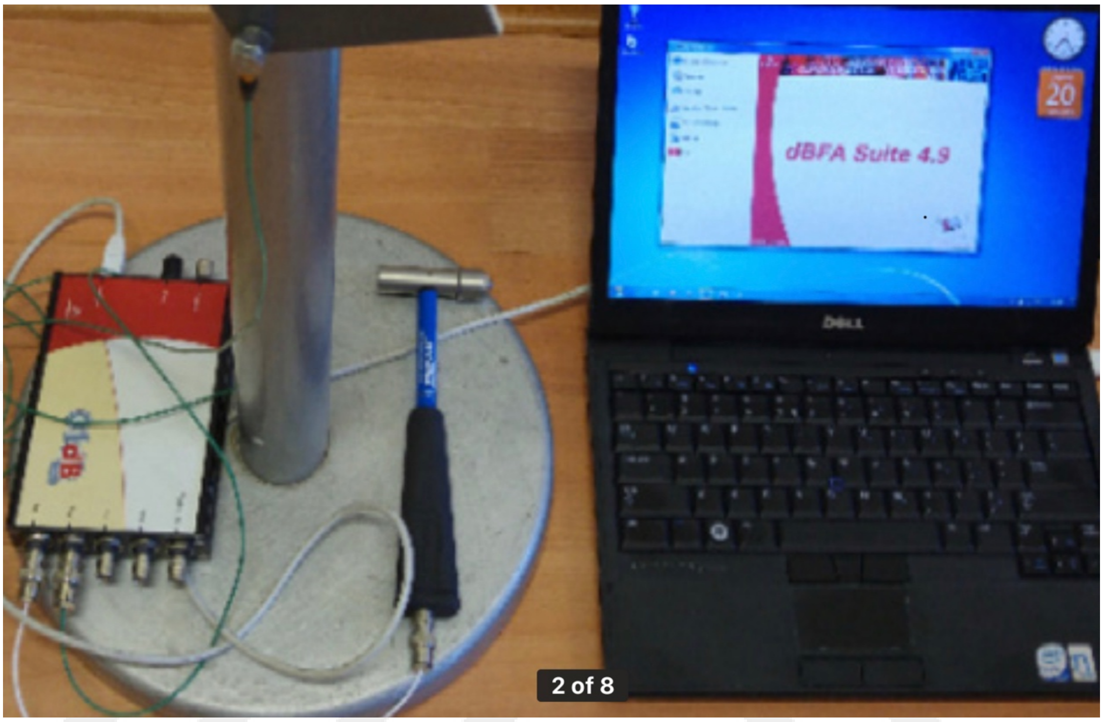

Figure 3.3: Equipment view for hammer test experiment.

3.2 Hammer Test Results

As the experiments are carried through with nine individual points on the top plate (see Figure 2.1), only one-point analysis will be described as an example. All points resonant frequencies on both woods can be seen from the Appendices A, B and C. Those data for nine individual points are collected and kept. Therefore, that would be a reference point from the real world parameters to compare and verify with the following 3D Physical Modeling stage.

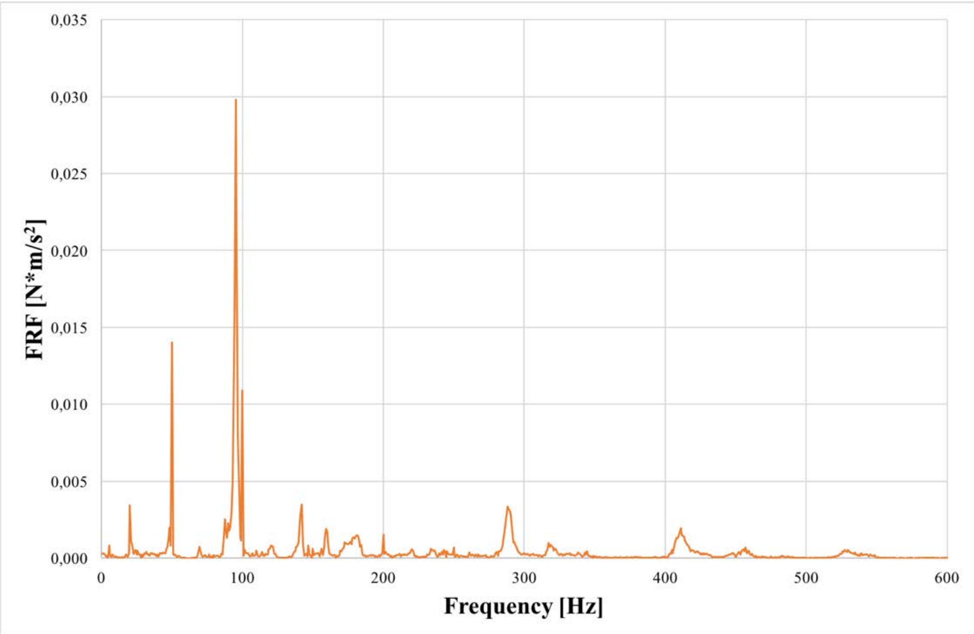

Figure 3.4: Spruce natural frequencies detected by the experiment on point 1.

x axis indicates the frequency while y axis indicates the FRF which presents the acceleration through the impact force.

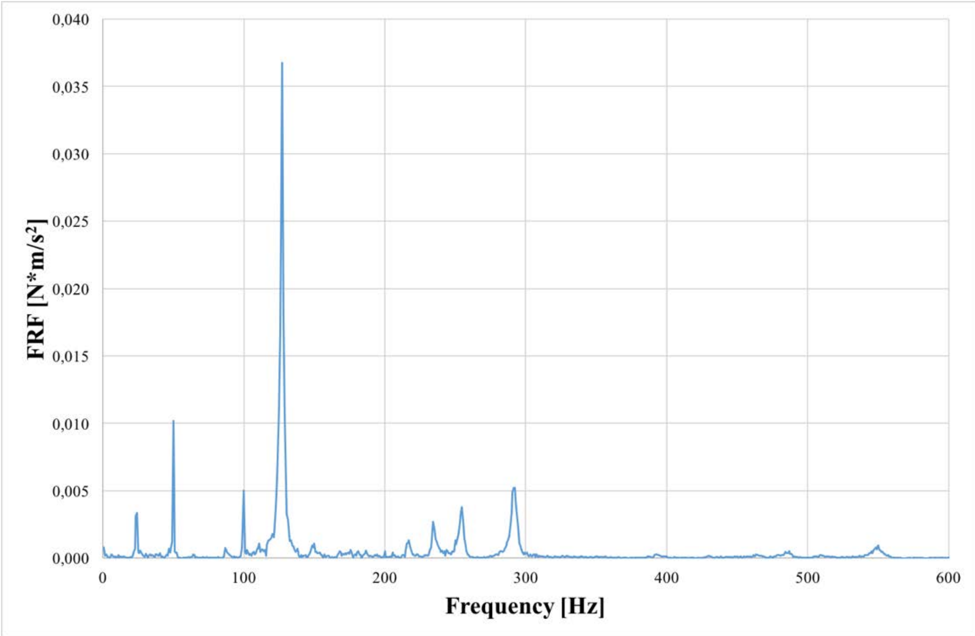

Figure 3.5: Plane tree natural frequencies detected by the experiment on point 1.

x axis indicates the frequency while y axis indicates the FRF which presents the acceleration through the impact force.

Figure 3.4 and Figure 3.5 shows the natural frequencies of spruce and plane tree respectively. The highly accelerated (so; displaced) parts of the frequency spectrum may be seen as peaks in the frequency range and those frequencies are listed below in Table 3.1 and Table 3.2 in order to make comparison.

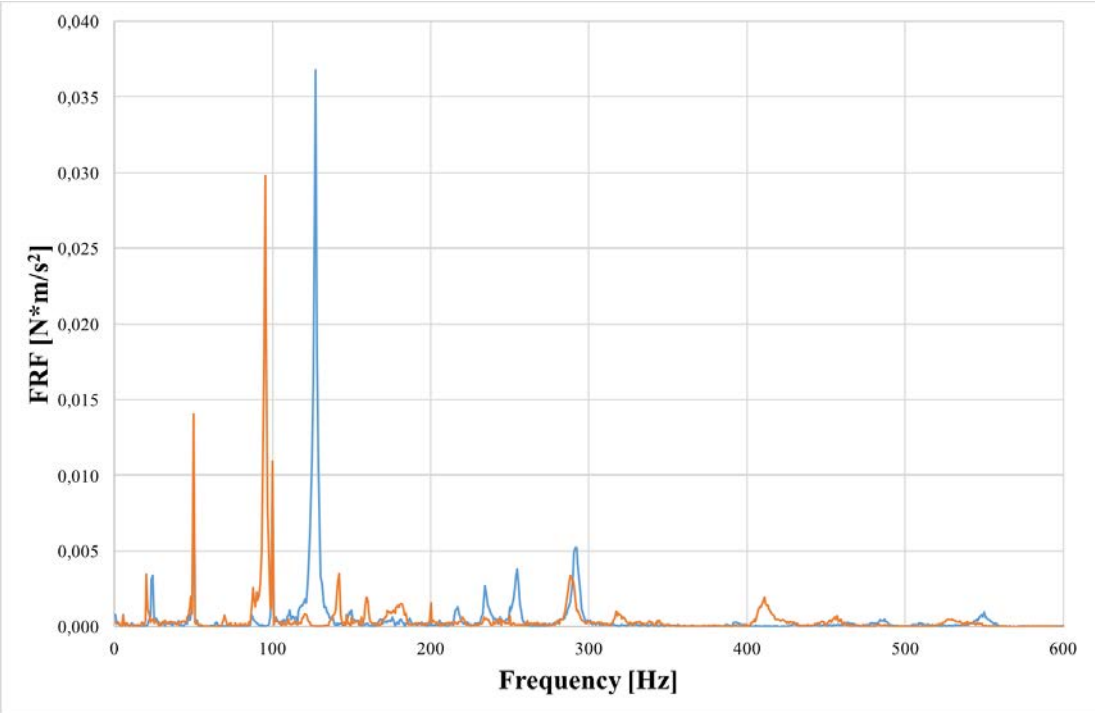

Figure 3.6: Spruce and Plane tree natural frequencies viewed together.

Figure 3.6 shows the natural frequencies of spruce and plane tree together for the experimental study. These frequencies are listed in Table 3.1 and Table 3.2 for spruce and the plane tree respectively.

Table 3.1: Spruce tree top plate natural frequencies of experiment results.

Frequency No	Point 1	Point 2	Point 3	Point 4	Point 5	Point 6	Point 7	Point 8	Point 9
1	20.31	20.31	20.31	20.31	20.31	30.47	31.25	50	20.31
2	50	50	50	30	48.44	50	50	87.5	30.47
3		69	69.53	50	95.31	95	96	100	50
4	95.31	95.31	94.53	95.31	141.41	120	120	141	87.5
5	142.19	120	121	121	159.38	141	141	159	120.31
6	288.28	158	158	159	182.81	159	159	200	141
7	410.94	182	182	200	200	182	182	408	184
8		236	200	288	289	200	223	431	305
9		378	289	409	378	239	250	485	339
10		483	340	460	482	289	271	525	459
11			410	484	546	319	289		484
12			544			339	341		545
13							432		
14							485		
15							546		

Natural frequencies of nine individual point are listed as experiment results viewed together in Hz for spruce.

Table 3.2: Plane tree top plate natural frequencies of experiment results.

Frequency No	Point 1	Point 2	Point 3	Point 4	Point 5	Point 6	Point 7	Point 8	Point 9
1	23.44	24.22	23.44	24.22	24.22	50.78	50	50	23.44
2	50	50	50	50	50.78	100	64.84	100	50
3	100	99.22	99.22	99.22	64.05	125	98.44	127	64.84
4	125	125.78	117	116.41	99.25	290	126.56	168	99.22
5	217	250	166	125	125.78	353	167.97	200	115
6	235	340	219	149	200	507	210.94	250	126
7	253.91		250	167	253.91	547	234.38	291	291
8	292.97		293	197.66	255.47		254.69	355	507
9			339	254.69	291.41		289.84	464	
10			355	290.63	392.19		339.84	489	
11			507	339.84	506.25		390.63	504	
12				355.47			507.81	547	
13				506.25			549.22		

Natural frequencies of nine individual point are listed as experiment results viewed together in Hz for plane tree.

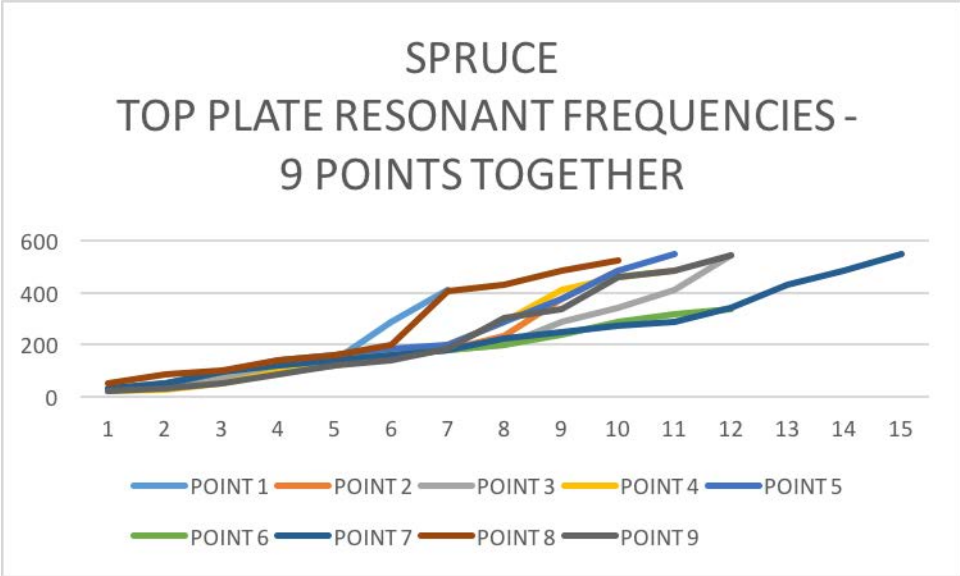

Figure 3.7: Spruce natural frequencies viewed together for nine points.

4. COMPUTER MODELING AND ANALYSIS

In this chapter, initially the finite elements method is presented as a computer modeling tool. And then, the top plate measured parameters like dimensions and the related material properties will be presented. Finally, the computer modeling results will be assembled. Spruce and plane tree top plates will be compared in terms of their natural frequencies.

4.1 Finite Elements Method

Virtually every mechanical phenomenon in nature can be described as the laws of physics in terms of algebraic, differential or integral equations. These equations can be related to various quantities of interest. There are two major tasks which involves most engineers and scientists studying physical phenomena, first; mathematical formulation of the physical process and the second; numerical analysis of the mathematical model.

The laws of physics and most often certain mathematical tools could be used as a background to obtain a mathematical formulation of a physical process. Mathematical formulations are usually expressed in differential equations. These would be related parameters or quantities of interest within the design aspect of a physical process. Concerning how the process works, development of a mathematical model of a process is achieved through assumptions. In numerical simulation, a numerical method and a computer is used to evaluate the mathematical model and estimate the characteristics of the physical process.

Derivation of the governing equations for most problems are not so difficult while their solution by exact method of analysis would be a challenging task. In this circumstances, approximate methods of analysis provide alternative means of solutions. The finite difference method and the variational methods such as the Rayleigh-Ritz and Galerkin methods are most frequently used in the literature.

