ACTIVE AND PASSIVE DAMPING OF STRUCTURES

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Abstract

Modern structures are often driven by design constraints to be extremely lightweight and hence very flexible and subject to increased vibration problems. In addition, improved manufacturing techniques often produce very good joints in structures reducing the amount of natural damping in structures. For instance, removing welds in bladed disc assemblies caused increase blade fatigue because of the reduced damping. As a result, active and passive damping methods are increasingly in demand. Here the basic areas of passive and active damping are reviewed and compared. Emphasis is placed on damping treatments, smart materials and applications to large flexible (inflated) space structures and automobile components.

Passive methods discussed include a summary of standard constrained layer damping treatments and piezoelectric based shunt dampers with focus on the various modeling methods and comparisons. Active methods focus on those that are obtained by using piezoelectric based materials: films, ceramics, and composites, as the sensor and actuation devices. The main example consists of a 300-meter/ 552 kg inflated satellite proposed for flight in the next 10 years called the Innovative Space Based Radar Antenna Technology (ISAT) program. This truss like structure holds a radar platform and is intended to rotate around its mid point for surveying the earth's surface. The rotation along with other maneuvering forces potentially causes large vibration interfering with the satellites ability to take measurements. Hence, active vibration means are required to remove these unwanted vibrations. Theoretical and numerical results are presented along with experimental validations of the modeling and vibration suppression methods.

INTRODUCTION

Passive damping methods have centered round adding layers of damping highly dissipative materials, basically viscoelastic in nature to metal objects to increase the damping in the total structure. The most common example of this can be found under foot in any automobile. Under the carpet, cars are treating with a layer of viscoelastic material (VEM) that serves to damp the floor vibrations and control road and other noise in the interior of the car. The more expensive the car, the more damping material is used to give the “solid feel” and quiet interior expected in luxury cars. Damping treatments have been successfully used in a wide variety of applications. Most treatments up until the mid 80’s are discussed and analyzed in [1]. Other books on passive damping focus on analysis [2, 5, 6, 9].

Active methods for providing damping have come to maturity only recently (see for instance [2,7]) and are slowly moving from the research labs to practice. Active methods involve substantially more cost and hardware then passive methods, thus applications of active methods focus largely on performance. For thin flexible structures, or structures that have broad operating environments, active damping may be the only possible solution. While active damping methods are expensive and require more complex hardware the level of performance the can be obtained by active methods is far better then passive treatments. However, because of the complexity and cost, passive methods should be tried first and if they fail to meet the required performance then active methods must be used.

In the following a brief summary of passive methods is presented followed by an introduction to active methods using piezoceramic based methods with applications to satellite systems.

PASSIVE DAMPING TREATMENTS

Passive damping treatments have consisted of using either free layer or constrained layer VEM to extract energy from a host structure. More recently, researchers have extended these VEM layer concepts to included electrical shunts, which also extract energy from a host structure. These are summarized in Figure 1 and reviewed here. Of course various combinations of these have also been considered in the literature. In a typical example of application of a VEM damping treatment a metal is covered (totally or partially) with a think layer of VEM. This is called *free layer damping treatment*, and although not free is the cheapest of the passive methods. As the metal bends in the transverse direction, the VEM extends causing it to dissipate energy through heat.

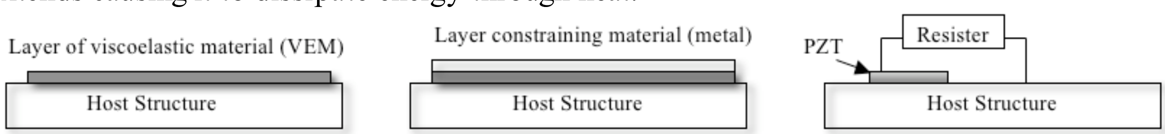

Figure 1 From left to right, a free layer treatment, a constrained layer treatment and a shunt treatment.

