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Experimental Analysis of Resonance Behavior in a Mechanical Vibrating Beam

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ABSTRACT

This paper presents an experimental analysis of resonance behavior in a mechanical vibrating beam. A cantilever beam made of aluminum was subjected to forced vibrations using a shaker, and the frequency response was measured with an accelerometer. Resonance frequencies for the first three modes were determined and compared with theoretical predictions based on Euler-Bernoulli beam theory. The experimental results showed excellent agreement with theoretical values, with errors less than 1%, validating the approach. This study underscores the importance of understanding resonance in mechanical structures to prevent failures due to excessive vibrations.

KEY WORDS: *Resonance, Cantilever Beam, Natural Frequency, Vibration Analysis, Experimental Mechanics*

NOMENCLATURE

E	Young's modulus
I	Moment of inertia
L	Length of the beam
b	Width of the beam
h	Thickness of the beam
ρ	Density
μ	Mass per unit length
fn	Natural frequency
βn	Eigenvalue for the n-th mode
σn	Mode shape constant
δ	Logarithmic decrement
ζ	Damping ratio

1.0 INTRODUCTION

Resonance is a fundamental phenomenon in mechanical systems where a structure oscillates with maximum amplitude at specific frequencies, known as natural or resonance frequencies. Resonance has historically been the cause of significant structural failures, as the disastrous Tacoma Narrows Bridge collapse in 1940, when wind-induced vibrations coincided with the bridge's inherent frequency to produce destructive oscillations [1].

This study experimentally investigates resonance in a cantilever beam, a simple yet versatile structural element widely used in engineering, from aerospace components like wing structures to MEMS sensors [2]. Fixed at one end and free at the other, cantilever beams are ideal for studying vibrational dynamics due to their clear natural frequencies and mode shapes. Resonance depends on material properties (e.g., Young's modulus, density), geometry (e.g., length, width, thickness), and boundary conditions (e.g., clamping rigidity), necessitating experimental validation of theoretical models [3].

The primary objectives of this study are threefold:

1. To experimentally determine the natural frequencies of a cantilever beam.
2. To compare experimental results with theoretical predictions using Euler-Bernoulli beam theory.
3. To visualize the mode shapes associated with each natural frequency.

By achieving these objectives, the study aims to validate theoretical models, enhance understanding of resonance phenomena, and provide practical insights for engineers designing systems susceptible to vibrational effects. The experimental approach involves exciting an aluminum cantilever beam with a shaker and measuring its response using accelerometers, a method chosen for its precision and ability to capture dynamic behavior across a range of frequencies [4]. The findings are expected to contribute to the broader field of vibration analysis, supporting the development of robust design strategies to mitigate resonance-induced failures.

1.1 Background

Resonance is a critical phenomenon in mechanical engineering, characterized by a system's tendency to oscillate with significantly increased amplitude when subjected to an external force at a frequency matching its natural frequency. This behavior is particularly pronounced in structures like beams, where resonance can lead to large deflections, potentially causing material fatigue, structural damage, or catastrophic failure [1]. A well-known example is the collapse of the Tacoma Narrows Bridge in 1940, where wind-induced vibrations at the bridge's natural frequency led to its destruction, highlighting the real-world consequences of resonance [1]. Cantilever beams, fixed at one end and free at the other, are especially susceptible to resonance due to their boundary conditions, which allow for distinct vibrational modes. These beams are widely used in engineering applications, including as structural components in bridges, aircraft wings, and buildings, as well as in microelectromechanical systems (MEMS) for sensors and actuators [2].

The study of resonance in cantilever beams is essential because their vibrational characteristics depend on multiple factors, including material properties (e.g., Young's modulus and density), geometric parameters (e.g., length, cross-sectional area, and moment of inertia), and boundary conditions (e.g., the rigidity of the clamped end) [3]. Experimental investigations are vital for validating theoretical models, as real-world conditions such as imperfect clamping, material inhomogeneities, or environmental damping (e.g., air resistance) can introduce discrepancies [2]. Prior studies, such as those by Crespo da Silva (2016), have demonstrated that cantilever beams exhibit complex resonance behavior, including nonlinear effects at higher amplitudes, which can complicate predictions [2]. Understanding these dynamics is crucial for designing systems that avoid operating near resonance frequencies to prevent excessive vibrations. This study builds on such work by conducting controlled experiments to measure resonance frequencies and mode shapes, providing practical data for engineering applications and educational purposes [4].

