University of Saskatchewan

Department of Mechanical Engineering

ME 497 – Acoustics and Vibrations in Design

Modal Analysis Laboratory

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Date of Submission: September 27, 2025

No table of Contents or abstract

general report formatting is

- Abstract
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Abstract/Summary: 2/4
Explanation of Lab: 5.5/6
Results & Discussion: 5/6
Writing Quality: 2/2
Overall impression: 2/2

Total 16.5/20

Introduction

Very long first sentence

Vibrations play a large role in modern designs for structures and machinery, since ignoring can result in various negative consequences ranging from annoying noises to catastrophic failure, which can cause property damage and present a severe danger to the public. The main issue with vibration is an external force causing a structure to oscillate near its natural frequency which causes the amplitude of the oscillations to become very large, inducing large stresses on the structure and often leads to rapid failure ~~of the structure~~ Unnecessary. To prevent these failures in future designs, this lab will investigate the modes of vibration for a plate under various support conditions to obtain the resonant frequency.

Purpose

To analyze the modes of vibration of two plates with different support conditions by measuring the acceleration response produced by the excitation at several points using an impact hammer.

Theory

Maybe since was supposed to be Continuous? Either way I'm not sure what the sentence means

Since structures can have an infinite number of natural frequencies because they depend on the mode of vibration. To identify these modes of vibration and the natural frequency, experimental modal analysis will be used by applying a known force to a known location of the plate and measuring the acceleration response of the plate. With the measured data, a frequency response function can be made which allows us to extract several parameters including the natural frequencies, amplitude of vibration and the damping ratio. The damping ratio can be found using the following formulas:

$$Q = \frac{f_n}{(f_2 - f_1)}$$

Little more detail on the FRF would be nice, sample image maybe

$$\zeta = \frac{1}{2Q}$$

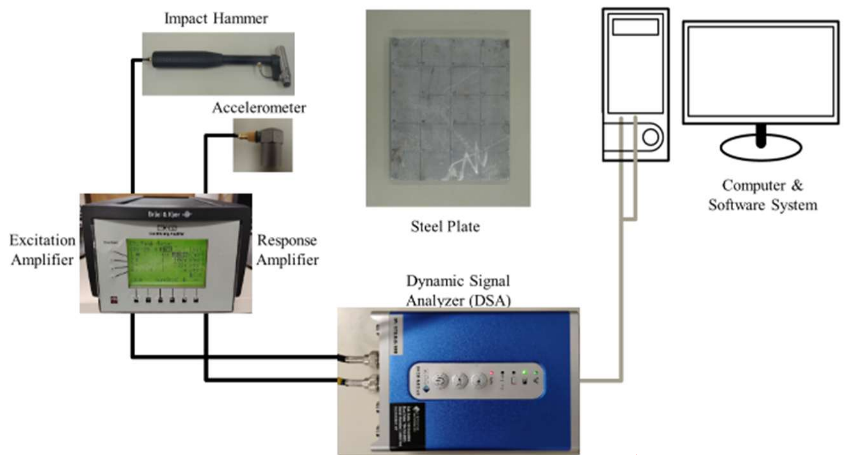
Where f_n is the resonant frequency and f_1 and f_2 are the half power points where the amplitude is equal to 0.707 times the peak response.

No discussion of damping ratio/mode shape

Experimental Investigation

Apparatus

A diagram of the experimental setup is shown in Figure 1. The diagram was obtained from ME 497 Acoustics and Vibrations in Design Modal Analysis Lab Manual.



Take a picture next time & label yourself

Figure 1: Experimental setup (Bitner and Fotouhi 2025)

Equipment

- Impact hammer (B&K 8202)
- Accelerometer (B&K 4371)
- Dynamic System Analyzer (Crystal Instruments Spider–HE)
- EDM-Modal software (Crystal Instruments)
- Amplifier (B&K 2692)
- Computer with Windows 11 operating system

Procedure

Don't direct reader away from report, ref is enough

A quick overview of the lab procedure is presented below. For more in-depth procedures, see ME 497 Acoustics and Vibrations in Design Modal Analysis Lab Manual (Bitner and Fotouhi 2025).

-Start the EDM-Modal program and select "Hammer Impact"

-Set up the plate geometry

-5" width, 4" height with a width count of 5 and a height count of 4

-Enable the *Measurement Point#* option under the *Point* menu

-The accelerometer should be placed at the corner corresponding to point 1 of the plate

-Select *Input Channel Setup*, Channel 1 should be the impact hammer and Channel 2 should be the acceleration response

-To set up the test select the *Measurement tab*, followed by *Measurement Point* and *Roving Setup*

> -Set *DOF's* to "+1Z", *Meas. Point* to "1", *Coordinate* to "+Z", *Roving Increment* to "1" for Ch1 and "0" for Ch 2, finally Ch1 is the "Excitation" and Ch2 is the "Response"

-Open *Control Panel* and set the *Frequency range(Hz)*: to "4.5(kHz)", the *Block size/Line* to "4096/1800", *Window* to "Force-Exponential" and the *Average mode* to "Linear".

