Journal of Sound and Vibration (2000) **232**(3), 601–617
doi:10.1006/jsvi.1999.2755, available online at http://www.idealibrary.com on IDEAL®

AN ACTIVELY TUNED SOLID-STATE VIBRATION ABSORBER USING CAPACITIVE SHUNTING OF PIEZOELECTRIC STIFFNESS

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(*Received* 10 *August* 1998, *and in final form* 2 *November* 1999)

A tunable solid-state piezoelectric vibration absorber and an active tuning method were developed and demonstrated. A passive vibration absorber generally acts to minimize structural vibration at a specific frequency associated with either a tonal disturbance or a lightly damped structural vibration mode. Because this frequency is rarely stationary in real applications, damping is usually added to ensure some level of effectiveness over a range of frequencies. Maximum response reductions, however, are achieved only if the absorber is lightly damped and accurately tuned to the frequency of concern. Thus, an actively tuned vibration absorber should perform better than a passive one and, furthermore, could be made lighter. In its simplest form, a vibration absorber consists of a spring–mass combination. A key feature of the tunable vibration absorber described herein is the use of piezoelectric ceramic elements as part of the device stiffness. The effective stiffnesses of these elements were adjusted electrically, using a passive capacitive shunt circuit, to tune the resonance frequency of the device. The tuning range of the absorber is thus bounded by its short- and open-circuit resonance frequencies. An alternative tuning approach might employ resistive shunting, but this would introduce undesirable damping. Another feature of the device is the ability to use the piezoelectric elements as sensors. A control scheme was developed to estimate the desired tuning frequency from the sensor signals, to determine the appropriate shunt capacitance, and then to provide it. The shunt circuit itself was implemented in 10 discrete steps over the tuning range, using a relay-driven parallel capacitor ladder circuit. Experimental results showed a 20 dB maximum, and a 10 dB average improvement in vibration reduction across the tuning range, as compared to a pure passive absorber tuned to the center frequency, with additional benefit extending beyond the tuning range.

1. INTRODUCTION

Vibration is an important aspect of many engineering systems, from machine tools to structure-borne noise in aircraft [1]. In most cases, such vibration is undesirable and requires attenuation. Attenuation techniques range in complexity from relatively simple narrowband passive elastomeric vibration absorbers to fully active broadband vibration control systems.

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Passive attenuation methods represent an important class of approaches to the control unwanted structural vibrations [2]. One particular method of passive vibration suppression involves the use of passive vibration absorbers (PVAs). Passive vibration absorbers are conceptually simple devices consisting of a mass attached to a structure via a spring or via a parallel spring–damper combination. PVAs are commonly constructed of an elastometric material sandwiched between the structure and a reaction (or proof) mass. The primary function of these devices is to increase the effective dynamic stiffness of a structure over a relatively narrow frequency band. Increasing the dynamic stiffness of a structure reduces its dynamic displacement (assuming the forcing level remains constant). In practice, PVAs are typically used to minimize vibration at a specific frequency associated with either a lightly damped structural mode or a tonal disturbance. The advantages of using vibration absorbers are low cost, low weight, and ease of attachment. The fact that a PVA may only be used effectively at a single frequency, however, can sometimes be a significant drawback.

While a common use of PVAs is to reduce vibration in tall buildings or towers [3], they have also been successfully used in the aviation industry for some time. For example, the DC-9 for many years used a set of four PVAs attached to each engine pylon to reduce aft cabin noise associated with the operating spool frequency of the engines [4]. Similarly, both the Fokker F27 and the Saab 340 aircraft use PVAs attached directly to fuselage frames to reduce interior cabin noise levels [5].

In these applications, the absorbers provide considerable vibration attenuation at specific frequencies. Performance can be seriously degraded, however, if the disturbance source changes frequency. If this occurred, the PVA could, in principle, be physically re-tuned to maintain optimal performance, but this is generally impractical. Thus, there is a need for vibration absorbers with tunable variable properties.

Tunable vibration absorbers are passive vibration absorbers having stiffness, mass, or damping that can be (actively) adjusted to change one or more device resonance frequencies. Such tunable absorbers are sometimes used to track frequency-varying disturbances or to increase the bandwidth of a vibration attenuation method. Recently, Northwest Airlines initiated plans to upgrade 173 of its DC-9s with active tuned mass absorbers built by Bary Controls [6, 7]. Due to the increased weight and complexity of using such devices, however, they have not found widespread use.

Recently, PCB Piezotronics, Inc. and the Center for Acoustics and Vibration at Penn State were involved in the development of a piezoelectric ceramic inertial ("proof mass") actuator [8]. When used passively, this actuator behaves like a vibration absorber. This device provided a starting point for the development of an actively tuned vibration absorber, based on a previously unexploited tuning mechanism.

Two key features of a tunable vibration absorber are the method by which the stiffness or mass of the device is altered and the magnitude of the resulting change in resonance frequency. Because it is difficult to vary the effective mass of a solid-state device, the present work focused on a way to vary the effective stiffness. Because they exhibit relatively strong coupling between electrical and mechanical behavior, piezoelectric materials offer an attractive means of implementing variable stiffness: changing the electrical boundary conditions can change the effective stiffness of the material. If some of the stiffness of a vibration absorber is provided by piezoelectric elements, the resonance frequencies of the device could be modified by using an adjustable external electric circuit.

2. BACKGROUND

A mathematical model was developed to assist in the development of insight concerning tuning of a vibration absorber in which some of the effective stiffness is provided by

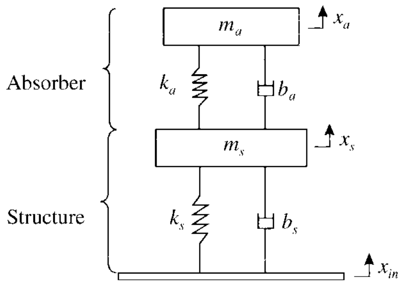

Figure 1. Lumped parameter model of passive vibration absorber attached to a s.d.o.f. structure.

piezoelectric elements. First, a single-degree-of-freedom (d.o.f.) structure and passive absorber model is created. Next, this model is modified to incorporate a piezoceramic stiffness element. Finally, an external adjustable electrical shunt circuit is added for frequency tuning.

2.1. VIBRATION ABSORBER/STRUCTURE INTERACTION

Consider a damped vibration absorber attached to a single d.o.f. structural system, as shown in Figure 1. Let m_s, k_s, and b_s represent the effective mass, stiffness, and damping of the structure, and m_a, k_a, and b_a represent the mass, stiffness, and damping of the absorber. This is a textbook model of the simplest system that can be used to study the behavior of a structure including a vibration absorber [9].

