ME 352

Group No.- 2

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Experiment 2: *Tuned mass damper (for a beam with unbalanced rotor)*

Under the guidance of
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1. Introduction

The objective of this experiment is to reduce the amplitude of vibrations of a beam with an unbalanced rotor (essentially a harmonic excitation) using tuned mass damper. When an external force is applied to a beam, it undergoes vibrations, and these vibrations can become significant, especially if the force is exerted at or close to the beam's natural frequency. In certain instances, these vibrations may lead to damage to the beam or the supporting structure.

A tuned mass damper serves as a device used to reduce the amplitude of vibrations in a structure. This device comprises a mass connected to the structure through a spring and a damper. The spring is utilized to adjust the device to the natural frequency of the structure, while the damper dissipates energy and minimizes the amplitude of the vibrations.

An unbalanced rotor refers to a rotating component with an uneven distribution of mass. As the rotor rotates, it generates harmonic excitation that induces vibrations in the beam.

Experiment [Video]

2. Experimental Design and Fabrication Details

2A. Experimental Design

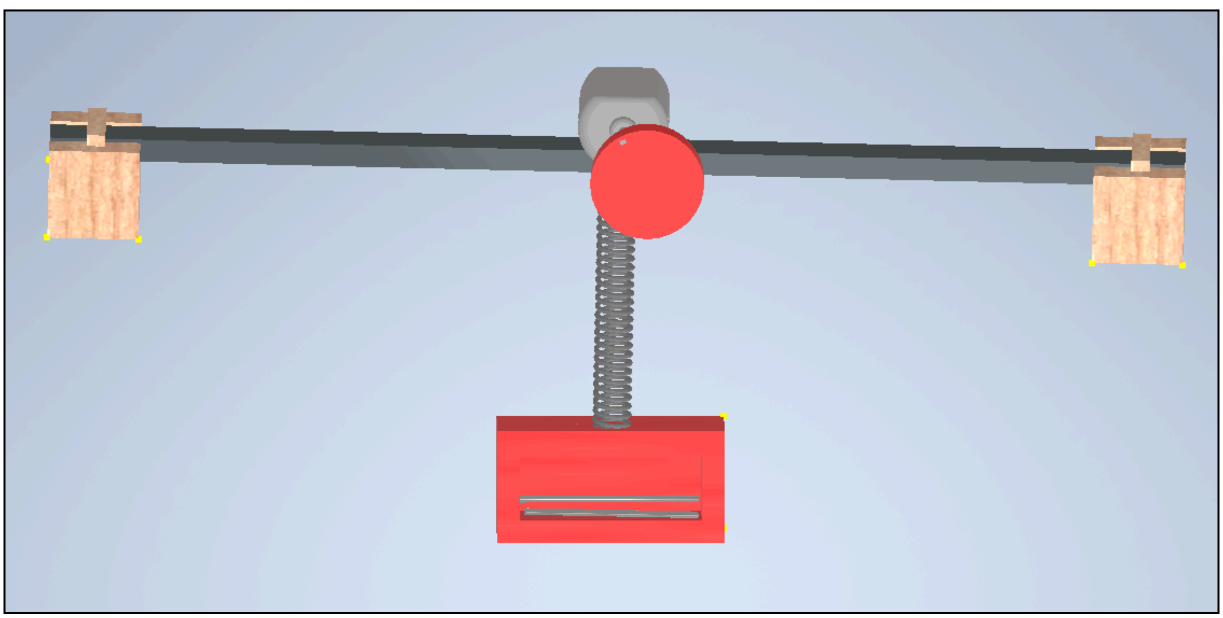


Fig. CAD Experimental design

The experiment involves a tuned mass damper attached to the beam and a harmonic excitation applied to the beam using an unbalanced rotor.

