DEVELOPMENT OF VIBRATION ABSORBER FOR BEAM STRUCTURE USING VARIABLE DAMPING AND STIFFNESS

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ABSTRACT-

The vibration absorbers are frequently used to control and to minimize excess vibration in structural systems. To reduce the vibration of the main system or machine, the frequency of absorber should be equal to the excitation frequency. This result in subcomponent of total structure adding large input impedance to the primary structure, thus ‘absorbing’ the internal energy transferred to form primary structure. The aim of this project is to develop the variable stiffness and variable damping type vibration absorber to control the vibrations of primary system which is a cantilever beam structure using finite element method through ANSYS programming.Matlab software is used to find theoretical response of beam structure with vibration absorber. A variable stiffness vibration absorber with variable damping is developed and its stiffness can be varied by moving the support plate along the length of absorber plate and damping can be varied by adjusting distancebetween two permanent magnets.

The proposed absorber was manufactured and tested on cantilever beam and it was found that absorber is very effective to control the vibration of lower as well as higher modes. The proposed absorber is suitable for vibration isolation of beam structure with uniform cross-section and facilitates vibration attenuation at variable excitation frequencies.

***Keywords* ---**Cantilever beam as primary system, variable stiffness and variable damping, Lower mode and higher mode frequency,Vibration analysis, FFT analyzer

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Introduction

Due to devastating effect of vibrations, it is necessary to reduce the vibrational effect over mechanical components. The spring-mass absorber has difficulty with effectiveness when excitation frequency is variable in nature. Tunable dynamic vibration absorber can be effectively used to minimize the vibrations of beam structure over the changing excitation frequenciesVibrations of machines and structures vanish perfectly at a certain frequency if they have a vibration absorber without damping. But if forced frequencies vary from the anti-resonance frequency, their vibration amplitudes increase significantly. Then the absorber without damping cannot be applied to the structure subjected to variable frequency loads or the loads having high-frequency components. In the present work it is proposed to study variable stiffness vibration absorber by varying its stiffness by moving the support plate along the length of absorber beam to adapt the changes in excitation frequency to suppress the vibration of cantilevered beam structure as a primary system at multiple frequencies. For higher modes variable damping absorber will be used for greater effectiveness.The variable damping absorber will be used for greater effectiveness for higher modes.

Literature review

In reference [5], Simon S. Hill and Scott D. Snyder have described the design of vibration absorber (dual mass vibration absorber) using FEA in Ansys software to reduce structural vibrations at multiple frequencies with enlarged bandwidth for the practical vibration attenuation of at multiple resonant frequencies. In reference [6], W.O.Wong and et al have developed a dynamic vibration absorber by combining a translational-type and rotational-type absorber for vibration isolation of beam under point or distributed harmonic excitation. Finite element analysis and Euler–Bernoulli beam theory was used for evaluation of the performance of vibration isolation of the proposed absorber mounted on a beam. In reference [4], K.Nagaya, A.Kurusu and et al illustrated a variable stiffness vibration absorber is used for controlling a principal mode. The stiffness is controlled by the microcomputer under the auto-tuning algorithm for creating an anti-resonance state. In reference [3], H. Moradi et al. has been designed the tunable vibration absorber to suppress chatter vibrations in boring operation in which boring bar is modeled as a cantilever Euler-Bernoulli beam instead of it is considering as single degree of freedom system. In reference [2] H.D.Supekaret al has been designed combined variable stiffness and variable damping (VSVD) type vibration absorber for vibration control of primary beam system

In this paper, the design of an adaptive vibration absorber is reported. The aim is to theoretically develop a device which could potentially find use in the control of transformer noise radiation, where multiple independent absorbers could simply be attached to the transformer without the need for an all-encompassing control system. Vibration absorbers have been used on a wide variety of structures to reduce vibration in an attempt to reduce the radiated noise. The utilization of vibration absorbers as a noise control technique has been limited for many reasons, including; the cost of commercial devices, the longset-up time associated with tuning, the ability to vary the resonance frequency of the device in response to dynamic changes in the structure and the inability to provide attenuation at multiple frequencies. The latter is constrained by two factors, the excitation of the absorber higher order modes and their coincidence with a structural resonance frequency.

