Acoustic Analysis of Honeybee Hive

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Approval Sheet

This thesis entitled "Acoustic Analysis of Honeybee Hive" by Patkulkar Ajay Sanjay is approved for the degree of Master of Technology from IIT Hyderabad.

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Patkulkar Ajay Sanjay

This thesis is dedicated to my parents

for their love, endless support and encouragement

Abstract

The design of a panel with lower weight and high stiffness for passive noise control application is a quite challenging task. Generally, more mass per unit area is required to get high transmission loss at mid and lower frequencies. Transmission loss at lower frequencies can be improved by higher stiffness and damping. A well-known solution in structural design community for higher stiffness to weight ratio is honeycomb structure. It motivates us to conduct acoustical experiment on a natural honeybee hive. The experimental results are quite encouraging to mimic in practical applications. Theoretical models have been proposed to explain the acoustical performance of the honeybee hive.

Design parameters of a hive like shape of the cell, number of layers, wall thickness and cell size have considered for parametric studies. Three different hypothesis have proposed to understand the acoustical performance of the hive. A simplified analytical expression has developed and corroborated with experimental results.

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Chapter 1

Introduction

1.1 Motivation

Nowadays, noise pollution in urban areas is increasing due to increasing usages of vehicles on road, trains on a rail, also air traffic, especially near airport during landing and take off of aeroplanes, industrial noise produced during different processing and many more. This increased noise pollution is making trouble in human and animal life. Case studies [1, 2] tells, about 52% of population is suffering from noise pollution, where road traffic noise is the main reason. Noise pollution affects on human health and also on human behaviour. This causes hypertension, hearing disabilities, sleep disturbance and many serious problems. So, it is required to isolate residencies near roadway, railway, airport or industrial worker from such noise pollution. There are different ways of noise control as shown in Figure 1.1. In noise control solution strategy, solutions may be provided at source, path and receiver location.

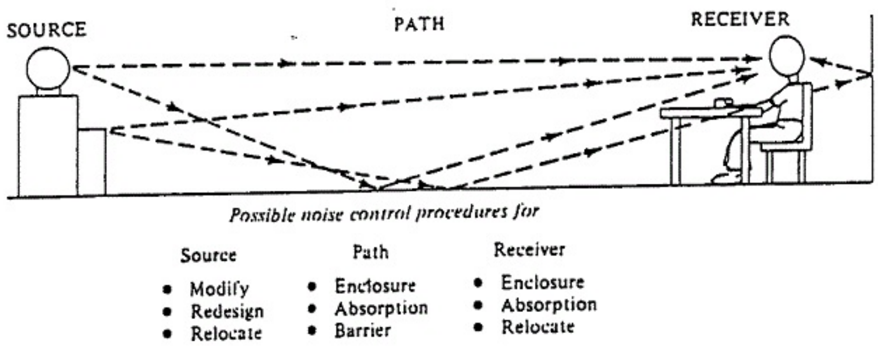


Figure 1.1: Noise control solution

1.2 Basics of Acoustics

1.2.1 Acoustic Wave

Acoustic wave is any disturbance that is propagated in an elastic medium, which may be a gas, a liquid, or a solid. We can say pressure perturbation over atmospheric pressure in a propagating medium is the acoustic wave. Speed of sound is related to the characteristics of the medium and function of temperature.

1.2.2 Noise

Noise is any unwanted sound, which makes human uncomfortable with that sound. It can be defined as loud, unpleasant, irritating or undesired sound. In general, sound is produced by vibration of any component, e.g. sound produced in machinery is due to vibration of its parts. Here sound produced by machine part will be noise, provided that, it is annoying.

1.2.3 Acoustic Variables

Sound pressure or acoustic pressure is the local deviation from the ambient (average, or equilibrium atmospheric pressure), caused by a sound wave. The solution of 1–D wave eqution i.e. Helmholtz eqution is written as [3],

$$p\left(z,t\right) = \left(A\ e^{-jk_z z} + Be^{jk_z z}\right) e^{j\omega t} \tag{1.1}$$

Where, first term represents propagating wave in z direction and second term is for reflected wave.

Particle velocity is the velocity $u(z)$ of a particle (real or imaginary) in a medium as it transmits in the wave. It can be written as eq. 1.2

$$u\left(z\right) = -\frac{1}{j\omega\rho_o}\frac{\partial p(z)}{\partial z} \tag{1.2}$$

For progressive plane wave, pressure is $p\left(z\right) = A\ e^{-jk_z z}$ and particle velocity will be $u\left(z\right) = \frac{p(z)}{\rho_o c}$. Where, ρ_o is a density of acoustic medium, and c is speed of sound is medium.

The acoustic impedance of a surface or medium is the ratio of the amplitude of the sound pressure and the amplitude of the particle velocity of an acoustic wave that impinges on the surface or medium.

$$Z\left(z\right) = \frac{p\left(z\right)}{u\left(z\right)} \tag{1.3}$$

Characteristic impedance of progressive wave is a product of density and speed of sound in that acoustic medium. It can be expressed as $Y_0\left(z\right) = \rho_o c$

1.2.4 Acoustic Properties

In noise control, barriers or absorptive materials are selected based on a frequency region of interest and type of application environment. Absorption coefficient and transmission loss are important acoustical properties to be considered for noise control solution along the path. Figure 1.2 shows incident sound wave interaction with a panel. Some part of incidence acoustic energy gets absorbed, partially get reflected and remaining part transmits through a panel. This analogy describes nicely absorption coefficient and transmission loss. Absorption coefficient is related to absorption capability of panel and transmission loss is an ability of not transferring sound through a panel.

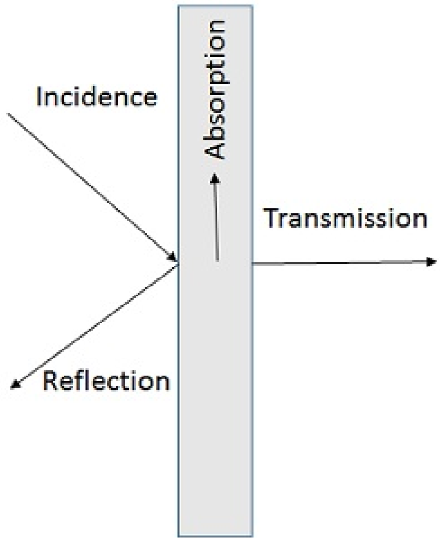


Figure 1.2: Sound incidence, absorption, reflection and transmission through panel

Absorption Coefficient

Absorption coefficient, α describes about how much sound energy gets absorbed by material from incidence sound energy. It can be defined as the ratio of sound power absorbed $W_{Absorbed}$ and sound power incident $W_{Incident}$.

$$\alpha = \frac{W_{Absorbed}}{W_{Incident}} \tag{1.4}$$

Transmission Loss

Transmission loss (TL) is ten times of logarithmic of base ten of reciprocal of transmission coefficient τ. Transmission coefficient is the ratio of sound power transmitted $W_{Transmitted}$ and sound power incident $W_{Incident}$. So more transmission coefficient, lesser the transmission loss. Transmission loss should be more, for good acoustic properties.

$$TL = 10 \log_{10} \left(\frac{1}{\tau} \right) dB \tag{1.5}$$

$$\tau = \frac{W_{Transmitted}}{W_{Incident}} \tag{1.6}$$

1.3 Problem Statement

Absorption coefficient depends on material porosity, flow resistance and thickness in sound traveling direction, etc. As a rule of thumb to get maximum absorption coefficient at particular frequency, thickness of material should be more than or equal to one-fourth of wavelength. Figure 1.3 shows that how a wave with wavelength (λ) gets trapped in a material while travelling through material. So in travel of wave most of the energy get absorbed by material, and we get a very high absorption coefficient. But problems come when we work at lower frequencies, we need very high thick material. For example at 100 Hz frequency, the required sample thickness is 0.85 m, which is very high value, and it increase total mass of a system. So this thesis discusses about getting high absorption coefficient at lower frequency.

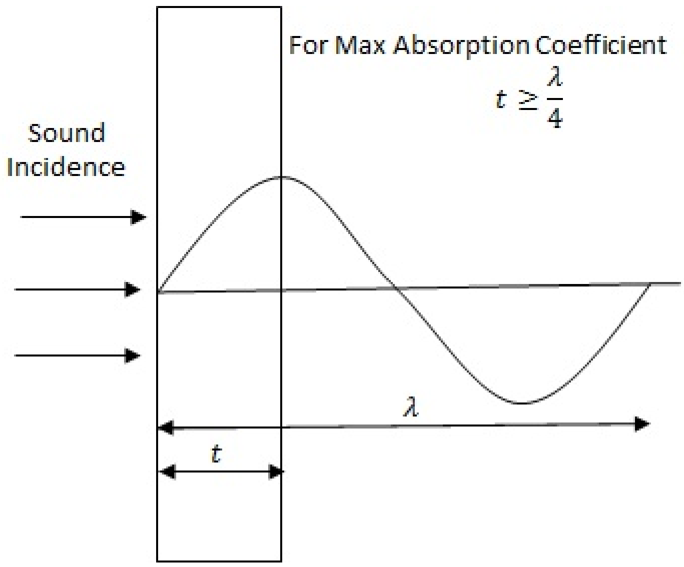


Figure 1.3: Effect of thickness on maximum absorption coefficient for particular frequency

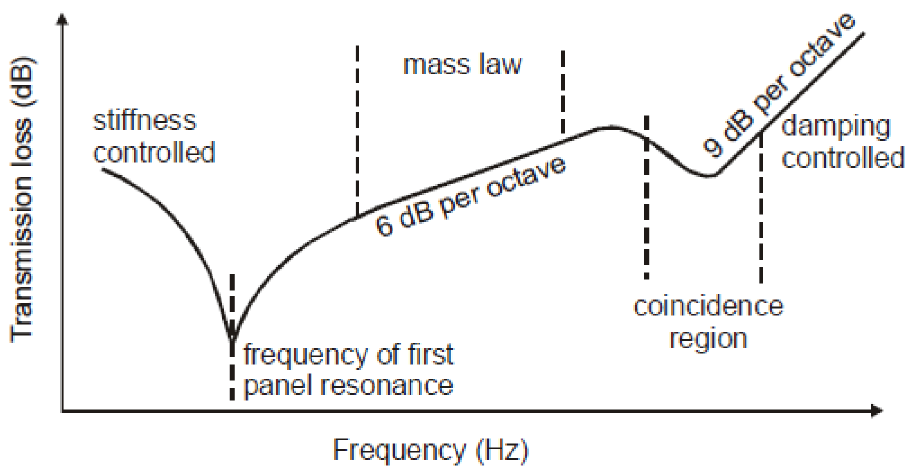


Figure 1.4: Transmission loss through single panel

According to mass law [3], transmission loss (TL) depends on surface density of panel and frequency. So as frequency decreases transmission loss reduces, for constant surface density. Our objective is to develop higher transmission loss for lower surface density. In aerospace, marines, automobile application, increasing mass is not feasible. Here, we need to think about another alternative to get higher transmission loss at lower frequencies. Figure 1.4 shows typical behavior of transmission loss as a function of frequency for isotropic single panel. At lower frequency, transmission loss is controlled by stiffness of the panel. At panel resonance frequency, there is a dip in TL and it is improved by introducing damping to the panel. After first resonance frequency, transmission loss is constantly increasing 6 dB per octave. This region is called mass controlled region, which follows the mass law. In the coincidence region, damping improves the TL values. Coincidence frequency can be calculated by equating bending wave number of panel and trace incident acoustic wave number of panel. It can be observed that stiffness of panel and damping improves TL at lower frequencies. There is only way that, we can make the panel stiff by adding stiffeners to a panel instead of increasing thickness of material.

1.4 Objectives

Panels are being used in building, locomotives, aircraft, automotive vehicles, transformers, enclosures, etc. as wall. Honeycomb structure panels have high stiffness to weight ratio. High stiffness and low weight are desired structural properties. Thicker absorptive material is used to get desired properties like absorption coefficient and transmission loss. In this process, we increase weight or geometrical dimensions of acoustical material. So it is required an alternative solution, which will give good structural and acoustic properties at lower frequencies. Our research focus is to develop an alternative design solution to achieve the structural and acoustical properties. One of the solution inspired from nature is honeybee hive. It has good structural properties and damping. So, the current research focus is to understand acoustical behavior of honeybee hive.

1.4.1 Design

Design of an acoustical panel is inspired from the natural honeybee hive. It is made with beeswax, and each cell of the hive is hexagonal in shape. These hexagonal cells are connected in parallel to each other as shown in Figure 1.5 which forms one layer of periodic hexagonal cells. All of these cells are connected to one more layer by centre layer called membrane. This centre membrane can be considered as simple panel and hexagonal cells as stiffeners attached to a panel.

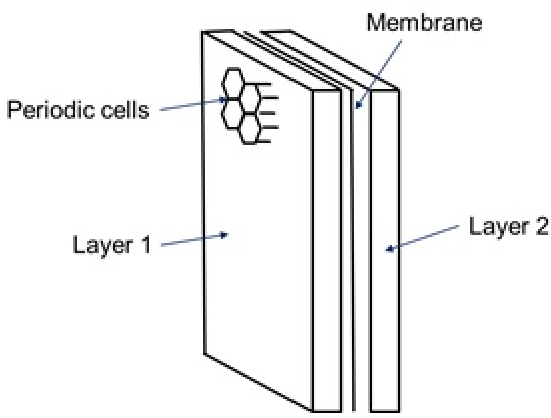


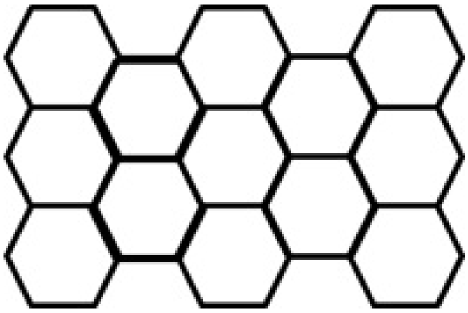
Figure 1.5: Schematic diagram of natural honeybee hive arrangement

1.4.2 Design Parameters

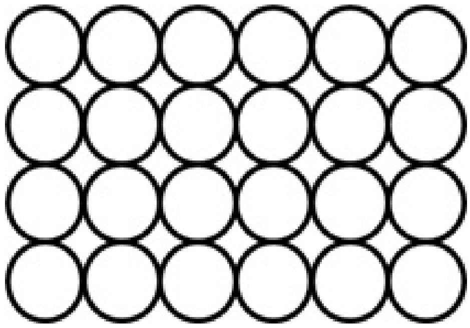
In design of an acoustical panel, number of parameters can be considered like shape of each cell, flexibility of cell and also membrane of panel, periodicity in cells, number of layers arranged, size of cell, which describes porosity of panel and thickness of membrane.

Cell shape

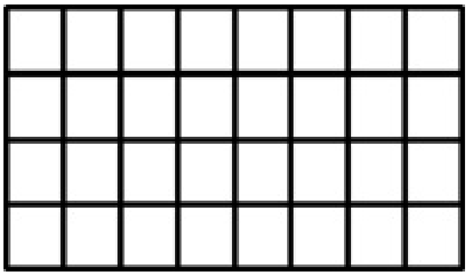
Panels can be made of cells with different geometrical shapes. Possible cell shapes are hexagonal, circular, rectangular and triangular. Every shape has its own speciality in stiffness, weight, strength and porosity. Out of all these shapes, hexagonal shapes have very good stiffness with low weight. Different cell shapes are shown in Figure 1.6.



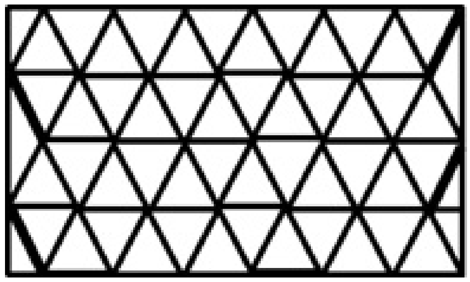
(a) Hexagon



(b) Circle



(c) Rectangle



(d) Triangle

Figure 1.6: Schematic diagrams of different cell shapes and its arrangement

Cell wall flexibility with higher damping

For good acoustic properties, panel should absorb incident sound energy or divert to uninterested direction, means it should divert sound wave traveling in axial direction. If panel membrane or cell walls are flexible, the incident sound wave excites flexible walls and converts as vibrational energy. The generated vibration energy is dissipated with help of damping. So, cell wall requires flexible wall with higher damping. Flexibility can be achieved by choosing appropriate material like natural bee's wax panel, polymer with higher damping, composites and metal with higher damping, etc.

