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DESIGN OF A COMBINED DYNAMIC VIBRATION ABSORBER FOR VIBRATION SUPPRESSION OF BEAM UNDER HARMONIC EXCITATIONS

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Abstract
A combined dynamic vibration absorber combining a translational and rotational type absorber is proposed. Finite element analysis and Euler-Bernoulli beam theory is used for evaluation of the performance of vibration isolation using combined dynamic vibration absorber mounted on beam. A computer program is written in Matlab and numerical experiments are carried out to predict the response of beam with absorber. The proposed translational and rotational absorbers are designed using ANSYS software. Both numerical and experimental tests have been done for verification of the theoretical prediction of vibration isolation in beam.

Keywords:
Vibration absorber, Rotational absorber, FEA, Experimental Testing, Simulation.

I. Introduction

The traditional dynamic vibration absorber which is a translational-type of absorber when correctly tuned and attached to a vibrating system subjected to harmonic excitations, causes to cease the steady state motion at the point of attachment. However, when applying dynamic vibration absorber to a continuous structure such as beam, vibration can be eliminated only at the attachment point of the beam while amplification of vibration may occur in other parts of the beam. In certain application, it may be of interest to suppress vibration for the particular span of elastic structure where sensitive instruments or components are mounted or attached. Research on suppressing vibration in a region or the particular span of an elastic structure by using the spring mass vibration absorber has been reported recently. Cha (1-3) proposed the scheme to impose node and to suppress vibration in a region or whole span of a beam structure by using the dynamic vibration absorber. This method requires the use of many translational-type spring-mass absorbers for creating a region of nearly zero amplitude in the beam structure. Recently Wong [4] developed a new dynamic vibration absorber combining a translational- type absorber and a rotational type absorber for isolation of beam vibration. The use of finite element analysis and experimental test for evaluation of the performance of vibration isolation of the combined absorber mounted on a beam is carried out. Patil[5] developed novel algorithm to find absorber parameters required to create node at desired location on beam using dynamic vibration absorber. Further Patil [6] has performed numerical simulations and experimental tests to find the required absorber resonance frequency and mass to impose a node at appropriate locations on a beam. Bassam [7] designed a control force to create nodal point(s) having zero displacement and/or zero slope at selected locations in a harmonically excited vibrating structure

The combined absorber is more effective than only translational type absorber alone for vibration isolation in a region of the beam. However the prototype absorber which includes spring-mass translational absorber is useful only for laboratory test. The purpose of present study is to develop combined dynamic absorber which can be useful for actual applications

II.Theory

The finite element model of a beam with a combined vibration absorber, which is a translational and rotational absorber, attached at node *i* is shown in Figure 1

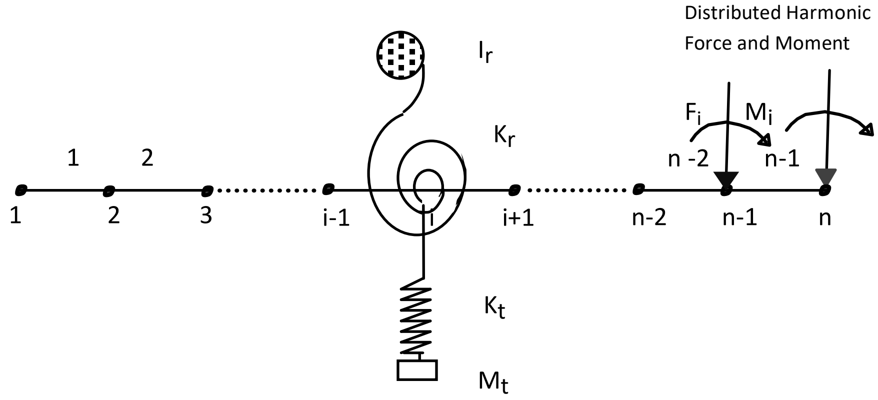

Figure 1: Finite element model of beam with translational and rotational absorber.

