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SYSTEM IDENTIFICATION AND ACTIVE MODAL CONTROL OF A FLEXIBLE TRUSS STRUCTURE

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Abstract: *The aim of this paper is to illustrate the active control of vibration of a flexible truss structure using a model-based digital controller. The state-space model of the system is derived using a system identification technique known as the Observer/Kalman Filter Identification (OKID) method together with Eigensystem Realization Algorithm (ERA). Based on the measured response of the structure to a chirp input, an explicit state-space model of the equivalent linear system is determined. To reduce the vibrations caused by an impulse force, two active strut members are installed along a vertical of the base bay of the truss. The active strut element consists of a piezoelectric ceramic stack actuator, a force transducer and mechanical interfaces. An integral controller is designed to suppress vibration of the truss. The controller, formulated with the root locus approach, is designed to maximize modal damping of a constructed truss structure. Experimental results illustrate that the active piezoceramic strut actuators and the integral controller can effectively reduce vibration of the truss.*

1 INTRODUCTION

Smart structures, which use actuators, sensors and a controller, can be used to suppress vibration in situations where passive measures are undesirable because of weight or space constraints. Frequently, piezoelectric actuators and sensors are used as they are light, cheap and convenient to bond to structures ([1], [2]). Lead zirconate titanate (PZT) is often used as an actuator because it is relatively stiff and couples well to a structure.

A truss structure is one of the most commonly used structures in aerospace and civil engineering [3]. Because it is desirable to use the minimum amount of material for construction, the trusses are becoming lighter and more flexible which means they are more susceptible to vibration. A convenient way of controlling a truss structure is to incorporate a piezoelectric stack actuator into one of the truss members [4]. Research on the damping of truss structures began in the late 80's. Fanson et al. [5], Chen et al. [6] and Anderson et al. [7] designed active members made of piezoelectric transducers. Preumont et al. [8] used a local control strategy to suppress the low frequency vibrations of a truss structure using piezoelectric actuators. Their strategy involved the application of integrated force feedback using two force gauges each collocated with the piezoelectric actuators, which were fitted into different beam elements in the structure. Carvalhal et al. [9] used an efficient modal control strategy for the active vibration control of a truss structure. In that approach, a feedback force is applied to each mode to be controlled according to a weighting factor that is determined by assessing how much that mode is excited by the primary source. To test the effectiveness of the control strategy it was compared with an alternative approach and the numerical results showed that with the proposed strategy it is possible to significantly reduce the control effort required, with a minimal reduction in control performance. Abreu et al. [10] used a standard H_∞ robust controller design framework to suppress undesirable structural vibrations in a truss structure containing piezoelectric actuators and collocated force transducers. Li and Huang [11] developed a linear-quadratic-Gaussian (LQG) model for vibration control for an adaptive truss. Numerical examples and the vibration control experiments were used to validate the efficiency of the proposed method. Abreu et al. [2] verified experimentally the application of a self-organizing fuzzy controller (SOFC) to suppress the vibrations of a truss structure using a pair of piezoceramic stack actuators. In that study, a decentralized active damping with local SOFCs connecting each actuator to its collocated force transducer were used. Experimental tests were performed, which illustrated the effectiveness of the controller in reducing the vibrations of a truss structure. The experimental results have shown that piezoceramic stack actuators control efficiently the vibrations of the truss structure.

Active damping of truss structures with integral control was introduced at the beginning of the 90's [8] and has since been thoroughly studied both theoretically and experimentally [12]. This paper investigates numerically an integral force feedback controller for suppressing the undesired structural vibrations in a truss structure containing piezoelectric stack actuators and collocated force sensors forming a so-called smart/intelligent truss structure. It is shown that the control system consists of independent SISO loops, i.e. a decentralized active damping with local controllers connecting each actuator to its collocated force sensor. It is also demonstrated that this control problem can be formulated with the root locus approach.

Although actuators and sensors are crucial elements in the design of a smart structure, they are not the focus of this paper. The focus is on the design of the integral controller. The model based controller can be further subdivided into two types; one type uses a numerical model of the structure derived theoretically, using finite element models ([13], [14]), for example. The second type involves the determination of a model of the structure using measured input and output data [15]. This paper concentrates on this approach, and demonstrates the procedure to design such a controller. The main objective is to investigate the combination of a system identification method and the integral control technique to actively control vibration in a truss structure.

Ljung [16] provides an excellent introduction to the subject of system identification, and describes the various methodologies that have been developed. Among the time domain methods, the Observer/Kalman Filter Identification (OKID) algorithm has shown to be efficient and robust ([17], [18]), and has been applied to space structures, such as the Shuttle Remote Manipulator System [19]. It has several advantages for the active vibration control application discussed here. First, it assumes that the system is a discrete linear time-invariant (LTI) state-space system. Second, it requires only input and output data to formulate the model (no priori knowledge of the plant is needed). Third, a pseudo-Kalman state estimator is produced, and lastly, any residual truncation errors will be small. Together with the OKID algorithm, the Eigensystem Realization Algorithm (ERA) ([18], [20]) generates a low order state-space model of the system to be controlled.

The paper is organized as follows. In section 2, the OKID and ERA approaches are summarized. Section 3 describes the experimental work in which the model and the controller of the truss structure, fitted with piezoelectric actuators and force sensors, is determined. Following this, real time control is implemented to demonstrate the efficacy of the integral control approach. Section 4 concludes the paper with some concluding remarks.

2 IDENTIFICATION OF THE DYNAMIC MODEL

In this section an overview is given of system identification technique used to determine a model of the system to be controlled. It consists of two parts: the OKID method to determine the system's Markov parameters, and the ERA to translate these parameters into a state-space model of the system.

2.1 Description of the Okid technique

The OKID method was developed to compute the Markov parameters of a linear system, which are the same as the sampled impulse response of the system. It is a time domain method which can work with general response data such as random vibration, impulsive signals or chirps. First, the observer Markov parameters are calculated, then the system Markov parameters are determined recursively from the Markov parameters of the observer system. The process of system identification using this method is described in ([17], [18]). In this section a brief overview of the process is given.

