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Design and Validation of a Mass Tuned Dynamic Vibration Absorber

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ABSTRACT

This paper presents the effectiveness of a new cantilever dynamic vibration absorber in reducing the vibrations of a primary system at resonance. The designed absorber is tested for cantilever and simply supported conditions of the beam with motor/rotor assembly to create harmonic excitations. The setup is tested numerically using modal and harmonic response analyses of ANSYS and further validated by building a prototype and conducting experiments using vibration analyzer. The proposed system was found to considerably reduce the vibrations in the primary system when the vibration absorber was tuned to the operating frequency.

Keywords: *Vibration, Cantilever Beam, Simply Supported Beam, Dynamic Vibration Absorber, Modal Analysis.*

1. INTRODUCTION

Vibration problem occurs where there are rotating or moving parts in machinery. The effects of vibration are excessive stresses, undesirable noise, looseness of parts and partial or complete failure of parts [1]. A machine or system may experience excessive vibration if it is acted upon by a force whose excitation frequency nearly coincides with a natural frequency of the machine. In such cases, the vibration of the machine can be reduced by using a vibration neutralizer or vibration absorber, which is simply another spring mass system. There have been many cases of systems not meeting performance targets because of resonance, fatigue and excessive vibration of a component. In general, each vibrating structure has a tendency to oscillate with larger amplitude at certain frequencies. These frequencies are known as resonance frequencies or natural frequencies [2].

At these resonance frequencies, even a small periodic driving force can result in large amplitude vibration. When resonance occurs, the structure will start to vibrate excessively. To suppress the vibration, the dynamic vibration absorber (DVA) is widely used as passive vibration control device. The concept of DVA is eliminating the vibration by a counter back motions. When force is applied to a structure, the DVA will react by producing same amount of force in the opposite direction henceforth restraining the beam motion at the desired frequency. When correctly tuned and attached to a vibrating body subjected to a harmonic excitation, eliminates steady-state motion of the point to which it is attached.

This paper deals with the modal and harmonic response analysis to check the effectiveness of the mass tuned dynamic vibration absorber in reducing the vibration at the operating frequency of the primary system. Further, a prototype model is built for the same and tested using vibration analyzer.

Modal Analysis: Modal analysis is used to determine the mode shapes and natural frequencies of a machine or a structure. It is the most basic form of dynamic analysis. The output of modal analysis can further be used to carry out a more detailed dynamic analysis like harmonic response analysis, transient analysis etc.

Harmonic Response Analysis: From the natural frequencies obtained by modal analysis, the harmonic analysis determines which vibration modes contribute more significantly to the dynamic response of the structure through frequency response curves [3].

2. THEORY AND FORMULATION

Theoretically, every vibration system can be modelled by an equivalent spring-mass vibration system. The excessive vibration can be reduced by means of a Dynamic Vibration Absorber (DVA). DVA is a device consisting of an auxiliary mass–spring system which tends to ‘absorb’ the vibration of a system to which it is attached [4]. A classical DVA consists only of a single pair of an auxiliary mass-spring system. This classical DVA is only useful for a single degree of freedom system [5], hence limiting its application prospects. Figure 1 illustrates the result using dynamic vibration absorber.

Figure 1 shows the response of a system with and without the use of an absorber. In general, without absorber the primary system has a high peak at a certain frequency which indicates the mode of the system. It can be seen from the figure that the absorber while eliminating vibration at the known impressed frequency of the primary structure, introduces two resonant frequencies Ω_1 and Ω_2. In practice, the operating frequency must therefore be kept away from the frequencies Ω_1 and Ω_2.