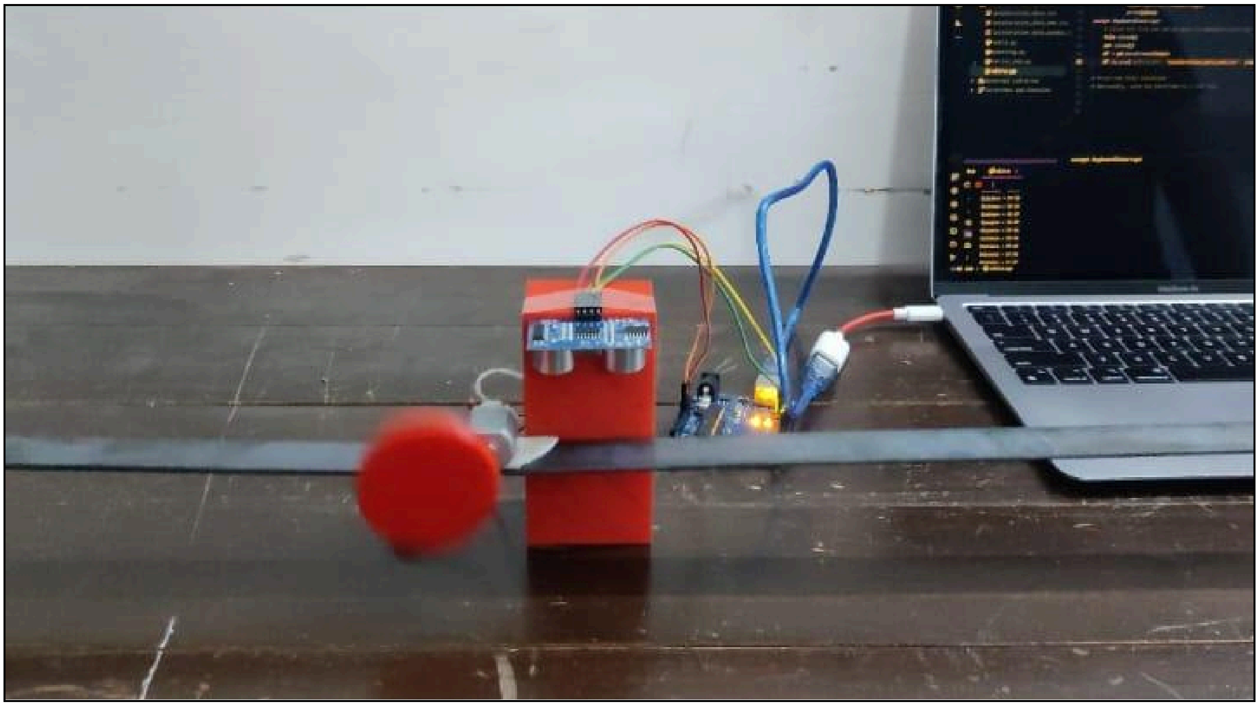


Fig.- Experimental setup without spring mass damper

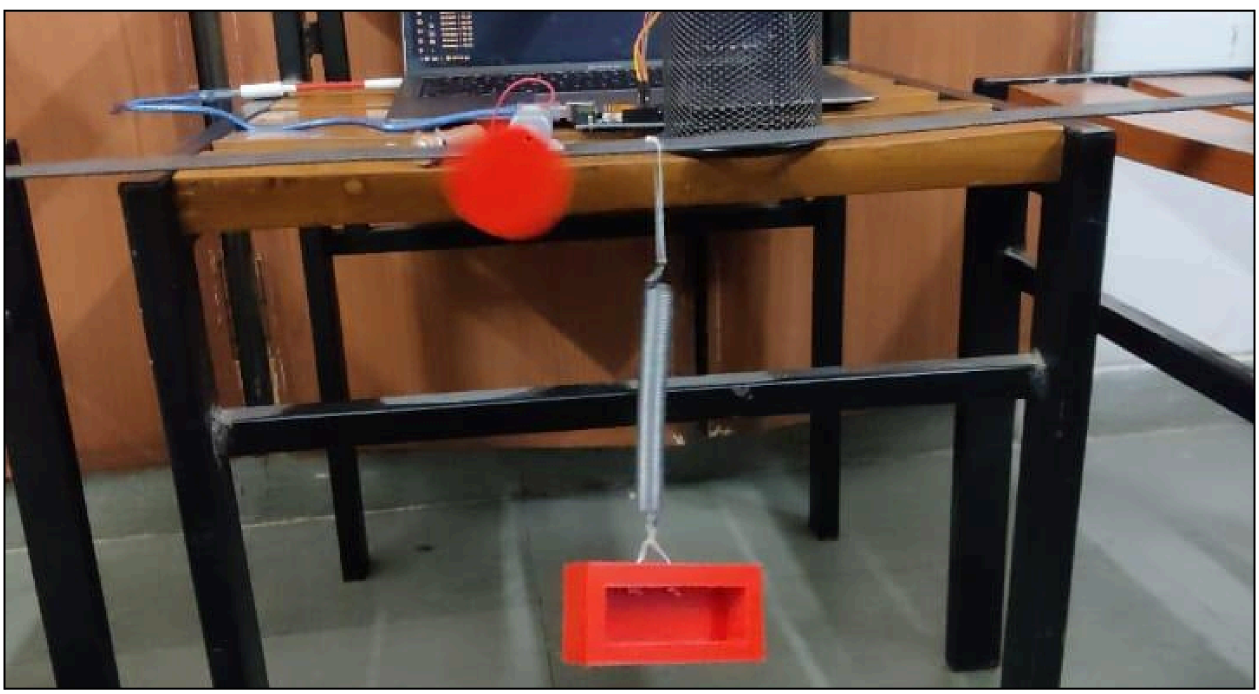
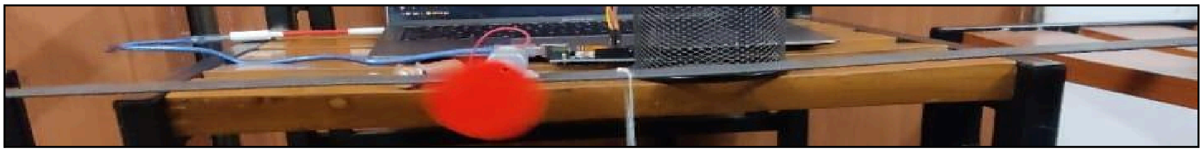