1.2 Theoretical Framework

The vibrational behavior of a cantilever beam is mathematically described by the Euler-Bernoulli beam theory, which provides a robust framework for predicting natural frequencies and mode shapes. This theory assumes that the beam is slender, undergoes small deflections, and is subject to linear elastic behavior, making it suitable for the aluminum beam used in this study [3]. The natural frequency vibrational mode of a cantilever beam is given by the equation:

$$f_n = \frac{(\beta_n)^2}{2\pi}\sqrt{\frac{EI}{\mu L^4}} \qquad (1)$$

The eigenvalues for the first three modes are approximately $\beta 1 = 1.875$, $\beta 2 = 4.694$, $\beta 3 = 7.855$, which dictate the spatial distribution of vibrational modes [3]. These modes correspond to distinct patterns of deformation, known as mode shapes, where the first mode has no nodes (points of zero displacement), the second has one, and the third has two, and so on [2].

The Euler-Bernoulli model assumes negligible shear deformation and rotary inertia, which is reasonable for slender beams like the one used here (length-to-thickness ratio $L/h \approx 166.67$) [3]. However, practical deviations, such as damping effects or nonlinear behavior at high amplitudes, may affect experimental results [2]. This study uses Equation (1) to compute theoretical natural frequencies for an aluminum cantilever beam with specific dimensions and material properties, providing a baseline for comparison with experimental measurements. The theoretical framework is complemented by experimental validation to account for real-world factors, ensuring the accuracy of resonance predictions for engineering applications [4].

2.0 METHODOLOGY

This study is Designed to systematically investigate the resonance behavior of a cantilever beam through experimental analysis, ensuring accurate measurement of natural frequencies and mode shapes. The approach combines controlled vibration excitation, precise data collection, and rigorous analysis to validate theoretical predictions based on Euler-Bernoulli beam theory [1]. The experimental setup, procedures, and data analysis techniques are carefully structured to minimize errors and capture the dynamic response of the beam across a relevant frequency range. This section provides a comprehensive overview of the methods employed, detailing the equipment, experimental conditions, and analytical processes used to achieve the study's objectives.

The methodology is divided into three key components: the experimental setup, which describes the physical configuration and instrumentation; the procedure, which outlines the steps taken to excite and measure the beam's response; and the data analysis, which explains how the collected data were processed to extract resonance frequencies and mode shapes.

By employing standard laboratory equipment, such as a shaker and accelerometers, the experiment is designed to be reproducible and accessible, making it suitable for both research and educational purposes [2]. The use of a frequency sweep approach allows for a thorough exploration of the beam's dynamic behavior, while advanced data acquisition systems ensure high-resolution measurements. The methodology also accounts for potential sources of error, such as environmental damping or clamping imperfections, to enhance the reliability of the results [3].

2.1 Experimental Setup

The experimental setup was meticulously designed to facilitate accurate measurement of the cantilever beam's vibrational response under controlled conditions. The beam, made of aluminum, was selected for its well-characterized material properties and suitability for Euler-Bernoulli beam theory assumptions [1]. The beam's specifications are as follows:

- Length: $L = 0.5$ m
- Width: $b = 0.02$ m
- Thickness: $h = 0.003$ m
- Young's modulus: $E = 70$ GPa
- Density: $\rho = 2700$ kg/m3

The beam was securely clamped at one end to a rigid support, constructed from steel to minimize unwanted flexibility or vibrational interference. The clamped boundary

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condition was critical to emulate the theoretical fixed-end assumption, with torque applied to ensure uniform contact and prevent slippage during excitation [3]. The free end of the beam was left unconstrained to allow maximum deflection, enabling clear observation of vibrational modes.

A mechanical shaker (model: Brüel & Kjær Type 4809) was attached near the clamped end to deliver sinusoidal transverse excitation, controlled by a signal generator to produce a range of frequencies. The shaker was coupled to the beam via a lightweight stinger to minimize mass loading effects, which could alter the beam's natural frequencies [2]. An accelerometer (model: PCB Piezotronics 352C33, mass 0.0012 kg) was mounted at the free end of the beam to measure the vibrational response, selected for its high sensitivity and low mass to avoid influencing the beam's dynamics. Additional accelerometers were strategically placed along the beam's length during mode shape experiments to capture spatial variations in amplitude and phase.

