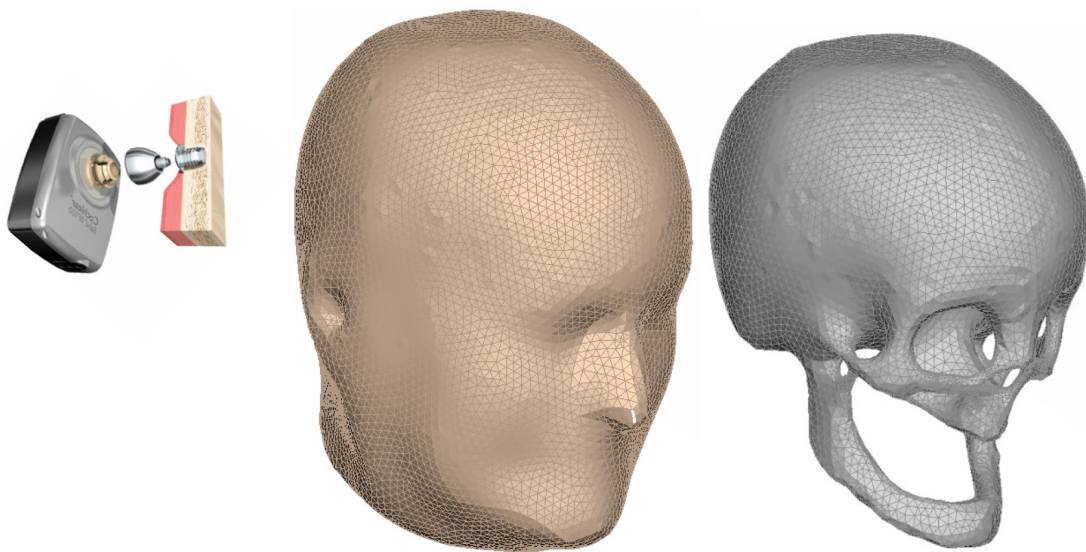




CHALMERS
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Development of 3D Finite Element Model of Human head

Numerical simulation of Mechanical Point Impedance in Human head model for Bone Anchored Hearing Aid (BAHA)

Master's Thesis in Mathematical Sciences

Li Jung Kim

MASTER'S THESIS 2017

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This thesis work was performed at Alten Sweden AB in collaboration with Cochlear Bone Anchored Solutions AB in Mölnlycke, Sweden.



Cover: FE model of human head and skull recreated from ANSA with picture of BAHA implant (Baha® BP100, Cochlear Bone Anchored Solutions AB) embedded in bone from article [1].

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Abstract

Bone-Anchored Hearing Aid (BAHA) is known as the surgically implantable hearing aid for patients whose medical conditions are beyond the stage of wearing the conventional air conductive hearing aid (ACHA) or those who suffer from bone diseases or chronic inflammation of the outer or middle ear. Numerous studies have been conducted to investigate the bone conduction mechanism. Nowadays, with the help of computational mechanics such as the Finite-Element Method (FEM), the performance of BAHA can be improved before the actual costly physical models are built.

This thesis aims to develop FE head models that are readily available for commercial use. This thesis presents two different head models, which enable the simulation of the vibration phenomenon, specifically the mechanical point impedance of the skull bone. One model addresses the artificial head model, and the other originates from the direct segmentation of CT scan with new segmentation software. The final goal is to identify critical parameters for effective bone conduction as well as to improve the current BAHA model.

The simulation results were compared with both test data and literature. This study concludes that both models were successfully able to reproduce results with the test data. Antiresonance frequencies in the simulation results were present at approximately 70 — 90 Hz in the simulator FE model and approximately 200 Hz in the actual human FE model.

The proposed modelling approach will be a stepping stone to quantitatively investigate the biomechanical behavior of bone conduction and provide platforms for the patient-specific optimization of the BAHA configuration with future improvements.

Keywords: Bone Anchored Hearing Aid (BAHA), Bone Conduction, Finite Element Method (FEM), Frequency response analysis, Mechanical Point Impedance, Nonstructural Mass, Fluid-Structure, MSC Nastran, ANSA, RETOMO, Medical Image Segmentation

Preface

This master's thesis has been carried out during the spring semester of 2017 in the department of Mathematical Sciences at Chalmers University of Technology. The project has been proposed by Cochlear Bone Anchored Solutions AB and performed at engineering consultancy firm Alten Sweden AB in Gothenburg under the supervision of Mikael Almquist, Océane Lançon and Saqib Dilshad. The examiner was professor Torbjörn Lundh in the Department of Mathematical Sciences at Chalmers University of Technology.

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Last but not least, I wish to thank my examiner professor Torbjörn Lundh at Chalmers University of Technology for great support throughout the project.

LJ Kim, Gothenburg, September 2017

Nomenclature

Acronyms

BAHA	Bone Anchored Hearing Aid
BC	Bone Conduction
CAE	Computer Aided Engineering
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Method
FS	Fluid Structure
FRF	Frequency Response Function
MPI	Mechanical Point Impedance
SPC	Single Point Constraint

Symbols

$[M]$	Mass matrix
$[K]$	Stiffness matrix
A	Square matrix
I	Identity matrix
λ	Eigenvalue
X	Eigenvector
$\{\ddot{u}\}$	Acceleration vector
$\{u\}$	Displacement vector
$\{\phi\}$	Eigenvector or Mode shape
ω	Angular Frequency

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1

Introduction

1.1 Background

Bone Anchored Hearing Aid (BAHA) is one type of hearing aid that uses the principle of osteo-integrated implants and bone conduction hearing. It is an established medical hearing aid with more than 25 years of clinical experience for patients with conductive or mixed hearing loss [13]. The concept of bone conduction with respect to hearing sensation, refers to sound transmitted via vibrations of the skull. Bone conduction has been a topic for many researchers for its characteristics used in auditory mechanism.

Franke *et al.*[14] investigated the lowest resonance frequency with dry skull filled with gel turned out to be 500 Hz and laid the groundwork for using spherical model for human skull. Then Håkansson group conducted an extensive investigation as to finding first the free skull resonance frequencies with mechanical point impedance from live humans: 1000 Hz and 1500 Hz with standard deviation 100 Hz and 270 Hz. From the work from Carlsson and Håkansson in 1997 [15], vibrations from the transducer of BAHA go directly to the skull bone bypassing the skin from the vibration pathway. Håkansson and Stenfelt [16] also investigated vibration pattern with dry skull filled with viscous polyurethane. Stenfelt and Goode [17] performed a study in six cadaver heads and measured mechanical point impedance and the acceleration response of the cochlear promontory in all three perpendicular directions. Stenfelt *et al.* [18] found the transcranial attenuation of bone conduction (BC) sound to be dependent on stimulation frequency and position.

With the development of computational method such as finite element method (FEM) part of advanced computer aided engineering (CAE), many researchers started to use this as a means to investigate bone conduction (BC) phenomenon. Kim *et al.* (2014) created 3D Finite Element (FE) dry skull filled with polyurethane and measured the mechanical point impedance and acceleration response at different positions in the skull. Chang *et al.* [19] created 3D whole head FE model including detailed bone structure, brain, cerebrospinal fluid (CSF), inner ear, eye balls, cartilages, and soft tissues.

Hence, Cochlear Bone Anchored Solutions AB in Mölnlycke has a need to get a better understanding of the biomechanics of bone conduction (BC) for hearing aid for its current and future projects. Cochlear Bone Anchored Solutions AB has identified

the FEM as a valuable tool for the study of vibration of its devices and would like to extend the use of FEM to a human head model. To the best of our knowledge, there is no commercially available model of the head offering satisfying simulation of bone conduction, but a lot of research has been done to offer a solid basis for such mode.

1.2 Aim

The objective of this study is to develop a valid 3D FE model of the human head, which can be used to investigate and simulate the vibration pattern of bone-conducted sound. Initially, one of the transfer functions (mechanical point impedance) was investigated and simulated in FE models.

The mechanical point impedance measures the motion resistance of skull bone when it is subjected to an oscillatory force of a given frequency range, which is generated by the BAHA. When the titanium implant of the BAHA approaches the ear canal, where the skull impedance level increases, there is an increased sensitivity, which suggests that there is a relationship between the mechanical impedance of the skull and the hearing sensitivity.

This model will enable us to investigate the factors that affect the bone conduction pathway, find the correct position to produce the vibration level for the hearing sensation, and further optimize the device for patient-specific way.

Hence, this study attempts to achieve the following goals:

- Development of FE head simulator model.
 - To understand the physical model and optimize it.
- Development of actual FE human head model from CT scan.
 - To compare with the Head simulator model and understand the effect of the geometry to optimize the Head simulator.
 - As the first step towards a standard FE human head model for commercial use.
- Simulation of a mechanical point impedance in different settings.

1.3 Process of the thesis work

Table 1.1 describes the process of this study. This study aims to create two different models: a head simulator from the CAD file and an anatomically correct 3D FE model of the human head directly from the CT scans and import it to a commercially available structural analysis software.

Table 1.1: Overview of the thesis process

Training Period	3D Geometry Setup	Pre-Process	Analysis	Post-Process	Validation	Final report
Literature Study	CAD model		Modal analysis			Conclusion
Software Training	Segmentation of CT scan with RETOMO	Mesh Generation with ANSA	and Frequency response analysis with Nastran	Visualization with Meta	Comparison with Clinical data and simulation	Discussion Future work

1.4 Delimitations

Due to the complexity of the human head geometry and tissues in the skull, the resulting test data may vary. Various studies have been conducted on bone-conduction hearing aids. Nevertheless, there are some limitations in this study:

- The BAHA device was modeled as point source in the model for further simplification.
- The material properties of foreign materials were assigned in the model instead of those of bone and soft tissue.
- Bone conduction was only limited to the head and not the entire body. Only the head section was examined.
- The effects of body heat and blood vessels were not discussed in this study.

2

Theory

This chapter summarizes the theoretical foundations pertaining to this project, including the knowledge in anatomy and physiology of the ear, fundamentals of the FEM, and essential part of structural analysis that is relevant to the bone conduction simulation.

2.1 Anatomy of ear and Hearing mechanism

Hearing is one of the five sensations critical to the perception of the surrounding and communication. The overall function of the ear is to convert physical vibrations to nervous impulses. In other words, the vibration from sound is transduced to electric signals in the ear, where they are processed by the central auditory system in the brain. To achieve this process, different parts of the ear participate in different manners [20].

2.1.1 Structure of ear

The human ear can be divided into three parts: outer ear, middle ear, and inner ear. The outer ear consists of external parts known as the pinna and ear canal. The pinna is made of cartilage covered by skin and collects the sound and channel into the ear canal. Because the pinna is angled, it can also localize the sound, although this effect is mainly for higher frequencies.

The middle ear is composed of a part of the ear canal and the tympanic membrane, which is also known as the eardrum. The sound is conducted from the outer ear to the ear canal, tympanic membrane and finally the inner ear by three bones (ossicles).

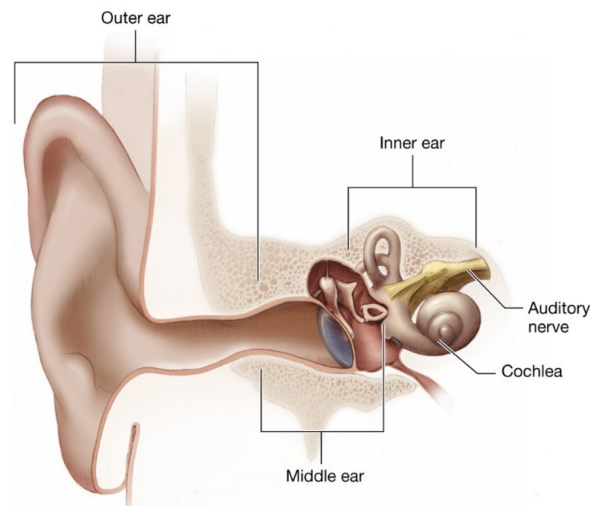


Figure 2.1: Structure of ear: Outer, middle and inner ear [2].

The outer wall of the middle ear is the eardrum, and the inner wall is the cochlea. The name “cochlea” comes from the shape of a snail because this part of the inner ear resembles a snail. It has two and half turns and contains the organ of hearing, which is known as the membranous labyrinth and perilymph fluid [20]. Assuming that the cochlea is unrolled, it would have two fluid-filled tubes, which are separated by an elastic membrane called the basilar membrane along the length of the cochlea (figure 2.2).

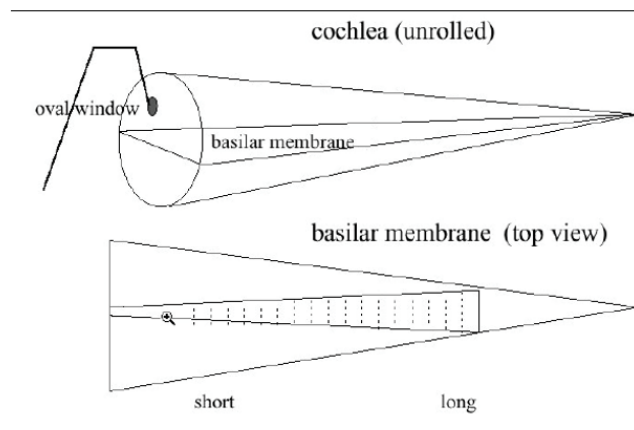


Figure 2.2: Schematics of unwound cochlea and basilar membrane [3].

When sound is transferred to this structure, the basilar membrane begins to oscillate, and the frequency of oscillation depends of the tone of the sound. In other words, this structure operates in the opposite manner of a piano: the sounds are generated by vibrations of strings in the piano, whereas the vibrations are first caused by the sound in the cochlea [21]. The basilar membrane responds to the highest frequencies at the stiff base nearest the oval window and to lower frequencies as one progresses toward the apex. The cochlea can be considered an efficient frequency analyzer.

To summarize anatomy of ear, the ear consists of parts responsible for sound conducting structure and sound transducing structure. The sound conducting structure has two parts: the outer ear, which is composed of pinna and ear canal and the middle ear consisting of tympanic membrane. The inner ear or cochlea transduces audible ranged vibration to a nervous impulse which is then taken to the brain where it is processed and finally perceived as sound.

2.1.2 Hearing pathway

There are two main sound transmission pathways with which we can hear sounds.

- Air conduction, where the sound is captured by the pinna, travels to the ear canal, reaches the cochlea, and is transmitted to the brain as an electrical signal.
- Bypassing the outer ear and middle ear so that the vibration directly reaches the cochlea via bone conduction.

Both air conduction and bone conduction support the conversion mechanism that transforms mechanical vibrations to electrical impulses in the inner ear (cochlea). This hypothesis was confirmed by von Békésy [22] by simultaneously sending the same tone of sound via both pathways and finding that they perceptually cancelled each other by amplitude and phase adjustments of the air conduction signal. If the processing mechanism in the cochlea for the two types of sound conduction were different, cancellation would not have occurred.

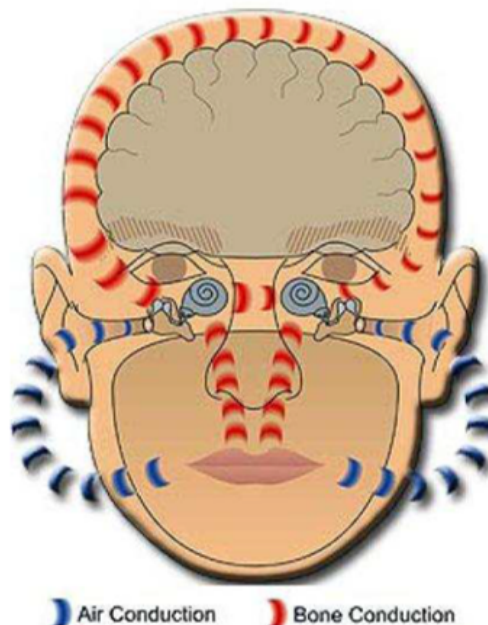


Figure 2.3: Schematics illustration of air conducted and bone conducted sound pathway [4].

2.1.2.1 Air conduction

Air conduction is referred to as the primary hearing pathway and indicated by the blue path in figure 2.3. As described, the airborne sound is captured by the pinna and travels into the ear canal. When the sound induces vibration in the tympanic membrane, the transmitted ossicles in the middle ear become a sound pressure in the cochlea. In short, air conduction is the mechanism with which the sound energy is captured by the outer ear and transmitted by the middle ear to the cochlea and eventually to the upper portion of the brain (auditory cortex), which causes a sound perception.

2.1.2.2 Bone conduction

Bone conduction is a method of auditory stimulation, where the stimulation is transmitted by the skull and results in sound pressure changes in the cochlea. The concept is sometimes divided into body conduction and bone conduction, but Stenfelt *et al.* [23] stated that both should be referred to as bone conduction (BC). The topic in this project only addresses bone conduction stimulation that causes hearing sensation and not the vibration stimulation that provides tactile sensations. This pathway is the secondary auditory pathway to supplement air conduction hearing. Since this process is almost negligible in the case of indirect stimulation, this type of hearing process can contribute to the hearing sensation when it is directly connected to the skull bone.

