World Applied Sciences Journal 12 (12): 2343-2351, 2011
ISSN 1818-4952

Adaptive Inverse Dynamics Control for Vibration Suppression of Flexible Panel with Piezoelectric Layers

Mohammad Azadi, Emad Azadi and Mehdi Roopaei

Department of Electrical Engineering, Science and Research Branch, Islamic Azad University, FARS, Iran

Abstract: In this paper an adaptive inverse dynamics control is applied to control and suppress the micro-vibrations of a flexible panel. The piezoelectric layers are used as sensors and actuators. Micro-vibrations, generally defined as low amplitude vibrations at frequencies up to 1 kHz, are now of critical importance in a number of areas. One such area is onboard spacecraft carrying sensitive payloads where the micro-vibrations are caused by the operation of other equipment. A rectangular simply supported flexible panel is considered and the equipments are located on it as lumped masses and concentrated forces. The concentrated loads induce micro-vibrations in the flexible panel. The governing equations of motion are derived based on Lagrange-Rayleigh-Ritz method. Finally the controller is applied and the system is simulated. Simulation results show the advantages of the adaptive inverse dynamics control algorithm.

Key words: Adaptive inverse dynamics control . flexible panel . piezoelectric sensors and actuators

INTRODUCTION

In recent years, the need to suppress the effects of micro-vibrations has increased. This is especially true for spacecraft structures where, due to ever increasing requirements to protect sensitive payloads, such as optical instruments or microgravity experiments, there is a pressing need to decrease the amplitude of the induced vibration.

Han and Lee [1] performed a refined analysis of composite plates with distributed piezoelectric actuators for vibration control. They used layerwise theory to modeling the in plane displacements through the thickness. Also they formulated the finite element method based on the developed mechanics. Chee *et al*. [2] presented a theoretical formulation for modeling composite smart structures, in which the piezoelectric actuators and sensors are treated as constituent parts of the entire structural system. They developed the mathematical model for a composite laminated plate structure using Hamilton`s variational principle with the finite element formulation. Amant and Cheng [3] extended a frequency domain model on the basis of a rectangular plate with symmetrically integrated piezo-elements to time domain suitable to use for online active vibration control simulations. Balamurugan and Narayanan [4] considered the mechanics for the coupled analysis of piezolaminated plate and piezolaminated curvilinear shell structures and their vibration control performance. Benjeddou *et al*. [5] proposed an exact two-dimensional analytical solution for the free vibration analysis of simply-supported piezoelectric adaptive plates. Mukherjee *et al.* [6] presented an active vibration control of stiffened plates. They formulated the stiffened plate finite element with piezoelectric effects. Benjeddou *et al.* [7] presented a two dimensional closed-form solution for the free vibrations analysis of simply supported piezoelectric sandwich plates. Raja *et al.* [8] used the piezoelectric actuator layers to suppress the vibrations of an aluminum panel. They applied LQR controller and simulated the system by using finite element method. In 2003, Gao and Shen [9] developed the incremental finite element equations for geometric nonlinear analysis of piezoelectric smart structures using a total Lagrange approach and virtual velocity incremental variational principle. Li and Cheng [10] investigated design of a controller for vibration control of a plate with piezoelectric patches. Ma [11] investigated the dynamic behavior and control of a clamped rectangular plate with bonded piezoelectric ceramic patches. He presented an adaptive nonlinear control scheme, which introduced a nonlinear function into the normal adaptive feedback control to non-linearize a reference signal. Caruso *et al.* [12] studied the vibration control of an elastic plate, clamped along one side and excited by an impulsive transversal force acting in correspondence of a free corner. Wang *et al.* [13] investigated the dynamic stability of negative velocity feedback control of piezoelectric composite plates using a finite element model. Moita *et al.* [14]

Corresponding Author: Dr. Mehdi Roopaei, Department of Electrical Engineering, Science and Research Branch, Islamic Azad University, FARS, Iran.

