Design of Variable Stiffness and Variable Damping Vibration Absorber

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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 work is to design the tunable vibration absorber (TDVA) for vibration control of primary beam system using finite element method through an Ansys Programming. Its stiffness can be varied to adapt the changes in excitation frequencies. Traditional means of vibration control have involved the use of passive and active methods. This study involves the design of variable stiffness vibration absorber by incorporating the use of lead screw for varying the stiffness of secondary system. Finite element analysis is done for the dynamic vibration absorber. FEA code is generated using Mat lab software to find numerical performance of beam structure with dynamic vibration absorber. The proposed absorber is suitable for vibration isolation of beam structure with uniform cross-section and facilitates vibration attenuation at variable excitation frequencies.

Keywords— FEA, TDVA, vibration attenuation, vibration control.

I. INTRODUCTION

Vibrations 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 aim of this project is to investigate the use of a Variable Stiffness and variable Damping type vibration absorber to control the vibrations in beam structure. This study will aim to develop a Variable Stiffness vibration absorber with variable damping to adapt the change in vibratory system. A tunable vibration absorber is developed and its stiffness can be varied by moving the support plate along the length of absorber beam also damping can be varied by varying distance between two magnets. The absorber system is mounted on a cantilevered beam acting as a primary system. The objective is to suppress vibration of the primary beam subject to a harmonic excitation whose frequency may vary. The variable damping absorber will be used for greater effectiveness for higher modes.

II. LITERATURE SURVEY

In reference [1], 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 [2], 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 [3], 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 [4], 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.

III. THEORY

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 long set-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

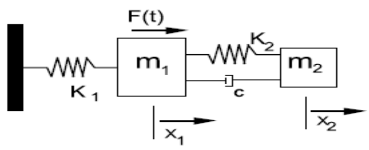

Fig.1: Primary system and absorber schematic

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, which is also outlined by Walsh and Lamancusa.

Referring to Figure 1, consider a primary system with mass m_1, and stiffness k_1 and hence resonance frequency

$w_p = \sqrt{k_1/m_1}$. 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

$$m_1\ddot{x_1} + c(\dot{x_1} - \dot{x_2}) + (k_1 + k_2)x_1 - k_2x_2 = F(t) \quad (1)$$

$$m_2\ddot{x_2} + c(\dot{x_2} - \dot{x_1}) + k_2(x_2 - x_1) = 0 \quad (2)$$

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.

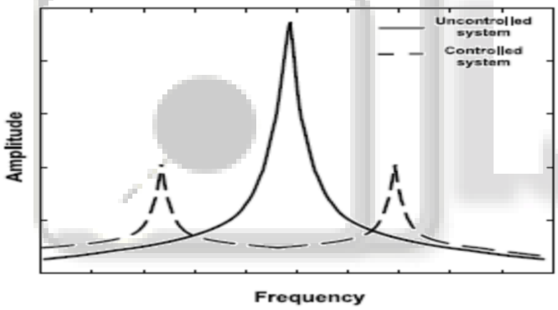

Fig.2: Frequency response with well-tuned absorber

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 distance between the peaks in the structure and the absorber response is 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

$$Z = M_{jaw}[(1 + ja/Q)/(1 - a^2 + ja/Q)] \quad (3)$$

Where $w = \sqrt{k_2/m_2}$ is the resonance frequency of the absorber (secondary system). In Equation 3, α is the ratio between the disturbance frequency, $w_u$and the resonance frequency of the absorber, α=w_u/w The term Q is the quality factor the absorber, which is related to the modal damping of the absorber, defined as

$$Q = \sqrt{k_2m_2}/c \quad (4)$$

For the case of no damping and with a well-tuned absorber (α=1), the system impedance becomes infinite and purely imaginary. The absorber will then attenuate vibration of anything that it is attached to by applying an infinitely large opposite force. The system's response for case of infinite damping, zero damping and intermediate damping are shown in Fig.3.An effective absorber will then have a large Quality factor, typically above 50, and when well-tuned will result in a disturbance ratio α of 1.

