DESIGN AND DEVELOPMENT OF TUNED VIBRATION ABSORBER FOR BEAM SUBJECTED TO HARMONIC EXCITATION WITH NONLINEAR PARAMETERS

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ABSTRACT

Vibration absorption is a method of adding a tuned spring-mass absorber to a system to create anti-resonance at a resonance of the original system. The dynamic vibration absorber is designed in such a way that the natural frequencies of the resulting system are away from excitation frequency. In this paper theoretical and numerical analysis of dynamic vibration absorber is carried out. Experimental setup for dynamic vibration absorber is also developed.

Keywords: Dynamic, Nonlinear, Resonance, Tuned, Vibration Absorber

I. INTRODUCTION

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 dynamic vibration absorber, which is simply another spring mass system [4].

II. THEORETICAL BACKGROUND

Vibration absorber with Two DoF system is as shown in Fig.2a

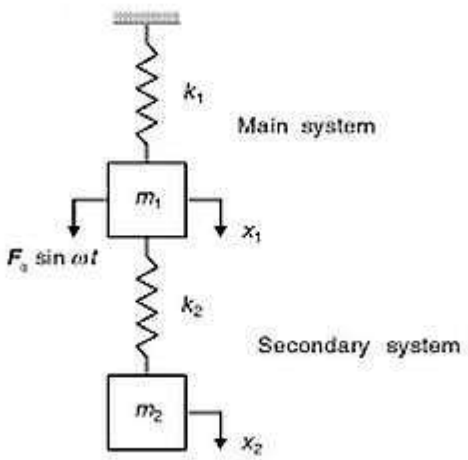

Fig.2a Two DoF Undamped Dynamic Vibration Absorber

For the system shown in Fig.2a let us assume $x_2 > x_1$,

Spring mass system k_1-m_1 as main system and spring mass system k_2-m_2 as absorber system.

Equations of motion for Two DoF system are as follows,

$$m_1\ddot{x}_1 + F_o \sin \omega t = -k_1x_1 + k_2(x_2 - x_1) \qquad (1)$$

$$m_2\ddot{x}_2 = -k_2(x_2 - x_1) \qquad (2)$$

Assuming solution under steady state condition,

$x_1 = X_1 \sin\omega t$

$x_2 = X_2 \sin\omega t$

Finally we get equation in dimensionless form as

$$\frac{x_1}{x_{st}} = \frac{1 - \frac{\omega}{\omega_2}^2}{\frac{\omega^4}{\omega_1^2\omega_2^2} - \left[(1+\mu)\frac{\omega}{\omega_1}^2 + \frac{\omega}{\omega_2^2}^2\right] + 1} \qquad (3)$$

$$\frac{x_2}{x_{st}} = \frac{1}{\frac{\omega^4}{\omega_1^2\omega_2^2} - \left[(1+\mu)\frac{\omega}{\omega_1}^2 + \frac{\omega}{\omega_2^2}^2\right] + 1} \qquad (4)$$

From this equations it is clearly seen that $x_1 = 0$, when $\omega = \omega_2$ i.e. When excitation frequency is equal to the natural frequency of the absorber system, the main system amplitude becomes zero even though it is excited by harmonic force. This the principle of an undamped dynamic vibration absorber [5].

Also it can be seen that when $\omega = \omega_2$, we get,

$$F_o = -k_2x_2 \qquad (5)$$

The above equation shows that the spring force k_2 x_2 on main mass due to amplitude x_2 of absorber mass is equal and opposite to the exciting force on main mass resulting in no motion of main system.

2.1 Tuned Absorber

For the effectiveness of absorber at operating frequency corresponding to natural frequency of main system alone we have,

$$\omega_1 = \omega_2 = \frac{k_2}{m_2} = \frac{k_1}{m_1}$$

When this condition is fulfilled absorber is called as Tuned Absorber. To have a tuned absorber we can have many combinations of k_2, m_2 as long as their ratio is equal to $\frac{k_1}{m_1}$ to satisfy the above condition.