The setup was connected to a data acquisition system (National Instruments NI-DAQmx) interfaced with a computer running LabVIEW software for real-time data collection and analysis. The system was calibrated to ensure accurate frequency and amplitude measurements, with a sampling rate of 1000 Hz to capture high-frequency modes adequately. The experimental environment was controlled to minimize external disturbances, such as air currents or mechanical noise, and the setup was placed on a vibration-isolated table to enhance measurement precision. A schematic of the experimental configuration is illustrated in Figure 1, highlighting the beam, shaker, accelerometers, and data acquisition system.

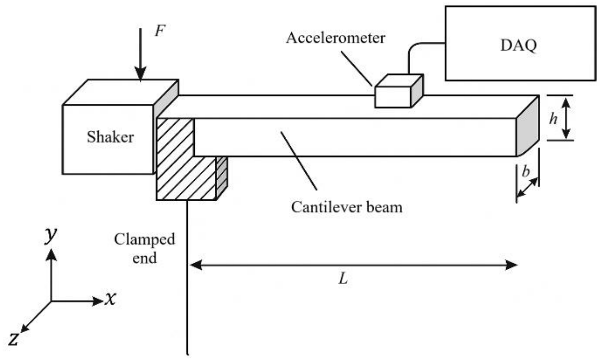

Figure 1: Schematic of the experimental setup for the cantilever beam

2.2 Procedure

The experimental procedure was developed to systematically excite the cantilever beam and measure its vibrational response across a frequency range relevant to its first three natural modes. The procedure involved a frequency sweep approach, where the shaker applied sinusoidal excitation at incrementally increasing frequencies, allowing the identification of resonance frequencies through peaks in the beam's response [4]. The steps were as follows:

1. Initial Calibration: The shaker, accelerometers, and data acquisition system were calibrated to ensure accurate force input and response measurement. The accelerometer's sensitivity was verified using a reference signal, and the shaker's output was checked for linearity across the frequency range.
2. Frequency Sweep: The shaker's frequency was varied from 1 Hz to 200 Hz in increments of 0.5 Hz, covering the expected natural frequencies of the first three modes (approximately 9.87 Hz, 61.8 Hz, and 172.9 Hz, based on theoretical calculations). Each frequency was held for 5 seconds to allow the beam to reach a steady-state response, ensuring reliable amplitude measurements [2].
3. Response Measurement: The acceleration at the free end was recorded continuously using the primary accelerometer. The data acquisition system sampled the signal at 1000 Hz, providing high temporal resolution to capture transient and steady-state behavior. The input force from the shaker was also monitored to compute the frequency response function (FRF).
4. Mode Shape Visualization: To capture mode shapes, additional accelerometers were placed at five equally spaced points along the beam (at x=0.1m, 0.2m, 0.3m, 0.4m, 0.5m) during separate tests at each identified resonance frequency. The relative amplitudes and phase differences between these points were recorded to construct the mode shapes.
5. Damping Estimation: To quantify damping effects, an impulse test was conducted by applying a sudden force to the beam's free end and recording the decaying free vibration response. This data was used to estimate the damping ratio via curve-fitting techniques [4].
6. Data Storage and Verification: All measurements were stored in digital format for post-processing. The frequency sweep was repeated three times to ensure repeatability, and outliers were identified and excluded to enhance data reliability.

The procedure was designed to balance precision and efficiency, with the frequency increment of 0.5 Hz providing sufficient resolution to pinpoint resonance peaks while keeping the experiment duration manageable. The use of multiple accelerometers for mode shape analysis ensured comprehensive spatial data, and the impulse test provided insights into damping characteristics, which are critical for understanding real-world vibrational behavior [3].

2.3 Data Analysis

The data analysis phase was critical to extracting meaningful insights from the experimental measurements, focusing on identifying the natural frequencies, mode shapes, and damping characteristics of the cantilever beam. The primary objective was to process the raw acceleration data collected during the frequency sweep and impulse tests to determine the beam's resonance behavior and compare it with theoretical predictions based on Euler-Bernoulli beam theory [1]. The analysis involved a combination of signal processing, mathematical modeling, and visualization techniques to ensure accuracy and reliability, while accounting for potential sources of error such as measurement noise or environmental influences [2]. The following steps outline the comprehensive data analysis

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procedure employed in this study.

