Model-Independent Vibration Control of Flexible Beam-Like Structures Using a Fuzzy Based Adaptation Strategy

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ABSTRACT: The present study deals with an AFCA (Adaptive Fuzzy Control Algorithm) for an Euler-Bernoulli approximation of a two-dimensional version of a cantilever beam-like orthogonal tetrahedral space truss. Transient disturbances, modeled as a unit impulse, excite all the modes of the beam. The resulting transverse displacement at the free end of the beam and its corresponding rate are observed by sensors placed there, and active control of the beam is provided by a collocated force actuator.

A design methodology for the closed-loop control algorithm that is independent of an exact mathematical model (space-state model, F.E.M., etc.) of plant dynamics and which is based on fuzzy logic is presented. First, the behavior of the open-loop system is observed. Then, the control force applied to the system emulates the behavior of a dynamic vibration absorber which is tuned to the measured fundamental frequency. This approach not only assures inherent stability associated with passive absorbers, but also circumvents the phenomenon of modal spillover. The damping and the mass ratios of the absorber adapt themselves by using a fuzzy decision-making process. This results in relatively quick settling times, low overshoots and dying out of vibration within a few seconds.

When the control force is turned off after a mere 16 seconds, almost all the vibrational energy is dissipated. In addition, the performance of the AFCA is insensitive to varying initial conditions. To demonstrate the robustness of the control system to changes in the temporal dynamics of the cantilever beam, the transient response to a considerably perturbed plant is simulated. The Young's modulus of the beam was raised as well as lowered substantially, thereby significantly perturbing the natural frequencies of vibration. The mode shapes, however, remain unchanged. For these cases, too, the AFCA provides similar settling times and rates of vibrational energy dissipation.

INTRODUCTION

Background

FUTURE aerospace applications include concepts such as large space stations, high resolution radar and communication antennas, astronomical observatories, solar power stations, and very large span high altitude long endurance unmanned aircraft. Qualifying as L.F.S. (Large Flexible Structures) these facilities may generally be comprised of repetitive latticed trusses, span large areas with a few intermediate supports, are light in weight and extremely flexible, and consequently are characterized by a large number of high density low frequency structural modes.

The L.F.S. may be characterized by low inertia, light inherent damping, undamped rigid modes, low natural frequencies, high modal density, some joint non-linearity, and includes sensors, actuators and computers, on-board power and a laser pointing system. The dynamic characteristics of the above are not well known and therefore make the analytical modeling of the structural dynamic problem of L.F.S. cumbersome with substantial uncertainties. Hence, it would be advantageous to develop control design strategies that literally require no a priori knowledge on the plant temporal model.

L.F.S. may need to meet target tracking, slewing, stringent line-of-sight and jitter control, pointing accuracies, and micrograsity acceleration requirements. However, when disturbed, the structure is likely to remain excited for some time because of its high structural modal density at low frequencies and possibly small damping. Therefore, it is vital to introduce means for passive energy dissipation, active control, or their combination to restrain the response of a given structure within an "in-mission displacement-time allowables envelope" by using vibration control methods.

Problem Definition

Following the detailed literature survey given in Cohen and Weller (1990), the problem of undesirable vibrational energy in large, flexible, beam-like repetitive lattices of space structures may be minimized by using passive means in the form of dynamic vibration absorbers for augmenting dissipation. Later, Cohen and Weller (1994) presented a procedure for obtaining optimal tuning and damping parameters of several absorbers that are applied to a cantilevered truss in order to minimize the settling time of the transient, transverse displacement response. These passive methods have proved to be simple, reliable, robust, and inherently stable.

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However, the disadvantages of model-dependent (knowledge of plant frequencies and mode shapes) absorbers lie in their considerable weight penalty, and their effectiveness is restricted to a narrow frequency band. In addition, their performance is limited to that obtained when using a linear feedback controller (mass, stiffness and damping parameters of an individual absorber are constant).

Approach and Scope of Investigation

To circumvent the above drawbacks, the absorber in the present effort is replaced by discrete, collocated actuators/sensors that actively control structural vibrations of the L.F.S. The feedback control law that commands the actuators emulates the behavior of a virtual dynamic vibration absorber tuned to the fundamental frequency of the L.F.S., which is easily obtained from the system sensor output of the displacement-time history. Ideally speaking, the parameters of the virtual absorber may be varied to provide quick rise times (low damping), for large deviations of the measured state of the plant from the desired state, and sluggish control (high damping) for small deviations. Thus, non-linear control actions, corresponding to a *lightly* damped absorber with a *large* mass ratio, which fully utilize the range of actuator displacements, send the plant state hurtling towards the desired state. On the other hand, in the vicinity of this desired state, the emulated virtual absorber is *heavily* damped, having a *small* mass ratio. Heuristic rules, based on basic "common sense" engineering insight, coupled with fuzzy reasoning provide crisp values for *lightly, large, heavily,* and *small.* The development of the adaptive control strategy which is model-independent, fuzzy, inherently stable, is presented in the following sections. Closed-loop performance and robustness, with respect to changes in the temporal dynamics, is examined.

Literature Survey

Robustness of a L.F.S control in the presence of uncertainties has been an area of intense research. One such method, based on the *positivity design* assures that the closed loop system of a L.F.S. having *collocated* sensors and actuators may be characterized by stability as well as an energy dissipation related to the input/output behavior (Benhabib, Iwens and Jackson, 1981). Hyland (1993) describes such an energy dissipative law that combines collocated actuator and sensor pairs as electromagnetically emulating passive structural damping.

Hollkamp and Starchville (1994) present a piezoelectric actuator that emulates an inherently stable dynamic vibration absorber. In addition, it was experimentally shown that the self-tuning piezoelectric absorber is made adaptive by tuning the electric resonance (i.e., by adjusting the shunt inductance and resistance). Within the past decade, experiments conducted on flexible structures validated the shunted piezoelectric models which predicted the optimal tuning parameters (Hagood and Von Flotow, 1991). Jaung and Phan (1992) developed a robust controller for L.F.S. which is based on a virtual passive approach and observe some of the basic properties of a Dynamic Vibration Absorber (DVA). These are:

- When a DVA is attached to any mechanical system, including L.F.S., the damping of the system is almost always augmented *regardless of system size.*
- The parameters of the DVA are arbitrary, model-independent and thus insensitive to system uncertainties.
- *No matter what happens,* the DVA will not destabilize the system because it is an energy dissipative device.