-To start the test on the free plate press *Connect*

-The press *Trigger,* this allows you to set the force that the software will record as a hit

> -Strike the plate at point 2 with the hammer and follow the instructions on the window. It will take at least 5 hits to set the trigger

-Press *Run* and set the test directory, then follow the instructions in the window which entails striking the shown point, viewing the data and either accepting it and moving on or striking the point again

-Press stop after all points have been struck

-Press the *Modal Data Selection* tab and click *Validate*

-Press the *Band Selection* tab and make sure the *End frequency (Hz):* is "4497.50" and the *Display style* is "Complex MIF"

-press the *Stability Diagram* tab followed by the *Calculate stabilization* button

-Press the *+* buttons that show up on the diagram that are closest to the peaks of the curve

-Blue lines will appear and these represent the natural frequencies of the plate

-Press *Calculate modes* and a new window will appear showing the modes of vibration, damping ratios and an animation of the mode

-Repeat for a cantilever plate and a cantilever plate where an extra corner is fixed

Table names are generally below table

Results

Table 1: Experimental natural frequencies and damping ratios for the free plate

MODE	NATURAL FREQUENCY [HZ]	DAMPING RATIO [%]	MODE SHAPE (IF AVAILABLE)
1	1070.789	0.35	Error! Reference source not found.
2	1668.333	1.222	Error! Reference source not found.
3	2168.352	0.327	Error! Reference source not found.
4	3060.985	0.82	
5	3605.919	0.608	

Table 2: Experimental natural frequencies and damping ratios for the cantilever plate

MODE	NATURAL FREQUENCY [HZ]	DAMPING RATIO [%]	MODE SHAPE (IF AVAILABLE)
1	68.094	2.203	Figure 7
2	184.956	0.734	Figure 8
3	394.934	2.198	Figure 9
4	641.571	0.3	Figure 10
5	756.732	0.342	Figure 11
6	1317.332	1.016	Figure 12
7	1823.916	1.941	Figure 13
8	2392.116	1.513	

Ref Appendix too

Table 3: Experimental natural frequencies and damping ratios for the cantilever plate with extra fixed point

MODE	NATURAL FREQUENCY [HZ]	DAMPING RATIO [%]	MODE SHAPE (IF AVAILABLE)
1	127.371	3.414	Figure 15
2	408.144	0.969	Figure 16
3	685.655	0.17	Figure 17
4	1152.679	2.068	Figure 18
5	1334.843	0.295	Figure 19

Table 4: Calculated damping ratios based on modes 3 and 4 for the cantilever plate. Sample calculations shown in Appendix 4: Damping Ratio Calculations

MODE	DAMPING RATIO [%]	CALCULATED DAMPING RATIO [%]	PERCENT DIFFERENCE [%]
3	2.198	2.41	9.20
4	0.3	0.546	58.2

Discussion

The mode shapes found for each case during the lab were reasonable. For the free plate the first mode of vibration occurs about the axis perpendicular to the longest side length as it has a lower area moment of inertia, $\frac{1}{2}hb^3\ vs \frac{1}{2}bh^3$ where b is the longer side length. Figure 22 shows the theoretical mode shape based off the equations presented in the ME 497 Acoustics and Vibrations in Design Modal Analysis Lab Manual. The equations presented are for a beam, but for the first mode matches the findings for the plate. For higher order modes there would be more deviation from the theoretical mode shapes as the plate does not satisfy the Euler-Bernoulli beam assumption.

some details on the actual shape ie V, S, W or 1 node, 2 node, etc

Experimentally, Figure 3 shows that the first mode of vibration is about the axis perpendicular to the longest side length. The second mode of vibration (Figure 4) is about the axis parallel to the longest side length. The second mode makes sense as it is likely the second easiest way for the plate to bend. The third mode (Figure 5) appears to be a combination or superposition of the first two modes, it is reasonable to assume that this would be the case.

For the cantilever plate the expected first mode would be for the free end to vibrate since one end is fixed, shown in Figure 7, which aligns with the theoretical first mode shape presented in Figure 23. The expected second mode is more difficult but the findings during the lab show that it follows a similar trend, with the sides vibrating about the axis perpendicular to the fixed end (Figure 8). The third mode (Figure 9) looks to be an extension of the first mode with vibration starting closer to the fixed end and the magnitude of the vibration being a lot larger than the first mode. Appendix 2: Experimentally Determined FRF and Mode Shapes for a Cantilever Plate contains some images of higher order modes for the cantilever plate. The mode shapes for the cantilever plate are the best-looking mode shapes obtained during this lab. The higher order mode shapes look reasonable and there are no unexpected or weird features.

discuss relative frequencies to cantilever, were they similar?

For the cantilever plate with the extra fixed point the first mode of vibration is straight forward. Since one other corner is fixed the easiest way for the plate to vibrate is for the free corner to vibrate, as shown in Figure 15. The second mode makes sense as well with the edge between the fixed corner and fixed edge of the plate vibrating, Figure 16. However, after this it is hard to tell how valid the next modes are as it is not intuitive about which way the plate will vibrate in this case. There are some unexpected results where a node with a large magnitude of vibration, but the surrounding nodes seem unaffected. Modes 4 (Figure 18) and 5 (Figure 19) have nodes with large magnitudes of vibration. These results were not expected, and further testing would have to be done to determine if the results are accurate.

Like what?

Is this true? Does the software account for this?

Is it the force that's the problem or saturating the readout?