Various variational methods such as Rayleigh-Ritz, Galerkin and least square methods, differ from each other in terms of the choices over integral form, weight functions and/or approximation functions. They all suffer from the disadvantage that the approximation functions for problems with arbitrary domains are not easy to construct.

FEM discards the disadvantages of the traditional variational methods by providing a systematic procedure for the derivation of the approximation functions over subregions of the domain.

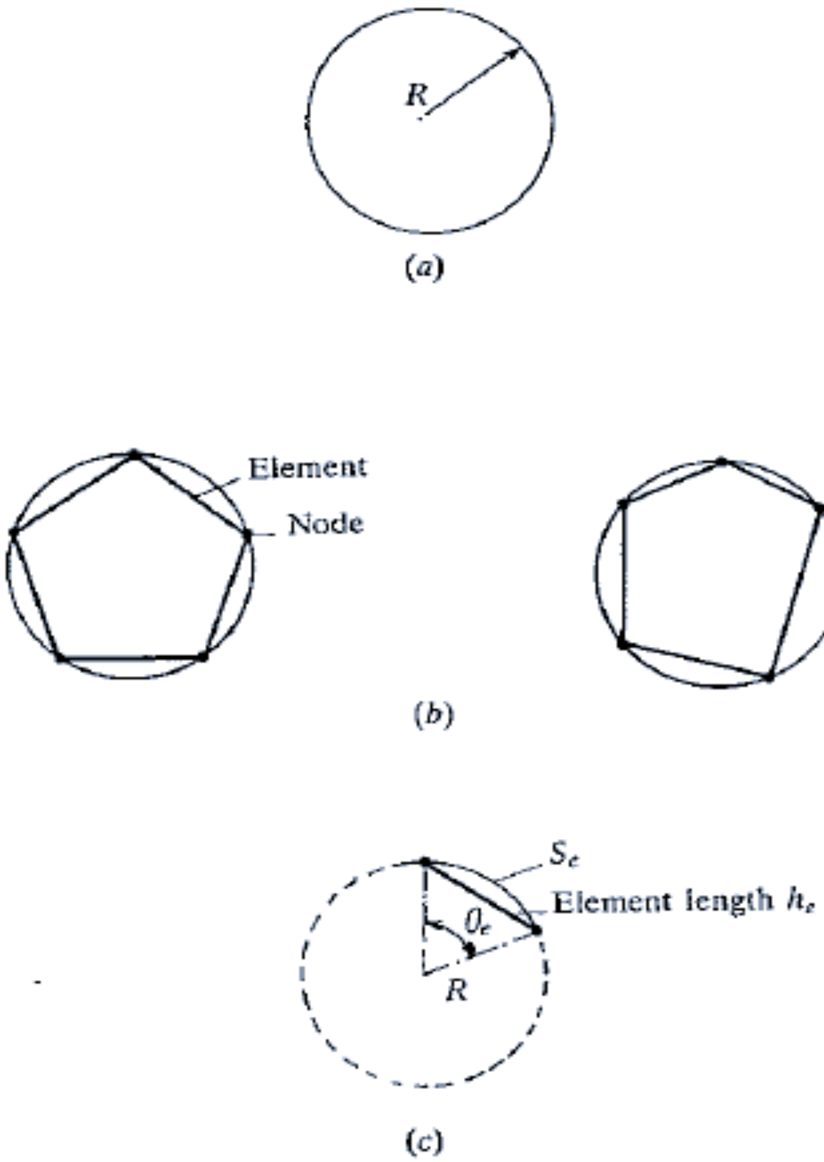

Figure 4.1: A basic problem of FEM. (Reddy, 1993)

R is the radius of circle (see Figure 4.1 a), the circumference of the circle is represented by uniform and nonuniform meshes which can be seen in Figure 4.1b, and a typical element h_e is given as the distance between two nodal points with an angle of θ_e is represented in Fig. 4.1c.

The method presents three basic features that also defines its advantage over competing alternative methods. First, the geometrically complex domain of the problem is represented as a collection of simple sub domains which are called finite elements. Second, over each finite element, the approximation functions are derived by a linear combination of an algebraic polynomials which are representing any continuous function. Third, algebraic relations among the undetermined coefficients like the nodal values are obtained by satisfying the governing equations, mostly in a weighted integral form over each element of the method. Thus the finite element method can be viewed, in particular, as an element wise application of Rayleigh-Ritz or weighted residual methods. (Reddy, 1993)

Approximation functions are mostly represented with algebraic polynomials and undetermined parameters represent the values of the solution at a finite number of preselected points called nodes, on the boundary and in the interior of the element.

Concepts from the interpolation theory are used for the derivation of the approximation functions that are therefore called interpolation functions. Degree of the interpolation functions depends on the number of nodes in the element and the order of the differential equations being solved.

Below are the basic ideas underlying the finite element method with an example like an approximation of the circumference of the circle:

This example presents the determination of the circumference of the circle using a finite number of line segments.

Figure 4.1: a: the radius of the circle, b: uniform and non uniform meshes used to represent the circumference of the circle , c: a typical element. (Reddy, 1993)

One can easily notice that when number of nodes within the domain are increased, the approximation function would behave ideally to this well known solution which is 2R times Pi. Ancient mathematicians estimated the value of the circumference by approximating by the line segments whose lengths were able to measure.

By summing the representing line 2segments, the approximate value of the circumference is obtained. This example illustrates several ideas and steps involved in the FEM analysis of the problem.

Considering the finite element discretization; the domain (the circumference of the circle) is represented as a collection of a finite number n of sub domains namely line segments. This is called the discretization of the domain. Each sub domain (i.e. line segment) is called an element. The collection of the elements is called the finite element mesh. The elements are connected at nodal points. In this circle example, circumference discretization is applied onto a mesh of five line segments. The line segments do not have to be the same length. If they do, the mesh is called a uniform mesh. If the line segment length is not constant these meshes are called non-uniform (see Figure 4.1 b)

Considering the element equations; a typical element; line segment is isolated and its required properties (i.e. length) are computed by some appropriate means. Let h_e be the length of element Ω^e in the mesh. For a typical element of Ω^e , h_e is given by;

$$h_e = 2R \sin\frac{1}{2}\,\theta_e \tag{4.1}$$

h_e: the length of the element
R: radius of the circle
θ_e: the angle subtended by the line segment; ($\theta_e < \pi$)

The above equations are called element equations. Ancient mathematicians most likely made measurements rather than using the formulas above to find the length of the element which is h_e in this particular example. (Reddy, 1993)

Assembly of element equations and solution can be seen with the Equations (4.2) and (4.3). The approximate value of the circumference of the circle is obtained by putting together the element properties in a meaningful way; this process is called the assembly of the element equations. It is based on a simple idea; the perimeter of the polygon is equal to the sum of each individual element.

$$P_n = \sum_{e=1}^{n} h_e \tag{4.2}$$

P_n: approximation to the actual perimeter. If the mesh is uniform, then h_e would be the same for each element in the mesh. Then; $\theta_e = 2\pi/n$, thus;

$$P_n = n\ (\ 2R \sin\frac{\pi}{n}\ ) \tag{4.3}$$

Convergence and error estimation for this case, we know the exact solution; $P = 2R\pi$. We can estimate the error in the approximation. We may shoe that the approximate solution of P_{an} converges to the exact P in limit as n goes to infinity. Considering the typical element Ω^e , the error of the approximation is equal to the difference between the length of the sector and that of the line segment;

$$E_e = |\ S_e - h_e\ | \tag{4.4}$$

where $S_e = R\ Q_e$ is the length of the sector.

Thus the error estimate of the given mesh would be;

$$E_e = R\ (\frac{2\pi}{n} - 2\ sin\ \frac{\pi}{n}) \tag{4.5}$$

The total error which is called the global error is given by multiplying E_e by n;

$$E_e = 2R\ (\ \pi - n\ sin\ \frac{\pi}{n}) = 2\pi R - P_n \tag{4.6}$$

Now it has been showed that E goes to zero if n goes to infinity.

Letting x = 1/n;

$$P_n = 2Rn\ \ sin\ \frac{\pi}{n} = 2R\ \frac{\sin \pi\ x}{x}$$

and

$$\lim_{n\to\infty} P_n = \lim_{x\to\infty} \left(2R\frac{\sin \pi x}{x}\right) = \lim_{x\to\infty} \left(2\pi R\frac{\cos \pi x}{1}\right) = 2\pi R \qquad (4.7)$$

As a result, it is presented that the circumference of a circle can be approximated as closely as possible by a finite number of piece wise linear functions.

As the number of elements is increased, the approximation is getting better. In other word, the error of the approximation is inversely proportional with the number of the elements.

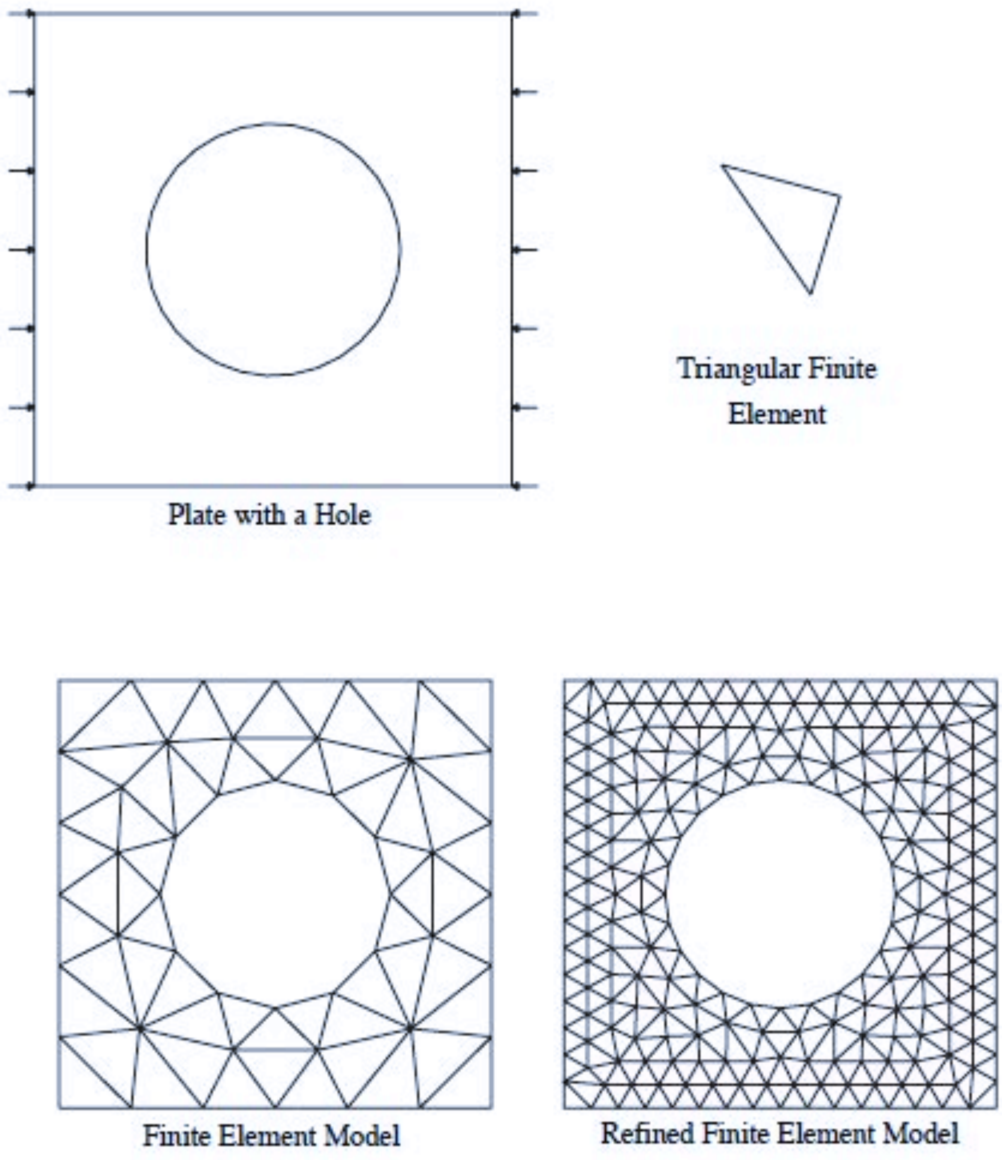

Figure 4.2 : Triangular mesh example with different number of nodes. (Fish and Belytschko, 2007)

A domain is divided into subdomains which are called finite elements and an approximate solution of the problem is developed over each of these elements in the finite element method. The subdivision process of a whole has two advantages;

It allows accurate representation of complex geometries and inclusion of dissimilar material properties.

It enables accurate representation of the solution within each element, to study and analyze local effects.

Three fundamental steps of the finite element method can be listed as; divide the whole into parts, seek an approximation to the solution over each part as a linear combination of nodal values and approximation functions, over each part, derive the algebraic solutions among the nodal values of the solution and assemble the parts to obtain the solution to the whole and FEM can also be presented as a numerical technique that employs the philosophy of constructing piecewise approximations of solutions to problems described by differential equations.

4.2 Physical Modeling Associated Parameters

Finite Elements Method (FEM) is used to identify the top plate resonant modes with the COMSOL software running on a PC computer.

Basically computer is used to obtain information about top plate vibration mode calculations. Therefore, the aim is to verify the results of the modeling study with the hammer test experiments. By modeling the geometry, defining the material properties and determining the physical conditions; modal patterns and their overtone frequencies may be seen after the computation of the related software.

Geometry, material density, Young modulus (including 3 axis;), Shear Modulus (including 3 axis; x, y, z) and the Poisson's ratio (3 planes; xy, yz, xz) are the variable parameters of the software calculations to identify top plate resonant modes.

At first the instrument top plate is 3D modeled with CAD module of the software and then the material properties like density, Shear and Young Modulus values and Poisson's ratio are defined for spruce and plane tree respectively.

Free modes are calculated first to approve the truly working 3D physical model. The goal is to get the hammer test experiment results as close as possible which are naturally present in the real world as a reference. And then the workflow order comes to fixed boundary mode calculations with the verified physical model. By doing this we ideally like to get the model much closer to real world conditions. Normally, the top plate is fixed with the instrument's structural carcass. So at this stage the model's four boundaries are fixed, letting the top plate's top and bottom surface for displacement.

Figure 4.3: Spruce top plate view.

Figure 4.4: Plane tree top plate view.

Spruce dimensions is given as follows:

Length bottom: 75 cm

Length top: 10,5 cm

Width: 44 cm

Height: 0,32 cm

Plane tree dimensions are given as follows:

Length bottom: 74 cm

Length top: 12 cm

Width: 43,5 cm

Height: 0,32 cm

Top plate final dimensions are practically being larger than the dimensions of the spruce and the plane tree so the top plates are actually laminated in practice. Note that the following practice of physical modeling process neglects this issue and studies the top plate as a one-piece homogenous vibrating surface which seems as it is to a listener eye.

Below are the material property values that have been used for physical modeling purposes. Note that the listed values are approximated and correlated with the concept of matching the model results to the experimental study results. Only the density and the Young Modulus values are approximated to match the modeling and the experimental studies. Material property values that are needed as a starting point were obtained from ''Wood Handbook - Wood as an Engineering Material'' by ROSS, R. J. (2010).

In order to initiate the numerical values of the material properties such as density can be seen below.