Periodicity

If two monopoles are arranged side by side in out phase, then that configuration will act as a dipole. In dipole, net sound radiated by two monopole will be canceled. This can be done, if we arrange all cells in a periodic way. In periodic cell as shown in Figure 1.7, each cell of periodic structure will act as monopole and radiate sound in all directions. In zoomed area, it shows that two monopoles are arranged in side by side. According to dipole theory, we get less sound radiation in periodic cell [4].

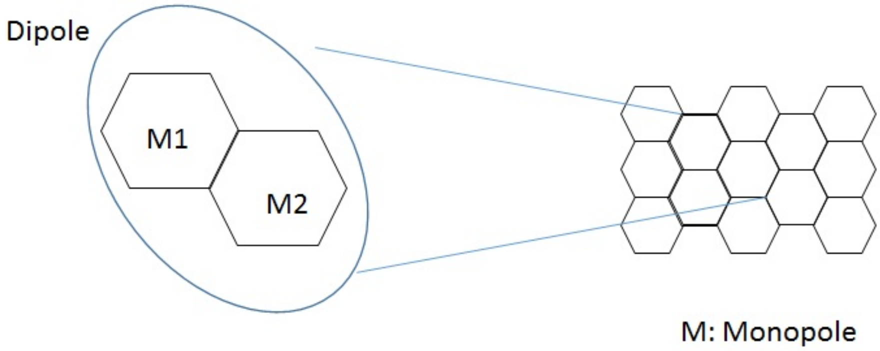


Figure 1.7: Periodic cells act as a dipole

Layer arrangement

Layer of periodic cell can be arranged parallel by keeping membrane in between the two layers to get required acoustical properties as shown in Figure 1.8. The overall thickness of compound panel increases as number of layers increases. However, it can be optimized based on the required acoustical properties and geometrical constraints.

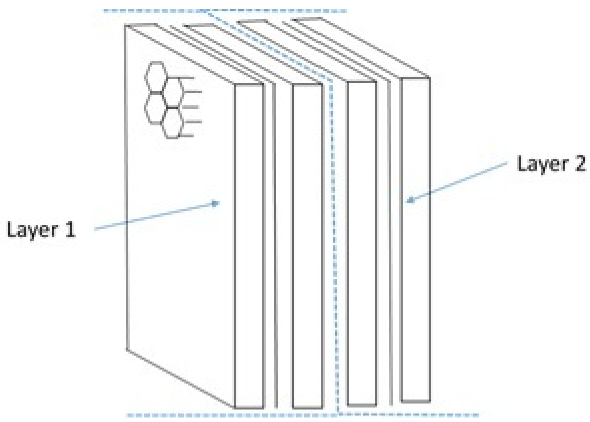


Figure 1.8: Schematic diagram of layer arrangement of panel

Cell size

Cell size can be defined as distance between opposite faces of cell as shown in Figure 1.9 . It is a measure of amount of air present in each cell. It describes the porosity of material.

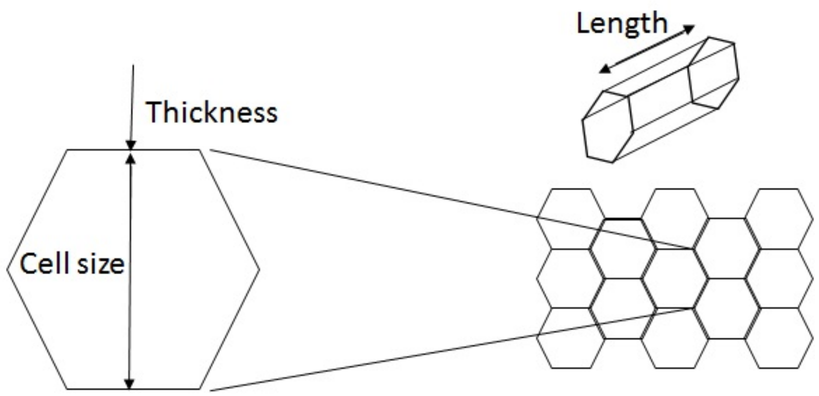


Figure 1.9: Schematic diagram of cell dimensions

Thickness

It is a thickness of cell wall and membrane present in between two arrays as shown in Figure 1.9. Wall thickness decides flexibility.

Length

Length of cell is defined as thickness of an array which decided based on final thickness of a panel as shown in Figure 1.9.

1.5 Literature Survey

Passive noise control along the path is done with the help of keeping barrier in between source and receiver or using enclosure for source. For such enclosures required panel thickness is very high especially for the design of enclosure wall for low frequency noise control application. As per mass law, high surface density panels are required, so that conventional method for blocking of low frequency sound needs heavy mass along cross-sectional area. For simple homogeneous panel, stiffness control and damping control region solution is not efficient in low frequency noise control, if we concern about mass of panel.

To overcome this problem, the periodic structure is the one of the solution for this case. Xin et al. [5] modelled analytical formulation for stiffened panel and calculated sound transmission loss through it. Rib stiffeners are equally spaced and identical in shapes. Tension, bending moments, torsion moments, and inertia due to mass of stiffeners are considered in analytical model. Due to stiffened ribs in panel, its properties changes from isotropic to orthogonal panel. This model discusses about sound transmission through air borne and structure borne path. They concluded that slight change in spacing of rib in panel will not change transmission loss curve. Its trend do not change with spacing of rib. When spacing increased, natural frequency of panel goes down, resulting in shifting of peaks and dips to lower frequency region.

Different configuration of periodic structure arrangement using rib stiffeners to panel is discussed by Lin et al. [6]. It discusses configurations like ribs in one direction, two ribs in perpendicular like square periodic structure, two ribs pattern attached to one panel where other panel free from pattern. The gain in the transmission loss in two ribs in perpendicular configuration is more than single rib panel configuration with same mass. A little advantage in transmission loss is seen when two perpendicular rib pattern is used with one free panel and other attached to pattern over the configuration of both panels attached to rib pattern.

Wang et al. [7], prepared theoretical model for sound transmission though panels attached to ribs. In this periodic model, ribs are modeled as set periodically spaced lumped mass attached to panels all over structure with the set periodically spaced springs having rotational stiffness and longitudinal stiffness. This model discussed about effect of incidence angle of sound waves and different parameters.

Analytical wave propagation through periodic structure is summarized by Mead [8]. He discussed about sound wave propagation in one, two and three dimensional continuous structure. This work can be applied to uniform beams, plates and shells, also energy method analysis can be used to work on non-uniform periodic structures. It consist computational method to calculate forced wave motion generated on infinite periodic structre by single point harmonic forces or plane harmonic pressure waves.

The behavior of multiple elements arranged in arrays is discussed by C. J. Naify [9]. According to him acoustic meta materials with negative dynamic mass density have been shown to demonstrate a five-fold increase in transmission loss (TL) over mass law predictions for a narrowband (100 Hz) at low frequencies (100 - 1000 Hz).

It can be understand that periodic structures are one of the feasible design option to provide noise control at low frequencies based on literature survey. Naturally available honeycomb hive is periodic structure, and it might show better acoustical performance at lower frequency. So next chapter will discuss details about acoustical properties of honeybee hive structure and its mechanism in noise control.

1.6 Organization of Thesis

- Chapter 2 discusses about experimental work which describe about experimental setup and sample preparation.
- Chapter 3 gives brief possible theoretical formulations to validate experimental results and discusses mass law including effect of stiffness and damping to calculate transmission loss.
- Chapter 4 discuss about computational analysis for calculating transmission loss and modal analysis. This analysis uncovers reasons of particular behaviours in acoustic properties.
- Chapter 5 discusses results of different samples, different configurations, hexagon and circular shape samples and their acoustic properties. Also it discuss about correlation between experimental and theoretical work.
- Chapter 6 is conclusions and future work.

Chapter 2

Experimental Work

This chapter discuss about experimental work done for measuring absorption coefficient and transmission loss of honeybee hive structure.

2.1 Experimental Setup and Instrumentation

For experimental testing of acoustic properties of honeybee hive and circular cell samples (straw samples), we used impedance tube as shown in Figure 2.1. The absorption coefficients and transmission loss of sample have measured according to ISO 10534-2 [10], ASTM E1050-08 [11], ASTM E2611-09 [12] standards.

Figure 2.1: Impedance tubes

BSWA SW series Impedance tube [13] follows ISO 10534-2, ASTM E1050-08, ASTM E2611-09 standards for measuring absorption coefficient and transmission loss. These are measured using transfer function method (TF). This method separates the incident and reflected sound pressure from the measured transfer function between two locations in tube, and then calculate the acoustic properties of sample installed in the tube. Impedance tube has two sizes of tubes, microphones, DAQ hardware and measurement software.

2.1.1 Tubes

There are two different size tubes, which are used for different frequency range for measuring absorption coefficient and transmission loss of samples. These tubes with series number SW422 and SW477 having internal diameter 100 mm and 30 mm, respectively. SW422 and SW477 series tube's working frequency range is in between 63 to 1600 Hz and 800 to 6300 Hz respectively. These sizes have chosen to maintain plane wave propagation inside the tube. These tubes have built-in speaker at one end of the tube and provision to mount the microphones along tube wall surface.

2.1.2 Microphones

There are four numbers of BSWA 1/4" microphones (MPA416). These microphones have a good phase match as per standard requirement. This microphones fit in mounting provided on tube and measures acoustic pressure inside tube at that position.

2.1.3 DAQ System

Data-acquisition hardware (MC3242) has 4-channels input and 2-channels of output. Microphones can be connected to input channel to collect sound pressures. Output channel goes to the speaker through Power Amplifier. DAQ has an USB cable connections for computer interface. VA-Lab software used for post-processing and provides sound absorption and transmission loss.

2.1.4 Power Amplifier

Power amplifier (PA50) provides gives the amplified signal to a speaker. Gain toggle is used to adjust the sound pressure level inside tube in between 90 to 110 dB. Output of DAQ system connects to input power amplifier, and its output connects to the speaker.

2.1.5 Calibrator

It is a sound source with 1 kHz tone of 114 dB SPL used for calibration purpose. It can be used to calibrate microphones. This calibrator has an adapter of 1/4-inch microphone. This calibrator follows IEC 60942:2003 Class 1, ANSI S1.40-1984 and GB/T 15173-1994. Using VA-Lab software and calibrator, microphones can be calibrated at 114 dB and 1000 Hz. Figure 2.2 shows the snapshot of calibration process in VA-Lab software.

2.2 Experimental Procedure

Experiments are carried out step by step from sample preparation to measurement. Following points will discuss about experimental procedure.

2.2.1 Sample Preparation

Honeybee hive samples

Impedance tube requires circular sample with 30 mm and 100 mm diameter. Honeybee hive samples are prepared by cutting with help of sharp cutter in required circular shape as shown in Figure 2.3.

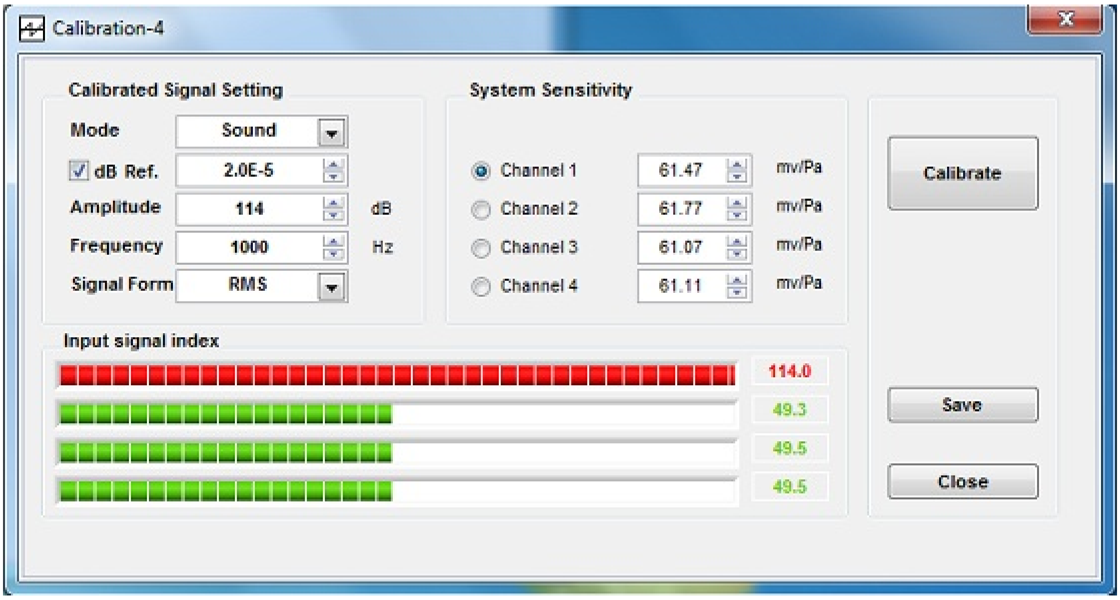


Figure 2.2: Microphone calibration interface snapshot

Total twelve samples of the honeybee hive were prepared for testing purpose. First, four were used to check repeatability of transmission loss and absorption coefficient of the honeybee hive. Remaining samples were used for parametric study. Effect of cell size and thickness have checked. Dimensions of first four samples are given in Appendix B and remaining samples dimension are mentioned in Appedix D.

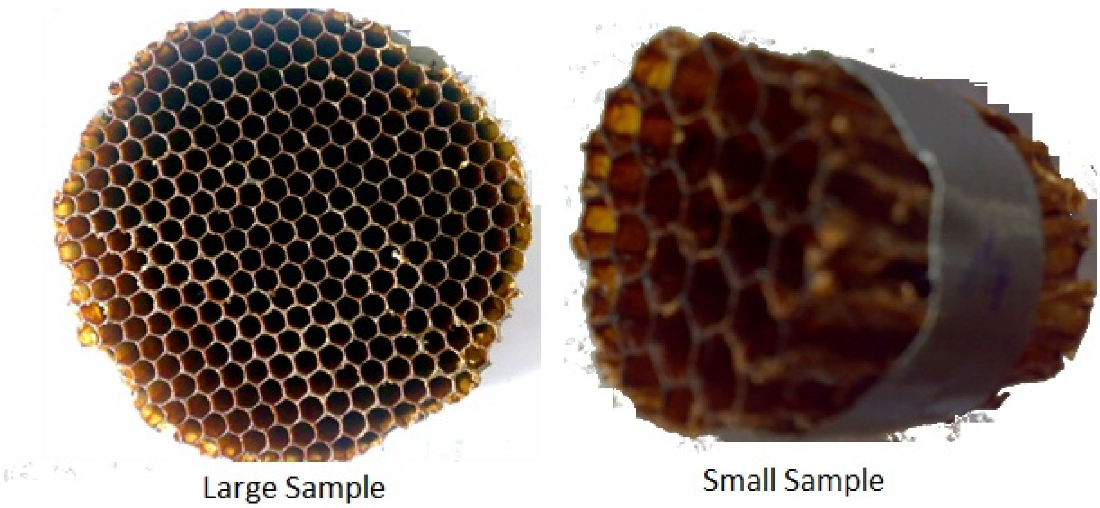


Figure 2.3: Honeybee hive samples prepared for testing

Circular cell samples

As discussed earlier, different cell shapes can be tried for acoustical panel. Along with hexagon cell shape, circular cell shape samples are prepared with the help of polypropylene tubes. In making of this samples, we used plastic straws, sellotape, sharp cutter and measuring scale. In first stage, stick straws on sellotape as shown in Figure 2.4. Second stage is to mark required length of tubes and cut it with sharp cutter. In next stage, straw row has to roll one other in a spiral, so to get the final circular array of straws. Different samples were prepared to do parametric studies and also typical dimensions information given in Table 2.1. Membrane thickness is 0.1 mm.