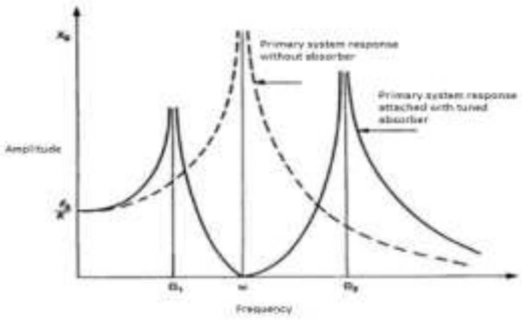


Fig. 1: Frequency Response of Primary System with and Without Absorber

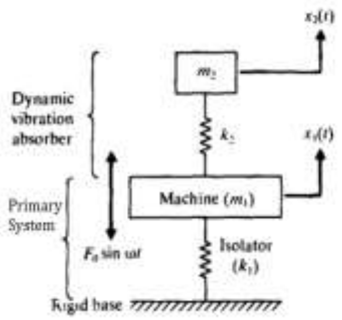


Fig. 2: Mathematical Model

Figure 2 illustrates a simplified equivalent model of a structure with mass m_1 (beam mass) and elasticity constant of k_1. A harmonic force, F_0 with forcing frequency of ω is exerted on the structure. If this forcing frequency is equal to the natural frequency of the beam, $\omega^{(n)}$ the structure will experience large vibration level where *n* is the number of mode of the beam structure. To suppress the vibration level of the structure a DVAs is attached to the structure. Hypothetically, the DVA with auxiliary mass, m_2 and elasticity constant, k_2 will absorb the fundamental mode (*n* = 1) of vibration of structure when tuned.

The equations of motion for mass m_1 and m_2 are,

$$m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2x_2 = F_0 \sin \omega t \tag{1}$$

$$m_2\ddot{x}_2 - k_2x_1 + k_2x_2 = 0 \tag{2}$$

Writing the equations of motion in matrix form to obtain,

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{Bmatrix} F_0 \sin \omega t \\ 0 \end{Bmatrix} \tag{3}$$

Assuming a harmonic solution and substituting in Eq. (3)

$$x = X \ \sin \omega t \tag{4}$$

X is the maximum displacement amplitude, *X* for each respective mass (X_1 for beam and X_2 for absorber mass), we get

$$\begin{bmatrix} (k_1 + k_2) - m_1\omega^2 & -k_2 \\ -k_2 & k_2 - m_2\omega^2 \end{bmatrix} \begin{Bmatrix} X_1 \sin \omega t \\ X_2 \sin \omega t \end{Bmatrix} = \begin{Bmatrix} F_0 \sin \omega t \\ 0 \end{Bmatrix} \tag{5}$$

Simplifying we can obtain,

$$X_1 = \frac{(k_2 - m_2\omega^2)F_0}{(k_1 + k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - k_2^2} \tag{6}$$

$$X_2 = \frac{k_2F_0}{(k_1 + k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - k_2^2} \tag{7}$$

We are primarily interested in reducing the amplitude of primary system to zero. In order to do so the numerator of Eq. (6) should be set equal to zero. This gives

$$\omega^2 = \frac{k_2}{m_2}$$

If the machine before the addition of DVA operates near its resonance, (i.e. $\omega^2 = \omega_1^2 = {k_1}/{m_1}$) large amplitude of vibration will be observed at its resonant frequency. Thus if the absorber is designed such that

$$\omega^2 = \frac{k_2}{m_2} = \frac{k_1}{m_1} \tag{8}$$

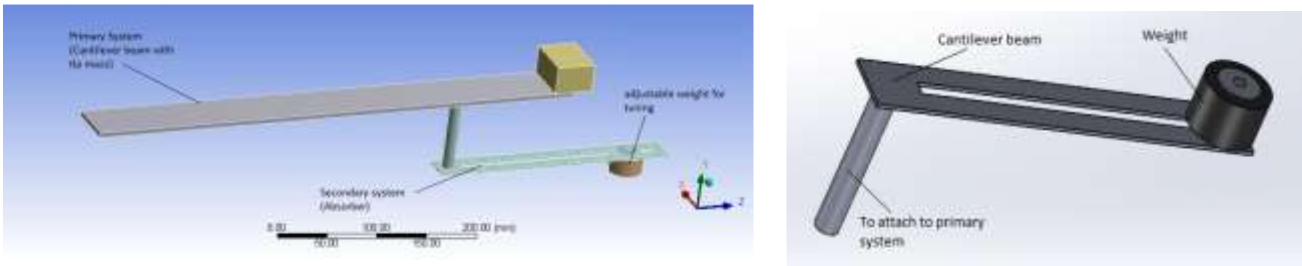
then the amplitude of vibration of the machine while operating at its resonant frequency, will be zero. Using Eq. (6), (7) and Eq. (8), the vibration level for any mode of primary system can be suppressed by tuning the DVA to appropriate mode accordingly.