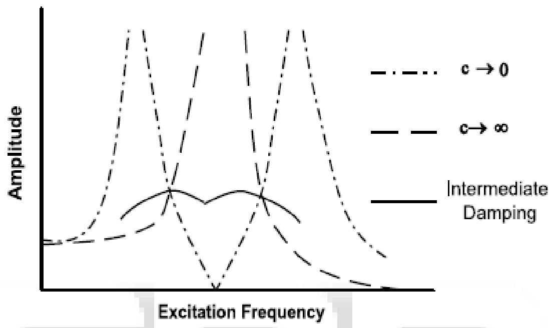

Fig.3: Theoretical response for different amounts of damping

IV. DESIGN OF TUNABLE DYNAMIC VIBRATION ABSORBER

The aim of work here is to develop a practical absorber that can be suitable for multiple resonance frequencies and easily tuned to the excitation frequency. The absorber device that can be investigated here is "cantilevered beam mass absorber" with variable stiffness and variable damping mechanism for tuning purpose. This cantilevered beam mass absorber acts a secondary system

The DVA consists of single concentrated mass attached to cantilever beam. The absorber can be easily tuned to excitation frequency by varying the stiffness of absorber with the help of variable stiffness mechanism.

The variable stiffness mechanism consist of mass and cantilever beam as absorber, lead screws, motor, variable support 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 hole 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.

Damping can be varied by varying distance between two magnets. The proposed model of absorber is as shown in Fig.4 below.

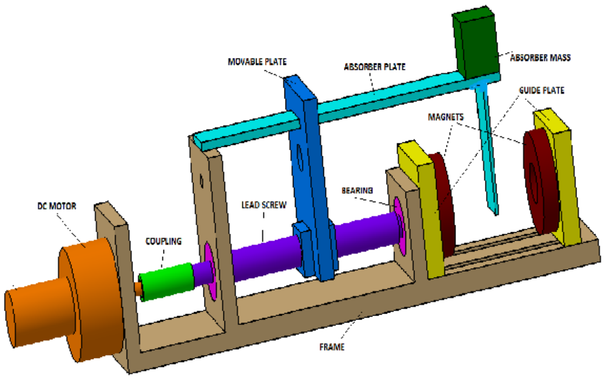

Fig.4: Variable stiffness and variable damping Vibration Absorber

A. *Design of Absorber using FEA in ANSYS*

The design of vibration absorber is carried out using 'FEA'. The absorber is modeled using ANSYS 12 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. The dimensions and material properties of DVA are given below in table I.

Dimensions of absorber	
Plate thickness	4 mm
Plate Length	175 mm
Mass Length	20 mm
Mass Height	25 mm
Magnet size	50*50*12.5 mm
Material Properties of Absorber	
Modulus of Elasticity	70 GPa
Density	2710 Kg/m^3

Table. 1 : dimensions and material properties of absorber

B. *Modal Analysis of Absorber*

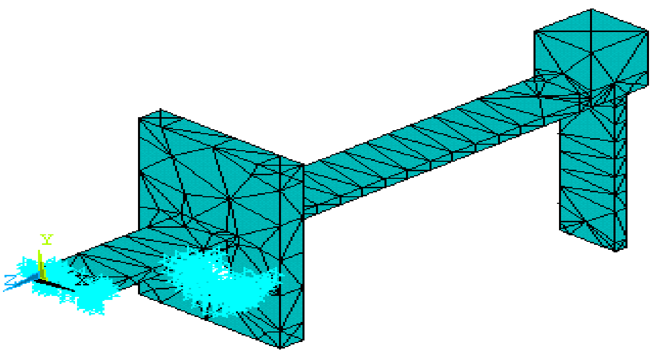

Fig. 5: FEA model of Absorber

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 rod. Absorber is designed by using FEA with software package ANSYS to have resonance frequencies around 50 Hz. First six mode shapes of absorber are found by ANSYS.

The suitability of this element is based on its bending and membrane properties. Also for the modeling of the two rods and absorber masses, SOLID 92 shown in Fig.6 is used.

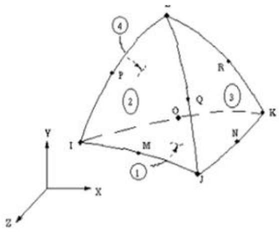

Fig.6: Solid92 3-D-Node tetrahedral structural solid

SOLID 92 is well suited to model of 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. An element used for all parts of absorber is solid 92. Its suitability comes from its ability to accurately model, plasticity, and creep. Here assumptions used are that the threaded rod is not important, so each rod is simplified to have a smooth surface. 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 rods and central section. These elements of the absorber are most important in determining how the absorber behaves at each resonance frequency

The meshed solid model produced in ANSYS is shown in Fig. 5. The model produced in ANSYS is similar to which will be manufactured for testing. The support is modeled by using 'BLOCK' command. Plate and masses are modeled by using 'RECTANGLE' command. The plates are glued to central section creating a single element. The same procedure is used for attaching masses to the plate. ANSYS program is made generalized by using parameters.