The approach proposed in here introduces for the first time a treatment, which is based on fuzzy logic control and provides continuous tuning of the parameters of the above described emulated absorber, for vibration control of L.F.S. subject to transient disturbances. The parameters of the above absorber may be adapted to provide fairly fast control for large deviations of the measured state of the plant from the desired state, and a minor amount of control for small deviations. The main advantages of using a fuzzy approach are the relative ease and simplicity of implementation and the robustness characteristics. The input to the control law is based on sensor readings of displacement/distortion, and their corresponding rates.

A philosophy similar to the above one was applied by Heckenthaler and Engell (1994) to develop a fuzzy, nonlinear control law which achieves robust, near minimum-time control of the water-level in a laboratory two-tank system. An overview of the key points in applying fuzzy control to various industrial applications is provided by Sugeno (1985). Schwartz et al. (1994) survey recent applications of fuzzy sets and approximate reasoning.

Meyer, Burke and Hubbard (1993) use velocity feedback with positive-definite feedback gain and collocated piezoelectric transducers in a control methodology integrating sliding mode control, distributed parameter systems theory, and fuzzy logic to develop vibration damping of a cantilever beam. A frequency-shaped LQR adaptive control scheme, based on a priori knowledge of the intervals of system-parameter variations and fuzzy-logic, is applied for vibration suppression of a cantilever beam using collocated piezoceramic transducers (Zeinoun and Khorrami, 1994).

CONTROLLER DEVELOPMENT

This section deals with the development of the fuzzy controller that emulates the behavior of an adaptive, virtual, dynamic vibrating absorber. An Euler-Bernoulli continuum model is used to approximate a two-dimensional version of the orthogonal tetrahedral lattice configuration in Figure 1. Such trusses are typical of large space structures (Necib and Sun, 1989).

The structure is subjected to transient disturbances, which are modeled as initial conditions corresponding to unit im-

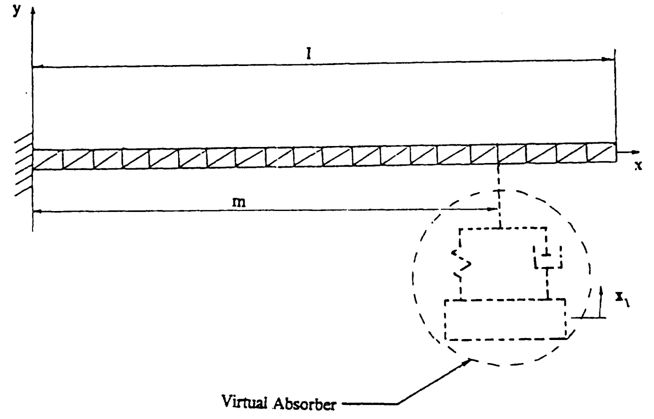

Figure 1. *A 2-D version of a beam-like lattice with an absorber attached.*

pulses. These kinds of disturbances may simulate shuttle docking or crew motion for a L.F.S. (Sesak, Gronet and Marinos, 1987). As a result of the unit impulse, all the modes of the beam are excited and in the absence of structural damping the L.F.S. will continue to vibrate until means of vibration control are introduced. Therefore, the controller will have to suppress all of the modes that make any significant contribution to the vibration in order to bring the beam to the prescribed state.

A major issue that arises in the design of L.F.S. concerns the placement and number of sensors/actuators. Obviously, for increased effectiveness, a minimum number of actuators and sensors is sought that will enable the control and observation of all modes that make a significant contribution with respect to a given performance index. In principle, for a given number of transducers, the global optimum of all possible candidate locations may be evaluated. However, this exhaustive search is cumbersome and often computationally infeasible. Cohen and Weller (1994) suggest a heuristically based iterative procedure that enables the selection of placing/number/tuning of DVA's for the vibration suppression of large beam-like truss structures. The lessons learned, which are incorporated into this research, are as follows:

- One DVA, tuned to a specific targeted mode, is sufficient for the vibration suppression of that particular mode. In addition, the DVA may also be effective in absorbing energy from neighboring modes for a L.F.S. having a high modal density.
- A DVA tuned to a certain frequency does not affect the characteristics of the modes above that frequency. Hence, after identifying the targeted modes, the highest mode is first tuned and then the structure is modified. In the next step, the second highest mode is tuned and so on (see details in flowchart presented in Figure 6, Cohen and Weller, 1994).
- The energy absorbing property of the tuned DVA is most effective when placed at the maxima of the respective mode shape.

Let us define a virtual dynamic vibration absorber (Figure 1) placed at a distance m from the fixed end of the structure. For the sake of convenience, we denote the natural frequency of an undamped vibration of the absorber as $\sqrt{a}$. Based on the optimal tuning ratio of a 2-DOF system, for a given open-loop frequency of vibration ω and a mass ratio of the absorber μ, a is obtained by equating the first two peaks of the steady state response (Den Hartog, 1956), and given by:

$$a = \left(\frac{\omega}{1+\mu}\right)^2 \tag{1}$$

Jacquot (1978) extended Den Harton's 2-DOF theory to continuous systems (a dynamic vibration absorber was applied at a distance m from the origin to an Euler-Bernoulli Beam), thereby transforming Equation (1) to:

$$a = \left[\frac{\omega}{1+(\phi_i(m))^2 \cdot \mu}\right]^2 \tag{2}$$

where ϕ_i is the value of the normalized mode shape of the natural mode i at a distance m from the origin. Following Equation (2), the only plant information required to define the stiffness of an optimally tuned absorber is ω (easily identified from the displacement-time sensor output of the open-loop system) and ϕ_i. *However, it should be noted that here we are interested in developing a model-independent controller and therefore it is assumed that no knowledge of the mode shape exists.* Consequently, we introduce a constant, μ^* which is used in the tuning process to make up for the loss in performance due to the lack of information on mode shapes. Therefore, the product of the mass ratio of the absorber, μ, and ϕ_i^2 may be written as a function of the damping factor, δ, and μ^* as follows:

$$\mu \cdot [\phi_i(m)]^2 = \mu^* \cdot \left[\frac{1}{\delta} + \left(\frac{1}{\delta}\right)^2\right] \tag{3}$$