Consider first a general linear system expressed in discrete-time state-space form as

$$\begin{aligned} \mathbf{x}(k+1) &= \mathbf{A}\mathbf{x}(k) + \mathbf{B}\mathbf{u}(k) \\ \mathbf{y}(k) &= \mathbf{C}\mathbf{x}(k) + \mathbf{D}\mathbf{u}(k) \end{aligned} \qquad \text{(1a,b)}$$

where $\mathbf{x}$ is an $n \times 1$ state vector, $\mathbf{u}$ an $m \times 1$ input or control vector and $\mathbf{y}$ a $q \times 1$ output vector. Matrices $\mathbf{A}$, $\mathbf{B}$, $\mathbf{C}$ and $\mathbf{D}$ are the state, input, output, and direct influence matrix, respectively. The integer k represents sampled time.

The input–output description of the system with zero initial conditions can be obtained from Eq. (1) recursively as

$$\mathbf{y}(k) = \sum_{i=0}^{k-1} \mathbf{Y}_i \, \mathbf{u}(k-i-1) + \mathbf{D}\mathbf{u}(k) \tag{2}$$

where $\mathbf{Y}_i = \mathbf{C}\mathbf{A}^i\mathbf{B}$ and $\mathbf{D}$ are the Markov parameters of the system. They are also samples of the system impulse response. For a lightly damped system, many Markov parameters are needed because the impulse response takes a long time to decay away. To reduce this number, an observer is introduced to artificially add damping and hence reduce the length of the impulse response of the combined system. If $(\mathbf{A}, \mathbf{C})$ is an observable pair, then there exists an observer of the form

$$\begin{aligned} \hat{\mathbf{x}}(k+1) &= \mathbf{A}\hat{\mathbf{x}}(k) + \mathbf{B}\mathbf{u}(k) - \mathbf{M}[\mathbf{y}(k) - \hat{\mathbf{y}}(k)] \\ &= (\mathbf{A} + \mathbf{M}\mathbf{C})\hat{\mathbf{x}}(k) + (\mathbf{B} + \mathbf{M}\mathbf{D})\mathbf{u}(k) - \mathbf{M}\mathbf{y}(k) \\ \hat{\mathbf{y}}(k) &= \mathbf{C}\hat{\mathbf{x}}(k) + \mathbf{D}\mathbf{u}(k) \end{aligned} \tag{3a,b}$$

Matrix $\mathbf{M}$ can be interpreted as an observer gain matrix. Consider the special case where all eigenvalues of $\mathbf{A} + \mathbf{M}\mathbf{C}$ are zero. Thus, the estimated state $\hat{\mathbf{x}}$ converges to the true state $\mathbf{x}(k)$ after at most n steps where n is the order of the system. Equation (3) then becomes

$$\begin{aligned} \mathbf{x}(k+1) &= (\mathbf{A} + \mathbf{M}\mathbf{C})\mathbf{x}(k) + (\mathbf{B} + \mathbf{M}\mathbf{D})\mathbf{u}(k) - \mathbf{M}\mathbf{y}(k) \\ \mathbf{y}(k) &= \mathbf{C}\mathbf{x}(k) + \mathbf{D}\mathbf{u}(k) \end{aligned} \tag{4a,b}$$

The input–output description of the system described by Eq. (4) is given by (for $k \geq n$)

$$\mathbf{y}(k) = \sum_{i=0}^{n-1} \bar{\mathbf{Y}}_i \, [\mathbf{u}(k-i-1) \quad \mathbf{y}(k-i-1)]^T + \mathbf{D}\mathbf{u}(k) \tag{5}$$

where

$$\bar{\mathbf{Y}}_i = \left[\mathbf{C}(\mathbf{A} + \mathbf{M}\mathbf{C})^i(\mathbf{B} + \mathbf{M}\mathbf{D}) \quad -\mathbf{C}(\mathbf{A} + \mathbf{M}\mathbf{C})^i\mathbf{M}\right] = \left[\bar{\mathbf{Y}}_i^{(1)} \quad \bar{\mathbf{Y}}_i^{(2)}\right],$$

in which $\bar{\mathbf{Y}}_i$ and $\mathbf{D}$ are the Markov parameters of the observer system.

A particular feature of this type of observer is that the Markov parameters $\bar{\mathbf{Y}}_i$ will become identically zero after a finite number of time steps. A standard recursive least-squares technique is used to solve Eq. (5) and then the observer Markov parameters are computed. Once the Markov parameters of the observer system are identified, the actual system Markov parameters can be calculated. The relationship between the Markov parameters of the observer system and those of the actual system is given by

$$\mathbf{Y}_i = \mathbf{C}\mathbf{A}^i\mathbf{B} = \bar{\mathbf{Y}}_i^{(1)} + \sum_{k=0}^{i-1} \bar{\mathbf{Y}}_k^{(2)} \, \mathbf{Y}_{i-k-1} + \bar{\mathbf{Y}}_i^{(2)}\mathbf{D} \tag{6}$$

Once the system Markov parameters have been determined, a state-space model of the system can then be derived using the ERA, which is described in the following section.

2.2 Minimum realization of the system model using the ERA

The estimated state-space model $(\hat{\mathbf{A}}, \hat{\mathbf{B}}, \hat{\mathbf{C}}, \hat{\mathbf{D}})$ of a system is determined from the system Markov parameters $\mathbf{Y}_i$ obtained by OKID using the ERA. Details of this approach can be found in ([18],

[20]), so only a brief overview is given here. The algorithm begins by forming the $l \times l$ block Hankel matrix $\mathbf{H}(l,i)$ given by

$$\mathbf{H}(l,i) = \begin{bmatrix} \mathbf{Y}_i & \mathbf{Y}_{i+1} & \cdots & \mathbf{Y}_{i+l-1} \\ \mathbf{Y}_{i+1} & \mathbf{Y}_{i+2} & \cdots & \mathbf{Y}_{i+l} \\ \vdots & \vdots & & \vdots \\ \mathbf{Y}_{i+l-1} & \mathbf{Y}_{i+l} & \cdots & \mathbf{Y}_{i+2l-2} \end{bmatrix} \tag{7}$$