There are sources of error that are prevalent in this lab. There is always the possibility of uncalibrated instruments, uncertainty in the measurement accuracy of these instruments, but the largest source of error is human error. Human error comes into play when hitting the plate with the hammer. Ideally the hammer strikes the plate with the same amount of force, and in the exact same way every single time. However, there were multiple measurement points taken during the lab that were questionable because time was a constraint. This error from the hammer strikes can be seen in the differences between the stability diagrams shown in Figure 2 (free plate), Figure 6 (cantilever plate), and Figure 14 (cantilever plate with extra fixed point). Notice how the stability diagram for the cantilever plate has a lot more peaks, leading to more modes of vibration. The modes of vibration found in this lab for the cantilever plate also looked nicer. It should also be noted that the data for the cantilever plate was provided by Doug Bitner. Likely there was more care taken and the experiment was completed by a more experienced individual leading to better looking data.

The calculated damping ratios were different than the damping ratios provided by the EDM software. One case had a percentage difference of 9.2% while the other case was 58.2% (Table 4). The large discrepancies are probably due to the uncertainty caused by reading measurements off a graph. The way that the EDM software calculated damping ratios was also not provided so it is difficult to say what truly caused the differences.

Conclusion

The modes of vibration obtained during the lab were physically reasonable. The free plate was a good verification that the setup was working. All the modes shapes for the cantilever plate looked good and there were no unexpected findings. The cantilever plate with the extra fixed point was hard to validate and it was tough to determine if the higher order mode shapes were reasonable. The first three modes shapes made sense but there were some unexpected results in modes 4 and 5, specifically large amplitudes of vibration at single nodes while the nodes around it did not seem to be vibrating very much at all. The damping ratios obtained during the lab seemed to have a large amount of error when compared to the results provided by the software, 9.2-58.25%. The difference in the damping ratio and the unexpected results in the mode shapes for the cantilever plate with extra fixed point can largely be explained by human error during the experiment. This specifically shows up in the force of the hammer strike on the plate which differed between every hit, and not all collected data points were hit perfectly which leads to skewed data. Overall, the lab provides a good understanding between the differences in different modes of vibration.

References

Bitner, Doug, and Reza Fotouhi. *Modal Analysis Laboratory*. Revised August 30, 2025. Department of Mechanical Engineering, University of Saskatchewan.