Equation (3) presents an empirical relation, whereby a *lightly* damped absorber corresponds to a *large* mass ratio and vice-versa. Inserting the above equation into Equation (2) gives:

$$a = \left[\frac{\omega \cdot \delta^2}{\delta^2 + \mu^* \cdot (1+\delta)}\right]^2 \tag{4}$$

Let the damping parameter, b, be defined as the ratio of the damping coefficient of the absorber to the mass of the absorber, for a typical second-order system, written as (Meirovitch, 1986):

$$b = 2 \cdot \left(\delta \cdot \sqrt{a}\right) \tag{5}$$

Inserting Equation (4) into Equation (5) yields:

$$b = 2 \cdot \delta \cdot \left[\frac{\omega \cdot \delta^2}{\delta^2 + \mu^* \cdot (1+\delta)} \right]^2 \qquad (6)$$

The resulting force applied by the absorber may be written as:

$$f = a \cdot [x_i - y(m,t)] + b \cdot \left[\frac{dx_1}{dt} - \frac{dy(m,t)}{dt} \right] \qquad (7)$$

where $y(m,t)$ and x_1 are the transverse displacements of the beam and virtual absorber respectively. As shown in Figure 1, x_1 is obtained by solving the following 2-DOF equation of motion of the absorber.

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -a & -b \end{bmatrix} \cdot \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ a & b \end{bmatrix} \cdot \begin{bmatrix} y(m,t) \\ dy(m,t)/dt \end{bmatrix} \qquad (8)$$

The approach employed herein is based on MRAC [Model-Reference Adaptive Control, (Friedland, 1993)]. This method may be used in control of systems whose models possess significant uncertainties, and hence the application of a model-independent control law is a distinct advantage.

As shown in Figure 2a, first, the reference inputs are applied and the performance errors, ε_y and ε_y, become the input to a tuner which adjusts the damping factor of the virtual dynamic vibration absorber. The value of δ inferred from the measurements of $y(m,t)$ and $y(m,t)$ is reached by using a fuzzy control gain weighting adaptation strategy, which will be described in the following section. In the next step the values of the parameters, which characterize the absorber, $a(\delta)$, $b(\delta)$, and the product $\phi_i^2 \cdot \mu(\delta)$ are found using Equations

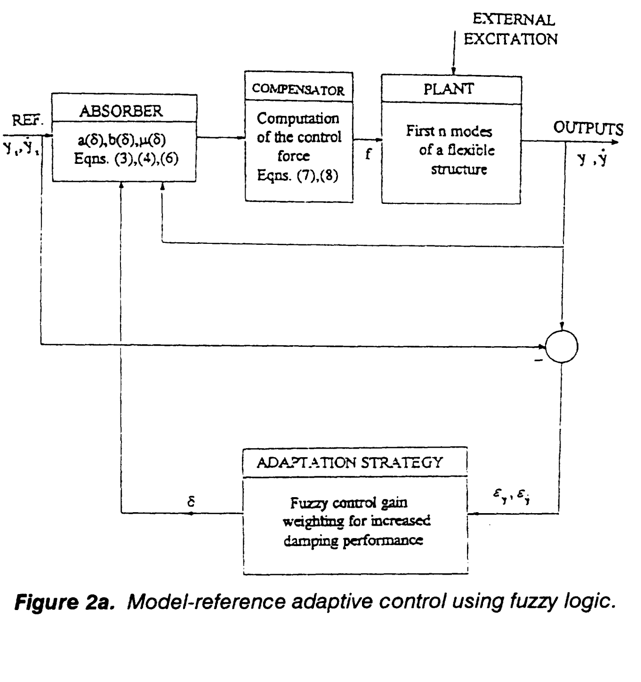

Figure 2a. *Model-reference adaptive control using fuzzy logic.*

(3), (4) and (6). Then, the 2-DOF equations of motion of the absorber, given by Equation (8), are numerically computed to give the displacement and the velocity of the virtual absorber. Finally, the input control force, obtained from Equation (7), is applied to the plant, which consists of the first n modes of a large, flexible, beam-like structure.

A very legitimate question asked is why fuzzy? Cohen, Weller and Ben-Asher (1996) applied the AFCA to a two-mass-spring system having a flexible as well as a rigid body mode. The settling time of the fuzzy based algorithm was 35% faster than that obtained in comparison to the "best" linear constant parameter, virtual passive controller. Both controllers were subject to identical hard constraints on the maximum allowable control. In addition the AFCA displayed far better stability and performance robustness characteristics. The major contribution to the success of the fuzzy controller is the ability to vary the damping of the virtual DVA. The variation, which is conducted smoothly, is based on structural insight and described by using simple linguistic terms. This structural insight was obtained when applying fixed parameter passive DVA's to a L.F.S. (Cohen and Weller, 1994).

FUZZY ADAPTATION STRATEGY

Introduction

One of the gifts of nature is the inherent human capability of making effective decisions based on inexact linguistic information. This all important characteristic has often led to the preference of man-in-the-loop type control as opposed to *autonomous controlled machines.* Hence, the performance of electromechanical controllers designed for "uncertain" dynamic systems may be improved by their modeling in a manner that emulates human reasoning. The logical system that captures the spirit of our approximate, imprecise world was introduced by Lotfi Zadeh (1965), as the theory of fuzzy sets, which in time proved to be a very powerful tool for dealing quickly and efficiently with imprecision and nonlinearity.

