VIBRATIONS

DESIGN & TESTING OF A MTD FOR VIBRATIONAL CONTROL

Names	Reg No.	Work breakdown
Saad Ahmad	369471 43 ME A	Theoretical analysis, MATLAB, Report writing
Ghufran Ahmed Talib	369250 43 ME A	Experimentation, FEA analysis
Ahmed Saeed	398405 43 ME A	Brainstorming, Experimentation
Ghulam Mohi ud Din	336368 42 ME A	Experimentation, Solidworks, Prototype

Table of Contents

Design and Testing of a Mass Tuned Damper for Vibrational Control

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ABSTRACT

This project investigates the dynamic behavior of a one-floor building structure, comprising a wooden plate supported by four steel rods, and the influence of a ***pendulum mass tuned damper***[1] [2], [3]on its vibrations. Initially, ***theoretical*** analysis was conducted to derive the ***equations of motion***, calculate the ***natural frequency***, and estimate the damper's secondary mass. These theoretical predictions were validated through software simulations and experimental measurements. Experimental data were obtained by analyzing the structure's response without and with the damper, focusing on frequency versus amplitude behavior. ***MATLAB*** was employed to plot frequency-response curves and calculate the ***experimental damping ratio*** from the damper's performance. Additionally, ***ANSYS*** software was utilized to compute the natural frequency of the structure and the damping ratio with the damper. Theoretical, experimental, and software results were compared to assess the effectiveness of the damper and validate the analysis. This comprehensive study highlights the interplay between theoretical modeling, experimental validation, and computational tools in understanding and mitigating structural vibrations.

Introduction

Structural vibrations are a critical concern in engineering design, particularly in systems exposed to dynamic loading, such as buildings, bridges, and mechanical components. Excessive vibrations can compromise structural integrity, reduce operational efficiency, and pose safety risks. To address these challenges, various vibration mitigation techniques have been developed, including the use of tuned mass dampers (TMDs). ***TMDs*** are widely employed for vibration control due to their simplicity, effectiveness, and adaptability across various applications. This project focuses on the design, implementation, and analysis of a pendulum mass tuned damper integrated into a simple structural model to study its effects on vibration reduction.

The study begins with the development of a one-floor building structure, represented by a wooden plate supported by four steel rods. This simplified model provides a controlled environment to study the dynamic characteristics of the system. Initially, the structure is analyzed without the damper to determine its natural frequency and response under dynamic excitation. Theoretical equations of motion are derived to characterize the system, and the natural frequency is calculated using analytical methods. Subsequently, a pendulum-type mass tuned damper is introduced, designed based on theoretical approximations of the damper mass and its optimal tuning parameters.

The project progresses through three stages of analysis: ***theoretical, experimental, and computational***. Theoretical analysis involves deriving the equations of motion, calculating the natural frequency, and determining the expected damping behavior with the damper. Experimental analysis involves exciting the structure and recording its frequency-response data both without and with the damper. Two sets of data are obtained: one for the bare structure and the other for the structure with the damper installed. ***MATLAB*** is then used to plot frequency-response curves and extract key parameters, such as the ***experimental damping ratio***, from the observed data.

The computational aspect of the study leverages ***ANSYS software*** to perform modal and harmonic analyses of the structure. These simulations provide insights into the natural frequency of the structure without the damper and the modified response when the damper is added. Additionally, ANSYS results are used to extract theoretical damping ratios, which are then compared with experimental findings.

The final stage of the project involves a comprehensive comparison of theoretical, experimental, and software-derived results. This comparison serves to validate the analytical models and computational tools while assessing the performance of the pendulum mass tuned damper in mitigating structural vibrations. The findings contribute to a deeper understanding of vibration control techniques and demonstrate the practical application of theoretical and computational methods in real-world engineering problems.

This report details the methodology, analysis, and findings of the project, highlighting the interplay between theoretical modeling, experimental testing, and computational simulations in understanding and optimizing vibration control strategies.

Problem Statement

Excessive vibrations in structural systems can lead to reduced stability, compromised safety, and potential structural failures. This project aims to address these issues by designing and analyzing a pendulum mass tuned damper for a simplified one-floor building structure. The study involves theoretical modeling, experimental validation, and computational simulations to evaluate the damper's effectiveness in ***reducing vibrations*** and improving structural damping.

Objectives

1. To derive the theoretical equations of motion for a one-floor building structure and calculate its natural frequency.
2. To design and estimate the optimal parameters of a pendulum mass tuned damper for vibration mitigation.
3. To perform experimental analysis of the structure's frequency response with and without the damper and extract key parameters such as damping ratio.
4. To simulate the structure's dynamic behavior using ANSYS software and validate the experimental and theoretical results.
5. To compare theoretical, experimental, and computational results to evaluate the effectiveness of the pendulum mass tuned damper in reducing vibrations.

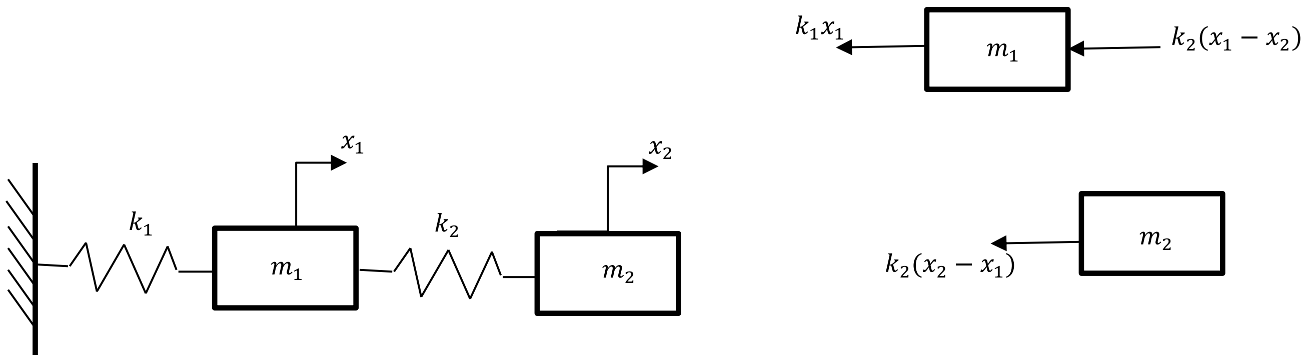
CEP Justification

- The project requires the integration of theoretical modeling, experimental validation, and computational simulations, involving advanced techniques in dynamics and vibration analysis.
- It addresses the real-world challenge of vibration mitigation, accounting for interdependent variables such as natural frequency, damping ratio, and the dynamic interaction between the structure and the tuned mass damper.
- The need to validate and reconcile results across theoretical, experimental, and computational domains adds complexity, highlighting the problem's multidisciplinary and non-routine nature.

