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AN ADAPTIVE VIBRATION ABSORBER

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1 INTRODUCTION

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 [1,2]. The utilisation 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.

The limitation of being a single frequency device has been overcome by using additional modes. A successful example of this type of device is The Dog Bone vibration absorber [3]. The Dog Bone absorber consists of a clamping device and a rod with masses on each end. By using an offset weight the manufactures claim that a torsional vibration mode can be induced which, combined with its normal bending modes, facilitates absorption by vibration modes beyond those available with the usual Stockbridge damper design [4,5]. To achieve attenuation at multiple (target) system resonances, several absorber resonance frequencies must also have the corresponding resonance frequency. The tuning of an absorber to meet this characteristic is presented in this paper.

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, which is also outlined by Walsh and Lamancusa [6].

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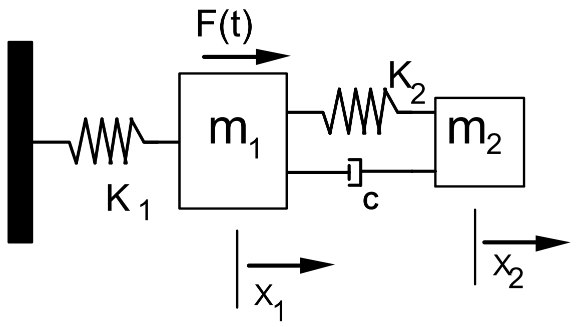

Figure 1: Primary system and absorber schematic

2 Background: Absorber with a single resonance

Referring to Figure 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 and viscous damping c is added to the system, then the differential equations describing the above system are

$$m_1\ddot{x}_1 + c(\dot{x}_1 - \dot{x}_2) + (k_1 + k_2)x_1 - k_2x_2 = F(t) \tag{1}$$

$$m_2\ddot{x}_2 + c(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \tag{2}$$

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 Figure 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[7] 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, $v = m_2/m_1$, 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. Considering the system as infinitely rigid in all but the direction normal to the resonant surface, then the mechanical impedance, Z of such a system is given by

$$Z = Mj\alpha\omega \left[\frac{1 + j\alpha/Q}{1 - \alpha^2 + j\alpha/Q}\right] \tag{3}$$

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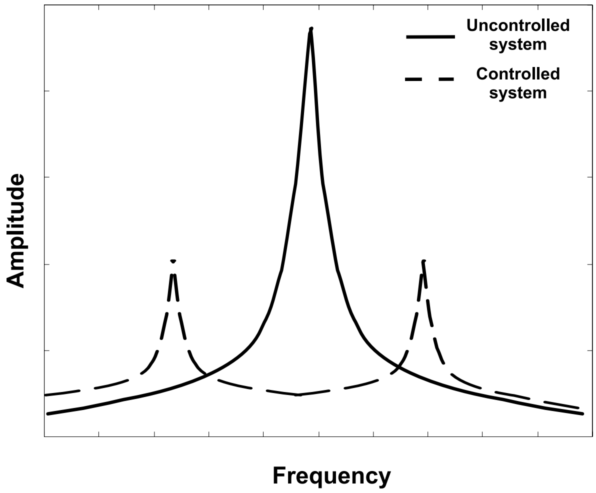

Figure 2: Frequency response with a well tuned absorber

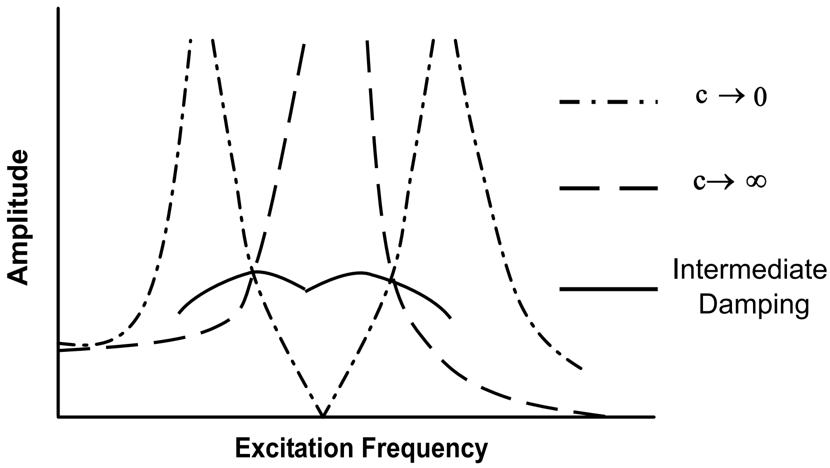

Figure 3: Theoretical response for different amounts of damping

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

$$Q = \frac{\sqrt{m_2 k_2}}{c} \tag{4}$$

For the case of no damping and with a well-tuned absorber ($\alpha = 1$), 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 Figure 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.