The finite element equation of the beam under harmonic excitation may be written as

$$M\frac{\partial^2 u(x,t)}{\partial t^2}+Ku(x,t)=F(x)\sin\omega t, \tag{1}$$

where M and K are the mass and stiffness matrices respectively, ω is the excitation frequency, u is the nodal displacement and F is a vector of the nodal amplitudes of excitations written as

$$F=[F_1,M_1,F_2,M_2,\cdots,F_n,M_n]^T, \tag{2}$$

where F_j and M_j for $j=1,2,3,\ldots,n$ are the force and moment applied at j respectively.

For the steady state response of the nodal displacement of the beam in the equilibrium state,

$$u=U\sin\omega t, \tag{3}$$

where U is the vector of nodal amplitudes of vibration written as

$$U=[U_1,\theta_1,U_2,\theta_2,\cdots,U_n,\theta_n]^T, \tag{4}$$

where U_j and θ_j for $j=1,2,3,\ldots,n$ are the nodal amplitudes of translation and rotation at node j, respectively.

Equation (1) may be written as

$$(K-\omega^2 M)U=F. \tag{5}$$

Based on Euler-Bernoulli beam theory with Hermite shape function, the stiffness matrix of the *jth* element of the beam is

Column: $2j-1$ $2j$ $2j+1$ $2j+2$

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$$k_{ej} = \frac{E_e I_e}{I_e^3}\begin{bmatrix} 0 & & & & & & & & \\ & \ddots & & & & & & & \\ & & 0 & & & & & & \\ & & & 12 & 6l_e & -12 & 6l_e & & & \\ & & & 6l_e & 4l_e^2 & -6l_e & 2l_e^2 & & & \\ & & & -12 & -6l_e & 12 & -6l_e & & & \\ & & & 6l_e & 2l_e^2 & -6l_e & 4l_e^2 & & & \\ & & & & & & & 0 & & \\ & & & & & & & & \ddots & \\ & & & & & & & & & 0 \end{bmatrix} \tag{6}$$

According to Figure 1, translational absorber in the form of spring-mass with stiffness k_t and a rotational absorber in the form of pendulum with stiffness k_r are attached at node i of the beam. The stiffness matrix of combined absorber K_a to be added to the global stiffness matrix of the beam is

Column: $\quad 2i-1 \;\; 2i \qquad\qquad\qquad 2n-1 \;\; 2n$

$$K_a = \begin{bmatrix} 0 & & & & & & & & 0 & 0 \\ \vdots & \ddots & & & & & & & \vdots & \vdots \\ & & & 0 & 0 & \ldots & 0 & 0 & 0 & 0 \\ & & & k_t & 0 & \cdots & 0 & 0 & -k_t & 0 \\ & & & 0 & k_r & \cdots & 0 & 0 & 0 & -k_r \\ & & & & & \ddots & & & \vdots & \vdots \\ & & & & & & & \ddots & & \\ 0 & \ldots & 0 & -k_t & 0 & 0 & \cdots & 0 & k_t & 0 \\ 0 & \cdots & 0 & 0 & -k_r & 0 & \cdots & 0 & 0 & k_r \end{bmatrix} \tag{7}$$

The global stiffness matrix of the beam with the absorbers may be written as

$$K = \sum_{i=1}^{n} k_{ej} + K_a. \tag{8}$$

The mass matrix of the jth finite element of beam is

Column: $\quad 2j-1 \;\; 2j \;\; 2j+1 \;\; 2j+2$

$$m_{ej}=\frac{\rho_e A_e l_e}{420}\begin{bmatrix} 0 & & & & & & & & \\ & \ddots & & & & & & & \\ & & 0 & & & & & & \\ & & & 156 & 22l_e & 54 & -13l_e & & \\ & & & 22l_e & 4l_e^2 & 13l_e & -3l_e^2 & & \\ & & & 54 & 13l_e & 156 & -22l_e & & \\ & & & -13l_e & -3l_e^2 & -22l_e & 4l_e^2 & & \\ & & & & & & & 0 & \\ & & & & & & & & \ddots \\ & & & & & & & & & 0 \end{bmatrix} \tag{9}$$