Fuzzy logic, which is the logic on which fuzzy control is based, is a convenient way to map an input space into an output space (Jang and Gulley, 1995). The major mechanisms of the fuzzy logic controller (FLC) are: a set of if-then statements called linguistic control rules; and a fuzzy inference system that *interprets* the values in the input vector and, based on the linguistic rules, *assigns* values to the output vector. The structure of a fuzzy logic controller is depicted in Figure 2b. The experience of the past decade, with the successful marketing of a wide variety of products based on the FLC (Kosko, 1994), has shown that for certain applications FLC provides superior results to those obtained by other conventional means.

Fuzzy Membership Functions

The first stage in building the fuzzy part of the controller, described in Figure 2, is referred to as *Fuzzification* of the input/output parameters. The transverse displacement $y(m,t)$ and the transverse velocity $\dot{y}(m,t)$ are normalized to the dimensionless variables y_N and $\dot{y}_N$ respectively, using the following relations:

$$
\begin{aligned}
y_N &= N_y \cdot y(m,t) \\
\dot{y}_N &= N_{\dot{y}} \cdot \dot{y}(m,t)
\end{aligned}
\tag{9}
$$

where N_y and $N_{\dot{y}}$ function as tuning parameters for the arbitrarily chosen membership functions illustrated in Figures 3 and 4, respectively. The use of these "tuning knobs" substantially cuts the degrees of freedom involved in reaching the required membership functions. In addition, once N_y and $N_{\dot{y}}$ are found, the sensitivity of individual fuzzy sets to the closed-loop performance is examined. Since no improvement is obtained, no additional changes are made to the arbitrarily selected membership functions of y_N and $\dot{y}_N$.

Fuzzy sets for the normalized transverse displacement, y_N, are characterized by membership functions μ_N, μ_{NS}, μ_Z, μ_{PS} and μ_P that map elements of the universe of discourse y_N, into the closed interval [0,1] as follows:

$$
\mu_L = y_N \rightarrow [0,1] \quad \text{for } L = \text{N, NS, Z, PS, P} \tag{10}
$$

where L stands for one of the linguistic terms used in this effort to categorize y_N, i.e., N (negative), NS (negative small), Z (zero), PS (positive small) and P (positive). The membership functions given in Equation (10) express the degree to which y_N belongs to some category L. These fuzzy sets may be viewed by plotting y_N versus μ_L as shown in Figure 3. For example, for the normalized displacement, y_N, a crisp value of 1.0 corresponds to 25 percent positive small and 75 percent zero.

Similarly, fuzzy sets for the normalized transverse velocity, $\dot{y}_N$, are characterized by the membership functions μ_Q, for Q = N, Z and P [where N(negative), Z(zero) and P(positive)]. In Figure 4, $\dot{y}_N$ is mapped to the characteristic fuzzy sets. In this case, it was sufficient to describe the mapping by using only three membership functions.

The fuzzy sets for the damping factor, δ, presented in Figure 5, are characterized by four membership functions μ_s, μ_M, μ_L and μ_{EL} [where S (small), M (medium), L (large) and EL (extra large)]. This mapping was initially based on intuition and previous work (Cohen and Weller, 1994). However, some fine-tuning, of the medium and large fuzzy sets was required to improve performance (reduce settling times).

Fuzzy Rule-Base and Inference

The fuzzy adaptation strategy, presented in this effort, is based on rules of the form "if... then..." that convert inputs

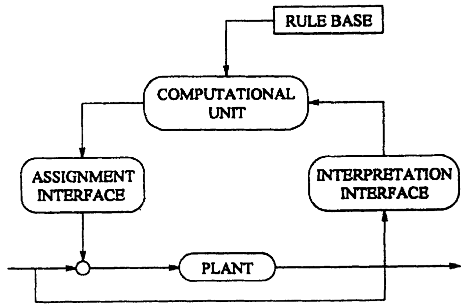

Figure 2b. Fuzzy logic control system.

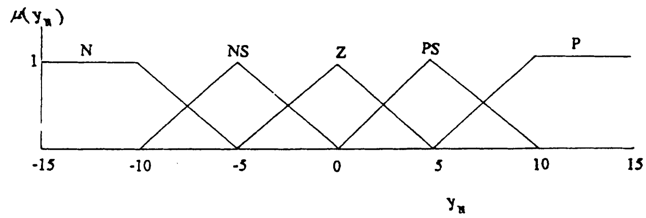

Figure 3. Membership functions for the fuzzy sets characterizing y(m,t).

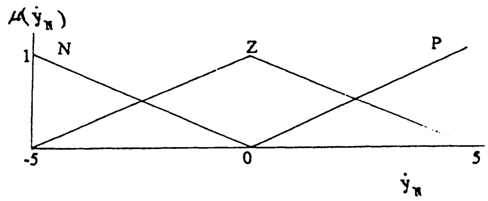

Figure 4. Membership functions for the fuzzy sets characterizing ẏ(m,t).

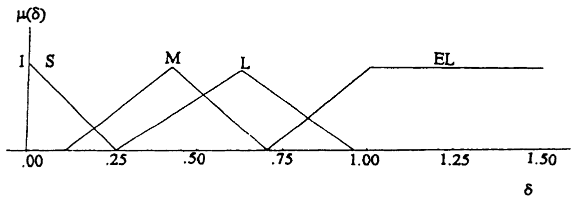

Figure 5. Membership functions for the fuzzy sets characterizing δ(t).

Table 1. Rule-base for computing the viscous damping factor (δ).

	y_N Negative	y_N Negative Small	y_N Zero	y_N Positive Small	y_N Positive
$\dot{y}_N$, Positive	SMALL	MEDIUM	LARGE	MEDIUM	SMALL
$\dot{y}_N$, Zero	MEDIUM	LARGE	EXTRA LARGE	LARGE	MEDIUM
$\dot{y}_N$, Negative	SMALL	MEDIUM	LARGE	MEDIUM	SMALL

(normalized transverse displacement and velocity) to a single output (damping factor), i.e., conversion of one fuzzy set into another (Kosko, 1991). The fuzzy rules were inspired by "common sense" engineering reasoning whereby *large* values of the inputs require a *lightly* damped absorber, which would provide quick rise times. However, when the plant state is in the vicinity of the desired state the damping factor is *large* to reduce the overshoot and steady state error. The resulting rule-base that converts fuzzified inputs into a fuzzy output is presented in Table 1. For example, the rule described by the first row, first column, in Table 1, reads as "if y_N is negative AND $\dot{y}_N$ is positive, then the damping factor, δ, is small".