We can have a small spring k_2 and small mass m_2 or k_2 large and large mass m_2. In all these cases main system response will be zero at $\omega = \omega_2$.

However, Eq. shows that for same exciting force the amplitude of absorber mass is inversely proportional to its spring rate. In order to have small amplitude of absorber mass m_2, we must have a large k_2 and therefore large m_2 which may not be desirable from practical considerations. So a compromise is usually made between amplitude and mass ratio μ. The mass ratio is usually kept between 0.05 to 0.2. A proper design of absorber spring is also necessary which depends upon its amplitude and available space.

The denominators of above equation and are identical. At a value of ω when these denominators are zero, the two masses have infinite amplitudes of vibration. The expression for the denominators is quadratic in ω^2 and therefore there are two values of ω, for which these expression vanish. These two frequencies are resonant frequencies or natural frequencies of the system [3].

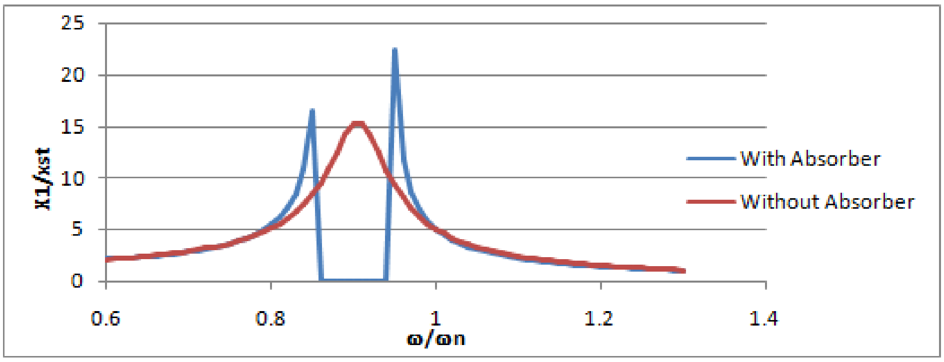

Fig.2b Magnification Factor Vs Frequency Ratio for Theoretical Analysis

When excitation frequency equals to any of the natural frequency of the system, all the points in the system have infinite amplitudes of vibration or the system is in resonance.

To find the two resonant frequencies of the system, when $\omega_2=\omega_1$, the denominator of either of equation is equated to zero.

$$\frac{\omega^4}{\omega_1^2\,\omega_2^2} - \left[(1+\mu)\frac{\omega^2}{\omega_1^2} + \frac{\omega^2}{\omega_2^2}\right] + 1 = 0 \qquad (6)$$

Solving we get,

$$\left(\frac{\omega}{\omega_2}\right) = \left[\left(1+\frac{\mu}{2}\right) \pm \sqrt{\mu + \frac{\mu^2}{4}}\right] \qquad (7)$$

This relationship is plotted in Fig. 2c

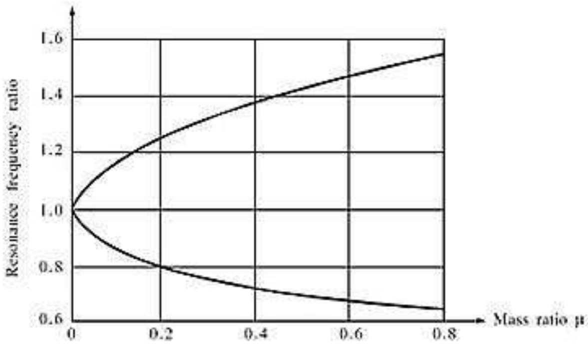

Fig.2c Resonant Speed Vs Mass Ratio

2.2 Experimental Parameters

Table1 Experimental Parameters of Cantilever Beam

Parameter	Symbol	Value
Material of Beam	MS	-
Total Length	L	0.9 m
Width	B	0.025 m
Thickness	T	0.005 m
Moment of Inertia	I	2.6042 x 10^-10 m4
Young's Modulus	E	2 x 10 ^11 N/ m²
Mass Density	P	7830 m³