Recall that the natural frequency of the undamped s.d.o.f. structure without an attached vibration absorber, ω_s, is

$$\omega_s = \sqrt{\frac{k_s}{m_s}}. \tag{1}$$

The transfer function from the input disturbance displacement, X_{in}, to the structural displacement (at the point of attachment of the vibration absorber), X_s, may be found as

$$\frac{X_s(s)}{X_{in}(s)} = \frac{k_s(m_a s^2 + k_a)}{m_s m_a s^4 + (m_a(k_s + k_a) + m_s k_a)s^2 + k_s k_a}. \tag{2}$$

The frequencies of the poles (ω_1 and ω_2) and zero (ω_{abs}) of this coupled structure/absorber system are

$$\omega_{1,2} = \frac{\sqrt{2}}{2}\sqrt{\frac{m_{\mathrm{a}}(k_a + k_s) + m_s k_a \mp \sqrt{m_a k_a(m_a(k_a + 2k_s) + 2m_s k_a) + (m_a k_s + m_s k_a)^2}}{m_a m_s}}$$

$$\text{and} \quad \omega_{abs} = \sqrt{\frac{k_a}{m_a}}. \tag{3a, b}$$

Note that ω_{abs} defines the frequency of minimum response of the coupled structure/absorber system, and is equal to the natural frequency of the (undamped) vibration absorber itself.

Figure 2 shows the magnitude of the frequency response functions from an input base displacement to the structural displacement, both with and without an attached vibration

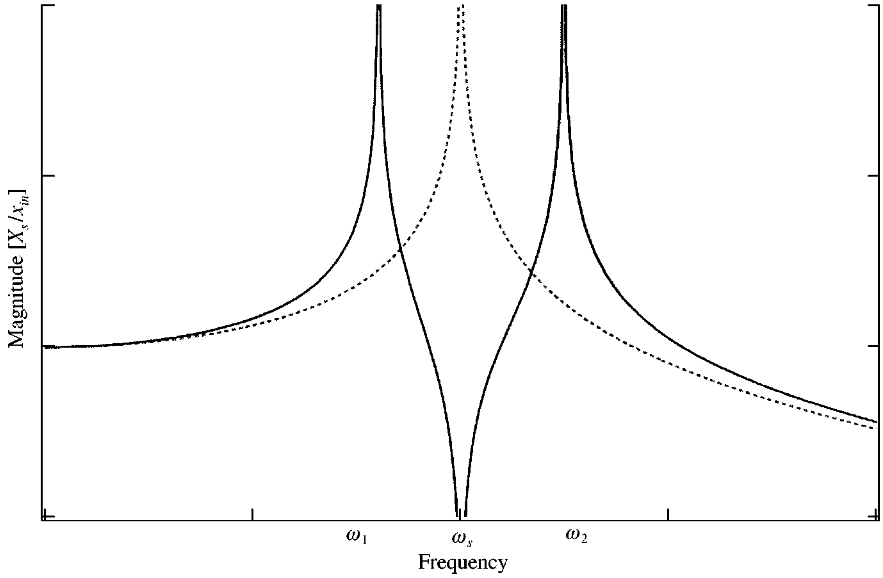

Figure 2. Sample frequency response functions for the s.d.o.f. structure and the s.d.o.f. structure with attached PVA: - - - - -, without PVA; ——, with PVA.

absorber. The structure without a vibration absorber exhibits high response at the frequency ω_s. In the case where a vibration absorber is used, its natural frequency is tuned to that of the structure alone. The structural response is reduced dramatically at the frequency of the absorber, but increased at frequencies both above and below this frequency (i.e., at the frequencies of the poles of the coupled system, ω_1 and ω_2). Note that damping reduces the amplitude of the peaks as well as the depth of the minimum, and that the mass ratio affects the spacing of the two resonance frequencies [9].

Evidently (from equation (2)), the frequency at which structural response is a minimum depends only on the absorber mass and stiffness. Thus, structural response may be greatly reduced at any disturbance frequency, by tuning the natural frequency of the absorber to that of the disturbance, and by using an absorber with little damping. This observation may be exploited to develop a tuning control strategy that maintains optimum absorber performance when the structure is subjected to tonal forcing.

2.2. FREQUENCY TUNING VIA PASSIVE ELECTRICAL SHUNTING

With the knowledge that the natural frequency of the absorber defines the frequency of minimum structural response, the next step involved developing a method for changing the stiffness of the absorber in real time. Electrical shunting of a piezoelectric device has the effect of changing the effective stiffness (and thus the natural frequency) of the device. The theory behind such a shunting approach is presented in the conference paper by Davis *et al.* [10] as well as in the thesis by Davis [11], and is summarized here for completeness.

As already noted, a vibration absorber may be modelled using lumped parameters such as a spring, k_a, and mass, m_a, as shown in Figure 3(a). Similarly, an inertial actuator may be modelled as a spring–mass system with a forcing element, F_p, in parallel with the spring element, k_a, of the absorber, as shown in Figure 3(b). Placing the effective stiffness of the forcing element of an internal actuator in parallel with the inherent structural stiffness of the

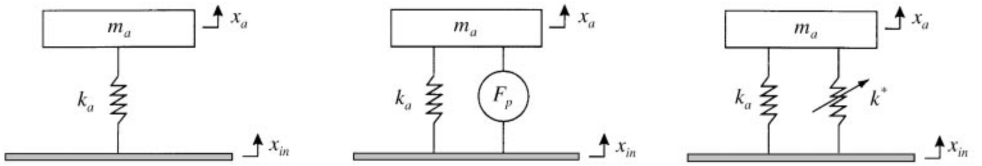

Figure 3. (a) Passive vibration absorber lumped parameter model; (b) inertial actuator lumped parameter model; (c) passively shunted inertial actuator lumped parameter model.

absorber, as shown in Figure 3(c), results in a passive device stiffness that is the sum of the two stiffnesses.

Electrically shunting the piezoceramic elements in a piezoceramic vibration absorber can change their effective stiffness. An expression for the effective stiffness, k^*, of an electrically shunted piezoceramic elements is [11]

$$k^* = k^E\left(1 + \frac{k_p^2}{1 - k_p^2 + \alpha(s)}\right), \tag{4}$$

where k^E is the effective short-circuit stiffness of the piezoelectric ceramic element, k_p is its electromechanical coupling coefficient, $\alpha(s)$ is the non-dimensional ratio of the electrical impedance of the piezoceramic (i.e., $1/sC_p^T$, where C_p^T is the capacitance of the piezoceramic measured under constant stress) to the electrical impedance of the shunt circuit, and s is the Laplace parameter (i.e., $s = \mathrm{i}\omega$ where ω is radian frequency).