In general, the source of BC sound transmission involves several pathways, and there is no definitive method to clearly distinguish them. There are five main theories regarding the bone conduction sound pathways: sound pressure in the ear canal, occlusion effect (blocking the outer ear), inertia of the middle ear ossicles, inertia of the cochlea fluid and fluid pressure transmission, and alteration of the cochlear space.

It is well known that the bone conduction sound perception is mainly caused by the basilar membrane vibration. A direct measurement of this motion shows similar excitation patterns if the stimulation is by air conduction or bone conduction. Since the inherent response of the basilar membrane is a wave motion that begins at the base of the cochlea and travels toward the apex, the expected important parameter for the bone-conduction-induced wave motion on basilar membrane is the pressure difference between the upper and lower ducts of the cochlea.

2.1.3 Types of hearing loss

There are three main types of hearing losses: conductive, sensorineural, and mixed hearing loss. A conductive hearing loss is caused by the reduced transmission of sound by the outer or middle ear. This condition may occur because of ear wax in the ear canal, eardrum damage by infection or trauma, damage or stiffening of the ossicles in the middle ear or fluid in the middle ear by infection. These issues result

in the attenuation of sound that approaches the cochlea, so one can perceive a quieter sound than normal. Since the amount of tone is associated with the frequency, one may also perceive a different tone. Unlike sensorineural hearing loss, conductive hearing loss does not usually cause catastrophic distortions or abnormalities in sound perception. It can often be treated by proper medications to treat infections or surgery.

Sensorineural hearing loss, which involves the cochlea and neural structures, is caused by a damage of the structures inside the cochlea. This condition is caused by intense sounds or ototoxic (toxic to the ear) chemicals such as certain types of antibiotics, high-blood-pressure drugs, or solvents. It can also be caused by infection, allergies, metabolic disturbances or autoimmune by hereditary factors. The combinations of those issues can produce diverse damage to the cochlea or even beyond the cochlea [24].

Mixed hearing loss refers to the combined conductive and sensorineural hearing loss. It can be caused by birth defects such as the large vestibular aqueduct syndrome and disorders of the ear bone, both of which can damage the inner ear and middle ear. Middle-ear infections can also cause mixed hearing loss, but the sensorineural parts can be recovered [25].

2.1.4 Bone Anchored Hearing aid (BAHA)

Bone-Anchored Hearing Aid, which is commonly referred to as BAHA, is a type of Bone Conduction Devices (BCDs) as illustrated in figure 2.4. In the BAHA, percutaneous (skin penetrating) titanium is directly integrated to the skull bone.

Since Per-Ingvar Br  memark identified that titanium implants could be firmly attached to bones in the late 1960s, the concept of osteointegration has led to the development of hearing aids to treat hearing loss that cannot be solved with the conventional air conduction hearing aids. Unlike conventional BCDs, BAHA eliminates the circulatory problem of compressing the skin, and the transmission of vibrations becomes more efficient because it is directly connected to the bone instead of through the skin, as illustrated in figure 2.5

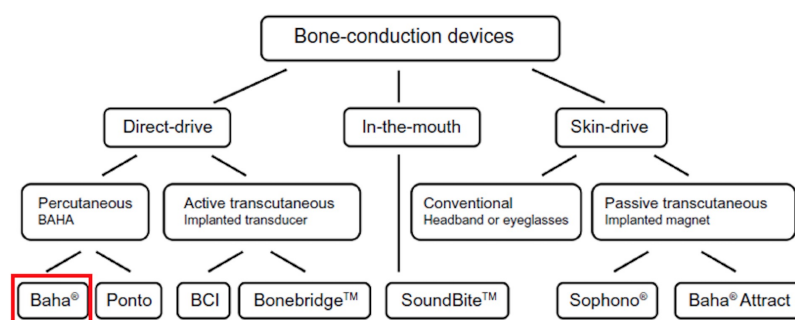


Figure 2.4: Category of different kinds of Bone Conduction Devices (BCDs) [1].

The BAHA device became commercially available from 1987 since its onset of a project collaboration between Chalmers University of Technology, Sahlgrenska University Hospital and Brånemark Osteointegration Center [1]. In addition, the BAHA procedure was cleared by the U.S Food and Drug Administration (FDA) in 1997 [26]. The device has been improved over many years and treated more than 150,000 patients till 2014.

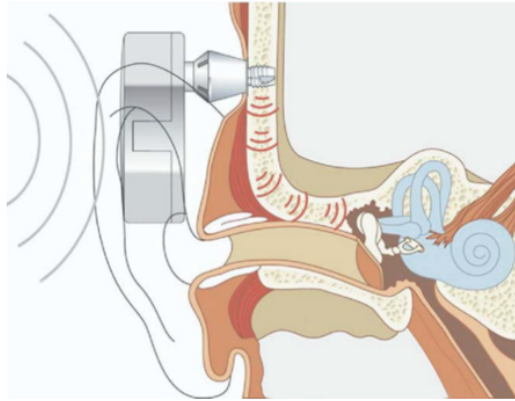


Figure 2.5: Schematic overview of BAHA implant in function [5].

The BAHA device requires a surgical procedure where a 2 x 3 cm retro curricular area (behind the ear) is made hairless, the subcutaneous tissue is incised, and the titanium implant with its skin penetrating coupling is placed at approximately 55 mm from the ear canal at the parietal bone (figure 2.6). After the surgical procedure, it takes approximately three months for the titanium implant to integrate to the bone for adult patients.

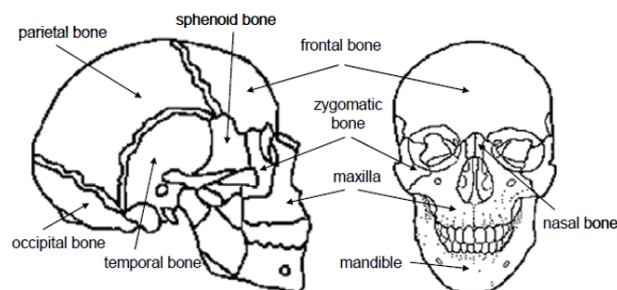


Figure 2.6: Different parts of skull and facial bone [3].

The BAHA consists of a microphone, an amplifier, and a transducer in one external housing, as illustrated in figure 2.7. The microphone captures the surrounding sounds, and the sound signals are amplified by the amplifier. The amplified sound signals vibrate the transducer, and this mechanical vibration is transferred to the skull bone via titanium abutment [6].

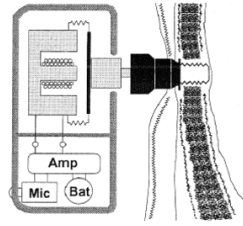


Figure 2.7: BAHA components including sound capturing, amplifying and vibration generating parts [6].

2.2 Finite element method (FEM)

The finite-element method (FEM) or finite-element analysis (FEA) is one of the most widespread approaches for numerical modelling in engineering design processes. Its name was coined by Clough in 1960. Since then, the FEM has been applied to various engineering problems [27]. Because the human mind faces many challenges to understand the behavior of complex systems and their creations in one operation, if the system is divided into individual components or “elements”, the system is readily understood, and one can rebuild the original system to study its behavior [28].

In other words, the FEM simplifies the structure by dividing it into small pieces (finite elements) that are interconnected by common points, which are called “nodes”. The nodes are at the corners of the elements but can also be on the sides or in the middle of the element. A set of elements is called a mesh, and there are many computational algorithms to formulate a mesh to increase the accuracy of the result. The analysis calculates the displacement at the nodes for the specifically designed loading that is applied to the FE model.

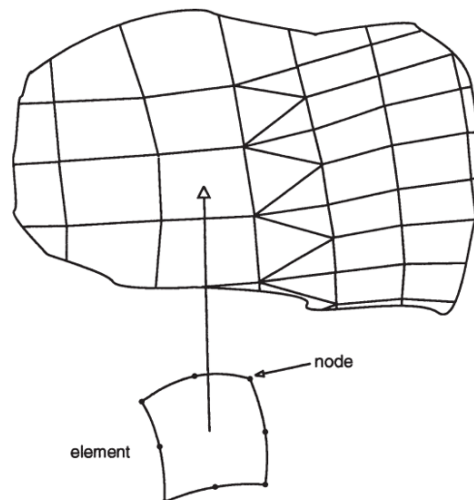


Figure 2.8: Finite elements and nodes [7].

To formulate finite-element equations to describe a physical problem, several approaches can be used. If the physical problem can be described by a differential equation, the most common method to formulate a finite element is Galerkin method. Suppose that the model is a simple one-dimensional linear element, whose length is $2L$ as illustrated in Figure 2.9

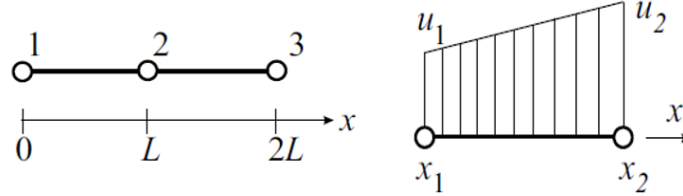


Figure 2.9: One-dimensional linear elements along the x-axis. The figure in the right shows interpolation inside the element [8].

Assuming we are numerically solving following differential equation, from 0 to $2L$ where u is unknown solution.

$$a \frac{d^2 u}{dx^2} + b = 0 \quad 0 \leq x \leq 2L \quad (2.1)$$

With following boundary conditions

$$u_{x=0} = 0 \quad a \frac{du}{dx} \bigg|_{x=2L} = R \quad (2.2)$$

The element has two nodes as shown in right side of figure 2.9 and approximation of the function can be done like below.

$$u = N_1 u_1 + N_2 u_2 = [N] \{u\} \quad (2.3)$$

$$[N] = [N_1 \ N_2] \quad (2.4)$$

$$\{u\} = \{u_1 \ u_2\} \quad (2.5)$$

Those N_i are called shape functions which are used for the interpolation of $u_{x=0}$ using their nodal values

$$N_1 = 1 - \frac{x - x_1}{x_2 - x_1} \quad N_2 = \frac{x - x_1}{x_2 - x_1} \quad (2.6)$$

Nodal values u_1 and u_2 are unknowns and those should be determined from the discrete global equation system.

After the substitution of u which is expressed through its nodal values and shape functions, it has following approximate form in the differential equation.

$$a \frac{d^2}{dx^2} [N] \{u\} + b = \psi \quad (2.7)$$

ψ is non zero residual due to approximate representation of a function inside a finite element. The Galerkin method implements residual minimization by multiplying terms of the aforementioned equation by shape functions, integrating over the element and equate to zero, it results in

$$\int_{x_1}^{x_2} [N]^T a \frac{d^2}{dx^2} [N] \{u\} dx + \int_{x_1}^{x_2} [N]^T b dx = 0 \quad (2.8)$$

Using integration by parts leads to the discrete form if the differential equation for the finite element form

$$\int_{x_1}^{x_2} \left[\frac{dN}{dx} \right]^T a \left[\frac{dN}{dx} \right] dx \{u\} - \int_{x_1}^{x_2} [N]^T b dx - \left\{ \begin{matrix} 0 \\ 1 \end{matrix} \right\} a \frac{du}{dx} \Big|_{x=x_2} + \left\{ \begin{matrix} 1 \\ 0 \end{matrix} \right\} a \frac{du}{dx} \Big|_{x=x_1} = 0 \quad (2.9)$$

In finite element process and solid mechanics, those are usually presented as following

$$[K] \{u\} = \{F\} \quad (2.10)$$

$$[K] = \int_{x_1}^{x_2} \left[\frac{dN}{dx} \right]^T a \left[\frac{dN}{dx} \right] dx \quad (2.11)$$

$$\{F\} = \int_{x_1}^{x_2} [N]^T b dx + \left\{ \begin{matrix} 0 \\ 1 \end{matrix} \right\} a \frac{du}{dx} \Big|_{x=x_2} - \left\{ \begin{matrix} 1 \\ 0 \end{matrix} \right\} a \frac{du}{dx} \Big|_{x=x_1} \quad (2.12)$$

$[K]$ is called stiffness matrix, $\{F\}$ is force or load vector. Now, go back to our considered simple case, where element length is L and stiffness matrices and the force vectors can be calculated by

$$[K_1] = [K_2] = \frac{a}{L} \begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \quad (2.13)$$

$$\{F_1\} = \frac{bL}{2} \begin{Bmatrix} 1 \\ 1 \end{Bmatrix} \quad \{F_2\} = \frac{bL}{2} \begin{Bmatrix} 1 \\ 1 \end{Bmatrix} + \begin{Bmatrix} 0 \\ R \end{Bmatrix} \quad (2.14)$$

Those two relations above provide finite element equations for the two separate finite elements. A global equation system for the system with two elements and three nodes can be calculated by an assembly of element equations. In our system, it can be seen that the two elements are connected by node number 2 in as it is seen in figure 2.9. Therefore, the final global equation system is

$$\frac{a}{L} \begin{bmatrix} 1 & -1 & 0 \\ -1 & 2 & -1 \\ 0 & -1 & 1 \end{bmatrix} \begin{Bmatrix} u_1 \\ u_2 \\ u_3 \end{Bmatrix} = \frac{bL}{2} \begin{Bmatrix} 1 \\ 2 \\ 1 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ R \end{Bmatrix} \quad (2.15)$$

After applying the boundary condition which is $u_{x=0} = 0$, the appearance of global equation will be

$$\frac{a}{L} \begin{bmatrix} 1 & 0 & 0 \\ 0 & 2 & -1 \\ 0 & -1 & -1 \end{bmatrix} \begin{Bmatrix} u_1 \\ u_2 \\ u_3 \end{Bmatrix} = \frac{bL}{2} \begin{Bmatrix} 1 \\ 2 \\ 1 \end{Bmatrix} + \begin{Bmatrix} 0 \\ 0 \\ R \end{Bmatrix} \quad (2.16)$$

The finite element solution with the use of simplest element is piece –wise linear. More and more accurate finite element solution can be obtained by increasing the number of simple elements (hence the concept of mesh convergence will appear in Method section) or by using elements with more complex shape functions [8].

Back to the equation $[K]\{u\} = \{F\}$, qualitatively speaking, the stiffness matrix represents the material property and force vector shows action and nodal degree of freedom, , represents the system’s behavior as a result of properties and actions assigned and usually engineers want to obtain this unknown variable at nodes. That is to say, to solve this,

$$\{u\} = [K]^{-1}\{F\}$$

Table 2.1: Representation of three parameters in other physical problems [11].

	Property [K]	Behavior {u}	Action {F}
Elastic	Stiffness	Displacement	Force
Thermal	Conductivity	Temperature	Heat source
Fluid	Viscosity	Velocity	Body force

One may wonder how the mesh is formulated on such a complicated geometry if the elements are created based on the geometry of interest. It is crucial to consider the balance between deforming the continuum, i.e., the system, and the creation of a finite grid or mesh, so that the numerical method can handle the large distortions and provide an accurate resolution for the material interfaces and moving boundaries.

In other words, when developing a computer code to create a mesh, one must select the appropriate *kinematical description* (branch of physics concerned with pure motion of body without considerations of the involved forces) of the continuum. This selection determines the relationship between the system deformation and the mesh of the calculating zones.

There are two classical methods to describe motion: Lagrangian description and Eulerian description. The Arbitrary Lagrangian-Eulerian (ALE) description was developed to combine the advantages of these classical kinematical descriptions to minimize the failure of capturing large deformations.

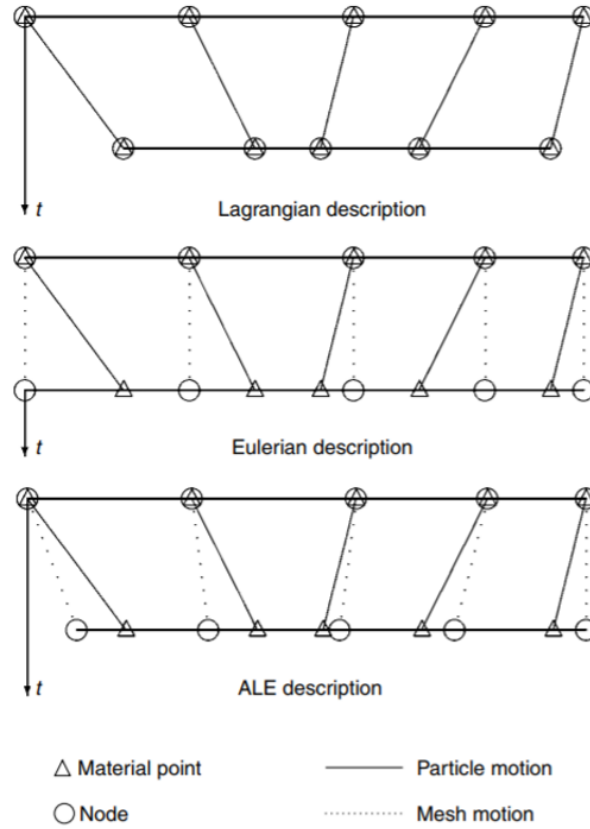


Figure 2.10: Example of Lagrangian, Eulerian and ALE method in one dimension [9].