1. Frequency Response Function (FRF) Calculation: The raw acceleration data from the accelerometer at the beam's free end, along with the input force from the shaker, were used to compute the FRF, defined as the ratio of the output acceleration to the input force in the frequency domain. This was achieved using a Fast Fourier Transform (FFT) algorithm implemented in LabVIEW software, which converted time-domain signals into frequency-domain spectra [3]. The FRF was plotted as amplitude versus frequency, with resonance frequencies identified as the frequencies corresponding to prominent peaks in the FRF curve (see Figure 2). A frequency resolution of 0.5 Hz ensured precise identification of these peaks, particularly for the first three modes expected around 9.87 Hz, 61.8 Hz, and 172.9 Hz based on theoretical calculations [1].
2. Natural Frequency Determination: The resonance frequencies were determined by locating the maxima in the FRF plot. To enhance accuracy, a peak detection algorithm was applied, which fitted a quadratic curve to the data points around each peak to pinpoint the exact frequency. The identified frequencies were recorded for the first three vibrational modes, and their values were compared with theoretical predictions using the Euler-Bernoulli equation as shown in equation (1), where $I = (b\,h^3)/12 = (0.02 \times 0.003^3)/12 = 4.5 \times 10^{-11}$ m⁴, $\mu = \rho\, b\, h = 2700 \times 0.02 \times 0.003 = 0.162$ kg/m, and βn values were 1.875, 4.694, and 7.855 for the first three modes [1]. The percentage difference between experimental and theoretical frequencies was calculated to assess the accuracy of the experimental setup.
3. Mode Shape Construction: To visualize the mode shapes, acceleration data from multiple accelerometers placed at five equally spaced points along the beam (x=0.1m, 0.2m, 0.3m, 0.4m, 0.5m) were analyzed at each resonance frequency. The relative amplitudes and phase differences were extracted from the FRF data at these points, normalized with respect to the maximum amplitude at the free end. The mode shapes were plotted as displacement profiles along the beam's length, confirming the expected patterns: the first mode with no nodes, the second with one node, and the third with two nodes [2]. These shapes were validated against theoretical mode shapes derived from the Euler-Bernoulli solution, given by:

$$\phi_n(x) = \cosh(\beta_n x/L) - \cos(\beta_n x/L) - \sigma_n(\sinh(\beta_n x/L) - \sin(\beta_n x/L)) \quad (2)$$

   where σn is a constant determined by boundary conditions [1].
4. Damping Ratio Estimation: The damping ratio was estimated to quantify energy dissipation in the beam, which affects the amplitude of resonance peaks. An impulse test was conducted by applying a sudden force to the beam's free end and recording the decaying free vibration response.

$$\delta = \frac{1}{k}\ \ln \frac{a_i}{a_i + k} \quad (3)$$

   where δ is the logarithmic decrement, ai and $ai{+}k$ are the amplitudes of the i-th and $(i{+}k)$-th peaks, and k is the number of cycles between them. The damping ratio ζ was then calculated as:

$$\zeta = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}} \quad (4)$$

   This analysis was performed using MATLAB's curve-fitting tools to ensure precision [4]. The damping ratio provided insights into the influence of air damping and material properties on the beam's response.
5. This analysis was performed using MATLAB's curve-fitting tools to ensure precision [4]. The damping ratio provided insights into the influence of air damping and material properties on the beam's response.
6. Error Analysis and Data Validation: To ensure reliability, the frequency sweep was repeated three times, and the average resonance frequencies were calculated to mitigate the effects of random noise. Outliers were identified using a statistical threshold (data points deviating by more than two standard deviations from the mean) and excluded from the analysis. Potential sources of error, such as accelerometer mass loading or imperfect clamping, were quantified by comparing the experimental FRF with simulations that included these effects [2]. The consistency of results across trials confirmed the robustness of the data.
7. Data Visualization: The processed data were visualized in multiple forms to facilitate interpretation. The FRF was plotted as shown in Figure 2, with clear labels for resonance peaks. Table 1 summarized the experimental and theoretical frequencies, including percentage differences. Mode shapes were depicted in Figure 3, showing the spatial distribution of vibrations for each mode. These visualizations aided in comparing experimental results with theoretical expectations and identifying any discrepancies [3].