Fig.- Experimental setup with spring mass damper

2B. Fabrication details

a. Simply supported beam



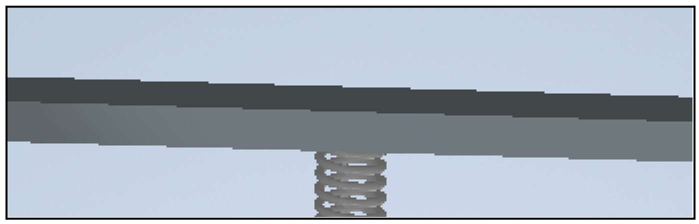


Fig. Simply supported beam

1. A mild steel of 90cm was cut from the manufacturing workshop, IITGN.
2. Mild steel rod is placed on two wooden blocks for support.

b. Wooden Block

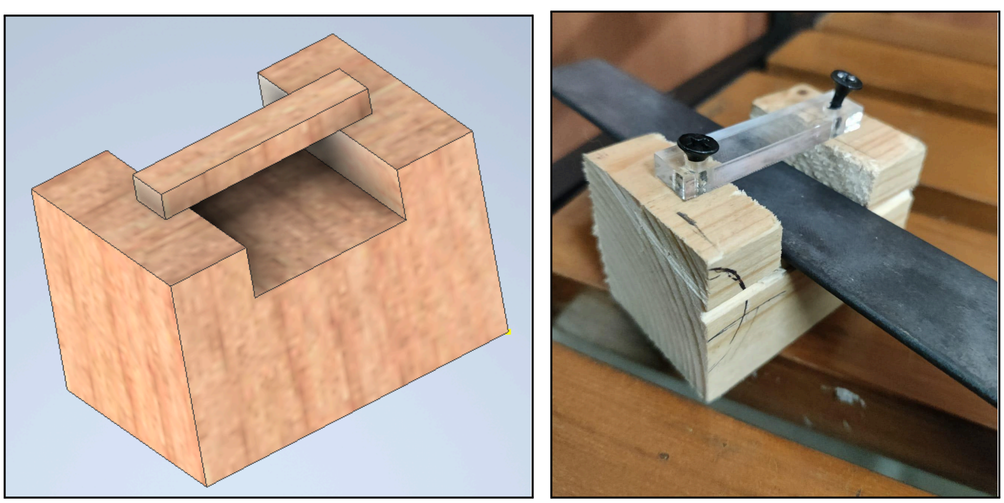
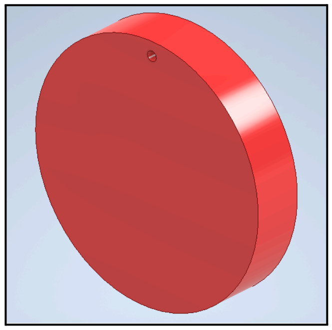
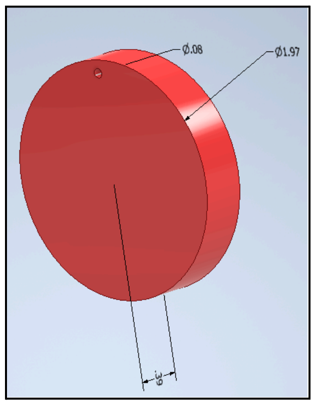