An aim of the work presented here is to develop a practical absorber that facilitates vibration attenuation at multiple frequencies. A secondary aim is to investigate the possibility of using multiple, closely spaced resonances to expand the effective bandwidth of the absorber. What follows is a description of the design and implementation of a tunable, multiple resonance vibration absorber. The absorber uses variable stiffness for tuning.

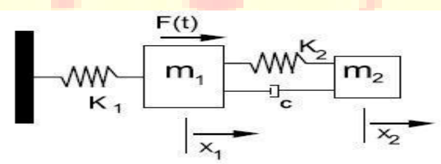


Fig.1: Primary system and absorber schematic [2]

Referring to Figure 1, consider a primary system with mass $m_{1,}$ and stiffness k_1 and hence Resonance frequency

$$\omega p = \sqrt{\frac{k_1}{m_1}} \text{-----} \qquad (1)$$

If a secondary device with mass m_2, stiffness k_2 and viscous damping c is added to the system, then the differential equations describing the above system are

$$m_1\ddot{x}_1 + c(\dot{x}_1 - \dot{x}_2) + (k_1 + k_2)x_1 - k_2x_2 = F(t) \quad \text{--(2)}$$
$$m_2\ddot{x}_2 + c(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \quad \text{---(3)}$$

It can then be shown that the response of m_1 vanishes, if the resonance frequency of the secondary system corresponds to that of the primary system. This is a well-known result of applying a vibration absorber. The addition of a properly tuned absorber will cause the system previously characterized by a single resonance to have two resonances, as shown in Fig. 2.

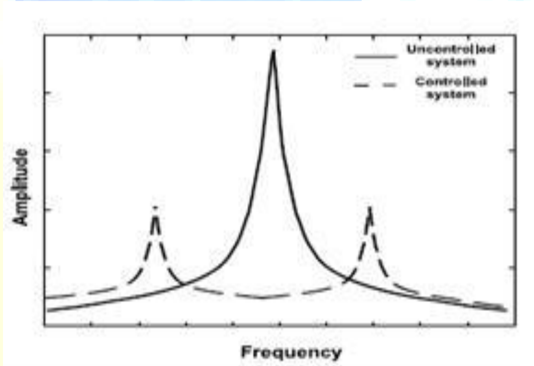


Fig.2: Frequency response with well-tuned absorber [2]

The two frequencies appear on either side of the single resonance. While the response at the previous resonance has dramatically dropped, the response at the two new resonance frequencies is much larger than before. This variation is controlled by the absorber damping. It is known that the controlled by the ratio, ν of the absorber mass m_2 and target structure mass m_1, The effective mass, ν= m_2/m_1 , plays an important part in determining if the absorber is effective. The effective mass is balance between the magnitude of the force applied to a structure and the ability of the structure to excite the absorber. The problem of attenuating structural resonances cannot be simply solved by the addition of a secondary system with corresponding resonance. Considering the system as infinitely rigid in all but the direction normal to the resonant surface, then the mechanical impedance, of such a system is given by

$$Z = M_{jaw}[(1 + ja/Q)/(1 - a^2 + ja/Q)] \quad \text{---- (4)}$$

Where $\omega = \sqrt{\frac{k2}{m_2}}$

Is the resonance frequency of the absorber (Secondary system).

Design of vibration absorber using FEA in ANSYS

The design of vibration absorber is carried out using 'FEA'. The absorber is modeled using ANSYS 5.4 software using SOLID 92 element. The advantages of this absorber are that it can be easily tuned to the excitation frequency, so it can be used to reduce the vibration of the system subjected to variable excitation frequency.

Modal analysis of absorber

In order to gain an accurate prediction of modes of the absorber, a numerical analysis using finite elements is used. This analysis allows determination of resonance frequency of each mode, which is a function of location of mass along two rods. Absorber is designed by using FEA with software package ANSYS. First six mode shapes of absorber are found by ANSYS.

The solid model produced in ANSYS is shown in Fig. 3.