In Lagrangian algorithm, the individual nodes of the mesh follow the material particle during motion. This method is commonly used in structural mechanics and enables an easy tracking of free surfaces and interfaces between different materials as shown in figure 2.10, where the nodal points move with the mesh. The disadvantage is that it cannot follow large distortions of the computational domain without relying on frequent remeshing [9].

Eulerian algorithm is generally prevalent in modelling fluid dynamics. As illustrated in figure 2.10, the mesh is fixed, and the system moves with respect to the grid. In the Eulerian description, a fixed void mesh is created in the computational space and filled with the system material at the points where the model must be present. This approach demands a fine mesh to create sufficiently accurate results; thus, it requires longer computational time [29].

Because of the inconvenience of using purely Lagrangian and purely Eulerian algorithms, the combined technique, which is the ALE description, was developed, where the nodes of the mesh move with the continuum in the normal Lagrangian manner and held fixed in Eulerian manner. In addition, unlike Eulerian method, the mesh is not fixed in space, so there are fewer void space elements, and the material is free to move with the mesh. Thus, this method has the stable nature of Eulerian algorithm and low computational cost of Lagrangian method [9],[29].

2.3 Structural Analysis

Structural analysis is a crucial part of any design, implementation, and maintenance process in vehicles, machine tools, spacecraft, buildings and even medical devices. One of the mechanical aspects discussed in this project is vibration. Vibration is the movement when a body is under an oscillatory motion at a reference position. Vibration can be periodic or random such as tires on a road. The rate of vibration cycles is called the frequency. For general terminology, repetitive motions with low frequencies and regular behaviors are categorized as “oscillation”, whereas any repetitive motion regardless of frequency, with low amplitudes and irregular, random motion is categorized as “vibration”. Those terms are often used interchangeably in the literature.

Vibration can naturally occur in engineering systems and are representative of their free and natural dynamic behavior, whereas the system can be forced via stimulation, which can be derived from an internal system or an external source. When the frequency of the force is identical to its natural frequency, the system will rigorously respond, which is known as resonance, and its frequency is called the resonance frequency. The alternative terms for resonance frequency are characteristic frequency, fundamental frequency, and normal frequency. It is imperative to study the system with consideration of the resonance frequencies because operations at those frequencies are not desirable and may even be fatal [30].

2.3.1 Modal Analysis

Modal analysis is the first step and plays an important role in the sound and vibration analysis. By performing modal analysis, one can find the system’s natural frequency and mode shapes (shape of vibration) without external force and damping. The results of modal analysis characterize the basic dynamic behavior of the structure and indicate how the structure will respond under dynamic loading.

In this analysis, the resonance frequency will be referred to as the eigenfrequency, and the shapes of vibration will be indicated as eigenvectors because of the mathematical background. In its mathematical representation, when the structure is assumed to be linear and the eigenvalue analysis is performed, one can begin with the calculating equation [31].

$$[\mathbf{M}]\{\ddot{\mathbf{u}}\} + [\mathbf{K}]\{\mathbf{u}\} = \mathbf{0} \quad (2.17)$$

To solve equation 2.17 under harmonic solution of the form. This harmonic form is the key to the numerical solution as well as physical significance. This harmonic form of the solution indicates that all the degrees of freedom of the vibrating structure move in a synchronous manner. The overall structural configuration does not change its basic shape during in motion but only its amplitude changes.

$$\{u\} = \{\phi\} \sin \omega t \quad (2.18)$$

Substituted into the equation of motion which is denoted in equation 2.17, the following is obtained. After simplifying the equation,

$$-\omega^2[M]\{\phi\} \sin \omega t + [K]\{\phi\} \sin \omega t = 0 \quad (2.19)$$

This equation 2.20 is called the eigen-equation which represents a set of homogeneous algebraic equations for components of the eigenvector and makes the basis for the eigenvalue problem.

$$([K] - \omega^2[M])\{\phi\} = 0 \quad (2.20)$$

The eigenvalue problem is a specific equation form that has many applications in linear matrix algebra. The basic form of an eigenvalue problem is shown in equation 2.21.

$$(A - \lambda I)x = 0 \quad (2.21)$$

Physical representation of natural frequencies and mode shapes is done by stiffness and mass in the eigen-equation in structural analysis. Hence, the eigen-equation is written in terms of stiffness matrix, mass matrix, and angular frequency as shown in equation 2.20.

There are two possible solution forms for equation 2.20

If the determinant matrix of $([K] - \omega^2[M]) \neq 0$, the only solution is $\{\phi\} = 0$ and this is the trivial solution which does not provide any important information from a physical point of view because it represents the case of no motion.

On the other hand, if the determinant matrix of $([K] - \omega^2[M]) = 0$, then non-trivial solution is obtained. The general mathematical eigenvalue problem reduces to one of solving the equation of the form from a structural engineering viewpoint where $\lambda = \omega^2$

$$([K] - \omega^2[M])\{\phi\} = 0 \quad (2.22)$$

$$([K] - \lambda[M])\{\phi\} = 0 \quad (2.23)$$

The determinant is zero only at a set of discrete eigenvalues λ_i or ω_i^2 where eigenvector satisfies and corresponds to each eigenvalue so it can be rewritten as equation 2.24.

$$([K] - \omega_i^2[M])\{\phi_i\} = 0 \quad i = 1, 2, 3, \dots, \quad (2.24)$$

Each eigenvalue and eigenvector defines a free vibration mode of the structure of interest. The i-th eigenvalue is related to its i-th natural frequency as below.

$$f_i = \frac{\omega_i}{2\pi} \quad (2.25)$$

The number of possible eigenvalues and eigenvectors is equal to the number of degrees of freedom that have mass or the number of dynamic degrees of freedom. Moreover, there are many characteristics of natural frequencies and mode shapes that make them useful in various dynamic analyses. In the field of vibration analysis, common type of analysis is based on the linear behavior of the structure. That is to say, its response is linear and when a load is removed, the structure returns to its original position or shape when load is removed.

Therefore, when a linear elastic structure is vibrating in free or forced vibration, its deformed shape at any given time is a linear combination of all of its normal modes.

$$\{u\} = \sum_i (\phi_i) \xi_i \quad (2.26)$$

Moreover, if $[K]$ and $[M]$ are symmetric and real the following equations hold.

$$\{\phi_i\}^T [M] \{\phi_j\} = 0 \quad \text{if } i \neq j \quad (2.27)$$

$$\{\phi_j\}^T [M] \{\phi_j\} = m_j = \text{jth generalized mass} \quad (2.28)$$

$$\{\phi_j\}^T [K] \{\phi_j\} = 0 \quad \text{if } i \neq j \quad (2.29)$$

$$\{\phi_i\}^T [K] \{\phi_j\} = k_j = \text{jth generalized stiffness} = \omega^2 m_j \quad (2.30)$$

From equations 2.28 and 2.30, Rayleigh's equation is obtained as below.

$$\omega_j^2 = \frac{\{\phi_j\}^T [K] \{\phi_j\}}{\{\phi_j\}^T [M] \{\phi_j\}} \quad (2.31)$$

Equations 2.27 and 2.29 are the orthogonal property of normal modes, which explain that each normal mode is different from all others. The orthogonality of the modes means that each mode shape is unique, and one mode shape cannot be obtained by a linear combination of any other mode shapes in the physical interpretation [31].

In the modal analysis, the mass participation factor or modal mass can answer the question of the “strong” response of a system. In other words, it is notably useful to calculate how much mass is associated with each natural frequency. Thus, each natural frequency captures a certain percentage of mass of the structure, and its total dynamic response is the sum of all resonance frequencies of the system.

Moreover, those few natural frequencies at approximately the first resonance frequency have most of the total mass of the model. Thus, it is notably important to examine the first few resonance frequencies.

Lanczos method

Lanczos method is one of the real eigenvalue extractions commonly used in software MSC Nastran for normal mode analysis. In other words, Lanczos method is one of the numerical approaches to solve for the natural frequencies and mode shapes. From numerical point of view, it is iterative method when calculating real eigenvalues. Its standard form is $A\nu = \lambda\nu$.

The generalized eigenvalue problem if we revisit equation 2.20 and simplify it by assigning eigenvector $\{\phi\} = x$, and removing all the matrix brackets it becomes

$$Kx = \omega^2 Mx \quad (2.32)$$

The solution of equation 2.32 by the Lanczos method would produce the largest eigenvalues. However, when it comes to vibration analysis, the lowest or lower part of eigenvalues are the ones of interest. Therefore, a shift of σ , close to the eigenvalues of interest is introduced and then the problem is inverted because we want to have lowest eigenvalue.

From $\lambda = \omega^2$ applying shift σ , then invert

$$\omega^2 = \frac{1}{\lambda} + \sigma \quad (2.33)$$

Then substitute 2.17 in equation 2.16 above,

$$Kx = \left(\frac{1}{\lambda} + \sigma\right) Mx \quad (2.34)$$

Then multiplying σ in each side of equation 2.18 yields,

$$Kx\sigma = Mx + Mx\sigma\lambda \quad (2.35)$$

Rearrangement of 2.19 will give

$$\lambda[K - \sigma M] = Mx \quad (2.36)$$

Multiplying both sides with $[K - \sigma M]^{-1}$ will produce

$$\lambda x = [K - \sigma M]^{-1} Mx \quad (2.37)$$

Let $A = [K - \sigma M]^{-1} M$ and substitute this in equation 2.37 and rearrange terms will give standard Lanczos form.

$$Ax = \lambda x$$

Application of Lanczos algorithm requires the computation of the vector in a given eigenvalue. At the same time, it is important to avoid excessive computational cost and time for finding the inverse of the matrix in equation 2.37 which will result in full matrix. This will lead to losing the advantages of having banded matrix (matrix

where all of the nonzero quantities lie only on diagonal band). Instead, the following procedure can be implemented [32].

Let complex conjugate of stiffness matrix as $\overline{K} = [K - \sigma M]$

$$\overline{K} = LDL^T \quad (2.38)$$

Where L is a lower triangular matrix with unit diagonal D is a diagonal matrix and transposed form L^T and due to its symmetric nature it becomes the upper triangular matrix. Then rearrange the terms and introduce y

$$A\nu = (LDL^T)^{-1}M\nu \quad \text{or} \quad A\nu = (L^T)^{-1}(LD)^{-1}M\nu \quad (2.39)$$

$$L^T A\nu = (LD)^{-1}M\nu = y \quad (2.40)$$

$$M\nu = LDy \quad (2.41)$$

Solve for $A\nu$,

$$L^T(A\nu) = y \quad (2.42)$$

2.3.2 Frequency response analysis

The frequency response analysis is a method to compute the structural response under steady-state oscillatory excitation. It can be stated that the frequency response of a dynamic system is the response to a sinusoidal excitation. The response of the system over a range of excitation frequencies can be changed, which represents the frequency response of a system. Here, the frequency is the independent variable, so this section addresses the frequency domain.

The signals can certainly be transformed to the frequency domain using the Fourier transform, even when the signals are not periodic. A given time signal, which is equivalent to a Fourier spectrum, contains all frequency components of the signal and can be analytically or computationally determined. Therefore, at least for the linear dynamic systems, the time domain and frequency domain can be equally represented. The response to a sinusoidal form of excitation is considered the “frequency” domain analysis; in particular, this part addresses the “forced response” analysis.

A frequency response function or complex frequency response function is called a transfer function, and it is a frequency domain input-output relationship, which can also be considered equivalent to the time domain impulse response function via Fourier transformation [7]. Since the FRF is a complex function, it can also be represented in terms of magnitude and phase. This FRF expresses the structural response to an external force as a function of frequency, and its response can be calculated in terms of the displacement, velocity and acceleration. Moreover, those parameters can be present in the denominator or numerator [33].

To return to its basic concept, the transfer function $H(\omega)$ as a function of the frequency is expressed as the quotient of the output parameters (displacement, velocity and acceleration), and the input force is expressed as a function of the frequency. A more detailed explanation can be found in the **Glossary**.

Table 2.2: Different types of frequency response (transfer) functions [12].

Definition	Name
Displacement / Force	Receptance, admittance, dynamic compliance
Velocity / Force	Mobility
Acceleration / Force	Accelerance, inertance
Force / Displacement	Dynamic stiffness
Force / Velocity	Mechanical impedance
Force / Acceleration	Apparent Mass

Mechanical point impedance

The mechanical point impedance (MPI) or driving point mechanical impedance Z_{mech} is one of the transfer functions that describes the complex force-to-velocity ratio as follows. where

$$Z_{mech} = \frac{F(f)}{v(f)}$$

Where $F(f)$ is the force as a function of frequency, and $v(f)$ is an output velocity, which is also a function of frequency. In the case of modal testing, Z_{mech} is calculated from the acceleration with integration in the frequency domain using equation 2.43 from [34].

$$a(f) = 2\pi f \times v(f) \quad (2.43)$$

which will result in

$$Z_{mech}(f) = 2\pi f \times \frac{F(f)}{a(f)} \quad (2.44)$$

The mechanical impedance in this project is only limited to the measurement of force and acceleration or velocity at the “driving point” (hence the name) such as a vibrating surface with which the subject is in contact. This is the first category (force-motion method) of whole-body vibration biomechanical research [35].

In particular, in this thesis, the mechanical point impedance approach is the first step in investigating the bone conduction because it measures the local resistance of motion at the skull bone and key concepts to investigate the input impedance of the skull at the implant site. This will form a basis for transmission of vibration into the head and improvement of the device output impedance.

3

Methods

This section will present an overview of the simulation and methodology in FE modeling, including medical image segmentation. The use of segmentation tools followed by the procedures of FE modeling and simulation will be described.

3.1 Overview of FE model simulation

FE analysis approximates the geometry or original model in several different types of elements. The behavior of the structure of interest is obtained by analyzing the overall behavior of the elements. The figure below illustrates the process from the real model to visualization of result. Each of these procedures requires several sets of commercial software.

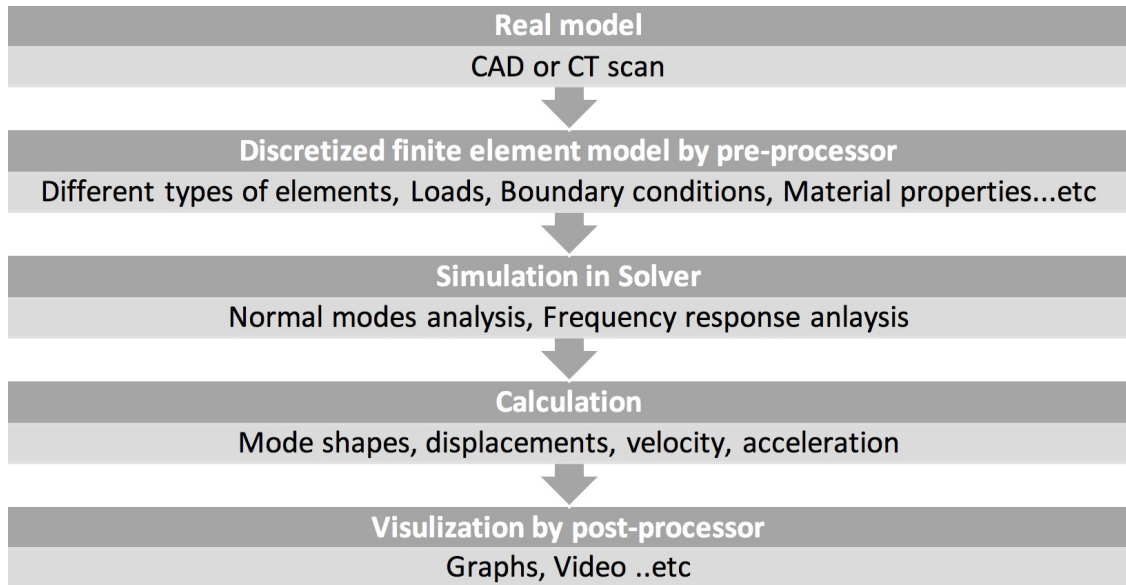


Figure 3.1: Finite element process. Inspired by [10].

Generally, the real model is formulated in CAD such as vehicles, airplanes, buildings, and bridges. In this study, in order to create actual human head model, the medical image was segmented, and a rough mesh was formed; this rough mesh served as the ‘geometry’. This procedure was performed by RETOMO 17.1.0 (BETA CAE Systems, Thessaloniki, Greece).

Then, this file was transferred to the pre-processor ANSA 17.1.1 (BETA CAE Systems, Thessaloniki, Greece) to generate meshes for the different parts of the human head with appropriate boundary conditions. The full FE model was computed by a traditional structural analysis solver, MSC Nastran 2014.1 (MSC Software Corporation, Newport Beach, CA, United States). The computation result was post-processed with META 16.1.0 (BETA CAE Systems, Thessaloniki, Greece).