The mass matrix of the absorber M_a, with m_t is the lumped mass of translational absorber and I_r is the moment of inertia of the rotational absorber, to be added to the global mass matrix of the beam is

Column: $2n-1$ $2n$

$$M_a=\begin{bmatrix} 0 & \cdots & & \cdots & 0 & 0 \\ \vdots & \ddots & & & \vdots & \vdots \\ & & & & & \\ \vdots & & & 0 & 0 & 0 \\ 0 & \cdots & & 0 & m_a & 0 \\ 0 & \cdots & & 0 & 0 & I_r \end{bmatrix} \tag{10}$$

The global mass matrix M of the beam with the absorber is given as

$$M=\sum_{i}^{n} m_{ej}+M_a. \tag{11}$$

The matrix in equation (5) may be written as

$$\begin{bmatrix} \Delta_{11} & \Delta_{12} & & & & & 0 & 0 \\ \Delta_{21} & \Delta_{22} & & & & & & \\ & & \ddots & & & & & \\ & & & \Delta_{2i-1,2i-1}+k_t & & & -k_t & \\ & & & & \Delta_{2i,2i}+k_r & & & -k_r \\ & & & & & \ddots & & \\ & & & -k_t & & & k_t-\omega^2 m_a & 0 \\ & & & & -k_r & & 0 & k_r-\omega^2 I_r \end{bmatrix} \tag{12}$$

where $\Delta_{pq}=K_{pq}-\omega^2 M_{pq};\ p,q=1,2,3,\ldots n$

When the absorber are tuned such that $k_t/m_t = k_r/I_r = \omega^2$

$$\Delta = \begin{bmatrix} \Delta_{11} & \Delta_{12} & & & & 0 & 0 \\ \Delta_{21} & \Delta_{22} & & & & & \\ & & \ddots & & & & \\ & & & \Delta_{2i-1,2i-1}+k_t & & -k_t & \\ & & & & \Delta_{2i,2i}+k_r & & -k_r \\ & & & & & \ddots & \\ & & & -k_t & & 0 & 0 \\ & & & & -k_r & 0 & 0 \end{bmatrix} \quad (13)$$

Referring to Figure1 and consider a harmonic distributed excitation with frequency ω, written as $F^{'}(x)\sin\omega t$, where $F^{'}$ is a vector of the nodal amplitudes of excitations written as

$$F = [F_1, M_1, F_2, M_2, \cdots, F_i, M_i, 0\cdots 0]^T, \quad (14)$$

The vector of nodal amplitudes of the beam may be written as

$$U = (K - \omega^2 M)^{-1} F^{'} \quad (15)$$

$$U = \Delta^{-1} F^{'} \quad (16)$$

The equation (16) can be used to find the response of beam with absorbers subjected to point or distributed loading.

III.Numerical Experiment

A computer program was written in Matlab to perform numerical experiment on beam to predict response of beam for vibration isolation during harmonic excitations. The M.S beam of dimension 1000 mm $\times$ 65 mm $\times$ 10 mm was modelled with hundred standard Euler Bernoulli beam element. Young's modulus and density of the beam are 210 GN/m^2 and 7500 Kg/m^3. The stiffness matrices, mass matrices, displacement matrix and the force matrix are prepared using equation (4) - (14). Two different cases of boundary conditions were considered for the beam. In the first case, the beam was assumed as fixed-free with absorber attached at 0.4L and point load with frequency 28 Hz applied at free end of the beam as depict in Figure 2 . Second case is for the simply supported boundary condition with absorber at 0.3L and uniformly distributed force for the span of 0.2L to 0.26L of frequency 35 Hz is applied in beam as illustrated in Figure 3. The response of beam with translational absorber, combined absorber and without absorber is shown in Figure 4 and Figure 5 for both cases.