Density $_{\text{SPRUCE}}$: 355,6 kg/m^3

Density $_{\text{PLANE TREE}}$: 365 kg/m^3

Young Modulus values of spruce and Plane tree can be seen in Table (4.1). Elasticity implies that deformations produced by low stress are completely recoverable after loads are removed. Elastic ratios, as well as the elastic constants themselves, vary within and between species and with moisture content and specific gravity.

Table 4.1: Young modulus values for spruce and plane tree.

Young Modulus (MPa)	SPRUCE	PLANE TREE
E_X	2100	2500,2
E_Y	21000	12430
E_Z	1400	830,8

Table 4.2: Poisson's ratio values for spruce and plane tree.

Poisson's Ratios	SPRUCE	PLANE TREE
μ_{XY}	0,030	0,033
μ_{XZ}	0,467	0,604
μ_{YZ}	0,530	0,641

Table 4.3: Shear modulus values for spruce and plane tree.

Shear Modulus (MPa)	SPRUCE	PLANE TREE
G_{XY}	850	1653,2
G_{YZ}	726	919,82
G_{XZ}	35	12,43

4.3 Free Modes of Vibration Modal Analysis

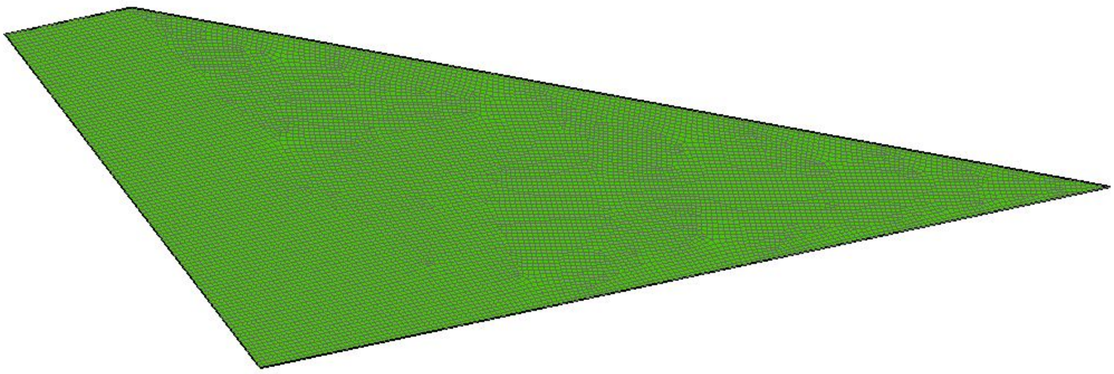

Figure 4.5: Top plate static view without vibration.

This static phase without vibration mimics the initial mass spring system shown in Fig. 2.1. The whole plate as a system rest in its equilibrium position when no external force is applied on it like the performers plucks or touches.

First two harmonics of spruce are given as follows:

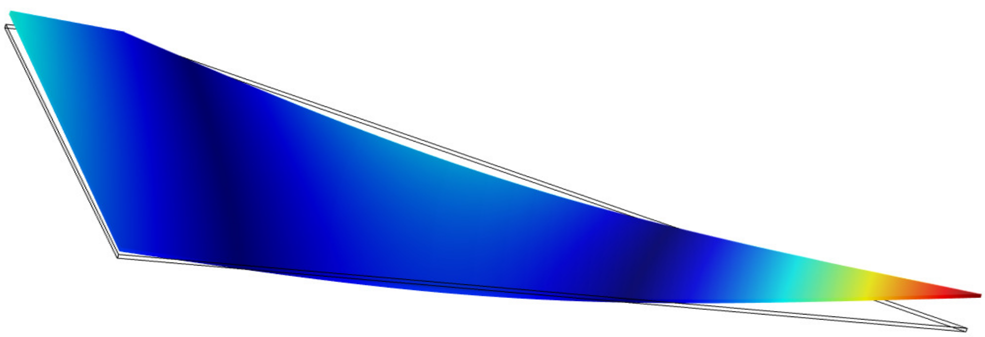

Figure 4.6: Free mode first harmonic frequency view for spruce – 20,31 Hz.

Above and below the modal shape of the natural frequencies can be seen for spruce; displacement from the equilibrium position is shown through colors. Starting from the low values from deep blue, blue, green, yellow and finally the highest values to the red. Plate's static view is also presented with black lines.

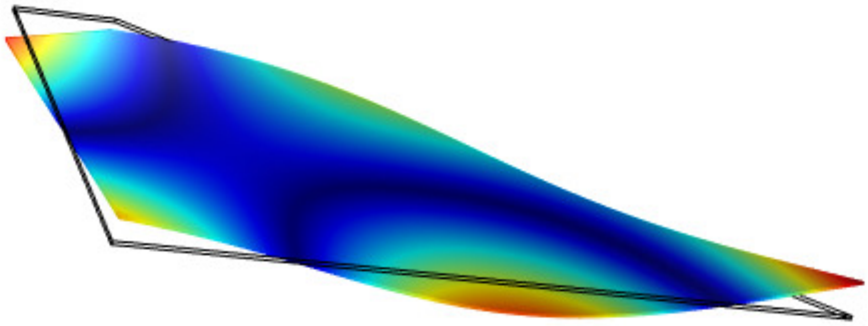

Figure 4.7: Free mode second harmonic frequency view for spruce – 43,88 Hz.

This harmonic series can be seen individually like Fig 4.6 and 4.7 and so forth. The natural frequencies are listed below with the hammer test results for a comparison to verify the model is truly and efficiently working. The results for first ten harmonics of spruce can be seen in Table 4.4.

First two harmonics of plane tree are given as follows:

Figure 4.8: Free mode first harmonic frequency view for plane tree – 23,44 Hz.

Above and below the modal shape of the natural frequencies can be seen; displacement from the equilibrium position is shown through colors. Starting from the low values from deep blue, blue, green, yellow and finally the highest values to the red.

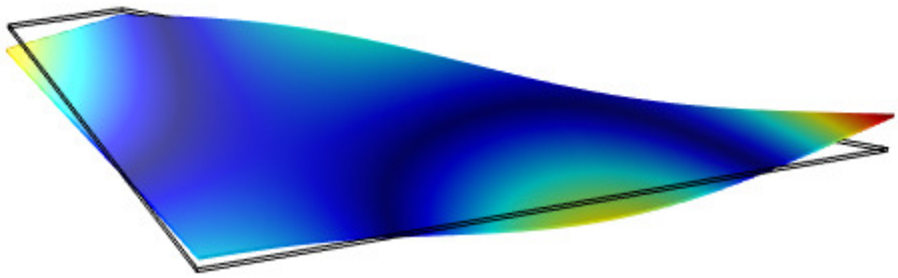

Figure 4.9: Free mode second harmonic frequency view for plane tree – 55,23 Hz.

As it is performed before, this harmonic series can be seen individually like Fig 4.8 and 4.9 and so forth. The natural frequencies are listed below with the hammer test results for a comparison to verify the model is truly and efficiently working.

Table 4.4: Free modes comparison for spruce and plane tree.

FREE MODES COMPARISON	SPRUCE	PLANE TREE
MODE NO	FREQUENCY (Hz) - FREE MODE	FREQUENCY (Hz) – FREE MODE
1	**20,31**	**23,44**
2	**43,88**	**55,23**
3	**54,94**	**62,33**
4	**87,50**	**10,04**
5	**106,18**	**106,18**
6	**144,57**	**120,07**
7	**171,46**	**142,03**
8	**178,29**	**166,73**
9	**214,72**	**191,44**
10	**251,26**	**239,76**

Starting from the lowest border of the human audible frequency range, first ten of the natural frequency harmonic series is listed for Spruce and Plane Tree.

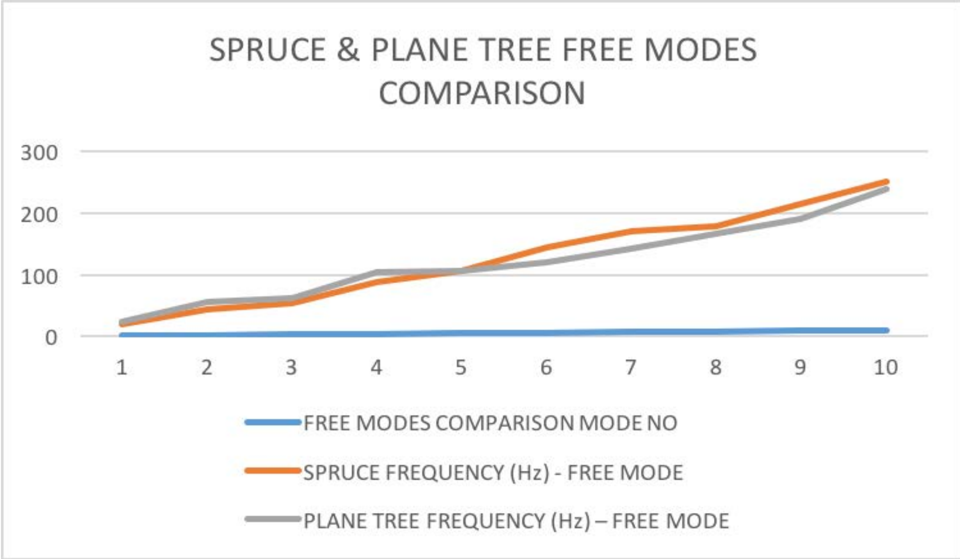

Figure 4.10: Spruce and plane tree free modes comparison.

x axis indicates harmonic series number while y axis presents the frequency in Hz.

First ten modes are graphically shown with Fig.4.10 in which one may notice that the below 100 Hz spruce and plane tree arguably behaves so similar. After that point the differences may become more clear to observe.

4.4 Free Mode Comparison of Analysis and Experimental Results

Free mode experiment and modeling results for spruce are given as follows:

Table 4.5: Free modes comparison of spruce.

FREE MODE	STUDY	1	2	3	4
SPRUCE	EXPERIMENT	20,31	50	95,31	142,19
SPRUCE	MODELING	20,31	54,94	87,50	144,57

Free mode experiment and modeling results for plane tree are given as follows:

Table 4.6: Free modes comparison of plane tree.

FREE MODE	STUDY	1	2	3	4
PLANE TREE	EXPERIMENT	23,44	50	100	125
PLANE TREE	MODELING	23,44	55,23	104,04	120,07

Experimental study and the model results differences are relatively small. The average difference between them can be said mostly around 5Hz. So it is assumed that the results of the computer model are verified with the free modes experimental analysis. it is possible to work with the model for the fixed modes of vibration.

Although the parameters that effect the model are checked, the plates are attached and fixed to the main carcass of the instrument. So, the next study will be to modify the computer model for this extra boundary conditions and calculate the natural frequencies one more time for the fixed boundary conditions.

4.5 Fixed Modes of Vibration Modal Analysis

Spruce fixed modes of vibration are given as follows:

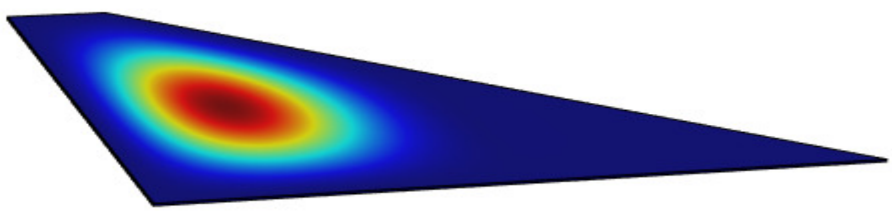

Figure 4.11: Spruce fixed mode first harmonic frequency view – 186,64 Hz.

Above and below the modal shape of the natural frequencies can be seen; displacement from the equilibrium position is shown through colors. Starting from the low values from deep blue, blue, green, yellow and finally the highest values to the red.

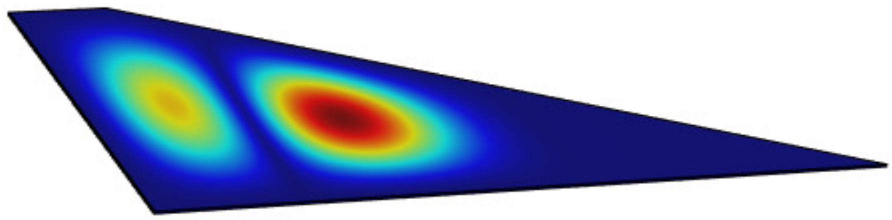

Figure 4.12: Spruce fixed mode second harmonic frequency view – 268,21 Hz.

This harmonic series can be seen individually like Fig 4.11 and 4.12 and so forth. The natural frequencies are listed below with the hammer test results for a comparison to verify the model is truly and efficiently working.

Plane tree fixed modes of vibration are given as follows:

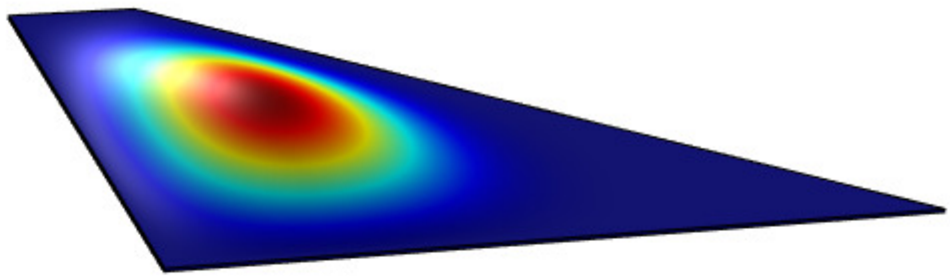

Figure 4.13: Plane tree fixed mode first harmonic frequency view – 155,6 Hz.

Above and below the modal shape of the natural frequencies can be seen; displacement from the equilibrium position is shown through colors. Starting from the low values from deep blue, blue, green, yellow and finally the highest values to the red.

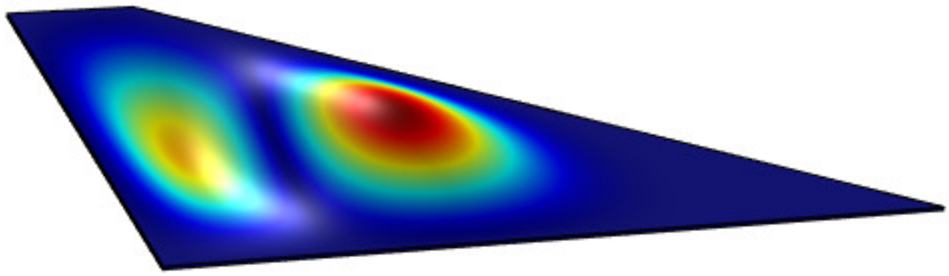

Figure 4.14: Plane tree fixed mode second harmonic frequency view – 233,42Hz.

As it is performed before, this harmonic series can be seen individually like Fig 4.13 and 4.14 and so forth. The natural frequencies are listed below with the hammer test results for a comparison to verify the model is truly and efficiently working.

4.6 Fixed Modes Natural Frequencies and Musical Pitch Relation

Table 4.7: Kanun's pitch and frequency values for equal temperament tuning.