The order of the system is determined from the singular value decomposition of $\mathbf{H}(l,0)$ which is given by

$$\mathbf{H}(l,0) = \mathbf{U\Sigma V}^T \tag{8}$$

where the matrix $\mathbf{U}$ and $\mathbf{V}$ are unitary matrices, $\mathbf{\Sigma}$ is an $n \times n$ diagonal matrix of positive singular values, and n is the order of the system. Defining a $q \times lq$ matrix $\mathbf{E}_q^T$ and an $m \times lm$ matrix $\mathbf{E}_m^T$ made up of identity and null matrices of the form

$$\begin{aligned} \mathbf{E}_q^T &= [\mathbf{I}_q \quad \mathbf{0}_{q\times(l-1)q}] \\ \mathbf{E}_m^T &= [\mathbf{I}_m \quad \mathbf{0}_{m\times(l-1)m}], \end{aligned} \tag{9a,b}$$

a discrete-time minimal order realization of the system can be written as

$$\hat{\mathbf{A}} = \mathbf{\Sigma}^{-1/2}\mathbf{U}^T\mathbf{H}(l,1)\mathbf{V\Sigma}^{-1/2} \tag{10}$$

$$\hat{\mathbf{B}} = \mathbf{\Sigma}^{1/2}\mathbf{V}^T\mathbf{E}_m \tag{11}$$

$$\hat{\mathbf{C}} = \mathbf{E}_q^T\mathbf{U\Sigma}^{1/2} \tag{12}$$

and the direct influence matrix $\hat{\mathbf{D}}$ can be identified by solving Eq. (5).

Obviously, the $\hat{\mathbf{A}}$, $\hat{\mathbf{B}}$, $\hat{\mathbf{C}}$, $\hat{\mathbf{D}}$ matrices describe the state space model, which are functions of the singular values of the collected data. Note that now the state space variables allow one to give a clear physical meaning to the identified state-space system.

3 EXPERIMENTAL WORK

The system identification procedure and the subsequent controller design methodology were carried out in a truss structure with a multiple-input multiple-output system. Figure 1 shows the experimental setup.

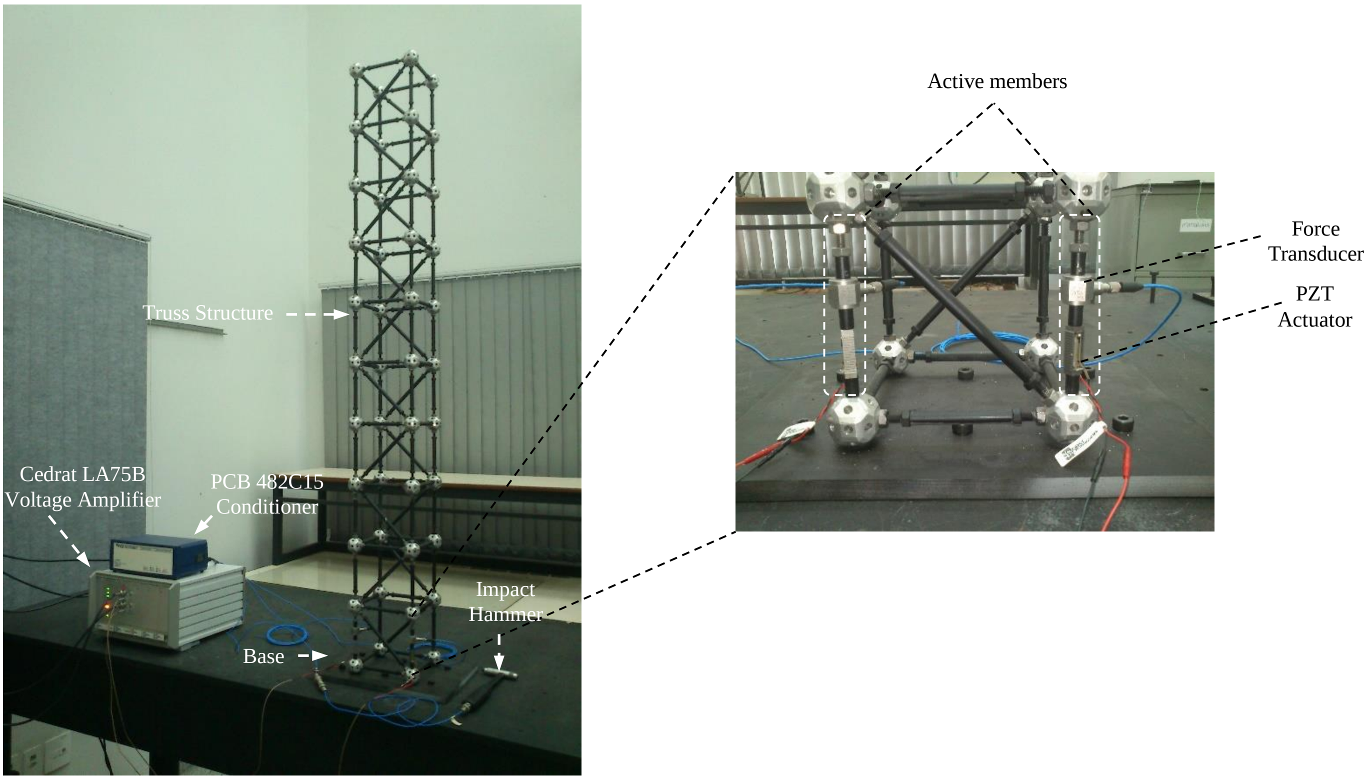


Figure 1. Truss structure used in the experimental tests. The detail shows the active member, with force transducer and the PZT stack actuator (Cedrat model PPA20M)

The truss structure (Fig. 1) is composed of 10 cubic bays assembled from a combination of 104 elements that begin and terminate in an aluminum node ball. There are a total of 44 node balls constituting the truss and the nodes at the bottom are clamped. The passive members are made of steel with a diameter of 8 mm. The structure is approximately 135 mm in length, 135 mm wide and 1.3 m tall (from the base plate).

To excite the vibration of the truss, an impact hammer (model PCB 086C04) was used. To achieve the maximum excitation effect, the truss was excited at its free end by the impact hammer. To achieve active suppression of the vibration of the truss, a pair of active members (Fig. 1) which consists of a force transducer (model PCB 208C03) and a PZT stack actuator were installed as vertical active members in the bay next to the base. Each active member replaces a regular strut member. A more detailed description of the truss and its finite element model and the positions of PZT actuator/force sensor can be found in reference [21].