Constrained layer treatments improve upon free layer treatments by applying a thin yet stiff (usually metal) layer on top of the VEM to force the VEM into shear as the host structure bends. This is illustrated in the middle of Figure 1. The shear motion in the VEM dissipates even more energy to heat, but at the added cost (both mass and Euros) of the constraining layer. Free layer and constrained layer damping treatments are a mature industry and have been used for decades in numerous products (see Nashif et al [5]).

The majority of applications employ add on, or designed in VEM, however other materials are also used. Materials used in passive damping treatments that exhibit a viscoelastic behavior are polymers, rubber, pressure sensitive adhesives, urethanes, epoxies and enamels. Adding these materials to a structure or material system improves the vibration response by

- Reducing the resonant peak response
- Reducing the settling time of the response
- Reducing noise transmission
- reducing the rattle space required for isolation

Shunting (depicted on the far right in Figure 1) uses the piezoelectric effect, which converts strain into an electric field) to add damping to structures. An excellent summary is provided by Lesieutre [4]. The basic concept is that as the host structure vibrates, the piezoelectric material (usually a ceramic such as PZT) stains inducing a voltage, which appears across a resistor shunted in series with the PZT's voltage source. The resistor then dissipates the energy as heat, causing a passive damping effect. The piezoelectric material is usually modeled as voltage source and a capacitor. A variety of electrical shunt circuits are then used to dissipate energy given a variety of different behaviors. The resistive shunt dissipates energy through Joule heating and behaves very much like a VEM. The difference between a VEM layer and a resistive shunt is that the resistive shunt is not as effective per added mass as the VEM layer but is also not as prone to performance degradation due to changes in ambient temperature. If an inductive and capacitive circuit is used as a shunt (LC circuit), the added damping behaves like a tuned mass damper. Compared to constrained layer damping treatments, the LC shunt adds much more damping, but only at a single frequency. Again, the shunt is much more robust to temperature changes then standard VEM treatments are.

Other "smart materials" have been investigated for providing passive damping. These are summarized by Baz [1]. These involve using the dissipative nature of shape memory alloys, magnetic fields and magnetic constrained layer damping methods. Sodano, et al [10] used Eddy Currents to provide damping. Other methods available include particle dampers (random motion of particles), stand off damping treatments and vibration absorbers. These approaches are not discussed here but only mentioned for completeness.

ACTIVE DAMPING TREATMENTS

Active damping can be accomplished by almost any type of control actuator coupled with a control law that involves velocity, such as simple velocity feedback. If one thinks of the model of a structure as the simple linear system

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t)$$

where x is the displacement and over dots denote time derivatives, then passive damping only has the potential to effect the relative magnitude of the damping matrix, C. However, active damping treatments can accomplish much more. For example active methods can introduce new degrees of freedom (such as the vibration absorber does) to help dissipate energy. To make this a little clearer, consider a method known as Positive Position Feedback. Consider a single degree of freedom system (alternatively a single decoupled mode of the system) defined as

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = bu. \tag{1}$$

where ζ and ω_n are the damping ratio and natural frequency of the structure, and b is the input coefficient that determines the level of force applied to the mode of interest. The PPF controller is implemented using an auxiliary dynamical system defines as (see for instance [2,7]

$$\begin{aligned} \ddot{\eta} + 2\zeta_f\omega_f\dot{\eta} + \omega_f^2\eta &= g\omega_f^2 x \\ u &= \frac{g}{b}\omega_f^2\eta \end{aligned} \tag{2}$$

where ζ_f and ω_f are the damping ratio and natural frequency of the controller (electronically determined) and g is a constant. These parameters are left to the designer to manipulate and obtain the best controller possible. Combining equations (1) and (2) and placing them in their second order form assuming no external forces gives us

$$\begin{bmatrix} \ddot{x} \\ \ddot{\eta} \end{bmatrix} + \begin{bmatrix} 2\zeta\omega_n & 0 \\ 0 & 2\zeta_f\omega_f \end{bmatrix}\begin{bmatrix} \dot{x} \\ \dot{\eta} \end{bmatrix} + \begin{bmatrix} \omega_n^2 & -g\omega_f^2 \\ -g\omega_f^2 & \omega_f^2 \end{bmatrix}\begin{bmatrix} x \\ \eta \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}. \tag{3}$$

The form of equation (3) illustrates how this chose of control law introduces damping into the structure electronically. The filter damping is controlled electronically by the choice of ζ_f, which in turn is coupled to the structure through the effective stiffness matrix. This form of control allows much more damping to be introduced then passive methods or simple velocity feedback because of the extra coordinate (η) that also absorbs energy.