In this project, there will be two different models to represent human head model: one is a head simulator and the other is the actual human head model which was directly recreated from CT scan. Both models share the same material except additional material for real head model.

Also, for modelling brain parts, two different techniques were applied. One model, the part of the model was modified as mass points which were distributed to all of the grid points of the other parts of the structure. In other words, the mass of the part was distributed as points or dots, and the simulation only considered the mass of the part, not the stiffness. The model with this technique will be referred to be as ‘Nonstructural Mass (NSM)’ model. This NSM model is only for the head simulator model not for actual human head model.

The other brain model assumes brain to have fluid property due to unknown parameters of material used in this model for vibration analysis. Hence the model with fluid brain skull structure will be named as Fluid – Structure model.

3.2 Formulation of Head Simulator model

3.2.1 Geometry

Due to the confidentiality from performing industrial Master's thesis, following description of head simulator will be limited meaning all the pictures and material properties of Head simulator are hidden.

The head simulator geometry was created with CAD software. Head simulator was designed to be exposed to frequency sweep sine wave representing vibration generated by BAHA. The head simulator model is composed of MATERIAL 1 and MATERIAL 2.

3.2.2 Construction of 3D FE modelling

3.2.2.1 Material properties

One of the important parts of the simulation with FE models is to assign the right material properties to the model. The structural shell domain was defined as MAT1. Young's modulus, density, and Poisson's ratio were assigned to the shell domain.

The brain simulant was modeled as fluid and defined as a MAT 10 entry in the Nastran deck. In addition, all grid points in the fluid model had a CD of -1 and the PFLUID property.

CD represents the identification number of the coordinate system where the displacements, degree of freedom, constraints, and solution vectors are defined at the grid point. Table 3.1 shows the material properties and the bulk modulus was used in fluid part instead of the Young's modulus.

Table 3.1: Material parameters for components of Head simulator model.

Material	Young's modulus (MPa)	Bulk Modulus (MPa)	Density (ton/mm ³)	Poisson's ratio
MATERIAL 1	X_1		X_3	X_5
MATERIAL 2		X_2	X_4	

3.2.2.2 Mesh (batch mesh)

To determine the initial element size, Nastran recommends six to eight elements to cover one wavelength. Since a higher frequency leads to a shorter wavelength, calculations should be performed with the highest frequency (10,000 Hz in this case).

After the selection of an adequate element size, collective FE elements or meshes were created with the ‘Batch mesh’ function in ANSA. This function allows different types of meshing cases depending on the user’s need to perform automatic mesh generation, which is based on mesh parameters and the quality criteria of each part.

Table 3.2: Shell Mesh parameters.

Mesh Parameters	Value
Mesh Type	<i>General</i>
Element Type	<i>Tria</i>
Element order	<i>1st</i>
Target length	5 mm
Minimum target length	1 mm
Maximum target length	6 mm
Distortion distance	20 %
Distortion angle	0

Table 3.3: Solid Mesh parameters.

Mesh Parameters	Value
Mesh Type	<i>Tetra FEM</i>
Element Type	<i>Tetra</i>
Element order	<i>1st</i>
Maximum growth rate (1-3)	1.2
Maximum element length	<i>max_shell_size</i>
Nastran Maximum aspect ratio	4

First, a 2D or shell mesh was created. Then, a 3D, solid or volume mesh was created based on the 2D mesh. Fluid parts were also created using the same approach. Tables 3.2 - 3.4 list the mesh parameters and quality criteria used in this model.

Table 3.4: Quality criteria for shell and solid elements for Head simulator model.

Criteria	Calculation	Shell mesh	Solid mesh
Aspect ratio	NASTRAN	3	8
Skewness	NASTRAN/ABAQUS	45	0.1
Warping	NASTRAN	10	10
Taper	NASTRAN	0.35	
Jacobian	NASTRAN	0.7	0.7
Minimum angle trias/tetras	NASTRAN/ABAQUS	30	20
Maximum angle trias/tetras	NASTRAN/ABAQUS	120	120

3.2.2.3 Stimulation points and local coordinates

The BAHA was approximated as a point to simplify the model and to easily measure the velocity response mentioned in section 1.4.

Since mechanical point impedance is defined as the ratio of the excitation force F to the velocity response v at the same position, the stimulation position would also be the same as the measurement position. In this study, the stimulation direction is normally incident upon the surface. The rectangular coordinate was also assigned to the stimulation point, and its location is described in Table 3.5.

Table 3.5: The vector components and node number of stimulation position (same stimulation and measurement position) for head simulator.

	Node	x	y	z
Stimulation point	2119	-0.40	278.33	-91.00

3.2.2.4 Boundary condition

Boundary condition represents the interaction of the parts in the model and parts that are not included in the current model. In this model, the displacements at the bottom side of the head were fixed to prevent rigid body motion.

Assigning this boundary condition will be more realistic for the situation of head simulator model where the head part is bound to the ground.

3.2.2.5 Mesh convergence

In FE modelling, a smaller or finer mesh produces a more accurate result. However, the finer the mesh, the longer the computing time. In mesh convergence, one can perform calculations and improve simulation accuracy and computing efficiency. The model was re-simulated with three different element sizes(2mm,3mm,7mm) analyzed at the same point as that in the original simulation.

3.2.2.6 Finalized FE model

The total mass of Head simulator is listed in table 3.6.

Table 3.6: Mass specification for Head Simulator model.

Head Simulator	
Total mass (kg)	Z_1

3.3 Formulation of human head model

3.3.1 Medical image segmentation

Medical imaging can be performed using various methods such as CT (computed tomography), MRI (magnetic resonance imaging), PET (positron emission tomography), and ultrasonography. In this study, the CT scan of an adult male was used to reconstruct the geometry of a human head model. Image segmentation of the CT scan is a key step in analyzing medical images for computer-aided engineering.

Segmentation is the process of dividing an image into several segments with similar properties such as brightness, gray level, color, texture, and contrast. In other words, image segmentation assigns a label to every pixel in an image so that pixels with the same label share specific characteristics. Most technical applications including automated segmentation algorithms can produce desirable results; however, this can be difficult for medical images due to the complex human body structure, deformation, and non-rigid characteristics of organs [36]. Therefore, in order to capture small and complex anatomical structures, precise segmentation is essential. Simultaneously, computational time should also be considered in the overall simulation process.

In order to accurately and rapidly reconstruct the 3D tissue-skeletal geometry of the head, RETOMO was used in this study. RETOMO is a software recently developed by BETA CAE Systems (This software was used before its official release date due to partnership between Alten Sweden AB and BETA CAE Nordic) which enables 3D geometry reconstruction from a series of tomography datasets such as DICOM files and automated segmentation algorithms. Subsequently, the resulting FE model would be ready for pre-processing in ANSA [37].

Figure 3.2 describes the overall process in RETOMO. Original CT scans (460 DICOM files) were imported. All three planes (sagittal, coronal, and axial) were displayed in the main interface. The number of materials in this setting was three including air (denoted in green). The yellow region represents the bone, and the red region represents the area except bone (skin and soft tissue), as highlighted in figure 3.3.



Figure 3.2: Process of image segmentation and 3D reconstruction in RETOMO.

Figure 3.3 shows the user interface after importing the DICOM files to RETOMO, including histograms (red box), to tune the threshold of the image.

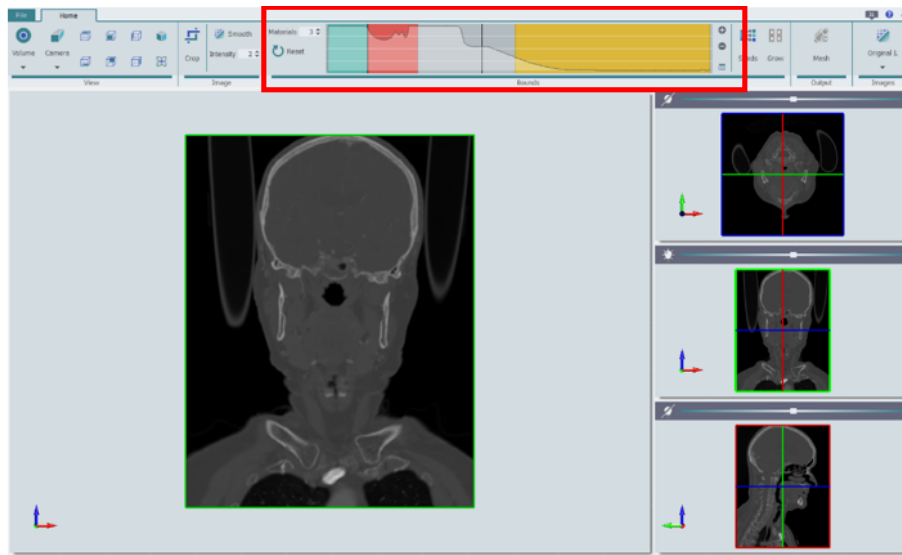


Figure 3.3: RETOMO User interface.

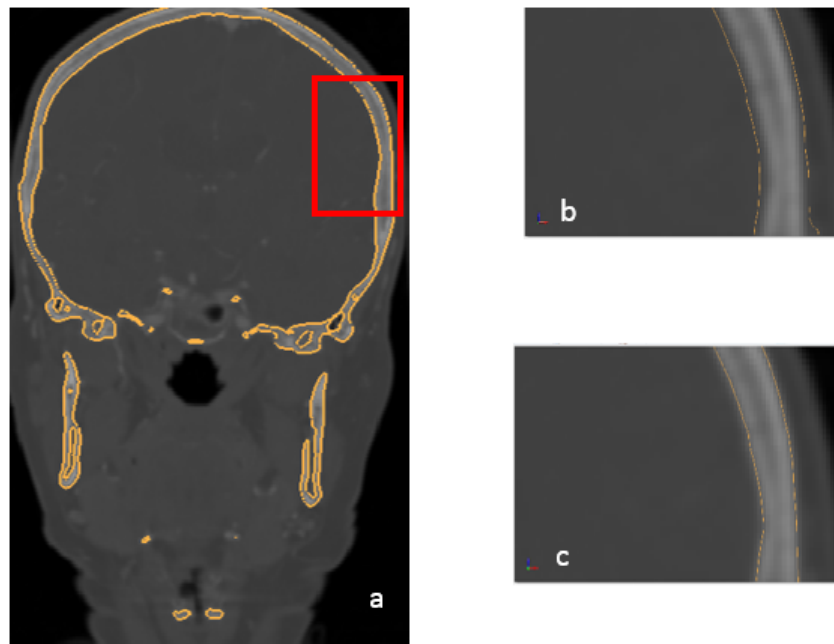


Figure 3.4: Setting boundaries with threshold. (a) Coronal plane of head CT. (b) Threshold bar toward the right (higher intensity). (c) Threshold bar toward the left (lower intensity).

By adjusting the black bar in the colored region in the upper part, the boundaries between air and skin as well as skin and bone can be set. After placing the threshold bar, the histogram was separated into several ranges (green, red, and yellow) according to the material.

3. Methods

In each range, the user can expand or shrink the range of the color applying the ‘Seeds’ function followed by ‘Growing’.

The purpose of seeding and growing is to label the voxel according to its material. In order to avoid extra layers of materials that do not actually exist in the model, the seeding and growing approach is used. With this method, voxels at the interface can be avoided by assigning different materials with the same grey value.

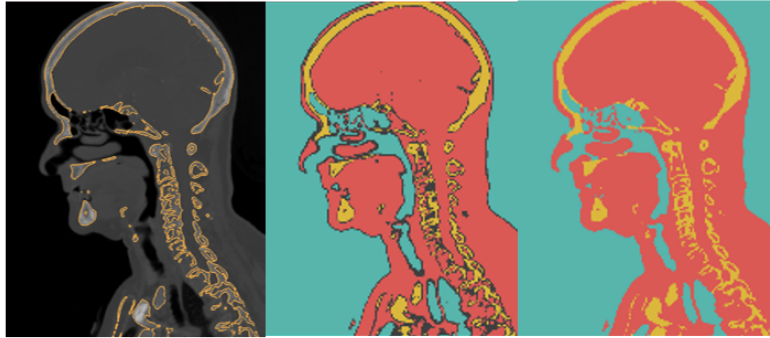


Figure 3.5: After placement of threshold, seeding, growing process (from left to right).

After labeling all the voxels, a 3D surface mesh was generated by pressing the ‘Mesh’ button on the upper right part. The ‘Extremely Low’ level is highly recommended for smoothing and simplification before increasing to higher levels because of the long processing time. The mesh generated in RETOMO was a very rough mesh; thus, the final mesh generation step with the correct mesh criteria was performed in the pre-processor ANSA.

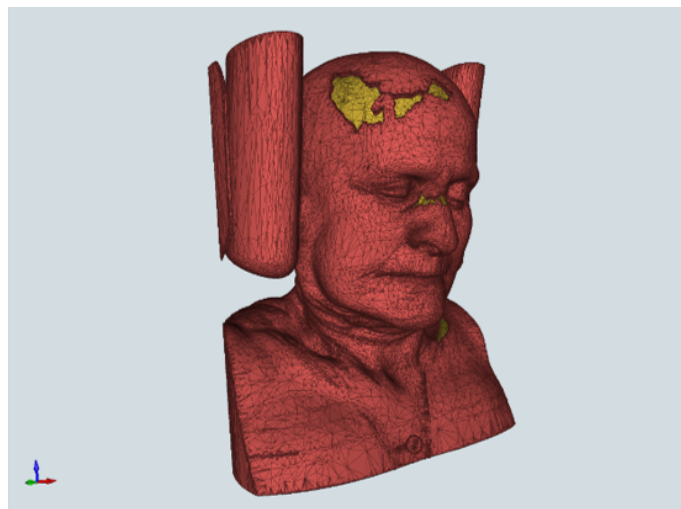


Figure 3.6: Rough mesh ready for ANSA.

3.3.2 Construction of three dimensional FE model

In the second step, a proper mesh was built on the geometry or 3D rough shell mesh that was formulated in the first step mentioned in FE process in section 3.1. The pre-processor ANSA was used to create the mesh and add boundary conditions, local coordinates for the BAHA, and material properties. This step would ensure that the model is more equipped for FE model simulation with reproducible and reliable simulation results.

In this model, the brain was modeled as fluid. A brief description of the solid and fluid mesh and their simulations is given below. In these acoustic simulations, the pre-processor usually applies 3D elements with special fluid nodes, as described below.

3.3.2.1 Rough mesh

The geometry of real human head model was processed by the segmentation software RETOMO mentioned in section 3.3.1. The OBJ files (.obj) were created and opened in ANSA. Since the CT scan was taken with the equipment in the CT machine and the clothes worn by the patient, the recreated model also included parts of them next to the ears and other parts of body, which are unnecessary as shown in figure 3.7. The aim of this step is to remove the redundant parts in the model for optimal simulation, to reduce computational time, and to apply the mesh without drastically changing the original model. Geometry preparation is the quintessential step before the meshing step; hence, a balance between the elimination of parts and preserving the original geometry will be crucial in this process. To achieve this and to create different parts separately, the ANSA function ‘MESH-Elements-Wrap’ was used. In addition, to create the head model, several elements below the head were deleted in the process. The final model had three parts: the skin, bone, and brain.

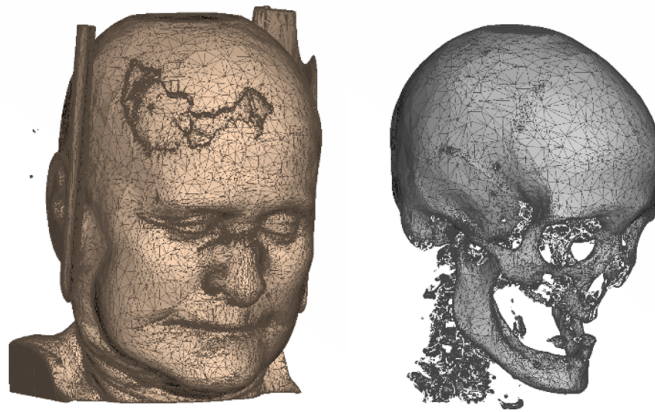


Figure 3.7: Raw anatomical data of skin and skull obtained from RETOMO to ANSA.

3.3.2.2 Material properties

One of the important parts of the simulation with FE models is to assign the right material properties to the model. The skin and bone used in the solid domain were defined as MAT1. Young's modulus, density, and Poisson's ratio were assigned to the solid domain. The brain was modeled as fluid and defined by MAT 10 in the Material Database.

In addition, all grid points in the fluid model had a CD of -1 with PFLUID property. CD represents the identification number of the coordinate system where the displacements, degree of freedom, constraints, and solution vectors are defined at the grid point.

Table 3.7 shows the material properties and fluid part using the bulk modulus instead of Young's modulus.