The ratio of absorber mass m_2 to machine mass m_1 plays an important role in separating the two natual frequencies of the primary system with and without absorber as shown in Fig. 2. The ratio of absorber mass to the primary mass m_2/m_1 is usually taken between 0.05 and 0.25 to separate the two natural frequencies of the primary and the absorber system as shown in Fig.1 [6]. In our case this ratio was however taken as 0.29, due to experimental design changes.

3. MODAL AND HARMONIC RESPONSE ANALYSIS

A. Cantilever Beam with Unbalanced Motor/Rotor

A simple rectangular beam of dimensions 520 x 50 x 5 mm with unbalanced motor/rotor assembly was chosen



(a) **(b)**

Fig. 3: (a) Cantilever Beam Attached with Absorber, (b) Vibration Absorber

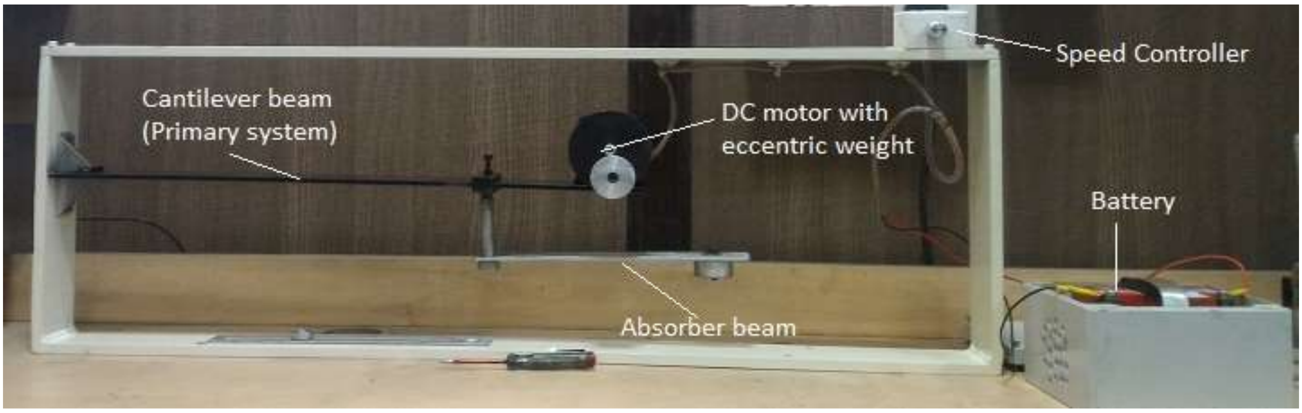


Fig. 4: Experimental Setup for Cantilever Beam with Vibration Absorber

as the primary system. The absorber was chosen to be a cantilever beam with dimensions 250 x 30 x 2 mm with a slot mounted with a movable mass (Fig. 3b). The proposed setup was tested for i) cantilever beam with motor/rotor assembly at the end attached with and without absorber, ii) simply supported condition with motor/rotor assembly at the center with and without absorber. For simplification of theoretical and numerical analysis, the motor/rotor assembly was modeled as a rectangular block with equivalent weight without significantly affecting the solution.

First, the natural frequencies of cantilever beam with motor/rotor at the free end were found using modal analysis of ANSYS. Then Harmonic response analysis was carried out for the same system to check the dynamic response of the system at the natural frequency. The results of modal analysis using ANSYS are tabulated in Table-1.

Further, the tuning of absorber was carried out for the fundamental natural frequency of 9.0235 Hz and again the model was evaluated using modal and harmonic response analysis of ANSYS. The frequency response of the system with and without the use of vibration absorber using ANSYS is shown in Fig. 5. From the figure we can see that there is a peak at 9.0235 Hz which is the fundamental natural frequency of the primary system, after attaching the tuned absorber to this frequency we now see that, at 9.0235 Hz the amplitude of vibration of the primary system is almost zero. Fig. 6 shows the comparison of the two systems with and without absorber.