As observed in Table 1, the rule-base contains quite a few rules relating to the same output variable. Therefore, to obtain an overall output in the fuzzy state, an inference method is applied. First, the degree of fulfillment of each and every rule is found by applying the fuzzy "AND" operation. Let us represent the individual elements of the rule-base "matrix", presented in Table 1, as δ_{ij} (i = 1,3; j = 1,5), where:

$$\delta_{ij} = \text{Minimum}(\mu_Q, \mu_L) \tag{11}$$

for

Q = positive, zero and negative
L = negative, negative small, zero, positive small and positive

Then, from Table 1, the union of the fuzzy sets for the same output variable is taken to reach the membership functions of the output, described in Figure 5, as follows:

$$\begin{aligned}\mu_M &= \delta_{12} + \delta_{14} + \delta_{21} + \delta_{25} + \delta_{32} + \delta_{34} \\ \mu_S &= \delta_{11} + \delta_{15} + \delta_{31} + \delta_{35} \\ \mu_L &= \delta_{13} + \delta_{22} + \delta_{24} + \delta_{33} \\ \mu_{EL} &= \delta_{23}\end{aligned} \tag{12}$$

The rule-base, which was not made to be part of the tuning process, underwent only a single closed iteration to rid inactive rules. However, the sensitivity of closed-loop performance to changes in the rule-base was examined to verify the continued use of these heuristic rules.

Defuzzification

Finally, in order to reach a practical controller a control action comprised of a single numerical value is required. Therefore, the space of the fuzzy damping factor, obtained using the method described in the previous section, is mapped into a non-fuzzy space (crisp) in a process known as defuzzification.

There are various strategies aimed at producing a crisp value. Some of the commonly used strategies are the center of area (COA), the mean of maximum and the max criterion (Lee, 1990). However, there is no accepted systematic methodology for selecting a defuzzification strategy. Herein, the COA scheme is adopted. This strategy was found to yield better steady-state performance when compared to the other above mentioned strategies (Lee, 1990). The COA method projects the centroid of the output membership function, μ_R (for R = S, M, L and EL), defined in Equation (12), as the crisp value of the output viscous damping factor, δ:

$$\delta = \frac{\sum_R \mu_R \cdot A_R \cdot c_R}{\sum_R \mu_R \cdot A_R} \tag{13}$$

where

A_R = area under the Rth fuzzy set defined in Figure 5
c_R = centroid of the area A_R
R = small, medium, large and extra large (see Figure 5)

DYNAMIC SIMULATION

The effort is primarily aimed at examining the closed-loop performance of the fuzzy based adaptive controller. For this purpose, MATLAB (MathWorks, 1992) simulations were conducted to observe the unit impulse response and damping performance. As a means of comparison, the unit impulse response of the structure, with a passive dynamic vibration absorber attached to it, is also presented.

Table 2. Natural frequencies of the structure in Figure 1 (rad/sec).

Mode	First	Second	Third	Fourth	Fifth
Frequency	1.59	9.98	27.95	54.77	90.53

The Euler-Bernoulli continuum beam model of the orthogonal tetrahedral lattice structure (see Figure 1) is defined by the following parameters (Cohen and Weller, 1994):

Equivalent flexural rigidity	$7.17 \cdot 10^7$ Nm^2
Equivalent mass density (ρ)	0.69 Kg/m
Length of the structure (l)	150 m

The first five modes of the structure are considered in the simulation. The respective natural frequencies, given in [rad/sec] and corresponding to the beam model, are displayed in Table 2.

The structure, subjected to an initial condition unit impulse, provides similar transient disturbance to each of the first five bending modes, thereby exciting all of them. The closed-loop controller is applied at the lapse of a second, at which time the beam is in the vicinity of the maximum open-loop amplitude (see Figure 6). The collocated, discrete, transducers are placed at the free end of the cantilevered structure ($m = l$). In addition, a sensor, positioned midway between the fixed and the free ends, at 1/2, observes the transverse displacement-time history. The value of the fundamental frequency, ω, is obtained from the first ten cycles of the open-loop transverse displacement sensor output (see Figure 6) as:

$$\omega = \frac{10 \cdot (2 \cdot \pi)}{39.5} = 1.591\,\mathrm{rad/sec} \tag{14}$$

The "measured" value of ω, given in Equation (14), is inserted into Equations (1), (2), (4) and (6) for the development and tuning of the model-independent closed-loop controller.

A passive dynamic vibration absorber tuned to the fundamental frequency, having a mass ratio of 0.1 and an optimal damping factor of 0.5 (Cohen and Weller, 1994), is placed at the free end of the structure (at $t = 1$ second). The time history of the displacement sensor outputs, presented in Figure 7, indicates that the vibrations have essentially died out after approximately 36 seconds. These results, obtained using a passive dynamic vibration absorber, serve as a mere baseline whose sole purpose is to grant the reader a sense of orientation. An increase in the mass ratio, corresponding to the respective selection of the damping factor, would obviously improve the settling time. However, the mass ratio of passive absorbers are constrained on account of the considerable weight penalty. The closed-loop fuzzy based adaptive controller, developed in this effort, is now applied to the nominal