In this experiment, the PZT stack actuator (model PPA20M) manufactured by Cedrat was used. This preloaded PZT actuator is a high resolution linear translator for static and dynamic applications. It provides sub millisecond response and sub nanometric resolution. The translators are equipped with high reliability multilayer PZT ceramic stacks protected by an internally spring-preloaded non-magnetic stainless steel case. The actuator provides a displacement up to 20 µm, a push force and a pulling force up to 800 N and 400 N, respectively, and an operating voltage range of -20 to 150 V. The voltage amplifier (model Cedrat LA75B) and the charge amplifier (model PCB 482C15 with gain of 20 V/V) shown in Fig. 1 were used to power the PZT stack actuator and to condition the signal from the force transducers, respectively. The truss response was measured by the force sensors collocated with the PZT stack actuators. The dSPACE system along with Matlab/Simulink® was used for digital data acquisition and real-time control.

3.1 Frequency response identification

Prior to designing the active controller, it was necessary to identify the frequency response of the truss system [2], whose input was the force of the impact hammer that excited at free end of the truss and whose output is each force sensor. The frequency response functions of the impact hammer-sensor systems were obtained using the Matlab/Simulink® software together with a PC and the dSPACE 1103 board. The frequency responses (calculated from 20 averages) of the system (in terms of the impact force applied by the hammer and the voltage measured from the force transducer) from 0 to 100 Hz are displayed in Fig. 2.

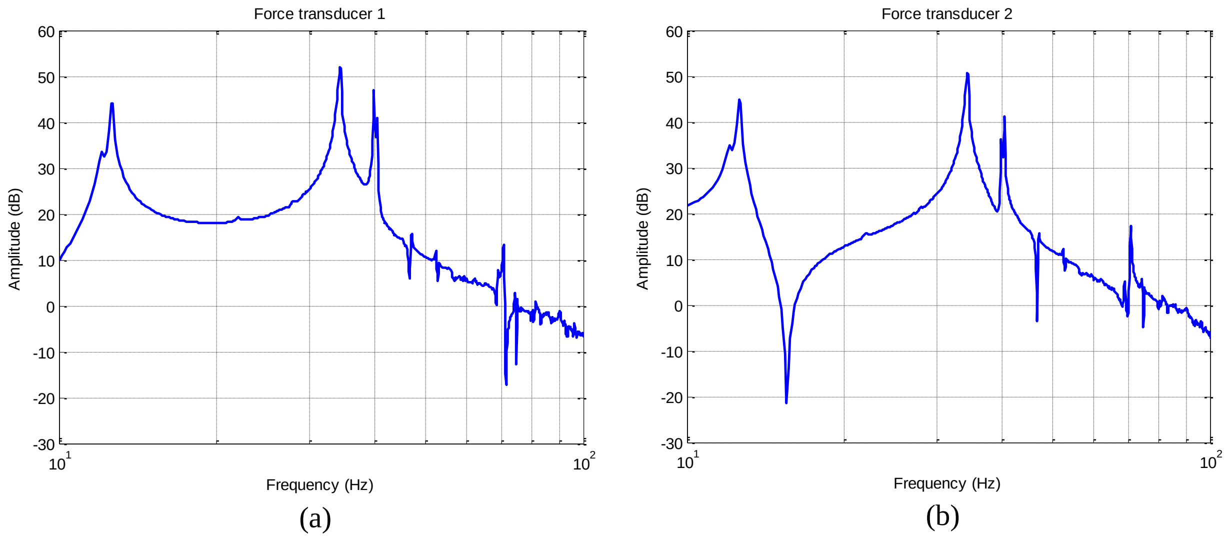


Figure 2. Frequency response of the force transducers: (a) 1 and (b) 2.

By examining the frequency response plots, the frequency of the dominant mode below 20 Hz is determined to be at 12.57 and 34.42 Hz. The strategy is to control simultaneously the first two modes ($\omega_1 = 78.98$ rad/s and $\omega_2 = 216.27$ rad/s) by using two active members (PZT struts) positioned in the elements shown in Fig. 1, and two decentralized integral controllers connecting each actuator to its collocated force transducer.

3.2 Identification of a model for the truss structure

To identify a model of the system, the experimental setup shown in Fig. 3 was used. A dSPACE 1103 board together with the Matlab/Simulink® software were used to generate and process the signals. The truss was driven with chirp signal (with frequency of 0 up to 100 Hz in 41 seconds) through the PZT actuators and the truss responses were measured using the force sensors. A sampling frequency of 1 kHz was used.

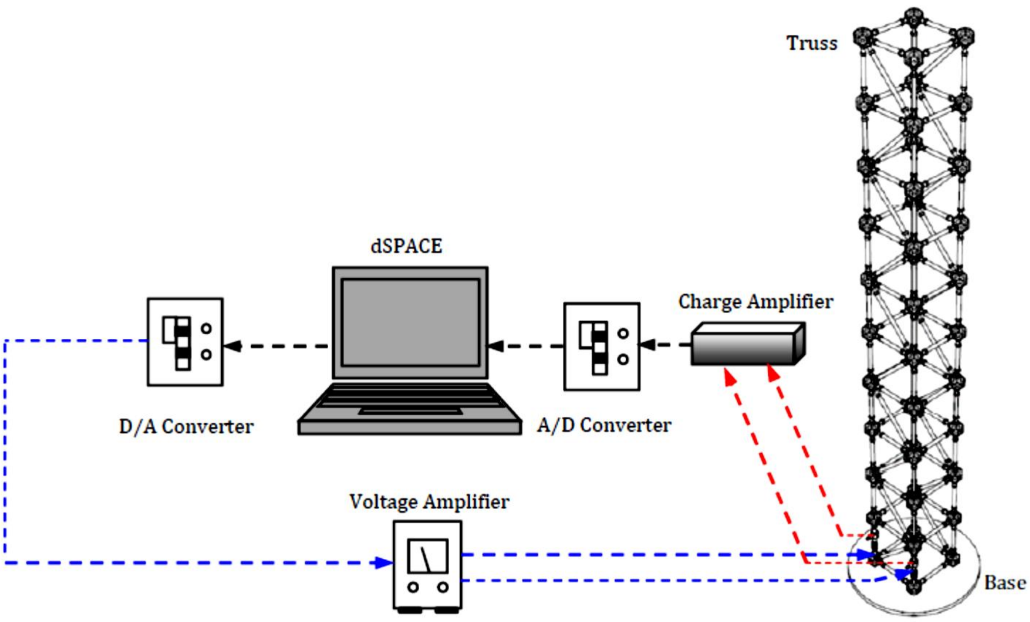
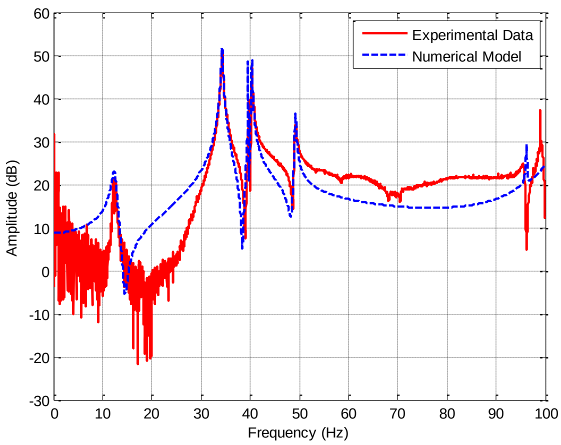