String Group No	PITCH	FREQUENCY (Hz) (EQUAL TEMPERAMENT)	PITCH (TURKISH MUSIC)
1	B2	123,5	E - Kaba Hüseyni Aşiran
2	C3	130,8	F - Kaba Acem Aşiran
3	D3	146,8	G - Kaba Rast
4	E3	164,8	A - Kaba Dugah
5	F#3	185	B - Kaba Buselik
6	G3	196	C - Kaba Çargah
7	A3	220	D - YEGAH
8	B3	247	E - Hüseyni Aşiran
9	C4	261,6	F - Acem Aşiran
10	D4	293,7	G - Rast
11	E4	329,6	A - Dugah
12	F#4	370	B - Buselik
13	G4	392	C - Çargah
14	A4	440	D - Neva
15	B4	493,9	E - Hüseyni
16	C5	523,3	F - Acem
17	D5	587,3	G - Gerdaniye
18	E5	659,3	A - Muhayyer
19	F5	698,5	B - Tiz Buselik
20	G5	784	C - Tiz Çargah
21	A5	880	D - Tiz Neva
22	B5	987,7	E - Tiz Hüseyni
23	C6	1047	F - Tiz Acem
24	D6	1175	G - Tiz Gerdaniye

Above the frequencies and their corresponding pitches in Turkish music (with related Capitals) are listed for Kanun when the instrument is tuned in equal temperament.

Table 4.8: Spruce fixed modes versus pitch.

FIXED MODES COMPARISON	SPRUCE	
MODE NO	FREQUENCY (Hz)	PITCH&OCTAVE – (FREQUENCY)
1	**186,64**	**F# 3 (185 Hz)**
2	**268,21**	**C 4 (261,6 Hz)**
3	**358,95**	**F 4 (349,2 Hz)**
4	**431,38**	**A 4 (440 Hz)**
5	**460,94**	**A# 4 (466,2 Hz)**
6	**554,52**	**C# 5 (554,4 Hz)**
7	**577,78**	**D 5 (587,3 Hz)**
8	**677,86**	**F 5 (698,5 Hz)**
9	**708,62**	**F 5 (698,5 Hz)**
10	**780,97**	**G 5 (784 Hz)**
11	**807,13**	**G# 5 (830,6 Hz)**
12	**848,69**	**A 5 (880 Hz)**
13	**942,59**	**A# 5 (932,3 Hz)**
14	**955,81**	**A# 5 (932,3 Hz)**
15	**999,07**	**B 5 (987,7 Hz)**
16	**1089,3**	**C# 6 (1109 Hz)**
17	**1124,2**	**C# 6 (1109 Hz)**
18	**1162,5**	**D 6 (1175 Hz)**
19	**1230,3**	

Nine different musical pitches such as F# / C / F / A / A# / C# / D / G / G# are naturally produced on spruce top plate in the frequency range of the instrument; due to boundary fixed conditions as well as material properties.

Table 4.9: Plane tree fixed modes versus pitch.

FIXED MODES COMPARISON	PLANE TREE	
MODE NO	FREQUENCY (Hz)	PITCH&OCTAVE – (FREQUENCY)
1	**155,6**	**D# 3 (155,6 Hz)**
2	**233,42**	**A# 3 (233,1 Hz)**
3	**318,69**	**D# 4 (311,1 Hz)**
4	**352,44**	**F 4 (349,2 Hz)**
5	**413,46**	**G# 4 (415,3 Hz)**
6	**516,9**	**C 5 (523,3 Hz)**
7	**594,19**	**D 5 (587,3 Hz)**
8	**623,29**	**D# 5 (622,3 Hz)**
9	**631,73**	**D# 5 (622,3 Hz**
10	**715,26**	**F 5 (698,5 Hz)**
11	**749,88**	**F # 5 (740 Hz)**
12	**789,67**	**G 5 (784 Hz)**
13	**839,15**	**G# 5 (830,6 Hz)**
14	**878,38**	**A 5 (880 Hz)**
15	**944,24**	**A#5 (932,3 Hz)**
16	**967,33**	**B 5 (987,7 Hz)**
17	**982,08**	**B 5 (987,7 Hz)**
18	**1013,8**	**B 5 (987,7 Hz)**
19	**1092,9**	**C# 6 (1109 Hz)**
20	**1109,4**	**C# 6 (1109 Hz)**
21	**1152,4**	**D 6 (1175 Hz)**
22	**1173,4**	**D 6 (1175 Hz)**
23	**1230,4**	

Twelve different musical pitches such as D# / A# / F / G# / C / D / F# / G / A / B / C# / D are naturally produced on plane tree top plate in the frequency range of the instrument; due to boundary fixed conditions as well as material properties.

Table 4.10: Spruce and plane tree fixed modes comparison.

FIXED MODES COMPARISON	PLANE TREE	SPRUCE
MODE NO	FREQUENCY (Hz)	FREQUENCY (Hz)
1	**155,6**	**186,64**
2	**233,42**	**268,21**
3	**318,69**	**358,95**
4	**352,44**	**431,38**
5	**413,46**	**460,94**
6	**516,9**	**554,52**
7	**594,19**	**577,78**
8	**623,29**	**677,86**
9	**631,73**	**708,62**
10	**715,26**	**780,97**
11	**749,88**	**807,13**
12	**789,67**	**848,69**
13	**839,15**	**942,59**
14	**878,38**	**955,81**
15	**944,24**	**999,07**
16	**967,33**	**108,.3**
17	**982,08**	**1124,2**
18	**1013,8**	**1162,5**
19	**1092,9**	**1230,3**
20	**1109,4**	
21	**1152,4**	
22	**1173,4**	
23	**1230,4**	

Twenty-two individual harmonics for plane tree versus eighteen individual harmonics for spruce. The last highest pitches do not taken into account because the frequency range of the instrument is ended up with 1175 Hz. Each of the woods start to resonate in the third musical octave range. Additionally, the range of the plane tree is wider than the spruce. Although they both end up around 1230 Hz, plane tree starts to resonate a minor third musical interval lower than the spruce.

4.7 Alternative Material Studies via Computer Modeling

In this section, different kinds of materials for the top plate will be studied as an alternative for woods. Metal and composite materials such as glass fiber reinforced plastic (GFRP) and carbon fiber reinforced plastic (CFRP) will be analyzed via computer modeling. Finally, the results will be compared in terms of top plate design parameters.

4.7.1 Using metal as a top plate material

Aluminum 3003-H18 is used for this process. Plate dimensions are given as follows:

Length bottom: 74 cm, Length top: 12 cm, Width: 43,5 cm, Height: 0,2489 cm

Material property values used in this modeling process are given as follows:

Density: 2730 kg/m^3 , E: 69 GPa , μ: 0,33

All of the material property values are taken from Comsol software material library data.

Table 4.11: Aluminum 3003-H18 top plate fixed modes of vibration.

FIXED MODES	ALUMINUM 3003-H18
MODE NO	FREQUENCY (Hz)
1	**155,67**
2	**254,96**
3	**330,65**
4	**381,9**
5	**470,6**
6	**533,24**
7	**577,77**
8	**638,61**
9	**711,69**
10	**763,44**
11	**829,5**
12	**896,53**
13	**915,58**
14	**973,82**
15	**1052,2**
16	**1136,3**
17	**1145,2**

For the given material properties above, Aluminum 3003-H18 pronounces natural frequencies from 155,67 Hz to 1145,2 Hz.

4.7.2 Using GFRP as a top plate material

Toray GFRP is used for this process. Plate dimensions are given as follows:

Length bottom: 74 cm, Length top: 12 cm, Width: 43,5 cm, Height: 0,3883 cm

Material property values are taken from Esacomp software material library data.

Density $_{\text{GFRP TORAY}}$: 1835,88 kg/m^3

Table 4.12: Young modulus values for Toray GRFP.

Young Modulus (GPa)	TORAY GFRP
E_X	24,40
E_Y	24,40
E_Z	10

Table 4.13: Poisson's ratio values for Toray GFRP.

Poisson's Ratios	TORAY GFRP
μ_{XY}	0,14
μ_{XZ}	0,35
μ_{YZ}	0,35

Table 4.14: Shear Modulus values for Toray GFRP.

Shear Modulus (GPa)	TORAY GFRP
G_{XY}	4,368
G_{YZ}	4
G_{XZ}	4

Table 4.15: Toray GFRP fixed modes of vibration.

FIXED MODES	TORAY GFRP
MODE NO	FREQUENCY (Hz)
1	**155,6**
2	**256,59**
3	**324,47**
4	**385,78**
5	**465,2**
6	**538,96**
7	**561,31**
8	**633,97**
9	**718,09**
10	**746,12**
11	**827,39**
12	**865,21**
13	**920,57**
14	**952**
15	**1049,5**
16	**1100,3**
17	**1148,7**

For the given material properties above, GFRP Toray pronounces natural frequencies from 155,6 Hz to 1148,7 Hz.

4.7.3 Using CFRP as a top plate material

Toray CFRP is used for this process. Plate dimensions are given as follows:

Length bottom: 74 cm, Length top: 12 cm, Width: 43,5 cm, Height: 0,3223 cm

Material property values are taken from Esacomp software material library data.

Density $_{\text{CFRP TORAY}}$: 1555 kg/m^3

Table 4.16: Young modulus values for Toray CRFP.

Young Modulus (GPa)	TORAY CFRP
E_X	119
E_Y	9
E_Z	9

Table 4.17: Poisson's ratio values for Toray CFRP.

Poisson's Ratios	TORAY CFRP
μ_{XY}	0,309
μ_{XZ}	0,309
μ_{YZ}	0,350

Table 4.18: Shear Modulus values for Toray CFRP.

Shear Modulus (GPa)	TORAY CFRP
G_{XY}	4
G_{YZ}	3,3
G_{XZ}	4

Table 4.19: Toray CFRP fixed modes of vibration.

FIXED MODES	TORAY CFRP
MODE NO	FREQUENCY (Hz)
1	**155,6**
2	**260,52**
3	**319,13**
4	**396,4**
5	**461,67**
6	**541,19**
7	**561,79**
8	**639,13**
9	**717,97**
10	**754,53**
11	**823,29**
12	**849,41**
13	**934,03**
14	**977,8**
15	**1032,6**
16	**1087,4**
17	**1164,2**

For the given material properties above, CFRP Toray pronounces natural frequencies from 155,6 Hz to 1164,2 Hz.

4.7.4 Material comparison in terms of top plate design parameters

Table 4.20: Top plate design parameters for selected materials.

	TOP PLATE MATERIAL TYPES				
	METAL	**WOOD**		**COMPOSITE**	
Properties	**Aluminum 3003-H18**	**Plane**	**Spruce**	**GFRP Toray**	**CFRP Toray**
Density (kg/m3)	2730	365	355,6	1835,88	1555
Young Modulus E (GPa)	69				
Poisson's ratio	0,33				
E_x		2,5002	2,100	24,404250000	119
E_y		21	12,430	24,404250000	9
E_z		1,400	0,8308	10	9
μ_{xy}		0,030	0,030	0,14	0,309
μ_{xz}		0,467	0,604	0,35	0,309
μ_{yz}		0,530	0,641	0,35	0,350
Shear Modulus G (GPa)					
G_{xy}		1653,2	850	4,368	4
G_{yz}			726	4	3,3
G_{xz}		12,43	35	4	4
Plate thickness (mm)	**2,489**	**3,200**	**3,200**	**3,883**	**3,223**
Number of harmonics	**17**	**22**	**18**	**17**	**17**
Fundamental Frequency (Hz)	155,67	155,60	186,64	155,60	155,60
Mass (kg)	1,27	0,218	0,214	1,33	0,937

Top plate design parameters for selected materials are listed above. Plane tree fundamental frequency is taken as a reference between them to easily compare the plate thickness. The number of harmonics within the frequency range of the instrument is also presented.

4.8 Top Plate Design Alternatives

Aluminum 3003-H18 and GFRP Toray will be analyzed at this stage to present their choice would be an alternative for traditional use of wood in manufacturing.

4.8.1 Tuning the top plate according to instrument's frequency range

Until this stage, the fundamental frequency is studied around 155,6 Hz. This is the point where the plane tree introduces 22 number of modes within the frequency range of the instrument as a reference for comparison (See Table 4.20).

4.8.2 Tuning Aluminum 3003-H18 top plate

Table 4.21: Aluminum 3003-H18 top plate tuned to lowest pitch of Kanun.

FIXED MODES	ALUMINUM 3003-H18
MODE NO	FREQUENCY (Hz)
1	**123,5**
2	**202,4**
3	**262,68**
4	**303,39**
5	**374,24**
6	**423,91**
7	**459,51**
8	**508,2**
9	**566,31**
10	**608,24**
11	**660,54**
12	**714,18**
13	**729,56**
14	**776,87**
15	**838,71**
16	**906,79**
17	**913,76**
18	**957,34**
19	**1024,7**
20	**1041,1**
21	**1113**
22	**1120,4**
23	**1171,9**

This stage presents the tuning of the fundamental frequency of the top plate to the instrument lowest musical pitch B2. This frequency is 123,5 Hz. The tuning can be achieved by changing the thickness of the plate. All the other material property values follow the conditions according to Table 4.20.

According to material properties of Table 4.20, Aluminum 3003-H18 produces more harmonics than the plane tree, starts from the instrument's lowest pitch equivalent frequency which is 123,5 Hz. The plate thickness has to be 1,9722 mm.

4.8.3 Tuning GFRP Toray top plate

Table 4.22: GFRP Toray top plate tuned to lowest pitch of Kanun.

FIXED MODES	GFRP TORAY
MODE NO	FREQUENCY (Hz)
1	**123,5**
2	**202,4**
3	**262,68**
4	**303,39**
5	**374,24**
6	**423,91**
7	**459,51**
8	**508,2**
9	**566,31**
10	**608,24**
11	**660,54**
12	**714,18**
13	**729,56**
14	**776,87**
15	**838,71**
16	**906,79**
17	**913,76**
18	**957,34**
19	**1024,7**
20	**1041,1**
21	**1113**
22	**1120,4**
23	**1156,9**

According to material properties of Table 4.20, GFRP Toray produces more harmonics than the plane tree, starts from the instrument's lowest pitch equivalent frequency which is 123,5 Hz. The plate thickness has to be 3,079 mm.

4.9 Form and Geometry Practice via Computer Modeling

In this section, the plates are drilled via computer modeling. The effect of a single hole and a pair of three holes within the plate for selected materials will be analyzed and compared. Single hole drill is with a radius of 7 cm. Alternatively, a radius of 4 cm of three identical drills are studied comparatively.

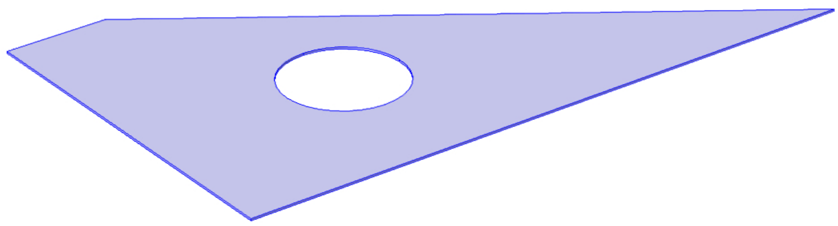

Figure 4.15: Top plate model view with a single hole.

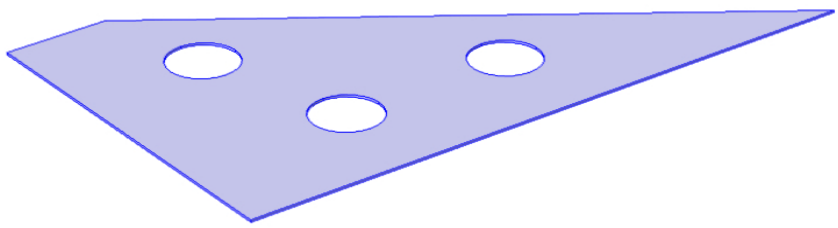

Figure 4.16: Top plate model view with three holes.