Methodology

Theory

The picture of the experimental setup is attached at the end of this document. Whereas the setup is modelled as an equivalent system. The masses of the structure components were accounted individually, then the combined mass of the system is modelled as m_1. The total calculated stiffness is modelled as k_1. Then the mass for damper is calculated in theory and taken equivalent as m_2 and k_2 for the stiffness value of damper.



Calculation for m_1:

$$mass\ of\ steel\ bars + wooden\ plate = 0.335\ kg\ \times 4 + 0.56\ kg = 1.9\ kg$$

Calculation for k_1:

For a steel bars the stiffness is calculated as follow. As, they are in parallel combination, so stiffness adds directly.

$$k = \frac{AE}{L} = \frac{\left(\frac{\pi 0.0076^2}{4}\right)(200 \times 10^9)}{0.33} = 27450 \frac{N}{m}$$

$$k_1 = 4 \times k = 109800 \frac{N}{m}$$

Calculation for k_2:

$$k_2 = \frac{mg}{L} = \frac{0.62 \times 9.81}{0.12} = 50.685 \frac{N}{m}$$

Calculation for m_2 (damper):

The calculations for damper are done based on theory of mass tuned damper, where the natural frequencies are equivalent to excitation frequencies and results developed on the basis of theory is:

$$m_2 = \frac{F}{\omega^2 x_2}$$

The assumption is made for the maximum amplitude of damper that is $5cm$. The frequency and force value taken from experimentation; the tunning of damper is done.

$$m_2 = \frac{300}{157^2 \times 0.05} = 0.6kg$$

Equation of motion

For m_1:

$\rightarrow +ve \sum F = m_1 \ddot{x}$

$-k_1 x_1 - k_2(x_1 - x_2) = m_1 \ddot{x_1}$

$$m_1 \ddot{x_1} + (k_1 + k_2)x_1 - k_2 x_2 = 0 \qquad eq\ A$$

For m_2:

$\rightarrow +ve \sum F = m_2 \ddot{x}$

$-k_2(x_2 - x_1) = m_2 \ddot{x_2}$

$$m_2 \ddot{x_2} - k_2 x_1 + k_2 x_2 = 0 \qquad eq\ B$$

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x_1} \\ \ddot{x_2} \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

Assuming S.H.M response, the solutions are

$$x_1(t) = X_1 \cos(\omega t + \theta)$$

$$x_2(t) = X_2 \cos(\omega t + \theta)$$

Differentiating both equations,

$$\dot{x_1}(t) = -X_1 \omega \sin(\omega t + \theta)$$

$$\ddot{x_1}(t) = -X_1 \omega^2 \cos(\omega t + \theta)$$

And

$$\dot{x_2}(t) = -X_2 \omega \sin(\omega t + \theta)$$

$$\ddot{x_2}(t) = -X_2 \omega^2 \cos(\omega t + \theta)$$

Substituting values into $eq\ A$, we get,

$$-m_1 X_1 \omega^2 \cos(\omega t + \theta) + (k_1 + k_2) X_1 \cos(\omega t + \theta) - k_2 X_2 \cos(\omega t + \theta) = 0$$

$$-m_1 X_1 \omega^2 + (k_1 + k_2) X_1 - k_2 X_2 = 0 \qquad eq\ AA$$

Substituting values into $eq\ B$, we get,

$$-m_2 X_2 \omega^2 \cos(\omega t + \theta) - k_2 X_1 \cos(\omega t + \theta) + k_2 X_2 \cos(\omega t + \theta) = 0$$

$$-m_2 X_2 \omega^2 - k_2 X_1 + k_2 X_2 = 0 \qquad eq\ BB$$

$$\begin{bmatrix} -m_1 \omega^2 + (k_1 + k_2) & -k_2 \\ -k_2 & -m_2 \omega^2 + k_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

Theoretical Frequency Calculations

Now,

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} \neq \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

So,

$$\begin{bmatrix} -m_1\omega^2 + (k_1 + k_2) & -k_2 \\ -k_2 & -m_2\omega^2 + k_2 \end{bmatrix} = 0$$

For a non-trivial solution.

$$\begin{vmatrix} -m_1\omega^2 + (k_1 + k_2) & -k_2 \\ -k_2 & -m_2\omega^2 + k_2 \end{vmatrix} = 0$$

Putting values calculated above,

$$\begin{vmatrix} -1.9\omega^2 + 109850.685 & -109800 \\ -50.685 & -0.62\omega^2 + 50.685 \end{vmatrix} = 0$$

Let $\lambda = \omega^2$, simplifying

$$1.178\lambda^2 - 68161.55\lambda + 2568.969 = 0$$

$$\lambda = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

Solving, the roots are

$$\lambda_1 = 0.0037689 \Rightarrow \omega_1 = 0.06 \approx 0 \; (rigid\ body\ motion\ at\ first\ frequency)$$

$$\lambda_2 = {\omega_2}^2 = 40425.89963 \left(\frac{rad}{s}\right)^2$$

$$\omega_2 = 201.06 \frac{rad}{s}$$

$$f = 32\ Hz$$

Note: Important results regarding these calculations are discussed at the end.

Experimental Procedure

1. The primary structure was made with wooden block attached with the four steel bars (observations for dimensions are in next section). This part is modelled as first mass and its stiffness value in the above analysis. The setup picture is attached at the end of this document.
2. Then the value for the damper mass is calculated and chosen. The mass was attached to the primary structure and acted as a damper mass. At this point the prototype was ready.
3. Then it was brought for experimentation. The sensors were placed onto the primary structure, and it was hit with a hammer showing the values onto the software output screen. From there the data set for natural frequency vs amplitude was extracted. Later it was put into excel to make full real time plot and then in MATLAB for plot and its frequency value.