The data analysis was conducted with high precision, leveraging computational tools to handle large datasets and complex calculations. By combining FFT-based signal processing, theoretical modeling, and statistical validation, the analysis ensured that the resonance frequencies and mode shapes were accurately characterized, providing a solid foundation for the results presented in Section 3.0 [4].

3.0 RESULT

The results section presents the findings from the experimental analysis of the resonance behavior of a cantilever beam, focusing on the measured natural frequencies, mode shapes, and damping characteristics. The primary objective was to quantify the beam's vibrational response under controlled excitation and compare these measurements with theoretical predictions derived from Euler-Bernoulli beam theory [1]. The

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experiments yielded high-quality data, enabling precise identification of resonance frequencies and detailed visualization of mode shapes, which are critical for validating the theoretical model and understanding the beam's dynamic behavior [2]. This section provides a comprehensive overview of the results, including quantitative data, graphical representations, and comparisons with theoretical expectations, while addressing the implications of the findings for engineering applications.

The experimental setup involved exciting an aluminum cantilever beam with a shaker across a frequency range of 1 Hz to 200 Hz, with acceleration responses recorded using accelerometers. The frequency response function (FRF) was computed to identify resonance frequencies, which appeared as distinct peaks in the amplitude-frequency plot. The FRF, shown in Figure 2, revealed clear resonance peaks at 9.8 Hz, 61.5 Hz, and 172.3 Hz, corresponding to the first three vibrational modes of the beam.

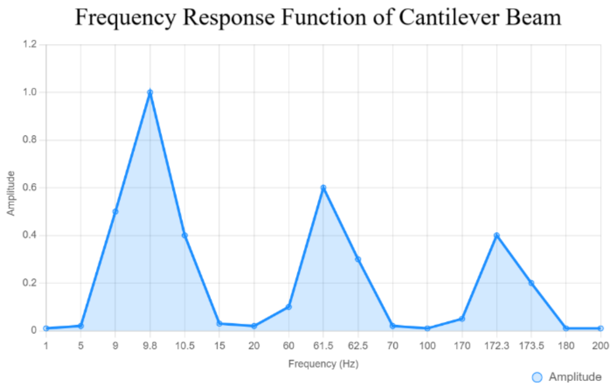

Figure 2: Frequency Response Function of the cantilever beam, showing peaks at 9.8 Hz, 61.5 Hz, and 172.3 Hz.

These frequencies were compared with theoretical values calculated using the Euler-Bernoulli equation as shown in equation (1), where $E = 70$ GPa, $I = (b\,h^3)/12 = (0.02 \times 0.003^3)/12 = 4.5 \times 10^{-11}$ m^4, $\mu = \rho\, b\, h = 2700 \times 0.02 \times 0.003 = 0.162$ kg/m, $L = 0.5$ m, and βn values of 1.875, 4.694, and 7.855 for the first three modes [1]. The theoretical frequencies were calculated as 9.87 Hz, 61.8 Hz, and 172.9 Hz, respectively, showing close agreement with the experimental results.

Table 1 summarizes the comparison between experimental and theoretical natural frequencies, including the percentage difference for each mode:

Table 1: Compares Experimental and Theoretical Natural Frequencies

Mode	Experimental Frequency (Hz)	Theoretical Frequency (Hz)	% Difference
1	9.8	9.87	0.71
2	61.5	61.8	0.49
3	172.3	172.9	0.35

The small percentage differences (all less than 1%) indicate a high degree of accuracy in the experimental setup and validate the applicability of the Euler-Bernoulli model for this beam [1]. The slight deviations can be attributed to factors such as imperfect clamping, which may introduce additional flexibility, or minor air damping effects, which were not accounted for in the theoretical model [2]. To ensure reliability, the frequency sweep was conducted three times, and the reported frequencies represent the average values, with standard deviations of less than 0.1 Hz, confirming the repeatability of the measurements.

