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A REVIEW OF ADAPTIVE VIBRATION ABSORBERS FOR VIBRATION ISOLATION OF BEAMS

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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 vibration absorbers are frequently used to control and to minimize excess vibration in structural systems. The Dynamic vibration absorbers are used to reduce the undesirable vibrations in many applications such as electrical transmission lines, pumps, machine tools, gas turbines, engine, bridges, structures etc. 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 review work is to study various types of dynamic vibration absorber with variable stiffness mechanism for tuning and to study design of the tunable vibration absorber for vibration control of primary system (beam structure) using finite element method through an Ansys. 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 and implementation of control system to make absorber tunable by incorporating the use of ball screw for varying the stiffness of secondary system. The objective of this review work is to introduce methods of vibration control and describe practical methods for their application. Several scenarios and case studies will be presented, with emphasis on pragmatic solutions to industrial vibration problems.

KEYWORDS- FEA, TDVA, vibration attenuation, vibration control.

INTRODUCTION

A dynamic vibration absorber (DVA) is essentially a secondary mass, attached to an original system via a spring and damper. The natural frequency of the DVA is tuned such that it coincides with the frequency of unwanted vibration in the original system. This results in a subcomponent of the total structure adding a large input impedance to the primary structure, thus 'absorbing' the inertial energy transferred from the primary structure. Active DVAs are characterized by a secondary mass, mounted to a vibratory primary system, that is directly connected to an actuator and electronic control system. This study is different in the way that it involves an adaptive component in the design of DVA using a novel design for adaptive mechanism i.e. variable stiffness for the absorber device. Adaptive tuned means the ability of the DVA to change its internal properties to reflect the changes in the vibratory environment. This study is different in the way that the "active" component is implemented in the design of absorber. Here, actuator will be controlling the natural frequency of the absorber system (secondary system). Adaptive-passive method involves the use of passive elements which can be optimally tuned to perform over a certain frequency range. . The active part is often achieved with an actuator and control loop in a similar manner to full active control. The actuator is often an electromagnetic, piezoelectric, and magnetostrictive or shape memory alloy actuator .The passive part can be achieved with a physical mass spring-damper set. Beam structures are frequently used in investigations of combined absorption.

Beams are utilized for this application because of the simplicity of the structure and the inherent mass, stiffness and damping values they possess. The equivalent material properties of a beam can be altered with active control. Some methods for achieving this are:

1. A beam with a mass that can be moved along the beam. Different locations of the mass will result in different resonant frequencies of the structure.
2. A beam with mass in which support can be moved along the length of beam to vary the stiffness of beam to have different resonant frequencies.
3. A natural frequency of dynamic vibration absorber is automatically tuned to by adjusting distance between magnets as a repelling force system.

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

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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. In reference [5], Prof.H.D.Desai, Prof.Nikunj Patel presented an analytical and experimental investigation of a tuned undamped dynamic vibration absorber in torsion.

ADAPTIVE TUNABLE VIBRATION ABSORBER

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.

Absorber with Single Resonance

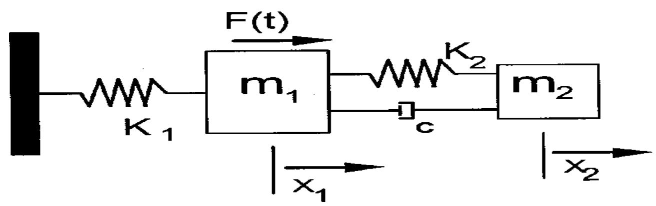

Figure 1: Primary system and absorber schematic

Referring to Fig.1, consider a primary system with mass m_1, and stiffness k_1, and hence resonance frequency $\omega_p = \sqrt{k_1/m_1}$.

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

$$m_1 \ddot{x}_1 + k_1 x_1 + k_2(x_1 - x_2) = f_0 \sin\omega t$$

$$m_2 \ddot{x}_2 + k_2(x_2 - x_1) = 0 \qquad (1)$$

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. 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.

The effective mass, m_2/m_1, plays an important part in determining if the absorber is effective. The effective mass is a 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.