presented a finite element formulation for active vibration control of thin plate laminated structures with integrated piezoelectric layers, acting as sensors and actuators. Benassi and Elliott [15] investigated strategies for the suppression of plate vibration by considering approximations to the equivalent impedance of power-minimizing vibration controllers. Sloss *et al.* [16] formulated and validated a maximum principle for the vibration control of an annulus plate with the control forces acting on the boundary. Lin and Nien [17] investigated modeling and vibration control of a smart beam using piezoelectric damping modal actuators/sensors. Moita *et al.* [18] studied an active control to vibration suppression of PZT plates. Alessandroni *et al.* [19] presented a dynamic passive controller for thin smart plate vibrations. They used a four-node first order shear plate element model with reduced and selective integration. They also investigated geometrically nonlinear transient vibration response and control of plates with piezoelectric patches subjected to pulse loads. Lin [20] used the piezoelectric actuators to actively control the vibrations of smart panels by using a decomposed parallel fuzzy control approach. He demonstrated a general methodology by decomposing a large-scale system into smaller subsystems in a parallel structure so that the fuzzy control methodology could be used for studying a complex system. Moon and Hwang [21] applied an optimal control strategy to suppress the flutter of a supersonic composite panel using piezoelectric actuators. They investigated optimal control design based on the nonlinear model in order to obtain the maximum suppressible dynamic pressure with a lower control input as compared to a controller based on the linear model. Heidary and Eslami [22] outlined the governing equations of the linear response of piezothermoelastic plate based on the Hamilton`s principle and finite element methods. Robaldo *et al.* [23] presented some finite elements for the dynamic analysis of laminated plates embedding piezoelectric layers based on the principle of virtual displacements and an unified formulation. Moon [24] presented a finite element formulation of an optimal control scheme based on a Linear Quadratic Regulator (LQR) with output feedback for nonlinear flutter suppression of a composite panel with piezoelectric actuators and sensors. Ma and Ghasemi-Nejhad [25] studied vibration suppression of smart composite panels with piezoelectric patches. They extended the filtered Least Mean Square (LMS) Adaptive Feedforward Control (AFC) for vibration/noise reduction. To and Chen [26] studied optimal random vibration control of large-scale complicated shell structures with distributed piezoelectric components under nonstationary random excitations. Zhang and Shen [27] presented an analytical formulation for structural vibration control of laminated plates consisting of piezoelectric fiber-reinforced composite layers and orthotropic composite layers. Lin and Nien [28] discussed adaptive modeling and shape control of laminated plates with piezoelectric actuators. They developed a finite element formulation for modeling the dynamic and static response of laminated plates containing discrete piezoelectric ceramics subjected to both mechanical and electrical loadings. Ebrahimi and Rastgo [29] investigated the free vibration behavior of thin circular plates with distributed actuator layers made of piezoelectric material based on classical plate theory. Qiu *et al.* [30-31] presented acceleration sensor-based modal identification and active vibration control methods for the first two bending and the first two torsional modes vibration of a cantilever plate. Kapuria *et al.* [32] presented of an exact two-dimensional (2D) piezoelasticity solution for free vibration and steady-state forced response of simply supported piezoelectric angle-ply laminated circular cylindrical panels in cylindrical bending under harmonic electromechanical loads, with and without damping. Tavakol[pur [33] presented an active vibration control incorporating active piezoelectric actuator and self-learning control for a flexible plate structure. Thinh and Ngoc [34] developed a finite element model based on the first-order shear deformation theory for the static flexural shape and vibration control of a glass fiber/polyester composite plate bounded with piezoelectric actuator and sensor patches.

In this paper, an adaptive control scheme is presented to suppress the vibrations of a simply supported panel with equipments as lumped masses and concentrated loads. The piezoelectric layers are used as sensors and actuators. The governing equations of the system are derived by using the Lagrange-Rayleigh-Ritz method. The whole system is simulated and the results illustrate the effectiveness and capabilities of the control scheme.

SYSTEM DYNAMICS

We consider the case of a mass loaded panel with piezoelectric layers as sensors and actuators. The sensors and actuators employed are twin patches of piezoelectric material bonded onto opposite faces of the panel. The Lagrange-Rayleigh-Ritz based procedure used to model this system is based on Lagrange's equations of motion which take the form:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\psi}}\right)-\frac{\partial T}{\partial \psi}+\frac{\partial U}{\partial \psi}=Q \tag{1}$$

where

$$T = T_{pl} + T_{lm} + T_{pz}$$

$$U = U_{pl} + U_{pz}$$

and Q are the kinetic energy, potential energy and the vector of generalized forces, respectively. pl, lm and pz refer to panel, lumped mass and piezoelectric layers, respectively.