Table 3.7: Material parameters for skin, skull and brain

Parts applied	Material	Young's modulus (MPa)	Bulk modulus (MPa)	Shear modulus (MPa)	Density (ton/mm ³)	Poisson's ratio
Bone	MATERIAL 1	Y_1			Y_4	Y_7
Brain	MATERIAL 2		Y_2		Y_5	
Skin	MATERIAL 3			Y_3	Y_6	Y_8

The skull bone was modeled as a homogeneous material for simplification while typical bone structure has three layers including strong and dense cortical bone and spongy cancellous bone.

3.3.2.3 Mesh (batch mesh)

The properties are determined in the same way as the head simulator model above except the few changes in quality criteria.

Table 3.8: Mesh parameters for starting shell mesh before the volume mesh.

Mesh Parameters	Value
Mesh Type	<i>General</i>
Element Type	<i>Tria</i>
Element order	<i>1st</i>
Target length	4 mm
Minimum target length	1 mm
Maximum target length	6 mm
Distortion distance	20 %
Distortion angle	0

Table 3.9: Quality criteria for shell and solid elements in human head model.

Criteria	Calculation	Shell mesh	Solid mesh
Aspect ratio	NASTRAN	3	8
Skewness	NASTRAN/ABAQUS	45	0.1
Warping	NASTRAN	10	10
Taper	NASTRAN	0.35	
Jacobian	NASTRAN	0.7	0.7
Minimum angle trias/tetras	NASTRAN/ABAQUS	30	20
Maximum angle trias/tetras	NASTRAN/ABAQUS	120	127

3.3.2.4 Stimulation points and local coordinates

All measurements of the mechanical point impedance of the skull were performed in the BAHA position, which is 55 mm behind the opening of the ear canal at the parietal bone which was mentioned in subsection 2.1.4. In addition, the BAHA was also approximated as a grid point to simplify the model and to easily measure the velocity response mentioned in section 1.4.

Since mechanical point impedance is defined as the ratio of the excitation force (F) to the velocity response (v) at the same position, the stimulation position would also be the same as the measurement position. In this study, the BAHA was approximated as a point source normally incident upon the surface of the skull bone. The rectangular coordinate was also assigned to the BAHA point, and its location is described in table 3.9.

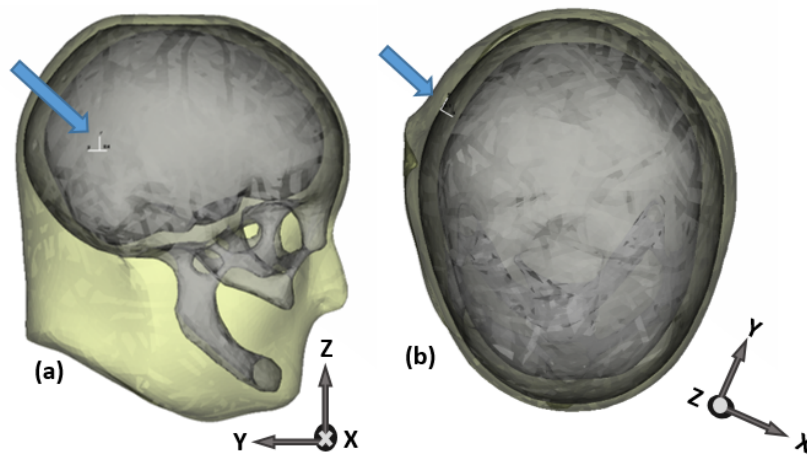


Figure 3.8: (a) The BAHA point from right side view (b) The BAHA point from top view. Blue arrows indicate BAHA position.

The skull surface is curved rather than flat, the input force was designed to align normal to the skull surface.

Table 3.10: The vector components and node number of BAHA position (same stimulation and measurement position).

	Node	x	y	z
BAHA point	334668	-52.58	111.11	-611.27

3.3.2.5 Boundary condition

Boundary condition represents the interaction of the parts in the model and parts that are not included in the current model. In this model, the displacements at the bottom side of the head were fixed to the ground in order to prevent rigid body motion.

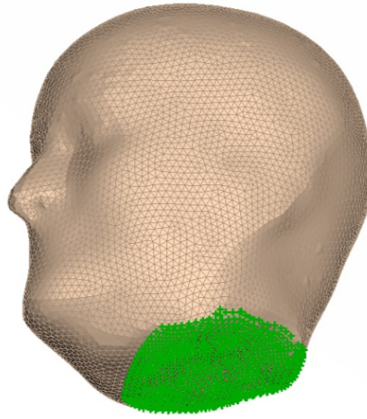


Figure 3.9: Constrained neck part with SPC in green points.

Moreover, it would be more realistic to prevent rigid body motion since test subjects were alive and not decapitated. This was performed by single point constraints (SPCs) to the grid points where the constraint was applied. This function assigns SPCs of zero displacement to selected grid points.

3.3.2.6 Mesh convergence

Same mesh convergence study was performed to increase the accuracy of human FE head model as well. The model was re-simulated with two different element sizes (2mm, 3mm) for real head model for both NSM and Fluid-structure model and analyzed at the same point as that in the original simulation.

3.3.2.7 Finalized FE model

Figure 3.10 illustrates FE human head model formulated in ANSA. The tables below shows the mass of the FE human head model and number of elements in the model.

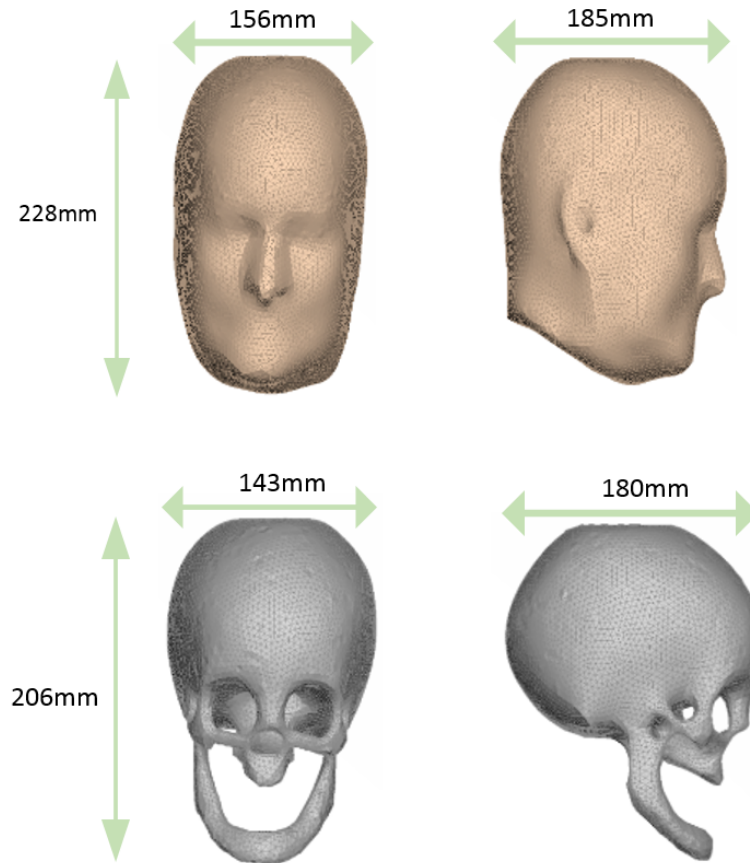


Figure 3.10: The dimensions of the whole head for skin and skull part after completing FE model.

Table 3.11: Mass specification for Human head model.

Real head model	
Total mass (kg)	Z_2

Table 3.12: Number of elements and grid points for the FE model.

	Number of elements	Number of grid points
Skin	240215	49656
Skull	100034	14472
Brain	86779	16792

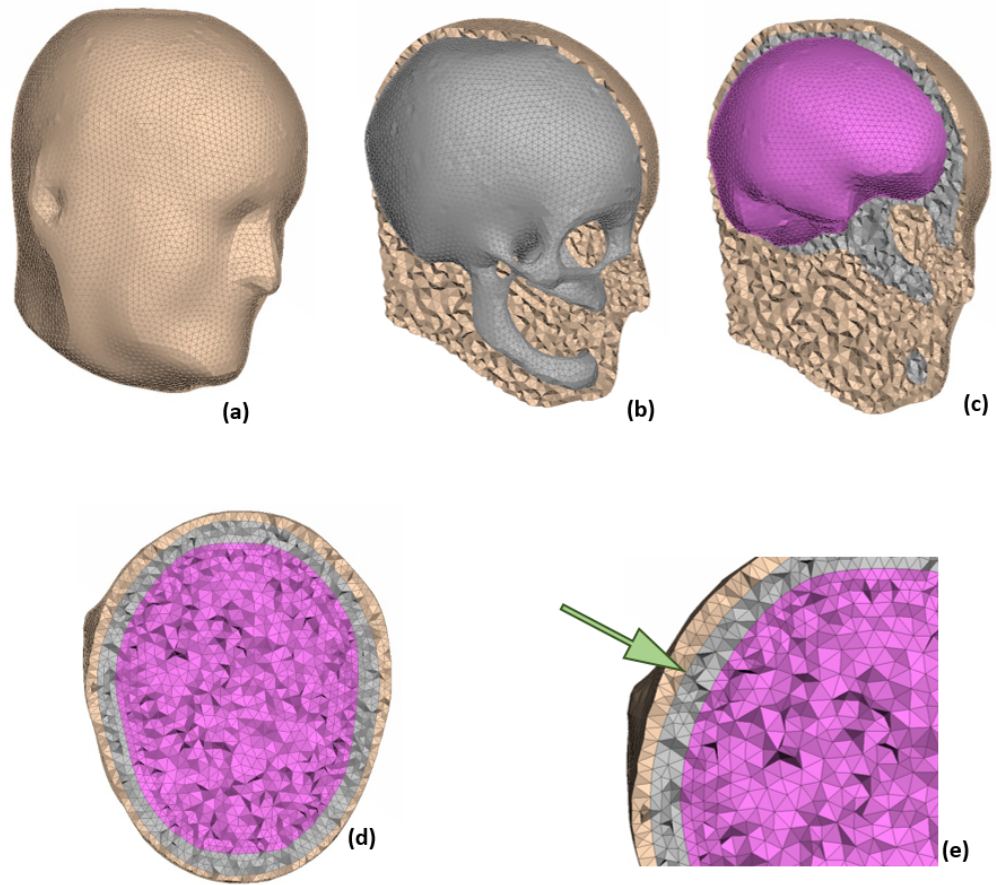


Figure 3.11: Components of the FE head model and connection in oblique view. All in tetrahedral four-noded mesh. (a) Skin part of the head (b) Skull bone and part of skin (c) Brain, skull bone and skin parts of the head (d) The cross sectional top view of the whole head model (e) Skin, skull and brain connected by grid points from BAHA position (arrow).

3.4 Dynamic simulation setup in Nastran

Mechanical point impedance (MPI) or driving point mechanical impedance measurements were used to determine the dynamic response of the Head Simulator head and human head to chirp signal with sweep from 1 Hz to 10,000 Hz. Application in Nastran was done at both models.

The dynamic frequency response was simulated using a commercially available structural analysis software, MSC Nastran. For this model, loads, frequency range, analysis output, and damping coefficients were assigned in the form of Nastran codes. Two types of analysis were performed in this study: normal mode and frequency response analysis [31].

In general, the human hearing frequency range ranges from 20–20,000 Hz. In this case, the frequency range of interest was set to 0–10,000 Hz based on previous studies on bone conduction. However, some results do not have a lower frequency range because of the lack of test data.

3.4.1 Normal mode analysis

Usually, when performing any structural analysis with vibration, identification of the natural frequencies or eigenfrequencies and mode shapes are important [31]. The Lanczos method was used to calculate eigenvalues. This method can be applied by using EIGRL entry in Nastran. In this analysis, the frequency range was set higher than the usual frequency range in order to capture the modes in the higher frequency. The calculation of normal modes was performed by SOL 103. This step is the basis for dynamic modal frequency response analysis, which will be discussed in section 3.4.2.

3.4.2 Modal frequency response analysis

Frequency response analysis is a method to compute the structural response to oscillatory excitation, which, in this case, involves the BAHA device using SOL 111 in Nastran.

$$P(f) = A[C(f) + iD(f)] \cdot e^{i(\theta - 2\pi f\tau)}$$

The force was defined by setting ‘Load’ at the BAHA position. The input force was defined by the DAREA and RLOAD1 command. RLOAD 1 defines a frequency-dependent dynamic load, as shown above [31], and the input force was set to a unit force of 1 N. In this analysis, two kinds of methods were applied to the current model: nonstructural mass (NSM) model and fluid structure model.

The NSM model only takes mass of the brain into account instead of modeling it as a structure. The total mass of the brain would be distributed to the grid points of the skull. This was performed by the CONMi2 function in ANSA. In the other model (the fluid structure model), the brain was modeled with fluid elements, and this was performed by the aforementioned PFLUID and assigning CD as -1 in ANSA. Moreover, there was an additional ACMODL Fluid-Structure Interface Parameters entry in Nastran to couple the structure mesh and fluid mesh. This would define the parameters for the fluid-structure interface.

Several different damping coefficients for both structural and fluid models were also investigated by simulation. Different damping coefficient from literature were also investigated. This modal damping was defined by SDAMPING followed by TABDMP1 command in Nastran.

Table 3.13: Different damping coefficients settings.

	Structural modal damping	Fluid modal damping
Original NSM and FS model	0.16	$1.6 \cdot 10^{-4} \cdot f$
Case 1: Chang <i>et al.</i> [38]	0.01	$1 \cdot 10^{-4} \cdot f$
Case 2: Bernard <i>et al.</i> [39], Mrozek <i>et al.</i> [40]	0.03958	0.3

4

Results and Discussion

The mechanical point impedance (MPI) of the head was calculated and compared with the test results from Cochlear. Additionally, a parametric study was performed by varying the values found in previous studies for Fluid Structure (FS) models. All the measurement outputs were taken perpendicular to the surface instead of from all three directions. The stimulation direction was toward the center of mass and the test data was average of all four points. The mesh size was 5 mm for Head simulator results.

4.1 Result of Head simulator model

4.1.1 Nonstructural mass (NSM) model

The response of the skull surface of the surrogate was simulated and the output was the velocity for mechanical point impedance. This was done by post-processor META and compared with the test data.

4.1.1.1 Initial mechanical point impedance result

Figure 4.1 illustrates the results of MPI magnitude calculation with the test data. The peak which represents the forced antiresonance was 75 Hz in this model, as opposed to 70.56 Hz in the test results. The overall trend shows the similarity between the simulation results and test data with a maximum 7 dB difference in the higher frequency domain (3,000 Hz – 10,000 Hz), as in the previous study [19]. The simulation shows multiple resonance frequencies at a higher frequency level, but due to excessive damping in the test subject, the test data did not represent all of the resonance frequencies or fluctuations.

At frequencies below 3,000 Hz, the magnitude of the calculated MPI has good agreement with the test data. The magnitudes of the experimental data were within a 7 dB difference, and typically fell between 3 – 4 dB, which is also in agreement with the previous study [19].

The fluctuations between the frequencies of 400 – 4,000 Hz indicated that there were several resonance frequencies within this range, compared with previous study, which only showed their presence from 150 – 400 Hz.

Before the peak at 75 Hz, the magnitude of the MPI from the calculation increased with frequency, indicating that the mass dominating area starts to decrease after 75 Hz magnitude, which also indicates that the stiffness mainly affects the impedance at those frequencies.

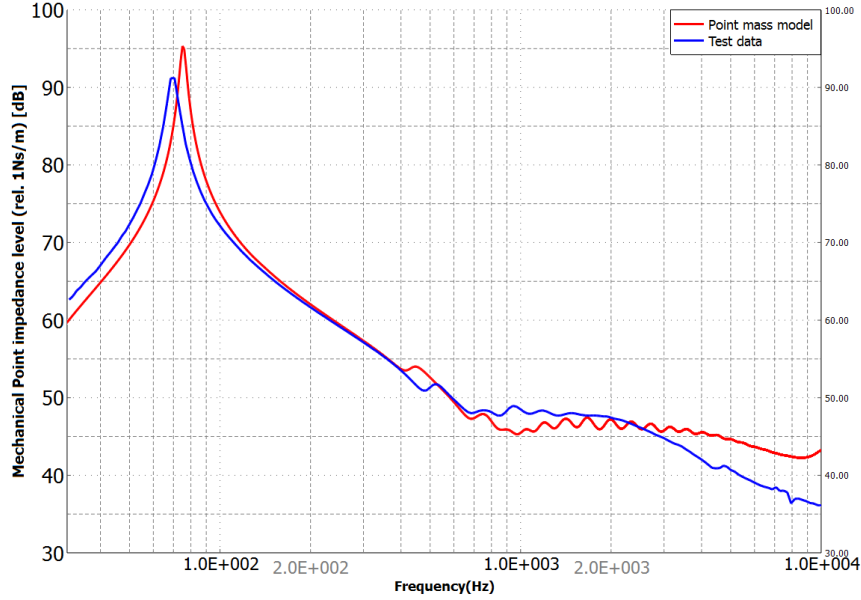


Figure 4.1: Mechanical point impedance magnitude at BAHA position in NSM model. The red line represents the simulation result with NSM model and the blue line indicates the experimental data from Cochlear.