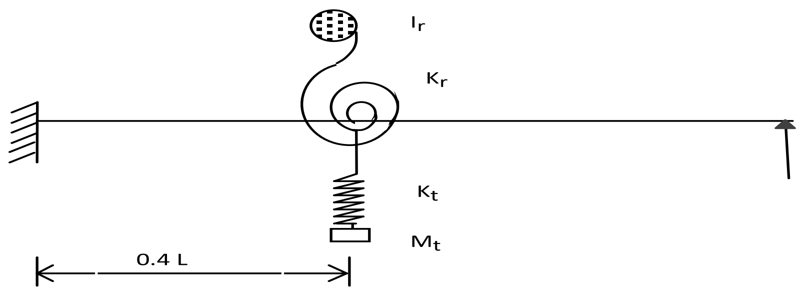

Figure 2: Schematic of cantilever beam with combined absorber under point harmonic excitation

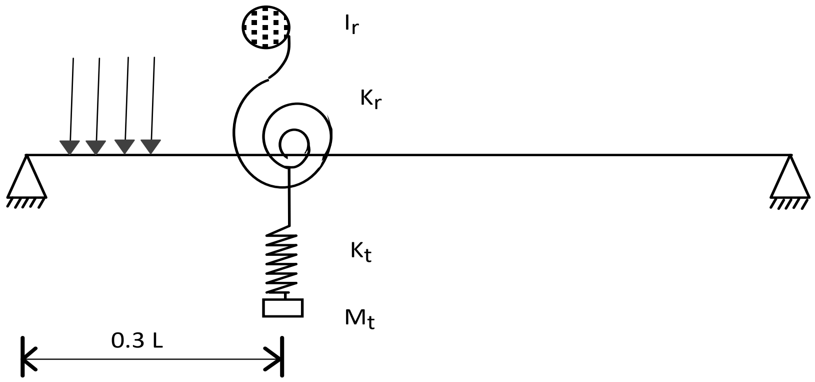

Figure 3: Schematic of simply supported beam with combined absorber under distributed harmonic excitation

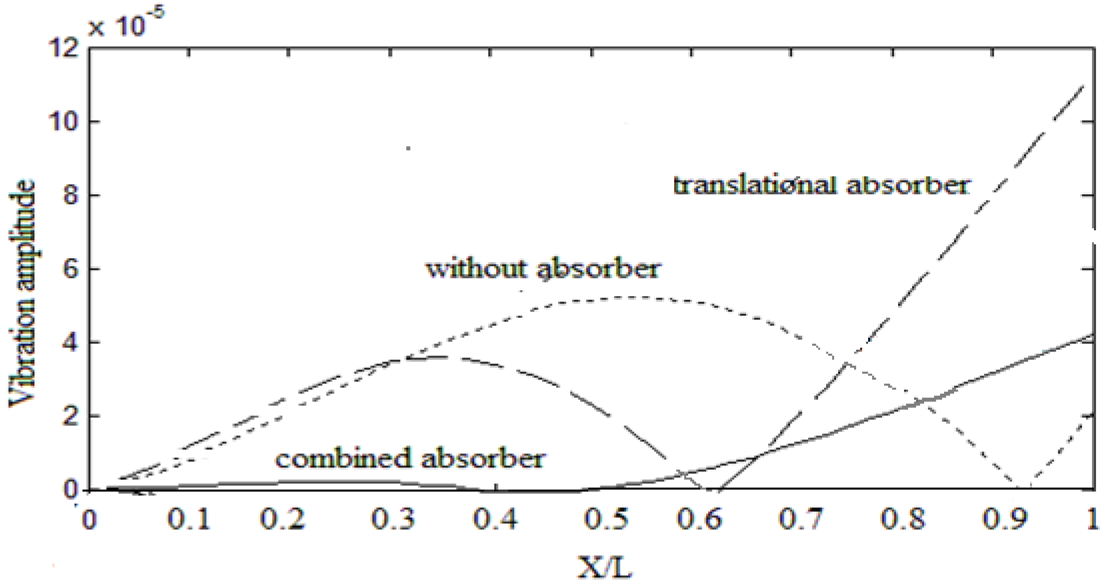

Figure 4: Response of cantilever beam with combined, translational and without absorber under point harmonic excitation