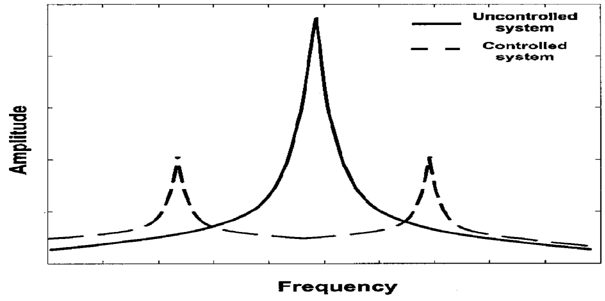

Figure 2: Frequency Response with well tuned absorber

ABSORBER WITH MULTIPLE RESONANCES

The aim of the work here is to develop a practical absorber that facilitates extension of the previously described absorption scenario to multiple resonance frequencies. There are several reasons to pursue multiple resonance absorbers, including: the potential for attenuating vibration at multiple frequencies with a single device, potential for expanding the bandwidth of the device by having multiple absorber resonances in close proximity; and the potential for using the device in different orientations for different applications, where different orientations will excite different absorber resonances and so the device will provide attenuation at different frequencies.

The absorber device being investigated here, the ‘‘dual mass absorber’’, is shown in the sketch of Fig.3. The dual mass absorber consists of two rods supporting two equal masses on either side of a center section, which is attached to the target structure. One rod is smooth and the other threaded, with a stepper motor used to rotate the threaded rods to move the masses in or out. The absorber resonances are changed by this movement of the masses, within bounds set by parameters such as the mass and dimensions of the bells, supporting rod thickness and material, and the separation distance between the supporting rods.

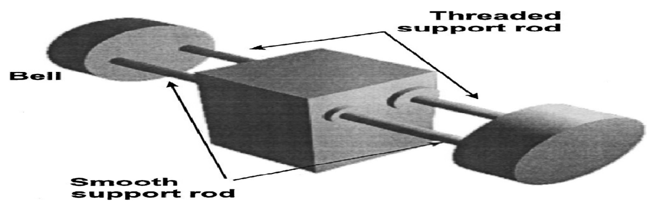

Figure 3: Dual Mass Vibration Absorber

The first six mode shapes of the absorber for a typical set of material properties, Poisson's ratio of 0.3, steel density of 7800 kg/m3 and Young's modulus 2073109 Nm22 as evaluated using finite element analysis, are illustrated in Fig.4. Observe in Fig.4 that there is a pattern in the mode shapes, with modes pairs [1+2],[3+4] and [5+6] having similar motion with the exception that the masses are moving in phase with the odd numbered mode and out of phase with the even numbered mode. For example, modes 1 and 2 are both characterized by vertical displacement of the masses, with the masses in phase for mode 1 and out of phase for mode 2. Observe also the close proximity of the frequency for these modal pairs. This pairing of modes will become important in terms of expanding the bandwidth of the absorber device.

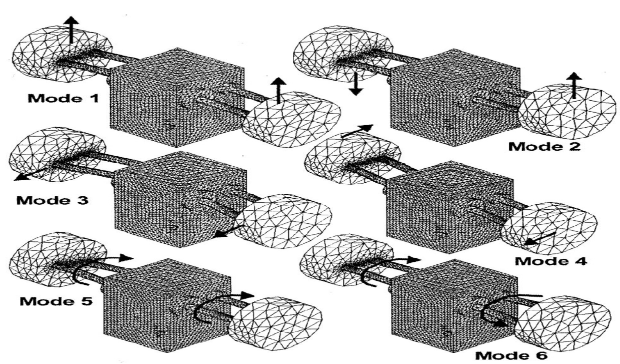

Figure 4: Absorber Modes, clockwise from top left, mode 1, 2, 3, 4, 5 and 6 for Modal frequencies 95, 105, 196, 204, 295, 304 Hz.

A valuable characteristic of this absorber is that the absorber resonances can be modified independently. This can be explained by examining the mode shapes of the absorber. To modify a modal frequency in general two parameters can be changed; mass and/or stiffness. As the absorber's mass will remain relatively constant, a method of modifying the stiffness as seen by the various absorber modes is sought. The varying the separation distance of the supporting rods has a negligible impact upon the placement of the first two modes.