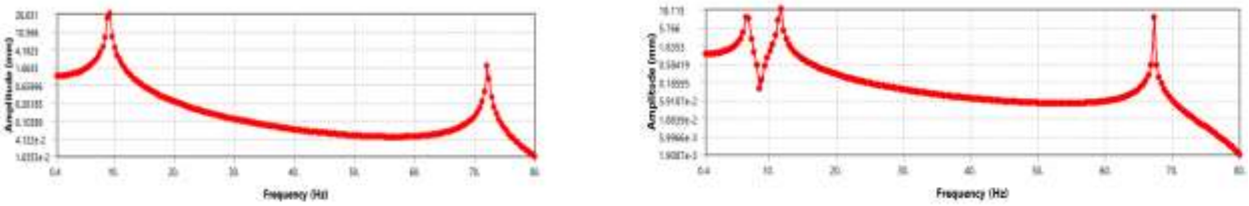
Table-1: Natural Frequencies of Cantilevered Beam with Motor/Rotor without Absorber using ANSYS

Mode	Frequency in (Hz)
1	**9.0235**
2	72.138
3	84.855
4	185.9
5	212.81
6	416.97

Tuning of Absorber

To tune the absorber for a particular operating frequency we need to adjust the stiffness of the absorber such that, the natural frequency of absorber coincides with the operating frequency eq. (9). This is achieved by the following steps.

i. We know that the natural frequency is given by



(a) **(b)**

Fig. 5: (a) Frequency Response of Cantilever Beam Model without Absorber, (b) Frequency Response of the Model Attached with Tuned Absorber

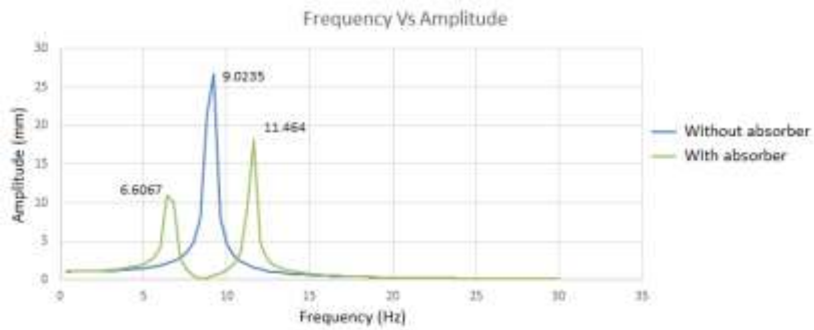


Fig. 6: Comparison of Frequency Response for Cantilever Beam with Motor/Rotor

$$f = \frac{\sqrt{k_a/m_a}}{2\pi}$$

Where k_a = stiffness of absorber beam. (N/mm)

m_a = mass attached to absorber beam. (kg)

m_a is chosen randomly to fit the requirement. Normally m_a is chosen to be 0.05-0.25 of the mass on the primary system.

ii. Since the absorber is a cantilever beam in our case, its stiffness is given by

$$k_a = \frac{3EI}{l^3}$$

Where, E = Young's modulus of the absorber material. (Mpa)
I = moment of inertia of cantilever beam cross section. (mm^4)
l = is the length of position of absorber mass from the fixed end. (mm)

iii. In the above formula E and I are known from absorber beam material and its cross section dimensions. Also the mass is chosen randomly within space and design constraints and is a known quantity. The only unknown quantity is *l* which is required to be found out. Once we find *l* we adjust the mass location to this length.

Performing the above calculations for cantilever beam model the length was evaluated to be 214.13 mm for the weight on the absorber from the fixed end of the absorber cantilever beam. From Fig.5 and Fig. 6 we can see that the proposed vibration absorber was successful in reducing the vibrations at the operating frequency (in this case resonance) of the primary system significantly.

B. Simply Supported Beam with Unbalanced Motor/Rotor

A similar analysis as performed for cantilever model was also carried out for simply supported model with unbalanced motor/rotor at the center as the primary system. The same absorber was used as the secondary system. The dimensions of the primary simply supported beam were 550 x 50 x 5 mm. A similar procedure was adopted, first finding the natural frequencies of the simply supported beam with motor/rotor without absorber (Table-2) and then attaching the tuned absorber to the primary system and then carrying out the modal and harmonic response analysis.