Regardless of the type of simple electrical shunt circuit used (e.g., resistor, capacitor, inductor), there are limits on the range of values k^* can take on. These limits are conveniently defined in terms of the short- and open-circuit stiffnesses of the piezoelectric element, as well as the shunt circuit electrical impedances. When short circuited, the shunt impedance is effectively zero and equation (7) reduces to $k^* = k^E$. At open circuit, the shunt impedance is effectively infinite and $k^* = k^E(k_p^2/(1 - k_p^2))$. Thus the tunable range of k' is bounded by its short- and open-circuit stiffnesses, which are related by the electromechanical coupling coefficient, k_p.

In the application described here, the value of k_p (the planar coupling coefficient of the piezoelectric material) was approximately 0·6. For a device made solely of this type of piezoceramic material, the change in stiffness from short to open circuit could be as high as 56%, resulting in an almost 25% change in natural frequency. In practice, however, the stiffness of the piezoceramic is in parallel with the inherent mechanical stiffness of the actuator, and only some fraction of the net device stiffness may be changed due to electrical shunting.

Note that equation (4) may be complex depending upon the type of shunt circuit used (e.g., if a resistor is used as the shunt circuit). A complex stiffness would indicate that the device has mechanical properties similar to those of an anelastic material, including hysteretic damping. In terms of vibration absorber performance, adding damping has the effect of increasing the response magnitude at the natural frequency of the absorber. Because the goal of this research was to maintain minimum structural response at ω_{abs}, shunt circuits that added damping to the system were not considered.

An ideal capacitor is a purely reactive element and does not dissipate energy or provide damping. Figure 4 shows the ratio of the effective stiffness of a capacitively shunted piezoceramic element to its short-circuit sitffness. Note that, for this case, the tuning ratio is the ratio of the shunt capacitance, C_{sh}, to the clamped capacitance of the piezoceramic. The tunable piezoceramic stiffness varies smoothly with increasing shunt capacitance. Also note

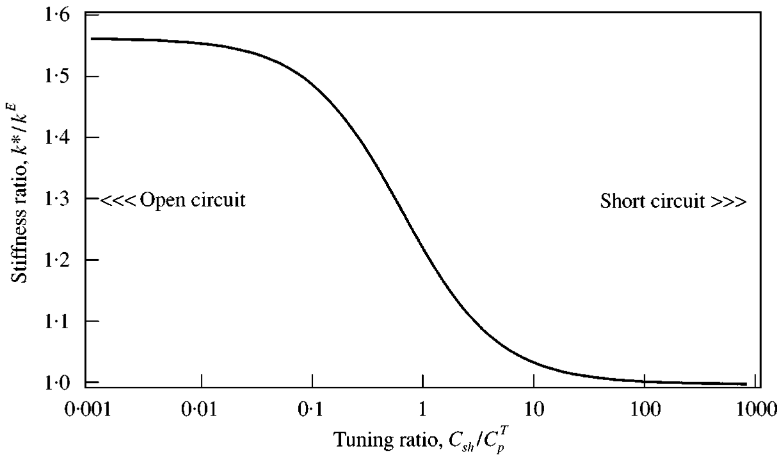

Figure 4. Effect of shunt capacitance on effective piezoceramic stiffness.

that roughly a four-order-of-magnitude change in shunt capacitance is necessary to effectively change from open to short circuit (or *vice versa*).

As mentioned previously, the total stiffness of a piezoceramic vibration absorber may be considered the sum of the effective tunable piezoceramic stiffness and the inherent stiffness of the device. The relative magnitudes of the two stiffnesses determine the net frequency change possible via electrical shunting.

An experiment was conducted using a commercially available piezoceramic inertial actuator as a passive, electrically shunted vibration absorber. Figure 5 shows a schematic of the device used for the experiments, PCB Model X712A02. The actuator (the lower flat cylinder in Figure 5) is approximately 2 in in diameter and approximately 3/8 in thick. The reaction mass is attached to its top by a standard 10-32 threaded stud, making it relatively easy to coarsely tune the device frequency by changing the mass. The base of the device also has a 10–32 threaded stud used for attaching it to a structure.

Note that the thickness and the radius of the piezoelectric element relative to that of the base metal disk was determined by design to maximize the device electromechanical coupling coefficient [8], and that this also maximizes the tuning range for this type of device configuration. Other device configurations might have higher device coupling coefficients and tuning ranges.

The purpose of the experiment was to measure the natural frequency and modal damping ratio of the device under a variety of capacitive shunt conditions ranging between short and open circuit. In the experiment, the device was attached to a shaker and accelerometers were used to measure both the input (i.e., the shaker) acceleration and the reaction mass acceleration. The ratio of the two acceleration measurements formed a frequency response function which was then curve-fit to approximate the natural frequency and modal damping ratio of the actuator for a given shunt condition. The electrodes of the inertial actuator were attached to a solderless breadboard where discrete values of capacitance could be used to shunt the device.

The results of the passive shunting experiment verified the ability to predictably tune the natural frequency of the piezoceramic vibration absorber between a short-circuit natural frequency of 313 Hz and an open-circuit natural frequency of 338 Hz. The resulting change in natural frequency was approximately 7·5% from short to open circuit. Next,

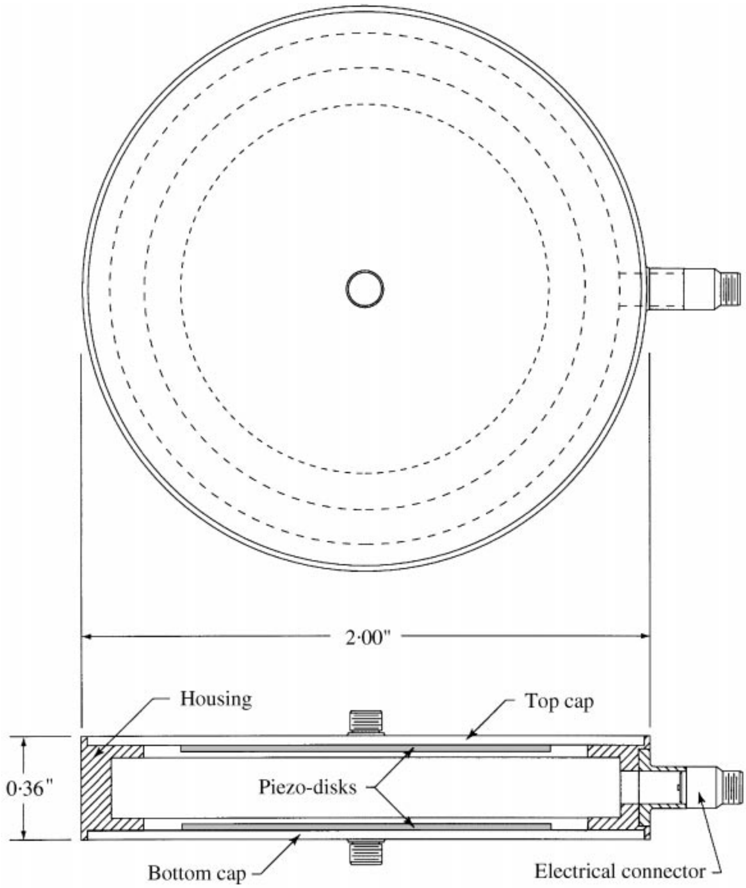

Figure 5. Schematic of the PCB Model X712A02 inertial actuator.

a frequency tuning control method for the shunted piezoceramic vibration absorber was developed.