Appendix 1: Experimentally Determined FRF and Mode Shapes for a Free Plate

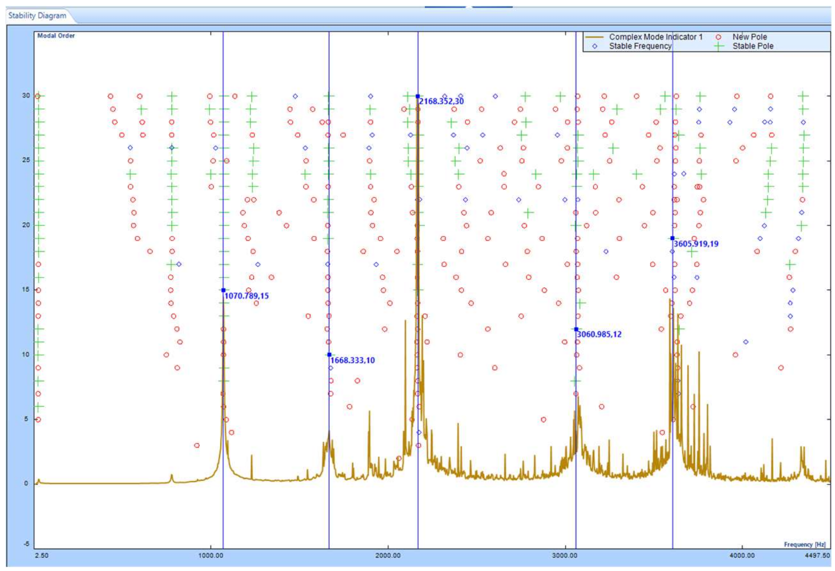


Figure 2: Stability diagram with natural frequency selections for a free plate

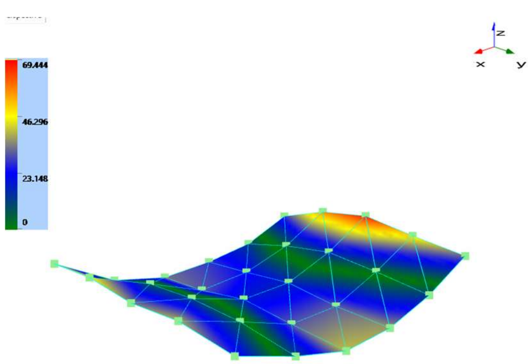


Figure 3: Mode 1, 1070.789 Hz

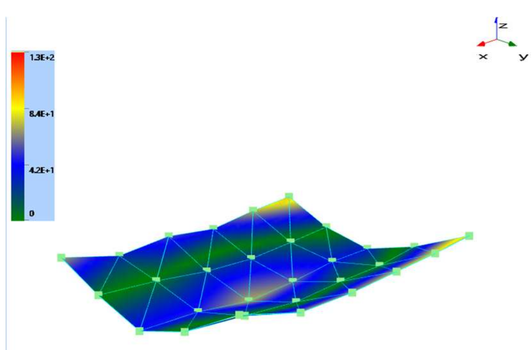


Figure 4: Mode 2, 1668.333 Hz

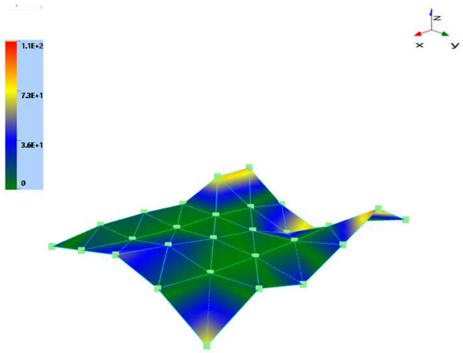


Figure 5: Mode 3, 2168.352 Hz

Appendix 2: Experimentally Determined FRF and Mode Shapes for a Cantilever Plate

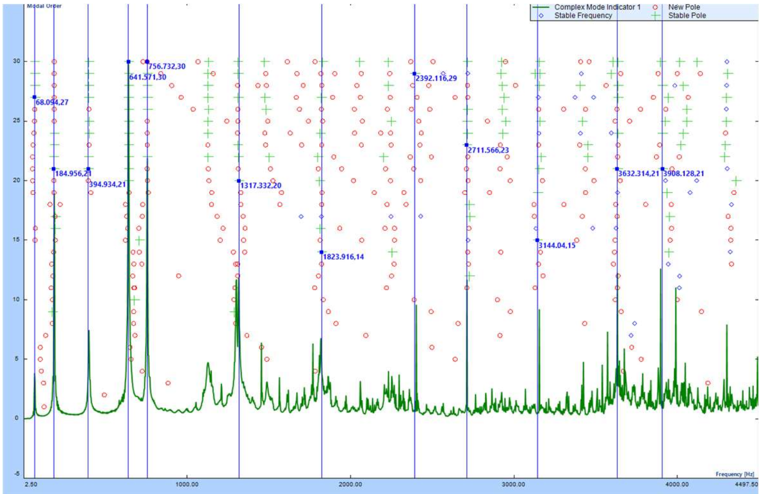


Figure 6: Stability diagram with natural frequency selections for a cantilever plate