The length *l* for the position of the absorber weight for the frequency of 29.932 Hz was evaluated to be 96.27 mm from the fixed end. Fig 7 shows the experimental setup developed for the simply supported model. The results of harmonic analysis without and with the absorber are shown in Fig. 8a and 8b respectively.

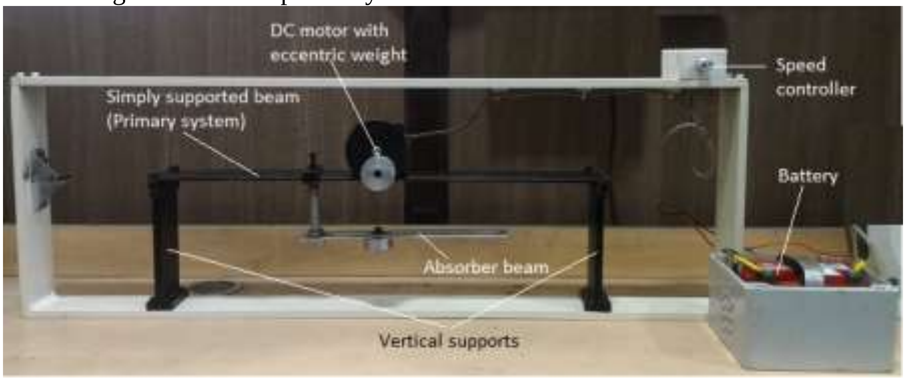
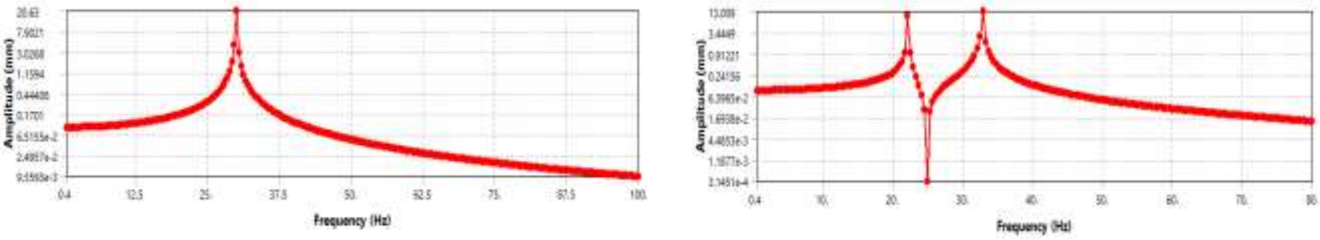


Fig. 7: Experimental Setup for Simply Supported Beam with Absorber



(a)

(b)

Fig. 8: (a) Frequency Response of the Simply Supported Model without Absorber, (b) Frequency Response of the simply Supported Model Attached with Tuned Absorber

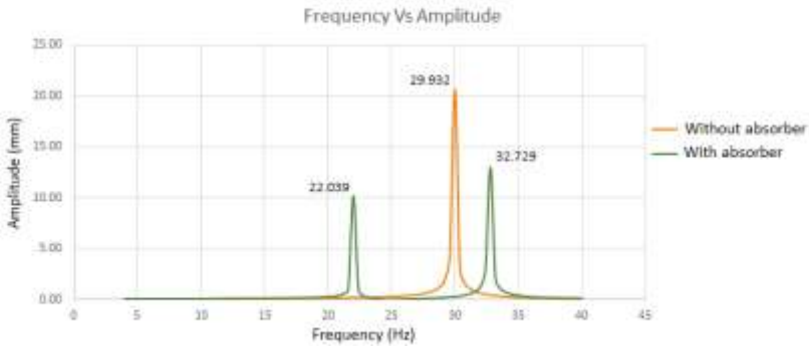


Fig. 9: Comparison of the Frequency Responses of the Simply Supported Model with and without Absorber

Table-2: Natural Frequencies of Simply Supported Beam with Motor/Rotor without Absorber using ANSYS

Mode	Frequency in (Hz)
1	**29.932**
2	147.98
3	273.04
4	310.84
5	444.5
6	538.02

From Fig. 8 and Fig. 9 we can see that the designed absorber model was successful in reducing the vibrations of the primary system at the operating frequency (in this case resonance frequency of the primary system without absorber). This is evident from Fig. 9, where we see a peak of 21 mm at 29.932 Hz for simply supported beam model without absorber, the same model when attached with tuned absorber for 29.932 Hz gives an amplitude of almost 0 mm for the operating frequency. However, it also creates two new peaks at 22.039 Hz and 32.729 Hz which are apart from the operating frequency. Thus, this designed absorber system is best suited when we want to operate a particular system at a desired operating frequency only.