One problem with feedback control is that the natural stability of the system is potentially destroyed. Thus, a stability analysis of any control law must be performed. Recall that a conservative system defined by

$$M\ddot{\mathbf{q}} + K\mathbf{q} = 0, \tag{4}$$

where M and K are symmetric, the system is stable if M and K are positive definite. This is simply because the eigenvalues of K are positive and hence the eigenvalues of the system are purely imaginary. Thus, the system is stable since the response of such system is always bounded by a constant.

One can also define a Liapunov function for the system in (4) as

$$V(\mathbf{q}) = \frac{1}{2}\left[\dot{\mathbf{q}}^T M\dot{\mathbf{q}} + \mathbf{q}^T K\mathbf{q}\right]. \tag{5}$$

Equation (5) is readily identified as the energy in the system. Stability can now be defined by the following conditions:

$$V(\mathbf{q}) > 0 \text{ for all values of } \mathbf{q}(t) \neq 0$$
$$\dot{V}(\mathbf{q}) \leq 0 \text{ for all values of } \mathbf{q}(t) \neq 0$$

If $\dot{V}(\mathbf{q})$ is strictly less the zero the system is asymptotically stable. One can see now that if M and K are positive definite $V(\mathbf{q})>0$, and thus the first condition is met. For the second condition the time derivative of (5) is taken resulting in

$$\frac{d}{dt}V(\mathbf{q})=\dot{\mathbf{q}}^T M\ddot{\mathbf{q}}+\dot{\mathbf{q}}^T K\mathbf{q}. \tag{6}$$

Multiplying equation (4) on the left by $\dot{\mathbf{q}}^T$ gives us

$$\dot{\mathbf{q}}^T M\ddot{\mathbf{q}}+\dot{\mathbf{q}}^T K\mathbf{q}=0. \tag{7}$$

Hence the second condition for the Liapunov function is $\dot{V}(\mathbf{q})=0$, and thus the equilibrium of equation (4) is stable.

Adding damping to the system in (4) leads to

$$M\ddot{\mathbf{q}}+C\dot{\mathbf{q}}+K\mathbf{q}=0. \tag{8}$$

$V(\mathbf{q})$ as defined by (5) is still a Liapunov function of (8). In this case the solution $\mathbf{q}(t)$ must satisfy

$$\dot{\mathbf{q}}^T M\ddot{\mathbf{q}}+\dot{\mathbf{q}}^T K\mathbf{q}=-\left(\dot{\mathbf{q}}^T C\dot{\mathbf{q}}\right) \tag{9}$$

which comes from premultiplying (8) by $\dot{\mathbf{q}}^T$. From this it can be seen that the time derivative of the Liapunov function is

$$\frac{d}{dt}V(\mathbf{q})=-\left(\dot{\mathbf{q}}^T C\dot{\mathbf{q}}\right). \tag{10}$$

Observing (10) shows that if C is positive definite $\dot{V}(\mathbf{q})<0$ and the system in (8) is asymptotically stable. If C is positive semidefinite the above argument is still valid and the system is stable. However, it is not clear if the system is also asymptotically stable (see [2] for stability conditions for semi definite damping).

Referring back to the system specified in (3) it can be said that this is a stable (asymptotically stable) closed loop system if the symmetric "stiffness" matrix is positive definite for appropriate choices of g and ω_f. That is if the determinant of displacement coefficient matrix is positive which happens if

$$g^2\omega_f^2<\omega_n^2.$$

In the case of the PPF the stability conditions depend only on the natural frequency of the system and not on the damping or mode shapes. This is good as natural frequency is the most accurate measurement of these three. Hence closed loop stability can be determined in terms of only measured quantities.