The displacement field is described by multiplication of mode shapes and generalized coordinates:

$$w(x,y,t) = \sum_{m=1}^{N_m}\sum_{n=1}^{N_n} S_{m,n}(x,y)\psi_{m,n}(t) = s^T\psi \tag{2}$$

Herein, w(x,y,t) is displacement and s and ψ are the vectors of mode shapes and generalized coordinates, respectively. The generalized forces are:

$$Q_i = \sum_{j=1}^{N_f} F_j \frac{\partial w_j}{\partial \psi_i} = S_f^T f \tag{3}$$

where f is vector of external forces and S_f is the vector of mode shapes at the corresponding force locations. By considering the kinetic energy as $T = 1/2\dot{\psi}^T M\dot{\psi}$, the mass matrices are obtained as follows.

$$M_{pl} = \iiint_{pl} \rho s s^T dxdydz$$

$$M_{pz} = \sum_{i=1}^{N_{pz}} \iiint_{pz_j} \rho_{pz_j} s s^T dxdydz \tag{4}$$

$$M_{lm} = \sum_{i=1}^{N_{lm}} M_{lm_j} s_{lm_j} s_{lm_j}^T$$

wherein N_{pz} and N_{lm} are the number of piezoelectric layers and lumped masses, respectively. The subscribes pz_i and lm_i mean the ith piezoelectric layer and lumped mass, respectively. The potential energy of the panel is

$$U = \frac{1}{2}\iiint_{Vol} \varepsilon^T \sigma . dxdydz = \frac{1}{2}\psi^T K_{pl}\psi \tag{5}$$

where ε is the strain vector and σ is the stress vector of the panel. By substituting the strain and stress vector in (5), the panel stiffness matrix will be obtained as follows

$$K_{pl} = \iiint_{pl} \frac{E_{pl} z^2}{(1-\upsilon)}\Big(\frac{\partial^2 s}{\partial x^2}\frac{\partial^2 s^T}{\partial x^2} + \frac{\partial^2 s}{\partial y^2}\frac{\partial^2 s^T}{\partial y^2} + 2\upsilon\frac{\partial^2 s}{\partial x^2}\frac{\partial^2 s^T}{\partial y^2} + 2(1-\upsilon)\frac{\partial^2 s}{\partial x \partial y}\frac{\partial^2 s^T}{\partial x \partial y}\Big)dxdydz \tag{6}$$

E_{pl} and υ, in (6) are the Young's modulus and Poisson's ratio of the panel, respectively. The potential energy of the piezoelectric layers is composed of mechanical and electrical terms:

$$U_{pz} = U_{pz_{elast}} + U_{pz_{elastelec}} + U_{pz_{elec}} \tag{7}$$

where $U_{pz_{elast}}$ is the energy stored due to the elasticity of the material, $U_{pz_{elastelec}}$ represents the additional energy due to voltage-driven piezoelectric effect and $U_{pz_{elec}}$ is the electric energy stored due to the dielectric characteristics of the piezoelectric material employed. To compute the elastic energy, an appropriate model for the stress-strain pattern in the piezoelectric patches must be selected and here we make the following assumptions: (i) The electrodes attached to the piezoelectric patches have negligible stiffness. (ii) The thickness of the layer of adhesive which connects each of the patches to the panel is negligible compared to that of the patches and is able to transfer all of the shear strain. (iii) The natural boundary conditions at the edges of each patch are not enforced and a uniform strain distribution is assumed through the whole patch. Considering these assumptions, the same procedure as employed for the panel can be used to write:

$$K_{pz_{elast}} = \sum_{pz_i}\iiint_{pz_i} \frac{E_{pz_i} z^2}{(1-\upsilon_i)}\Big(\frac{\partial^2 s}{\partial x^2}\frac{\partial^2 s^T}{\partial x^2} + \frac{\partial^2 s}{\partial y^2}\frac{\partial^2 s^T}{\partial y^2} + 2\upsilon_i\frac{\partial^2 s}{\partial x^2}\frac{\partial^2 s^T}{\partial y^2} + 2(1-\upsilon_i)\frac{\partial^2 s}{\partial x \partial y}\frac{\partial^2 s^T}{\partial x \partial y}\Big)dxdydz. \tag{8}$$