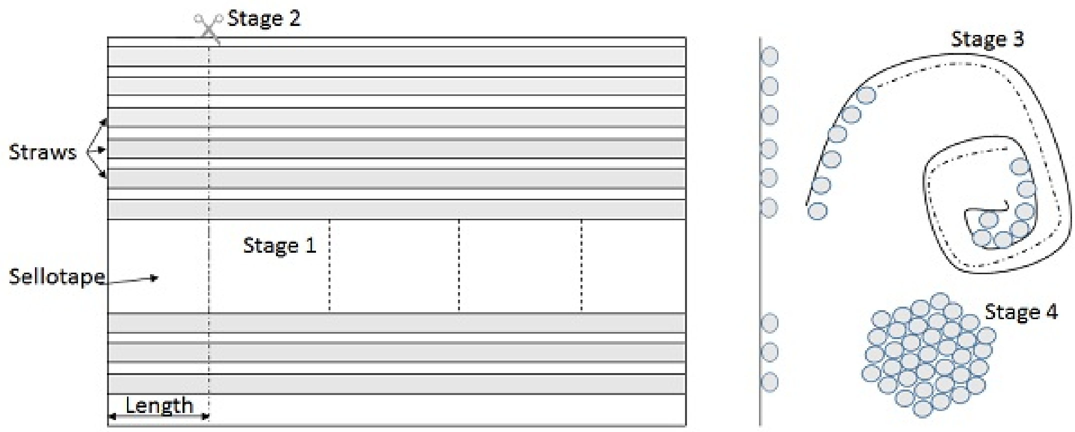


Figure 2.4: Preparation of circular cell samples

Table 2.1: Dimension of circular cell samples

Sample	Cell size (mm)	Thickness (mm)	Length (mm)
1	5.5	0.10	20
2	5.5	0.10	20
3	6.0	0.15	20
4	3.5	0.15	20

2.2.2 Measurement of Absorption Coefficient

Figure 2.5 shows the schematic diagram of impedance tube to measure absorption coefficient.

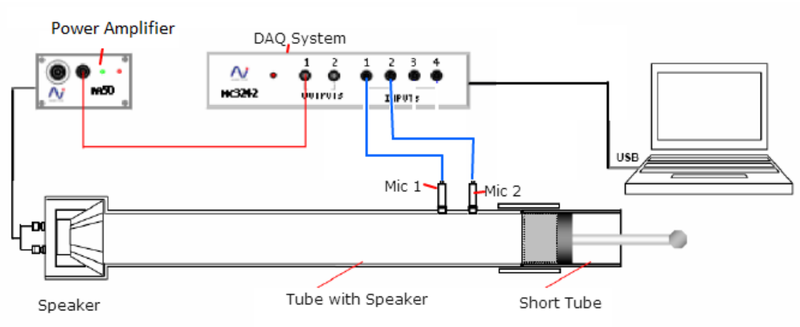


Figure 2.5: Schematic diagram for measurement of absorption coefficient

This measurement process requires two samples with a small and large diameter which fits in two different size tubes.

2.2.3 Measurement of Transmission Loss

Figure 2.6 shows the schematic diagram of impedance tube for measuring transmission loss. Measurement procedure is given in Appendix C.

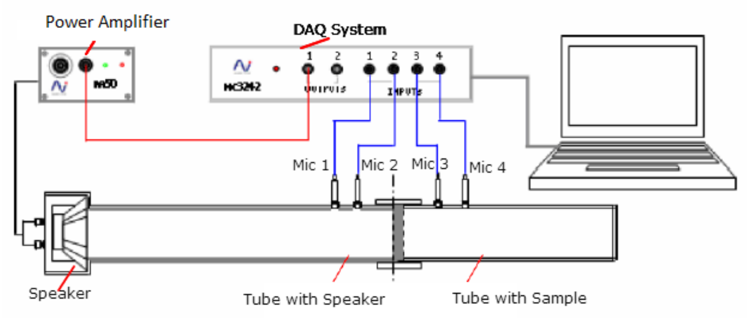


Figure 2.6: Schematic diagram for measurement of transmission loss

2.3 Test Configurations

Different types of test configurations have planned to measure absorption coefficient and transmission loss. These configurations are as follows:

2.3.1 Honeybee Hive Samples

1. Samples 1 to 12 made using honeybee hive is tested. This test measures acoustic properties for simple array of the hexagonal cylinder with centre membrane.

2. Honeybee hive sample 1 to 4 used to check repeatability of measurement. On three different days, three trials are taken for each sample and comparison of measurement is done for repetition of result.

3. Samples 5 to 12 used for studying parameter effect on acoustic properties. Following Table 2.2 shows different groups formed for parametric study. Small and large sample have different dimensions, so that there are two groups arranged in ascending order of respective parameter. Let's consider for absorption coefficient and for parameter thickness, large sample group is sample 8, 5, 7, 6, i.e. sample 8's thickness is less and sample 5's thickness is more than sample 8's, and so on. Where large samples are used for studying below 1600 Hz frequency and small sample for above 1600 Hz.

Table 2.2: Sample groups for parametric study on honeybee hive

Parameter	Absorption Coefficient		Transmission Loss	
	Large	Small	Large	Small
Cell Size	5, 7, 6	5, 7, 6	5, 7	5, 7
Thickness	8, 5, 7, 6	5, 8, 6, 7	8, 9, 5, 7	5, 9, 8, 7

4. Sample 5 to 8 is used in parallel configuration forming a layer of sample 5 and sample 6 and another layer of sample 7 and sample 8.

2.3.2 Circular Cell Samples

1. Acoustic properties of sample 1 to 4 measured. Sample dimensions are given Table 2.1. Where, sample 1 is prepared with glue instead of sellotape, else other samples are prepared using method described in section 2.2.1.

2. Studied the effect of centre membrane based on with and without centre membrane configuration. Simple array of circular cells is tested which give acoustic properties for circular cell sample without membrane.

3. Parametric study is done with help of samples 2, 3 and 4. These samples have different dimension. Table 2.3 shows sample groups for parametric study for acoustic properties. Cell size of sample 3 is more thane 4, while thickness of both sample is same. Thickness of 3 is more than 2, while these sample have negligible difference in cell size as compare to thickness difference.

Table 2.3: Sample groups for parametric study on circular cell

Parameter	Absorption Coefficient	Transmission Loss
Cell Size	4, 3	4, 3
Thickness	2, 3	2, 3

2.4 Dimensions of Honeybee Hive

Dimensions of honey bee hive is required for analytical and numerical work. Three different methods have been used to find the dimensions of naturally available honeybee hives.

2.4.1 3-D Scanner

Honeybee hive is made from wax, which is very soft and flexible, so dimensions cannot be measured accurately by contact type measuring instrument. 3-D Scanner is a non-contact type of instrument, which gives a scanned 3-D object with help of laser reflection. Figure 2.7 shows working principle of 3-D scanner.

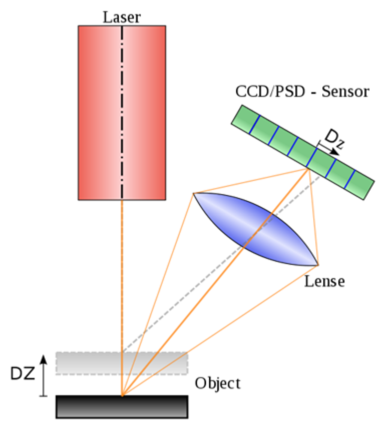


Figure 2.7: Working principle diagram of 3-D scanner

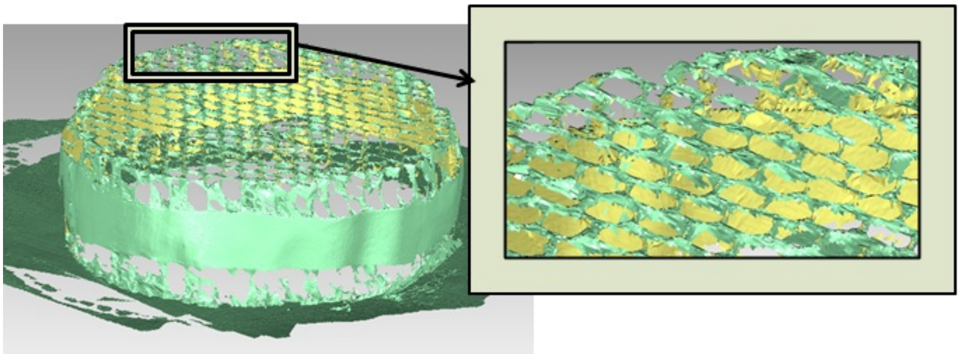


Figure 2.8: Scanned object captured by FARO Arm

After scanning and post-processing in Geomagic Studio software, object looks like in Figure 2.8 and 2.9. It can be observed from Figures 2.8 and 2.9 that obtaining depth information is difficult. Similarly, measuring small dimensions may not be accurate. So, 3-D scanner data is not used for further analysis.

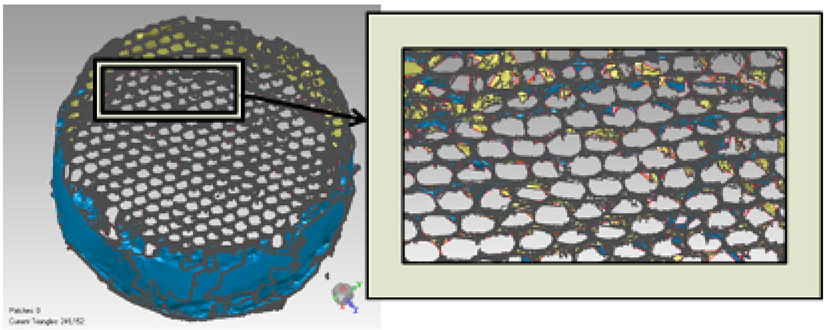


Figure 2.9: Honey bee hive image after post-processing in Geomagic Studio

2.4.2 Optical Microscope

Table 2.4 shows the dimension of the honeybee hive is taken using optical microscope. Values given in Table 2.4 is an average data for ten measurements for each parameter.

Table 2.4: Dimension of unit cell of honeybee hive using optical microscope

Parameter	Dimension in mm
Neck thickness	0.34
Wall thickness	0.12
Distance between two opposite surfaces of neck	4.75
Length of neck	0.34
Length of wall	16.66
Distance between two opposite surfaces of wall	4.97
Total length of two unit cell back to back	34.00

This method is somewhat time-consuming and focusing, adjustments are required for each reading. Image processing is used as an alternative to optical microscope measurement. Image Processing is used for measuring dimensions and also validating optical microscope measurement.

2.4.3 Image Processing

In this method, images are captured to measure dimension of a particular parameter with known distance, e.g. taking photograph of sample with measuring scale. In MATLAB using an Image-Processing toolbox, number of pixel for parameter and number of pixels required for known distance for scale in an image is measured. From this data, actual dimensions are calculated.

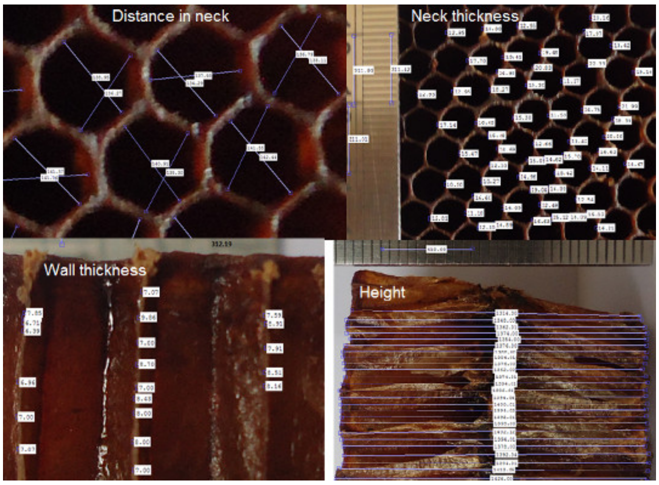


Figure 2.10: Analysis of cell dimensions using image processing tool box

In this 50 readings are taken for each parameter and statistical analysis is done to check for 99%, 95% and 90% confidence level. For each parameter statistical analysis is done in Appendix B

$$Confidence\ Interval = x \pm z\frac{s}{\sqrt{n}}$$

Where, x = mean, z= critical value, s = standard deviation, n = no. of reading,
$z\frac{s}{\sqrt{n}}$= Margin of error and value for z taken from following Table 2.5 for different confidence level.

Table 2.5: Critical values

For % Confidence	Critical Value z
99%	2.58
95%	1.96
90%	1.645

Chapter 3

Theoretical Formulation

3.1 Introduction

This chapter is related to analytical formulation considering three hypotheses to find physics behind special behavior of absorption coefficient and transmission loss. And also it includes transmission loss calculation using the mass law with effect of mass, stiffness and damping and using narrow tube theory.

3.2 Hypotheses

The acoustical performance of the natural honeybee hive might be explained theoretically based on three hypotheses. The proposed hypotheses are Helmholtz resonator, dipole theory and dissipation of energy. Honeybee hive cell structure is like Helmholtz resonator. Honeybee hives have higher damping, which supports dissipation mechanism, and cell periodicity may generate dipole sound radiation pattern.

3.2.1 Helmholtz Resonator

Helmholtz resonator can be considered as spring mass system [3]. Where the neck acts as lumped inertance (mass). Volume acts as lumped compliance (stiffness). Figure 3.1 shows the schematic diagram of Helmholtz resonator. If we tune these Helmholtz resonator dimensions for particular

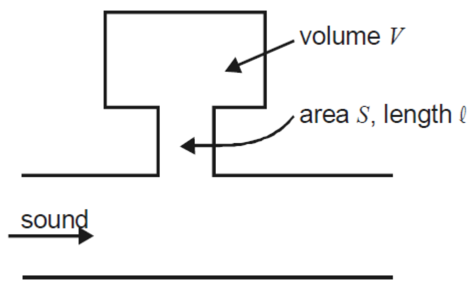


Figure 3.1: Schematic diagram of Helmholtz resonator

frequency, it will give good axial transmission loss for that tonal frequency region. Honeybee hive hexagonal tube's structure has approximated as Helmholtz resonator. Resonance frequency of resonator [3] can be calculated from eq. 3.1

$$f_H = \frac{c}{2\pi}\sqrt{\frac{S}{Vl}} \tag{3.1}$$

Where, c is speed of sound, S is a cross-sectional area of neck, l is a length of a neck and V is volume of Helmholtz resonator.

Consider honeybee hive as Helmholtz resonator, like opening as neck and hexagonal tube as volume of the resonator. So required S, V can be calculated as. $S =$ Cross sectional area of neck $= \frac{\sqrt{3}a^2}{2}$, $V=$ Volume of air in resonator $= \frac{\sqrt{3}a^2 h}{2}$, $L=$ Length of neck, $a =$ cell size, $f_H = 21730Hz$.

3.2.2 Dipole Phenomenon

When two sound sources kept nearby in out of the phase act as a dipole. Sound radiated by sources get canceled because of out of phase sources [14]. Dipole sound radiation efficiency is lower compared to the monopole. So, it may provide higher TL.

Figure 3.2 shows schematic diagram of base excitation problem [15]. Excitation base and mass have connected through a spring and damper elements. As we know in base excitation, when excitation frequency crosses natural frequency of a spring mass system then base and mass go out of phase. So, sources go out of phase their net effect of sound radiation will be vector addition of sound radiated by both sources. Figure 3.3 shows the dipole configuration with base excitation and cavity.