Looking at the transfer function of the controller, in equation (11), one can see that it rolls off at high frequencies avoiding spillover into higher modes, having the characteristics of a low pass filter. The controller transfer function is

$$\frac{\eta(s)}{X(s)} = \frac{g\omega_f}{s^2 + 2\zeta_f\omega_f s + \omega_f^2} . \tag{11}$$

Thus the approach is well suited to control modes that are well separated as the controller is insensitive to higher un-modeled dynamics.

In application the feedback signal can not always be chosen and it is readily determined by what is available or what is more feasible as sensors. Thus, much in the same way one can define a Positive Velocity Feedback (PVF) controller [11]. In this case the controller is defined as

$$\begin{aligned} \ddot{\eta} + 2\zeta_f\omega_f\dot{\eta} + \omega_f^2\eta &= g\omega_f\dot{x} \\ u &= \frac{g}{b}\omega_f\dot{\eta} \end{aligned} . \tag{12}$$

Combining equations (1) and (12) and placing them in their second order form assuming no external forces gives us

$$\begin{bmatrix} \ddot{x} \\ \ddot{\eta} \end{bmatrix} + \begin{bmatrix} 2\zeta\omega_n & -g\omega_f \\ -g\omega_f & 2\zeta_f\omega_f \end{bmatrix}\begin{bmatrix} \dot{x} \\ \dot{\eta} \end{bmatrix} + \begin{bmatrix} \omega_n^2 & 0 \\ 0 & \omega_f^2 \end{bmatrix}\begin{bmatrix} x \\ \eta \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{13}$$

It is seen that in this case the stability conditions have moved to the damping matrix. Because of the velocity feedback no longer is the stiffness matrix affected by the controller. In this case the stability conditions are more relaxed, since the condition for stability requires that the damping matrix be positive semidefinite [2]. Looking at the determinate of the velocity coefficient matrix the following condition arises for stability

$$\zeta\omega_n \geq \frac{g^2\omega_f}{4\zeta_f} . \tag{14}$$

In the case that it is strictly greater than (14) it guarantees asymptotical stability.

The PVF controller also has the roll off characteristic as that of the PPF. Its roll off though is not as fast as that of the PPF, which can be seen by observing the transfer function for PVF:

$$\frac{\eta(s)}{X(s)} = \frac{g\omega_f^2 s}{s^2 + 2\zeta_f\omega_f s + \omega_f^2} . \tag{15}$$

The slower roll off is caused by the s term in the numerator of the transfer function consequential of the velocity feedback.

Again in the similar manner as for the PVF we can design a Positive Acceleration Feedback (PAV) controller as follows,

$$\begin{gathered}\ddot{\eta}+2\zeta_f\omega_f\dot{\eta}+\omega_f^2\eta=g\ddot{x}\\ u=\frac{g}{b}\ddot{\eta}\end{gathered}. \tag{16}$$

Combining equations (1) and (16) and placing them in their second order form assuming no external forces gives us

$$\begin{bmatrix}1 & -g\\ -g & 1\end{bmatrix}\begin{bmatrix}\ddot{x}\\ \ddot{\eta}\end{bmatrix}+\begin{bmatrix}2\zeta\omega_n & 0\\ 0 & 2\zeta_f\omega_f\end{bmatrix}\begin{bmatrix}\dot{x}\\ \dot{\eta}\end{bmatrix}+\begin{bmatrix}\omega_n^2 & 0\\ 0 & \omega_f^2\end{bmatrix}\begin{bmatrix}x\\ \eta\end{bmatrix}=\begin{bmatrix}0\\ 0\end{bmatrix}. \tag{17}$$

In this case it is required that the mass matrix be positive definite for stability (asymptotical stability). That leaves the condition that the gain g must be chosen with the following condition for stability

$$g^2<1. \tag{18}$$

Unlike the other two controllers this one those not roll off. This can be seen by looking at the transfer function of the controller

$$\frac{\eta(s)}{X(s)}=\frac{gs^2}{s^2+2\zeta_f\omega_f s+\omega_f^2}. \tag{19}$$

This effect needs to be taken into consideration when using this type of controller since it can disturb the higher modes. Figure 2 shows the frequency response of a 2^{nd} order plant model (solid line) and the controlled closed loop plant using the three methods described above. As we can see the PAF does not roll off. With the design of such controllers we have covered a wide range of sensors that could possibly be used for feedback for the purpose of adding damping.