4. EXPERIMENTAL OBSERVATIONS WITH VIBRATION ANALYZER

The above models for cantilever system and simply supported system with unbalanced rotor were tested experimentally with the help of OROS 34 vibration

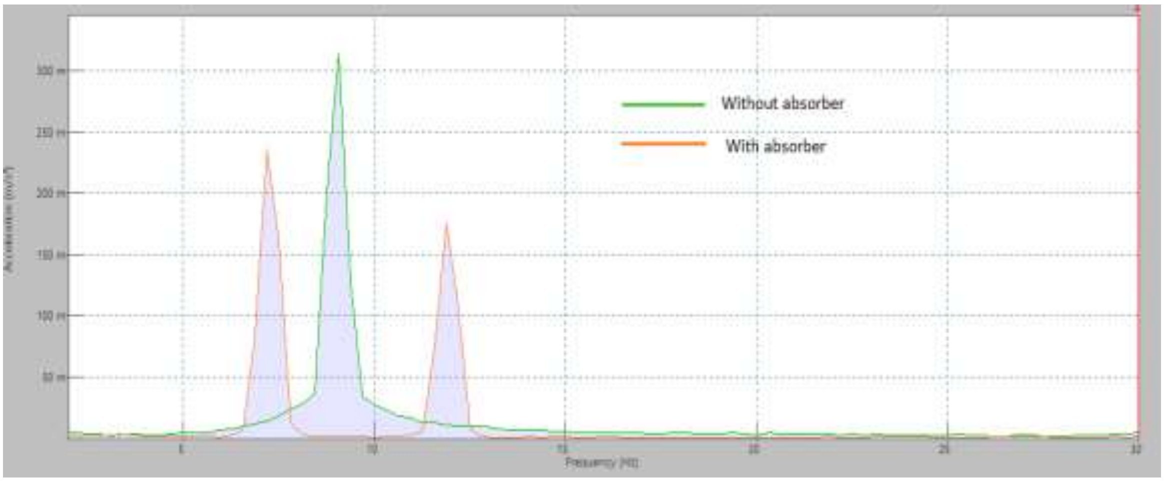


Fig. 10: Frequency Response of Cantilever Model with Unbalanced Rotor with and Without Absorber using OROS

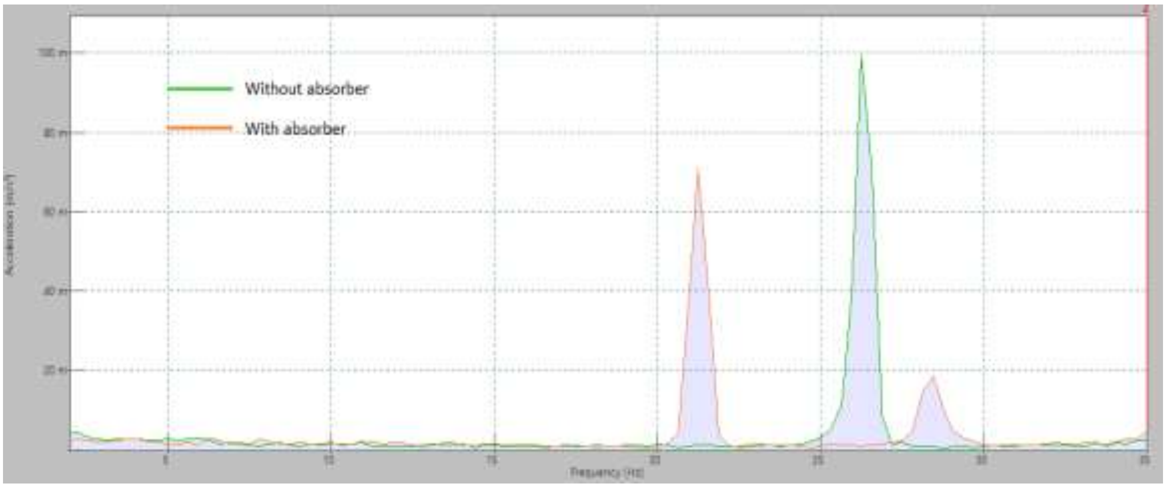


Fig. 11: Frequency Response of the Simply Supported Model with Unbalanced Rotor with and without Absorber using OROS