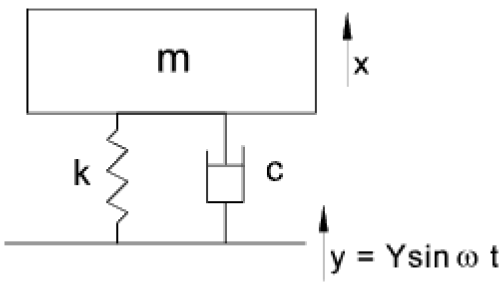


Figure 3.2: Lumped parameter representation of base excitation

Source strength of base excitation can be calculated by $Q_s = S_s V_s$, where, S_s is the surface area sound radiating source and Source strength for mass can be calculated using eq. 3.2

$$Q_c = \int_0^a V(r)\ 2\pi r\ dr, \tag{3.2}$$

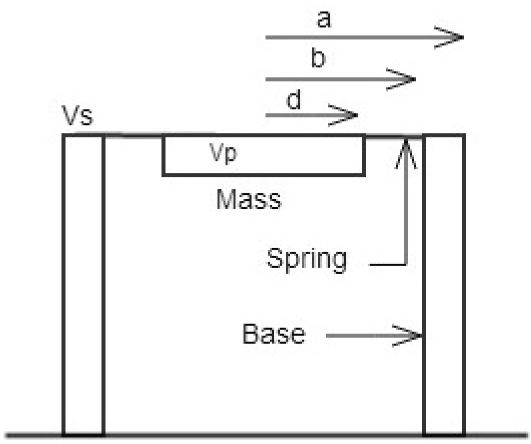


Figure 3.3: Dipole configuration in hexagonal cell

where $V(r)$ is given by eq. 3.4

$$\begin{aligned} V(r) &= V_p(r) = \frac{c_m + \frac{k}{j\omega}}{j\omega m_p + c_m + \frac{k}{j\omega}}, \quad 0 \leq r \leq d \\ &= V_h(r) = \frac{V_p + V_s}{2} + \cos\frac{\pi r}{b - c} \quad \frac{V_p - V_s}{2}, \quad d < r < b, \\ &= V_s, \quad b \leq r \leq a \end{aligned} \tag{3.3}$$

Where, k is the addition of mechanical and fluid stiffness, c_m is mechanical damping of spring, m_p is mass of top plate.

Acoustic power radiated by base and excited mass can be calculated as eq.3.4

$$W = \frac{1}{2}\pi\rho c\frac{|Q|}{\lambda} \tag{3.4}$$

where, λ is acoustic wavelength.

3.2.3 Dissipation of Sound Energy

Consider a cylinder with flexible wall as shown in Figure 3.4. For this configuration, wave equation [16] can be written as following eq. 3.5.

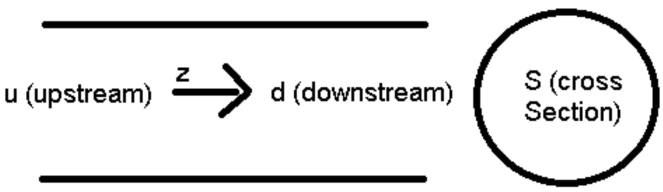


Figure 3.4: Transmission loss for flexible cylindrical cell

$$\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} + \frac{\bar{\beta}\ l}{c}\frac{\partial p}{\partial t} - \frac{\partial^2 p}{\partial z^2} = 0 \tag{3.5}$$

Where, c is sound speed in the cylinder, p is acoustic pressure, t is time, $\bar{\beta}$ is normalized admittance and l is the perimeter of the cylinder. Now considering the harmonic variation in time $e^{j\omega t}$ and space $e^{jk_z z}$, then eq. 3.5 reduces to eq. 3.6.

$$-\frac{\omega^2}{c^2}p + \frac{j\ \omega\ l\ \bar{\beta}\ p}{cs} + k_z^2\ p = 0 \tag{3.6}$$

Now k_z can derive as follows.

$$k_z = \pm k_i \left(1 - \frac{j\ l\ \bar{\beta}}{k_i s}\right)$$

$$k_i = \frac{\omega}{c_i}$$

In transfer matrix form [17] we can get the relation in upstream and downstream acoustic variables as eq. 3.7

$$\begin{pmatrix} p_u \\ v_u \end{pmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{pmatrix} p_d \\ v_d \end{pmatrix} \tag{3.7}$$

where,

$$\begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} = \begin{bmatrix} \cos(k_z l) & jY_z \sin(k_z l) \\ \frac{j \sin(k_z l)}{Y_z} & \cos(k_z l) \end{bmatrix}$$

Now, using the formula for calculating transmission loss from transfer matrix method [17] as eq. 3.8

$$TL = 20\ log_{10}\left(\frac{\left|T_{11} + \frac{T_{12}}{Y_0} + T_{21}Y_0 + T_{22}\right|}{2}\right) \tag{3.8}$$

Where, Z_o is terminal impedance and v_u is the velocity at upstream. Terminal impedance can be calculated as follows [16]

Anechoic termination, $Z_o = \frac{\rho_i c_i}{s} = Y_z$

Rigid end termination, $Z_o = \infty$

Open end termination, $Z_o = \frac{\rho_i c_i}{s}(\frac{k_i^2 r_o^2}{4} + j0.6\ k_i r_o)$ for $k_i r_o <\ 0.5$ where, $r_o = \sqrt{\frac{s}{\pi}}$ and s is cross-sectional area of duct.

Normalized admittance can be expressed as [16]

$$\bar{\beta} = \oint \frac{j\omega\rho_i c_i \mathrm{w}(\theta, z)}{P\ (e_z^{-jk_z z} + R_o e_z^{jk_z z})} d\theta \tag{3.9}$$

where $\mathrm{w}(\theta, z)$ is wall displacement along the radial direction [18] which can be calculated using eq. 3.10

$$\mathrm{w}(\theta, z) = \sum_{m=0}^{\infty} \mathrm{w}_m f(z) \cos m\theta \tag{3.10}$$

Where,

$$\mathrm{w}_m = \frac{PabA_m}{h\rho_p c_l^2}\left(\frac{pp\ tt - qq\ ss}{jk_z l\,(ss+tt)\sqrt{ss\ tt}}\left[\tan^{-1}\left(\frac{R_o\,(ss\ tt)\,e^{jk_z l}+(ss-tt)}{2\sqrt{ss\ tt}}\right)\right.\right.$$
$$\left.\left.-\tan^{-1}\left(\frac{R_o\,(ss\ tt)\,e^{-jk_z l}+(ss-tt)}{2\sqrt{ss\ tt}}\right)\right]+\left(\frac{pp+qq}{ss+tt}\right)\right)$$

And, $f(z) = e_z^{-jk_z z} + R_o e_z^{jk_z z}$, $A_m = 1$ is for circular duct.

3.3 Transmission Loss Calculation

3.3.1 Mass Law

Mass law states that transmission loss (TL) for isotropic or orthotropic panel with sound wave incidence angle θ on normal to panel surface, then TL is calculated as [3];

$$TL_{ML} = 10\log_{10}\left(1+\left(\frac{Z\cos\theta}{2\rho c}\right)^2\right) \tag{3.11}$$

Where, $Z = j2\pi fm =$ Panel impedance considering mass effect
$f =$ frequency at which TL is to be measured,
$m =$ surface mass density of panel,
$\rho =$ density of medium through which sound waves travelling,
$c =$ speed of sound in medium.

Consider the normal incident of sound wave on panel, eq. 3.11 becomes

$$TL_{ML} = 10\log_{10}\left(1+\left(\frac{\pi fm}{\rho c}\right)^2\right)$$

For the convenience, this equation can be used as by taking $\rho = 1.223\ kg/m^3$ and $c = 340\ m/s$, then simplified $TL_{ML} = 20\log_{10}(fm) - 47\ dB$.

Honeybee hive is not a simple panel. It has hexagonal stiffeners on both sides of the panel. So surface mass density of honeybee hive can be calculated by measuring mass of honeybee hive sample and dividing by cross-sectional area of sample. Or for theoretical calculation of surface mass density of the honeybee hive can done as explained in Appendix E.

3.3.2 Inclusion of Stiffness and Damping in Mass Law

TL of the panel can be calculated by adding an effect of bending stiffness of the panel in the panel impedance.

Panel impedance can be expressed as in terms of mass and stiffness as follows,

$$Z = j2\pi fm - j\frac{k}{2\pi f} \tag{3.12}$$

Where, k = stiffness which can be written as follows

$$k = \pi^4 D \left(\left(\frac{m}{a} \right)^2 + \left(\frac{n}{b} \right)^2 \right) \tag{3.13}$$

$$\begin{aligned} \text{Where, } a, b &= \text{panel width and length} \\ m, n &= \text{integers 1,2,...} \\ D &= \text{flexural rigidity} \end{aligned}$$

Flexural rigidity can be written including damping effect in Young's modulus as follows,

$$D = \frac{E\,(1 + j\eta)\,I}{(1 - \nu^2)} \tag{3.14}$$

$$\begin{aligned} \text{Where, } E &= \text{Young's modulus} \\ \eta &= \text{damping loss factor} \\ \nu &= \text{Poission's ratio} \\ I &= \text{Area moment of inertia} \end{aligned}$$

3.3.3 Narrow Tube Theory

In previous wave equation, effect of boundary layer is not included. In narrow slits, small tubes, effect of boundary layer becomes dominant. We need to include this effect in wave equation to get effective propagation constant. With the help of this propagation constant, transmission loss can be calculated. In boundary-layer effect, two losses need to be included, which are viscosity loss and thermal conduction loss [19].

Viscosity Loss

It is associated with stored kinetic energy, and it dissipates through viscosity of a medium at the boundary. The viscosity loss is expressed as series impedance and calculated [19] as eq. 3.15.

$$Z = j \left(\frac{\omega\rho}{\pi a^2} \right) (1 - F_v)^{-1} \tag{3.15}$$

Where,

$$F_v = \frac{2}{r_v\sqrt{-j}} \frac{J_1(r_v\sqrt{-j})}{J_0(r_v\sqrt{-j})}$$

$$r_v = a\sqrt{\frac{\rho_0\omega}{\mu}}$$

Where, a is the internal radius of the tube and μ is viscosity of a medium through sound is propagating.

Thermal Conduction Loss

This loss is associated with the potential energy in a medium due to compressibility. This energy associates with adiabatic condition at the boundary of a medium. This loss is expressed as shunt admittance. The following eq. 3.16 gives shunt admittance [19]. C_p is specific heat at a constant pressure and K is thermal conductivity of a medium.

$$Y = j\left(\frac{\omega\pi a^2}{\rho_0 c^2}\right)(1 + (\gamma - 1)F_t) \tag{3.16}$$

Where,

$$F_t = \frac{2}{r_t\sqrt{-j}}\frac{J_1(r_t\sqrt{-j})}{J_0(r_t\sqrt{-j})}$$

$$t_t = \sigma r_v$$

$$\sigma = \sqrt{\frac{\mu C_p}{K}}$$

So, finally equivalent impedance Z_0 and propagation constant Γ can be calculated from series impedance and shunt admittance as follows:

$$Z_0 = \sqrt{\frac{Z}{Y}} \tag{3.17}$$

$$\Gamma = \sqrt{ZY} \tag{3.18}$$

After simplification of propagation constant (Γ) expression [20] and it becomes:

$$\Gamma = k\sqrt{\frac{J_0\left(j^{\frac{3}{2}}r_v\right)}{J_2\left(j^{\frac{3}{2}}r_v\right)}}\sqrt{\frac{C_p}{n}} \tag{3.19}$$

Where,

$$n = \left(1 + \frac{C_p - 1}{C_p}\frac{J_2\left(j^{3/2}\sigma r_v\right)}{J_0\left(j^{3/2}\sigma r_v\right)}\right)^{-1}$$

In Appendix E, expression for equivalent impedance Y_0 is derived. Putting the value of Y_0 in eq. 3.8, transmission loss can be calculated for narrow tube effect as shown in eq. 3.20

$$TL_{NT} = 10log_{10}\left(\frac{1}{4}|2 + \tanh(\Gamma L)|^2\right) \tag{3.20}$$

3.3.4 Total Transmission Loss

Transmission loss calculated from mass law with inclusion of stiffness and damping and transmission loss due to effect of the narrow tube can be summed to calculate total transmission loss as given in eq. 3.21.

$$TL_{Total} = 10\log_{10}\left(10^{-0.1TL_{ML}} + 10^{-0.1TL_{NT}}\right) \tag{3.21}$$

Chapter 4

Numerical Models

4.1 Modal Analysis of Honeybee Hive

Honeybee hive shows tonal behaviour in absorption coefficient and good transmission loss. So modal analysis is done to calculate natural frequency and mode shapes of honeybee hive for unit cell and total sample. Unit cell model is used to verify the presence of local modes. Total sample modal analysis provides an idea of mode shapes and representation of a dipole. Acoustic modal analysis is done to verify resonator effect. Structural modal analysis is done in ANSYS APDL[21] and for acoustic modal analysis LMS Virtual.Lab [22] is used.

4.1.1 Structural Modal Analysis of Unit cell

Geometry

Unit cell of a honeybee hive is considered from periodic structure. This is a hexagonal cylinder with centre membrane. Distance between two opposite faces is 4.5 *mm*, length of half part of cell, i.e. from opening to membrane is 17 *mm*, thickness of complete structure is 0.217 *mm*. All sides of hexagons are equal and opposites are parallel to each other.

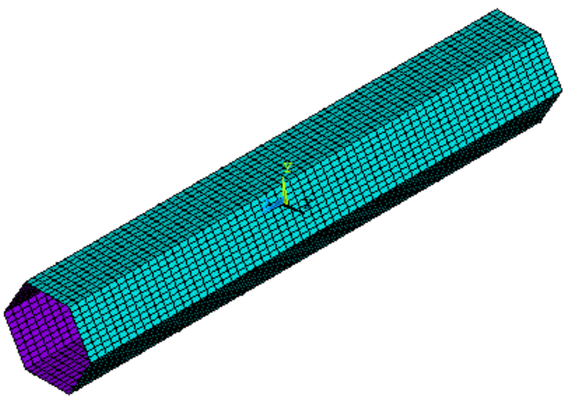


Figure 4.1: Unit cell mesh for structural modal analysis

Mesh Generation

Unit cell is modeled and meshed in ANSYS APDL. Since this mesh has to use for structural modal analysis, it should be sufficient fine mesh as all modes up to 6000 Hz capture. Lines forming hexagon are divided into 10 elements and lines along length of cell are divided into 30 elements. Model area is meshed using element Shell 63 with real constant equal to thickness of cell as shown in Figure 4.1.

Material Properties

Material properties used in the simulation are given in Table 4.1.

Boundary Conditions

In this analysis, assumption is that unit cell is simply supported in axial direction at the end. Ends of the hexagonal cylinder are constrained to translation, but free for rotation. Remaining part of unit cell is free to vibrate.

Table 4.1: Material properties

	Honeybee Wax	Steel	Aluminium		Air
Young's Modulus (GPa)	0.39	210	80	Speed of Sound (m/s)	340
Poisson's Ratio	0.326	0.3	0.33	Density (kg/m^3)	1.22
Density (kg/m^3)	854	7800	2800		

4.1.2 Structural Modal Analysis of Total Complete Sample

Geometry

Total sample is modeled with the help of the previous unit cell model. The same unit cell is copied at location such a way that slanted surface of unit cell will match with opposite slant surface of copied unit cell. Now copy this two areas in x and y direction in such a way that rectangular panel will form. To get the circular panel, select the areas which are not part of 100 mm diameter circle and delete it. Areas are selected using the cylindrical coordinate system. Multiple areas are formed in copying process. Multiple areas are combined to single area using overlap tool option. Prepared geometry is as shown in Figure 4.2

Mesh Generation

All lines forming hexagon are divided into three elements and all remaining lines in parallel to z axis are meshed to ten elements. Complete area of a circular panel model is meshed using element Shell 63. Thickness of the wall which is 0.217 mm is used as real constant for this element.

Boundary Conditions

In the impedance tube, sample is fixed along its circumference in a tube, and acoustic properties are measured. Same boundary condition is used for modal analysis. All nodes which are out of diameter of 100 mm is kept constraint in all directions.

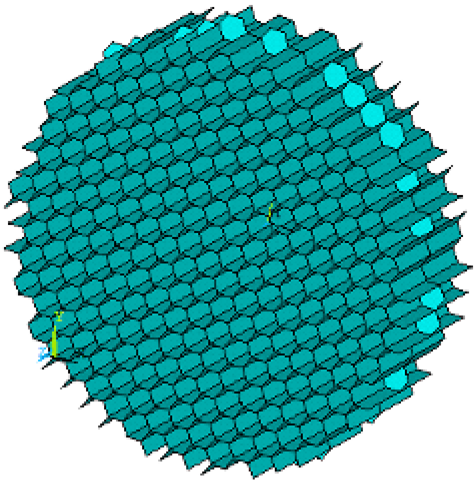


Figure 4.2: Geometry of total sample for structure modal analysis

4.1.3 Acoustic Modal Analysis of Hexagonal Tube

Geometry

Honeybee hive has hexagonal empty spaces in each unit cell to fill with honey. In experimental measurement of acoustical properties, it was filled with air. So geometry for acoustic modal analysis becomes a hexagonal tube with a dimension of air volume inside the unit cell of a honeybee hive.