BEAM EXPERIMENTS

Laboratory tests were next performed on a more rigid structure by clamping the absorber to a simply supported beam of dimensions of 1700 x 50 x 25 mm. It is important to perform these tests, as the transformer that is the target of this work is also very rigid. Referring to the sketch in Fig.5, a shaker was to input broadband excitation to the beam, and an accelerometer placed at the base of the absorber to measure the beam frequency response.

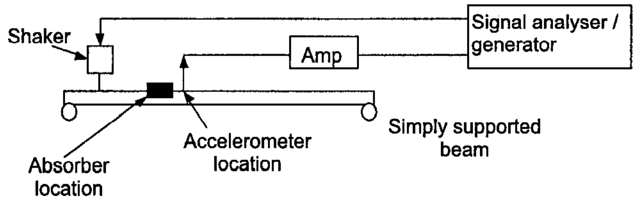

Figure 5: Experimental Setup for Beam Testing

Plotted in Fig.6 is the ''uncontrolled'' beam response and the beam response with a tuned absorber. To measure the uncontrolled response, the absorber was attached to the beam and the masses wound all the way in against the base section, to stop any motion of the bells. Observe that there is a significant resonance peak at approximately 67 Hz in the uncontrolled spectrum. This frequency is within the adjustment range of the first two resonance frequencies of the constructed absorber, and so the locations of the masses were manually adjusted to target this. The result, also plotted in Fig.6, is a notching of the resonance peak.

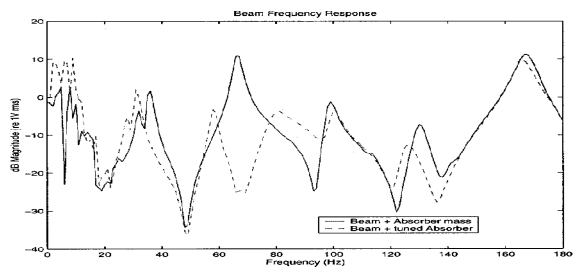

Figure 2: Notch at Beam Resonance

ANTI-RESONANCE TUNABLE VIBRATION ABSORBER

The present article discusses a method of vibration control of a structure by using the vibration absorber without damping. In this method, 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. The analyses and the algorithm for the auto-tuning control are developed. In order to validate the control method and the analysis, experimental tests have been carried out.

In the present article, a method of vibration control for structures has been presented with consideration of higher modes of vibrations. In our method, the principal vibration mode is controlled by use of auto-tuning anti-resonance control of the tunable vibration absorber without damping.

RESPONSE OF STRUCTURE WITH ABSORBER

In order to suppress vibrations in a wide frequency region, the present article proposes a system consisting of a tunable vibration absorber and a vibration absorber with a magnetic damper. The tunable absorber suppresses the principal mode, and the vibration absorber with the damper suppresses higher vibration modes.

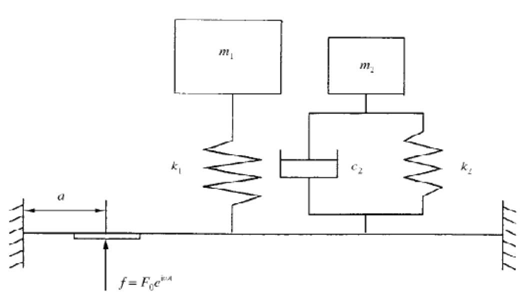

Figure 7: Geometry of the beam with vibration absorbers

Fig.7 shows the geometry of the structure with the absorbers in which one of the absorber has no damping (the tunable absorber for anti-resonance is called absorber 1), and the other has the magnetic damper (the absorber for higher modes is called absorber 2). The locations of the absorbers are important, because the system parameters vary with the stiffness of the tunable absorber.

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In order to use the absorber for higher modes, the tunable absorber (absorber 1) lies on the nodal point of the second mode of the structure, and so the motion of absorber 1 does not affect the second mode, because, the second mode vibration is not generated due to the excitation at the second nodal point. Absorber 2 lies at the anti-nodal point of the second mode.
In order to indicate the validity of the present system, consider a simple beam with both built-in edges. The theoretical model is shown in Fig.7. The vibration response can be obtained by using the transfer matrix method.