4. Then the damper was attached, again the analysis was done and data set for frequency vs amplitude was generated and analyzed. The plot was obtained from excel and again from MATLAB and its damping ratio ζ experimental was also obtained from MATLAB for this experiment.
5. These natural frequency and damping ratio value then compared with those of ANSYS ones and theoretical frequency value in results section.

Observations

These are the measured physical measurements from experiment.

$\boldsymbol{Quantity}$	$\boldsymbol{Value}$
$\boldsymbol{Length\ of\ a\ structure}$	$0.3302\ m$
$\boldsymbol{Radius\ of\ a\ steel\ rod}$	$0.0076\ m$
$\boldsymbol{Area\ of\ a\ wooden\ block}$	$0.96\ m^2$
$\boldsymbol{Length\ of\ a\ pendulum}$	$0.12\ m$
$\boldsymbol{Mass\ of\ a\ damper\ (calculated)}$	$0.620\ kg$
$\boldsymbol{Mass\ of\ a\ structure}$	$1.9\ kg$

Computations

In this section, the procedure and setup for computations (theoretical, experimental & numerical) are shown, while the ***results*** are shown and compared in the results section with plots.

Theoretical Values

1. The value of frequency for first mode is $\boldsymbol{\omega_1 = 0.06 \approx 0\ (rigid\ body\ motion\ at\ first\ frequency)}$.
2. The value of natural frequency found out *to be* $\boldsymbol{f = 32\ Hz}$.

Experimental

The data set for frequency vs amplitude were generated first without damper for natural frequency and then second one after with damper for damping ratio and plot showing dying out amplitude. The MATLAB scripts are shown in this section whereas the plots on both excel, and MATLAB are shown in results sections.

Without damper

```
% Data for natural frequency vs amplitude

frequency = [0, 0.806452, 1.612903, 2.419355, 3.225806, 4.032258, 4.83871, ...

    5.645161, 6.451613, 7.258065, 8.064516, 8.870968, 9.677419, ...

    10.483871, 11.290323, 12.096774, 12.903226, 13.709677, ...

    14.516129, 15.322581, 16.129032, 16.935484, 17.741935, ...

    18.548387, 19.354839, 20.16129, 20.967742, 21.774194, ...

    22.580645, 23.387097, 24.193548, 25, 25.806452, 26.612903, ...

    27.419355, 28.225806, 29.032258, 29.83871, 30.645161, ...

    31.451613, 32.258065, 33.064516, 33.870968, 34.677419, ...

    35.483871, 36.290323, 37.096774, 37.903226, 38.709677, ...

    39.516129, 40.322581, 41.129032, 41.935484, 42.741935, ...

    43.548387, 44.354839, 45.16129, 45.967742, 46.774194, ...
```

```
    47.580645, 48.387097, 49.193548, 50];

amplitude = [-10.819573, -12.096677, -45.501293, -46.239852, -55.581419, ...
    -47.772725, -41.880334, -37.754729, -34.905035, -32.714019, ...
    -30.085326, -28.064619, -26.215868, -24.984212, -24.419576, ...
    -22.334911, -19.823124, -18.647535, -17.200715, -15.45298, ...
    -14.341031, -12.783555, -11.529432, -10.230539, -8.323766, ...
    -6.709328, -5.405042, -3.959216, -1.896982, 0.876048, ...
    3.877329, 7.439527, 11.299654, 12.405765, 9.492175, ...
    3.728478, 1.308552, -0.436584, -1.602838, -2.30439, ...
    -2.398705, -2.350015, -2.866457, -3.294854, -3.659834, ...
    -4.403565, -4.637664, -4.517318, -4.955492, -5.596757, ...
    -6.21742, -6.352384, -6.288849, -6.689764, -7.373242, ...
    -7.533552, -7.440044, -7.739709, -8.150861, -8.399297, ...
    -8.71626, -8.835929, -8.707808];

% Create the plot
figure;
plot(frequency, amplitude, '-o', 'LineWidth', 1.5, 'MarkerSize', 5);
grid on;

% Add labels and title
xlabel('Natural Frequency (Hz)');
ylabel('Amplitude (dB)');
title('Natural Frequency vs Amplitude');
xlim([0, 50]); % Set x-axis limits
ylim([-60, 10]); % Set y-axis limits
```

With damper (for ζ)

```
% Data for frequency vs amplitude with secondary mass (mass tuned damper)
frequency = [0, 0.806452, 1.612903, 2.419355, 3.225806, 4.032258, ...
    4.83871, 5.645161, 6.451613, 7.258065, 8.064516, ...
    8.870968, 9.677419, 10.483871, 11.290323, 12.096774, ...
    12.903226, 13.709677, 14.516129, 15.322581, 16.129032, ...
    16.935484, 17.741935, 18.548387, 19.354839, 20.16129, ...
    20.967742, 21.774194, 22.580645, 23.387097, 24.193548, ...
    25, 25.806452, 26.612903, 27.419355, 28.225806, ...
    29.032258, 29.83871, 30.645161, 31.451613, 32.258065, ...
    33.064516, 33.870968, 34.677419, 35.483871, 36.290323, ...
    37.096774, 37.903226, 38.709677, 39.516129, 40.322581, ...
```

```
41.129032, 41.935484, 42.741935, 43.548387, 44.354839, ...
45.16129, 45.967742, 46.774194, 47.580645, 48.387097, ...
49.193548, 50, 50.806452, 51.612903, 52.419355, ...
53.225806, 54.032258, 54.83871, 55.645161, 56.451613, ...
57.258065, 58.064516, 58.870968, 59.677419, 60.483871];
amplitude = [2.17887, 2.611507, -36.691347, -47.125508, -46.46793, ...
-41.969869, -34.883933, -35.593308, -32.965939, -30.458907, ...
-28.207974, -24.841593, -28.060548, -22.373238, -21.669219, ...
-17.878414, -20.977175, -13.699288, -18.569909, -14.030842, ...
-7.244444, -15.184765, -11.130495, -8.553603, -3.791722, ...
-11.357853, 0.533974, 2.844556, 9.237347, 14.811617, ...
25.285591, 35.700015, 12.435653, -0.343017, -0.222005, ...
2.566909, -7.575285, 1.713266, -9.818035, -7.077247, ...
-4.275504, -6.635405, -0.588422, -10.747763, -5.476512, ...
-10.573223, -1.896611, -9.52754, -10.284295, -10.033071, ...
-7.541253, -7.742991, -8.719222, -6.583139, -8.838422, ...
-5.756194, -11.278326, -8.380058, -8.195005, -7.824788, ...
-9.183283, -8.305484, -13.744573, - 7.893325, -5.546677, ...
-11.368567, -10.850377, -8.175314, -13.766724, -7.093848, ...
-12.268889, -8.481792, -10.55583, -9.320408, -10.990333, -7.665496];
% Create the plot
figure;
plot(frequency, amplitude, '-o', 'LineWidth', 1.5, 'MarkerSize', 5);
grid on;
% Add labels and title
xlabel('Frequency (Hz)');
ylabel('Amplitude (dB)');
title('Frequency vs Amplitude with Mass Tuned Damper');
xlim([0, 60]); % Set x-axis limits
ylim([-60, 40]); % Set y-axis limits
% Calculate the damping ratio (zeta) using logarithmic decrement
% Find peaks in the amplitude data
[peaks, locs] = findpeaks(amplitude);
% Initialize zeta
zeta = NaN; % Default value if not enough peaks
% Calculate the logarithmic decrement if there are enough peaks
if length(peaks) > 1
    delta = log(peaks(1:end-1) ./ peaks(2:end));
```