The electric field is $e = v/h_{pz}$, where v is the applied voltage and h_{pz} is the piezoelectric thickness. The stress due to the applied voltages is given by:

$$\sigma_{elect} = \begin{pmatrix}\sigma_{x_{elect}} \\ \sigma_{y_{elect}}\end{pmatrix} = \frac{E_{pz}}{1-v^2}\begin{pmatrix} d_{xz} + vd_{yz} \\ d_{yz} + vd_{xx}\end{pmatrix} e \tag{9}$$

Here, d_{xz} and d_{yz} are the piezoelectric constants of the material, which are assumed to have polling direction z perpendicular to the plate. By using equation (9) one may write the elastoelectric energy stored in the piezoelectric layers as:

$$U_{pz_{elastelect}} = V^T K_{pz_{elastelect}} \psi$$
$$K_{pz_{elastelect}} = \sum_{i=1}^{N_p} \iiint_{pz_i} \frac{E_{pz_i} d_{z_i} p_i}{2(1-v_i)} \left(z\frac{\partial^2 s^T}{\partial x^2} + z\frac{\partial^2 s^T}{\partial y^2}\right) dxdydz, \quad (10)$$

where d_{xz} and d_{yz} are considered to be the same and are equal to d_{x_i} . The electrical energy stored in the piezoelectric material can be expressed as:

$$U_{pz_{elec}} = \frac{1}{2}\iiint_{pz} e.d.dxdydz \quad (11)$$

where d is the electrical displacement. For each patch, the electrical displacement is [35]:

$$d_i = \varepsilon_{pz_i} \frac{v_i}{h_{pz}} \quad (12)$$

herein ε_{pz_i} is the dielectric constant of the piezoelectric material which forms the ith patch. Hence, the electrical energy will be:

$$U = \frac{1}{2} V^T K_{pz_{elect}} V,$$
$$K_{pz_{elect}} = \sum_{i=1}^{N_P} \iiint_{pz_i} \varepsilon_{pz_i} p_i\, p_i^T dxdydz \quad (13)$$

where p_i is a zero entries vector except for entry i which is equal to $1/h_{pz}$.Substituting (3-13) in (1) the governing equation of motion of the system will be obtained as:

$$(M_{pl} + M_{pz} + M_{lm})\ddot{\psi} + (K_{pl} + K_{pz_{elast}})\psi + K^T_{pz_{elastelect}} V = Q$$
$$K_{pz_{elastelect}} \psi + K_{pz_{elect}} V = 0 \quad (14)$$

The elastoelectric stiffness matrix can be divided into two parts corresponding to sensors and actuators.

$$K_{pz_{elastelec}} = [K_{pza_{elastelec}}, K_{pzs_{elastelec}}] \quad (15)$$

where ν_a and ν_s are the vectors of the voltages applied to the actuators and received from the sensors, respectively. Eq. (14) can be rewritten as:

$$M\ddot{\psi} + C_s\dot{\psi} + (K_{elast} + K_{pzs})\psi = -K^T_{pza_{elastelect}} v_a + s_f^T f \quad (16)$$

wherein all inertia elements are included in the inertia matrix, M and all stiffness elements in the stiffness matrix, K_{elast} and C_s is the damping matrix that can be added to the system. K_{pzs} shows the elastic energy stored in the piezoelectric sensors and is as follows

$$K_{pzs} = -K^T_{pzs_{elastelect}} K^{-1}_{pzs_{elec}} K_{pzs_{elastelect}} \quad (17)$$

ADAPTIVE INVERSE DYNAMICS CONTROL

Controllers that can handle regulation and tracking problems without the need of knowledge of the process parameters are by themselves appealing procedure. Such controller schemes belong to the class of adaptive control. Equation (16) is linear in dynamic parameters and can be written as [36]:

$$M\ddot{\psi} + C_s\dot{\psi} + K\psi = Y\bar{\rho} \quad (18)$$

where *Y* is a known matrix of the measurable variables and $\bar{\rho}$ is the vector of parameters of the system and $K = K_{elast} + K_{pzs}$. The control input can be determined by:

$$-K_{pza_{elastelect}} v_a = \hat{M}(\ddot{\psi}_d - K_D\dot{\tilde{\psi}} - K_P\tilde{\psi}) + \hat{C}_s\dot{\psi} + \hat{K}\psi \quad (19)$$