NUMERICAL TESTING

Numerical calculations are carried out for the beam with vibration absorbers used in the experiment as mentioned below. The dimensions for the variable stiffness absorber used in the calculation are also depicted in Table 1. An aluminum beam with channel shape is used. Table 2 shows the dimensions of the beam. The nodal point of the second mode is calculated as x= 46 cm from the left end of the beam where the auto-tuning absorber is attached. Numerical calculations are carried out under the assumption that the sinusoidal force with amplitude 10 N acts on the point measured from 24.5 cm from the left end of the beam.

Table 1. Dimensions of the variable stiffness absorber for anti-resonance

Attached mass at the tip of beam	0.6 Kg
Eigen frequency of the absorber	13-29 Hz Variable
Spring of the absorber	Beam made of SS Bar
Total mass of the absorber	4.29 Kg

Table 2. Dimensions of the beam

Material	Aluminum
Shape of the cross-section	Channel
Size of the cross-section (mm)	21.5*60.4*3.3
Length of the beam	980 mm
Young's modulus	7.06 E^{10} N/m^2
Density	2.7 E^3 Kg/m^3
Total mass of the beam	0.845 Kg
Moment of the cross-section	$1.2*10^{-2}$ m^4
Mass of the attached plate	0.5 kg

The geometry of anti-resonance vibration absorber which is also called as variable stiffness vibration absorber is shown in Fig. 8.

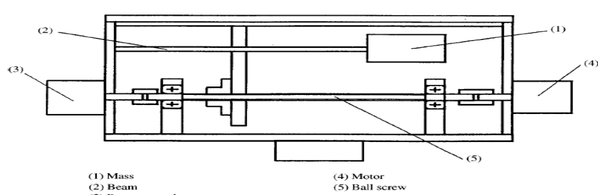

Figure 8: Geometry of the anti-resonance absorber

EXPERIMENTAL TESTING

In order to validate the present control method and analyses, experimental tests have been carried out for the same model mentioned in the numerical calculation. Fig.9 illustrates the geometry of the control system used for experimental testing of vibration absorber with simply supported beam.

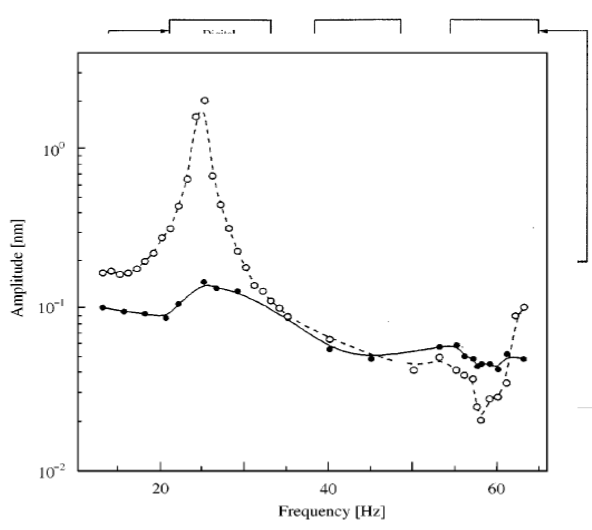

Figure 9: Geometry of Experimental setup

The acceleration sensors were attached at two points at x=46 cm (point A) and x=65 cm (point B) from the left end of the beam, and detected signals were input to the FFT analyzer as shown in Fig.5. The forced excitation was applied at the point x=24.5 cm from the left end of the beam by an electromagnet. Since the electromagnet generates attractive force only, and the force is in proportion to the square of the current, the exciting force becomes the square of half sine pulse under the sinusoidal input current. This implies that the force has various frequency components. Then the absorber to suppress higher modes is needed.

Figure 10: Frequency response at point A in the beam with control and without control

The frequency response at the point A of the beam is depicted by the solid line in Fig.10, where the anti-resonance states are created. Experimental tests have been carried out for a flexible beam, and it is clarified that the vibrations can be controlled to be significantly small by use of the present method.