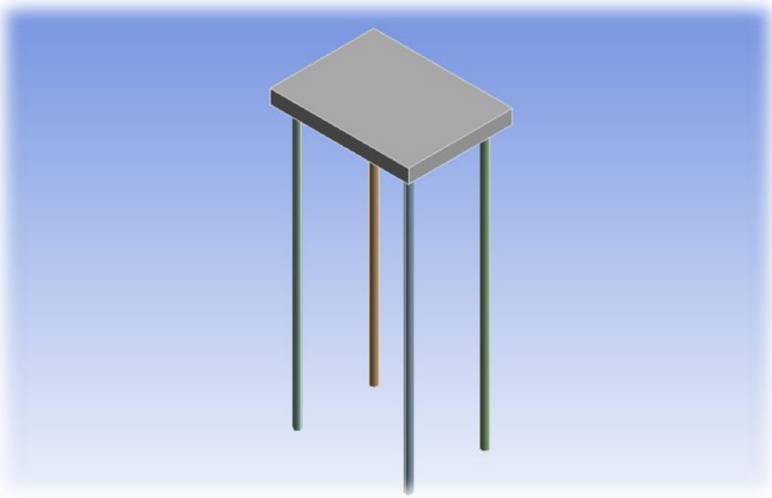
```
zeta = mean(delta) / sqrt((2 * pi)^2 + mean(delta)^2);
end
```

Numerical (ANSYS)

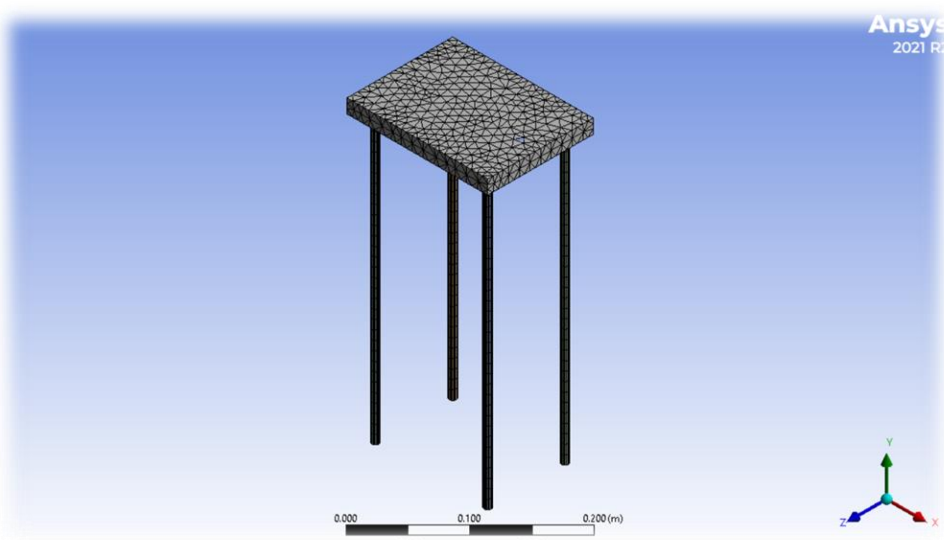
In this section, the geometry creation in SOLIDWORKS, mesh creation, boundary conditions are shown for both with and without damper. Then outcome graphs of frequency and damping ratio values are shown in results sections. Also, the MATLAB was used to extract the theoretical damping ratio value from the underdamped plot given by ANSYS, and its script is attached here.

Without damper

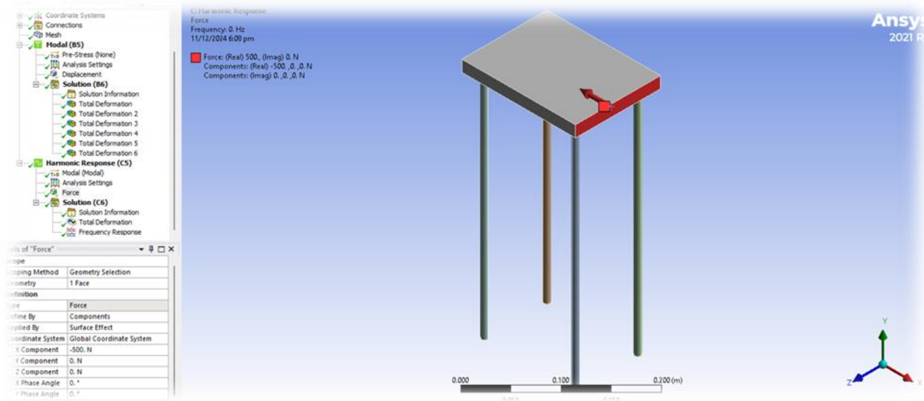
1. **Geometry on SOLIDWORKS**
   The geometry was created on SOLIDWORKS with the same dimensions.