4.9.1 Aluminum 3003-H18 top plate drilled center

Table 4.23: Aluminum 3003-H18 top plate modes when drilled with a single hole.

FIXED MODES	ALUMINUM 3003-H18
MODE NO	FREQUENCY (Hz)
1	**123,5**
2	**180,42**
3	**226,73**
4	**305,03**
5	**338,61**
6	**371,4**
7	**407,88**
8	**474,19**
9	**509,04**
10	**570,01**
11	**600,02**
12	**647,17**
13	**697,35**
14	**743,01**
15	**782,54**
16	**824,41**
17	**846,34**
18	**916,38**
19	**956,01**
20	**1004**
21	**1042,3**
22	**1079,1**
23	**1091,9**

According to material properties of Table 4.20, Aluminum 3003-H18 produces 23 harmonics when a single hole is drilled. The plate thickness has to be 1,7638 mm.

4.9.2 Aluminum 3003-H18 drilled with three holes

Table 4.24: Aluminum 3003-H18 top plate modes with three holes.

FIXED MODES	ALUMINUM 3003-H18
MODE NO	FREQUENCY (Hz)
1	**123,5**
2	**199,45**
3	**269,34**
4	**304,41**
5	**377,49**
6	**424,35**
7	**444,49**
8	**489,69**
9	**548,12**
10	**580,38**
11	**656,93**
12	**689,98**
13	**740,31**
14	**798,24**
15	**814,85**
16	**862,52**
17	**911,89**
18	**940,21**
19	**1009,4**
20	**1049,5**
21	**1071,5**
22	**1122,5**

According to material properties of Table 4.20, Aluminum 3003-H18 produces 22 harmonics with three drills on the top plate. The plate thickness has to be 1,9356 mm.

4.9.3 GFRP Toray drilled center

Table 4.25: GFRP Toray top plate modes with a single hole.

FIXED MODES	GFRP Toray
MODE NO	FREQUENCY (Hz)
1	**123,5**
2	**181,04**
3	**222,28**
4	**300,49**
5	**337,68**
6	**371,87**
7	**397,57**
8	**462,73**
9	**502,86**
10	**562,98**
11	**590,1**
12	**630,16**
13	**674,65**
14	**726,74**
15	**773,88**
16	**785,32**
17	**844,41**
18	**889,99**
19	**928,11**
20	**969,5**
21	**1014,8**
22	**1040,3**
23	**1043,2**
24	**1125,6**
25	**1162,1**

According to material properties of Table 4.20, GFRP Toray produces 25 harmonics when a single hole is drilled. The plate thickness has to be 2,7942 mm.

4.9.4 GFRP Toray drilled with three holes

Table 4.26: GFRP Toray top plate modes with three holes.

FIXED MODES	GFRP Toray
MODE NO	FREQUENCY (Hz)
1	**123,5**
2	**200,57**
3	**263,1**
4	**308,36**
5	**366,2**
6	**426,54**
7	**431,96**
8	**482,75**
9	**536,83**
10	**564,41**
11	**632,65**
12	**661,09**
13	**729,98**
14	**763,02**
15	**801,88**
16	**840,78**
17	**887,57**
18	**904,93**
19	**970,89**
20	**1002,7**
21	**1048,2**
22	**1082,8**
23	**1150,1**

According to material properties of Table 4.20, GFRP Toray produces 23 harmonics with three drills on the top plate. The plate thickness has to be 3,0528 mm.

4.10 Top Plate Material Comparison

The results about the material choices, plate thickness and number of natural frequency harmonics within the frequency range of the instrument have been found and discussed so far are given as a table below for an easy comparison. The lowest musical pitch of the Kanun is equivalent to 123,5 Hz and the highest pitch of the instrument is equivalent to 1175 Hz which are already given in Table 4.7.

Table 4.27: Top plate material design parameters comparison.

	Lowest Natural Frequency (Hz)	**Highest Natural Frequency (Hz)**	**Plate Thickness (mm)**	**Number of harmonics within instrument's frequency range**
Plane Tree	155,6	1173,4	**3,2**	**22**
Spruce	186,64	1162,5	**3,2**	**18**
Aluminum 3003-H18	123,5	1171,9	**1,9722**	**23**
Aluminum 3003-H18 Single Hole	123,5	1091,9	**1,7638**	**23**
Aluminum 3003-H18 Three Holes	123,5	1122,5	**1,9356**	**22**
GFRP Toray	123,5	1156,9	**3,079**	**23**
GFRP Toray Single Hole	123,5	1162,1	**2,7942**	**25**
GFRP Toray Three Holes	123,5	1150,1	**3,0528**	**23**

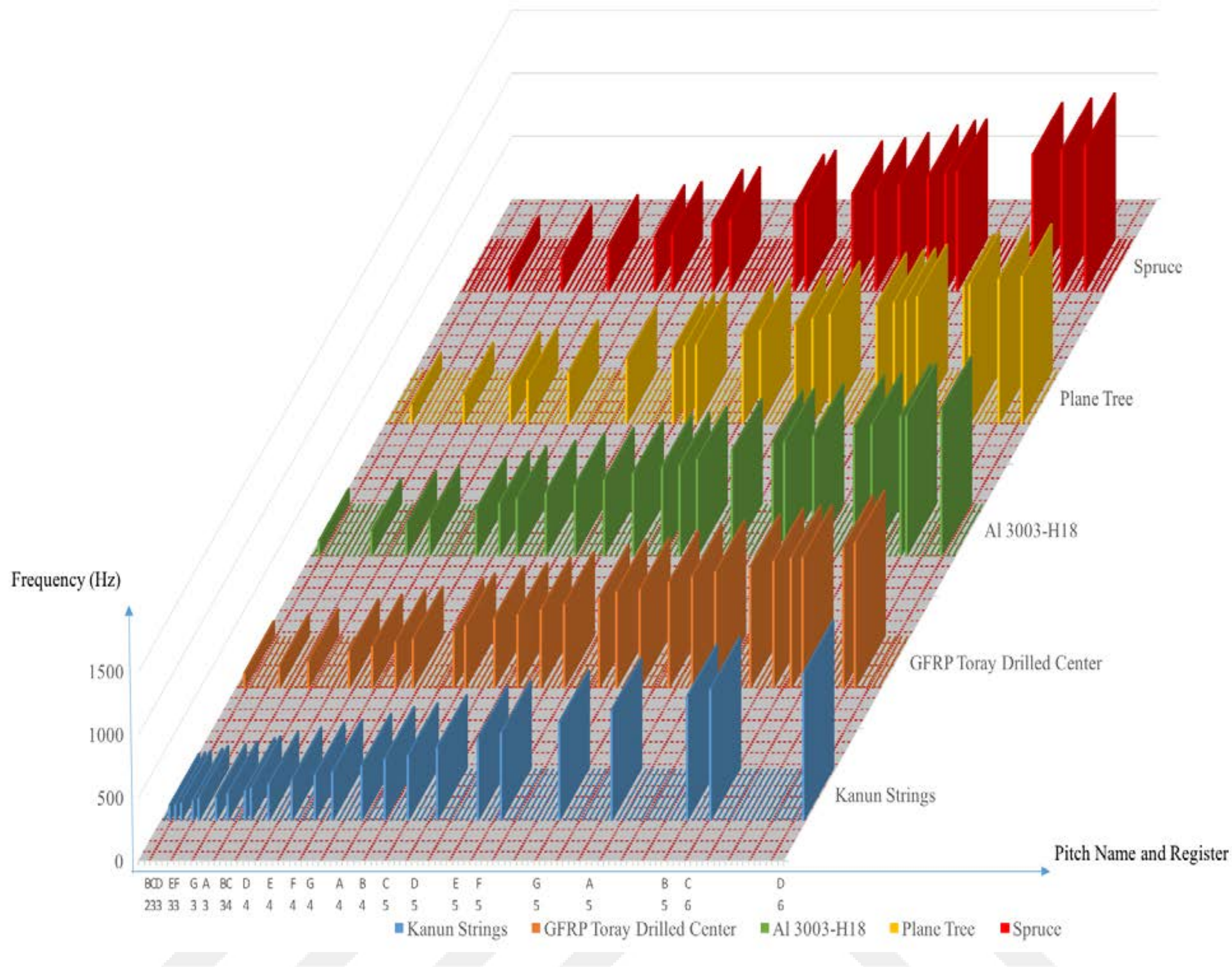

Figure 4.17: Visual representation of material comparison.

A visual material comparison of the top plates is presented above in Figure 4.17. Blue color displays the Kanun strings and the entire frequency range of the study.

Introducing alternatives for woods, brown color is indicated as GFRP Toray which produces 25 harmonics. Green color presents Al 3003-H18 which produces 23 harmonics through the entire range.

Yellow color presents plane tree and the red color presents spruce natural frequencies through the entire range of the instrument.

5. CONCLUSION

A computerized physical model can be used effectively in designing traditional acoustic instruments along with manufacturing process. Although only one parameter (top plate) of a complex structure is modelled and analyzed in the study, it is possible to improve the model by adding other acoustically related parameters.

Material of the top plate which effects the playing range of the music instrument can be selected with the help of the physical computer modeling.

It is possible to integrate acoustical aspects of the instrument and the analysis results of the instrument with manufacturing design alternatives which were formed traditionally and intuitively.

Timbre of the instrument is related with the frequency and the amplitude interactively as well as the envelope of the sound source. Therefore, Kanun's playing style by using two hands may create spectral differences depending on the string impact locations and their ratios due to constant string lengths.

Even one cm change in any dimension of the geometry effects the natural frequency results of the top plates. Therefore, the sound's harmonic content and the radiation intensity in terms of frequency and the corresponding amplitudes might change in the perceived sound field. Dimensions determine the wavelengths. Therefore, the natural frequency changes in accordance with the change in dimensions as the sound velocity keeps constant.

Depending on the sympathetic vibration of the woods, the top plates determine a dominant role in spreading the energy of the sound throughout the ambiance. Sound propagation is supported effectively by the natural frequencies of the top plate.

After handling several top plates during the manufacturing process, although all plates had the same geometry and identical dimensions, the weights were variable in practice. In brief; the density and the relative humidity of the wood have a strong effect on the natural frequency. Along with the density, the sound velocity is related with the Young Modulus in solids. So, when the density parameter is decreased only, the natural frequencies tend to go higher inversely.

Plane and spruce woods resonate around different parts of the frequency spectrum. In this case, again the geometries are almost identical. In addition to density, Young Modulus, Shear Modulus and Poisson's ratios together determine those resonant frequency differences.

During the check process of the fixed mode results comparison of two woods concerning natural frequencies; it's seen that plane tree has much more harmonics (22) placed in the actual sounding frequency range of the instrument than the spruce (18). Depending on this fact, we can say that the plane tree has much more potential in terms of loudness and sound radiation via top plate vibrations.

Twelve different individual pitches are pronounced for plane tree as natural frequencies. This number is counted nine in spruce. The exact names and register of the pitches are given in Table 4.7 and 4.8

Alternative materials for the top plate instead of woods are also presented. To make comparison more easy; tuning the materials to the same fundamental frequency of the plane tree, metal and composite materials such as Aluminum, GFRP and CFRP are studied via computer modeling. Aluminum 3003-H18, GFRP Toray and CFRP Toray are used as alternative materials for wood in this process.

Regarding Table 4.20, Aluminum 3003-H18 with a plate thickness of 2,489 mm produces 17 harmonics within the frequency range of the instrument.

GFRP Toray with a plate thickness of 3,883 mm produces 17 harmonics within the frequency range of the instrument.

CFRP Toray with a plate thickness of 3,223 mm produces 17 harmonics within the frequency range of the instrument.

Aluminum thickness of 2,489 mm is lower than the woods which is 3,2 mm in spruce and plane tree. Inversely, the composite material alternatives need to be thicker with the given material property values. These properties are again the density, the Young Modulus, Poisson’s ratio and Shear Modulus.

Top plate design alternatives are presented in the section 4.8, which aims to tune the alternative materials through the entire range of the instrument. This could be achieved to tune the fundamental frequency of the top plates to the lowest pitch equivalent frequency of the Kanun which is 123,5 Hz.

Aluminum 3003-H18 and GFRP Toray produce 23 harmonics while these plates are tuned to 123,5 Hz with a plate thickness of 1,972 and 3,079 mm respectively.

At this stage the threshold of the plane tree which has 22 harmonics over the frequency range of the instrument is exceeded by Aluminum 3003-H18 and GFRP Toray.

A further step for the experiments on computer physical modeling is presented in section 4.9 which is labeled as form and geometry practice via computer modeling. Basically, a single centered hole and a pair of three holes around the center with a radius of 7 and 4 cm are drilled respectively via modeling.

Drilling a single hole on a plate presents more number of natural frequencies within the entire frequency range of the instrument than a pair of three holes.

Aluminum 3003-H18 produces 23 harmonics with a single hole while it pronounces 22 harmonics with a pair of three holes.

GFRP Toray produces 25 harmonics with a single hole while it pronounces 23 harmonics with a pair of three holes.

Drilling o single centered hole gives the maximum number of harmonics with GFRP Toray throughout the entire modeling experiments. Another result with this single hole option comes out with a reduction of top plate thickness compared to alternatives with no drill and a pair of three holes.

As a result, all GFRP Toray or Al 3003-H18 top plate design options can produce more harmonics than the woods if the plate thickness and the fundamental frequency is tuned via computer physical modeling.

All these given material design alternatives with a metal or a composite material also present another option to reduce the plate thickness compared to woods.

Composite materials and therefore metals can be seen and used more often than before, which may behave free of direction unlike the orthotropic nature of the woods.

Formulas, algorithms and further modifications of the finite elements method model parameters can also be used as a specific data sources for digital sound synthesis subject like physical modeling.

Although only one particular top plate component of the entire instrument construction is analyzed in this study, the results might likely to be considered within the scope of narrowed research concept.

5.1 Practical Application of This Study

Hopefully, this study would stand as meaningful, useful and practical step for instrument designers and luthiers. With the help of the computer physical model, any related parameter change on top plate design could be previewed, analyzed and evaluated during the design stage.

It is presented that; designing a top plate which pronounces more harmonics with its natural frequencies through the entire range of the Kanun is possible also with a reduced thickness of the plate, by using a metal or a composite material instead of the woods.

Comparison studies presented as tables in this work would be a useful tool for instrument designers and manufacturers. Available material choices can be evaluated to contribute to design process.

In this study; the change in density, Young Modulus, Shear Modulus, Poisson's Ratio is evaluated along with dimensions, geometry and finally the form of the top plates. The effect on the natural frequencies of the top plate can be derived from the properties of the material. Additionally, any change of form and geometry can be previewed and analyzed.

5.2 Recommendations

The following step for this particular study would be; analyzing the whole Kanun body construction with the same method and the same study model. After one particular component analysis like the top plate vibration is confirmed, additionally the following topics could be physically modelled and studied within the complete system, including; ribs and bridge attached to the body, the air inside the body, Helmholtz resonators on the top plates and the skin membrane attached to the bridges.

Another aspect for the following practices of the study may be to design a top plate of a composite material with a natural frequency series engaging with the open strings of the instrument.