Mesh Generation

In acoustic analysis mesh, requirement is to form an element of size, which is equal to one-sixth of minimum wavelength. Our maximum interested frequency is 6000 Hz. So the interested wavelength becomes 56 mm and element size becomes 9 mm. But cell size is 4.5 mm which is less than required element size. So that volume is meshed is such a way that at least six elements will be in all directions. Figure 4.3 shows the mesh for acoustic analysis. Using Brick 8 node element, volume is meshed.

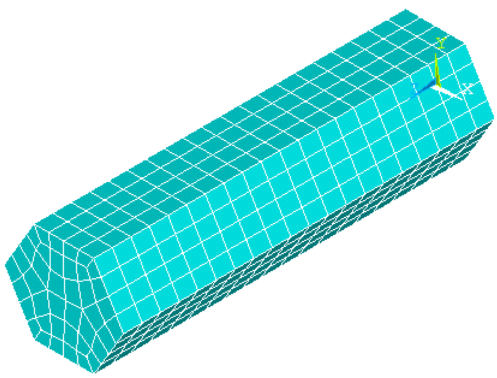


Figure 4.3: Hexagon cylinder mesh for acoustic modal analysis

Material Properties

These volumes are occupied in between hexagonal wall and membrane with one side open. It consists air, which is in room standard condition. For this analysis, air properties are considered, which are discussed in Table 4.1

Boundary Conditions

In acoustic analysis, boundary conditions are given in the form of pressure, velocity and impedance [22, 23]. For acoustic modal analysis; pressure is used as boundary condition. In this analysis unit pressure is applied at opening side of hexagonal cell.

4.1.4 Coupled Modal Analysis

This analysis is done to verify the coupling phenomenon between structure and acoustic modes. Uncoupled structure and acoustic mode sets are used, which is already solved earlier. Structure and acoustic meshes mapped at the boundary surfaces using mapping tool in LMS Virtual.Lab.

4.2 Vibro-acoustic Response Analysis

4.2.1 TL Calculation for Cylinder Open at Both Sides

In one of the hypotheses, dissipation of energy due to flexible structure and damping is discussed. This analysis is done to validate results obtained from analytical formulation for cylinder.

Geometry

The purpose of this analysis is to find out sound energy dissipation when sound waves pass through the flexible cylinder. To demonstrate this, considered model are two cylinders connected with flexible cylinder. Upstream and downstream cylinders have a thickness of 1 mm and radius 50 mm with length 200 mm. Middle flexible cylinder has a thickness of 0.1 mm and radius 50 mm with length 20 mm. These three cylinders connected to each other form structural model and air inside it, forms acoustic model.

Mesh Generation

Uncoupled structural and acoustical mesh as discussed in previous section have been used for coupled modal analysis. Structure and acoustic surface elements mapped to couple pressure and velocity in acoustic mesh and displacement of structure mesh as shown in Figure 4.4

Material Properties

Flexible cylinder material is aluminium, and other two cylinders material is steel. An acoustic medium is air. Other material properties are given in Table 4.1.

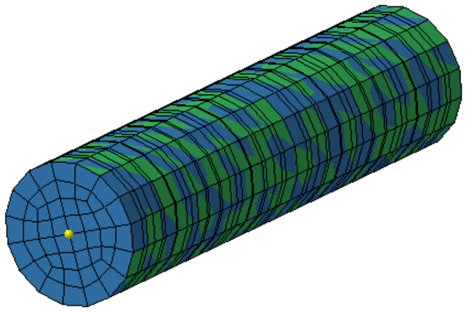


Figure 4.4: Structure and acoustic mesh of circular tube open at both sides

Boundary Conditions

In structural mesh, ends of cylinders are simply supported i.e. Ends of the cylinder are constrained to translation and free for rotation. Unit velocity at inlet face and characteristic impedance (ρc) as outlet boundary condition has applied in the acoustic model.

After solving for vibro-acoustic response analysis, pressure at the inlet (p_{inlet}) and outlet (p_{outlet}) is calculated[22]. These pressures are used to find transmission loss as given in eq. 4.1.

$$TL = 20\log_{10}\left|\frac{p_{inlet} + \rho c u_{inlet}}{2p_{outlet}}\right| \tag{4.1}$$

4.2.2 TL Calculation for Unit Cell of Honeybee Hive

Effect of membrane on TL is studied in this analysis. Here same structural and acoustic mesh is used as discussed in section 4.1. Material properties are same as Table 4.1. Mapping of mesh is done wherever acoustic mesh is in contact with structure mesh. Final mesh used for analysis is as shown in Figure 4.5. Three different models are simulated with three structure boundary conditions. All models are with simply supported boundary, only difference in edges of cell. Boundary conditions applied in first model at all sides (edges) of rectangles, in second model ends of cell and in third model edges of centre membrane. Acoustic boundary conditions are same as mentioned in section 4.2.1. After completion of vibro-acoustic response, calculated transmission loss using formula mentioned in eq. 4.1.

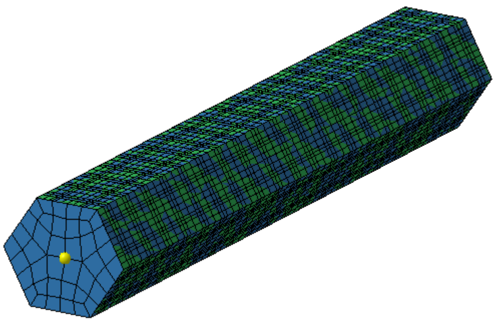


Figure 4.5: Coupled structure and acoustic mesh for unit cell of honeybee hive

Chapter 5

Results and Discussion

5.1 Repeatability

Experiments are carried out on honeybee hive samples 1 to 4 to verify the repeatability of absorption coefficient and transmission loss measurements. Measurements are repeated on three different days. It can be observed that results are repeatable with a minimum deviation in an audible frequency region. Figures 5.1, 5.2, 5.3 and 5.4 shows repeatability of the absorption coefficients for samples 1 to 4, respectively.

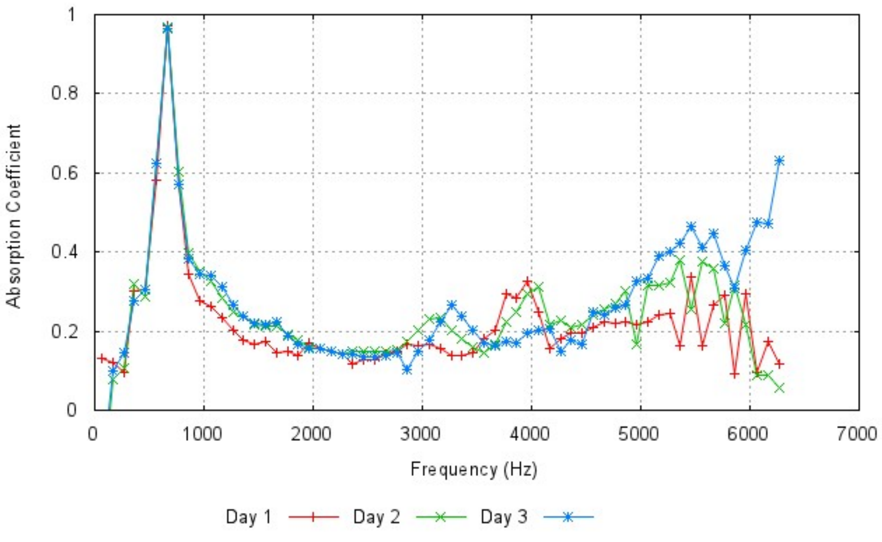


Figure 5.1: Repeatability test for absorption coefficient on sample 1

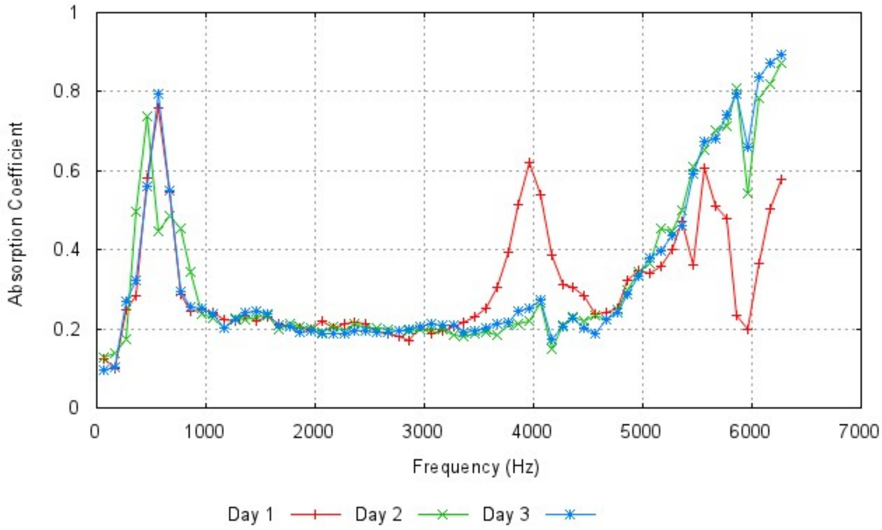


Figure 5.2: Repeatability test for absorption coefficient on sample 2

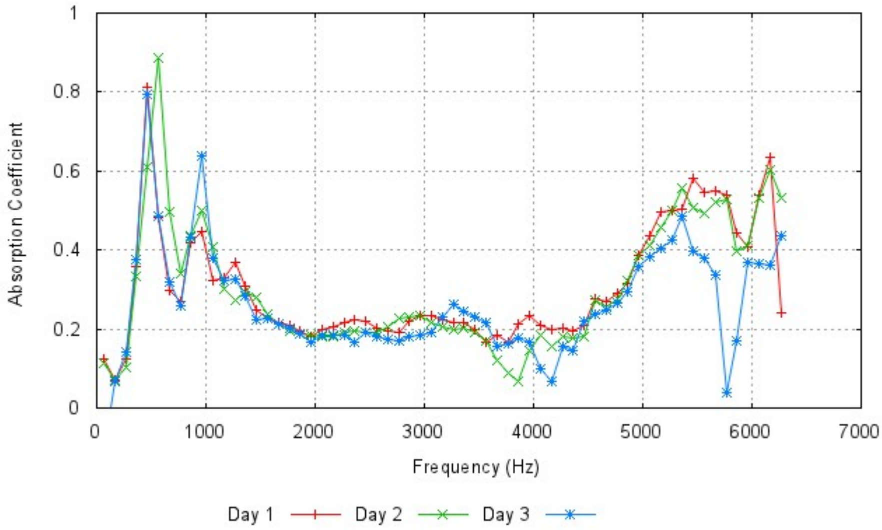


Figure 5.3: Repeatability test for absorption coefficient on sample 3

Figures 5.1, 5.2, 5.3 and 5.4 shows absorption coefficient of samples 1, 2, 3 and 4, respectively. It is observed that every sample shows tonal behaviour in between 350 to 700 Hz. Sample 1 showed high absorption coefficient 0.95 at frequency 700 Hz. Peak value of absorption coefficient observed in sample 2 and 3 is in between 0.8 and 0.9. It is similar with sample 4, but it is observed that one tone is split into two tones forming two peaks and one dip. Also in sample 3, two peaks and one dip is observed below 1000 Hz. In all samples, pattern of absorption coefficient is same with one or two peaks below 1000 Hz and after 1000 Hz constant coefficient of nearly 0.2 till 3500 Hz followed by one peak at 4000 Hz.

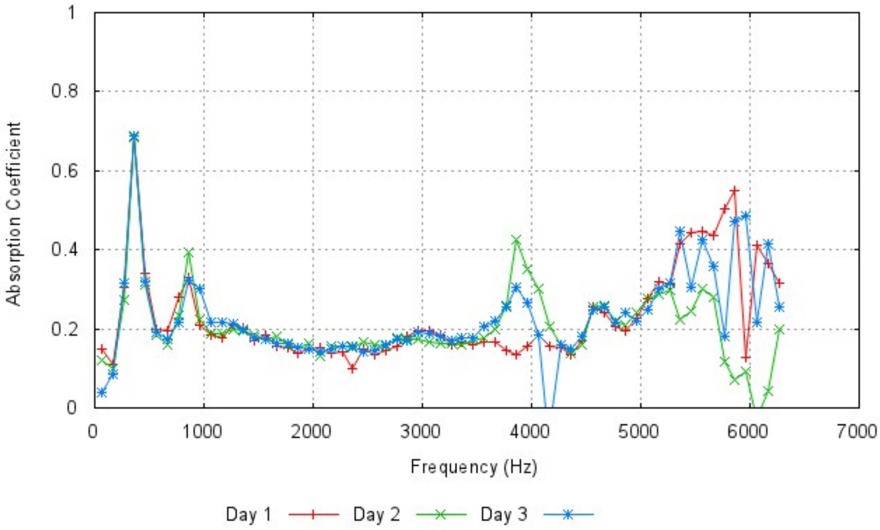


Figure 5.4: Repeatability test for absorption coefficient on sample 4

One interesting finding, about tonal behavior of the absorption coefficient is that, it is at a frequency of honeybee wing. So there is possibility that nature designed their hive in such a way that wing's vibrations and noise will be absorbed so that queen and bees staying in the hive should not disturb. Appendix A discusses about honeybee wing frequency and their hearing capability.

Sample 1 to 4 shows repetitive results in measurement of an absorption coefficient. Sample 1 and 3 shows perfect match of all peaks and dips and overall behavior. Sample 2 and 3 shows small deviation at 4000 Hz frequency, else all dips and peaks are matching.

Figures 5.5, 5.6, 5.7 and 5.8 shows measurement taken on day 1 to 3 for transmission loss of samples 1 to 4, respectively. This results shows repetition of value on all three days reading.