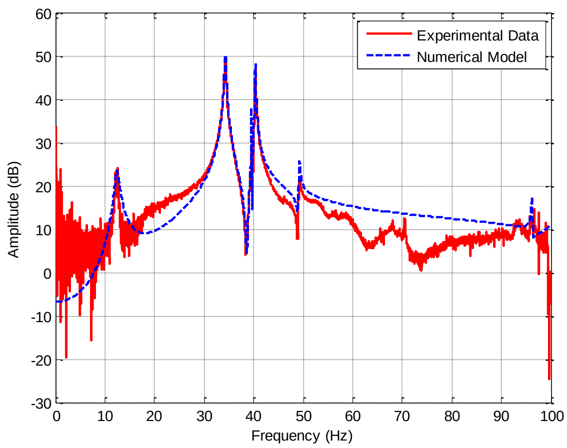
Figure 3. Experimental setup for model identification of the system.

Using the method described previously, the Markov parameters of the observer and the system were calculated, and consequently a state-space model of the truss structure was determined by using 45 states. As it was intended to control the first two modes of the system only, it was necessary to reduce the state space model. The Hankel norm model reduction technique [22] was used generate a fourteenth order model of the system.

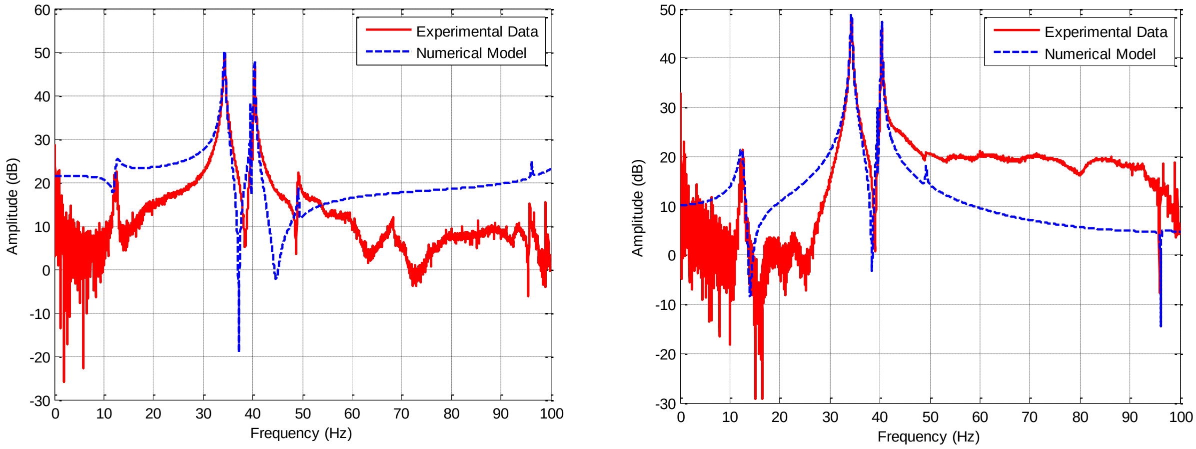
The measured frequency response functions (calculated from 20 averages) of the system (in terms of the voltage applied to the PZT actuators and the force signals measured from the force sensors) together with the reconstructed frequency response functions from the model are shown in Fig. 4. It can be seen that the frequency responses of the identified model are a reasonable match to the frequency responses of the actual system for the first two modes.



(a)



(b)



(c) (d)

Figure 4. Frequency response functions: (a) actuator 1 to force sensor 1; (b) actuator 1 to force sensor 2; (c) actuator 2 to force sensor 1 and (d) actuator 2 to force sensor 2.

3.3 Controller design

According to the integral control technique [12], the collocated i-th force sensor (y_i) is integrated and fed back to the i-th control input voltage (u_i):

$$u_i(s) = -\frac{k_i}{s+\varepsilon} y_i(s) \tag{13}$$

where k_i is the gain of the i-th controller and the constant ε is a forgetting factor that can be introduced by slightly moving the pole of the compensator from the origin to the negative real axis. In this paper, ε is assumed be equal to $\varepsilon = \frac{\omega_1}{2} = 39.49$ rad/s. The integral term 1/s introduces a 90º phase shift in the feedback path and thus damping in the system. It also introduces a -20 dB/dec slope in the open-loop frequency response, and thus reduces the risks of spillover instability [6].

The control law described by (13) can be implemented in a decentralized manner, with each actuator interacting only with the collocated sensor. In this case, the control system consists of independent SISO loops, whose stability can be readily established from the root locus of [12]:

$$k_i C_i(s) G_i(s) \tag{14}$$

where $G_i(s)$ is the structure transfer function between the actuator and the collocated sensor (Figs. 4a and 4d), and $C_i(s)$ is the active compensator given by $C_i(s) = \frac{1}{s+\varepsilon}$.