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APPENDICES

APPENDIX A: Hammer Test Experiment Results (Spruce)

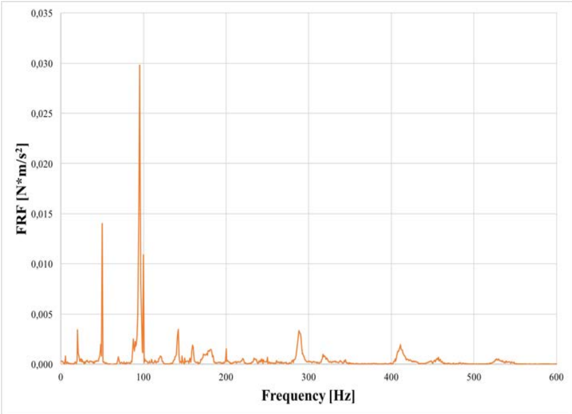
APPENDIX B: Hammer Test Experiment Results (Plane Tree)

APPENDIX C: Hammer Test Experiment Results (Spruce & Plane Tree Together)

APPENDIX D: Software Model Report for Spruce

APPENDIX E: Software Model Report for Plane tree

APPENDIX A: Hammer Test Experiment Results (Spruce).



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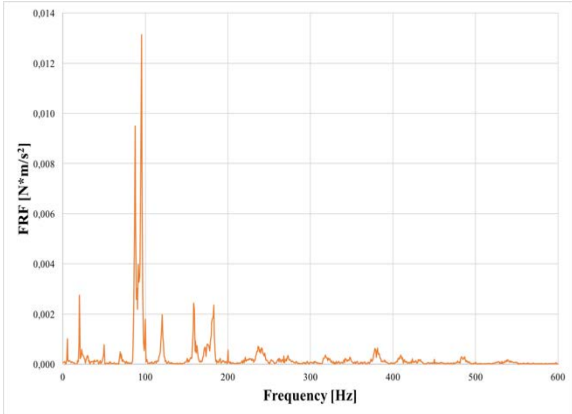


Figure A.2: Point 2.

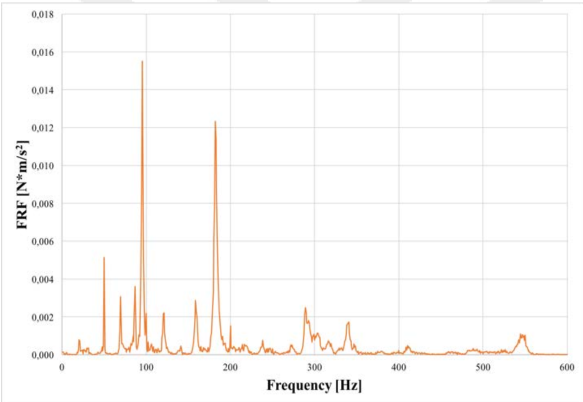


Figure A.3: Point 3.

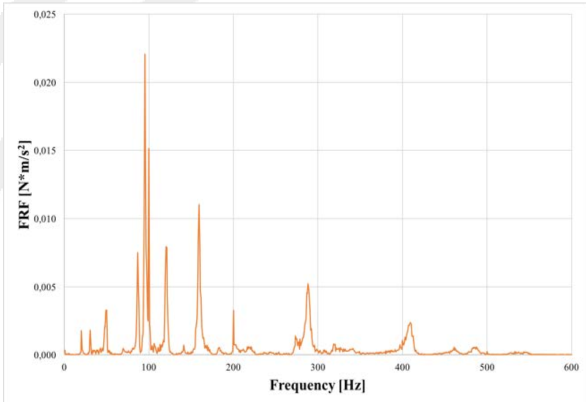


Figure A.4: Point 4.

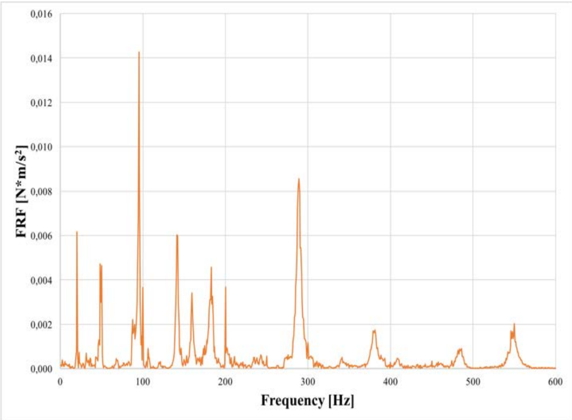


Figure A.5: Point 5.

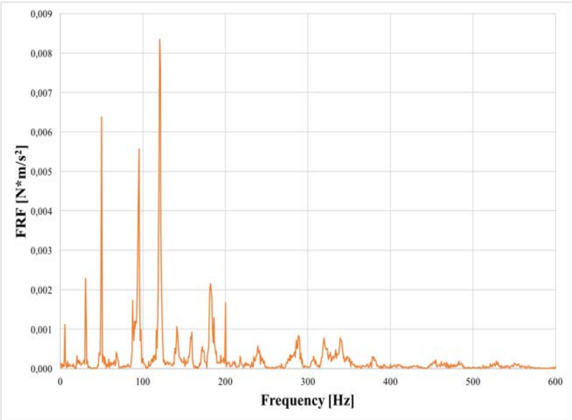
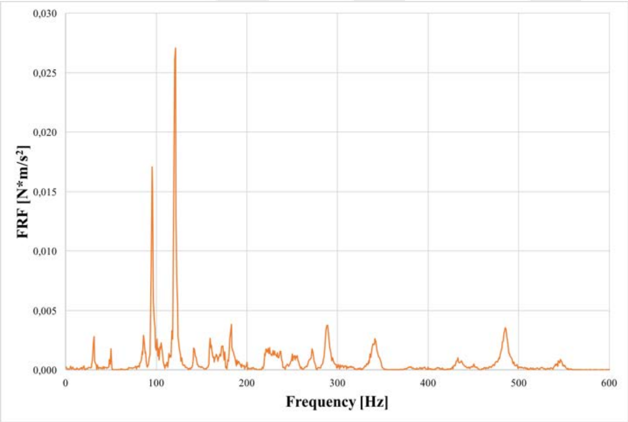


Figure A.6: Point 6.



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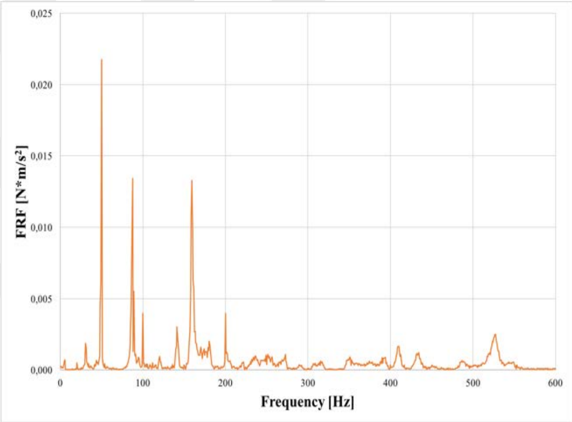


Figure A.8: Point 8.

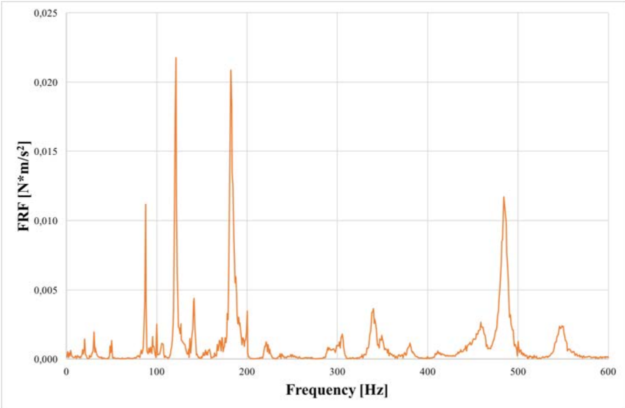


Figure A.9: Point 9.

APPENDIX B: Hammer Test Experiment Results (Plane Tree).

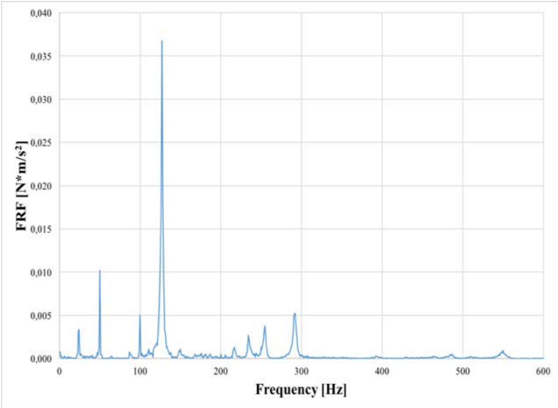
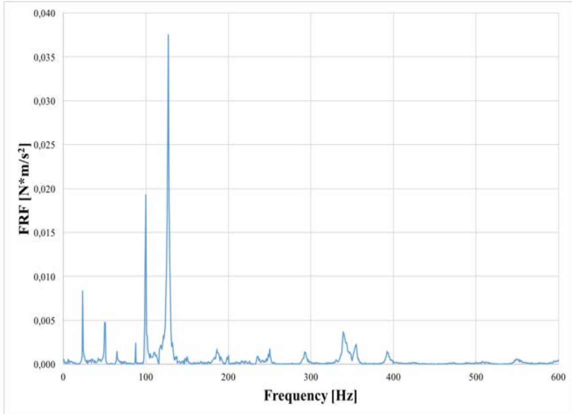


Figure B.1: Point 1.



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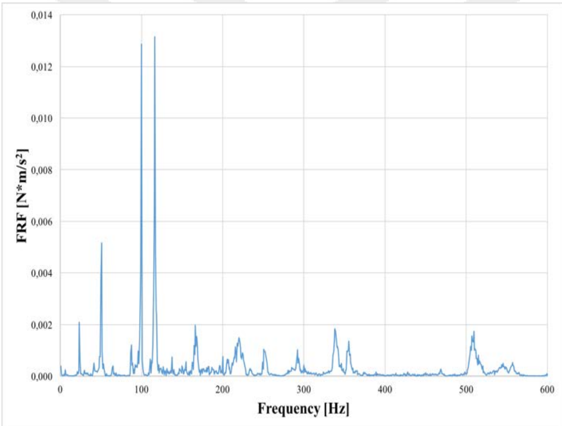


Figure B.3: Point 3.

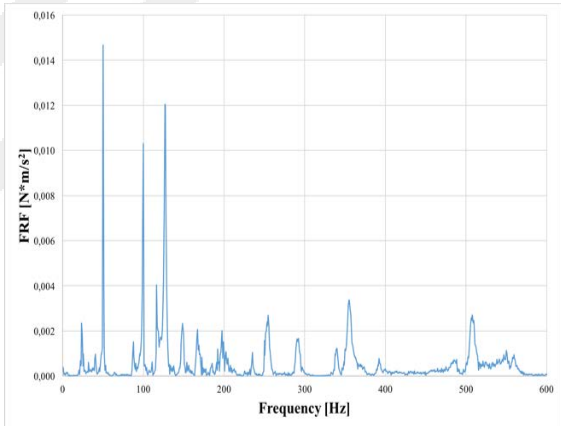
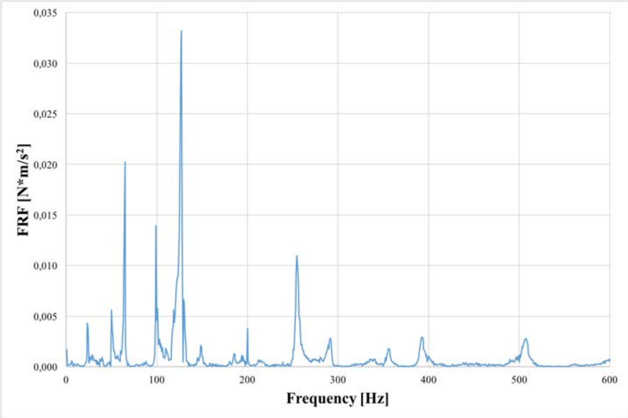


Figure B.4: Point 4.



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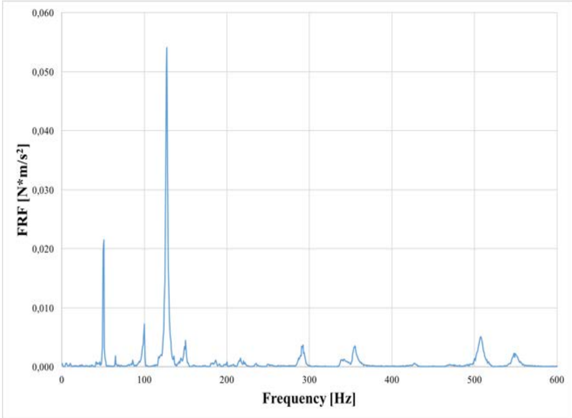


Figure B.6: Point 6.

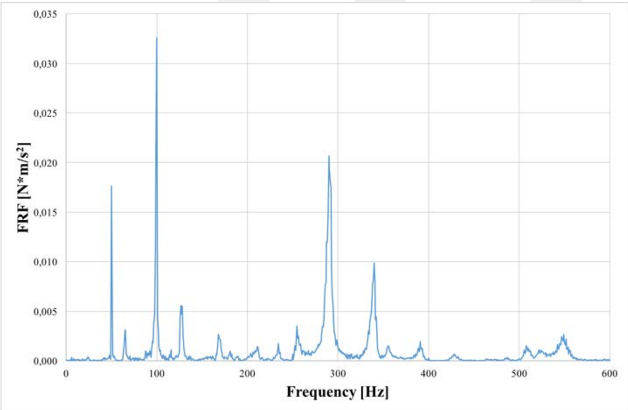


Figure B.7: Point 7.

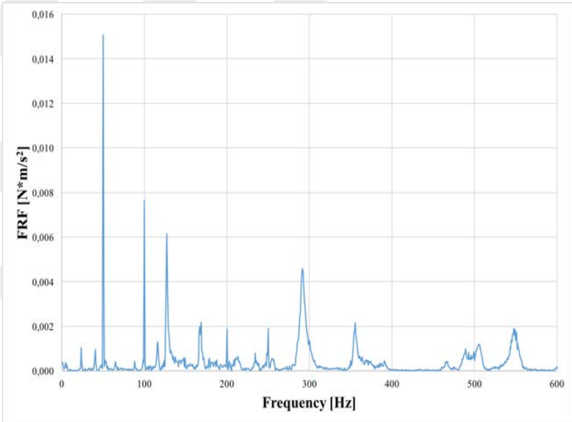


Figure B.8: Point 8.

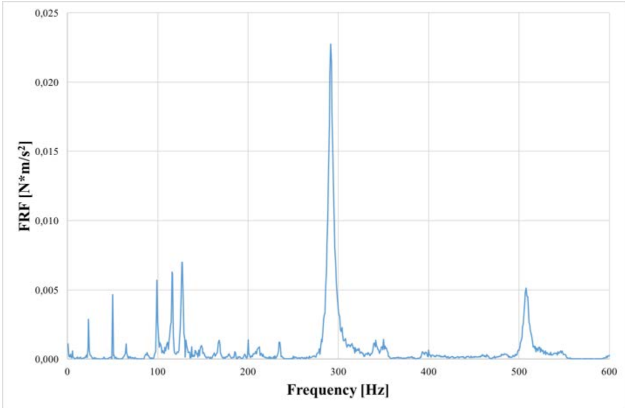


Figure B.9: Point 9.

APPENDIX C: Hammer Test Experiment Results (Spruce and Plane Tree).