We assume here that $\hat{M}$, $\hat{C}_s$ and $\hat{K}$ have the same form as M, C_s and K with estimated parameters $\hat{\bar{\rho}}$. Taking advantage of (18) we can write the following equation.

$$-K_{pza_{elastelect}} v_a = Y\hat{\bar{\rho}} \quad (20)$$

Substituting (19) into the governing equations of the system gives the following closed-loop error equation

$$\hat{M}(\ddot{\tilde{\psi}} + K_D\dot{\tilde{\psi}} + K_P\tilde{\psi}) = Y\tilde{\bar{\rho}} \quad (21)$$

where $(\tilde{\cdot}) = (\cdot) - (\cdot)_{desired}$ and

$$Y\tilde{\bar{\rho}} = (\hat{M} - M)\ddot{\psi} + (\hat{C} - C)\dot{\psi} + (\hat{K} - K)\psi \quad (22)$$

The error dynamics of (21) can be rewritten as

$$\ddot{\tilde{\psi}} + K_D\dot{\tilde{\psi}} + K_P\tilde{\psi} = \hat{M}^{-1}Y\tilde{\bar{\rho}} = \bar{\Phi}\tilde{\bar{\rho}} \quad (23)$$

This equation can be transformed to state-space form by choosing

$$\xi_1 = \tilde{\psi}, \xi_2 = \dot{\tilde{\psi}}, \xi = (\xi_1^T\ \xi_2^T)^T, \text{ i.e.,}$$

$$\dot{\xi} = A\xi + B\bar{\Phi}\tilde{\bar{\rho}} \quad (24)$$

with

$$A = \begin{bmatrix} 0 & I \\ -K_P & -K_D \end{bmatrix}, B = \begin{bmatrix} 0 \\ I \end{bmatrix} \quad (25)$$

By choosing the Lyapunov function as

$$V = \xi^T P \xi + \tilde{\bar{\rho}}^T \Gamma \tilde{\bar{\rho}} \quad (26)$$

where P is the unique symmetric positive definite solution to the equation $A^T P + PA = -Q$, for a given symmetric positive definite matrix Q. Tacking the time derivative of V yields

$$\dot{V} = -\xi^T Q \xi + 2\tilde{\bar{\rho}}^T (\bar{\Phi}^T B^T P \xi + \Gamma \dot{\hat{\bar{\rho}}}) \quad (27)$$

Choosing the update law as

$$\dot{\hat{\bar{\rho}}} = -\Gamma^{-1} \bar{\Phi}^T B^T P \xi \quad (28)$$

equation (27) will be reduced to the following

$$\dot{V} = -\xi^T Q \xi \quad (29)$$

It can be shown that $\xi \in L_2 \cap L_\infty$, $\hat{\bar{\rho}} \in L_\infty$ and then the control input ν_a in (19) is bounded. It follows that $\ddot{\psi} \in L_\infty$ so that $\dot{\xi} \in L_\infty$. Then ξ is uniformly continuous and, since $\xi \in L_2$, it can be concluded that ξ asymptotically converges to zero.

SIMULATION RESULTS

The parameters of the panel and piezoelectric layers are summarized in Table 1. The panel is considered as simply supported where a harmonic concentrated load and a lump mass are applied to it. The location of lump mass and the point force has been considered as (0.0508, 0.1524) and (0.2, 0.1), respectively. In order to measure the ability of the controller, two cases are simulated and in both of them the controller gains are similar. First, the point force is considered as *f=sin(pt/10)*. Figure 1 shows the panel vibration in t=0.7s when no voltages are applied to the actuators. If voltages are applied to the actuators the vibration of the panel will be damped. Figure 2 shows the vibration of the panel in t=0.7s when controller are applied to the panel. Figure 4 and 5 show the vibration of the location of point force and lump mass, respectively. The effects of controller are noticeable in these figures. Second, the point force is considered with different frequency namely *f=sin(10pt)*. Figure 6 illustrates the forced vibration of the panel when no voltages are applied to piezoelectric layers. The effect of using the controller is shown in Fig. 7. The vibration of the location of point force and lump mass are indicated in Fig. 8 and 9, respectively. Comparing these figures illustrates that this adaptive inverse dynamics control has acceptable ability to suppress the vibrations of the panel in different conditions and there is no need to change the controller gains in different states.