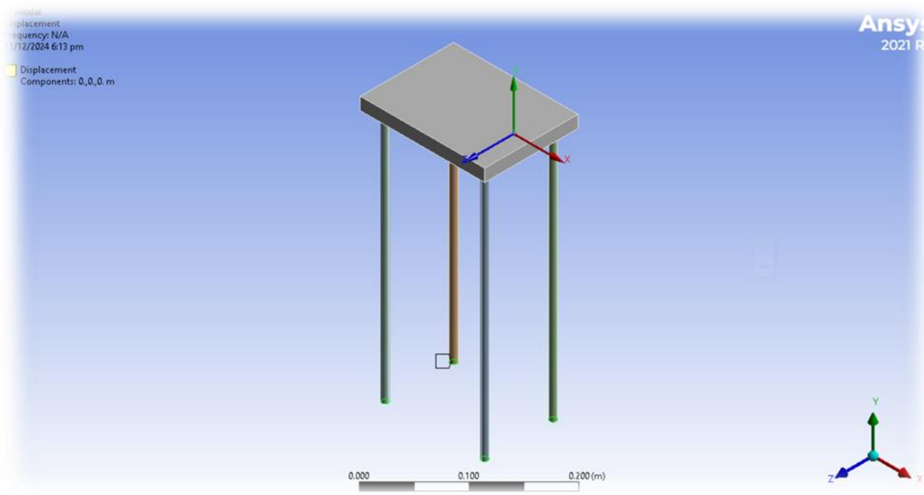
2. **Meshing**
   The ***tetrahedral*** meshing element was chosen, and meshing was done.



3. **Force**
   The load was applied at one end for its natural frequency.

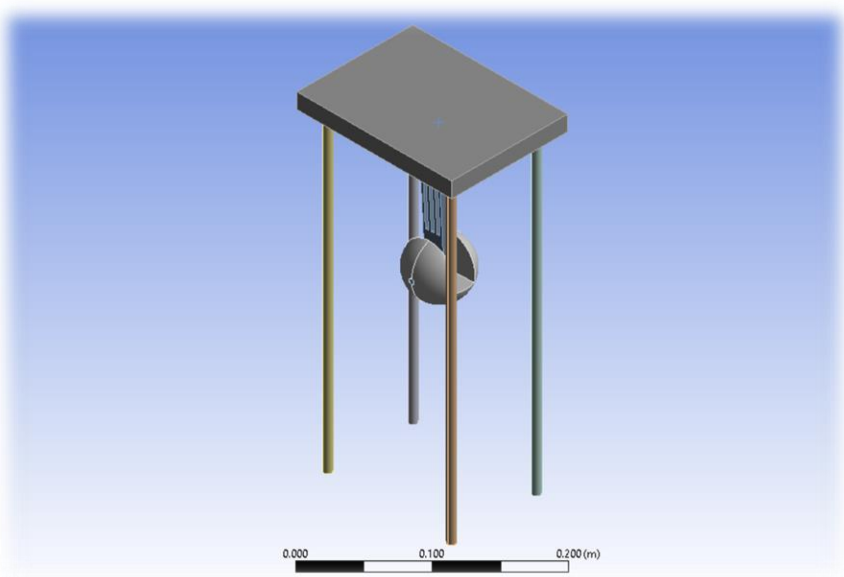


4. **Fixed Support**
   The ends were fixed for proper supporting and analysis.



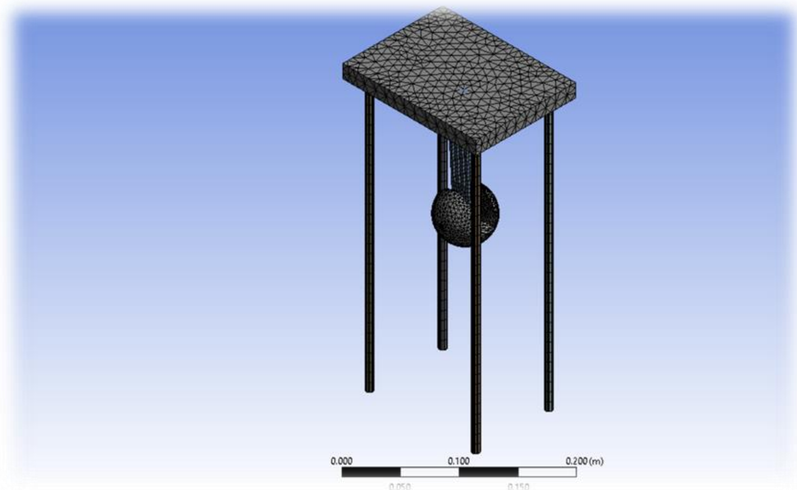
With damper

1. **Geometry**
   The geometry was updated with damper attached.



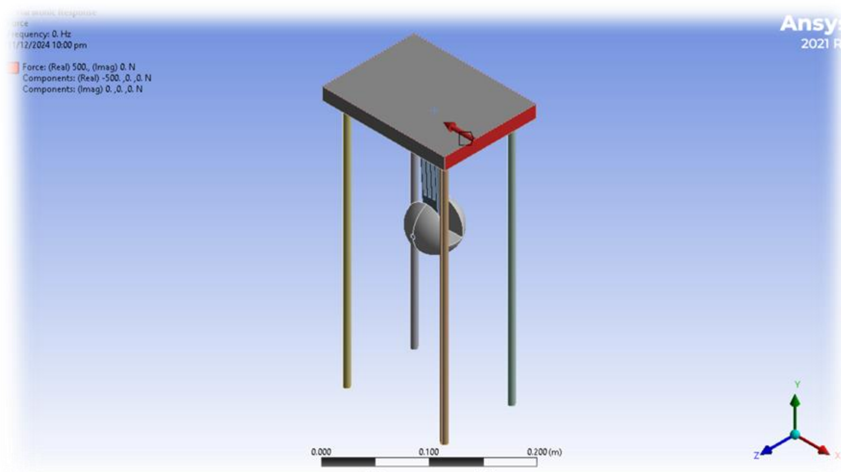
2. **Meshing**

The meshing was done for analysis. Fine mesh was created on the damper for better results.