The first six natural frequencies obtained are tabulated in table II.

Frequency (Hz)	1	2	3	4	5	6
	65.48	247.14	330.04	375.48	1060.07	1982.30

Table. 2 : First six natural Frequencies of Absorber at 0.4 L

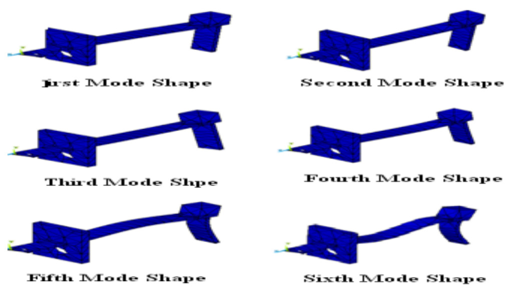

Fig. 7: Mode Shape of absorber for first six natural frequencies at 0.4 L

An understanding of absorber's mode shapes is required so that significant parameters that affect each mode can be determined allowing for possibility for adjusting different modal frequencies. In first mode, mass is moving laterally in vertical direction and in second mode of mass moves in horizontal direction. In third mode plate is moving laterally in horizontal direction. In fourth mode of absorber, plate is moving up and down perpendicular to direction. In fifth mode, plate is moving in torsional mode. In sixth modes, plate is moving similar to fifth mode. The corresponding mode shapes of the absorber is evaluated using finite element analysis by writing an ANSYS program and these are shown above in Fig.7.

C. Effect of Absorber Parameters on Natural Frequencies

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. For higher mode variable damping affects on higher mode. The results from ANSYS are listed below in Table III.

Rod distance (mm)	First freq. (HZ)	Second freq. (HZ)	Third freq. (HZ)	Fourth freq. (HZ)	Fifth freq. (HZ)	Sixth freq. (HZ)
40	65.48	247.14	330.04	375.48	1060.0	1982.3
50	73.17	260.75	364.12	396.70	1664.2	2136.6
60	82.51	275.50	405.49	420.69	1279.6	2289.6
70	93.76	291.85	446.14	456.45	1401.8	2358.9
80	107.3	310.43	475.89	517.98	1535.0	2331.2
90	124.6	332.47	509.0	598.93	1677.3	2259.9

Table. 3 :Effect of Support Position on Absorber Natural Frequencies

D. 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 Fig. 8.

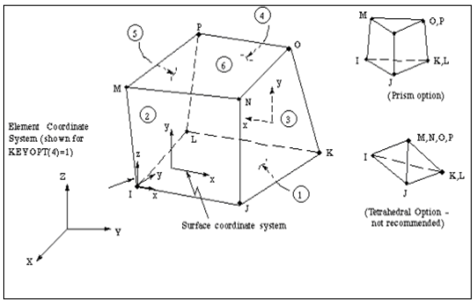

Fig.8 Solid 45 –structural solid

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.

Length of beam	1000 mm
Thickness of beam	10 mm
Width of the beam	65 mm
Density of material	7800 kg/m^3
Modulus of elasticity	210 Gpa
Material	Mild Steel

Table. 4: DIMENSIONS AND MATERIAL PROPERTIES OF BEAM

The FEA mesh model of Fixed-Free beam acting as primary system used in experimentation is shown in Fig. 9.

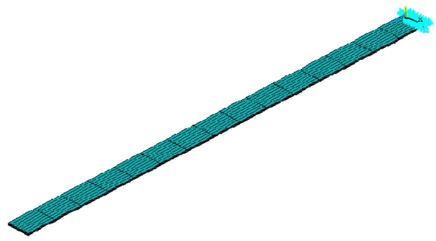

Fig.9: FEA Mesh Model of Cantilever beam

V. CONCLUSION

A cantilevered mass absorber, which uses cantilevered beam and concentrated masses, 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 and further work will be planned for this device. Also, Variable damping is incorporated for higher mode shape vibration suppression.

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 cantilevered mass absorber its first six resonances have been moved to create a multiple resonance absorber.

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