III. NUMERICAL ANALYSIS

In order to carry out numerical analysis of vibration absorber ANSYS software is used. This analysis allows determination of resonance frequency of each mode, which is a function of location of mass along absorber plate. The type of element is used for analysis is SOLID186. It is a higher order 3-D 20-node solid element that exhibits quadratic displacement behavior. The element is defined by 20 nodes having three degrees of freedom per node: translations in the nodal x, y and z directions. The element supports plasticity, hyperelasticity, creep, stress stiffening, large deflection and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastoplastic materials and fully incompressible hyperelastic materials [1].

Using ANSYS analysis of linear and nonlinear cantilever beam is carried out as below,

3.1 Linear Analysis

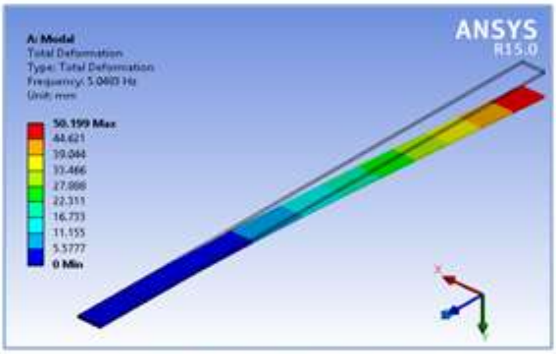

Fig.3a First Mode Shape of Linear Cantilever Beam

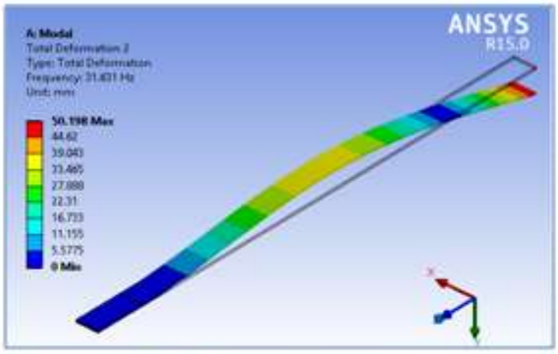

Fig.3b Second Mode Shape of Linear Cantilever Beam