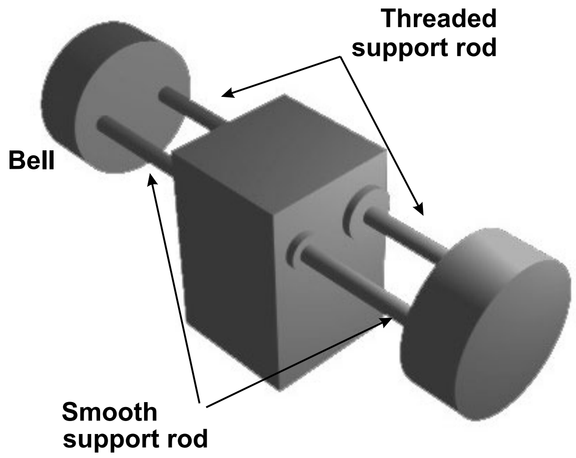

Figure 4: The dual mass absorber

3 Designing a Multiple Resonance Absorber

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; the 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 Figure 4. The dual mass absorber consists of two rods supporting two equal masses on either side of a centre 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.

The first six mode shapes of the absorber are illustrated Figure 5. The absorber was modelled using a Poisson's ratio of 0.3, steel density of 7800 kg/m^3 and Young's modulus 207 x 10^9 Nm^{-2} as evaluated using finite element analysis, are illustrated in Figure 5. Observe in Figure 5 that there is a pattern in the mode shapes, with modes pairs (1+2, 3+4, 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 characterised 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.

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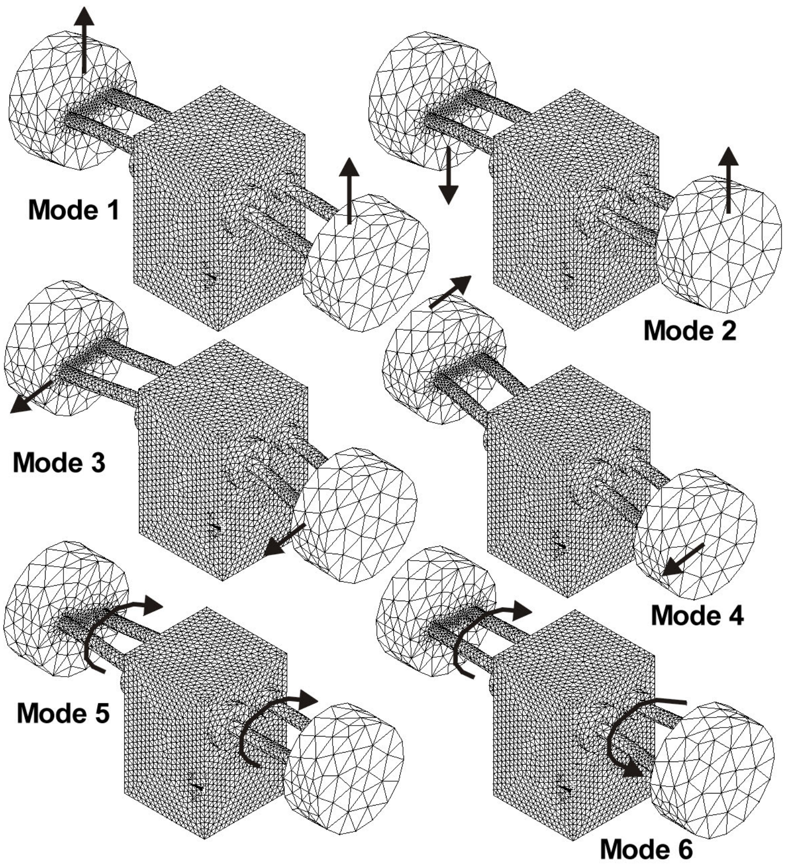

Figure 5: Absorber Modes. Clockwise from top left, mode 1,2,3,4,5 and 6. Modal frequencies 95, 105, 196, 204, 295 and 304 Hz respectively.

A valuable characteristic of this absorber is that the 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. Viewing the modes shapes in Figure 5, is it apparent that the rod diameter, mass and rod length contribute to the stiffness of the first two modes. The same applies for modes 3 and 4, however here the distance between the support rods is an additional factor affecting the stiffness of these modes. Furthermore, as modes 5 and 6 are torsional, the moment of inertia becomes important. This is affected by the dimension of the masses. There then exists absorber dimensions which effect "relatively" independently the adsorber modes, (at least close enough to facilitate design tuning).

As mentioned previously, the target of interest for the absorber work being described here is electrical transformers in Australia. With a mains electricity frequency of 50 Hz, the rectified sinewave excitation will be at 100 Hz, 200 Hz, 300 Hz, etc. The above methodology for designing an absorber with resonances around these frequencies will be demonstrated (theoretically) with this in mind. The resonance frequency of the first first mode is placed at approximately 100 Hz by first selecting a bell mass, and then choosing an appropriate supporting rod diameter and length. The bell masses were selected to be 2 kg each based upon the practicality of building and testing a prototype device. The supporting rod diameter was 10mm, with a length of 75mm. With these dimensions, finite element analysis predicted that the resonance frequencies of the first two modes would be 97 and 103 Hz. Note that these modes flank the 100 Hz excitation frequency target, and taken together yield an absorber with a large bandwidth over which significant attenuation can be achieved. This will greatly simplify tuning and improve the robustness of the device to design and environmental factors.