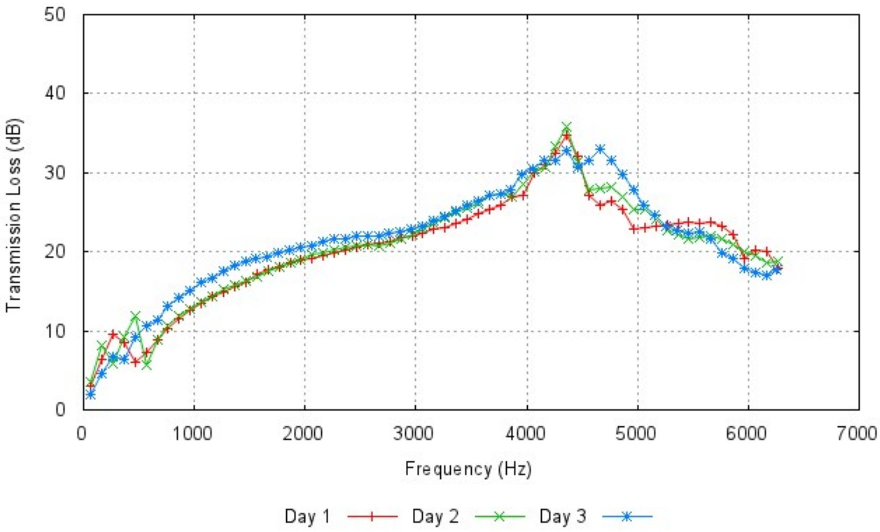


Figure 5.5: Repeatability test for transmission loss on sample 1

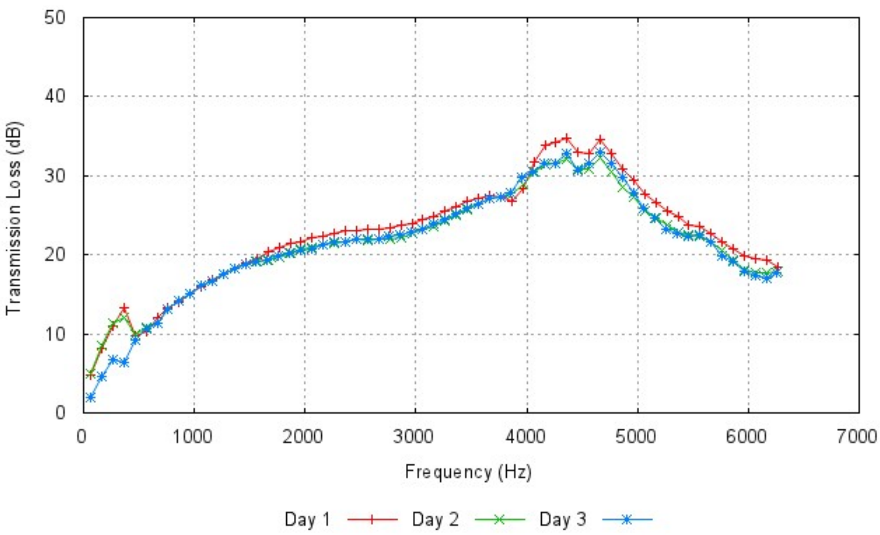


Figure 5.6: Repeatability test for transmission loss on sample 2

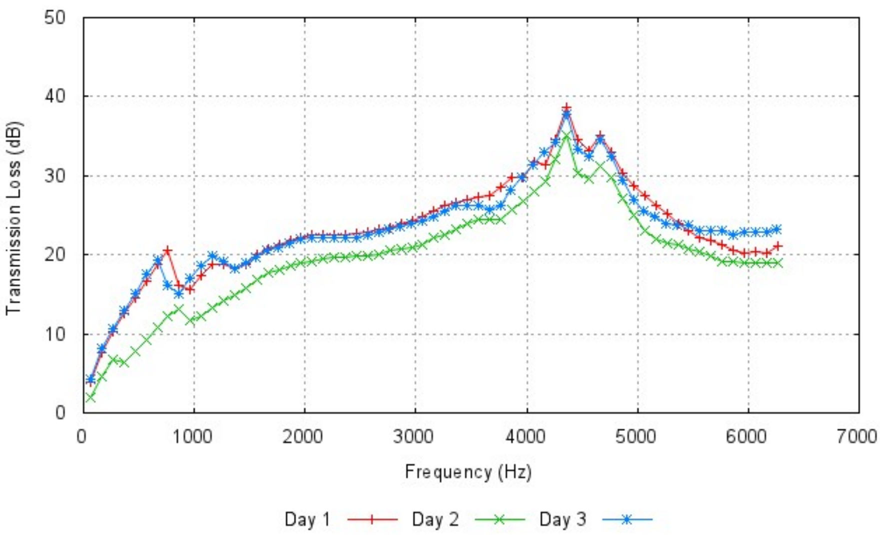


Figure 5.7: Repeatability test for transmission loss on sample 3

Figures 5.5, 5.6, 5.7 and 5.8 shows transmission loss of sample 1, 2, 3 and 4, respectively. All samples shows one peak in transmission loss below 1000 Hz and other at 4500 Hz. In sample 3, maximum transmission loss of 40 dB is observed, while other samples shows maximum transmission loss of 35 dB. Sample 3 and 4 gives peak near to 800 Hz with transmission loss of 20 dB. For all samples transmission loss below 1000 Hz is varying from 5 to 20 dB and above 1000 Hz, it's average value is 25 dB.

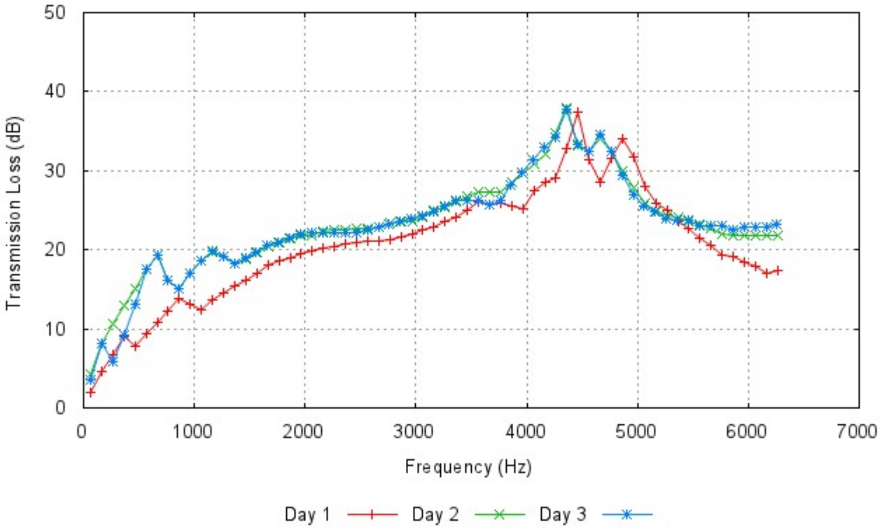


Figure 5.8: Repeatability test for transmission loss on sample 4

From above graph, it is concluded that results are consistent and whatever measurement process we followed has the optimum error.

5.2 Parametric Study using Honeybee Hive Samples

5.2.1 Cell Size

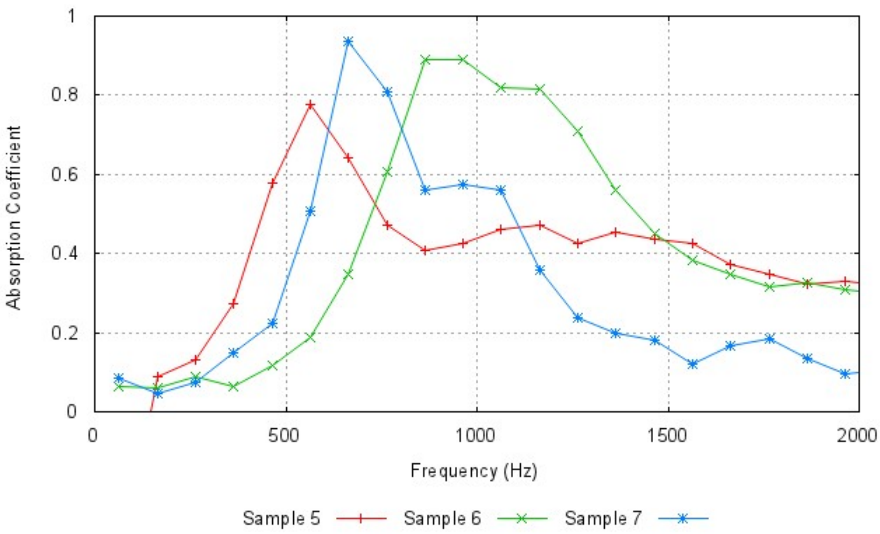


Figure 5.9: Effect of cell size on absorption coefficient

Figure 5.9 shows the effect of cell size on the absorption coefficient. It is clear that peak frequency is a shift accordingly cell size of sample. Cell size of sample 5 to 7 is increasing, also peak is shifting towards higher frequency. This means that according to interested frequency, we can choose cell size to get a higher absorption coefficient. Cell sizes of samples are in Table D.1. Bigger size cell samples broaden the effective frequency range and also peak value of an absorption coefficient.

Figure 5.10 shows transmission loss in samples 5 and 7, where sample 7′s cell size is more than sample 5. It concludes that as cell size increase, overall behavior of transmission loss dropped down in lower and higher frequency range and not significant change in mid frequency range. Also peaks shifted to lower frequency as cell size increased.

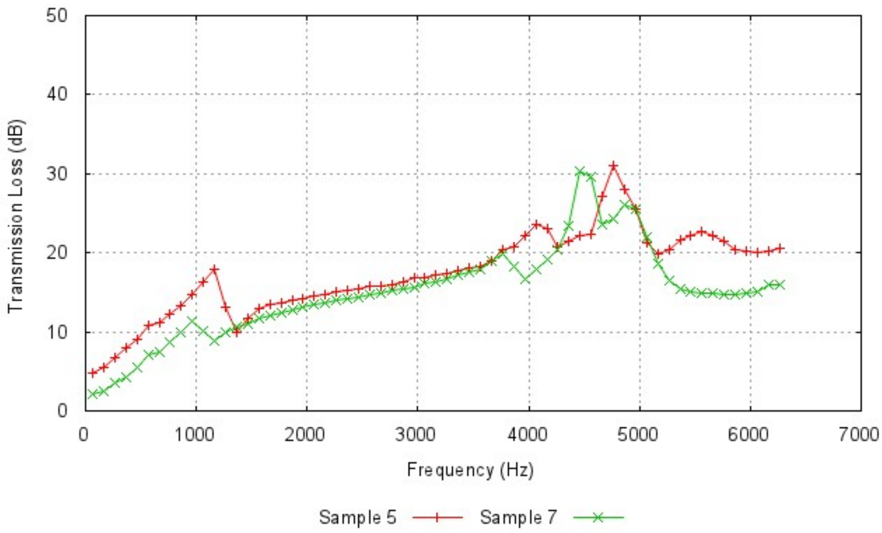


Figure 5.10: Effect of cell size on transmission loss

5.2.2 Cell Wall Thickness

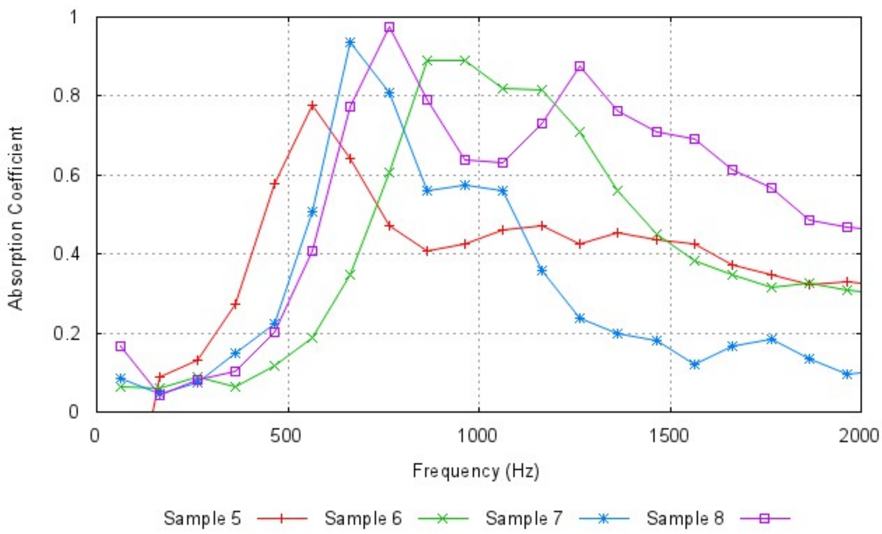


Figure 5.11: Effect of cell wall thickness on absorption coefficient

As shown in Figure 5.11, overall absorption coefficient is directly proportional to cell thickness. As the thickness increasing, absorption coefficient becoming broader and higher. Also peak shift is seen with the increase of thickness. Thickness of samples is in Table D.2.

Transmission loss of samples is dropping down as the thickness of cell increases. Figure 5.12 also shows that peaks are shifting towards lower frequency as cell thickness increases.

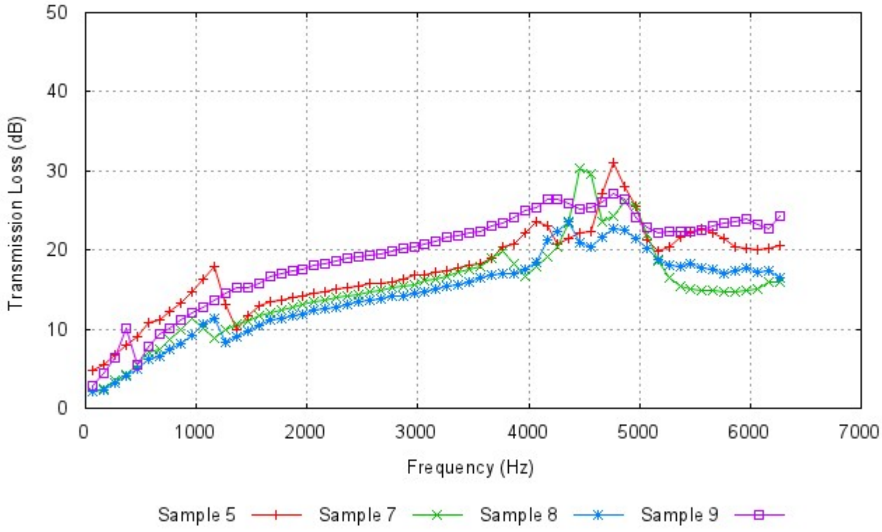


Figure 5.12: Effect of thickness on transmission loss

5.2.3 Number of Layers

This configuration shows the good amount of amplification in transmission loss and peak frequency of an absorption coefficient is shifted. Figures 5.13 and 5.14 shows the absorption coefficient of layer formed with samples 5, 6 and 7, 8, respectively. And Figures 5.15 and 5.16 shows transmission loss for layer of samples 5, 6 and 7, 8 respectively.

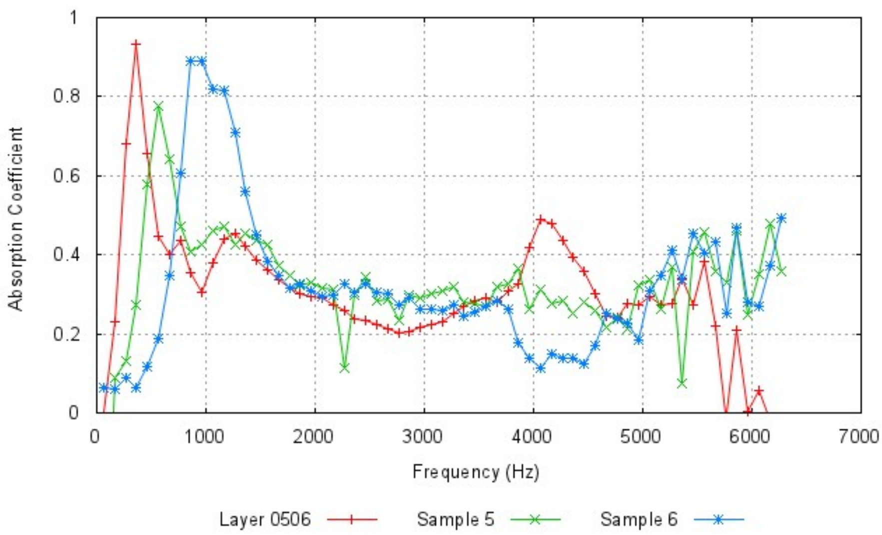


Figure 5.13: Effect of number of layers on absorption coefficient for sample 5 and 6

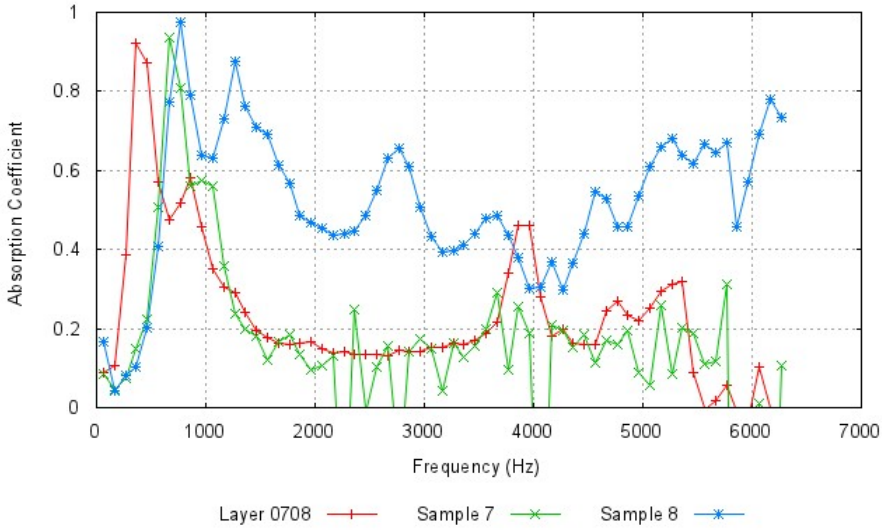


Figure 5.14: Effect of number of layers on absorption coefficient for sample 7 and 8

Results of the absorption coefficients shows dominant tonal behavior in multi layer configuration compared to single samples. If two samples attached in layers, it acts like impedances in parallel. Equivalent frequency calculated from the equivalent impedance, and it matches with experimental results. Each sample layer is approximated as equivalent spring and mass system and layer arrangement is represented as the series model. If the peak of the absorption coefficient is at f_a and f_b, then in layer that peak comes at $\frac{f_a f_b}{f_a+f_b}$.