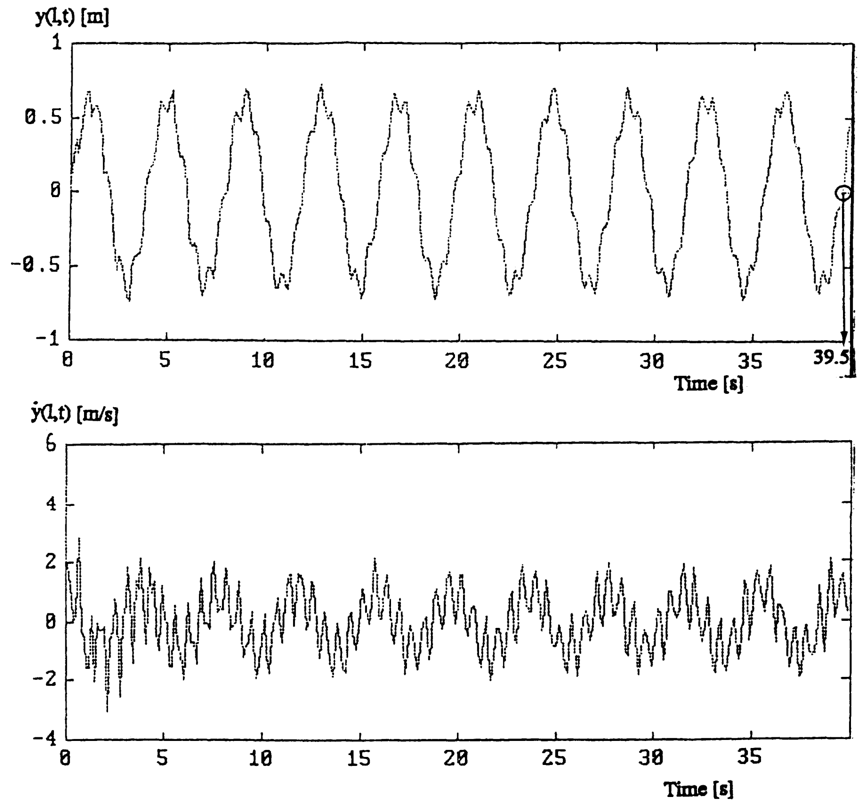

Figure 6. *Sensor output response of the open-loop system.*

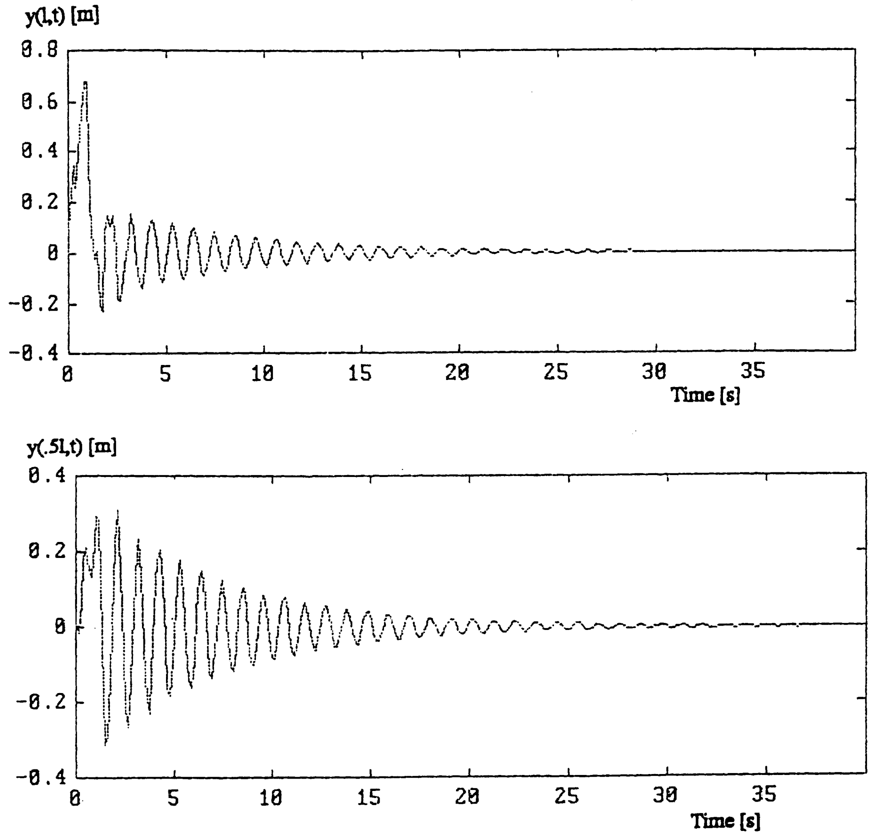

Figure 7. *Sensor output response of the structure with a passive absorber.*

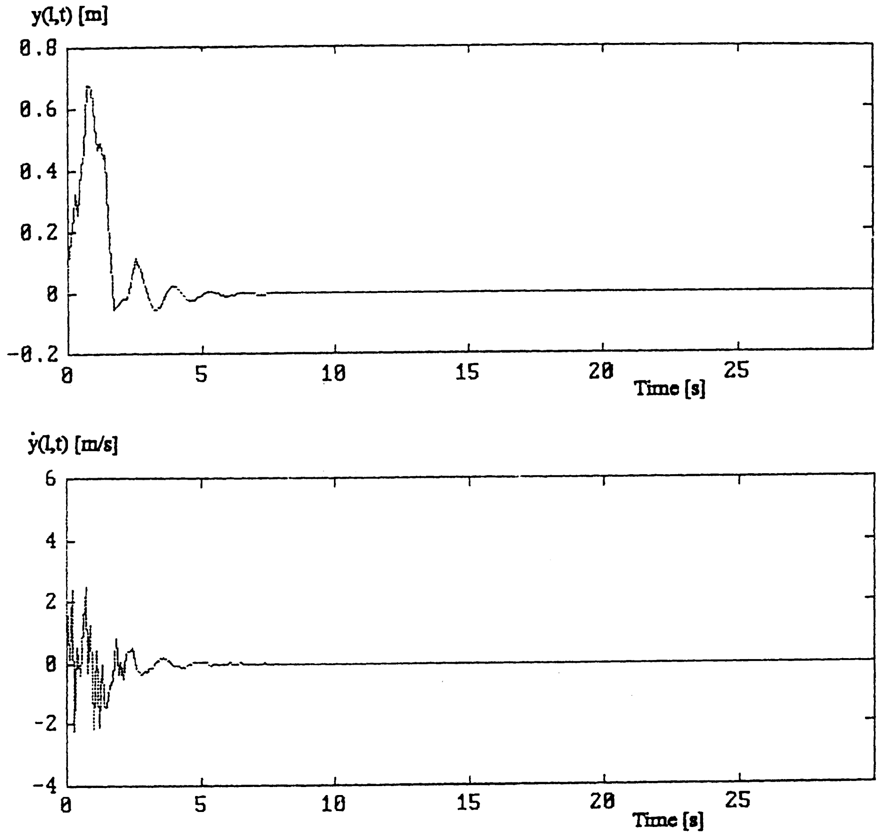

Figure 8. *Free end sensor outputs of the nominal closed-loop system.*

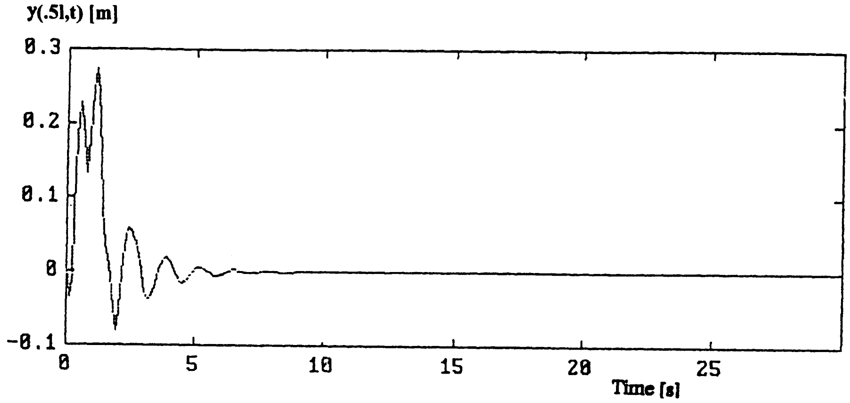

Figure 9. Beam center sensor output of the nominal closed-loop system.

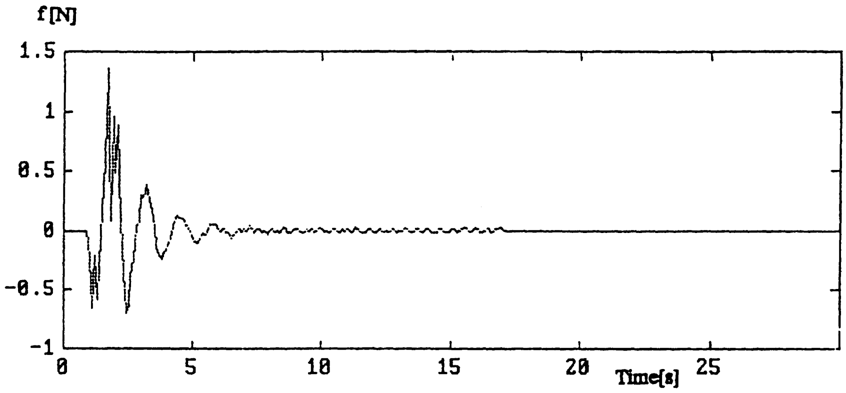

Figure 10. Control input versus time for the nominal closed-loop system.

Table 3. Natural frequencies of the nominal and perturbed plants.

	Nominal Plant	First Perturbed Plant	Second Perturbed Plant
EI ($\times 10^7$ Nm^2)	7.17	11.472	4.481
$\omega(\omega_1)$ (rad/sec)	1.59	2.01	1.26
ω_2 (rad/sec)	9.98	12.62	7.89
ω_3 (rad/sec)	27.95	35.35	22.10

plant. After some tuning, to a variety of initial conditions, the values of the tuning parameters, N_y, $N_{\dot{y}}$ and μ^*, are frozen:

$$N_y = 15\,[1/\text{m}]$$
$$N_{\dot{y}} = 0.5\,[\text{s/m}] \qquad (15)$$
$$\mu^* = 0.6$$

The resulting sensor outputs, placed at the free end, are shown in Figure 8. The time history of the transverse displacement at 1/2 is presented in Figure 9. The settling time is reduced by a factor of about 4.5, to approximately 8 seconds, when compared to the output of the passive system in Figure 7. The corresponding control input is illustrated in Figure 10.

About 16 seconds after the time the control force, which was activated a second after the application of the initial unit impulse, is "switched off", almost all of the vibrational energy has been dissipated. From this point in time onward, the structure continues to remain in the desired state not only at the free end but also at beam center (1/2). Therefore, it may be deduced that this desired state exists throughout the length of beam. The above observations, which are present irrespective of the initial conditions, suggest that for the configuration studied herein a single discrete actuator producing the required control force may be sufficient to control all of the 5 excited modes, without causing any control spillover.

To demonstrate the robustness of the control system to changes in the temporal dynamics of the Euler-Bernoulli continuum model, of the cantilevered lattice structure presented in Figure 1, the initial unit impulse response to a considerably perturbed plant is simulated. The Young's modulus, E, of the beam was subjected to two major perturbations. In one case, E was increased by 60 percent with respect to the nominal plant, while in the other instance E was reduced by 60 percent. These perturbations lead to changes only in the