Having placed the first two modes, the second pair of modes was tuned to flank the 200 Hz target frequency via adjustment of the separation distance between the supporting rods. Referring to Figure 6, observe that a separation distance of approximately 42 mm will have the desired result. Observe also that varying the separation distance of the supporting rods has a negligible impact upon the placement of the first two modes, which continue to flank the 100 Hz target frequency over the range of separations being considered here.

For the final target frequency of 300 Hz, the resonance frequencies of the fifth and sixth absorber vibration torsional modes, were adjusted by varying with dimensions (diameter and width) of the bells. Referring the Figure 7, varying the bell dimensions has a small effect upon the resonance frequency of modes 1-4, owing to an associated change in the effectively length of supporting rods. However, the dominant effect is on modes 5 and 6. Observe that a bell length of approximately 44 mm will produce the desired result, where the resonance frequencies of modes 5 and 6 flank the 300 Hz target.

4 Conclusions

A design procedure for a vibration absorber capable of providing attenuation at multiple frequencies has been described. Using Finite Element Analysis of the dual mass absorber it's first six resonance frequencies have been moved to create a multiple resonance absorber.

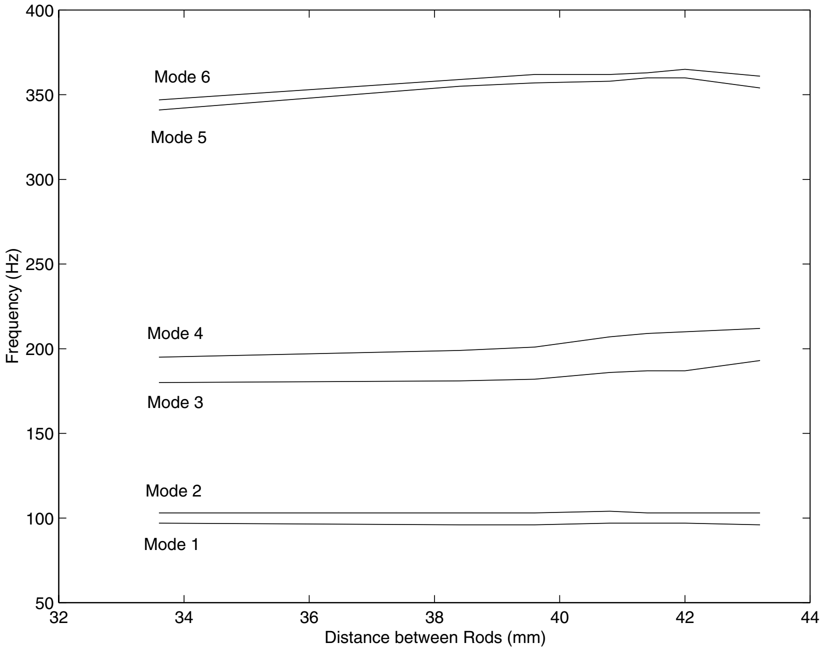

Figure 6: Tuning Mode three and four

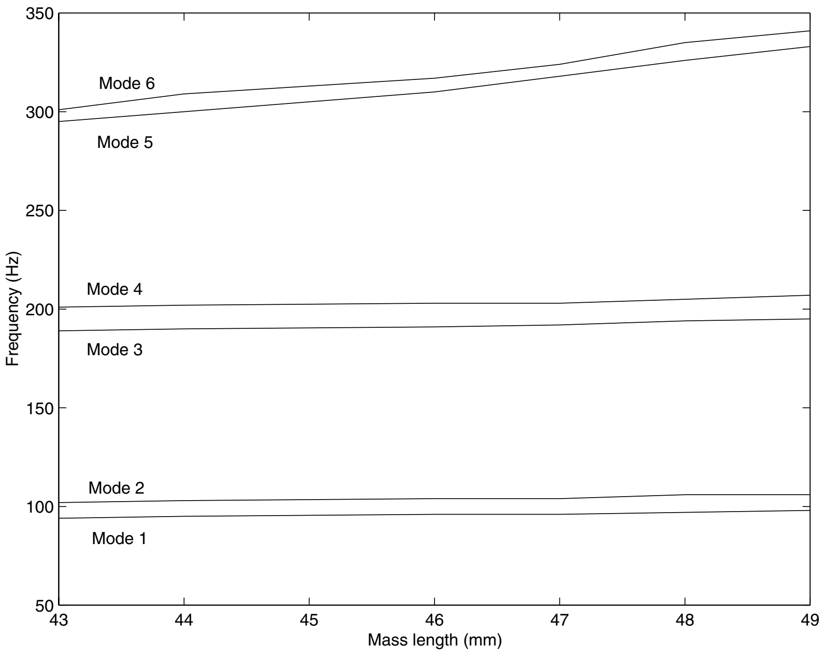

Figure 7: Tuning Mode five and six

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