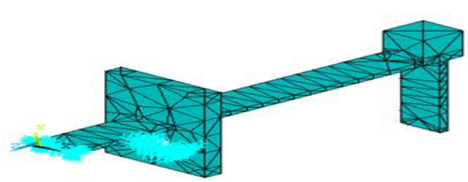


Fig. 3 FEA Model of absorber

The model produced in ANSYS is similar to which will be manufactured for testing. The support is modeled by using 'BLOCK' command. Plate and mass are modeled by using 'CYLINDER' command. The plate is glued to central section creating a single element. The same procedure is used for attaching masses to the plate. Complete program used for ANSYS is given in Appendix-B. The ANSYS program is made generalized by using parameters.

Assumptions

In order to gain an accurate prediction of the modes of the absorber, a numerical analysis using finite elements is used. This analysis allows determination of the resonance frequency of each mode, which is a function of the location of the mass along the plate, for which the square hole in the center of moving plate (which attaches the absorber plate) is used. The suitability of this element is based on its bending and membrane properties. Also for the modeling of the plate and absorber masses, SOLID 92 shown in Fig. 4.9 is used. SOLID 92 is well suited to model irregular meshes. The element is defined by 10 nodes having three degree of freedom at each node translations in the nodal in x, y, and z direction.

To get accurate values for the resonance frequency the maximum mesh density is used. To further increase the accuracy of the results a coarser mesh density is used for the masses as the deformation of these elements has little effect on absorber resonance frequency. Very fine mesh is applied on the plate and section. These elements of the absorber are most important in determining how the absorber behaves at each resonance frequency. So fine mesh will give the better results. A meshed model is shown in Fig. 3

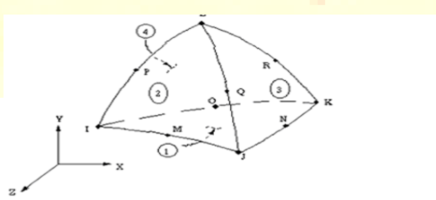


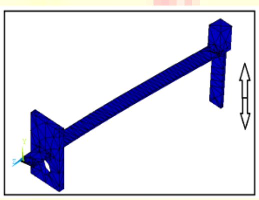
Fig.4Solid 92 3-D-Node tetrahedral structural solid

An understanding of absorber's mode shapes is required so that significant parameters that effect each mode can be determined allowing for possibility for adjusting different modal frequencies. Following mode shapes were found using model developed in ANSYS. The first six natural frequencies obtained are tabulated and corresponding mode shapes of the absorber are evaluated using finite element analysis by writing an ANSYS program as shown in Fig. 5 to Fig. 10

In first mode, mass is moving laterally in vertical direction and in second mode, mass moves in

Distance (mm)	1st natural Frequency (Hz)	2nd natural Frequency (Hz)	3rd natural Frequency (Hz)	4th natural Frequency (Hz)	5th natural Frequency (Hz)	6th natural Frequency (Hz)
0	9.58	74.95	112.82	105.73	271.82	442.76
10	9.96	81.12	113.06	112.43	284.97	502.43
20	10.32	87.57	114.49	119.65	301.29	583.57
30	11.03	93.62	116.37	126.77	318.61	662.76
40	11.75	98.91	118.93	133.62	335.84	706.55
50	12.78	104.59	121.80	140.76	355.12	456.14
60	13.11	110.63	125.01	148.36	376.23	807.77
70	13.63	116.92	128.84	156.58	400.91	865.44
80	14.29	123.00	133.97	165.06	428.02	924.55
90	15.38	128.48	140.91	174.27	460.29	989.18

horizontal direction. In third mode rod is moving laterally in torsional direction. In fourth mode of absorber, rod is moving up and down perpendicular to direction. In fifth mode, rod is moving similar to forth mode. In sixth mode, rod is moving up and down perpendicular to direction having wavy nature.