3. ACTIVE TUNING CONTROLLER

The preceding section showed that the natural frequency of a vibration absorber that uses a piezoelectric stiffness element can be passively tuned with an external capacitive shunt circuit. This section describes the approach used to implement an active tuning method.

3.1. CONCEPT

Consider a flexible structure with several well-spaced structural modes of vibration subjected to a tonal disturbance. Attaching a conventional passive vibration absorber to the structure, tuned to the tonal disturbance frequency, would reduce structural response at that frequency. Thus, as long as the disturbance frequency remained constant, a high level of attenuation would be achieved (i.e., structural vibration would be minimized). If, however,

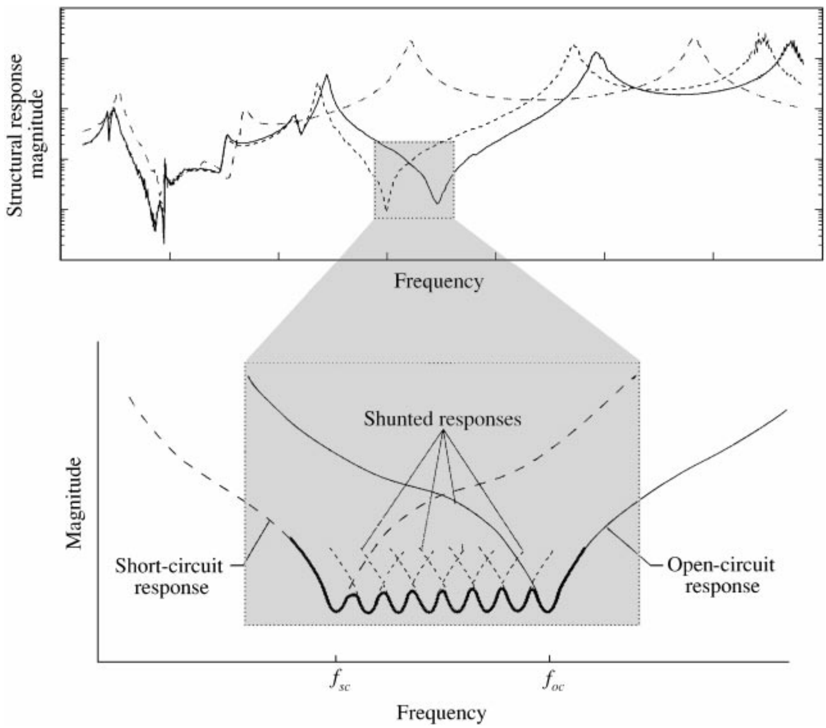

Figure 6. Conceptual tuned vibration absorber structural response: — — —, no absorber; – – – , short circuit; ——, open circuit.

the total disturbance frequency changed, performance would decrease (i.e., the vibration level would increase). Therefore, it would clearly be beneficial to have a tunable vibration absorber and a tuning method to track a changing disturbance frequency and appropriately re-tune the frequency of the absorber.

Developing a tuning method for the vibration absorber involved three steps: first, an appropriate system signal was identified, from which the desired tuning frequency could be estimated. Next, a method for actually estimating the frequency of structural vibration from the signal was developed. The final step involved formulating a control scheme to determine and provide the proper shunt capacitance.

The concept of monitoring the disturbance frequency and tuning the absorber to maintain minimum structural response is illustrated in Figure 6. The upper plot represents the broadband structural response for a structure both with and without an attached vibration absorber. The highlighted area of the upper plot is enlarged in the lower portion of Figure 6 to show the transfer function zeros (or minima) created by the addition of a short-circuit, shunted, and open-circuit vibration absorber to the structure.

First, consider the case when the piezoceramic element within the vibration absorber is short circuited (indicated by the dashed line in the highlighted section in the lower plot of Figure 6). If a tonal disturbance acted on the structure at a frequency equal to f_{sc}, the structural response would be minimal. If the disturbance frequency were to decrease while the PIA remained short circuited, the structural response would increase. Similarly, if the disturbance frequency were to increase to the frequency f_{oc}, while the piezoelement remained short circuited, the structural response would again increase. However, if the absorber were "re-tuned" by adjusting the electrical shunt impedance to an open-circuit condition, the structural response would remain minimal. In addition, if the disturbance frequency were to fall anywhere between f_{sc} and f_{oc}, there exists a shunt impedance that will deliver minimum structural response.

Consider the following situation: a structural mode is excited by a pure tone harmonic disturbance which varies in frequency by a few percent of some nominal frequency. In principle, a piezoelectric vibration absorber could be added to the structure such that at a shunt tuning ratio of one, the natural frequency of the absorber would be equal to the nominal disturbance frequency (i.e., the natural frequency of the absorber for a tuning ratio of one is half-way between f_{sc} and f_{oc}). If sensing (in the form of determining the frequency of the disturbance), control (in the form of a command signal based on the sensed frequency to alter the electric shunt impedance), and actuation (in the form of a means to alter the electrical shunt condition) were provided, then a minimum structural response could be maintained within the band defined between the open- and short-circuit frequencies.

The heavy solid line in Figure 6 illustrates the conceptual response of a discretely tuned vibration absorber. In this illustration, the shunt capacitance does not vary smoothly between f_{sc} and f_{oc}, but instead is discretized. Thus, determining a means to estimate the disturbance frequency, choosing the correct value of shunt impedance based on the estimated frequency, and physically changing the shunt impedance are the main subject of the next section. Two important questions to be addressed are: (1) what sensor(s) could be used and (2) once a sensed signal is acquired, how can frequency information be extracted from it?

3.2. IMPLEMENTATION

It is clearly desirable to try to use the fewest number of sensors in order to reduce control system complexity, weight, and power consumption. With this in mind, a method for sensing the tonal disturbance frequency based on the voltage produced by the piezoceramic within the PIA was developed.

Assuming a linear model for the coupled structure/absorber system, the voltage produced by the piezoceramic elements within the absorber is directly proportional to the piezoceramic strain and the shunt circuit electrical impedance. Thus, for an open-circuit vibration absorber being forced near resonance, the piezoelements would be under considerable strain due to the motion of the reaction mass (and corresponding small motion of the attached structure), producing considerable voltage. If, however, the vibration absorber were short circuited, there would be no measurable voltage across its terminals. Instead, short-circuiting would produce a large current (for the same forcing conditions). Thus, because shunting the absorber can vary the electrical impedance of the device from nearly short circuit to nearly open circuit, using the electrical state of the piezoceramic to estimate vibration frequency can only be effective if both voltage and current are used as sensor variables.