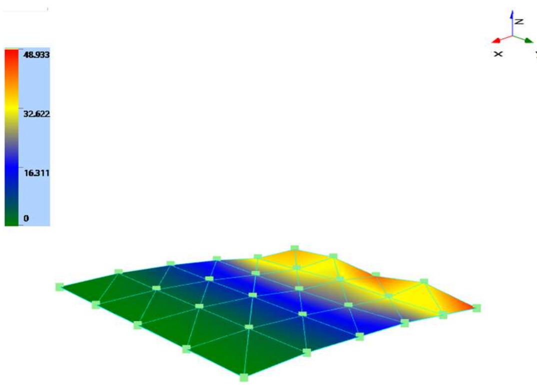


Figure 7: Mode 1, 68.094 Hz

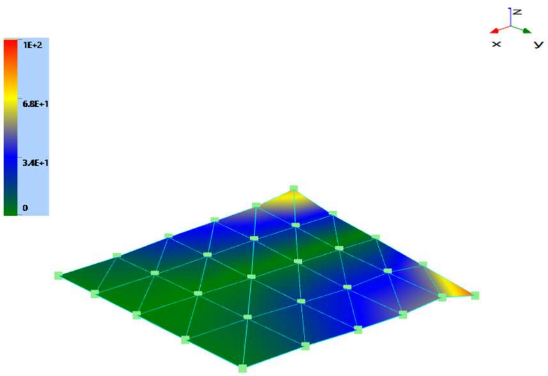


Figure 8: Mode 2, 184.956 Hz

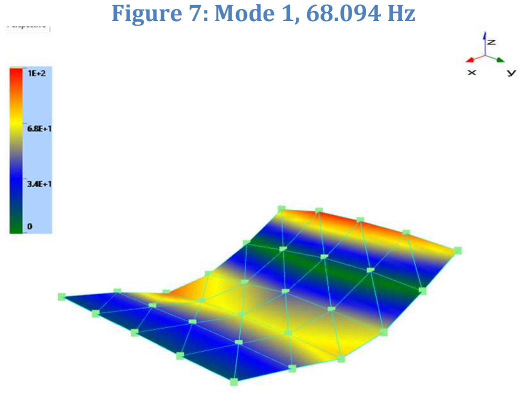


Figure 9: Mode 3, 394.934 Hz

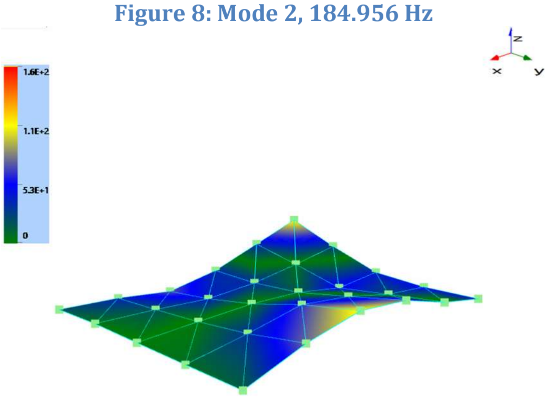


Figure 10: Mode 4, 641.571 Hz

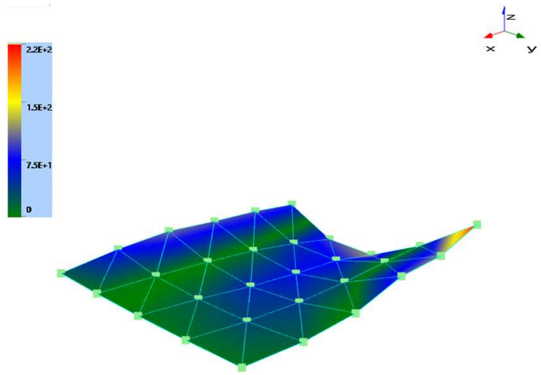


Figure 11: Mode 5, 756.732 Hz

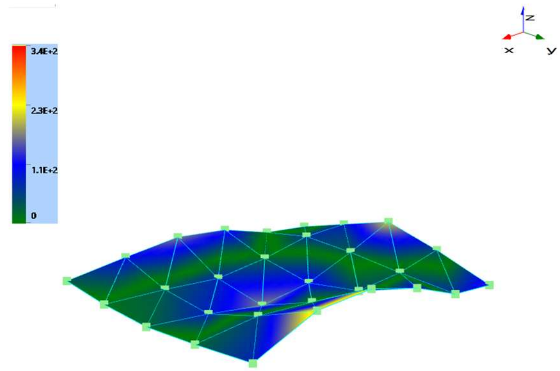


Figure 12: Mode 6, 1317.332 Hz

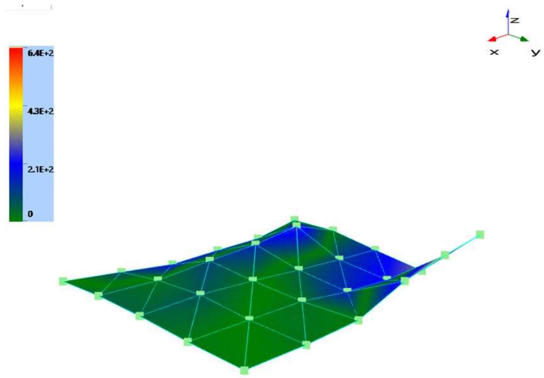


Figure 13: Mode 7, 1823.916 Hz

Appendix 3: Experimentally Determined FRF and Mode Shapes for a Cantilever Plate with Extra Fixed Point

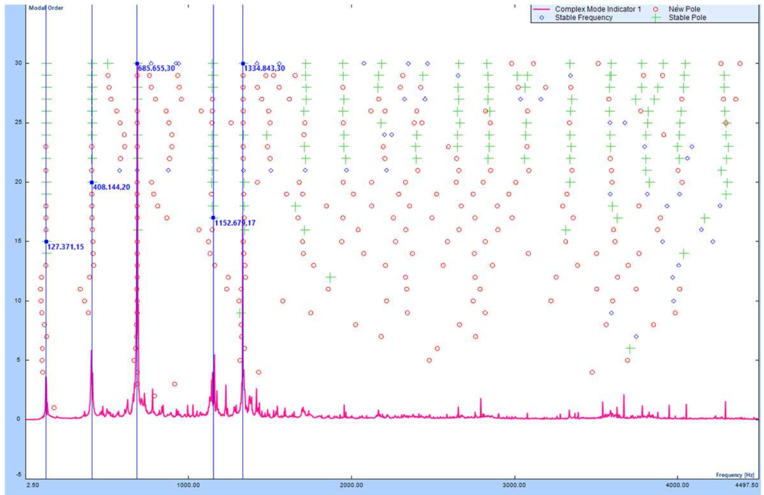


Figure 14: Stability diagram with natural frequency selections for a cantilever plate with extra fixed point