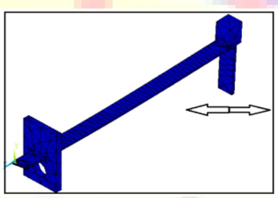
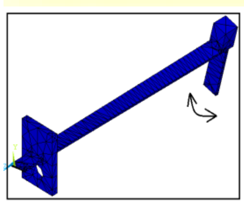


Fig 5 First mode shape **Fig 6 Second mode shape**



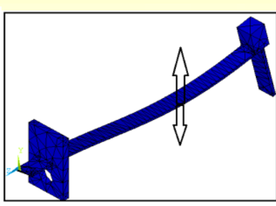


Fig 7 Third mode shape **Fig 8 Fourth mode shape**

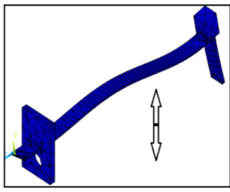
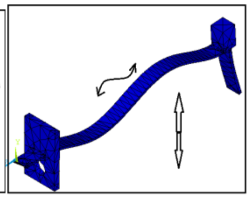
 

Fig 9 Fifth mode shape **Fig 10 Sixth mode shape**

Effect of absorber parameters on natural frequency

As absorber is cantilevered mass absorber, its length affects first and second mode. Stiffness of cantilever can be varied by changing effective length of absorber mass. Effective length can be varied by moving support along length of cantilever and thus frequency can be changed. The results from ANSYS are then listed.

Table I: Effect of absorber length on natural frequency

Modal analysis of beam

In order to predict frequencies and mode shapes of beam modal analysis is carried out using ANSYS software. The beam is modeled using SOLID 45 element as shown in Fig. 11.

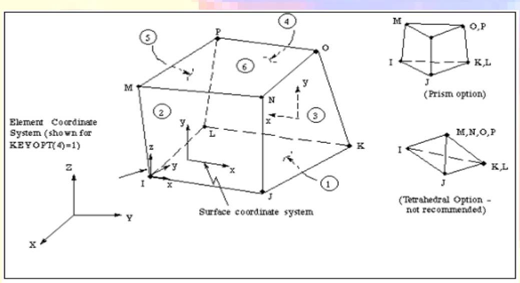


Fig. 11 Solid 45 –structural solid

SOLID 45 is used for three dimensional modeling of solid structure. The element is defined by eight nodes: translations in the node x, y and z direction. The nodes at the one end of beam are fixed i.e. all degree of freedom of the nodes are zero.

Dimension and properties of the beam are-

Length of beam = 1000mm,
Thickness of beam = 10 mm,
Width of the beam = 65 mm,
Density of material =7800 kg/m^3,
Modulus of elasticity = 210 Gpa.
Material = M.S.

The FEA mesh model of Fixed-free beam acting as primary system used in experimentation is shown in Fig.12.

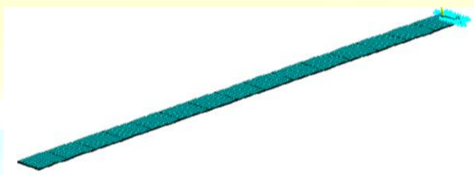


Fig.12 FEA Mesh model of cantilever beam

The first three frequencies of beam are listed in Table II.