Fig. Wooden blocks

1. From the manufacturing workshop, IITGN, Big wooden blocks were made available and according to need we cut them and designed them by applying carpentry basics that we studied in our Manufacturing Course.

C. Eccentric mass

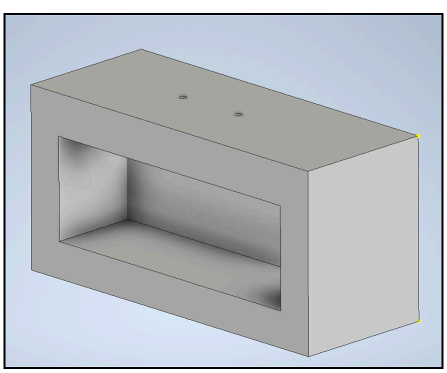




Mass - 10g
Diameter (d) =2 cm
Thickness (t) = 0.39 cm

We made the eccentric mass using 3D printing at the Tinkerer's lab in IITGN because using other materials, like wood, would have made it too heavy. With a low-voltage motor, a heavy mass would have made it difficult for the motor to rotate.

D. Mass of TMD



For hanging the mass to the spring, a cuboidal box with one side open is 3D printed. One side is kept open to add and remove mass as required. *Firstly we Manufactured TMD with the help of wood, but it was higher in weight than calculated mass.*

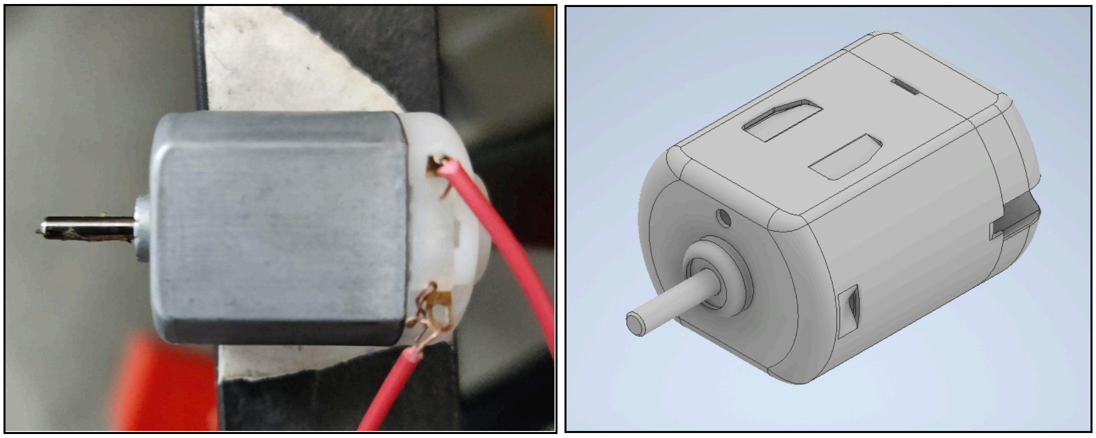


Fig.- DC Motor

Items/Equipments	Material used
Beam support	Wood
Beam	Mild Steel
Eccentric Mass	3D print
TMD Mass Box	Wood/3D print

3. Theoretical/mathematical modeling and analysis

Free vibration equation for transverse vibration of beam:

$EI\left(\partial^4 y/dx^4\right) + m\left(\partial^2 y/dt^2\right) = 0$

where:

EI is the flexural rigidity of the beam

y is the deflection of the beam as a function of position x and time t

m is the mass per unit length of the beam

Let $y(x, t) = \phi(x) sin\omega t$

$\left(\partial^4 \phi/dx^4\right) - \lambda^4 \phi = 0$, where $\lambda^4 = m\omega^2/EI$

$\phi(x) = A_1 cosh\,\lambda x + A_2 sinh\,\lambda x + A_3 cos\,\lambda x + A_4 sin\,\lambda x$

A_1 , A_2, A_3, A_4 are constants of integration that have to be found from boundary conditions.