Table 4.1: Resonance frequency comparison with the simulation and experimental data in NSM model. The first two modes are not presented due to the lack of test data. Units in Hz.

Mode	Simulation	Experiment	Difference (%)
1	22.51	x	x
2	22.93	x	x
5	453.16	472.2	4.0
7	696.26	686.6	1.4
12	852.76	840.0	1.5
27	1049.13	1058.0	0.8
52	1333.97	1332.0	0.2

Table 4.1 describes the resonance frequency found in both the simulations and the test data by comparing the ‘valleys’ in figure 4.1. Each peak represents antiresonance and each valley represents resonance in the MPI graph. The frequency range in the experimental data starts with 31.5 Hz, so the first two resonance frequencies could not be compared. Moreover, the trend near the highest frequency shows that MPI is on the rise compared to that of the test data. In addition, less resonance is observed from 3,000 Hz and above.

4.1.1.2 Mesh convergence

Mesh convergence was evaluated by changing the length of the element in both the NSM and the fluid structure (FS) model. The length of the element was changed to 2 mm, 3 mm, and 7 mm. For the NSM model in Figure 4.2, the maximum deviation for MPI was 1.1 dB, and the maximum frequency shift was 1 Hz. In the case of the FS model, the maximum difference of the MPI level was found to be 3 dB and 4 Hz in frequency.

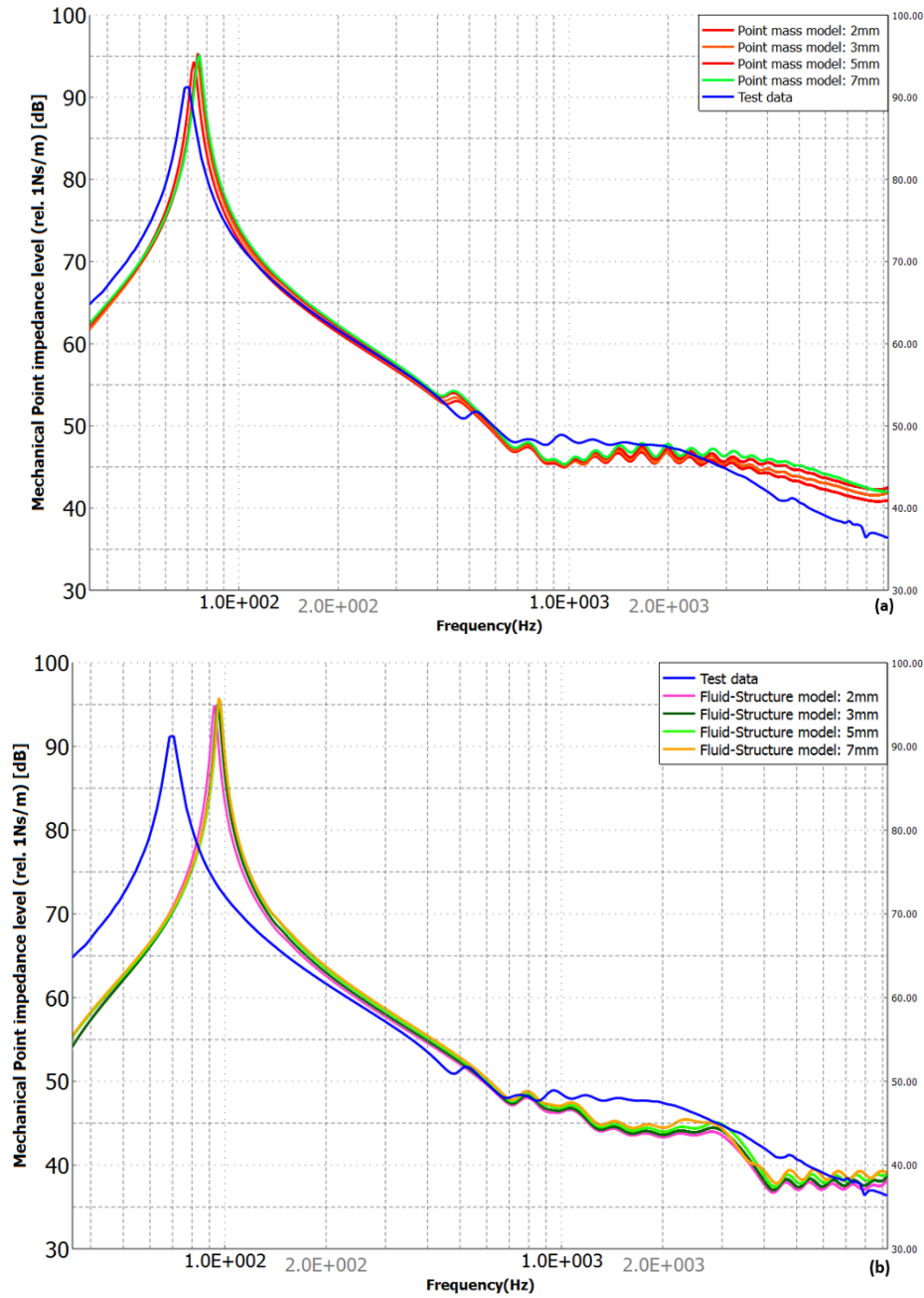


Figure 4.2: Mesh convergence results for NSM model (a) and FS model (b).

4.1.2 Fluid – Structure (FS) model

4.1.2.1 Initial mechanical point impedance result

The same simulation method was applied as in the previous NSM model, and Figure 4.3 shows the result of calculating the magnitude of the MPI compared to the test data. The overall trend resembles that of the test data, except that the antiresonance frequency is present at 97 Hz, which is a 27 Hz difference from the test data.

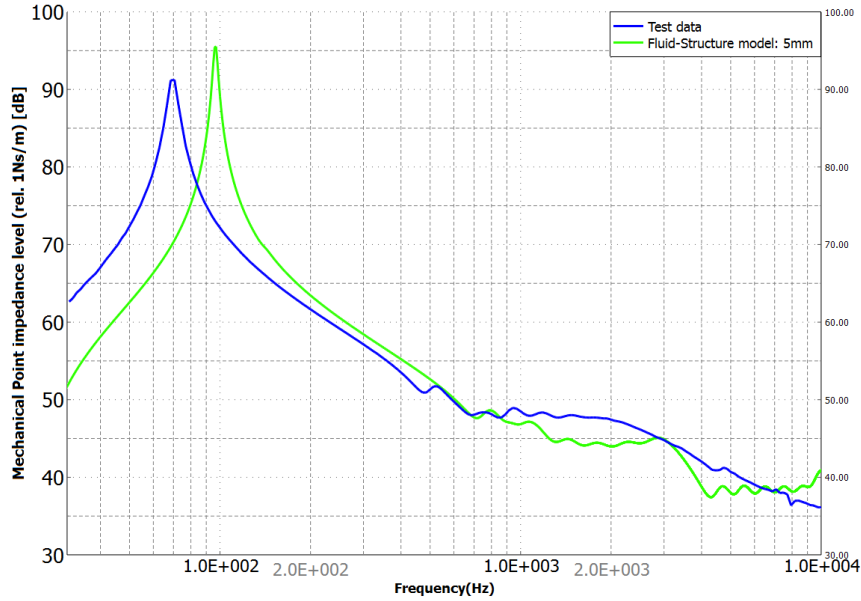


Figure 4.3: Mechanical point impedance level in FS model.

Similar to the NSM model, the simulation results and the test data have a maximum magnitude of a 5 dB difference in the higher frequency domain (3000 Hz – 10,000 Hz), similar to the previous study [19]. As it was stated in the previous NSM result, the simulation in this model also shows multiple resonance frequencies at a higher frequency level, and it can also be observed that there is an oblique line facing down in the frequency range from 3000 – 4,000 Hz. This sudden drop is mainly from the fluid structure interacting with the structure model.

At frequencies after 200 Hz, the magnitude of the calculated MPI has good agreement with the test data. However, there is an outlier in the lower frequencies due to the difference in the occurrence of the antiresonance frequency.

Several fluctuations were also present in the FS model to a lesser degree than in the NSM model. There are several fluctuations in the frequency domain from 800 Hz–3,000 Hz and from 4,000 Hz – 10,000 Hz. Moreover, the trend of the rising magnitude where the curve spikes up near the highest frequency is also shown in the FS model, as in the NSM model.

One possible explanation for the sudden drop in MPI magnitude is that the resonance frequencies of the fluid part are present beginning at 5,000 Hz and higher.

Table 4.2 shows the combined eigenfrequencies of the fluid and structure model, which are affecting the participation of fluid modal frequencies. Table 4.3 describes the resonance frequency only in fluid domain.

Table 4.2: Resonance frequencies in FS model compared with test data (Structure part).

Mode	Simulation	Experiment	Difference (%)
1	43.47	x	x
2	43.77	x	x
4	314.40	472.2	33.4
5	1010.79	686.6	47.2
6	1011.41	840.0	20.4
7	1647.38	1058.0	55.7
8	1647.39	1332.0	23.7

Figure 4.4 illustrates the NSM, FS model and the test data together. The difference in the antiresonance frequency of the two simulation models is 22 Hz. In both models, the frequency from 200 Hz to 1,000 Hz has a great resemblance to the test data. In the NSM model, the majority of the resemblance happens below a frequency of 3,000 Hz, and the FS model captures the magnitude of MPI more accurately in the frequencies above 400 Hz.

Table 4.3: Resonance frequencies in FS model compared with test data (Fluid part). RBM stands for Rigid Body Motion.

Mode	Simulation (Hz)
1	RBM
2	5719.99
3	5720.09
4	5725.72
5	9191.04
6	9191.47
7	9194.77
8	9194.83

Both of the models show multiple resonance frequencies in a higher frequency domain, primarily due to a higher modal density than in previous studies, which have shown multiple resonances before the peak of the mechanical point impedance magnitude level curve. The study by Chang *et al.*[19] states that the resonances at a lower frequency are due to rotational modes in addition to the translational motion of the head. In this model, the stimulation point was in line with the center of the mass and perpendicular to the surface of the skull. In addition, the model is constrained at the bottom part such that the effect of the rotational model is minimal. However, the origin of these multiple resonances needs further investigation.

4. Results and Discussion

Even though the test data does not show the very first resonance frequency, it is highly recommended that the first resonance be investigated due to its contribution to the overall structural response to the external frequency load. Figure 4.5 shows the first resonance frequency. The experimental data was left out due to the lack of data.

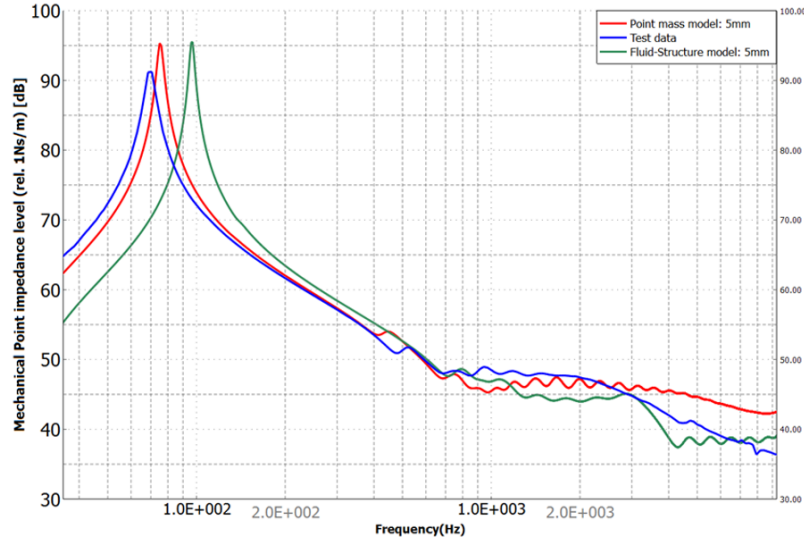


Figure 4.4: Magnitude of mechanical point impedance from NSM, FS model and test data (Point mass indicates NSM model) frequency from 30 Hz to 10,000 Hz.

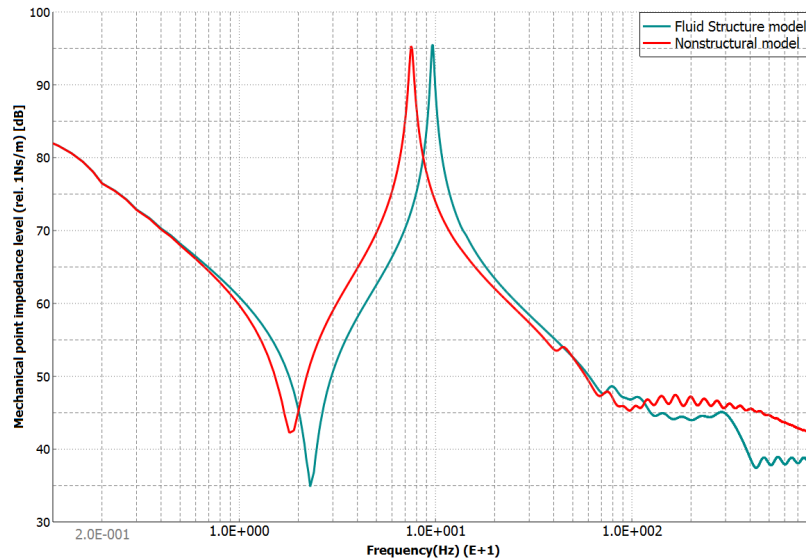


Figure 4.5: Comparison of magnitude of mechanical point impedance of NSM, FS model in whole frequency range from 1 to 10,000 Hz.

4.1.2.2 Parameter study

In order to better fit the test data and to find mechanical factors influencing the simulation results, several parameters were investigated, such as Young's modulus of structural part, and the bulk modulus of the fluid part.

Figures 4.6 and 4.7 present the influence of Young's modulus and bulk modulus on the FS model. These parameters were both increased and decreased by 20 %. In the case of the bulk modulus, the system remained the same until reaching a frequency of 1,000 Hz, at which point the high bulk modulus underwent a frequency shift of 10 Hz compared to the original FS curve. The sudden dip in the higher frequency range started to present earlier and had a longer 'dip' in this frequency domain. The lower bulk modulus had a shorter dip range in the MPI curve and manifested its dip at a later frequency than the original FS model by 650 Hz (16 %). Both the lower and higher bulk modulus resulted in a similar trend, especially in the higher frequency range, and had almost no influence in frequency below 1,000 Hz, while having no influence on the impedance level at the same time.

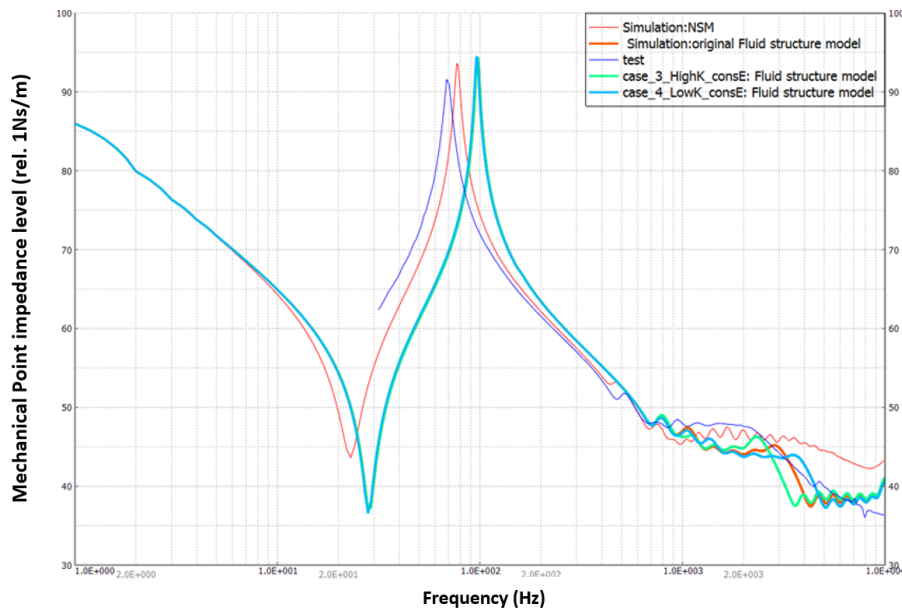


Figure 4.6: Effect of bulk modulus of fluid under constant Young's modulus of structure part. Comparison with original NSM, FS and test data.

Similarly, figure 4.7 shows that increasing and decreasing Young's modulus had a great influence, shifting whole curve by 10 Hz with respect to the original FS result. A higher Young's modulus led to a shift to the right, and a lower Young's modulus led to a shift to the left. The curves kept their original shape, and the changes in magnitude levels were minimal compared to the result of the unmodified original FS model.