For the prototype system described here, a control system was used in which the A/D conversion process required voltages within a prescribed range. Thus, it was necessary to convert the current to a corresponding voltage. The current estimation process was realized using an op-amp as an ideal current-to-voltage converter [12].

The controller used for tuning the vibration absorbed is shown in Figure 7. The controller used two inputs and one output. The inputs were the voltage, V_v, and a voltage proportional to the current V_i. The output was a voltage proportional to the tuning impedance of the shunt circuit. The main elements of the controller were: (1) the band-pass filters (2) the frequency estimation logic and (3) the control voltage calculation.

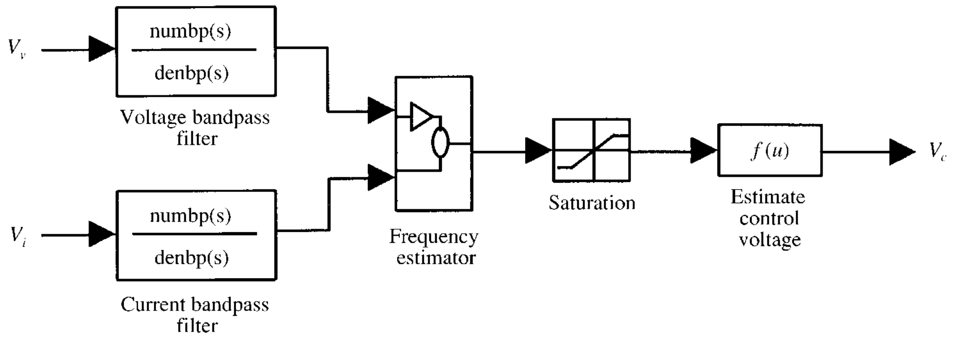

Figure 7. Control system block diagram.

The band-pass filters removed any DC-component of the input signals and attenuated high-frequency noise. The filters were second order with low and high cut-off frequencies of approximately 130 and 780 Hz respectively. Note that for the prototype system, the frequency range of interest (i.e., the range of frequencies defined by the short- and open-circuit resonance frequencies of the absorber) was approximately 290–350 Hz. Thus the pass-band encompassed the range of interest well.

The filtered signals were next used to estimate the frequency of the tonal disturbance. For the prototype version, frequency estimation was done by the control computer. For increased performance, analog circuitry could be implemented in the form of a phase-locked loop to convert the sensed voltages to a voltage proportional to frequency for use by the control system.

With a proper estimate for the frequency of vibration, the remaining task of the controller was to calculate an appropriate control voltage with which to vary the shunt impedance. Before the controller could be programmed, however, it was necessary to determine a method for physically altering the shunt capacitance. Recall that in Figure 4, the effective tuning ratio range for capacitive shunting is roughly 0·01–100 times the capacitance of the vibration absorber measured at constant stress. The measured capacitance of the prototype system at constant stress was approximately 0·072 μF. Therefore, to tune the natural frequency of the PIA between f_{sc} and f_{oc}, a shunt capacitance range of roughly 0·7 nF to 7 μF was required.

Variable capacitors do exist. However, the majority of variable capacitors have relatively small ranges (e.g., even a range of 12–100 pF is not common) and must be tuned by physical means. Programmable capacitors also exist, but due to their added complexity and resistance, they were not considered for the prototype system. Instead, a "ladder" circuit of discrete capacitors wired in parallel was used to tune the vibration absorber.

The effect of placing capacitors in parallel is a net capacitance equal to the sum of the individual capacitances. Figure 8 illustrates a conceptual shunt circuit with several parallel capacitors. If the frequency tuning band of the shunted absorber were discretized into a finite number of capacitive impedances, a control law could be developed to select a number of parallel capacitors whose sum would be the net electrical impedance needed to tune the actuator very close to the estimated disturbance frequency. Clearly, the number of discrete capacitance values used in a specific application will depend on the overall size of the frequency tuning range, the intrinsic damping of the absorber, and the acceptable deviation from minimum response. Finer discretization of the tuning band will yield a more uniformly low system response.

Consider a discretized shunt circuit with 10 discrete capacitance levels ranging from approximately 0·7 nF to 7 μF. Ideally, each discrete shunt capacitance will tune the

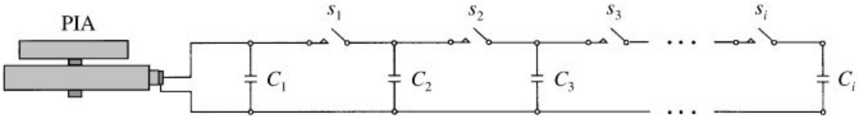

Figure 8. Conceptual "ladder" capacitive shunt circuit.

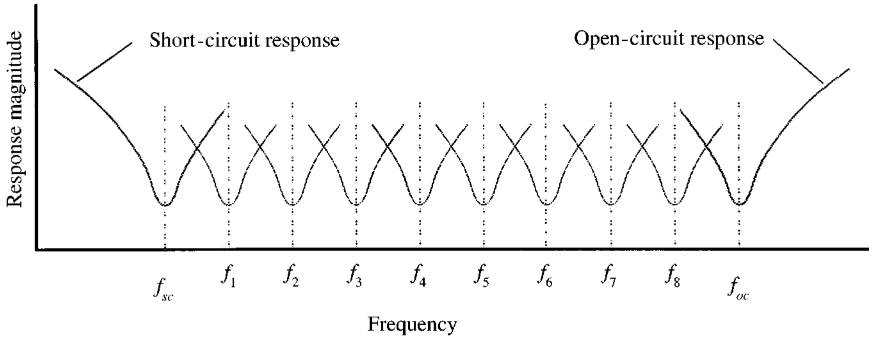

Figure 9. Frequency band shunt discretization.

absorber such that structural response will be a minimum for a prescribed frequency within the control bandwidth. Thus, the tunable frequency band was divided into nine frequency bands between the short- and open-circuit natural frequencies of the absorber, as shown in Figure 9.

The frequencies at the minima of each frequency band (labeled as f_{sc}, f_1 through f_8, and f_{oc} in Figure 9) define frequencies at which the cumulative combinations of shunt capacitance prescribe the shunted natural frequency of the absorber. First, consider the open-circuit shunt case. The open circuit corresponds to a very small (or zero) shunt capacitance. Thus if the capacitor C_1 in Figure 8 were very small (or removed) and all switches were open, the absorber would be in an approximately open-circuit shunt condition (i.e., the natural frequency of the absorber would equal f_{oc}). Next, consider closing the switch s_1 in Figure 8. The net capacitance of the shunt circuit increases to the sum of C_1 and C_2. Conversely, the natural frequency of the absorber decreases to f_8 (assuming C_1 and C_2 are chosen correctly). Similarly, closing switches s_1 and s_2 will increase the net shunt capacitance to the sum of the C_1, C_2 and C_3 and the absorber natural frequency (and thus the frequency of minimum structural response) will decrease to f_7. Closing all of the switches will increase the net shunt capacitance to the total of all of the capacitors in the ladder circuit. Thus, if C_1–C_i are chosen correctly, the sum of all the capacitors will be large enough to approximate a short-circuit shunt condition (i.e., the total parallel capacitance will be greater than or equal to $100 \times C_p^T$). The question remains, however, as to how to open and close the switches in the ladder circuit.