Another effective control law for adding damping to a structure is to use optimal control as a means to extract energy. Basically optimal control again uses measurements of the system out put to feed back to the input, as in PPF control, but this time the control law is chosen to minimize the energy in the time response. The method proceeds by forming the equations of motion (4) in the state space by defining the vector $\mathbf{x}=\begin{bmatrix}\mathbf{q}^T & \dot{\mathbf{q}}^T\end{bmatrix}$. This yields

$$\dot{\mathbf{x}}=A\mathbf{x}+B\mathbf{u},\quad \mathbf{y}=C^T\mathbf{x},\quad \mathbf{u}=-K\mathbf{x}$$

where $\mathbf{u}$ is chosen to minimize the function

$$J = \int_0^\infty (\mathbf{x}^T Q \mathbf{x} + \mathbf{u}^T \mathbf{u})dt, \quad Q = \begin{bmatrix} K & 0 \\ 0 & M \end{bmatrix}$$

This is know as the LQR problem and is a standard optimal control method (see for instance [2,7]). Each of these two control approaches are used in the following example.

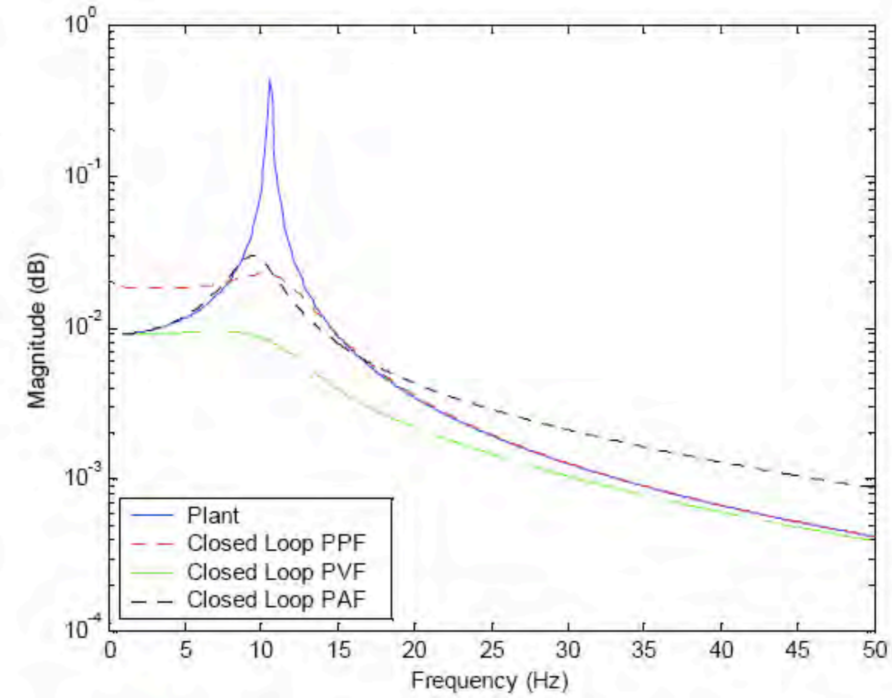

Figure 2 Transfer function of a 2nd order plant and the respective close loop transfer functions using the different controllers.

APPLICATION OF ACTIVE DAMPING FOR FLEXIBLE SATELLITES

Innovative Space Based Radar Antenna Technology (ISAT) program is intended to enable nonstop global surveillance of moving ground targets. These satellites (Fig. 1) are designed to operate in a medium earth orbit (MEO) to provide improved coverage when compared to low-orbiting satellites. Such technology will require fewer satellites for global coverage and thus reduce the overall cost involved. However, a radar antenna operating at MEO must be so large that it cannot be launched aboard existing rockets. Space inflatable structures, which can be compressed into far smaller packages, provide a viable means of placing large metrology systems in space. Inflatable structures have many advantages compared to mechanically deployed structures, such as lower weight, higher packaging efficiency, and easier maintenance. Such structures have a long history of use in aerospace applications, from the ECHO series of satellites in the 1960’s to the space shuttle launched Inflatable Antenna Experiment (IAE) in May 1996.