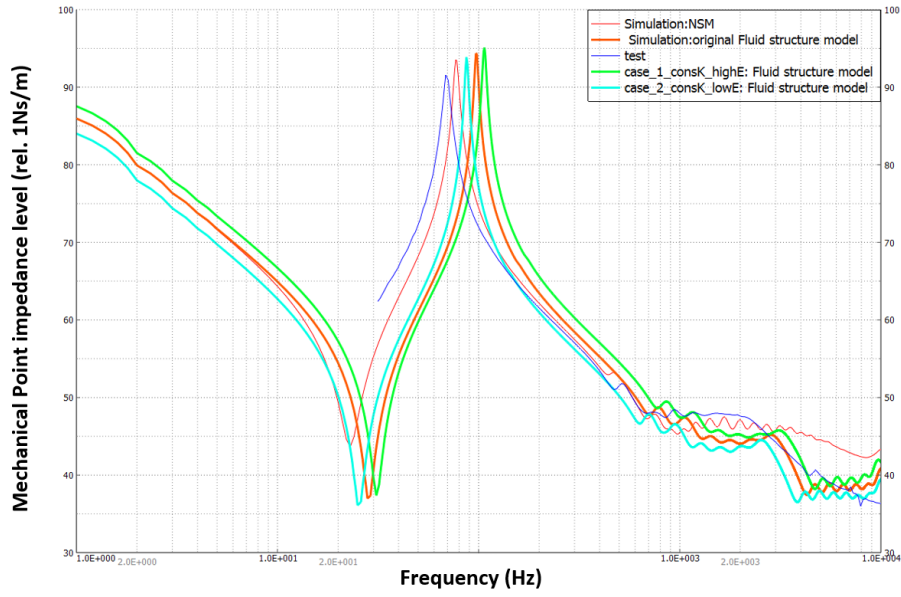


Figure 4.7: Effect of Young's modulus of structure part under same Bulk modulus of fluid part.

Figure 4.8 shows how the magnitude level changed between the mechanical point impedance in the FS model under normal circumstances and when varying Young's modulus in the skull and the bulk modulus in the fluid model at the same time. The changes were made by increasing and decreasing the value twofold.

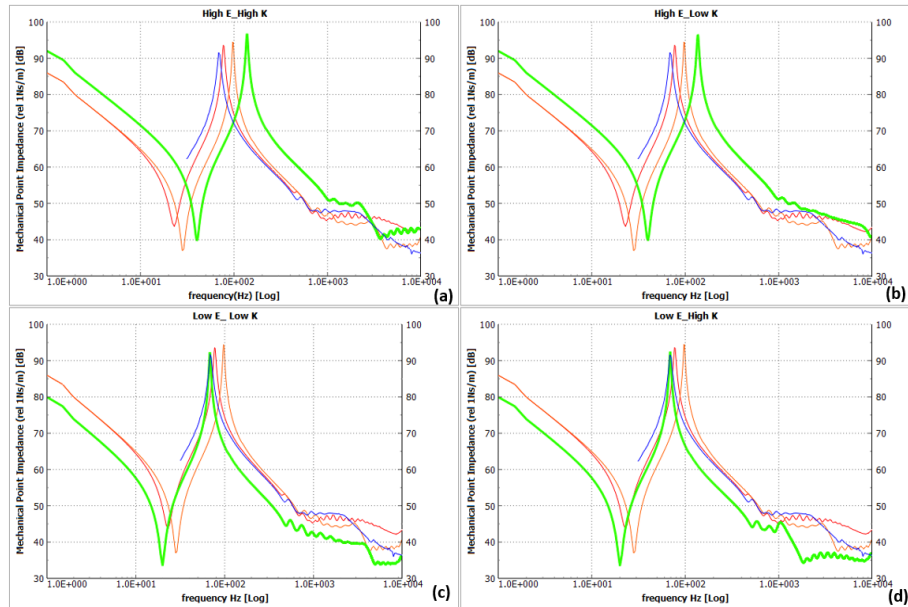


Figure 4.8: Parameter sensitivity in FS model (in green) compared with original NSM model (in red), original FS model (in orange), and test data (in blue). All the parameters were modified by half and double the original value. (a) MPI result: High Young's modulus in structure part with high Bulk modulus in fluid part. (b) MPI result: High Young's modulus in structure part with high Bulk modulus in fluid part.

The results shown from figures 4.6 – 4.8 conclude that Young’s modulus definitely shifts the whole curve to the right and left without changing the magnitude level of the MPI, while the bulk modulus plays an important role at higher frequencies only, leaving other frequency ranges unchanged.

Figures 4.6 – 4.8 did not include the effect of Poisson’s ratio in the structure part since Poisson’s ratio could not be defined in the fluid part. Figure 4.9 illustrates the effect of Poisson’s ratio on FS model. A decrease in Poisson’s ratio to 0.25 from 0.469 shifted the resonance frequency by a maximum of 250 Hz (6 %) and the magnitude level by a maximum of 0.5. Therefore, it can be concluded that changing Poisson’s ratio in the structure part has only a minor effect in the simulation of MPI.

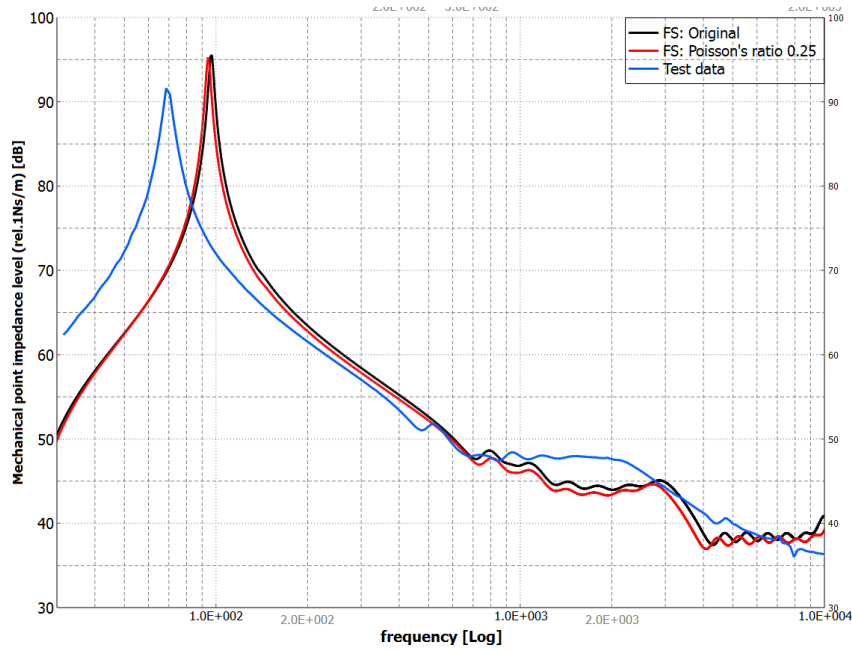


Figure 4.9: Comparison between original FS model, test data for Poisson’s ratio.

4.1.2.3 Damping coefficient

One of the factors of vibration analysis is the control of the damping coefficients. Here, unlike in a conventional material like steel, the materials used in this model have a very high damping effect due to a lack of observed fluctuations in the high frequency domain throughout the result. In order to investigate its effect, damping coefficients from different sets of articles were used. Their values are listed in Table 3.8 in section 3.4.2. It can be observed that in all cases, damping does not influence resonance frequency, which is in accordance with the general frequency relationship.

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}}$$

Moreover, Case 2, which has a higher damping coefficient than Case 1, has a lower damping effect in general in the frequency range 700 – 4,000 Hz. Case 1 also has frequency dependent fluid damping, which starts to show an effect from 4,000 Hz onwards corresponding to the fluid resonance frequency effects in this range. These findings indicate that it is highly recommended to use higher damping coefficient at frequency ranges beginning at 1,000 Hz. However, at the same time, these should be low enough to catch the resonance frequency ‘fluctuations’ in dynamic simulation.

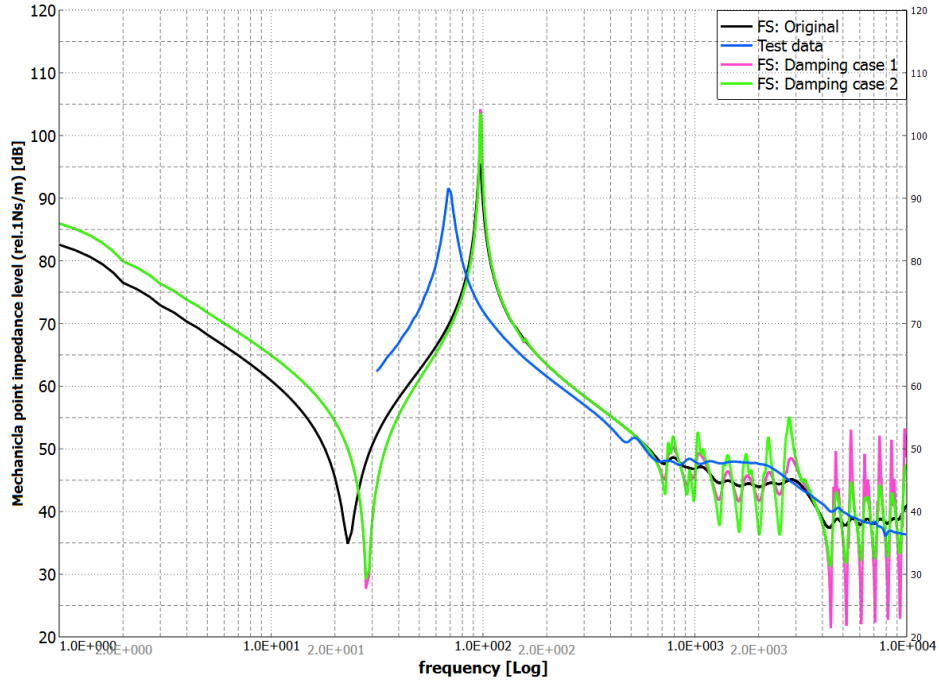


Figure 4.10: Comparison between original FS model, test data for Poisson’s ratio.

4.1.2.4 Organization of constraint

In order to recreate the environment where Head Simulator was placed, both models had to be constrained. Figure 4.11 shows that, when compared to the models without proper constraint, models with an SPC-assigned boundary condition showed a better fit to the test data.

This boundary condition did affect the occurrence of the first resonance shift in the whole range of resonance frequencies as well as the magnitude change. In this section, the results are shown with the whole frequency range, which is 1 – 10,000 Hz, and presented in Table 4.4. The FS model without constraint had a higher antiresonance frequency by 43.44 Hz than the test data, and was still higher by 17 Hz in the FS model with constraint.

In the same manner, the NSM model without constraint had higher antiresonance by 36.44 Hz with respect to the test data, and was still 32 Hz higher in the NSM with constraint. In the case of magnitude level, the FS model without constraint was 4.15 dB higher than the one with and 8.44 dB higher than the test data. The NSM

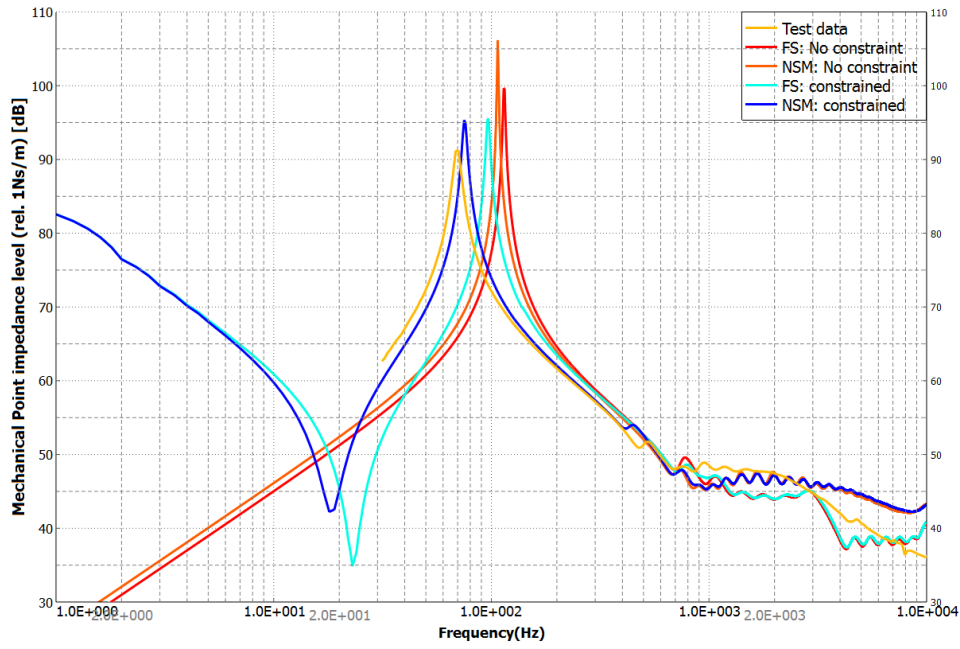


Figure 4.11: MPI in different settings of constraints.

model with SPC was 10.88 dB lower than the one without and 14.97 dB than the test data. The measurement results indicate that the FS model had less sensitivity than the NSM model when it came to constraint, but both models were affected by the assignment of constraint.

Table 4.4: Measurement of MPI in different models with and without constraint. Percentage next to each cell indicate deviation from the test data.

	FS with constraint	FS without constraint	NSM with constraint	NSM without constraint	Test data
Antiresonance Frequency (Hz)	97 (37.5 %)	114 (61.5 %)	75 (6.3 %)	107 (51.6 %)	70.56
MPI Magnitude (dB)	95.52 (4.7 %)	99.67 (9.3 %)	95.32 (4.5 %)	106.2 (16.4 %)	91.23

It can be concluded that assigning proper constraint can result in changes in resonance frequency and MPI magnitude. In addition, constraint mainly influences frequency domains lower than 200 Hz. Previous research shows that the head-neck junction is not significant for frequencies above 400 Hz, as indicated by these results, where the overall trend above 200 Hz was similar to that of the test data. In addition, modelling the whole body for BAHA would increase the computational cost and would demand more computational power, leading to an overly complicated system.

4.2 Result of Anatomical head with Fluid Structure model applied

The head geometry was based on a public CT scan and the mechanical properties were replaced with other non-biological materials. Therefore, the simulation results are not expected to be similar to the test data, which was the average of 43 patients with BAHA. However, it did show qualitative similarity with the experimental data, which presented an antiresonance peak around 150 – 200 Hz corresponding to the 147 Hz in the test data and the 145 Hz in the literature (Håkansson et al, 1986). These results were also shown with and without boundary conditions. The test data had a maximum and a minimum based on the data points recovered from the literature [41].

Figure 4.12 illustrates an MPI simulation of the real head finite element model compared to the clinical data from the literature [41] and investigates the effect of constraint in a similar manner to the Head Simulator head model. Similar to the

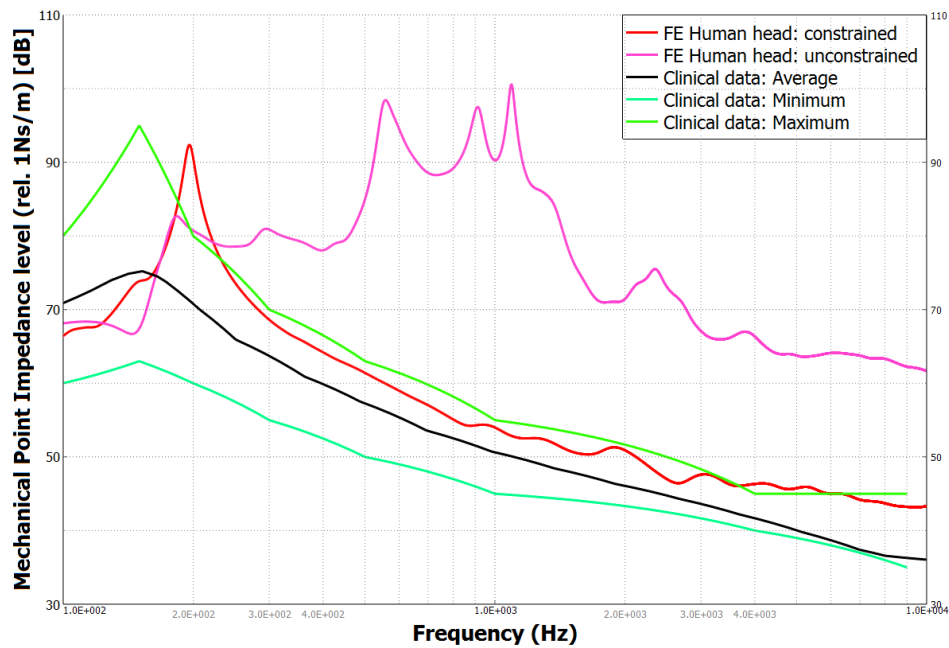


Figure 4.12: MPI measurement in actual human Finite element head model with clinical data.