3. **Force**

The force was applied for analysis to carried out.



4. **MATLAB code for theoretical damping ratio**

```
5. % Given frequency and amplitude data
6. data = [ 3, 9.1239e-003;6, 9.7953e-0039, 1.115e-002;12, 1.378e-002;15, 1.9612e-00218, 3.9582e-00221, 0.23725;24, 2.7039e-002; 27, 1.3792e-002;30, 9.0762e-003;33, 6.6973e-003;36, 5.2864e-003;39, 4.3685e-003;42, 3.7359e-003;45, 3.2823e48, 2.9823e-003;51, 2.72754, 2.5593e-003;57, 2.4509e-003;60, 2.3971e-003];
7. % Extract frequency and amplitude
8. frequency = data(:, 1);
9. amplitude = data(:, 2);
10. % Calculate logarithmic decrement
11. n = 1; % Number of cycles between measurements
12. log_decrement = log(amplitude(1:end-n) ./ amplitude(1+n:end));
13. % Calculate average logarithmic decrement
14. delta = mean(log_decrement);
15. % Calculate damping ratio
16. zeta = delta / sqrt((2*pi)^2 + delta^2);
17. % Display the damping ratio
18. fprintf('Estimated Damping Ratio (zeta): %.4f\n', zeta);
19. % Plotting the amplitude vs frequency
20. figure;
21. plot(frequency, amplitude, '-o', 'LineWidth', 2);
22. xlabel('Frequency (Hz)');
23. ylabel('Amplitude');
24. title('Amplitude vs Frequency');
25. grid on;
```

Results

Theoretical Calculations

1. The first mode frequency $\boldsymbol{\omega_1 = 0.06 \approx 0}$ $(\boldsymbol{rigid\ body\ motion\ at\ first\ frequency})$, shows that at the very start of vibration the damper mass moves along (in phase movement) with the primary mass but after the instant later the damping starts and frequency value changes.
2. The natural frequency calculated $\boldsymbol{\omega_2 = 201.062\frac{rad}{s}, f = 32\ Hz}$. This value of frequency might be subtle larger than the experimental and computational value because the ***damping was not considered in the theoretical analysis.***

Experimental Plots and Values

Without damper

1. The data set for natural frequency vs amplitude was put in excel first and the plot was generated.

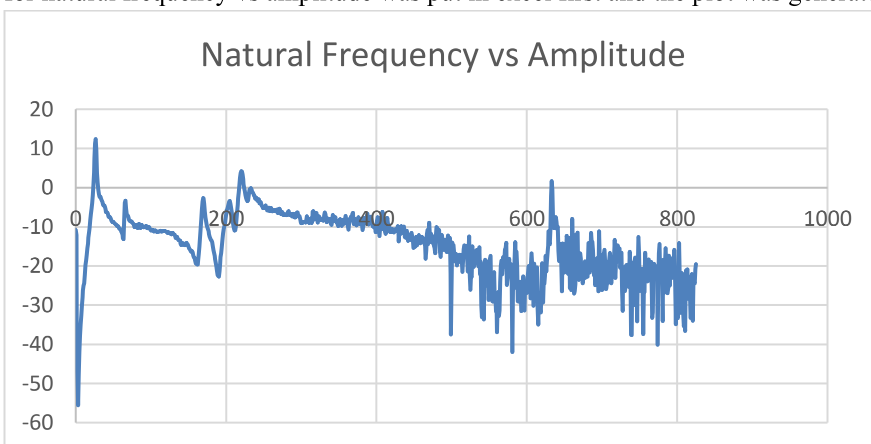


Figure 1 Experimental Natural Freq VS Amplitude (Excel)

2. Then the same data set was used to generate the same plot in MATLAB environment for the peak value. The value for natural frequency comes out to be $\boldsymbol{26.6129\ Hz}$.

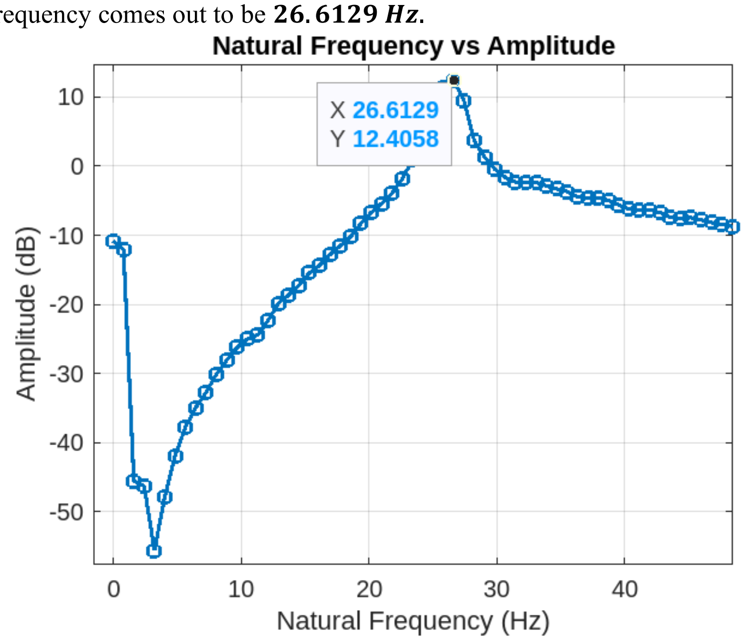


Figure 2 Experimental Natural frequency VS Amplitude (MATLAB)

With damper

1. The damp frequency vs amplitude plot was generated in excel first. The damping in amplitude after application of damper can be observed.