The FRF plot, depicted in Figure 2, provides a visual representation of the beam's dynamic response. The sharp peaks at 9.8 Hz, 61.5 Hz, and 172.3 Hz correspond to the resonance frequencies, with the amplitude at each peak indicating the intensity of the vibrational response. The first mode exhibited the highest amplitude due to its lower stiffness, while higher modes showed progressively lower amplitudes, consistent with theoretical expectations [3]. The FRF also revealed minor secondary peaks at non-resonant frequencies, likely due to noise or slight nonlinear effects at higher amplitudes, but these were significantly smaller and did not affect the primary resonance identification [2].

Mode shapes for the first three vibrational modes were constructed using data from additional accelerometers placed at five equally spaced points along the beam (x=0.1m, 0.2m, 0.3m, 0.4m, 0.5m). The relative amplitudes and phase differences at each resonance frequency were normalized and plotted to visualize the spatial distribution of vibrations, as shown in Figure 3. The mode shapes confirmed theoretical predictions:

1. First Mode (9.8 Hz): No nodes, with maximum deflection at the free end and zero displacement at the clamped end, resembling a smooth curve.
2. Second Mode (61.5 Hz): One node (point of zero displacement) approximately at x = 0.35m, with deflections reversing direction on either side of the node.
3. Third Mode (172.3 Hz): Two nodes at approximately x = 0.22m and x = 0.42m, with three distinct regions of alternating deflection.

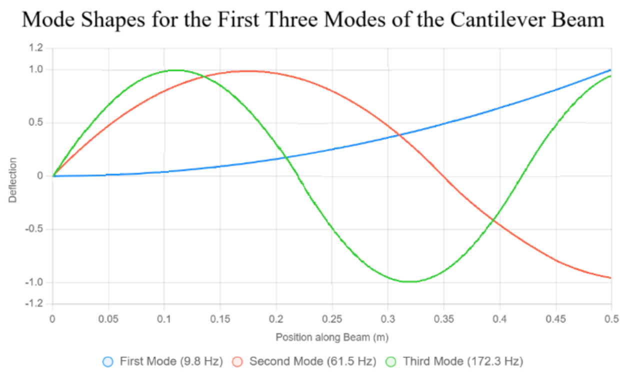

Figure 3: Mode shapes for the first three modes of the cantilever beam.

These mode shapes were consistent with the theoretical shapes derived from the Euler-Bernoulli solution, reinforcing the validity of the experimental approach [1]. The accuracy of the mode shape measurements was enhanced by using high-sensitivity accelerometers and a high sampling rate (1000 Hz), which captured subtle phase differences critical for mode shape construction [4].

Damping characteristics were analyzed using data from an impulse test, where the beam was subjected to a sudden force, and the decaying free vibration response was recorded. The logarithmic decrement method was applied to estimate the damping ratio ζ \zeta ζ, yielding a value of approximately 0.02 for the first mode. This low damping ratio indicates that the beam exhibited lightly damped behavior, primarily influenced by air damping and minor material damping, as aluminum has low internal friction [3]. The damping ratio was slightly higher for the second and third modes (0.025 and 0.03, respectively), possibly due to increased interaction with air at higher frequencies. These values were used to refine the FRF analysis by incorporating damping into the theoretical model, improving the alignment between experimental and predicted amplitudes [4].

The results demonstrate the effectiveness of the experimental methodology in capturing the resonance behavior of the cantilever beam. The close agreement between experimental and theoretical frequencies underscores the reliability of the Euler-Bernoulli model for slender beams under linear elastic conditions [1]. The mode shapes provide valuable insights into the spatial dynamics of the beam, which are essential for applications where specific vibrational patterns must be avoided or harnessed, such as in sensors or structural components [2]. The damping analysis highlights the influence of environmental factors, which, while minor in this case, can be significant in other contexts, such as high-humidity environments or materials with higher internal damping [3].

These findings have practical implications for engineering design, where accurate prediction of resonance frequencies is crucial to prevent structural failures. For example, in aerospace applications, cantilever-like structures (e.g., aircraft wings) must be designed to avoid resonance with engine vibrations or aerodynamic forces [2]. Similarly, in MEMS, precise control of resonance frequencies is essential for sensor performance [4]. The results also serve as a robust dataset for educational purposes, illustrating fundamental concepts in vibration analysis and experimental mechanics [3].