Simply supported beam:

$\phi(x) = A_1 cosh\,\lambda x + A_2 sinh\,\lambda x + A_3 cos\,\lambda x + A_4 sin\,\lambda x$

$\phi''(x) = A_1 \lambda^2 cosh\,\lambda x + A_2 \lambda^2 sinh\,\lambda x - A_3 \lambda^2 cos\,\lambda x - A_4 \lambda^2 sin\,\lambda x$

Boundary conditions for simply supported ends:

At x = 0, $\phi(0) = 0;\ and\ El\left(\partial^2\phi/dx^2\right)_{x=0} = 0$

At x = L, $\phi(L) = 0;\ and\ El\left(\partial^2\phi/dx^2\right)_{x=L} = 0$

First condition gives $A_1 + A_3 = 0$ and $A_1 - A_3 = 0$ which means $A_1 = A_3 = 0$

$\phi(x) = A_2 sinh\,\lambda x + A_4 sin\,\lambda x$

$\phi(L) = A_2 sinh\,\lambda L + A_4 sin\,\lambda L$

$\phi''(L) = A_2 \lambda^2 sinh\,\lambda L - A_4 \lambda^2 sin\,\lambda L$

Hence we get

$A_2 sinh\,\lambda L + A_4 sin\,\lambda L = 0$

$A_2 sinh\,\lambda L - A_4 sin\,\lambda L = 0$

$A_2 = 0$

Hence $sin\,\lambda L = 0$

$\lambda L = n\pi$ (n = 1,2,..)

$\lambda = n\pi/L$

$\omega_n = \lambda_n^{\ 2}\sqrt{EI/mL^4}$

$\omega_n = \pi^2 n^2 \sqrt{EI/mL^4}$

and

$\phi(x) = A_4 sin\, \lambda x$

Beam dimensions:

E = 210 GPa = 210 * 10^9 Pa
b = 2.5cm = 2.5 * 10^-2 m
h = 0.2cm = 2 * 10^-3
l = 0.9m
m = 1.18kg
I = bh^3/12 = 1.66 * 10^-11

$\omega_n = \pi^2 n^2 \sqrt{EI/\rho AL^4}$

$\omega_n = \pi^2 n^2 \sqrt{EI/mL^3}$

ω_1=3.14 * 3.14 $\sqrt{210 \ * \ 10^9 \ * \ 1.66 \ * \ 10^{-11} / 1.18 \ * \ 0.9^3}$ * 1*1
= 19.86 rad/sec

ω_2 =79.4 rad/sec
ω_3 =178.8 rad/sec
ω_4 =317.89 rad/sec

Values of **natural frequency**:
"When given an excitation and left to vibrate on its own, the frequency at which a simply supported beam will oscillate is its natural frequency. This condition is called Free vibration. The value of natural frequency depends only on system parameters of mass and stiffness" [4]

f = $\omega/2\pi$

$f_1 =$ 3.16 Hz

$f_2 = 12.63$ Hz
$f_3 = 28.45$ Hz
$f_4 = 50.59$ Hz

Spring constant:
To measure the spring constant, we acquire a known mass and measure the deflection of spring before and after the mass is suspended. We used the formula F=kx where k is the spring constant and x is the displacement of the spring in order to calculate the spring constant.

mass(in kg)	Initial length of spring (in m)	Final length of spring(in m)
0.5	0.12	0.45

F=0.5*9.8 N=4.9N
x=0.45m-0.12m=0.33m
k(spring constant)=F/x
=4.9/0.33
=14.84N/m

So,the spring constant will be 14.84N/m.