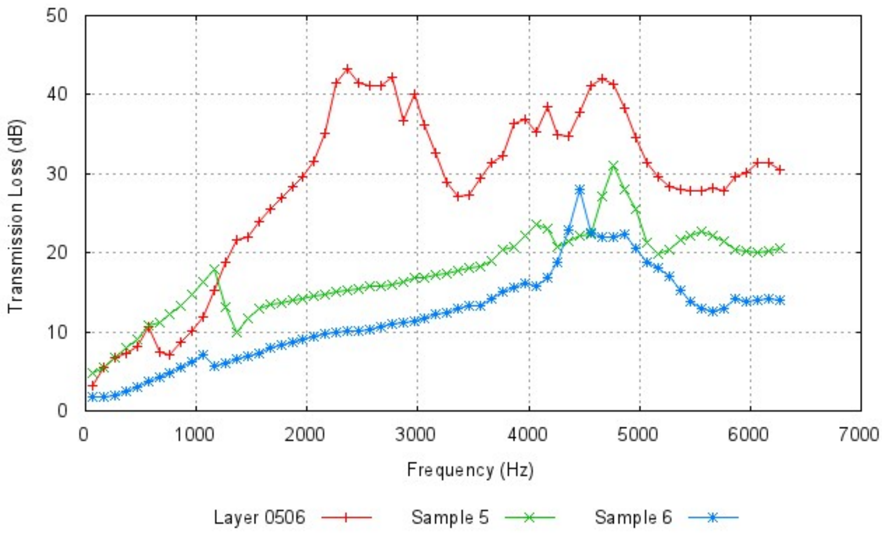


Figure 5.15: Effect of number of layers on transmission loss for sample 5 and 6

Along with stiffness, if we consider mass of sample, the expression for equivalent frequency becomes as shown in eq 5.1:

$$f_{eq} = \frac{f_a f_b}{\sqrt{m_a f_a^2 + m_b f_b^2}} \sqrt{\frac{m_a m_b}{m_a + m_b}} \tag{5.1}$$

Where, m_a and m_b are mass of first and second sample used in layer. For masses of samples refer to Table D.3. It is proved that if peaks are f_a and f_b for different samples, then in layer configuration that peak comes at f_{eq}.

Figures 5.15 and 5.16 shows transmission loss in layer configuration. For single sample, TL is in the range of 10 to 20 dB in the band of 2000 to 4000 Hz, which increases to 30 to 45 dB in layer configuration. Same way above frequency 4000Hz, TL is more than 30 dB. There is a small increase in TL below frequency 1000 Hz.

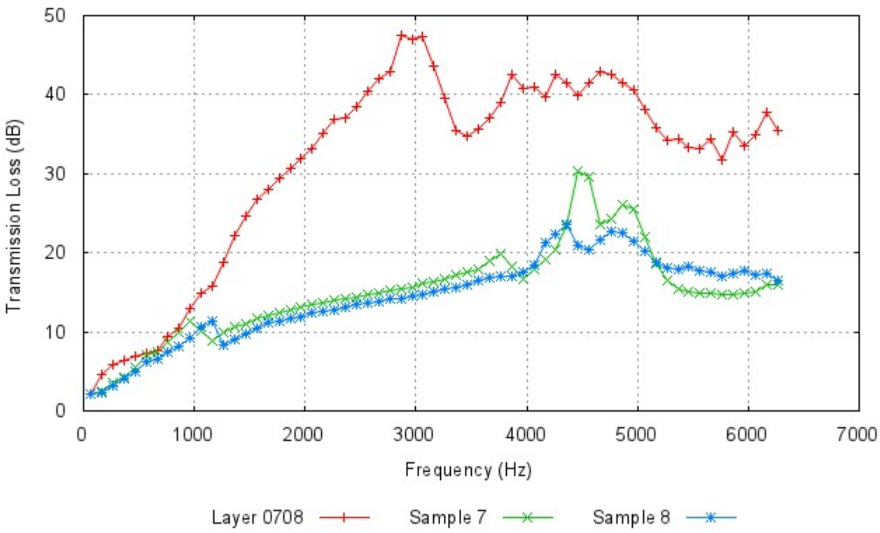


Figure 5.16: Effect of number of layers on transmission loss for sample 7 and 8

5.3 Parametric Study using Circular Cell Samples

5.3.1 Cell Size

Absorption Coefficient

Figure 5.17 shows the effect of cell size on the absorption coefficient for circular cell samples 1 to 4. It is observed that overall absorption coefficient increases as cell size increases and also peak frequency shifted towards higher frequency. The acoustical behaviors of circular cells are in consistent with natural honey bee hive samples.

Transmission Loss

Figure 5.18 shows transmission loss for circular cell samples 1 to 4. It shows as the cell size increases, overall TL decreases until 3000 Hz and after that it increases. Peak values are increasing and also shifted towards higher frequency with decreasing cell size.

5.3.2 Cell Wall Thickness

Absorption Coefficient

In Figure 5.17 shows the absorption coefficient for sample 2 and 3, but there is a shift of peak in the absorption coefficient towards lower frequency as the thickness increases. Moreover, overall behavior of the absorption coefficient decreases with an increase in thickness.

Transmission Loss

Figure 5.18 show that as the thickness increases transmission loss increase until 3000 Hz and after that it decreases. Peaks in transmission loss shift to higher frequency, when a thickness increase. Here, this behavior in the opposite to honeybee hive sample transmission loss behaviour.

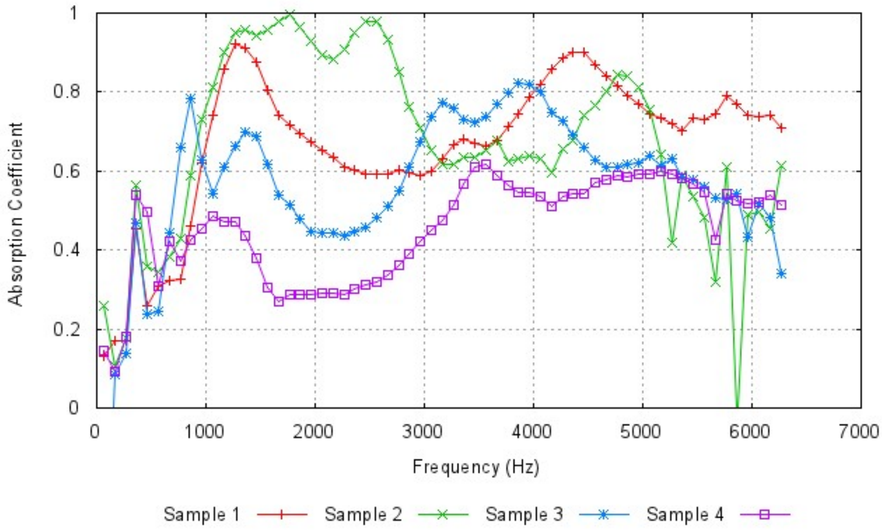


Figure 5.17: Absorption coefficient of circular cell samples 1 to 4

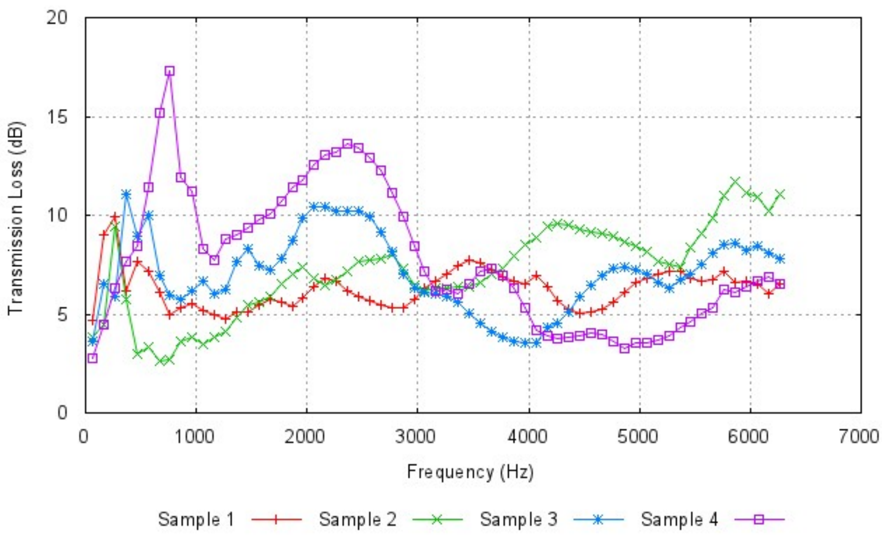


Figure 5.18: Transmission loss of circular cell samples 1 to 4

5.3.3 Membrane

Circular cell samples were prepared without membrane, and acoustic properties were measured. Both absorption coefficient and transmission loss values are as zero. It concludes that at least one membrane is required in between two arrays of periodic structure to get better acoustical properties.

5.4 TL using Mass Law, Narrow Tube and Experiment

Analytical modeling results have corroborated with experimental results for transmission loss of the honeybee hive. TL is calculated using mass law, narrow tube theory and combining both effect. Narrow tube formulation explains peaks at 4500 Hz in TL. In narrow tube theory formulation, very high TL is observed because of tube resonance. However, narrow tube formulation predicts lower TL values at other frequencies. Comparison of experimental and different analytical formulation results are shown in Figure 5.19.

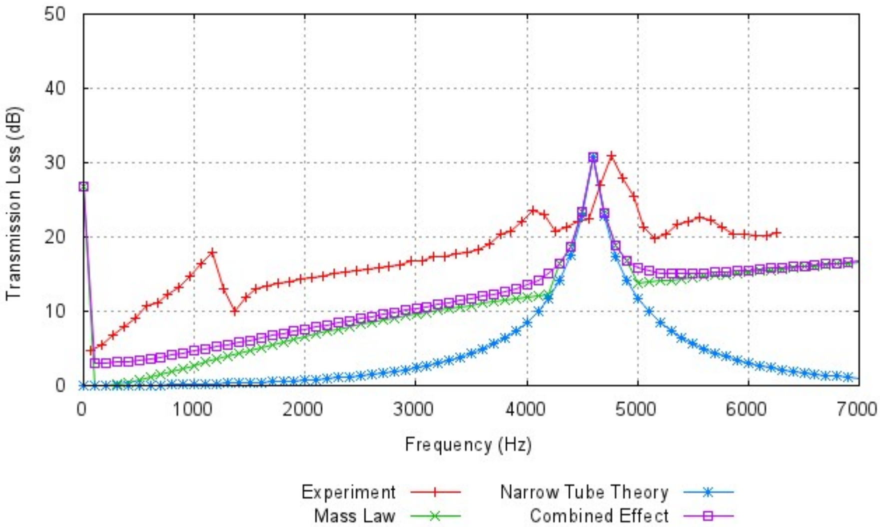


Figure 5.19: Validation of analytical models with experimental results for TL values

5.5 Analytical Results

5.5.1 Helmholtz Resonator

Calculated Helmholtz resonance frequency for hexagonal cell is 21730 Hz as discussed in section 3.2.1. It can be observed that resonant frequency is beyond audible frequency range. So, hypothesis based on Helmholtz resonator theory is not applicable.

5.5.2 Dipole Phenomenon

Dipole theory formulated here cannot be used as it is for honeycomb structure because of limitations. In formulation, it considered as inner wall will act as mass and wall as spring along with stiffness due to air inside tube. Dimensions of honeycomb are very less for assuming base dimension and excited mass dimension. Hence, hypothesis based on dipole theory is not applicable.

5.5.3 Dissipation of Energy

Analytical modeling has been developed for a open end cylinder with flexible walls, dissipating acoustic energy is not giving considerable results using derived formulation; axial transmission loss for cylinder is nearly zero, so there are not many contributions of only cylindrical flexible walls. So here conclusion is that we must add membrane wall in formulation to get experimental results for hexagonal cells.

5.6 Transmission Loss for With and Without Membrane Samples

Cylinder Open at Both Sides

Analytical and computational modeling has been conducted to verify the roll of membrane in transmission loss through flexible cylindrical tubes. Both models show zero transmission loss at all frequencies.

It is concluded that without membrane, it is not possible to get transmission loss for circular flexible tubes.

Hexagonal Cylinder with Membrane

This analysis is done using the computational model to check the effect of membrane in transmission loss. Computational model showed good transmission loss as shown in Figure 5.20. It shows different boundary condition effect on transmission loss. This behavior is same as the panel which shows a stiffness control region, damping region and mass control region.

With the help of computational method, it is proved that at least one membrane is required for getting transmission loss.

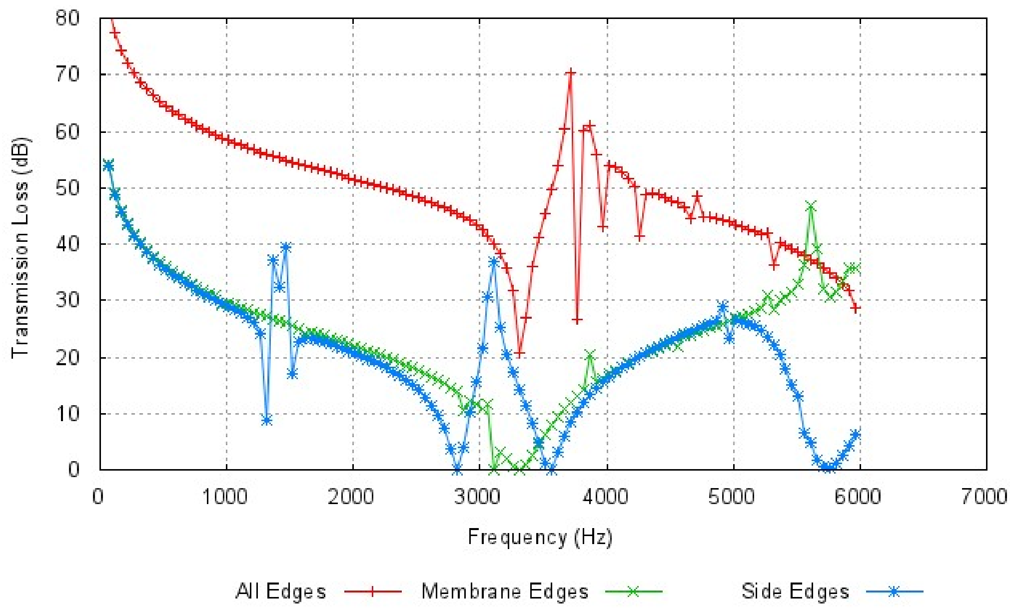


Figure 5.20: Effect of edge conditions on TL of hexagonal cell

5.7 Modal Analysis

5.7.1 Coupling between Structure and Acoustic

First three modes of a structural and acoustic model is shown in Table 5.1,which clearly shows that there is no coupling between structure and acoustic medium because first mode of structure and acoustic frequencies are well separated.

Table 5.1: Structural, acoustic and coupled modes

Mode	Structural(Hz)	Acoustical(Hz)	Coupled (Hz)
1	611	4935	610
2	960	14804	960
3	1604	24656	1604

5.7.2 Unit Cell

Modal analysis for unit cell is done for dimensions of honeybee hive sample 6. This sample shows tonal behavior at frequency of 630 Hz. Also for unit cell first modes at 633 Hz which is moving in transverse direction of sound propagation as shown in Figure 5.21. So it is concluded that this mode is helping for getting better absorption coefficient.

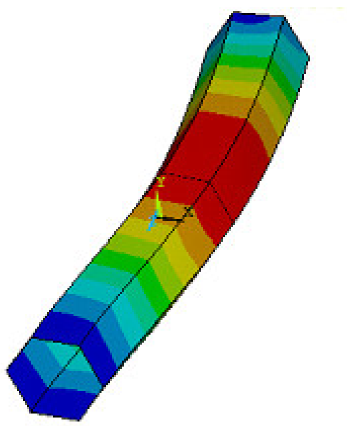


Figure 5.21: Mode Shape at 633 Hz for Unit Cell

5.7.3 Total Sample

In every sample, there is one peak in between 1000 to 1500 Hz. Modal analysis of total sample shows that one mode at 1213 Hz, which can be assumed as a dipole. In this, mode shape (0, 2), circular panel got divided into four parts with adjacent part in out of phase, i.e. one is moving in and other is moving out. It is concluded that dipole phenomenon works for getting transmission loss. Theoretical formulation is required to prove this theory.