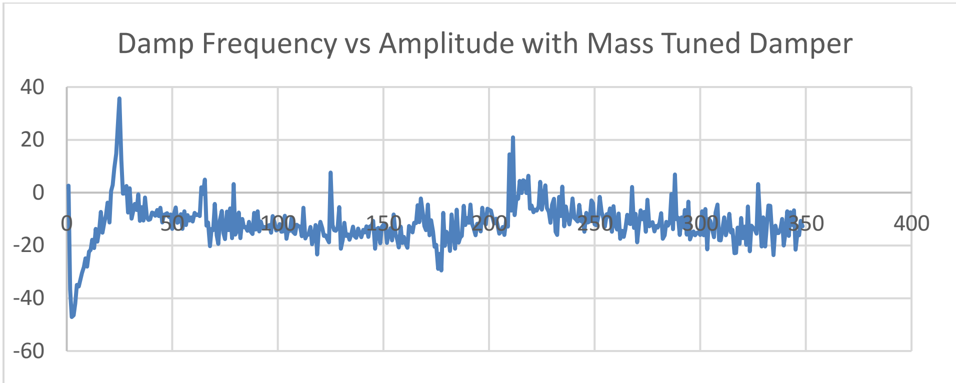


Figure 3 Experimental Damp Frequency vs Amplitude with Mass Tuned Damper (Excel)

2. The same plot was generated in MATLAB for experimental damping ratio value. It can be noted that the damp frequency also reduced after adding damper with value $\mathbf{25\ Hz}$.

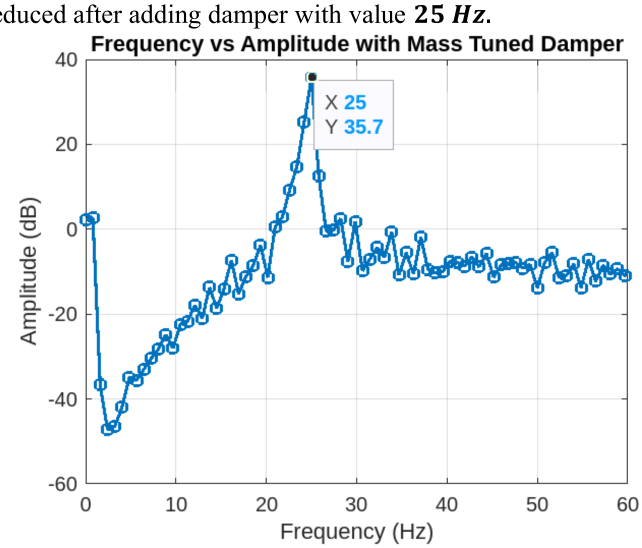
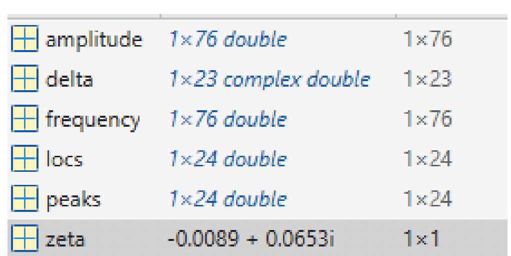


Figure 4 Experimental Damp Frequency vs Amplitude with Mass Tuned Damper (MATLAB)

3. Then the experimental value of damping ratio was calculated in MATLAB and has magnitude value $\mathbf{0.06594675}$. The value is very near to zero, it means that the structure damps very slowly with the passage of time.



amplitude	1×76 double	1×76
delta	1×23 complex double	1×23
frequency	1×76 double	1×76
locs	1×24 double	1×24
peaks	1×24 double	1×24
zeta	-0.0089 + 0.0653i	1×1

Figure 5 Experimental Zeta without magnitude taken (MATLAB)

4. It can also be verified that $\boldsymbol{\omega_d = \omega_n\sqrt{1-\zeta^2} = 25.7\ Hz}$ that is comparable to $\mathbf{25\ Hz}$.

Computational Plots & Values (ANSYS)

Without Damper

1. The natural frequency of structure in ANSYS environment comes out to be exactly $\mathbf{25}\ \boldsymbol{Hz}$ and the value is comparable with theoretical and experimental results.

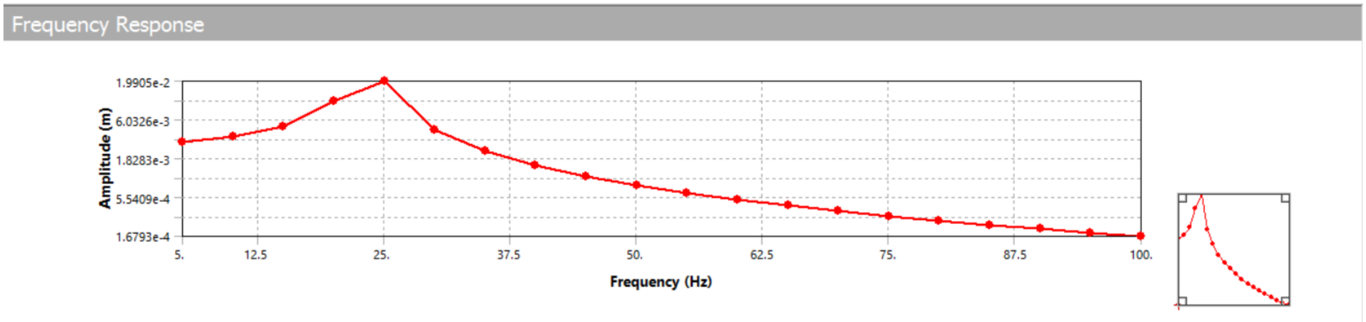


Figure 6 Theoretical Natural Frequency VS Amplitude (ANSYS)

With Damper

1. In damp frequency response the peak value for damp frequency comes out to be $\mathbf{21}\ \boldsymbol{Hz}$ that is comparable to the experimental one.