Head Simulator model, proper constraint around the neck area showed a similar trend to the test data with a 196 Hz antiresonance frequency compared to 147 Hz from the test data. The magnitude level differences are within 4 dB when comparing the FS model with the highest MPI clinical data after a frequency of 200 Hz. However, there were some inconsistencies in the magnitude in the lower frequency area.

One way to explain the differences between the simulation and the clinical test data is that the FE human head model was based on the CT scan of one person, whereas the clinical data was the average of 43 patients with BAHA device. Therefore, the geometry difference in individuals may have contributed to this difference.

Moreover, the material properties assigned to this head model are not consistent with actual skin, bone, and brain, which may explain these differences as well.

Lastly, it can be also observed that the boundary condition or constraint in the head neck junction also played an important role in human FE model as well as in FE head simulator. The result shows closer to the clinical data when the model is constrained.

5

Conclusion

In this project, the FE head simulator model was developed, and the frequency response analysis of the mechanical point impedance (MPI) was performed in the model. The results were remarkably consistent with physical test data in both Non-structural mass (NSM) and Fluid-Structure (FS) models with anti-resonance frequency at approximately 75 – 90 Hz; there was only a 5 % difference in magnitude level. In addition, an anatomically correct FE human head model was successfully created from the CT scan using a new segmentation software. Its result has shown close resemblance to clinical data.

The result of the FE human head model has proven to have a geometrical effect on the simulation of MPI compared to the numerical result of the FE head simulator model with identical material properties. This result indicates the promising result that the shape of the human head has more similarities to the clinical data than the artificial head simulator. In other words, the FE human head model better represents actual clinical data than the physical head simulator.

A mechanical point impedance (MPI) simulation for bone conduction was performed in this study. The procedure to evaluate the MPI was performed using the MSC Nastran with the segmentation tool RETOMO and meshed with ANSA. There are two different types of modeling for the simulation of MPI: FE head simulator model and FE human head model. In addition, a part of the head simulator was modeled in two different manners: Non-structural mass (NSM) and Fluid-Structure (FS) model. Both methods were validated with existing experimental data. Only the FS model was incorporated in the FE human head model for the brain part.

Comparing two different modeling methods (NSM and FS) in the FE head simulator model, we observe that the implementation of the NSM model is simple, and it performs slightly better than the FS model, although the NSM model does not consider the brain as a real structure. Nevertheless, both models must be improved in a higher frequency range because of the higher modal density in that domain. Moreover, more information on the first resonance is required to further validate the model because of the changes in the first resonances in different settings in the simulation.

For the parameter study in FE head simulator model, the effects of Young's modulus, bulk modulus, Poisson's ratio, boundary conditions and damping coefficients were evaluated. The results show that Young's modulus shifts the resonance fre-

quency, and the application of proper boundary conditions has a dominant effect on both MPI magnitude and anti-resonance frequency. However, Poisson's ratio has a negligible effect on the overall result.

The FE Human head model also exhibited a similar trend with clinical data from [41]; the anti-resonance frequency was approximately 200 Hz in this model. Moreover, it was illustrated that the model was better fitted to the clinical data when the model was constrained or assigned boundary conditions at approximately the head - neck junction. Thus, the boundary condition has a significant effect in both head simulator and human head model.

This study has validated the simulation result of the FE head simulator with physical test data, and the FE human head model displays similar trends with the clinical data despite the application of foreign materials.

6

Suggestions for Future work

To create a promising and standard FE head model, there are some suggestions for further improvement of the current study as follows.

First, the boundary conditions should be further examined for both models because they strongly affect any simulation and even change the occurrence of the first resonance frequency.

For the head simulator, the validation of simulation can be improved by adding transmission properties (equivalent to the transcranial properties in the actual human head) by having stimulation points and different points for measurements. This setting can increase the accuracy of the overall head simulator results, which increases the possibility of optimization with clinical data. Moreover, for the NSM method, an investigation in the damping properties in the high-frequency area can suggest a lower magnitude difference. This improvement can also be achieved by setting frequency-dependent damping coefficients.

The investigation of the mechanical point impedance of the skull or the entire head is only the first step toward full understanding of bone conduction. The next step should include the simulation of transmission acceleration, transcranial bone conduction, or transcranial attenuation of the skull part including the skin and brain, which is similar to the aforementioned suggestion for head simulator. The transmission acceleration, which is defined as the acceleration divided by the input force measured at different positions, should also be considered to complement the MPI, which is a local phenomenon. This includes the addition of an extra shell model to the head simulator model to recreate the effect of skin.

It will also be interesting to add the parts that were excluded during the creation of the initial FE head simulator model. Moreover, the head simulator can be further optimized with the clinical data, so there is a chance to recreate standard clinical data in the head simulator FE model.

For the FE human head model, the actual parameters for the bone, skin and brain should be applied to present similar trends with the data from the cadaver head measurements to validate the FE head model. An addition of detailed parts such as the muscle, fat, middle ear and inner ear can also be an improvement. Instead of the brain as a fluid model, a visco-elastic or hyper visco-elastic model should be applied in order to capture the elastic behavior of the brain and thus influence transcranial

transmission/attenuation.

The location of the BAHA is closer to the ear canal; thus, investigation of the location of excitation should also be conducted. Furthermore, the bone should be modeled as cancellous and cortical bone similar to a typical bone structure. Pure cortical bone can be excessively stiff for vibration.

For both models, inclusion of actual BAHA model with external case and simulation of acoustic feedback phenomenon can be also recreated by validating with acoustic test data by microphone.

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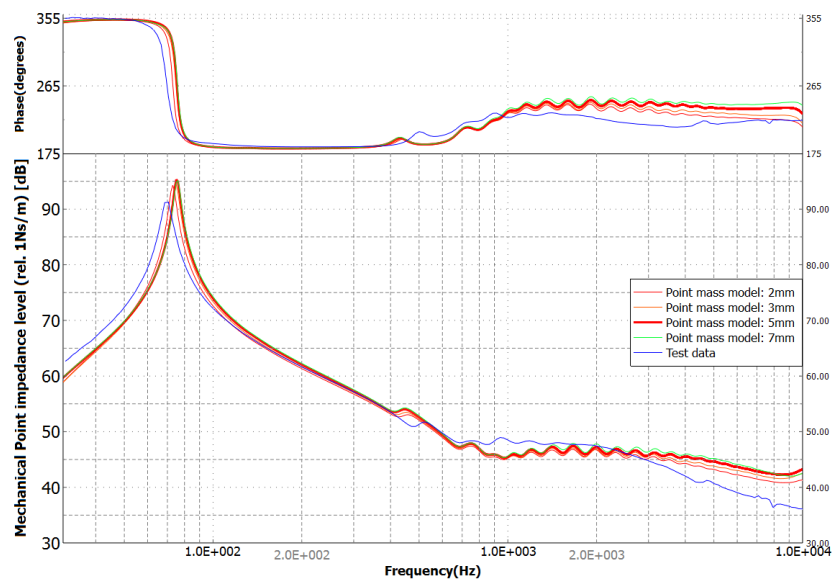
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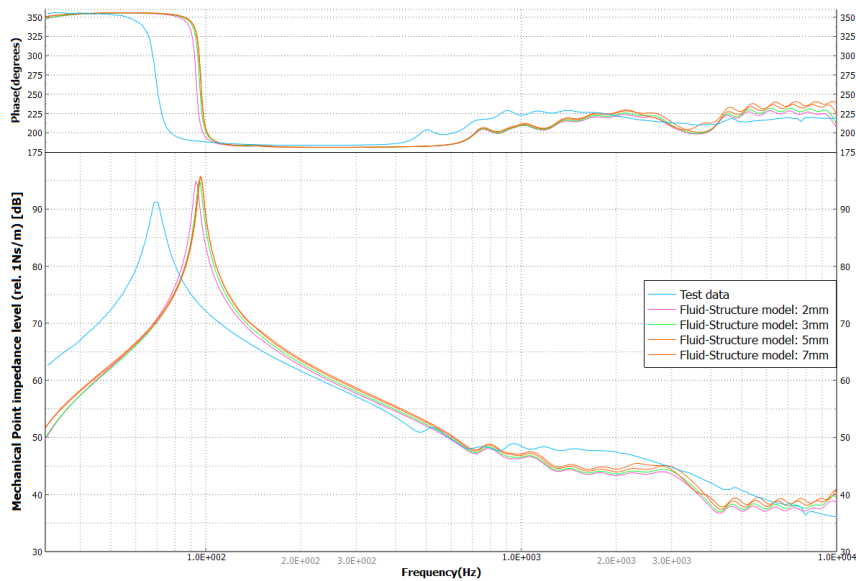
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Appendix

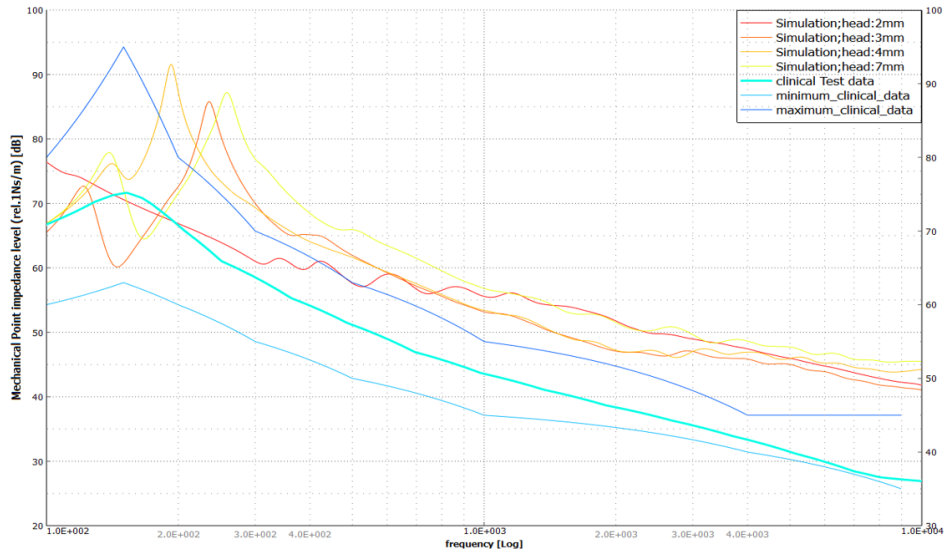
1. Head simulator model (NSM): Mesh convergence results with phase



2. Head simulator model (FS): Mesh convergence results with phase



3. Human head model: Mesh convergence results of Fluid-Structure model



Glossary

1. **Accelerance** : See ‘Frequency response function’
2. **Accelerometer** : An electro-mechanical transducer that measures acceleration
3. **Actuator**: A component of machine that is responsible for control or movement of a system
4. **Angular frequency** (ω): Also called circular frequency. Indicates the rate of change of phase in sinusoidal wave with unit of rad/s. $\omega = 2\pi f$ where f is frequency.
5. **Antiresonance**: Indicates minimum amplitude with a shift in oscillation phase. If a system reaches antiresonance, impedance reaches its maximum value and the mobility reaches a minimum. Antiresonance frequencies of measured FRF provide useful information on the dynamic properties of a system of interest.
6. **Bode plot**: The plot represents the behavior of dynamic system with respect to gain in decibel (dB) and phase in degree. In other words, this is a combination of a Bode magnitude plot, expressing the magnitude of the frequency response gain, and a Bode phase plot, expressing the frequency response phase shift.
7. **Damping**: The loss of energy of oscillating systems by dissipation or due to viscous materials.
8. **Decibel (dB)**: Measurement of sound pressure coined by Alexander Bell, literally meaning 1/10 of Bel. Uses logarithmic scale and converts 0.0002 to 2000 dynes/cm² to 0 to 140 dB HL. The decibel is not an absolute number rather a ratio between a standard and measured sound level.

$$\text{Number of dB} = 10 \cdot \log_{10}\left(\frac{\text{Sound intensity}}{\text{Reference intensity}}\right)$$

But intensity varies with square of pressure ($I \sim P^2$) and sound pressure P is not sound intensity.

$$\text{Number of dB} = 20 \cdot \log_{10}\left(\frac{\text{Sound pressure}}{\text{Reference pressure}}\right)$$

- (a) **Sound Pressure Level (SPL):** SPL can be defined as ratio of particular sound pressure to that of reference. Reference pressure is chosen by as the lowest sound pressure that can be easily detected by human beings, $20\mu\text{Pa}$ RMS. SPL is commonly used for describing phenomena such as hearing instruments, voice levels and environmental sounds.

$$\text{Intensity level in dB SPL} = 20 \cdot \log_{10}\left(\frac{\text{RMS sound pressure}}{20\mu\text{Pa}}\right)$$

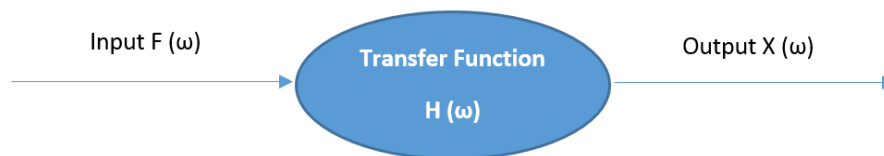
- (b) **Hearing Level (HL):** HL is used when using audiometer or audiogram to describe hearing threshold values.

9. **Dynamic stiffness:** Ratio between dynamic force and its dynamic displacement (response) such as vibration. Unit in N/m. Higher value of this means that system will vibrate less. It can be also described as

$$\text{Dynamic stiffness (restraint)} = \frac{\text{Force}}{\text{Observed vibration (response)}}$$

10. **Frequency response function (FRF):** FRF is a transfer function that expresses the structural response to an applied force as a frequency function. The response can be given in terms of displacement, velocity and acceleration. There are different forms of frequency response but acceleration is widely accepted method of measuring modal response.

FRF is a complex function so it possesses both real and imaginary parts it can be represented as both in magnitude and phase in frequency domain.



$$H(\omega) = \frac{X(\omega)}{F(\omega)}$$

11. **Transfer function:** Mathematically, transfer function is the Laplace transform of output divided by Laplace transform of input.

In order to investigate how the propagation of vibration takes place through the structure of interest, one must apply dynamic loading over a range of frequencies and measure the response at the point of application. This can provide as a means to estimate the resonance frequency of the drive point and used to calculate the drive point stiffness as well as attenuation between the two points.

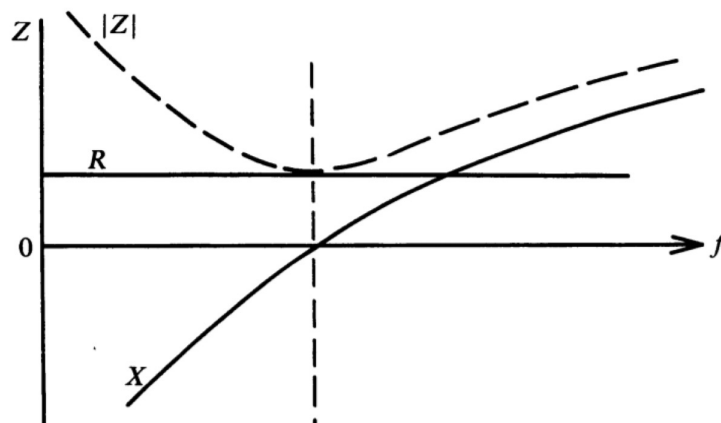
- (a) **Accelerance:** One of the basic transfer functions which is expressed by acceleration/force.

12. **Mechanical impedance (Z):** Indicates the resistance to motion.

$$\frac{F(\omega)}{v(\omega)}$$

Motion is described as force varying with respect to angular frequency (ω) at the point of application that leads to a velocity $v(\omega)$. This can be useful to determine the material property such as damping capacity or dynamic stiffness. It also forms a basis of determining natural frequencies and mode shapes of the system for vibration analysis.

- Materials respond differently to the sound. Relation between sound pressure and resulting velocity is described as $Z = \frac{F}{v}$ and this is a property of medium.
- High value means small vibration of the system and vice versa.
- Mechanical impedance is a complex property. It has real part which is called mechanical resistance (R) and its imaginary part as mechanical reactance (X_m). $Z = R + jX_m$. At resonance, the real part of Z becomes mechanical resistance while imaginary part becomes zero.



Real (R) and imaginary parts (X) of the mechanical impedance

13. **Modal analysis:** Defined as the study of the dynamic characteristics of a mechanical structure under excitation by vibration. The purpose of this is to find the modal shapes and frequencies at which the structure will amplify the effect of a load.

This also tells us at which frequency the structure will absorb all the energy applied to it, and what the shape looks like which corresponds to this frequency.

14. **Mode (mode of oscillation, mode of vibration, mode shape):** Shape of a vibration of system with respect to a natural frequency.

15. **Stiffness:** Resistance to deformation under load. Defined by ratio of a change of force to its linear displacement.
16. **Transducer:** Any device that converts one energy form to another. Usually the output is electric signals.