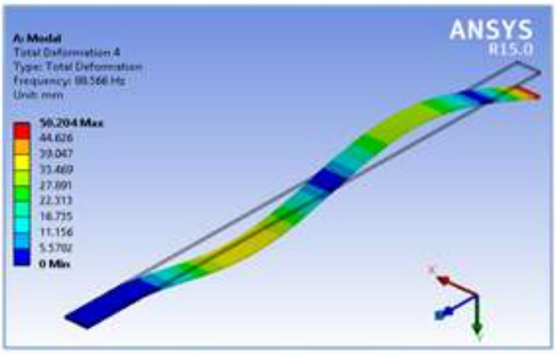

Fig.3c Third Mode Shape of Linear Cantilever Beam

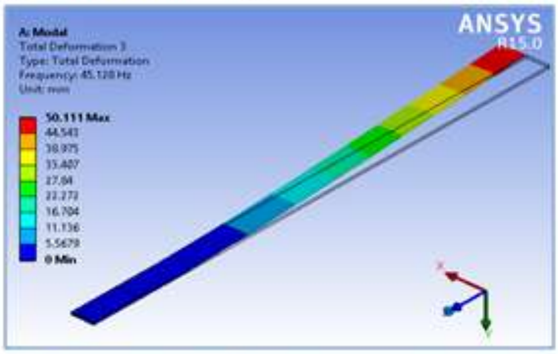

Fig.3d Fourth Mode Shape of Linear Cantilever Beam

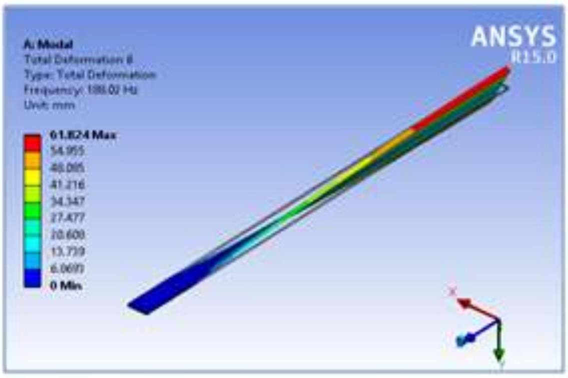

Fig.3e Fifth Mode Shape of Linear Cantilever Beam

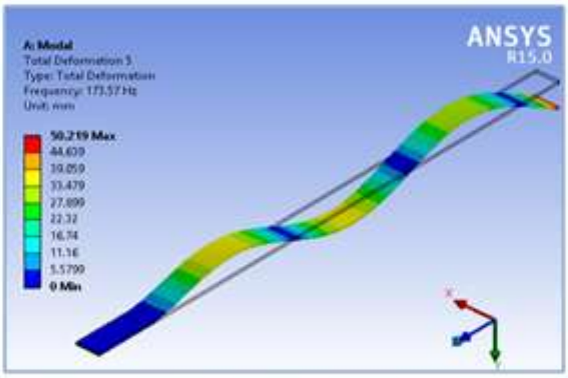

Fig.3f. Sixth Mode Shape of Linear Cantilever Beam

3.2 Nonlinear Finite Element Analysis

Non-Linear Analysis includes Material Nonlinearity: Force (stress) Vs. Displacement (strain) curve is nonlinear (polynomial). Geometric non-linearity: In real life, the stiffness [K] is a function of displacement [d]. This means in a geometric non-linear analysis, the stiffness K is re-calculated after a certain predefined displacement. Contact nonlinearity: In Contact analysis, the Stiffness K also changes as a function of displacement (when parts get into contact or separate) nonlinear analysis deals with true stress and strain (unlike engineering stress and strain in linear static analysis) [3].