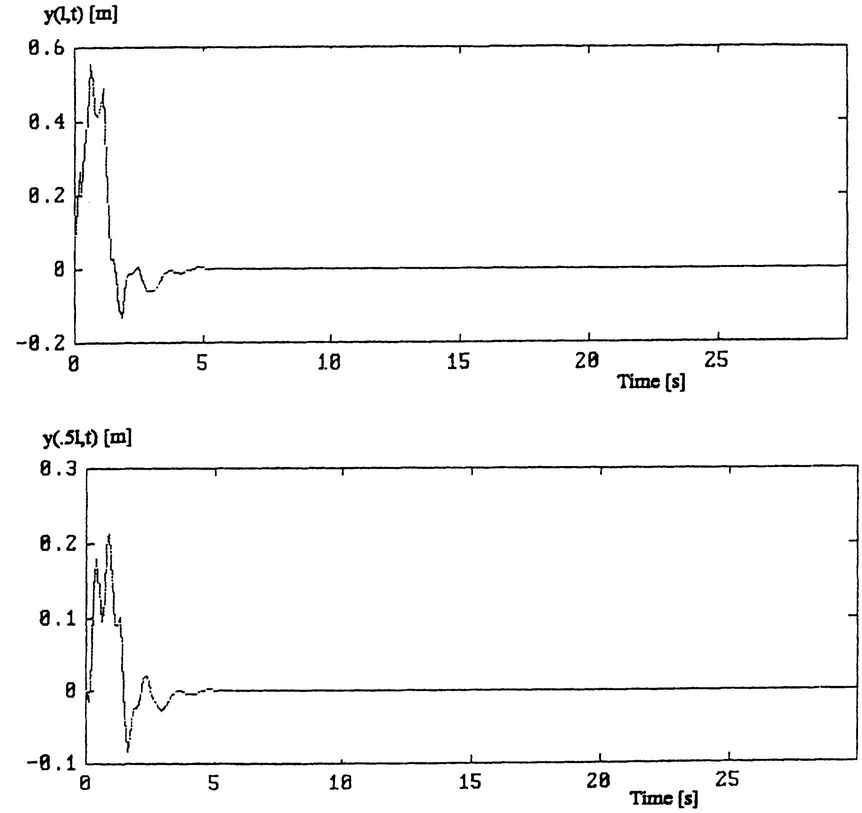

Figure 11. *Displacement sensor outputs for first perturbed plant.*

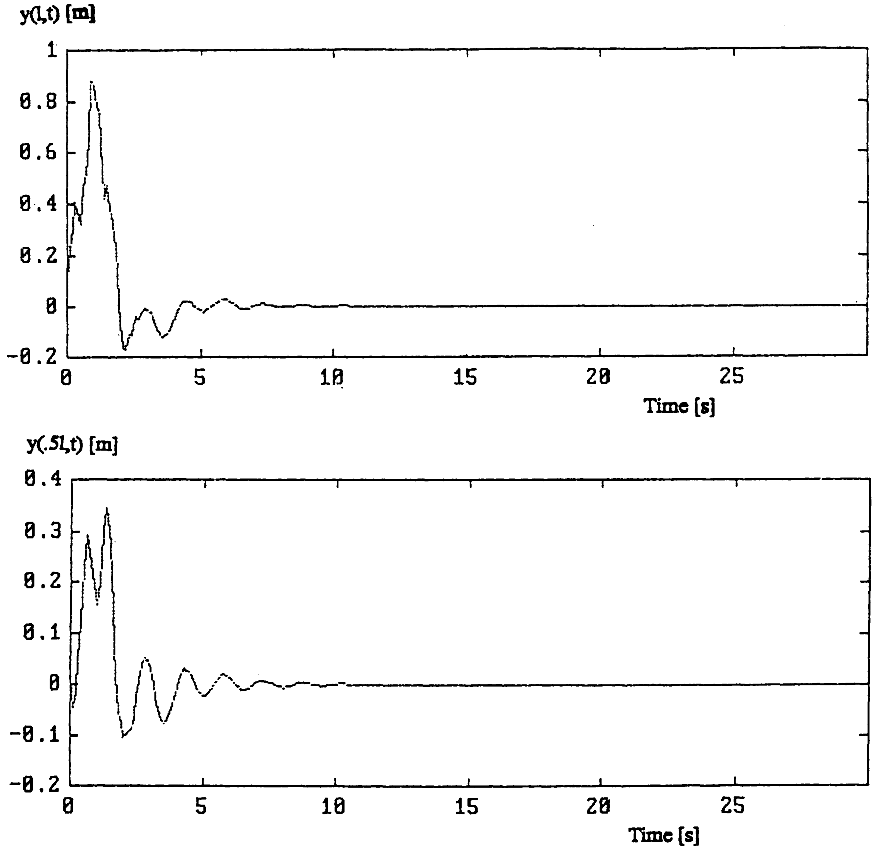

Figure 12. *Displacement sensor outputs for second perturbed plant.*

values of the natural frequencies. The mode shapes remain unchanged. The resulting frequencies, for the first three modes, of the Euler-Bernoulli beam model, are shown in Table 3.

The performance of the first and second perturbed plants is displayed in Figures 11 and 12, respectively. The settling times and rate of energy dissipation of the perturbed plants is approximately similar to that experienced with the nominal plant. This provides some indication of the robustness of the model-independent, fuzzy based adaptive controller to changes in the temporal dynamics of the plant. The importance of the relative insensitivity of the controller, developed in this effort, may be further highlighted by comparison with the robustness of a LQG/LTR compensated system. It was shown (Meyer, Burke and Hubbard, 1993) that the linear model-based controller, applied to an Euler-Bernoulli model of a cantilever beam with collocated distributed transducers, was unstable for a 2.5 percent change in the beam's natural frequencies corresponding to a 5 percent reduction only of the plant Young's modulus from the nominal.

CONCLUSIONS AND RECOMMENDATIONS

The present effort describes the development of a fuzzy based controller that emulates the functioning of an adaptive dynamic vibration absorber tuned to the fundamental frequency. The controller is applied to an Euler-Bernoulli cantilever beam which is subjected to an initial unit impulse disturbance. Collocated transducers, placed at the free end of the beam, were used in the control algorithm that assumed no a priori knowledge of the temporal plant model. MATLAB simulations of the closed-loop transient response, for the nominal and two other substantially perturbed plants, demonstrate quick settling times, a high rate of vibrational energy dissipation and no control spillover to the higher modes.

The controller presented may further be developed for the vibration suppression of higher order beam models, as well as 3-D beam-like and plate-like latticed structures, having different boundary conditions and which include coupling between the bending and twisting modes. Future work may include a stability analysis of the closed-loop system and performance comparisons with other controllers and the closed-loop robustness to variations in the location of the sensors relative to the actuators (non-collocated transducers).

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