Figure 5 shows the root locus of the closed-loop system when the controller is tuned on mode 1 (5a and 5b) and mode 2 (5c and 5d), respectively, i.e., it shows the evolution of the closed-loop poles when k_1 and k_2 increase from 0 to $+\infty$.

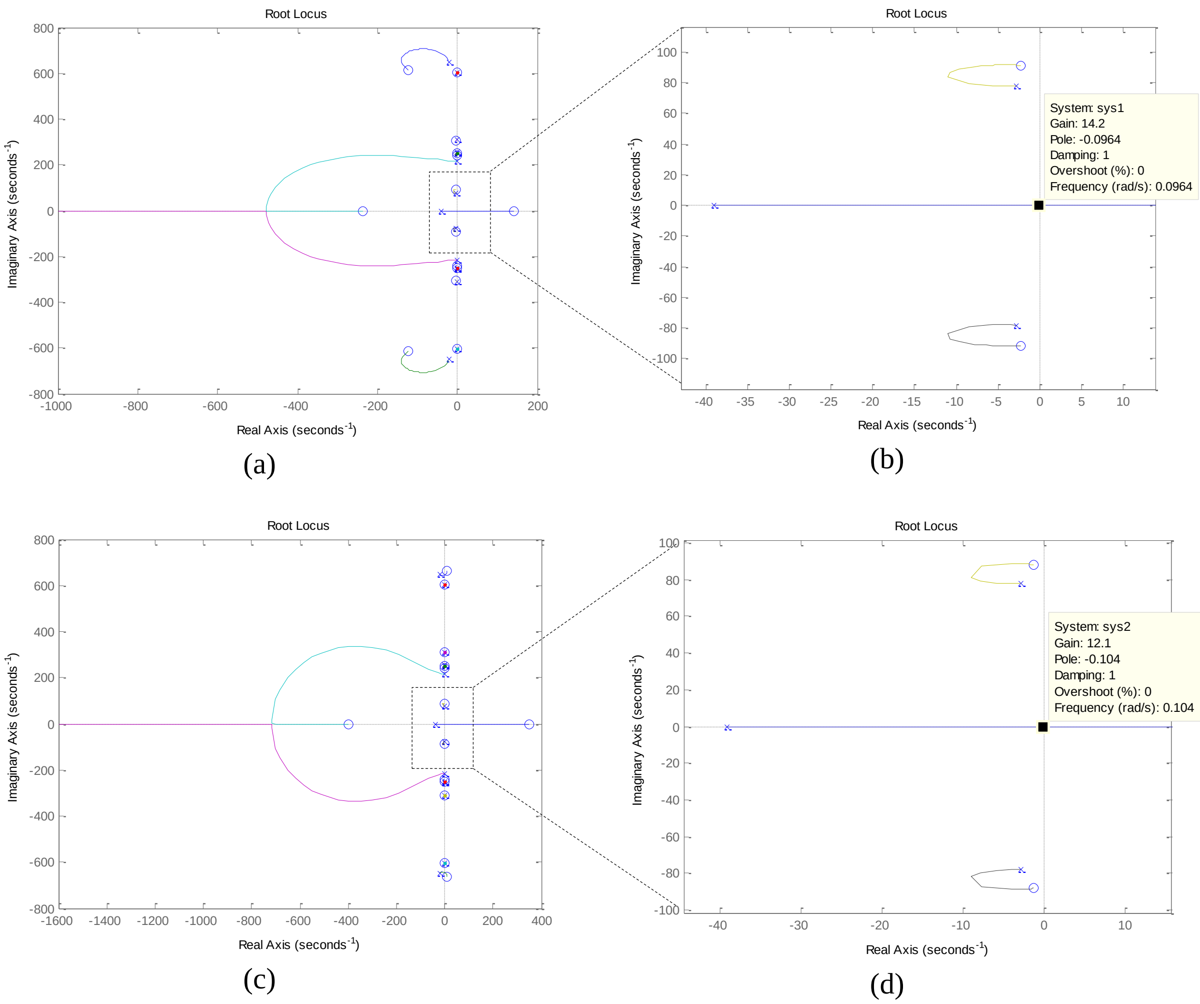


Figure 5. Root locus of the closed-loop system when the controller is tuned on (a), (b) mode 1 and (c), (d) mode 2.

Examining the Fig. 5, it is readily established from the root locus that the system is conditionally stable for a specific value of the gain k_i. Thus, as shown in Figs. 5b and 5d, there are stability limits which are reached when the closed-loop gains are equal to 14.2 for k_1, and 12.1 for k_2, respectively. A slightly upper value of the gain k_1 or k_2 would make the closed-loop system unstable. Therefore, the gains k_1 and k_2 can be chosen in such a way to avoid that condition. In this work, the gains k_i were chosen equal to 10 for both integral controllers.

3.4 Experimental results

As the controller was designed using the procedure discussed previously, some preliminary experiments were carried out with some initial control parameters set arbitrarily. The experiment was set up as shown in Fig. 3 but now the computer was set in control mode instead of system identification mode. The controller was implemented using Matlab/Simulink® software together with a PC and the dSPACE 1103 board. Figure 6 shows a Simulink® block diagram of the controller.

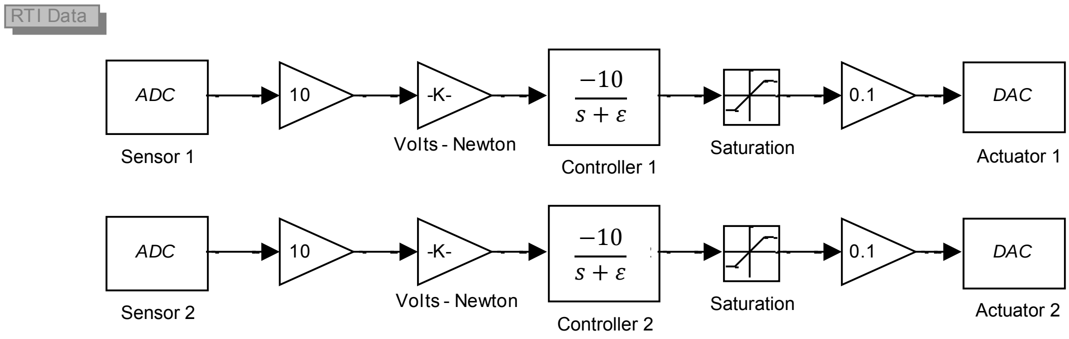


Figure 6. Simulink® block diagram of the controller.