The switches in the shunt circuit shown in Figure 9 determine the number of capacitors in parallel with the absorber. Ideally, these switches would operate in response to a prescribed signal from the control system. In doing so, however, the switches should not introduce any additional electrical impedance into the shunt circuit. A relay provides one possible solution to this problem.

The control voltage, V_c, shown in Figure 9, is used to turn the relays on in succession. Variable resistors wired between each relay driver circuit were adjusted such that for $V_c = 1$ V, relay #1 would turn "on" while others would remain "off". For $V_c = 2$ V, relays

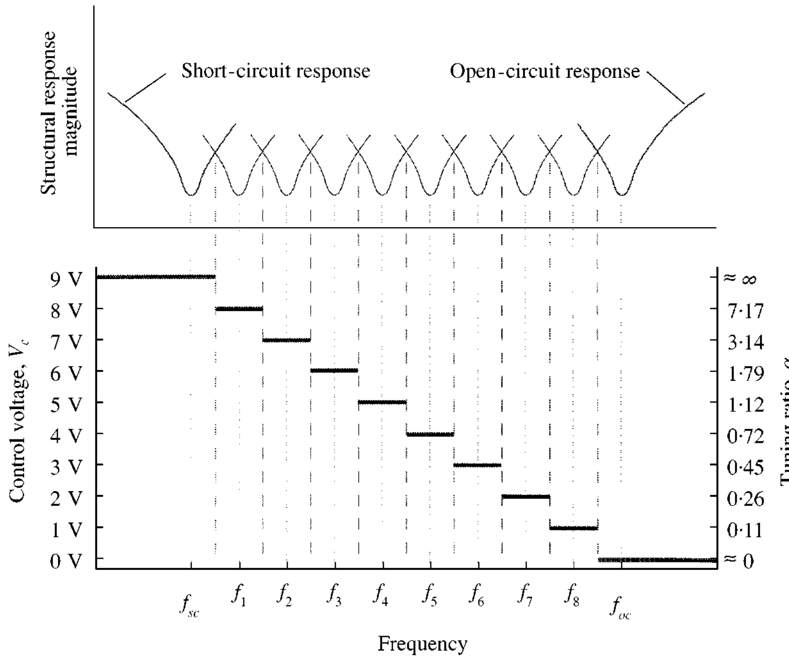

Figure 10. Control voltage to structural response correlation.

#1 and #2 would turn "on" while the other relays would remain "off". Similarly, more relays would turn "on" while the remaining relays remained "off" for increasing integer voltage levels up to and including 9 V. What remained was to program the control logic to output discrete integer voltage levels corresponding to desired shunt capacitance levels. Note that discrete voltage outputs were required to ensure the appropriate relays in the switching circuit were either "on" or "off".

Figure 10 illustrates the correlation between control voltage, V_c, absorber capacitive tuning ratio, α, and the frequency of minimum structural response. From Figure 10 it is clear that, for a control voltage of 0 V, no transistors are "on" and thus the absorber is shunted with one capacitor, C_1. As stated earlier, if C_1 is sufficiently small (say approximately $0{\cdot}01 \times C_p^T$), the absorber will behave as if it were open circuited. For a control voltage of 1 V, the first transistor turns "on" and the shunt capacitance is increased from C_1 to the sum of C_1 and C_2. If C_2 equal $0{\cdot}1 \times C_p^T$, the net tuning ratio would be approximately 0·11 and the natural frequency of the absorber and therefore the frequency of minimum structural response would be f_8.

As shown in Figure 10, varying the control voltage, V_c, from 0 to 9 V changes the frequency of minimum structural response from approximately the open-circuit natural frequency to approximately the short-circuit natural frequency. Referring to the control system block diagram, in Figure 7, the *estimate control voltage* block contains the code to convert the estimated disturbance frequency to an appropriate control voltage.

4. TUNING CONTROL EXPERIMENTS AND RESULTS

Several experiments were designed to evaluate the effectiveness of the tunable piezoceramic inertial actuator. In the experiments, a representative structure (a

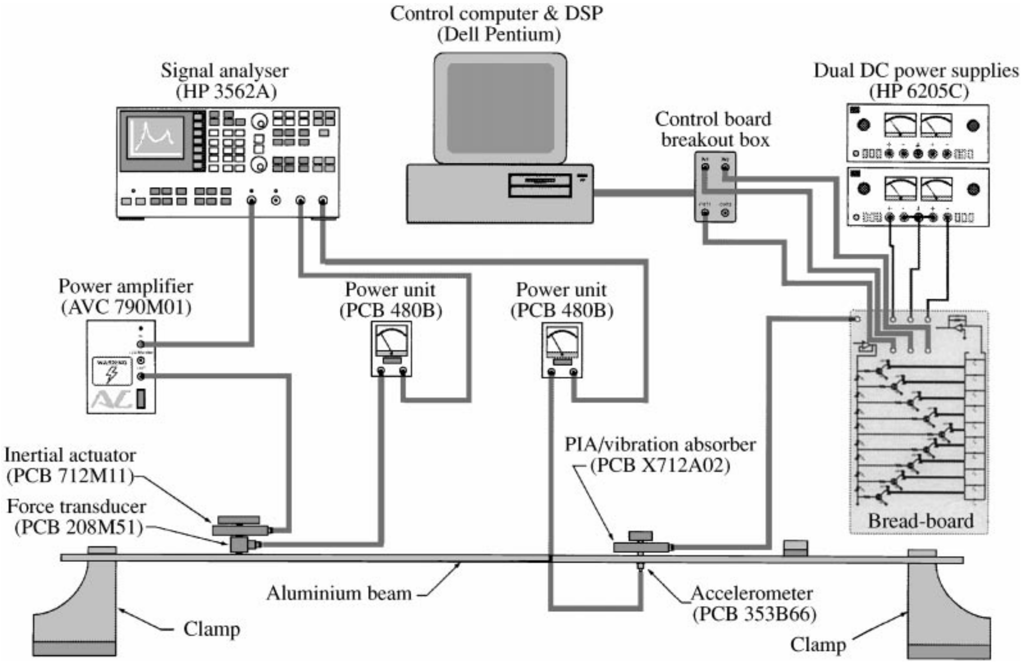

Figure 11. Experimental set-up.

clamped–clamped beam) was driven with a tonal force disturbance. First, structural accelerance measurements were taken both with and without the passive absorber attached to the structure. Measurements in which the absorber was both short and open circuited were used to define the effective tuning bandwidth for the shunt control system. Finally, a sine sweep was used to vary the disturbance frequency from just below the absorber short-circuit natural frequency to just above the absorber open-circuit natural frequency for the short circuit, open circuit, and actively tuned cases.