Table 1: Dimensions and material properties [35]

Panel		Piezoelectric	
Length =	304.8 mm	d =	1.66e-10 m/V
Wide =	203.2 mm	ε_{pz} =	1700 ε^0
Thickness =	1.52 mm	h_{pz}=	0.19 mm
E_{pl} =	71e9 Pa	E_{pz} =	63e9 Pa
υ =	0.33	υ_{pz} =	0.3
ρ =	2800 kg/m^3	ρ_{pz} =	7650 kg/m^3

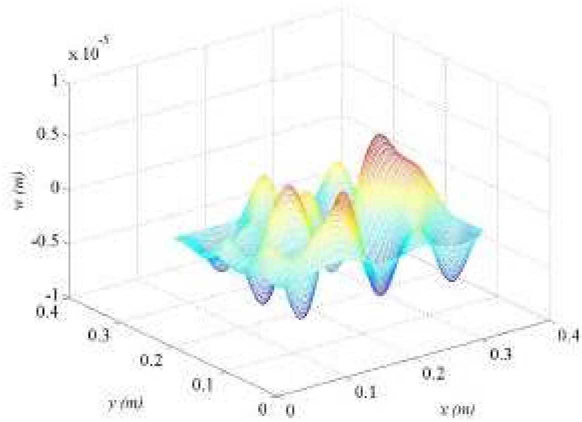


Fig. 1: The panel vibration when no voltage is applied to piezoelectric actuators (t=0.7s and *f*=sin(*p/10t*))

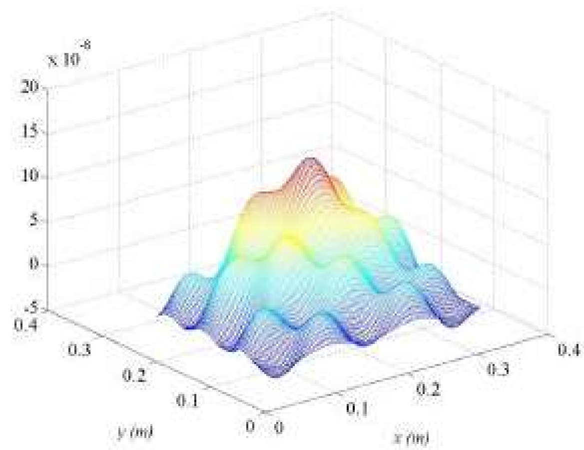
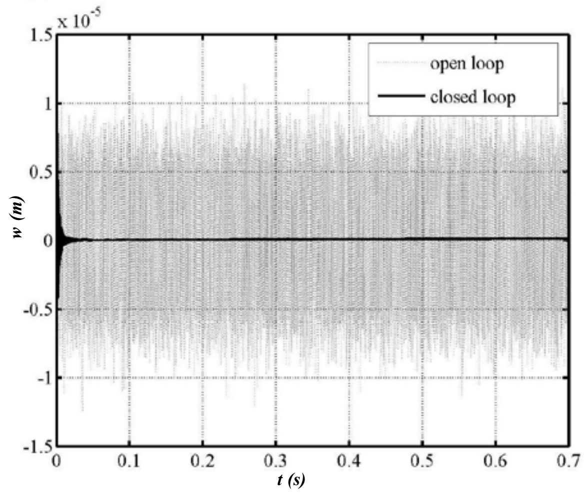
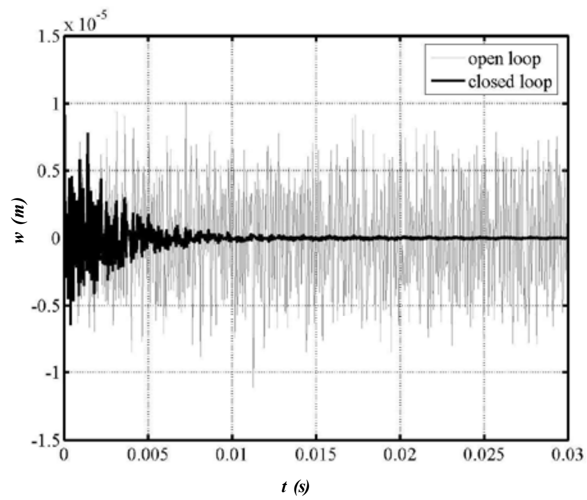