The truss was excited at its free end by using the impact hammer (model PCB 086C04) shown in Fig. 1. The responses of the truss were measured by the sensor forces, with and without control. The time-domain results of the control experiment are given in Fig. 7.

Figures 7a and 7b show the open and closed-loop responses of the force sensors, and Figs. 7c and 7d show the corresponding control voltages. It is thus clear that the main effect of the control was to add more damping to the system.

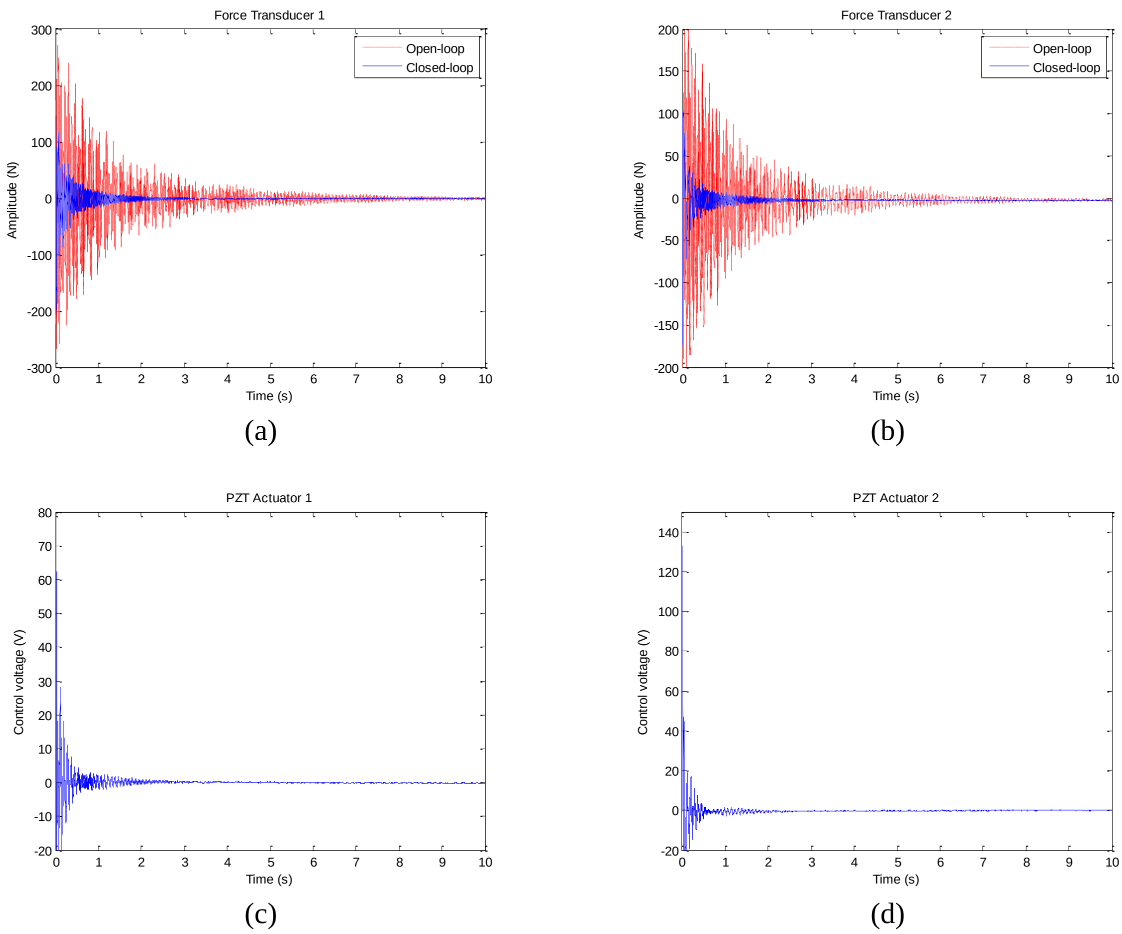


Figure 7. Time histories from the control experiments in which an impulsive force was applied to the free-end of the truss using a hammer, (a) and (b) force transducers 1 and 2 (c) and (d) control voltages applied to the PZT actuators 1 and 2.

Figure 8 shows the experimental open-loop and closed-loop frequency response functions determined from the time histories shown in Figs. 7a and 7b, and the time history of the force applied to the truss using the instrumented hammer. It can be noted that the control system reduced the vibrations in the frequency range containing the first two modes. From the results shown bellow, it can be observed that the frequency responses are reduced greatly, approximately 20 dB for first mode and 30 dB for second mode.

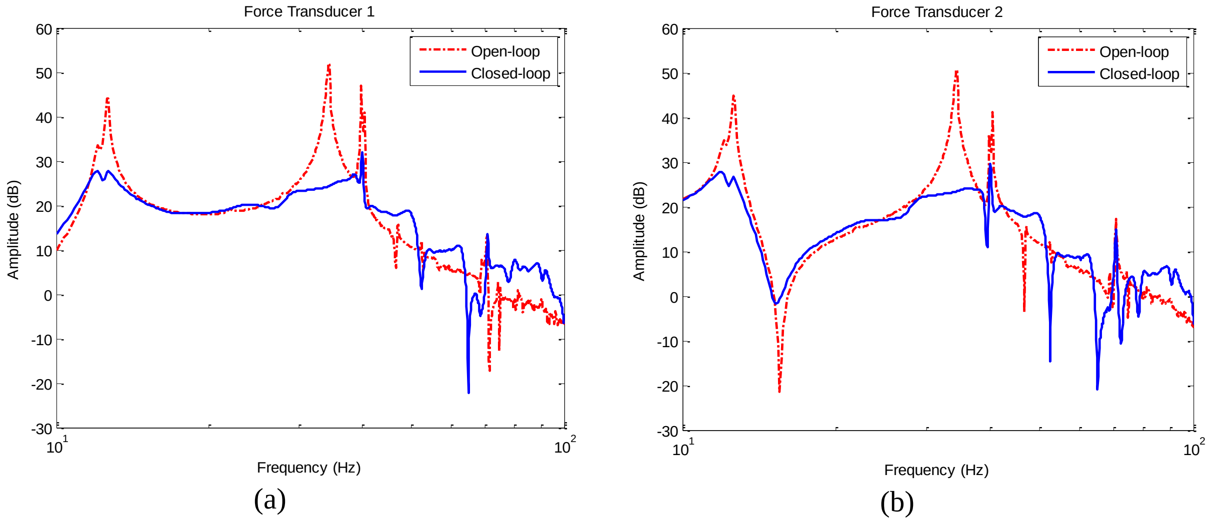


Figure 8. Open and closed-loop transfer functions of the truss in which an impulsive force was applied to the free-end of the structure using a hammer (a) force sensor 1, (b) force sensor 2.