f= $\frac{1}{2\pi}\sqrt{\frac{k}{m}}$
=> m = $\frac{k}{f^2\, 4\pi^2}$

So by putting the above values we get mass = 0.0376kg
=37.6 g
≃ 40 g

4. Comparison of experimental and theoretical results

Ultrasonic sensor was used to calculate the displacement

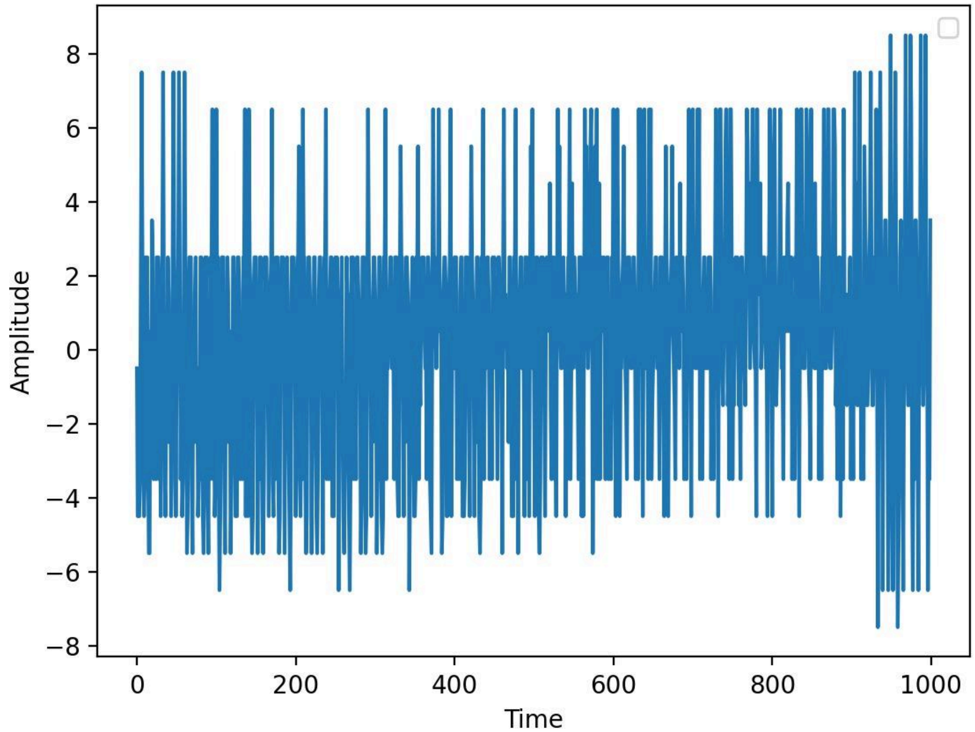


Fig.- without TMD

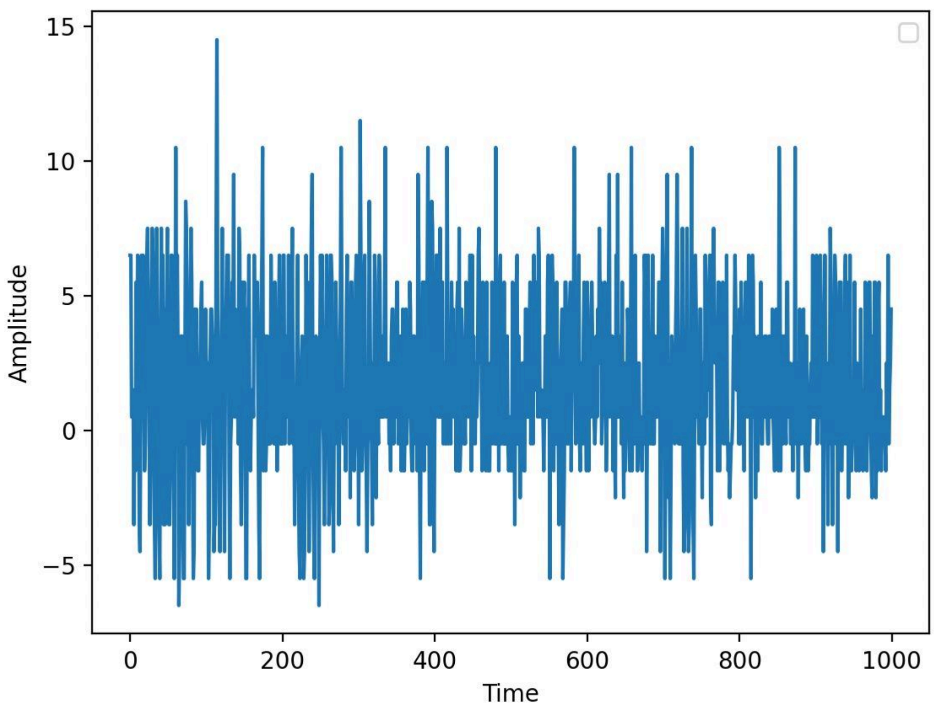


Fig. With TMD

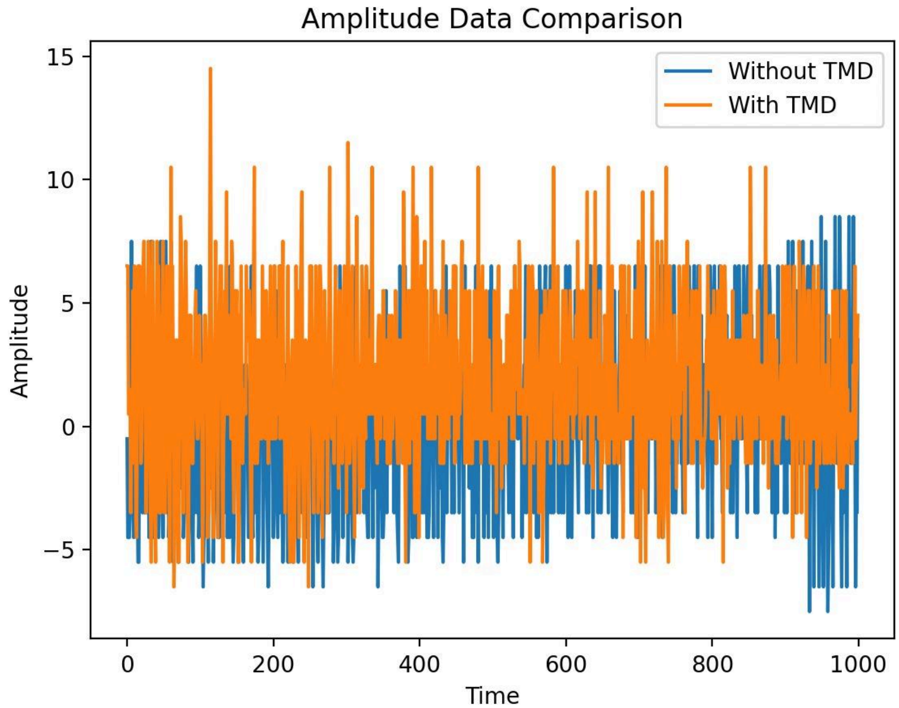


Fig. Comparison between Without TMD and With TMD

5. Results

1. The amplitude of vibrations with the damper was significantly lower than without the damper.
2. The results suggest that the effectiveness of the damper depends on its design parameters, such as the mass, stiffness, and damping ratio
3. Comparing the amplitude of the simply supported beam at resonance with and without TMD shows that the TMD has managed to dampen the amplitude.

6. Source of discrepancy/mismatch

1. The way we predicted the natural vibrations of the beam might be off because we didn't use specific values for instance density and stiffness.
2. We also didn't include the weight of the motor and the eccentric shaft attached at the end while figuring out how the beam would naturally vibrate.
3. Environmental factors such as temperature, humidity, and vibration from nearby equipment can also affect the accuracy of the experiment

7. Scope for improvement

1. Using a more powerful motor will allow for better-observing resonance.
2. By controlling environmental factors such as vibration from nearby equipment and reducing their impact on the experiment, it is possible to improve the accuracy of the results.
3. We can use the Fourier series for simply supported beam vibrations.

8. Acknowledgment

We would like to express our gratitude to Prof. Jayaprakash KR and TAs for their guidance and support throughout the completion of this project. We also would like to thank the Electrical lab staff for providing sensors, and required wires.

9. References:

1. https://onlinecourses.nptel.ac.in/noc23_ce21/preview
2. https://www.steelexpress.co.uk/steel-weight-calculator.html
3. https://youtu.be/JPJIg9soDYc?si=jkDG5t4b2uSNWTh0
4. https://rtlabs.nitk.ac.in/article/free-vibration-simply-supported-beam#:~:text=When%20given%20an%20excitation%20and,parameters%20of%20mass%20and%20stiffness.