Table II:First three natural frequencies of cantilever beam

Sr.No.	1	2	3
Frequency (Hz)	8.42	52.80	147.96

The respective corresponding mode shapes are as shown in Fig. 12to Fig.14

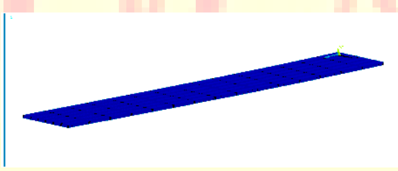


Fig. 12 First mode shape of beam

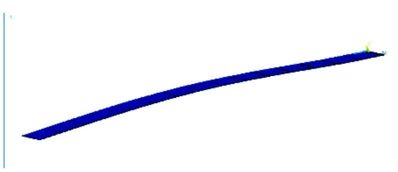


Fig.13 Second mode shape of beam

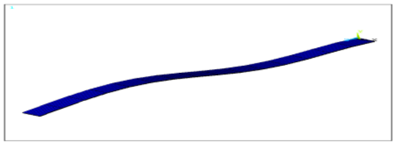


Fig.14 Third mode shape of beam

Experimental setup

Experimental setup to find response of beam without absorber

Experimental setup for response of primary system (cantilever beam) without absorber is shown in Fig. 15. Rectangular cantilever beam was fixed to I-Beam support which is fixed with foundation with the help of nut-bolts, which acts as fixed rigid support. Accelerometer was mounted on top surface of beam at a distance of 0.8L (800 mm from fixed end). Accelerometer was connected to FFT analyzer. Exciter was placed below cantilever between fixed end and accelerometer i.e. about 0.45L (450 mm) from fixed end of primary beam. When supply is given to exciter, its tip is moving up and down. Position of exciter was adjusted such that tip was in contact with beam. Beam was excited with sinusoidal force. Frequency of excitation can be gradually increased or decreased by using control over FFT. The set of readings are taken without absorber attached to the primary system.

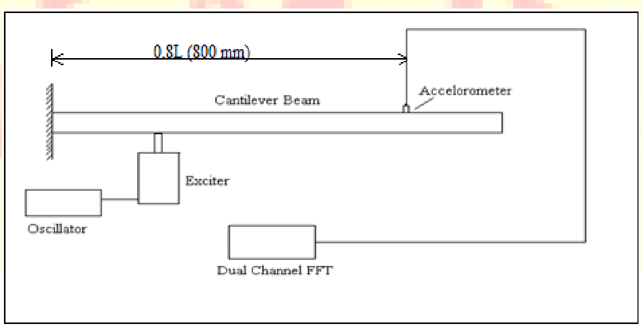


Fig.15 Block Diagram of experimental set up to find response of beam withoutabsorber

Experimental setup to find response of beam with absorber

The entire experimental setup consists of three subsystems: absorber, primary beam, and instruments required for experimentation. The cantilever beam was clamped to rigid support as shown in experimental setup Fig16. Then, absorber was clamped to cantilever beam at free end with help of bolts. Accelerometer was mounted on beam at same position i.e. at 0.8L as shown in Fig.4.2. Accelerometer was connected to FFT Analyzer. The beam was excited with help of exciter. Frequency of excitation can be gradually increased or decreased by using control over power oscillator. The set of readings are taken with absorber attached to the primary beam.

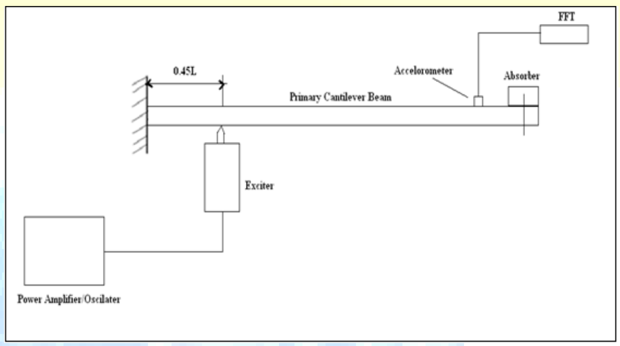


Fig. 16 Block Diagram of Experimental set up to find response of beam with absorber

Experimental procedure

a) For variable stiffness and variable damping condition

1. The beam is fixed with the help of nut-bolts to I beam support mounted on foundation, which acts as fixed support.
2. To find natural frequency of beam experimentally, excitation is given to beam at suitable location i.e. at 0.45L (450 mm from fixed end) and the frequency of excitation is varied from 2 Hz to 80 Hz. Displacement of beam is measured for different frequencies.
3. As per theory when the beam is vibrating at first natural frequency the displacement at the end is maximum.
4. For variable stiffness condition the length of absorber plate is changed to various positions (325 mm, 275 mm, and 225 mm) and displacement of the beam at various frequencies is recorded continuously for respective position of absorber plate.

5. For variable damping condition fix the magnets on guiding plate. Now by changing the distance between the magnets (35 mm, 25 mm and 15 mm) displacement of beam at various frequencies is recorded.
6. The amplitude of beam with and without absorber are tabulated which will be used to draw the response of system
7. For variable stiffness condition the length of absorber plate is changed to various positions (325 mm, 275 mm, and 225 mm) and displacement of the beam at various frequencies is recorded continuously for respective position of absorber plate.

b) For combined variable stiffness and variable damping condition

1. The beam is excited with different natural frequencies in the range of 2 Hz to 80 Hz with steps of 4 Hz at 0.45 L with the help of mechanical contact-type exciter.
2. The vibration amplitudes at the point of attachment (800 mm from fixed end) of accelerometer are measured by tuning the absorber to the excitation frequency to get minimal amplitudes by changing the length of absorber plate.
3. The amplitude of beam with and without absorber are tabulated which will be used to draw the frequency response of system.