The problem of concern here is to damp the vibrations of an ISAT like structure. Vibrations are induced into the satellite by slewing maneuvers as well as thrusting, space debris and thermal moments. Figure 3 shows a schematic of the proposed 300-meter IAST.

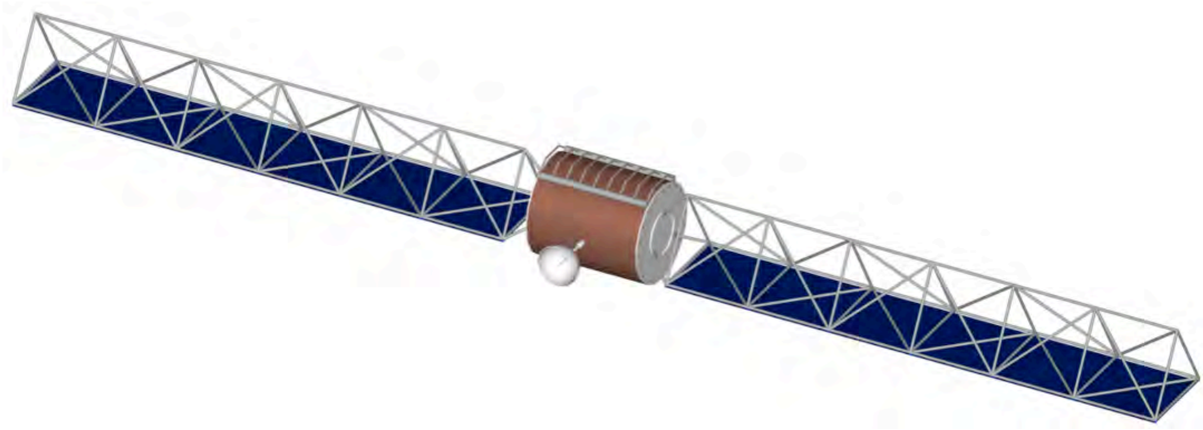

Figure 3 A schematic of the ISAT 300 meter satellite

Two approaches to adding damping to this structure are taken. The first is to use the PPF approach with integrated macro fiber composite actuators to add damping to each truss element. The second (global) approach is to add damping to the entire system by using optimal control to minimize the energy of the time response of the structure as certain key points along the structure.

First consider controlling the vibration of each individual component of the truss in Figure 3. The hardware used to implement a PPV controller on a single element of the repeated lattice structure is given in figure 4. The basic idea is to use an integrated macro fiber composite actuator (MFC) embedded in a single tube, coupled with a PPV control law to perform vibration suppression.