4 CONCLUSIONS

This paper has described the system identification, controller design, and subsequent implementation to control the vibration of a truss structure. An integral controller was designed and experimentally implemented on a truss structure containing a pair of piezoelectric linear actuators collinear with force transducers. It was demonstrated that this control problem can be formulated with the root locus approach by using a model obtained through OKID/ERA system identification technique, using inputs and outputs vibration data from the truss. Two modes of the truss were controlled. From the experimental results, it was observed that satisfactory performance of vibration attenuation was achieved.

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REFERENCES

[1] G. L. C. M. Abreu, S. M. Conceição, V. Lopes Jr., M. J. Brennan, M. T. S. Alves, *System Identification and Active Vibration Control of a Flexible Structure*, Journal of the Brazilian Society of Mechanical Sciences and Engineering, Vol. 34, pp. 386-392 (2012).

[2] G. L. C. M. Abreu, G. P. Melo, V. Lopes Jr., M. J. Brennan, *Active Modal Damping Control of a Smart Truss Structure using a Self-Organizing Fuzzy Controller*, Journal of the Brazilian Society of Mechanical Sciences and Engineering, Vol. 1, pp. 1-10 (2014).

[3] Y. J. Yan, L. H. Yam, *A synthetic analysis of optimum control for an optimized intelligent structure*, Journal of Sound and Vibration, Vol. 249, pp. 775-784, (2002).

[4] D. K. Anthony, S. J. Elliot, *On reducing vibration transmission in a two-dimensional cantilever truss structure using geometric optimization and active vibration control techniques*, Journal of the Acoustical Society of America, Vol. 110, pp. 1191-1194 (2005).

[5] J. L. Fanson, G. H. Blackwood, C. C. Chu, *Active Member Control of Precision Structures*, SDM Conference, pp. 1480-1494 (1989).

[6] G. S. Chen, B. J. Lurie, B. K. Wada, *Experimental Studies of Adaptive Structures for Precision Performance*, SDM Conference, pp. 1462-1472 (1989).

[7] E. H. Anderson, D. M. Moore, J. L. Fanson, *Development of an Active Truss Element for Control of Precision Structures*, Optical Engineering, Vol. 29, Nº 11, pp. 1333-1341 (1990).

[8] A. Preumont, J. P. Dufour, C. Malekian, *Active damping by a local force feedback with piezoelectric actuators*, Journal of Guidance, Control and Dynamics, Vol. 15, pp. 390-395 (1992).

[9] R. Carvalhal, V. Lopes Jr., M. J. Brennan, *An efficient modal control strategy for the active vibration control of a truss structure*, Shock and Vibration, Vol. 14, pp. 393-406 (2007).

[10] G. L. C. M. Abreu, V. Lopes Jr., M. J. Brennan, *Robust Control of an Intelligent Truss Structure*, 10th International Conference on Recent Advances in Structural Dynamics, Southampton University, United Kingdom-UK (2010).

[11] W. P. Li, H. Huang, *Integrated Optimization of Actuator Placement and Vibration Control for Piezoelectric Adaptive Trusses*, Journal of Sound and Vibration, Vol. 332, pp. 17-32 (2013).

[12] A. Preumont, *Vibration Control of Active Structures: An Introduction*, Kluwer (2002).

[13] H. Allik, T. J. R. Hughes, *Finite Method for Piezoelectric Vibration*, International Journal for Numerical Methods in Engineering, No. 2, pp. 151-157 (1970).

[14] H. S. Tzou, C. I. Tseng, *Distributed Piezoelectric Sensor/Actuator Design for Dynamic Measurement/Control of Distributed Parameter Systems: A Piezoelectric Finite Element Approach*, Journal of Sound and Vibration, Vol. 138, No. 1, pp. 17-34 (1990).

[15] Z. Wang, S. Chen, W. Han, *Integrated Structural and Control Optimization of Intelligent Structures*, Engineering Structures, Vol. 21, pp. 183-191 (1999).

[16] L. Ljung, *System Identification: Theory for the User*, Englewood Cliffs, 2nd Ed., NJ: Prentice-Hall (1999).

[17] J. N. Juang, M. Phan, L. G. Horta, *Identification of Observer/Kalman Filter Markov Parameters: Theory and Experiments*, Journal of Guidance, Control and Dynamics, Vol. 16, No. 2, pp. 320-329 (1993).

[18] J. N. Juang, M. Phan, *Identification and Control of Mechanical Systems*, Cambridge University Press, New York (2001).

[19] M. Scott, M. Gilbert, M. Demeo, *Active Vibration Damping of the Space Shuttle Remote Manipulator System*, Journal of Guidance, Control and Dynamics, Vol. 16, No. 2, pp. 275-280 (1993).

[20] J. N. Juang, R. S. Pappa, *An Eigensystem Realization Algorithm for Modal Parameter Identification and Model Reduction*, Journal of Guidance, Control and Dynamics, Vol. 8, No. 5, pp. 620-627 (1985).
[21] G. L. C. M. Abreu, V. Lopes Jr., M. J. Brennan, *A Self-Organizing Fuzzy Controller for the Active Vibration Control of a Smart Truss Structure*, Vibration Control, Mickail Lallart (Ed.), ISBN: 978-953-307-117-6, Sciyo, pp. 215-234 (2010).
[22] W. K. Gawronski, *Advanced Structural Dynamics and Active Control of Structures*, Institute of Technology of P Institute of Technology of Pasadena, California, USA (1998).