3.3 Nonlinear Analysis

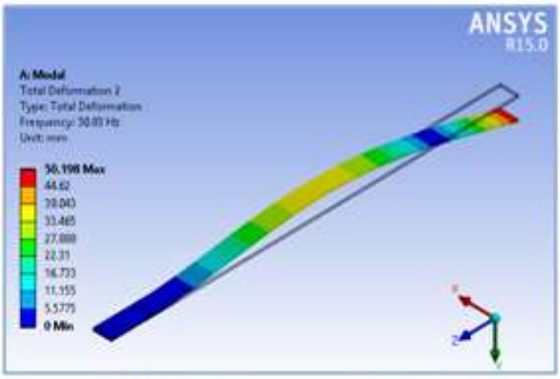

Fig.3g First Mode Shape of Nonlinear Cantilever Beam

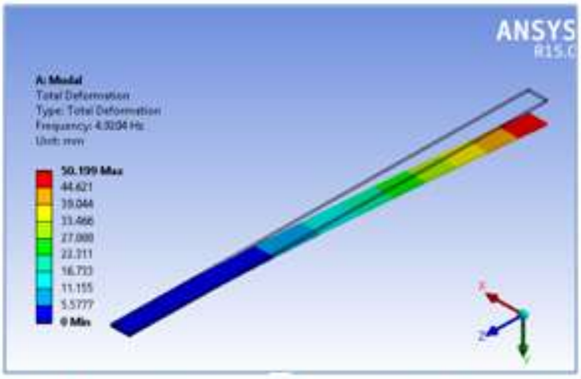

Fig.3h Second Mode Shape of Nonlinear Cantilever Beam

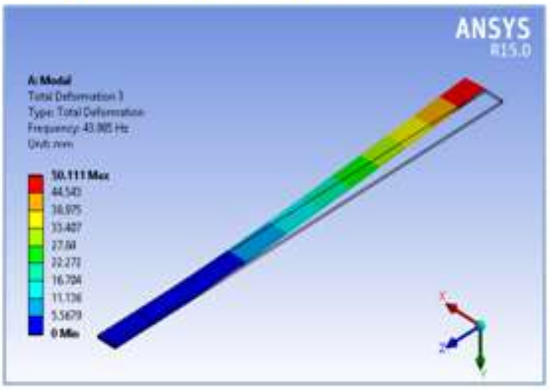

Fig.3i Third Mode Shape of Nonlinear Cantilever Beam

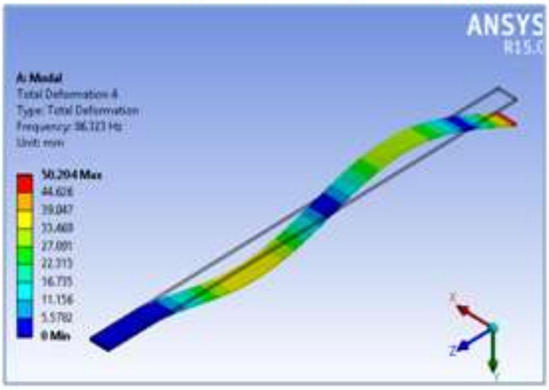

Fig.3j Fourth Mode Shape of Nonlinear Cantilever Beam

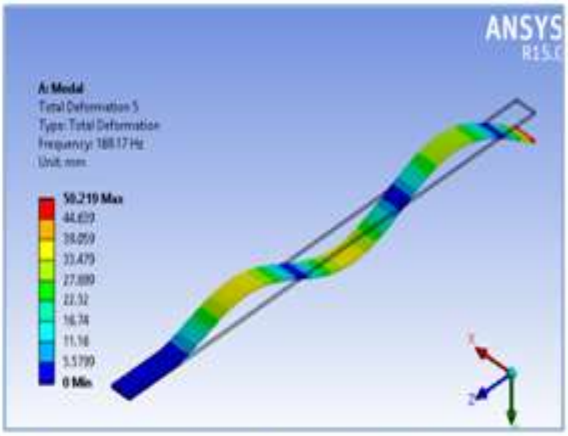

Fig.3k Fifth Mode Shape of Nonlinear Cantilever Beam

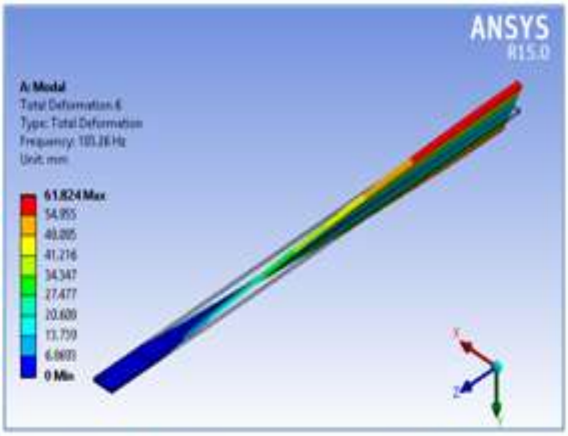

Fig.3l Sixth Mode Shape of Nonlinear Cantilever Beam

IV. DESIGN OF DYNAMIC VIBRATION ABSORBER

The variable stiffness and variable damping mechanism consists of mass and cantilever beam as absorber plate, lead screw, D.C. motor, variable support, guide plate, circular magnets and fixed frame to combine the mechanism as one. In the absorber, the spring constant at mass varies by moving the movable support along the cantilever beam. The movable support consists of plate with a rectangular slot in which a rectangular plate is inserted. Hence, Middle support moves when motor rotates. Hence, absorber resonances can be changed by the movement of support along the length of cantilevered beam.