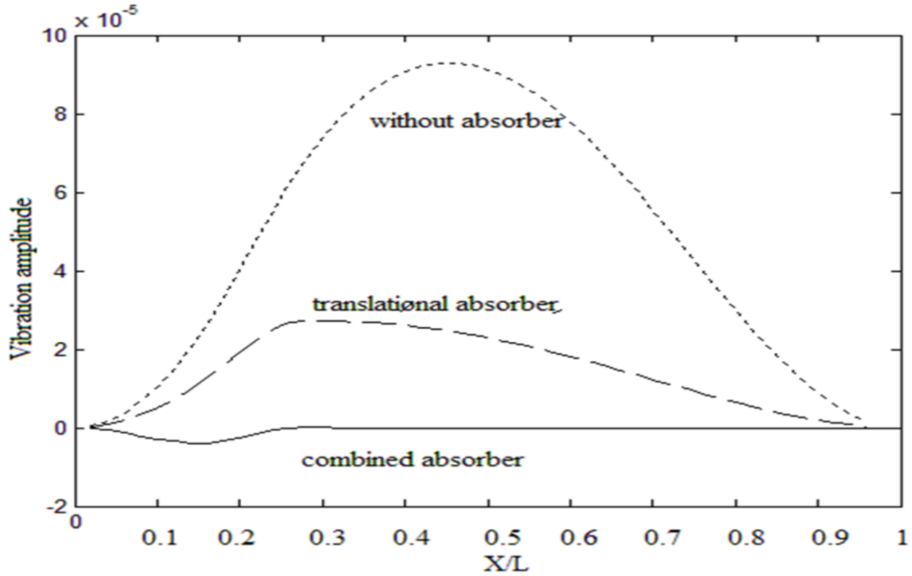

Figure 5: Response of simply supported beam with combined, translational and without absorber under point harmonic excitation

It is observed from Figure 4 and Figure 5 that translational absorber reduces vibration at perticuler location whereas combined absorber is effective in vibration rediction for a span of beam.

IV.Design of combined absorber

The aim of the work here is to develop a translational and rotational absorber that can be easily tuned to the excitation frequency. The dual mass absorber consists of central rod supporting two equal masses on either side of a center section, which is attached to a beam as illustrated in Figure 6. The absorber can be tuned to the required frequency by rotating masses so as to move in or out along the rod. The first five frequencies of the translational absorber for different position of the masses are listed in Table 1 using finite element analysis ANSYS software.

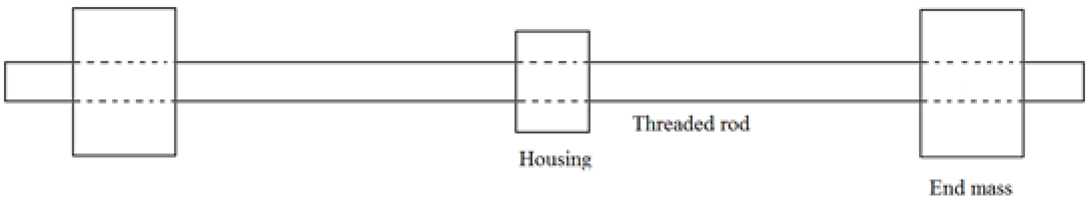

Figure 6: Dual mass variable stiffness traslational vibration absorber

Table 1: Frequency of translational absorber for various mass positions.