Results and discussion

The results of response of primary system (cantilever beam) without absorber and with absorber attached to it were plotted in Fig. 17 and Fig.18 for variable stiffness and variable damping arrangements respectively. These figures show that the vibration amplitudes are suppressed significantly in a wide frequency range.

a. Response of beam with variable stiffness and variable damping arrangement

Vibration amplitudes were measured at different frequencies on thecantilever beam by light weight accelerometer and recorded by FFT vibrationanalyzer. An exciter is used to excite the cantilever beam at 0.45L. The accelerometeris mounted on cantilever beam at 0.8L.

When absorber is attached to the primary system (cantilever beam) with variable stiffness arrangement, from the plot between amplitude vs frequency for it clearly shows that vibration amplitude of primary beam reduced effectively up to 60 to 80% when absorber is attached to beam. For the variable damping arrangement also the plot between amplitude vs frequency for it shows the sufficient reduction in vibration amplitude of primary beam near about 70 to 90% after the addition of absorber system to the primary system

Table III: Displacement of beam in µm at various lengths of absorber plate

Frequency Given by Exciter (Hz)	Displacement of beam (micron)			
	For variable stiffness arrangement			
	Without absorber	325 mm absorber plate length	275 mm absorber plate length	225 mm absorber plate length
2	1268	456	472	615
4	1435	528	628	795
6	2531	764	862	992
8	1983	750	854	982
12	1164	630	744	804
16	792	515	576	652
20	681	369	464	513
24	385	312	378	416
28	307	264	317	354
32	260	225	274	328
36	296	216	252	322
40	332	226	248	298
44	548	228	244	336
48	1226	254	288	399
52	409	258	282	365
56	243	209	232	238
60	163	144	160	162
64	124	105	136	145
68	110	83	88	113
72	96	60	72	98
76	87	54	68	77
80	78	41	42	58

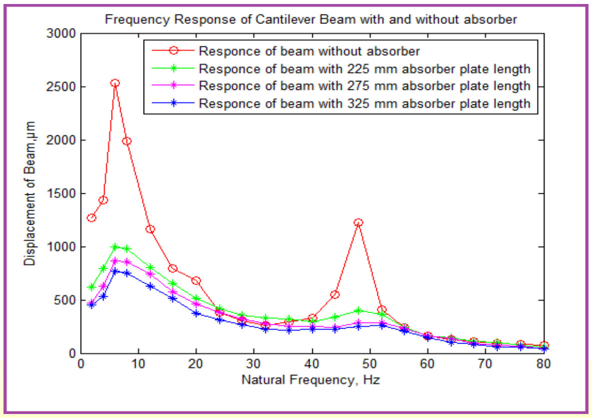


Fig 17 Plot between frequency vs amplitude (for variable stiffness arrangement)

Table IV: Displacement of beam in µm for various positions of magnets

Freq uency Hz	Displacement of beam (micron)			
	For variable damping arrangement			
	Without absorber	35 mm distance between magnets	25 mm distance between magnets	15 mm distance between magnets
2	1268	504	490	426
4	1435	626	541	508
6	2531	722	662	614
8	1983	702	648	602
12	1164	644	610	588
16	792	576	514	455
20	681	464	365	338
24	385	378	310	281
28	307	317	266	238
32	260	315	255	191
36	296	315	252	194
40	332	310	248	196
44	548	308	284	212
48	1226	375	344	257
52	409	302	300	228
56	243	252	221	159
60	163	185	152	114
64	124	146	112	95
68	110	122	95	85
72	96	108	80	67
76	87	98	78	62
80	78	92	72	60