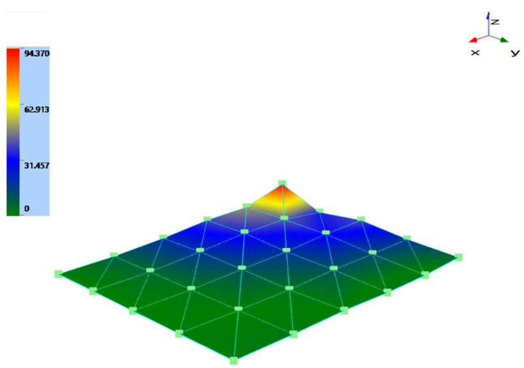


Figure 15: Mode 1, 127.371 Hz

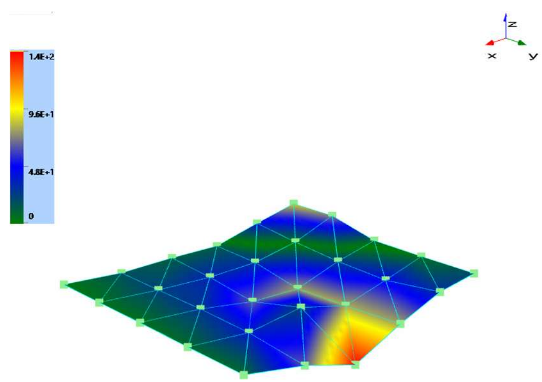


Figure 16: Mode 2, 408.144 Hz

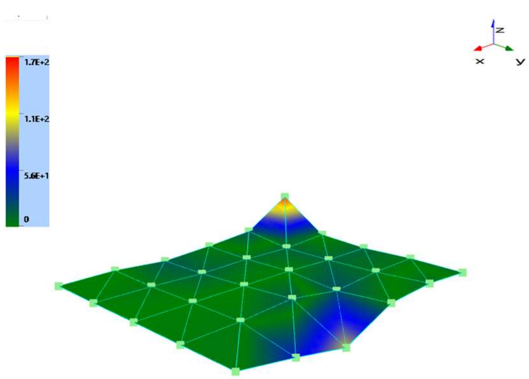


Figure 17: Mode 3, 685.665 Hz

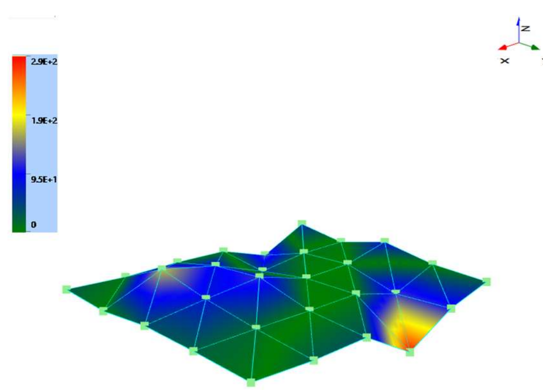


Figure 18: Mode 4, 1152.679 Hz

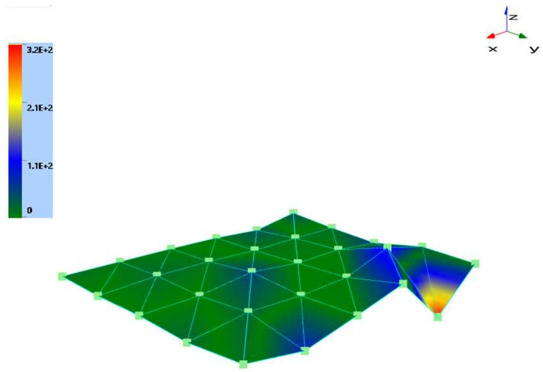


Figure 19: Mode 5, 1334.843 Hz

Appendix 4: Damping Ratio Calculations

This appendix contains the calculations for the damping ratio for two different mode shapes. The mode shapes chosen are modes 3 and 4 for the cantilever plate scenario. Sample calculations are shown for mode 3.

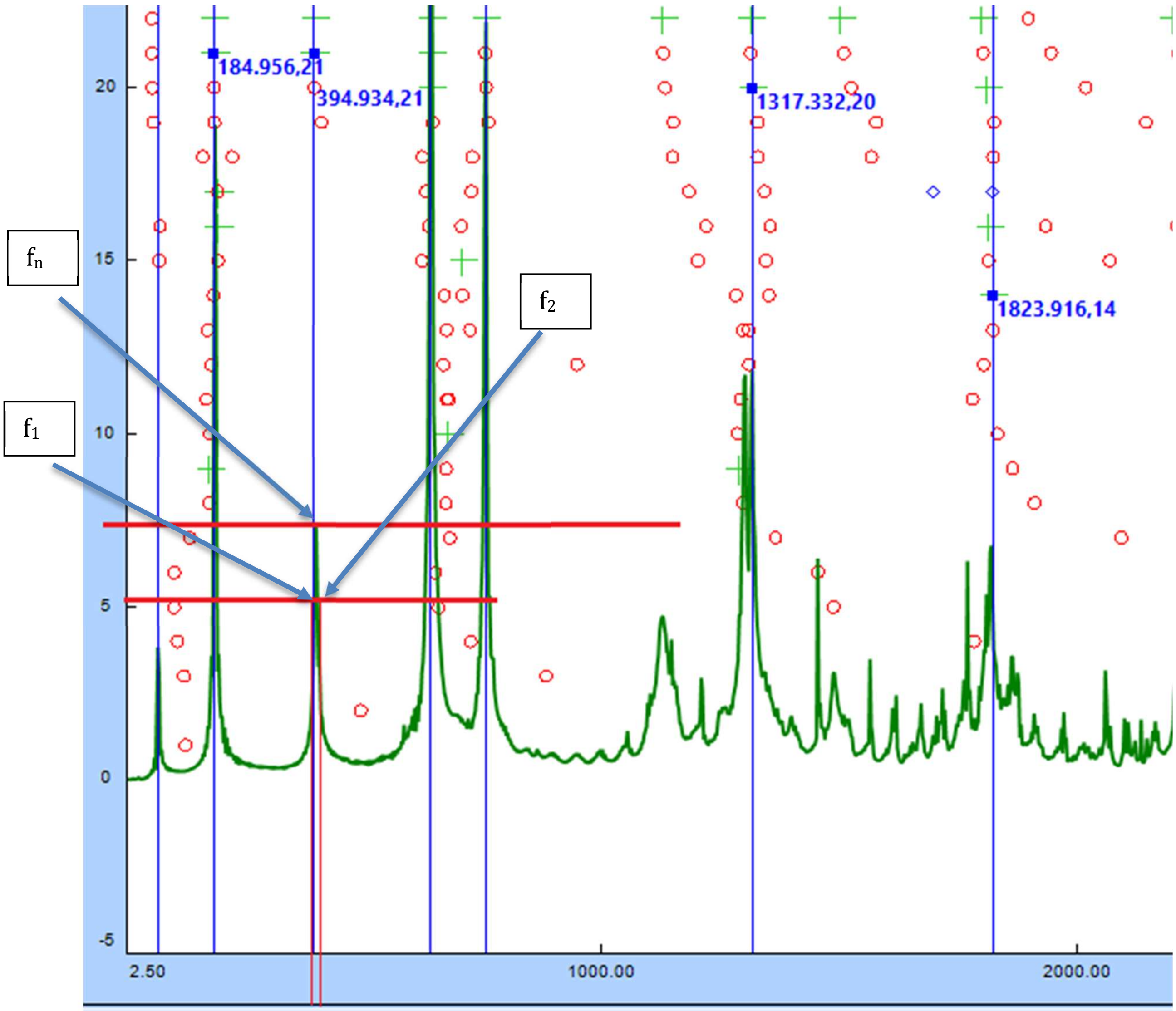


Figure 20: Stability diagram with peak power point (f_n) and half power points (f_1, f_2) shown for Mode 3 f_n=394.934[Hz]