Mass Position (mm)	Frequency (Hz)				
	1	2	3	4	5
270	30.9	147.1	247.4	416.1	744.3
260	31.8	151.3	259.4	412.0	786.8
250	32.7	155.5	270.3	418.7	801.0
240	33.6	159.7	279.5	426.2	814.0
230	34.5	164.1	286.4	433.7	814.9

The rotational absorber was made by attaching two rectangular plates to the on to a piece of steel strip The rectangular plates are clamped on to the steel strip by bolts and nuts so that their mounting position on the steel strip can be changed easily as shown in Figure 7. The fundamental frequency of the rotational absorber can be tuned easily by adjusting the location of plates on the steel strip. The first

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five frequencies of the rotational absorber for different position of the plates are listed in Table 2 using finite element analysis ANSYS software,. The dimensions and material properties of the translational and rotational absorber are given in Table 3.

Table 2: The frequency of rotational absorber for various plate positions.

Strip Position (mm)	Frequency (Hz)				
	1	2	3	4	5
140	28.9	29	29.0	29.0	85.7
130	31.8	31.9	31.9	31.9	88.8
120	35.2	35.3	35.4	35.4	92.4
100	44.1	44.1	44.3	44.3	101.1
90	49.9	49.9	50.1	50.1	106.4

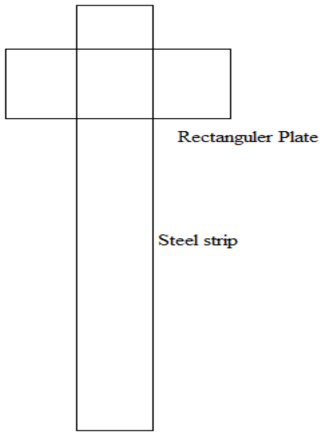

Figure 7: Rotational strip and plate type vibration absorbers

Table 3: Dimensions of absorbers

Dimensions of translational absorber	
Length of rod	170mm
Diameter of rod	8mm
Length of mass	25mm
Diameter of mass	80mm
Dimensions of rotational absorber	
Length of beam plate	300mm
Width of plate	24mm
Thickness of plate	5mm

V. Experimental Test

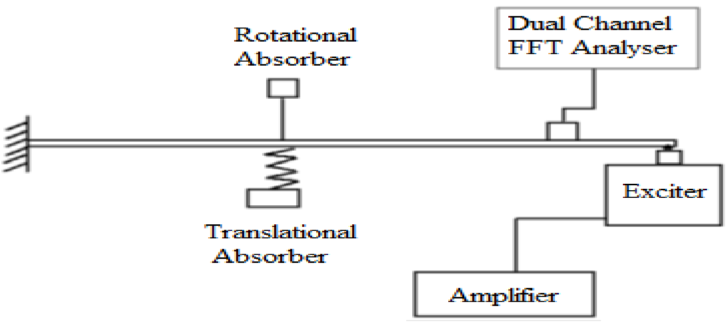

Figure 8: Illustration of experimental setup for testing of absorber

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Experimental test is carried for validation of the theoretical prediction of vibration reduction of beam using translational type and combined absorber. The experimental setup is illustrated in Figure 8. The dimensions and the physical properties of uniform beam used for experimentation are as given in section -4.