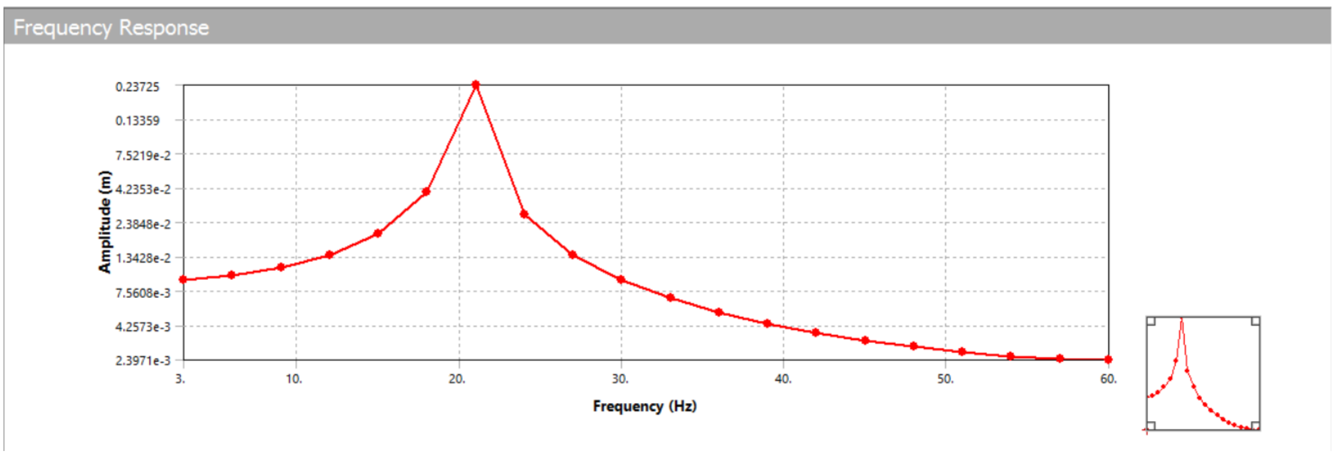


Figure 7 Theoretical Damp Frequency VS Amplitude (ANSYS)

2. The deflections in damper can be seen after the application of hammer force. As the red color on the bottom edge of damper shows its maximum deflection at that point. And negligible deflection on the primary structure. ***This analysis explicitly shows that the energy from the primary building has been transfer to the damper and the amplitude has died out for primary mass.***

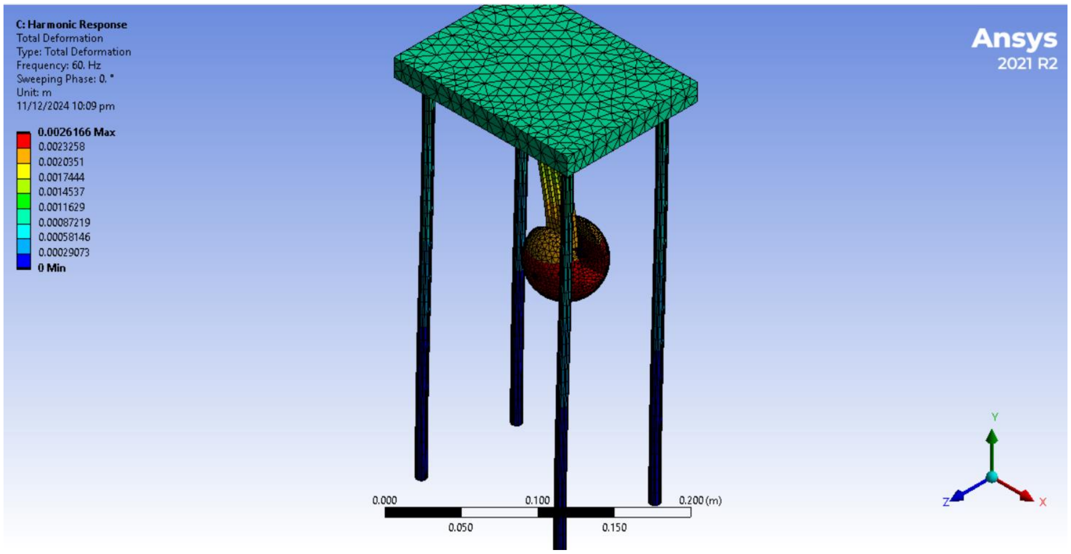


Figure 8 Amplitude of Damper

3. The theoretical value for damping ratio is generated in MATLAB by giving the plot coordinated given by ANSYS for damp response. The value comes out to be $\mathbf{0.0560}$ that is comparable to experimental value.

Workspace

Name	Value	Size
amplitude	*20×1 double*	20×1
data	*20×2 double*	20×2
delta	0.0703	1×1
frequency	*20×1 double*	20×1
log_decrem...	*19×1 double*	19×1
n	1	1×1
zeta	0.0560	1×1

Figure 9 Theoretical Zeta value from ANSYS (MATLAB)

Discussions

The comparison (validation) of different results can be seen in this section in one place in tabular format.

1. The validation of natural frequency.

Natural frequency	***Value*** (*Hz*)
Theoretical	32(*damping ignore in equations*)
Experimental	26.6129
Computational (***ANSYS***)	25

Figure 10 Comparison of Theoretical, Experimental & Computational Natural Frequency

2. The validation of damp frequency.

Damp frequency	***Value*** (*Hz*)
Experimental	25
Computational (***ANSYS***)	21

Figure 11 Comparison of Experimental & Computational Damp Frequency

3. The validation of damping ratio.

Damping Ratio ζ	***Value***
Experimental	0.06594675
Computational (***ANSYS***)	0.0560

Figure 12 Comparison of Experimental & Computational Damping Ratio

Sources of Error

1. There is error in theoretical natural frequency values because in theoretical analysis damping was not considered.
2. There is error in damp frequency values because of some modelling assumptions, tuned mass characteristics.
3. The error in damping ratio can be due to noise measurement in experimental data and approximations.

Conclusions

All the theoretical, numerical and experimental results match with minimum error. The analysis has been done. The reasons for possible errors have been quoted. However, ***errors can be minimized*** by:

1. Align numerical and experimental setups by accurately modeling material properties, damping mechanisms, and boundary conditions.
2. Reduce measurement errors with precise sensors, calibrated equipment, and standardized data analysis methods.

References

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[2] M. Basili, F. Busato, and M. De Angelis, "Integrated seismic and energetic rehabilitation of existing buildings based on the tuned mass damper concept," *Results in Engineering*, vol. 24, p. 103552, Dec. 2024, doi: 10.1016/J.RINENG.2024.103552.

[3] H. Wang and S. Zhu, "Latching control for pendulum tuned mass damper: Theoretical analysis and proof-of-concept experiment," *Mech Syst Signal Process*, vol. 224, p. 112158, Feb. 2025, doi: 10.1016/J.YMSSP.2024.112158.