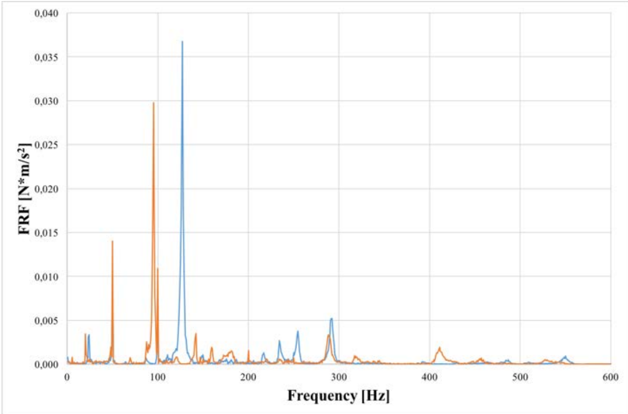


Figure C.1: Point 1.

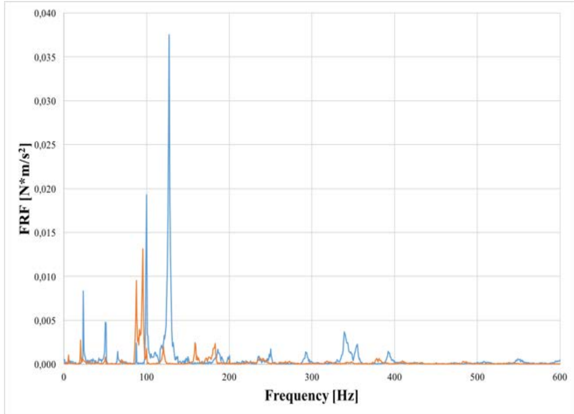


Figure C.2: Point 2.

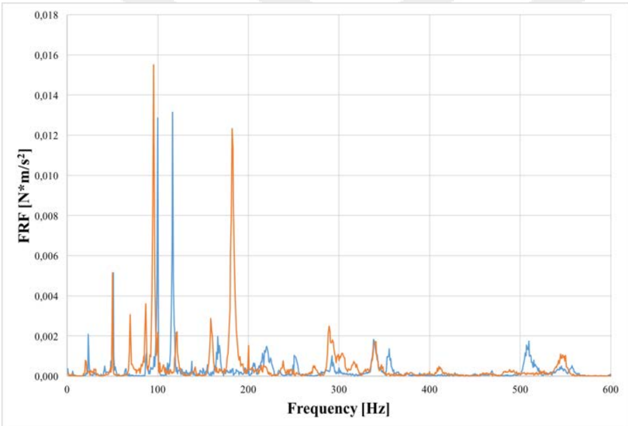
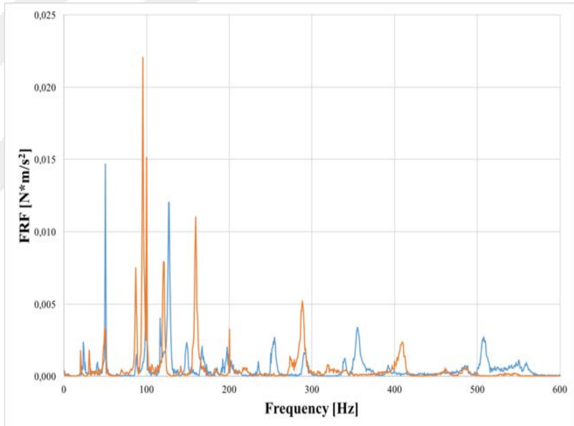


Figure C.3: Point 3.



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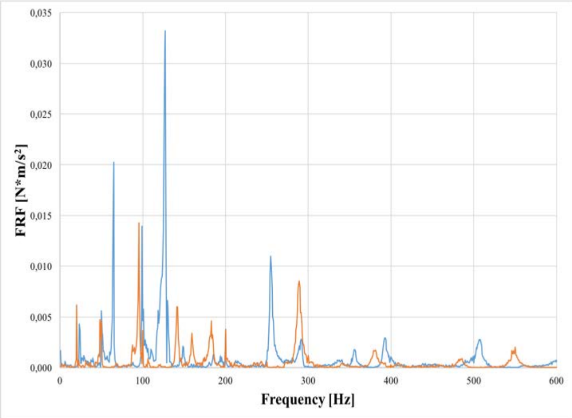

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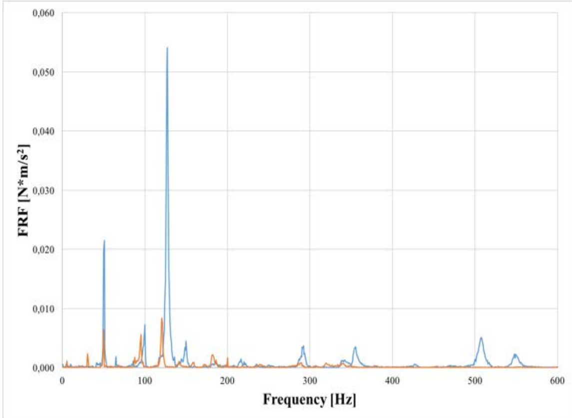

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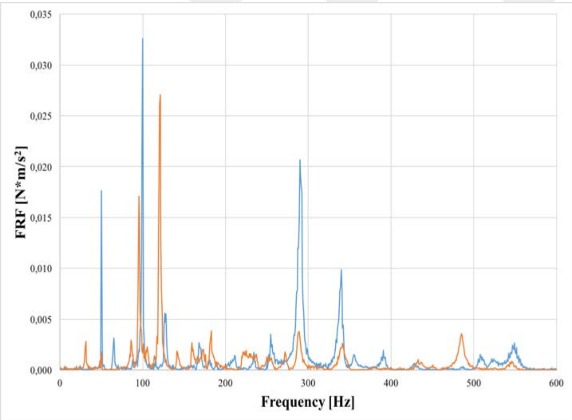

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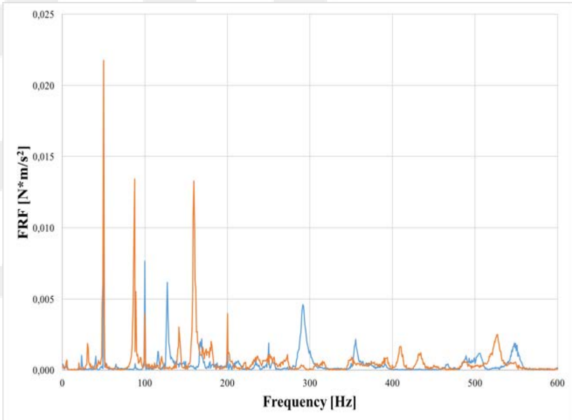


Figure C.8: Point 8.

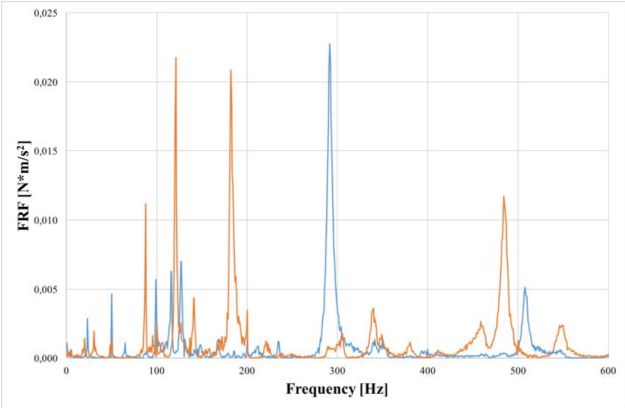


Figure C.9: Point 9.

APPENDIX D: Software Model Report for Spruce.

SPRUCE FREE MODES

Report date

Jan 23, 2020 12:49:22 AM

Contents

Global Definitions

Date	Jan 23, 2020 12:43:50 AM

Global settings

Name	SPRUCE - FREE MODES MODEL 20.mph
Path	C:\Users\SBL\Desktop\SPRUCE - FREE MODES MODEL 20.mph
Version	COMSOL Multiphysics 5.5 (Build: 306)

Used products

COMSOL Multiphysics
CAD Import Module
MEMS Module

Component 1

Settings

Description	**Value**
Avoid inverted elements by curving interior domain elements	Off

Definitions

Coordinate Systems

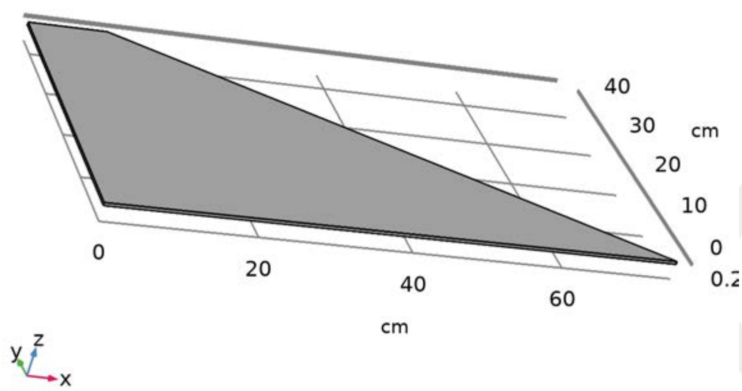
Boundary System 1

Coordinate system type	Boundary system
Tag	sys1

Coordinate names

First	Second	Third
t1	t2	n

Geometry 1



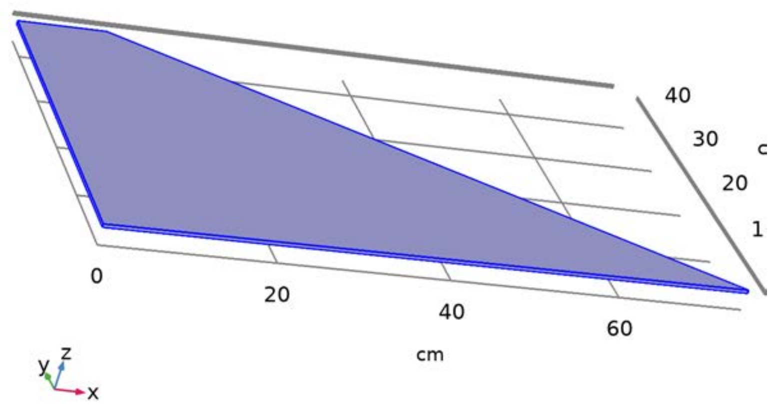
Geometry 1

Units

Length unit	cm
Angular unit	deg

Materials

Spruce [solid]



Spruce [solid]

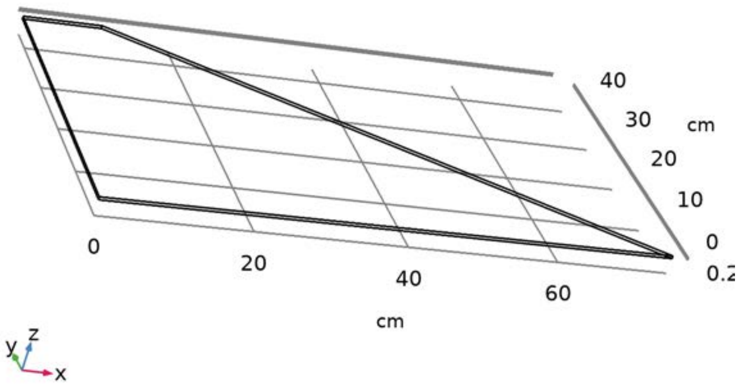
Selection

Geometric entity level	Domain
Selection	Geometry geom1: Dimension 3: All domains

Material Link 1

Link settings

Material	None

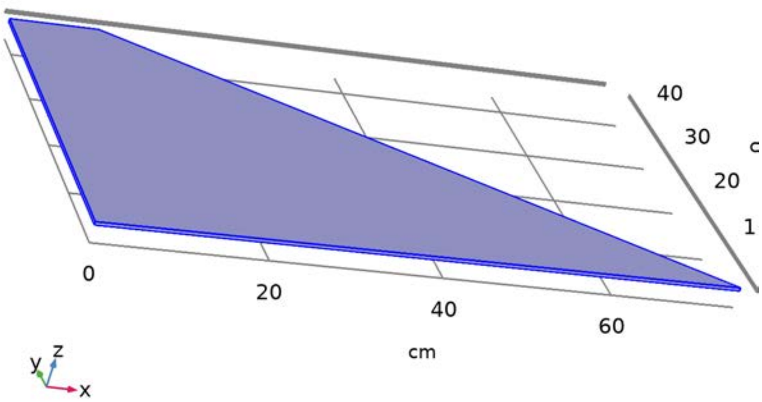


Material Link 1

Selection

Geometric entity level	Domain
Selection	Geometry geom1: Dimension 3: No domains

Solid Mechanics



Solid Mechanics

Equations

$-\rho\omega^2\mathbf{u} = \nabla \cdot S, \quad -i\omega = \lambda$

Features

Linear Elastic Material 1
Free 1
Initial Values 1

Linear Elastic Material 1

Equations

$-\rho\omega^2\mathbf{u} = \nabla \cdot S, \quad -i\omega = \lambda$

$S = S_{ad} + \mathbf{C} : \epsilon_{el}, \quad \epsilon_{el} = \epsilon - \epsilon_{inel}$

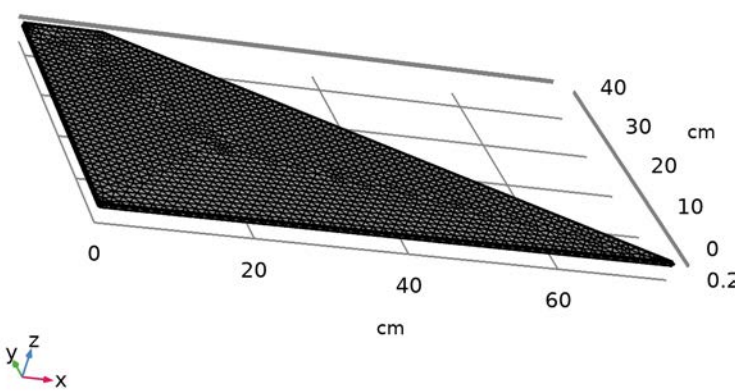
$\epsilon_{inel} = \epsilon_0 + \epsilon_{ext} + \epsilon_{th} + \epsilon_{hs} + \epsilon_{pl} + \epsilon_{cr} + \epsilon_{vp}$

$S_{ad} = S_0 + S_{ext} + S_q$

$\epsilon = \frac{1}{2}\left[(\nabla\mathbf{u})^T + \nabla\mathbf{u}\right]$

$\mathbf{C} = \mathbf{C}(\mathbf{E}, \nu, \mathbf{G})$

Mesh 1



Mesh 1

Study 1

Computation information

Computation time	22 s
CPU	AMD64 Family 21 Model 112 Stepping 0, 2 cores
Operating system	Windows 10

Eigenfrequency

Study settings

Description	Value
Include geometric nonlinearity	Off

Study settings

Description	Value
Desired number of eigenfrequencies	46
Desired number of eigenfrequencies	On

Physics and variables selection

Physics interface	Discretization
Solid Mechanics (solid)	physics

Mesh selection

Geometry	Mesh
Geometry 1 (geom1)	mesh1

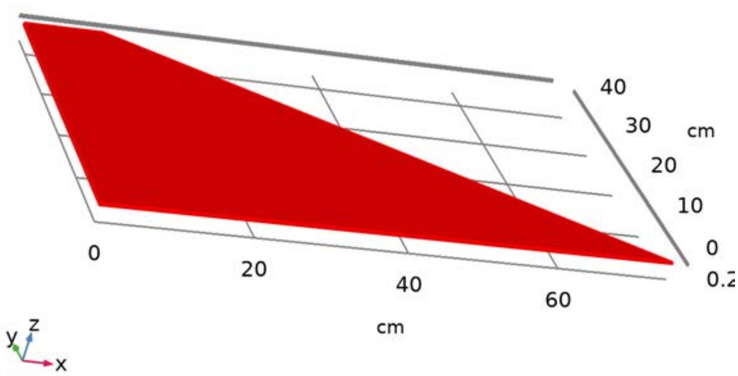
Results

Data Sets

Study 1/Solution 1

Solution

Description	Value
Solution	Solution 1
Component	Save Point Geometry 1



Dataset: Study 1/Solution 1

Tables

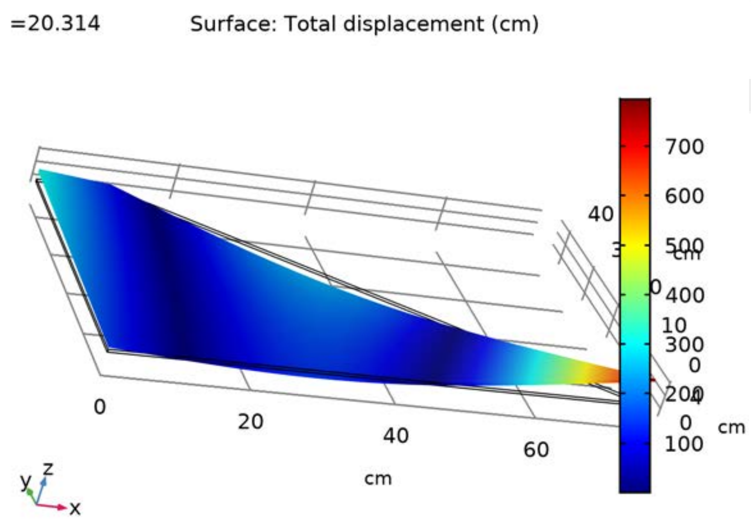
Evaluation 3D

Interactive 3D values

x	y	z	Value
9.5453	27.985	0.85660	59.080
13.982	8.9356	0.92292	124.94
9.1988	19.949	0.24501	20.316
33.661	16.156	1.1347	204.40
30.051	3.7729	1.0206	172.67
18.345	15.368	0.019694	105.84

Plot Groups

Mode Shape (solid)



Surface: Total displacement (cm)

APPENDIX E: Software Model Report for Plane tree.