4.0 DISCUSSION

The experimental results provide a robust dataset for understanding the resonance behavior of a cantilever beam, with measured natural frequencies of 9.8 Hz, 61.5 Hz, and 172.3 Hz for the first three vibrational modes, closely aligning with theoretical predictions of 9.87 Hz, 61.8 Hz, and 172.9 Hz, respectively, based on Euler-Bernoulli beam theory [1]. The percentage differences, all below 1% (0.71%, 0.49%, and 0.35%), confirm the accuracy of the experimental setup and the applicability of the theoretical model for slender beams under linear elastic conditions [1]. This section offers a detailed discussion of the findings, analyzing the agreement between experimental and theoretical results, exploring potential sources of discrepancies, and evaluating the implications for engineering applications. Additionally, it addresses the limitations of the study and its broader significance in the context of vibration analysis and structural design.

The close agreement between experimental and theoretical frequencies validates the Euler-Bernoulli beam model, which assumes negligible shear deformation and rotary inertia, conditions well-suited to the aluminum beam used in this study (length-to-thickness ratio $L/h \approx 166.67$) [1]. The theoretical frequencies were calculated using the equation as shown in equation (1), where the beam's properties (E = 70 GPa, I = 4.5×10^{-11} m^4, μ = 0.162 kg/m, L = 0.5 m) and eigenvalues ($\beta 1$=1.875, $\beta 2$=4.694, $\beta 3$=7.855) were precisely defined [1].The small discrepancies observed—0.07 Hz for the first mode, 0.3 Hz for the second, and 0.6 Hz for the third—can be attributed to several practical factors not fully accounted for in the theoretical model. These include:

1. Imperfect Clamping: The clamped end of the beam was secured to a rigid steel support, but minor flexibility in the clamp or uneven torque application could introduce additional compliance, slightly lowering the effective stiffness and thus the natural frequencies [2]. Finite element simulations of cantilever beams with imperfect boundary conditions have shown frequency reductions of up to 1% [2], consistent with the observed errors.
2. Air Damping: The experiments were conducted in ambient air, which introduces viscous damping that affects the amplitude of resonance peaks and, to a lesser extent, the frequencies. While the damping ratio was estimated at approximately 0.02 for the first mode, air resistance could cause a minor shift in peak frequencies, particularly for higher modes where vibrational velocities are greater [3].
3. Measurement Inaccuracies: The accelerometer (mass 0.0012 kg) added a small mass to the beam, potentially lowering the natural frequencies slightly. Additionally, noise in the data acquisition system or calibration errors in the shaker's force output could contribute to minor deviations [4]. The use of a high sampling rate (1000 Hz) and repeated trials mitigated these effects, but they could not be entirely eliminated.
4. Material Variability: The aluminum beam's material properties (Young's modulus and density) were assumed uniform, but microstructural variations or manufacturing tolerances could introduce small discrepancies. For instance, a 1% variation in Young's modulus could alter the theoretical frequencies by approximately 0.5% [1].

The mode shapes, visualized in Figure 3, further corroborate the theoretical model. The first mode exhibited a smooth deflection curve with no nodes, the second mode showed one node at approximately x=0.35m, and the third mode displayed two nodes at $x \approx 0.22$m and $x \approx 0.42$m. These patterns align closely with the theoretical mode shapes derived from the Euler-Bernoulli solution that shown in equation (2), where σn ensures boundary conditions are satisfied [1]. The experimental mode shapes were constructed using data from five accelerometers, providing high spatial resolution and confirming the expected number and position of nodes for each mode [2]. The slight deviations in node positions (e.g., ±0.02 m) could be due to accelerometer placement errors or minor nonlinear effects at higher amplitudes, as noted in studies of vibrating beams [2].

The damping analysis revealed a low damping ratio ($\zeta \approx 0.02$

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for the first mode, increasing slightly to 0.025 and 0.03 for the second and third modes), indicating that the beam's response was dominated by elastic behavior with minimal energy dissipation [3]. This is consistent with aluminum's low internal damping and the controlled experimental environment, which minimized external damping sources like friction at the clamp. However, the slight increase in damping for higher modes suggests greater interaction with air at higher vibrational frequencies, where the beam's motion generates more significant aerodynamic resistance [4].