Analyzer. Fig. 4 and Fig. 7 shows the experimental setups built for cantilever model and simply supported model attached with tuned vibration absorber. With the chosen parameters initial experiments have been conducted by exciting the structure with an initial impact near the tip for the primary system first without absorber to obtain the system response and the fundamental frequency. Typically, the acceleration vs frequency graphs have been obtained using NV Gate software using OROS 34 vibration analyzer. Further tests with the attached tuned dynamic vibration absorber have been conducted by tapping the structure appropriately to stimulate the natural frequencies and then measuring the response. In this case, with the absorber unit attached to the primary system, the system is anticipated to exhibit two clearly distinct natural frequencies along with the higher frequencies associated with the flexible beam structures. Once again the system response to initial excitation was recorded using vibration analyzer. Now the system exhibits more than one distinct natural frequency at which the new resonances will occur. On the other hand, as the frequency spectrums indicate, it can be anticipated that significant reductions in vibration amplitudes can be obtained at the natural frequency of the primary system. These experimental results have been obtained with the stationary unbalanced rotor.

Figure 10 shows the acceleration vs frequency graph for cantilever system with and without the dynamic vibration absorber. Without the absorber the fundamental natural frequency of the primary system was observed at 9.0625 Hz indicated by the peak. When tuned absorber was attached to the primary system, we can see significant reduction in the amplitude at the fundamental frequency of the primary cantilever system. The two peaks, one at 7.1875 Hz and other at 11.875 Hz with absorber were recorded away from the fundamental frequency of the primary system. This was further verified visually by first running rotor at the fundamental natural frequency of the primary system and observing the vibrations at resonance. Later, tuned absorber was attached to the primary system with motor running at the fundamental frequency and it was found that, now the primary system had significant reductions in the vibration amplitude to the extent that it was almost stationary.
A similar procedure was adopted for primary system, now composed of simply supported beam with unbalanced rotor. From Fig. 11 we can see the significant reduction in the vibrations of the primary simply supported system at its natural frequency of 26.25 Hz. The two peaks, one at 21.25 Hz and the other at 28.4375 Hz were recorded away from the natural frequency of the primary system.

5. RESULTS AND DISCUSSION

The results obtained using ANSYS and experimental results obtained using OROS vibration analyzer are compared and discussed for both cantilever and simply supported condition of the beam with motor/rotor assembly.

Table-3 shows the natural frequencies of primary cantilever system with and without vibration absorber found using ANSYS and using experimental tally. The comparison shows that the values of natural frequencies found by using ANSYS and by using experimental procedure are very close. Also from Fig. 6 and Fig. 10 we can see that the nature of graph obtained for primary cantilever system using ANSYS and from experiment are significantly similar.

Table-3: Natural Frequencies of the Primary Cantilever System

	Numerically using ANSYS	Using experimental	Deviation Δ (Hz)
Without absorber (Hz)	9.0235	9.0625	0.039
With tuned vibration absorber (Hz)	6.6067	7.1875	0.5808
	11.464	11.875	0.411

Table-4 shows the deviation of the natural frequencies for primary simply supported system with unbalanced rotor found using ANSYS and from experiment. We see a significant deviation of the two results in this case, this may have been due to errors in the manufacturing of the setup for simply supported condition, material deformities of the beam material etc. However, from Fig. 9 and Fig. 11 we see some similarities in the nature of the two graphs.

Figure 12 shows the frequency response of the cantilever system at resonance frequency with and without the use of DVA. It can be clearly seen that after attaching the vibration absorber the primary cantilever system shows negligible displacement. Figure 13 shows the frequency response of the simply supported system at resonance frequency with and without the use of DVA.