Figure 11 shows the experimental set-up for the semi-actively tuned vibration absorber experiments. The representative structure was a $0{\cdot}913 \times 0{\cdot}038 \times 0{\cdot}006$ m aluminum beam rigidly fixed at both ends. An inertial actuator placed 10 cm from the left end of the beam was used to apply a disturbance force to the structure. The drive signal for the actuator was generated by a Hewlett-Packard 3562A signal analyzer and amplified using a PCB/AVC high-power charge amplifier. A dynamic force transducer placed between the actuator and the structure measured the force applied to the structure, while a high-sensitivity accelerometer located 30 cm from the right side of the beam measured the dynamic response of the beam at the absorber location. Both the force transducer and the accelerometer signals were amplified via portable power units and then recorded by the HP signal analyzer. The analyzer was also used to process the force and acceleration signals to calculate accelerance FRFs.

Figure 12 shows the response of the system both with and without the vibration absorber attached to the clamped–clamped beam. The response of the system with no absorber attached has a prominent structural resonance at approximately 318 Hz. Clearly, the passive absorber significantly reduced the structural response in the neighborhood of the original structural resonance, and the frequency at which minimum structural response was obtained varied with the value of the electrical shunt. Also note that, because the absorber itself is lightly damped, the region of low response is bracketed by two regions of relatively

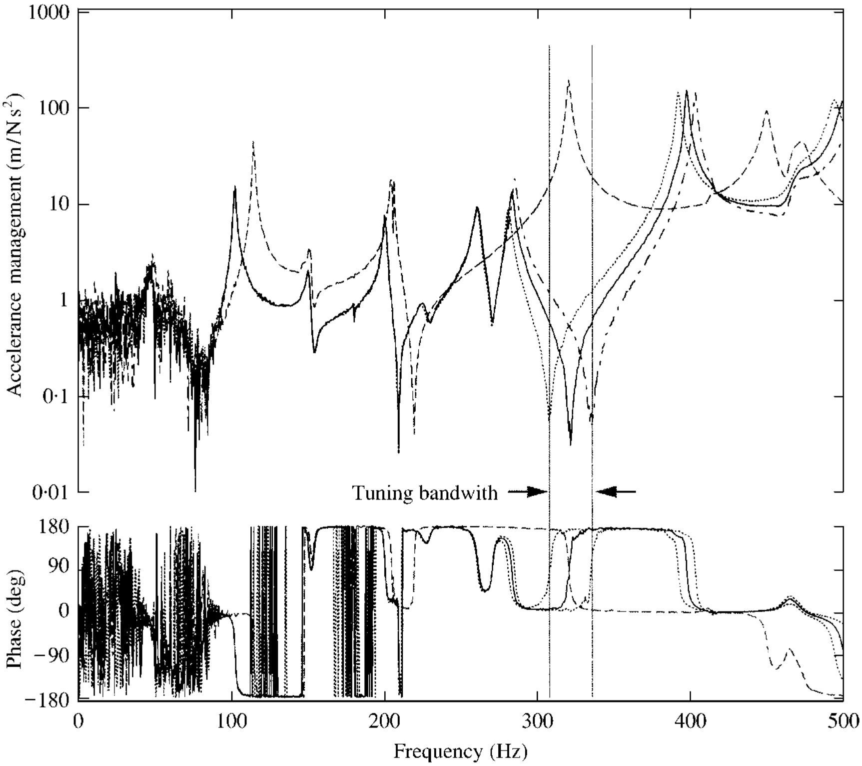

Figure 12. Accelerance for structure with passive tunable absorber. Forcing voltage 12·6 V sweep rate 312·5 Hz/s: — — —, no absorber; - - - - -, short circuit; ——, matched ($\alpha = 1$); – -- -, open circuit.

high response. This feature makes a lightly damped tunable vibration absorber most suitable for application to structures driven by variable frequency tonal disturbances.

In addition to the open- and short-circuit measurements (i.e., measurements taken for tuning ratios of 0 and ∞ respectively), the accelerance was also measured for a tuning ratio of 1·0. The response for $\alpha = 1$ defined the nominal or "control off" condition. In practice, a nominal disturbance frequency would be identified and the mass of the vibration absorber would be selected such that for $\alpha = 1$, minimal structural response would occur at the nominal disturbance frequency.

Next, the actuator was connected to the tuning circuit, which was implemented on an electronic breadboard. The breadboard was also connected to the real-time control computer to supply PIA voltage and PIA current estimates and receive shunt control voltages. The op-amps and relays used for the switching circuit were powered with Hewlett-Packard DC power supplies.

The control computer used for the experiments was a 100 MHz Dell Pentium. The Pentium housed a dSPACE DS1102 Floating-Point Controller Board with Texas Instruments TMS320P14 processor chip. The two A/D channels of the controller board had an input voltage range of ± 10 V and used 16-bit converters. The D/A channel had an output voltage range of ± 10 V and used a 12-bit converter. MATLAB's Real-Time Workshop was used to translate a SIMULINK block diagram into C-code, then to invoke the TI C-compiler. The compiled C-code was then downloaded to the processor on the controller board. The controller was designed and set to sample at 10 kHz which adequately accommodated the simulation of the analog linear filters.