From Figure 20 the peak response for f_n is approximately 7.4

The half power response f_1 and f_2 occur at 0.707 times the peak response

Half Power Response:

$half\ power = 0.707 * 7.4 = 5.23$

Q Factor Calculation:

$$Q = \frac{f_n}{f_2 - f_1}$$

Where:

$Q = Q\ factor$

$f_n = natural\ frequency\ [Hz] = 391.934\ [Hz]$

$f_1, f_2 = Half\ power\ points\ [Hz]$

From Figure 20:

$$f_1 \approx 391\ [Hz]$$

$$f_2 \approx 410\ [Hz]$$

$$Q = \frac{f_n}{f_2 - f_1} = \frac{391.934[Hz]}{(410 - 391)[Hz]}$$

$Q = 20.786$

Damping Ratio Calculation:

$$\zeta = \frac{1}{2Q}$$

$\zeta = damping\ ratio[\%]$

$$\zeta = \frac{1}{2 * 20.786}$$

$\zeta = 0.024055 = 2.41\%$

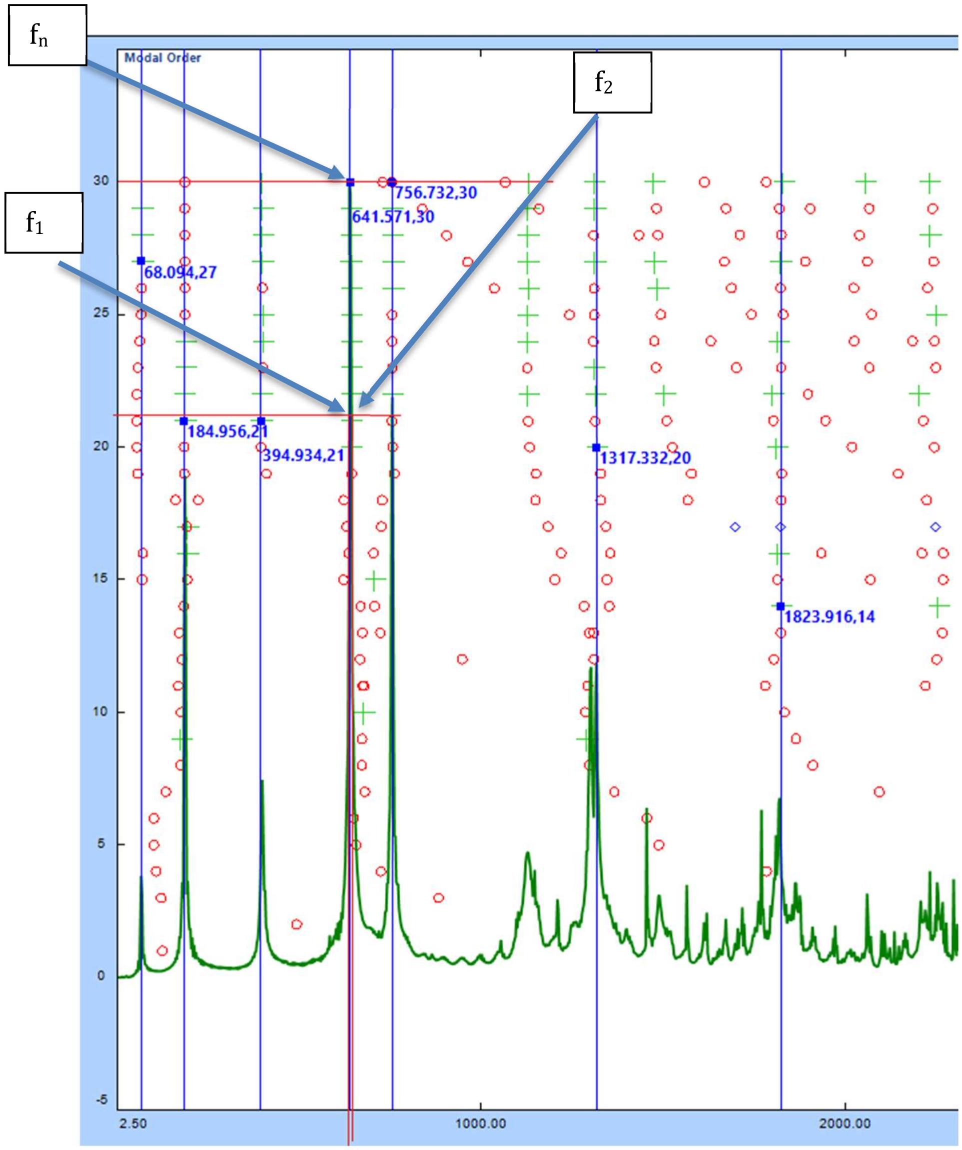


Figure 21: Stability diagram with peak power point (f_n) and half power points (f_1, f_2) shown for Mode 4 f_n=641.571 [Hz]

From Figure 21:

$$f_n = 641.571\ [Hz]$$

$$Peak\ Response\ \approx 30$$

$$Half\ Power\ \approx 21.21$$

$$f_1 \approx 639[Hz]$$

$$f_2 \approx 646[Hz]$$

$Q = 91.653$

$\zeta = 0.0054554 = 0.546\%$

Appendix 5: Theoretical Mode Shapes

This appendix contains the theoretical mode shapes for the first mode for the free plate and for the cantilever plate based off of the equations for free-free beam and fixed-free beam in the ME 497 Acoustics and Vibrations in Design Modal Analysis Lab Manual. Note that these equations are normalized to a beam length of one and a vibration amplitude of 1 for better presentation and easier understanding of the results.