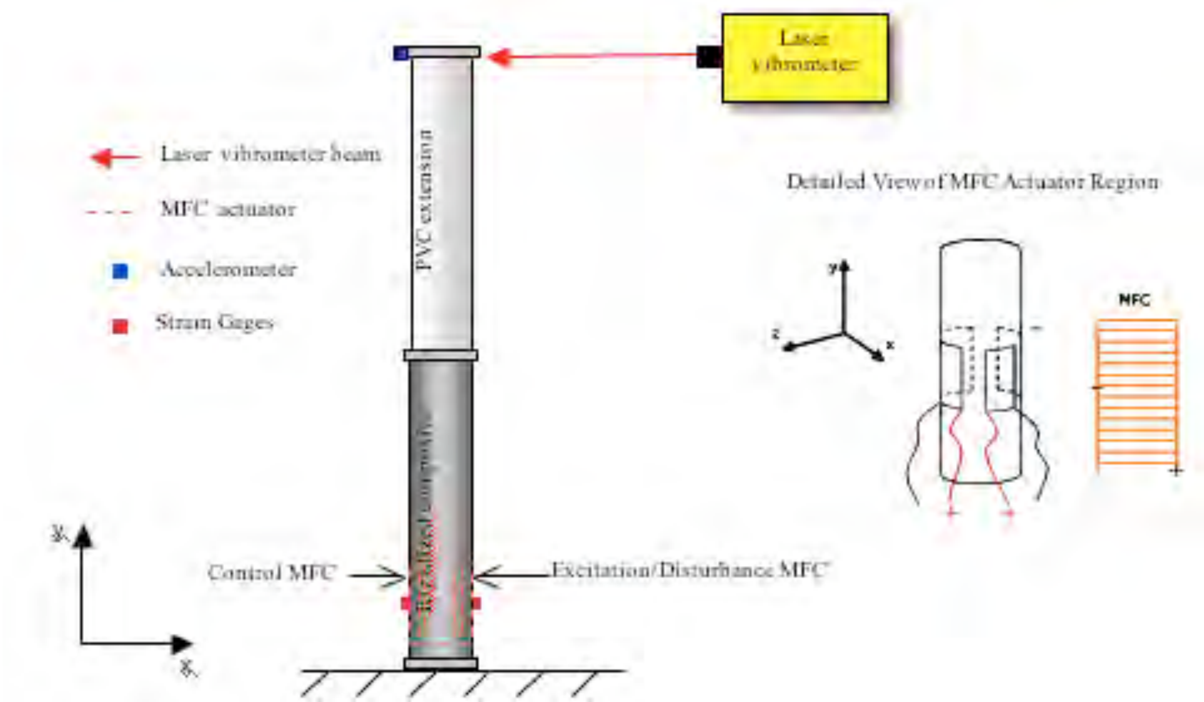

Figure 4 The hardware for implementing PPV control of a single tube of the ISAT

The control results are illustrated in Figure 5. The tube at the top simulated the mass of the remaining structure. A clamped boundary condition at the base was used to simulate connection to the rigid hub. The primary control design objective was to attenuate the vibratory response of the first boom-bending mode. Extension of this technique to multiple modes is simple and follows the same idea with cascading filters.

Feedback controllers were implemented using a dSPACE real-time digital control system. A MATLAB/Simulink-based front end was used for control design and programming. Controller gains using velocity proportional, acceleration proportional, and displacement (strain) proportional feedback signals were modified in real-time in order to optimize closed-loop performance. For each control signal case, the controller parameters where modified to obtain the best attenuation possible of the first mode. The controller parameters were control frequency (ω_f), control damping (ζ_f), and control gain (g). The optimal choices for these parameters were determined by observing the experimental frequency response between the excitation MFC input signal and output sensor while manually varying the controller parameters.

The first case examined utilized the laser vibrometer to provide a velocity-proportional feedback signal. A very large, 23 *dB* attenuation in the first bending mode response is demonstrated using the optimized controller parameters.

After determining the optimal controller in the frequency domain, its effectiveness was evaluated in the time domain. A sinusoidal disturbance at 10.5 Hz was applied to the structure with the excitation MFC. The laser vibrometer was used to read the response of the structure at the top of the boom. Figure 5 shows the time response before and after the controller was engaged. A 93% reduction in the response amplitude at the tip of the boom is shown.