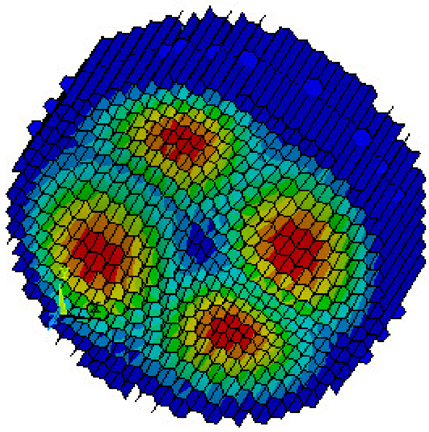


Figure 5.22: Mode shape at 1213 Hz for total sample

Chapter 6

Summary and Future Works

6.1 Summary

New approach for design of an acoustical panel, especially for low-frequency noise problem have been discussed in this thesis. It describes natural honeybee hive can be mimicked in real life as the acoustical panel. Acoustic properties of the honeybee hive are measured, and experimental results have used to validate analytical and numerical models.

Honeybee hive has very good acoustic properties, especially at low frequency. It shows tonal behavior in the absorption coefficient between 350 to 800 Hz with peak value 0.9. Its transmission loss is about 5 to 15 dB below 1000 Hz frequency, which is one of the good acoustic materials for low-frequency noise control problems. Furthermore, it shows nice transmission loss at higher frequencies about 40 dB. Layer configuration is a very efficient method to shift absorption coefficient to a lower frequency region. This configuration causes the drastic increase in transmission loss. It can be amplified from 20 dB to 40 dB using layers of the honeybee hive. Along with hexagon, different shapes can be used for preparing acoustic panels. Circular cell samples show a very high absorption coefficient above 0.9 for a broad band of frequency. Transmission loss is in between 5 to 15 dB, which are less as compare to hexagon shape, but can be improved with help of different sample material. Experiments and theoretical models proved that at least one membrane is required in arrangement of the array of periodic cells. Cell size and thickness can be used as a parameter to vary transmission loss and absorption coefficient accordingly.

6.2 Future Works

- Develop an analytical formulation which will be a function of frequency, structural properties and transmission loss. It will be useful to do optimization of dimensions and acoustical properties.

- Prove that tonal behavior of an absorption coefficient at particular frequency is because of mode shape, which shows movement in transverse direction. Develop theoretical formulation to do free vibration analysis of honeybee hive samples.

- There is a scope to improve theoretical model validation by considering bulk properties of structure.

- Develop a theoretical acoustical model which incorporates structure periodicity.
- An experimental study will be required for large panel structure with proposed design.

Appendices

Appendix A

Honeybee Hearing Capability

Honey bees speaks by Wangle Dance. According to literature, bees hear by Johnstons organ (JO) in the pedicel of the antenna as shown in Figure A.1.

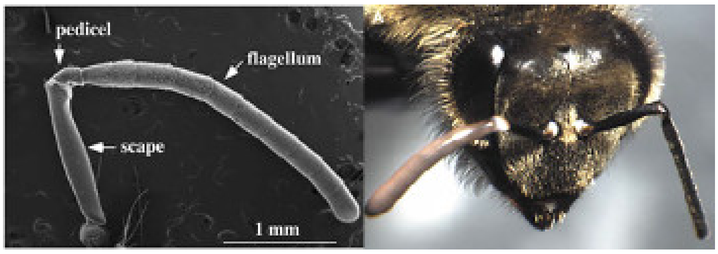


Figure A.1: Hearing system of honeybee

JO consists of about 300-320 scolopidia connected with about 48 cuticular knobs around the circumference of the pedicel JO neurons are best tuned to detect 250-300 Hz sound.

Frings and Little (1957) [24], induced vibrations of the combs by loud airborne noise and made use of the freezing response of worker bees to estimate the frequency range(300–1000 Hz) to which the bees respond behaviorally. They found the highest sensitivity at 500 Hz. Michelsen et al (1986b) [25] , who used the freezing response to vibrations of the combs of known amplitudes to study the behavioral thresholds. They showed that physiological threshold is lowest at 2.5kHz, whereas the behavioral threshold is lowest at about 500 Hz. Michelsen et al, (1987)[26], the dance sound signals are emitted as airborne sound by dorsoventral vibrations of the wings where frequency is 200–300 Hz. The sound consists of short pulses at a repetition rate of 15Hz. W. H. Kirchner (1990) [27], said bees are able to detect sound frequencies up to about 500 Hz. W. H. Kirchner (1993), concluded the fundamental frequency of bees beeping sounds was in 350–450 Hz.

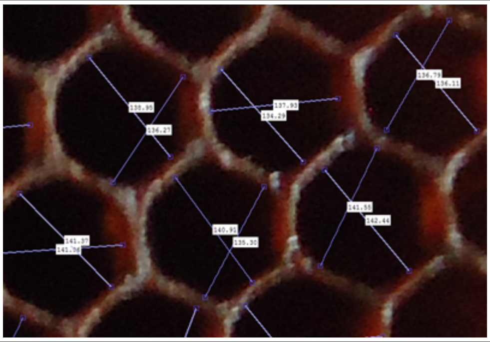
Seeing this literature, we are able say about tonal behavior of absorption coefficient in between 300–500 Hz frequency range and wing frequency of honeybee, there is some relation which is not known.

Appendix B

Statistical Analysis

All dimensions are in *mm*.

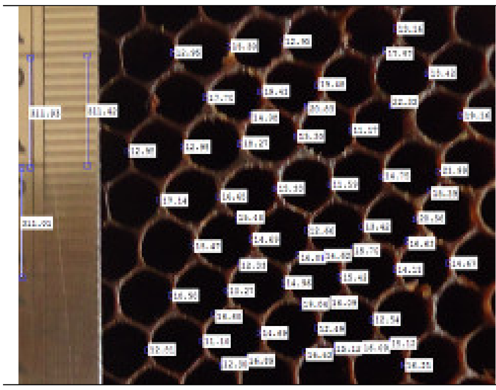
Table B.1: Neck dimensions



• No. of readings	50
• Average	4.521106109
• Standard Deviation	0.165279015
• Maxima	5.289389068
• Minima	4.179421222

Confidence Level %	Margin of error	Lower Limit	Upper Limit
99	0.060304875	4.4608012	4.581411
95	0.045813006	4.4752931	4.566919
90	0.038450201	4.4826559	4.559556

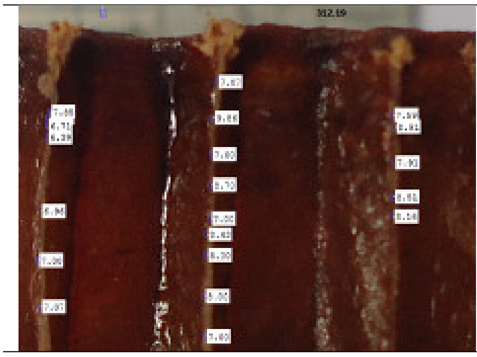
Table B.2: Neck thickness



• No. of readings	50
• Average	0.486231511
• Standard Deviation	0.087619148
• Maxima	0.707073955
• Minima	0.33022508

Confidence Level %	Margin of error	Lower Limit	Upper Limit
99	0.031969344	0.4542622	0.518201
95	0.024286789	0.4619447	0.510518
90	0.020383555	0.465848	0.506615

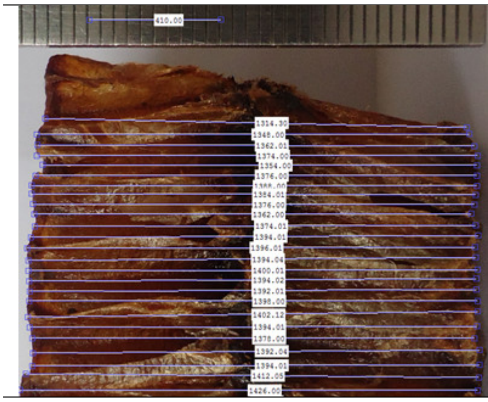
Table B.3: Wall thickness



- No. of readings 50
- Average 0.217089991
- Standard Deviation 0.046891328
- Maxima 0.315833307
- Minima 0.129728691

Confidence Level %	Margin of error	Lower Limit	Upper Limit
99	0.017109103	0.1999809	0.234199
95	0.012997613	0.2040924	0.230088
90	0.010908711	0.2061813	0.227999

Table B.4: Height



- No. of readings 50
- Average 34.44692683
- Standard Deviation 0.863856366
- Maxima 35.75634146
- Minima 32.05609756

Confidence Level %	Margin of error	Lower Limit	Upper Limit
99	0.315193	0.315193	34.76212
95	0.239449	0.239449	34.68638
90	0.200966	0.200966	34.64789

Appendix C

Experiment Procedure

C.1 Absorption Coefficient

Procedure of measurement as is as follows.

1. Do connections of whole setup as shown Figure 2.5 and place sample in tube.
2. Goto absorption coefficient measurement in VA-One software, select tube type and start speaker. Wait for some time until spectrum get stabilizes and after that stop the speaker. This readings will be used for calculating transfer function of SPL between two microphones.
3. Now change microphone position and repeat above procedure to measure transfer function. This reading will help to correct error produced due to phase mismatch in microphones.
4. Now calculate final transfer function and calculate absorption coefficient. Add this result to average.
5. Repeat above procedure to get good averaged result.

C.2 Transmission Loss

In this experimental work, impedance tube is used with extension tube and two more microphones.

1. Do connections of whole setup as shown Figure 2.6 and place sample in tube.
2. Goto insulation loss measurement in VA-One software, select tube type and start speaker for without end boundary condition. In this reading, remove the cap of extension tube. Wait for some time until spectrum get stabilizes and after that stop the speaker. This readings will be used for calculating auto spectrum and cross spectrum between four microphones.
3. Now attach cap to extension tube and repeat above procedure to get auto spectrum and cross spectrum between four microphones for with end boundary condition.
4. Using this two readings, four pole parameters and use them for calculation of transmission loss. Add this result to average.
5. Repeat above procedure to get good averaged result.

Appendix D

Dimensions and Mass of Samples

Table D.1: Cell size of honeybee hive samples

Sample	Cell Size (mm)			
	Large	% Change	Small	% Change
5	5.52	0	5.611	0
6	5.724	3.7	5.765	2.7
7	5.697	3.2	6.097	8.7
8	5.316	-3.7	5.723	2
9	5.379	-2.6	5.652	0.7
10	5.688	3	5.766	2.8

Table D.2: Thickness of honeybee hive samples

Sample	Thickness (mm)			
	Large	% Change	Small	% Change
5	0.665	0	0.568	0
6	0.737	10.9	0.76	33.7
7	0.672	1	0.665	17.1
8	0.781	17.5	0.768	35.2
9	0.439	-34	0.664	16.8
10	0.619	-6.9	0.618	8.9

Table D.3: Mass of honeybee hive samples

Sample	Mass (g)	
	Large	Small
5	18.2	2.08
6	20.9	2
7	20.9	2.04
8	21.69	2.17
9	19.77	1.34
10	39.65	1.83

Appendix E

Analytical Formulation

E.1 Characteristic Impedance

For narrow tube of length L including end correction, acoustic pressure and particle velocity at upstream and downstream for tube can be written as eq. 3.7.
Where,

$$\begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} = \begin{bmatrix} \cosh(\Gamma L) & jY_z \sinh(\Gamma L) \\ \frac{j\sinh(\Gamma L)}{Y_z} & \cosh(\Gamma L) \end{bmatrix}$$

Acoustic impedance at opening of tube Z_1 can be calculated as $Z_1 = \frac{T_{11}}{T_{21}}$. In periodic structure, impedance for tubes are in parallel can calculated as $Z_A = \frac{Z_1}{N}$, where N is number of tubes and transfer matrix of parallel impedance can be written as eq E.1

$$TM = \begin{bmatrix} 1 & 0 \\ \frac{1}{Y_0} & 1 \end{bmatrix} \tag{E.1}$$

Where, Y_0 is equivalent characteristic impedance for periodic structure and can calculated as $Y_0 = \frac{Z_A}{S}$. S is cross-sectional area of tube and $S_0 = NS$ is cross-sectional area of of periodic structure. So equivalent characterict impedance can be writtern as eq. E.2

$$Y_0 = \frac{Z_c}{S_0} \coth(\Gamma L) \tag{E.2}$$

E.2 Equivalent Surface Density

Honeycomb have different mass along cross sectional. So equivalent surface mass density ρ_{eq} is required to calculated to use in mass law. As shown in Figure E.1, cell is consider for calculating equivalent density of honeycomb sample.

First volume of each part is calculated and multiplied with density of honeybee hive material to get mass of that part. Total mass is calculated by adding all masses. Now, to calculate equivalent surface density this total mass is divided by surface area of unit cell considered as shown Figure E.1, which is rectangle.

Volume of hexagonal walls = $6a_{cell}tt_c$ where, t is thickness of wall and t_c is thickness of honeybee

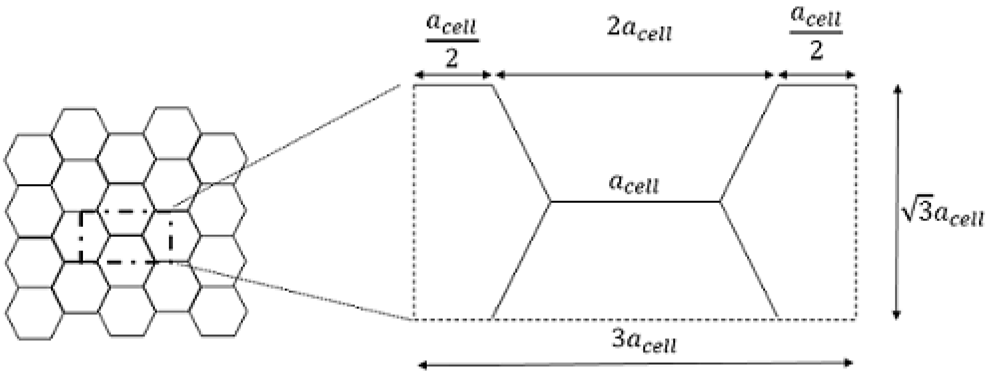


Figure E.1: Cell considered for equivalent density calculation

hive core i.e. half length of honeybee hive along cross section. And middle membrane volume $= 3\sqrt{3}a_{cell}^2 t_f$. Here t_f is thickness of membrane. So total mass of unit cell can be calculated as $\rho(2(6a_{cell}tt_c) + 3\sqrt{3}a_{cell}^2 t_f)$

Equivalent mass density can calculated as dividing cross sectional area $3\sqrt{3}a_{cell}^2$. After simplifying expression

$$\rho_{eq} = \rho \left(\frac{4}{\sqrt{3}} \frac{tt_c}{a_{cell}} + t_f \right) (t_f + 2t_c) \tag{E.3}$$

E.3 Area Moment of Inertia

For calculating stiffness, area moment of inertia is required. It can be calculated by using simplified Figure E.2 of honeybee hive.

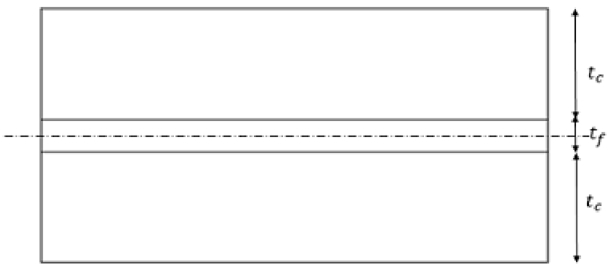


Figure E.2: Cut section of honeybee hive for calculating second moment of area

Area moment of inertia for honeybee hive is calculated as

$$I = \frac{t_f^3}{12} + \frac{t_f^2 t_c}{2} + t_f t_c^2 + \frac{2}{3} t_c^3 \tag{E.4}$$

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