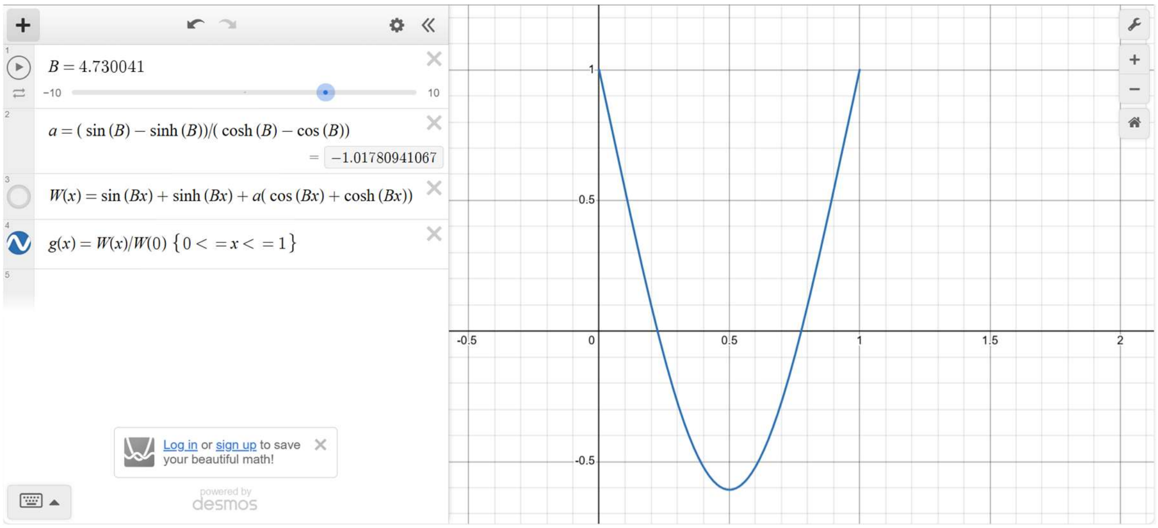


Figure 22: First mode shape for a free-free beam

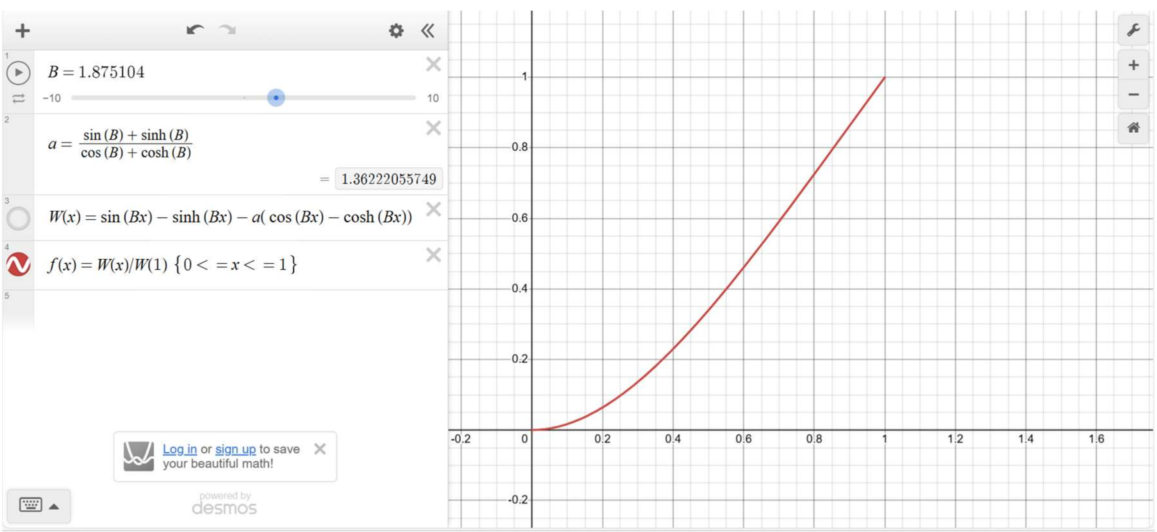


Figure 23: First mode shape for a fixed-free beam

Free-Free Beam

Equations were obtained from the ME 417 Modal Analysis Laboratory Manual (Bitner and Fotouhi 2025).

Mode Shape:

$$W_n(x) = C_n[sin\beta_n x + sinh\beta_n x + \alpha_n(cos\beta_n x + cosh\beta_n x)]$$

Where:

$$\alpha_n = \left(\frac{sin\beta_n l - sinh\beta_n l}{\cos\beta_n l - cosh\beta_n l}\right)$$

$$\beta_1 l = 4.730041$$

Fixed-Free Beam

Mode Shape:

$$W_n(x) = C_n[sin\beta_n x - sinh\beta_n x - \alpha_n(cos\beta_n x - cosh\beta_n x)]$$

Where:

$$\alpha_n = \left(\frac{sin\beta_n l + sinh\beta_n l}{\cos\beta_n l + cosh\beta_n l}\right)$$

$$\beta_1 l = 1.875104$$