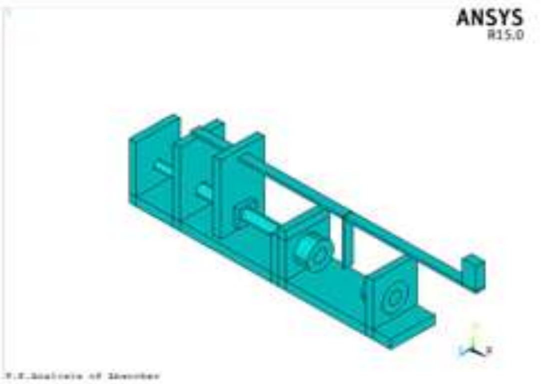

Fig.4a

Fig.4a CAD Model of Vibration Absorber

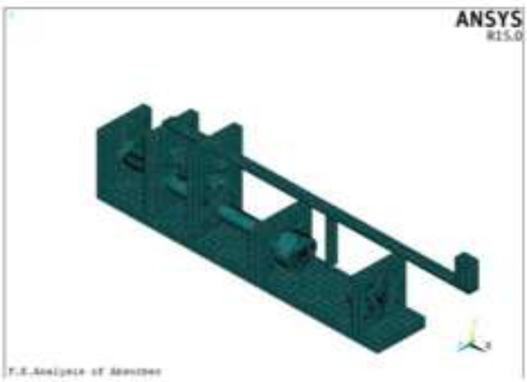

Fig.4b

Fig.4b Meshed Model of Vibration Absorber

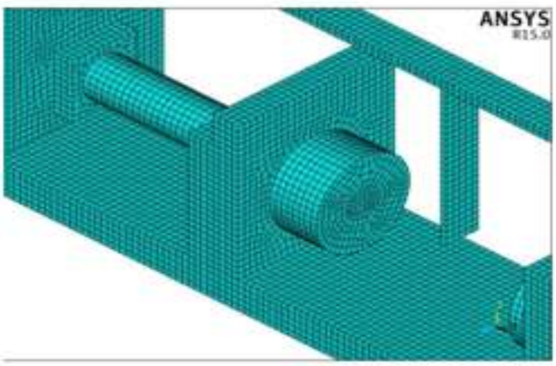

Fig.4c

Fig.4c Meshed Model of Vibration Absorber

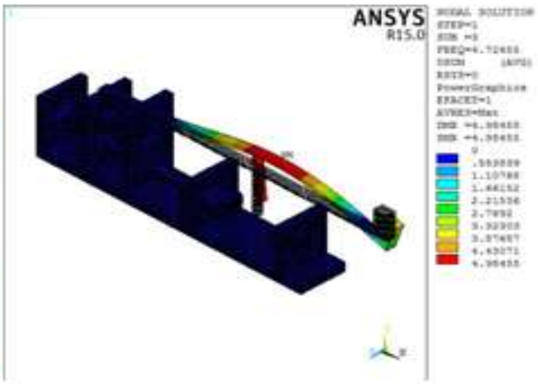

Fig.4d

Fig.4d Third Mode Shape Vibration Absorber

Frequency of absorber plate for nonlinear parameters,

When L (length of absorber plate) is 325 mm then,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{251.35}{0.258}}$$

$$f_n = 4.97\ Hz$$

V. RESULTS AND DISCUSSION

Table 2 First Six Natural Frequencies for a Linear and Nonlinear Cantilever Beam

	LINEAR BEAM	NONLINEAR BEAM	ABSORBER PLATE
Mode	Frequency (Hz)	Frequency (Hz)	Frequency (Hz)
1	5.2483	4.9604	4.98
2	31.631	30.83	30.88
3	45.128	43.985	43.99
4	88.566	86.323	86.35
5	173.57	169.17	169.21

From above table it is seen that natural frequency of nonlinear cantilever beam and absorber plate matches. Therefore it can be concluded that by using nonlinear analysis it is possible to design actual vibration absorber.

VI. CONCLUSION

As natural frequency of absorber system matches with natural frequency of nonlinear cantilever beam. Hence vibrations of beam will be absorbed for nonlinear cantilever beam. From the analysis it is seen that there is difference between frequencies of linear and nonlinear cantilever beam. Results of analyzed cantilever beam by considering nonlinear parameters tends to actual results. Therefore while designing the tuned vibration absorber for cantilever beam it is necessary to consider the nonlinear parameters to achieve real life results.

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