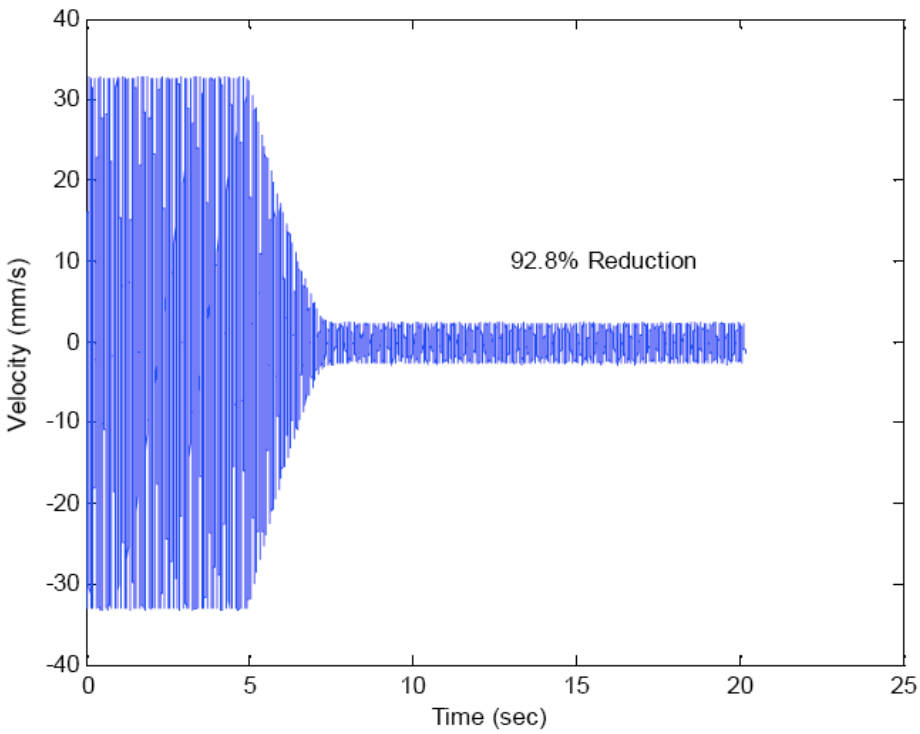

Figure 5 The closed loop response of the tip using PVF control

Next consider applying the optimal control, minimum energy method to controlling the vibrations of the entire assembled truss. This control must necessarily be numerical because the truss will not be assembled on the ground, but only in space. The method of modeling involves using and ad hoc homogenization procedure [8] to write a Timoshenko beam model of the entire 300 meter system. Then, a low order controller is built around the first few significant modes of the system. An LQR controller is then constructed and applied to the numerical model, the results are shown in the time domain in Figure 6.

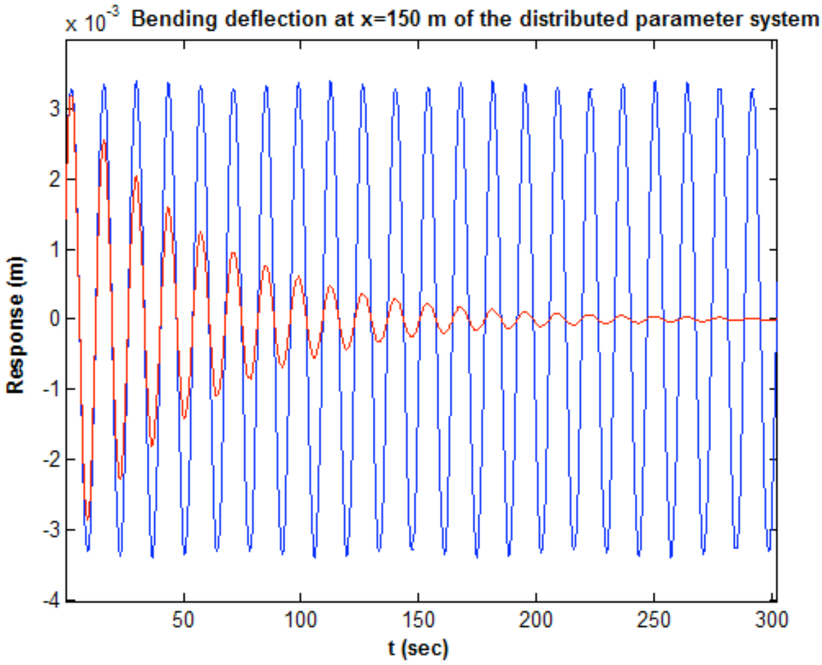

Figure 6 The closed loop and open loop response of the ISAT model at the 150 m point.

SUMMARY

Damping treatments for structures have been summarized. Two approached to active damping have been presented: positive velocity feedback and optimal control. Both of these methods are used to introduce damping into a large flexible structure. The PVF method is used to introduce damping into each element of the truss and optimal control is used to introduce damping into the entire system's global modes. The local PVF damping is verified experimentally. Active damping is chosen over passive damping in this case because of the potential for thermal disturbances and the need to compensate for a variety of inputs.

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