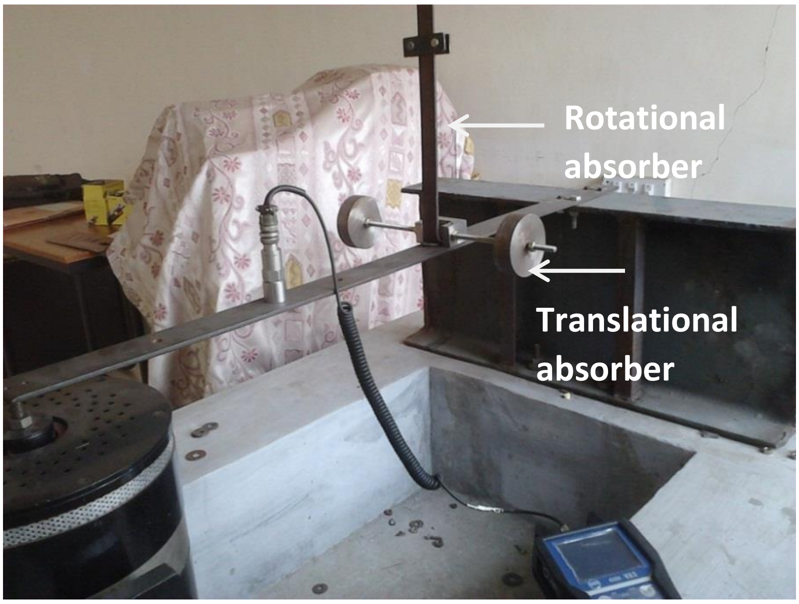

Figure 9: Experimental Testing on cantilever beam

To conduct the experiment an electromagnetic shaker (Syscon SI 230) is used to excite the beam at the end of beam with a single harmonic excitation of 28 Hz . Vibration amplitudes are measured at twenty points on the beam surface by the accelerometer (AC 102 CTC) and recorded by dual channel FFT vibration analyzer (Adash 4300) as shown in Figure 9. The measurement procedure of the forced vibrations amplitudes are repeated for the case of beam with only the translational absorber, combined absorber and also the case of beam without any absorber. Vibration amplitudes of the beam of all three cases is plotted in Figure 10.

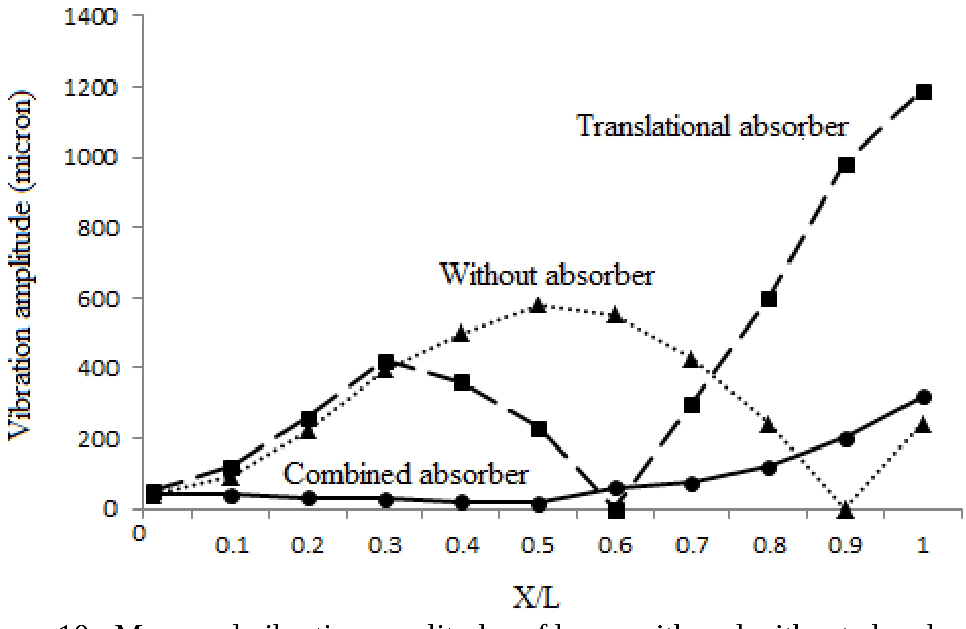

Figure 10: Measured vibration amplitudes of beam with and without absorber

It is observed from Figure 10 that translational absorber reduces vibration at particular location on beam. However combined absorber is effective for vibration minimization for span of beam.

Comparison if Figure 3 and Figure 10 shows that there is good agreement between the numerical results and the experimental results.

VI.Conclusions

The combined translational and rotational absorber has been designed and tested. Both numerical and experimental tests have been done for verification of theoretical prediction of vibration isolation of dynamic vibration absorber. The numerical and experimental results show that beam vibration under harmonic excitations can be isolated in a region of beam using combined absorber.

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