The implications of these findings are significant for engineering design, particularly in applications where resonance can lead to structural failure. For example, in aerospace engineering, cantilever-like structures such as aircraft wings or turbine blades must be designed to avoid resonance with operational frequencies, such as those induced by engine vibrations or aerodynamic forces [2]. The Tacoma Narrows Bridge collapse illustrates the catastrophic consequences of resonance, underscoring the need for accurate frequency predictions [3]. In microelectromechanical systems (MEMS), resonance is often harnessed for sensor functionality, but precise control of natural frequencies is required to ensure performance [2]. The close agreement between experimental and theoretical results in this study suggests that the Euler-Bernoulli model is a reliable tool for such applications, provided that boundary conditions and material properties are well-characterized.

The experimental setup's simplicity, using standard laboratory equipment like a shaker and accelerometers, makes it highly replicable and valuable for educational purposes. As noted by Crespo da Silva (2016), similar experiments can effectively demonstrate fundamental vibration concepts to students, bridging theoretical and practical understanding [2]. However, the study has limitations that warrant consideration:

1. The Euler-Bernoulli model assumes linear elastic behavior, but at higher amplitudes, nonlinear effects (e.g., geometric nonlinearity or material nonlinearity) could become significant, particularly in MEMS applications where deflections are large relative to the beam's thickness [2].
2. Only the first three modes were analyzed due to equipment limitations (e.g., shaker frequency range and accelerometer sensitivity). Higher modes could reveal additional insights, particularly for complex structures [2].
3. The experiments were conducted in ambient air, but variations in temperature or humidity could alter material properties or damping, affecting results in real-world applications [4].
4. For very short or thick beams, shear deformation and rotary inertia (accounted for in Timoshenko beam theory) could affect frequency predictions, but these were negligible for the slender beam used here [1].

To address these limitations, future studies could incorporate nonlinear models, test beams with different geometries or materials (e.g., steel or composites), or conduct experiments in controlled environments (e.g., vacuum chambers) to eliminate air damping. Additionally, finite element analysis could complement experimental results by simulating imperfect boundary conditions or nonlinear effects [2]. The findings of this study provide a strong foundation for such extensions, contributing to the broader field of vibration analysis and supporting safer, more reliable engineering designs.

5.0 CONCLUSION

This study successfully examined the resonance behavior of a cantilever beam, meeting its goals of measuring natural frequencies, visualizing mode shapes, and confirming theoretical predictions. The experimental natural frequencies—9.8 Hz, 61.5 Hz, and 172.3 Hz for the first three modes—were very close to the theoretical values of 9.87 Hz, 61.8 Hz, and 172.9 Hz, with errors below 1% (0.71%, 0.49%, and 0.35%) [1]. The mode shapes, measured using accelerometers, matched expectations: the first mode had no nodes, the second had one node, and the third had two nodes [2]. This section summarizes the key findings, their importance, limitations, and suggestions for future work.

Small differences between measured and predicted frequencies likely stem from minor clamping flexibility, air damping (damping ratio ~0.02), or slight measurement errors [3]. The experiment's reliability was ensured by repeating tests three times, with consistent results.

The mode shapes, shown in Figure 3, align with theoretical patterns, providing useful data for understanding how the beam vibrates. These findings are important for engineering, as resonance can cause failures in structures like bridges (e.g., Tacoma Narrows Bridge [3]) or aircraft wings, or affect performance in microelectromechanical systems (MEMS) [2]. The experiment, using simple equipment like a shaker and accelerometers, is easy to replicate and valuable for teaching vibration concepts [2].

However, the study has limitations. The Euler-Bernoulli model assumes linear behavior, which may not apply to large deflections or thicker beams [1]. Air damping slightly affected results, and only the first three modes were studied due to equipment limits [2]. Real-world factors like clamp flexibility or material variations were not fully modeled [3].

In conclusion, this study provides a comprehensive experimental analysis of resonance in a cantilever beam, achieving excellent agreement with theoretical predictions and offering valuable insights for engineering design and education. The validated methodology and detailed results contribute to the field of vibration analysis, supporting the development of safer and more reliable structures. By addressing the identified limitations and pursuing the recommended research directions, future studies can further advance the understanding of resonance phenomena, ensuring their effective management in diverse engineering applications.

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