6. CONCLUSION

In this study, we have designed and implemented a laboratory prototype of a cantilever and simply supported beam model with unbalanced rotor and a cantilever beam-type dynamic vibration absorber to control undesired vibrations under the influence of harmonic excitation source. The harmonic excitations in this study is caused by an unbalanced motor/rotor assembly. A finite element model was developed to investigate

Table-4: Natural Frequencies of the Primary Simply Supported System

	Numerically Using ANSYS	**Using experimental**	**Deviation Δ (Hz)**
Without absorber (Hz)	29.932	26.25	-3.682
With tuned vibration absorber (Hz)	22.039	21.25	-0.789
	32.729	28.4375	-4.29

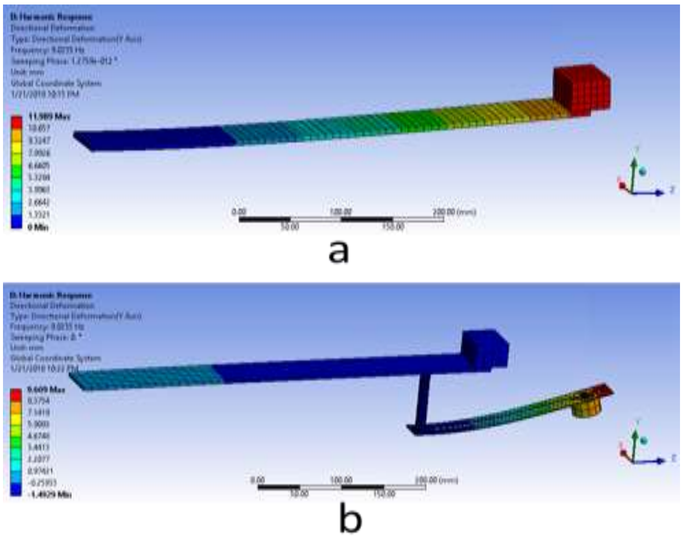


Fig. 12: (a) Harmonic Response of the Cantilever System at its Fundamental Natural Frequency, (b) Harmonic Response of the Cantilever System Attached with Tuned Absorber at the Fundamental Natural Frequency of the Primary System

The proposed prototype numerically. Experiments and numerical studies have revealed an adequate agreement and the proposed cantilevered dynamic vibration absorber system was capable of significant vibration reductions on a flexible cantilevered platform carrying an unbalanced motor/rotor at its tip and on a simply supported beam with unbalanced motor/rotor at its center.

The advantage of the designed absorber is that it can be practically tuned for a variety of operating frequencies by adjusting the position of the weight on the absorber beam. The proposed system is also compact and easy to attach to any vibrating structure. In the future work could be carried for auto-tuning of the absorber system using adequate actuators and feedback system.

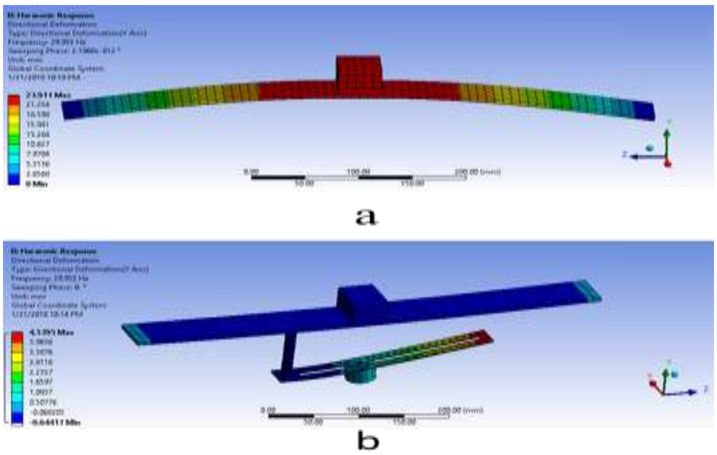


Fig. 13: (a) Harmonic Response of the Simply Supported System at its Fundamental Natural Frequency, (b) Harmonic Response of the Simply Supported System Attached with Tuned Absorber at the Fundamental Natural Frequency of the Primary System

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