VARIABLE STIFFNESS VIBRATION ABSORBER

A tunable vibration absorber is developed and its stiffness can be varied on-line. The absorber system is mounted on a clamped-clamped 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. A system modeling is conducted. The frequency response of the system is given to show the operating range of the absorber system. Using a simplified two-degree-of-freedom model, two auto-tuning methods are studied. The methods differ in the way of how to identify the exciting frequency. The first method follows a common practice that uses the frequency of the maximum peak in the response spectrum as the exciting frequency. The second method makes use of information of both the response spectrum and the natural frequencies. An experimental study is conducted to compare the two methods. The study has shown that the second method performs better than the first method in terms of frequency tracking ability and robustness to disturbance.

EXPERIMENTAL TESTING

Fig.11 shows the entire experimental setup that consists of three subsystems: absorber, primary beam, and computer control system. The main part of the absorber system is a cantilever beam (8) and its free end is attached by an absorber mass (9). The beam length can be varied by moving a movable support (7). The movable support is driven by two lead screws (5). The lower lead screw is driven by a DC motor (3) and through a pulley-belt set (4), the upper lead screw rotates simultaneously. An encoder (6) is attached to the upper lead screw to measure the position of the movable support. The absorber beam is an aluminum rod of 6.35 mm in diameter. The range of the beam length variation is 306 mm. The motor is a 12V DC permanent magnet reversible motor. An electromagnetic shaker (1) is used to generate a non-contact exciting force. A small permanent magnetic plate (2) is glued on the beam to interact with the electromagnetic force. The shaker is driven by a power amplifier. A Pentium III (550MHz) PC computer is used to control the system. The Data Acquisition (DAQ) Board is DS1102 from dSPACE. Control Desk (dSpace) is used to interface between Matlab, Simulink and DS1102.

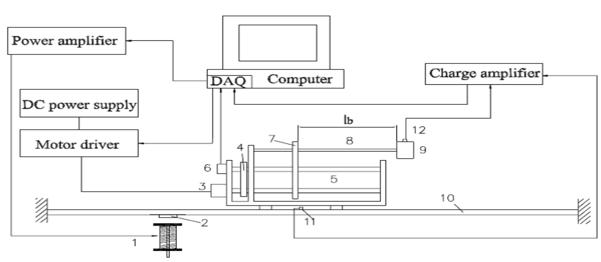

Figure 11: Experimental set-up to test VSVA

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The primary beam (10) is a clamped-clamped beam made of aluminum. The dimension of the beam is 50.8 mm (width) × 5.08 mm (thickness) × 1057 mm (length). Vibration of the primary beam is measured by an accelerometer (11). This signal is used for on-line tuning. Another accelerometer (12) is placed on the absorber mass and its signal is used for comparison. A charge amplifier is used to condition the accelerometer signals. A motor control command is generated by the computer and sent to a motor driver circuitry to regulate the current from a DC power supply.

EXPERIMENTAL FREQUENCY RESPONSE

Newton's second law of motion used to find differential beam element equations and mode summation method is used to transfer set of differential equations into ordinary differential form. The transfer function is obtained and frequency response shown in Fig. 12 is obtained from the same transfer function in Matlab.

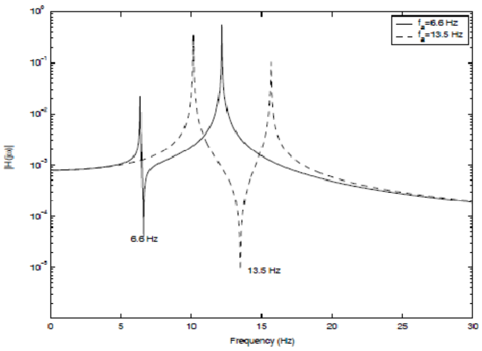

Figure 12: Frequency responses of the beam with VSVA Conclusion

The various types of vibration absorbers with their modeling, design procedure, construction, working, self-tuning strategy and experimental procedure were studied in depth.
In these types of vibration absorbers, it is possible to adjust its stiffness for the best performance in order to make the natural frequency of vibration absorber equal to exciting frequency, resulting in minimized or zero vibrations. Tuned vibration absorbers are particularly effective when the excitation frequency is close to the natural frequency of the primary system.

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