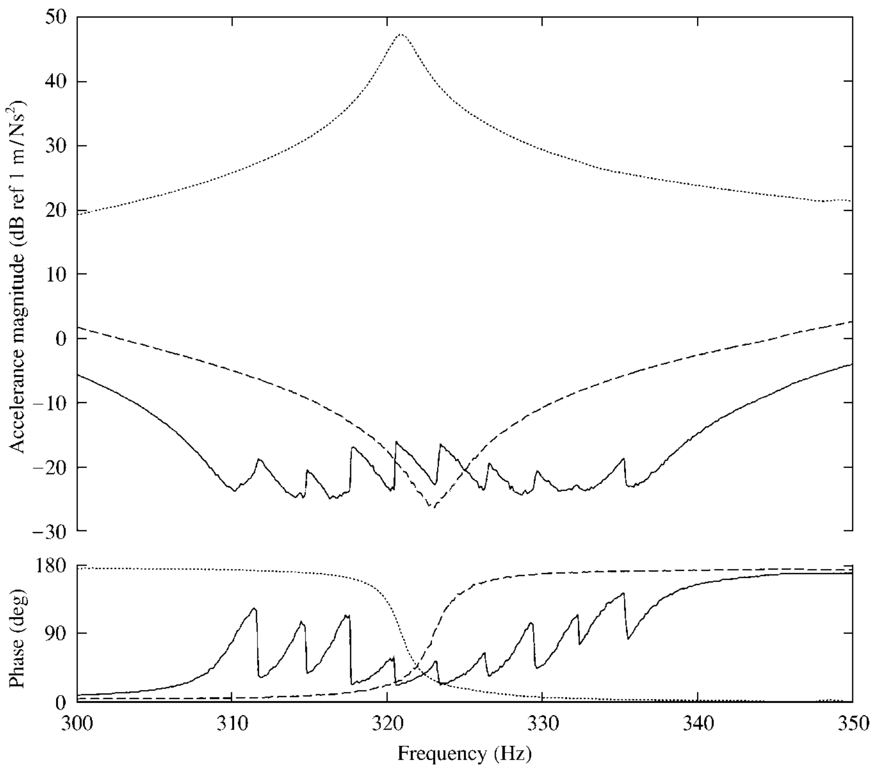

Figure 13. Actively tuned versus passive structural accelerance: - - - - -, no absorber: – – –, passive, matched ($\alpha = 1$); ——, actively tuned.

To gain a more accurate estimate of the passive and semi-active response of the system, swept sine measurements were made between 300 and 350 Hz. Note that the sweep rate for the measurements was set sufficiently low to ensure that the filters within the controller had time to settle before the analyzer moved to the next frequency in the sweep.

Figure 13 illustrates the effects of using a semi-active piezoceramic vibration absorber on structural accelerance due to a varying-frequency tonal disturbance. The dotted line in Figure 13 is the passive structural accelerance (i.e., structural response with a constant $\alpha = 1$). The solid line is the structural accelerance with the tuning controller turned on (i.e., structural response with a variable α). The changing discrete capacitances are evidently effective in increasing minimum accelerance over the previously defined tuning band. In addition, structural accelerance at frequencies below the short-circuit frequency of the absorber and above the open-circuit frequency of the absorber was improved.

Figure 14 shows the difference (in dB) between the passive and active responses. Maximum increases in performance of about 20 dB occurred at both the short- and open-circuit absorber natural frequencies. On average, an approximately 10 dB increase in performance was obtained over the frequency range shown ($\pm 7\%$ change in frequency from the center frequency of 325 Hz). A slight decrease in performance, however, was observed in a small frequency range in the center of the tuning band. This was attributed to the fact that the discretized shunt capacitance was not quite equal to the absorber capacitance measured at constant stress (i.e., $\alpha = 1$) at the tuning band center frequency.

Note that the speed of the control system was limited by the speed of the filters used in the control system. The electronics used to adjust the shunt capacitance operated very quickly

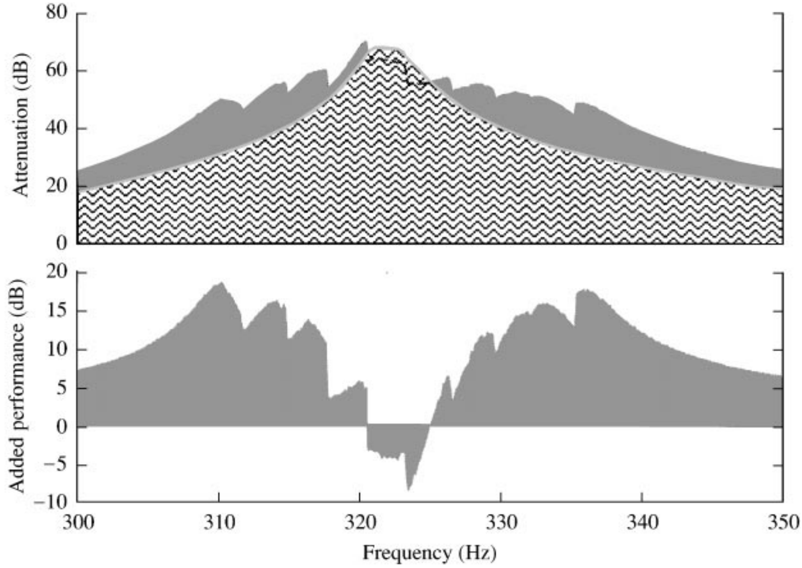

Figure 14. Actively tuned vs. passive structural attenuation performance; ■, actively tuned; ▨, passive.

compared to the filters. Thus, proper tuning was achieved as soon as the filters settled and the proper disturbance frequency was estimated. The rate of re-tuning due to a change in disturbance frequency was limited by the disturbance frequency estimation method. Small changes in disturbance frequency, less than about 11% of the tuning band, were effectively compensated in less than 60 ms. Detecting, then completely switching the absorber from short to open circuit, or *vice versa*, required less than 250 ms. A complete discussion of the effects of control speed as well as variations due to changing forcing amplitudes can be found in reference [11].

5. CONCLUSIONS

A solid-state tunable semi-active piezoceramic vibration absorber was developed. Electrically shunting a piezoelectric inertial actuator with a capacitive electrical impedance changed a fraction of the effective net stiffness of the device, thus changing the device's natural frequency. The control system used to tune the absorber monitored the voltage and current produced by the device to estimate a tonal structural vibration frequency and in turn adjust the net discrete shunt capacitance appropriately.

The semi-active vibration absorber had a $\pm 3{\cdot}7\%$ tunable frequency band relative to the center frequency. Additional attenuation effects extended beyond $\pm 7\%$ of the center frequency. Within the tuning band, increases in performance beyond passive performance were as great as 20 dB, and the average increase was over 10 dB.

This combination of tunable vibration absorber and active tuning method has several features that distinguish it from, and give it potential advantages over, others described in the literature. First, it is a piezoelectric-based device. Second, it uses capacitive shunting to accomplish an effective change in stiffness. Third, it is less complex than comparable devices, because it is completely solid state. Fourth, it requires no additional sensors, as the voltage and current signals generated in the piezoelectric elements may be used directly. Fifth, it has relatively low-power consumption relative to other PVA tuning methods; the electrical power required for shunt-circuit switching is far less than the power required for driving stepper-motors or heating viscoelastic materials. Furthermore, the tuning controller was novel in that it could be implemented as a completely solid-state analog system. Achieving this would require performing the frequency estimation and control voltage calculation in hardware instead of software. Frequency estimation could be accomplished using a phase-locked loop (PLL) circuit [11, 13]. PLLs require little power, react quickly, and are commercially available in compact integrated circuit packages. The output of the PLL is a DC voltage directly proportional to frequency, thus making the control voltage calculation largely a matter of scaling.

ACKNOWLEDGMENTS

This work was supported by the Office of Naval Research under MURI Grant N00014-96-1173 (Acoustic Transduction).

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