COMSOL MULTIPHYSICS®

PLANE TREE FREE MODES

Report date

Jan 23, 2020 1:10:09 AM

Contents

Global Definitions

Date	Jan 12, 2020 1:32:48 AM

Global settings

Name	PLANE - FREE MODES MODEL 20.mph
Path	D:\PLANE - FREE MODES MODEL 20.mph

Version	COMSOL Multiphysics 5.5 (Build: 306)

Used products

COMSOL Multiphysics
CAD Import Module
MEMS Module

Component 1

Settings

Description	**Value**
Avoid inverted elements by curving interior domain elements	Off

Definitions

Coordinate Systems

Boundary System 1

Coordinate system type	Boundary system
Tag	sys1

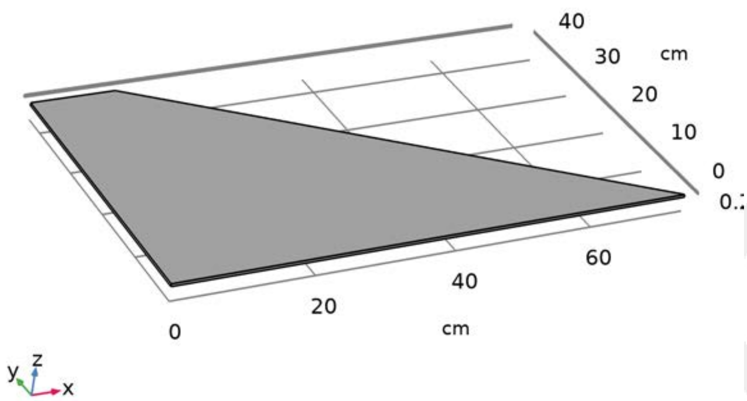
Coordinate names

First	**Second**	**Third**
t1	t2	n

Settings

Description	Value
Create first tangent direction from	Global Cartesian (material)

Geometry 1



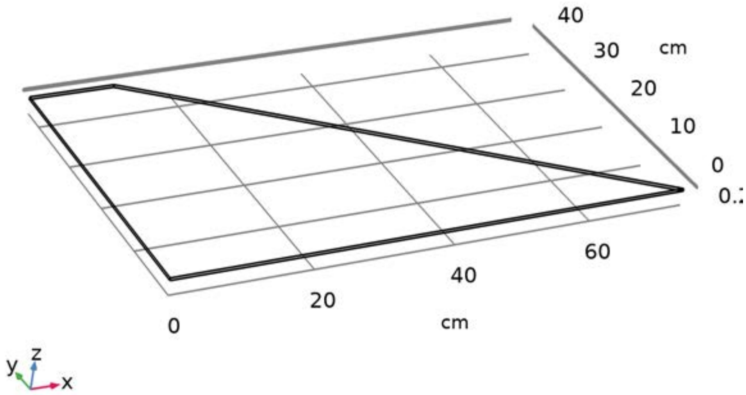
Geometry 1

Units

Length unit	cm
Angular unit	deg

Materials

Spruce [solid]



Spruce [solid]

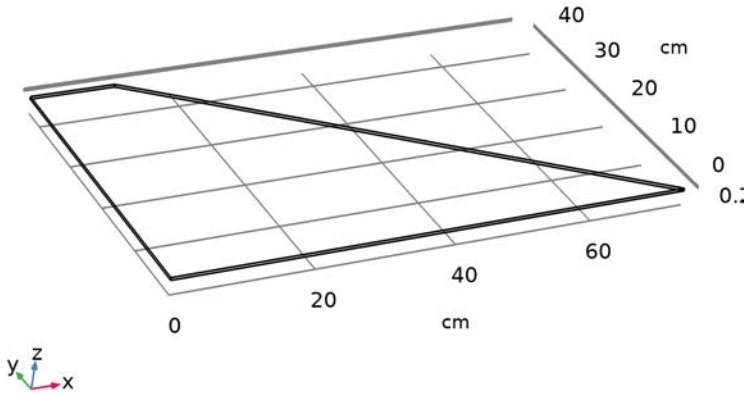
Selection

Geometric entity level	Domain
Selection	Geometry geom1: Dimension 3: All domains

Material Link 1

Link settings

Material	None



Material Link 1

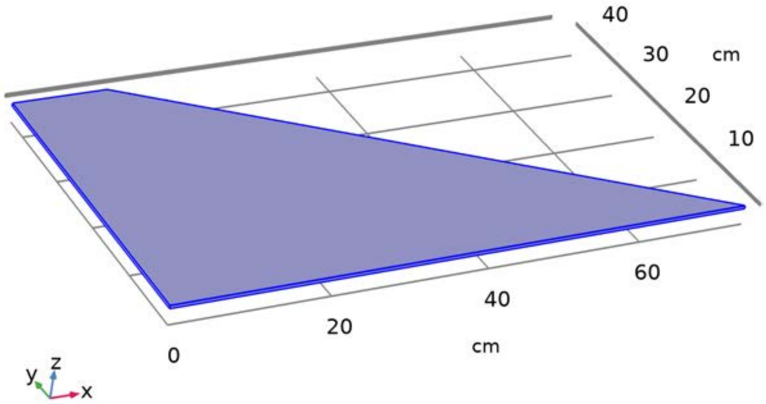
Selection

Geometric entity level	Domain
Selection	Geometry geom1: Dimension 3: All domains

Switch 1

Cases	

Solid Mechanics



Solid Mechanics

Equations

$-\rho\omega^2\mathbf{u} = \nabla\cdot(\mathrm{F\,S})^{\mathrm{T}}, \quad \mathrm{F} = \mathrm{I} + \nabla\mathbf{u}, \quad -i\omega = \lambda$

Features

Linear Elastic Material 1
Free 1
Initial Values 1

Linear Elastic Material 1

Equations

$-\rho\omega^2\mathbf{u} = \nabla\cdot(\mathrm{F\,S})^{\mathrm{T}}, \quad \mathrm{F} = \mathrm{I} + \nabla\mathbf{u}, \quad -i\omega = \lambda$

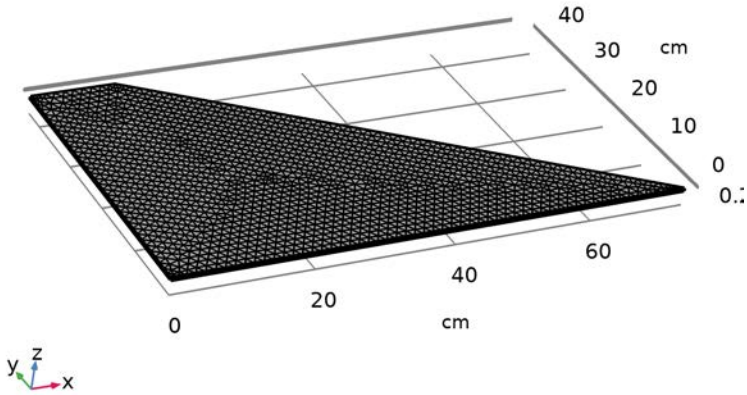
$\mathrm{S} = \mathrm{S_{ad}} + J_i \mathrm{F}_{\mathrm{inel}}^{-1}\left(\mathbf{C}:\epsilon_{\mathrm{el}}\right)\mathrm{F}_{\mathrm{inel}}^{-\mathrm{T}}, \quad \epsilon_{\mathrm{el}} = \frac{1}{2}\left(\mathrm{F}_{\mathrm{el}}^{\mathrm{T}}\mathrm{F}_{\mathrm{el}} - \mathrm{I}\right), \quad \mathrm{F_{el}} = \mathrm{F}\mathrm{F}_{\mathrm{inel}}^{-1}$

$\mathrm{S_{ad}} = \mathrm{S_0} + \mathrm{S_{ext}} + \mathrm{S_q}$

$\epsilon = \frac{1}{2}\left[(\nabla\mathbf{u})^{\mathrm{T}} + \nabla\mathbf{u} + (\nabla\mathbf{u})^{\mathrm{T}}\nabla\mathbf{u}\right]$

$\mathbf{C} = \mathbf{C}(\mathbf{E}, \nu, \mathbf{G})$

Mesh 1



Mesh 1

Study 1

Computation information

Computation time	36 s
CPU	AMD64 Family 21 Model 112 Stepping 0, 2 cores
Operating system	Windows 10

Eigen frequency

Study settings

Description	**Value**
Include geometric nonlinearity	On

Study settings

Description	**Value**
Desired number of eigenfrequencies	47

Description	Value
Desired number of eigenfrequencies	On

Adaptation and error estimates

Description	Value
Adaptation and error estimates	Adaptation and error estimates
Adaptation in geometry	Geometry 1
Weights for eigenmodes	1

Physics and variables selection

Physics interface	Discretization
Solid Mechanics (solid)	physics

Mesh selection

Geometry	Mesh
Geometry 1 (geom1)	mesh1

Results

Data Sets

Study 1/Solution 1

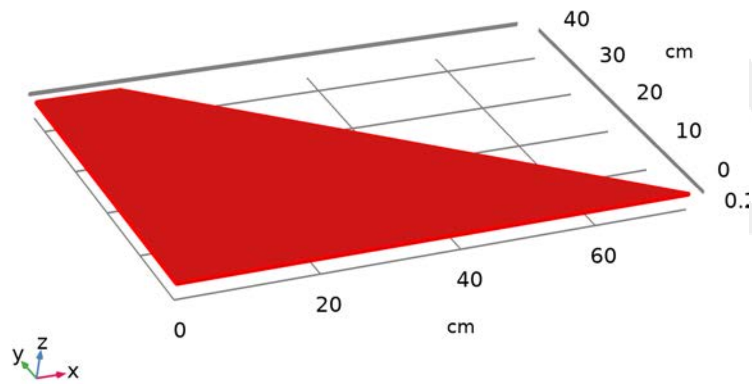
Selection

Geometric entity level	Domain

Selection	Geometry geom1

Solution

Description	Value
Solution	Solution 1
Component	Save Point Geometry 1



Dataset: Study 1/Solution 1

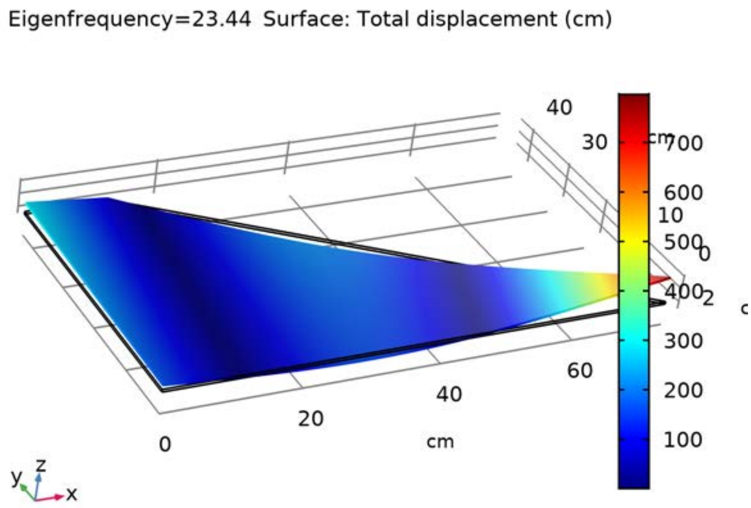
Study 1/Adaptive Mesh Refinement 1

Solution

Description	Value
Solution	Adaptive Mesh Refinement 1
Component	Geometry 1

Plot Groups

Mode Shape (solid)



Surface: Total displacement (cm)

CURRICULUM VITAE

Name Surname : Cem ÖMEROĞLU

Place and Date of Birth :

E-Mail :

EDUCATION :

- **M.Sc.** : 2003, Istanbul Technical University, Graduate School of Arts and Social Sciences, Music Department

- **B.Sc.** : 2000, Yıldız Technical University, Mechanical Engineering Faculty, Mechanical Engineering Department

PROFESSIONAL EXPERIENCE AND REWARDS:

- 2015 - 2020, Lecturing Acoustics and Music Production Techniques at Bahçeşehir University, Sound Technology and Design Graduate Program.
- 2009 – 2016, Lecturer at School of Audio Engineering ISTANBUL.
- 2003 – 2006, Lecturer at Yıldız Technical University, Art and Design Faculty Music Technology Program.
- 2007 – 2020, Front of House Live Sound Audio Engineer of the Turkish rock bands; Kurban and Adamlar.

- 1992 – 2020, Co-former band member, guitar and vocal performer of the Turkish rock band Nekropsi, released several albums (Mi Kubbesi, Sayı 2, 1998, Aylık), singles (Sekizler, Dedikodu, Tatadu) and performed numerous concerts.
- 2000 – 2020…, Recording, mixing and mastering as an audio engineer in the studio, produced musical band's albums, singles and projects like; Nekropsi, Kök, Kes, Helak, Adamlar, Melek, İhtiyaç Molası, Feza, Knight Errant, Karapaks, Angelika Akbar, Cem Tan Quintet and etc.

PUBLICATIONS, PRESENTATIONS AND PATENTS ON THE THESIS:

OMEROGLU, C. (2020). Top plate vibration analysis of the kanun, retrieved Winter 2020, Rast Musicology Journal, Volume 7, No: 3, from https://dergipark.org.tr/tr/download/article-file/1354985