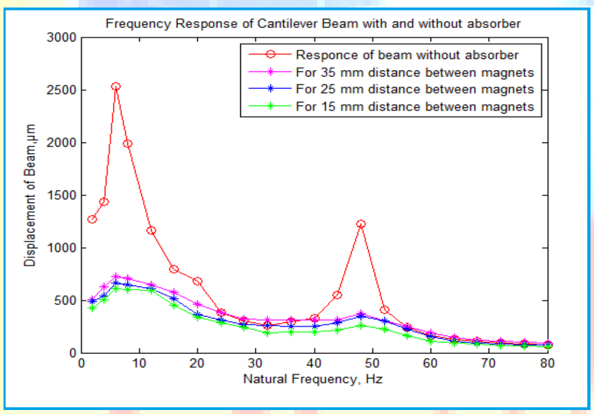


Fig 18 Plot between frequency vs amplitude (for variable damping arrangement)

Response of beam with combined variable stiffness and damping arrangement

The frequency response of a cantilever beam with and without combined variable stiffness and variable damping type vibration absorber attached to the cantilever beam.

To get theresponse of cantilever beam with combined absorber arrangement, vibration amplitudes were measured at different frequencies by tuning the absorber to excitation frequencies by changing the absorber plate length to get minimum amplitude at cantilever beam. When the excitation frequency is equal to natural frequency of absorber, amplitudes of beam are less and avoid the resonance condition. At the same time magnets are placed at optimum position to get minimum amplitude.

The frequency response of cantilever beam with and without absorber attached to the beam is depicted in Fig.19. The green colored line shows the response of beam with absorber attached to the primary cantilever beam. The red colored line is the result without vibration absorber.

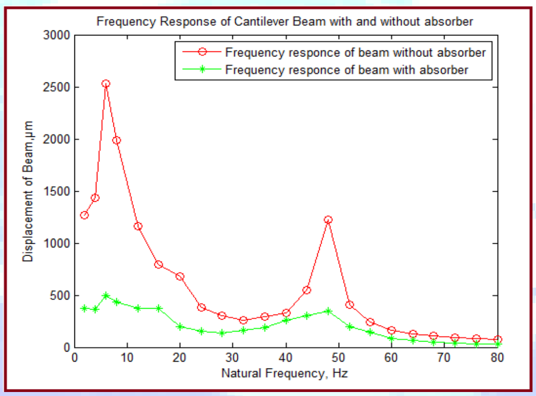


Fig 19 Frequency response of cantilever beam with and without absorber

The Fig.19 shows that the vibration amplitudes are suppressed significantly in a wide frequency range involving fundamental and higher mode resonant frequencies. The combined variable stiffness and variable damping absorber reduces the lower mode as well as higher mode vibrations significantly. The vibration reduction is about 80-90% with the use of combined variable stiffness and damping type vibration absorber. This is the greatest advantage of using present absorber.

The graph of frequency response of masses m_1 and m_2 obtained by theoretical modeling of second order vibration system using Matlab shows that vibration amplitude of primary system is less than that of secondary system.

The frequency response curve obtained by theoretical modeling and experimental testing shows that there is greatest suppression in vibration amplitudes of cantilevered beam due to addition of combined variable stiffness and variable damping type vibration absorber.

Thus variable stiffness and variable damping type of absorber was designed and modeled which can be used as a tunable vibration absorber at multiple frequencies to minimize the vibrations of cantilever beam structure.

CONCLUSIONS

- A cantilevered mass absorber, which uses cantilevered beam and concentrated mass, has demonstrated to be very effective in controlling the vibration in a cantilevered type of beam. This arrangement has been shown to be capable of being incorporated for adaptive use. Effective attenuation has been achieved with this absorber for varying resonance frequencies. This variable stiffness and variable damping type vibration absorber can be used as a tunable vibration absorber at multiple frequencies.
- A design procedure for a vibration absorber capable of providing attenuation at multiple frequencies has been described using ANSYS software using FEA theory. Using Finite Element Analysis of the Variable Stiffness and Variable Damping type vibration absorber, its first six resonances have been moved to create a multiple resonance absorber. A self tuning strategy for the absorber may also be implemented in future work.
- The theoretical modelling of the variable stiffness and variable damping type absorber is done in Matlab software and tested, results obtained shows that 80% reduction in amplitude of mass m_1 i.e. primary cantilever beam structure.

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