Fig. 2: The panel vibration when voltage is applied to piezoelectric actuators (t=0.7s and *f*=sin(*p/10t*))



(a)



(b)

Fig. 3: a) The panel vibration in the location of point force, b) zoom region (f=sin($p/10t$))

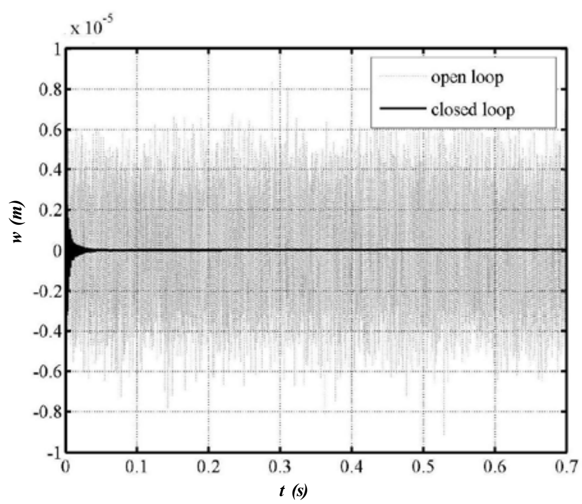


Fig. 4a: The panel vibration in the location of lump mass

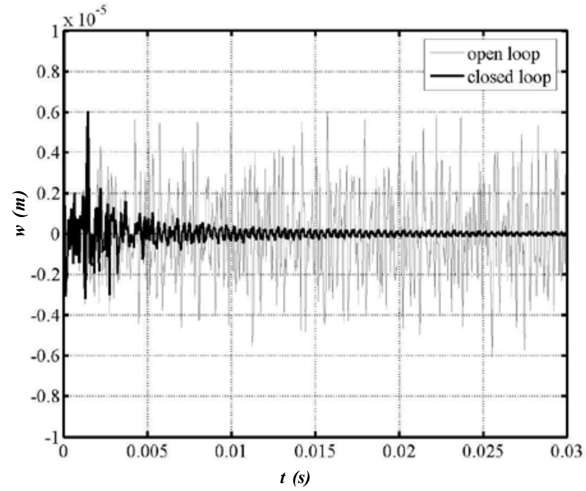


Fig. 4b: Zoom region (f=sin($p/10t$))

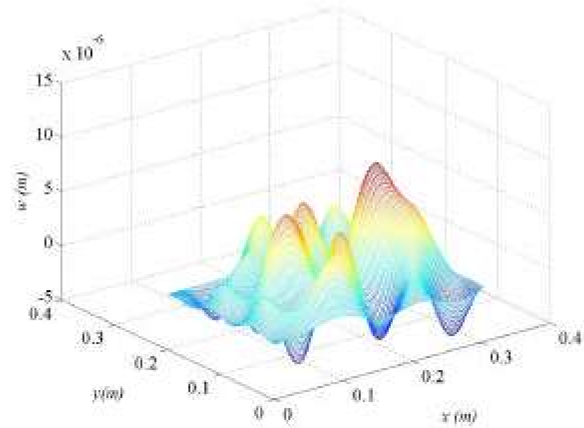


Fig. 5: The panel vibration when no voltage is applied to piezoelectric actuators (t=0.7s and f=sin(10pt))

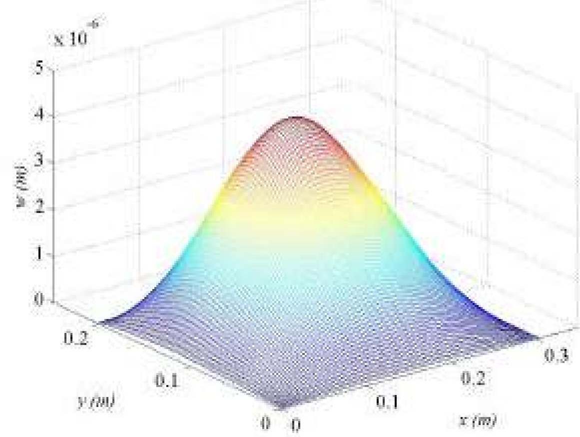


Fig. 6: The panel vibration when voltage is applied to piezoelectric actuators (t=0.7s and f=sin(10pt))

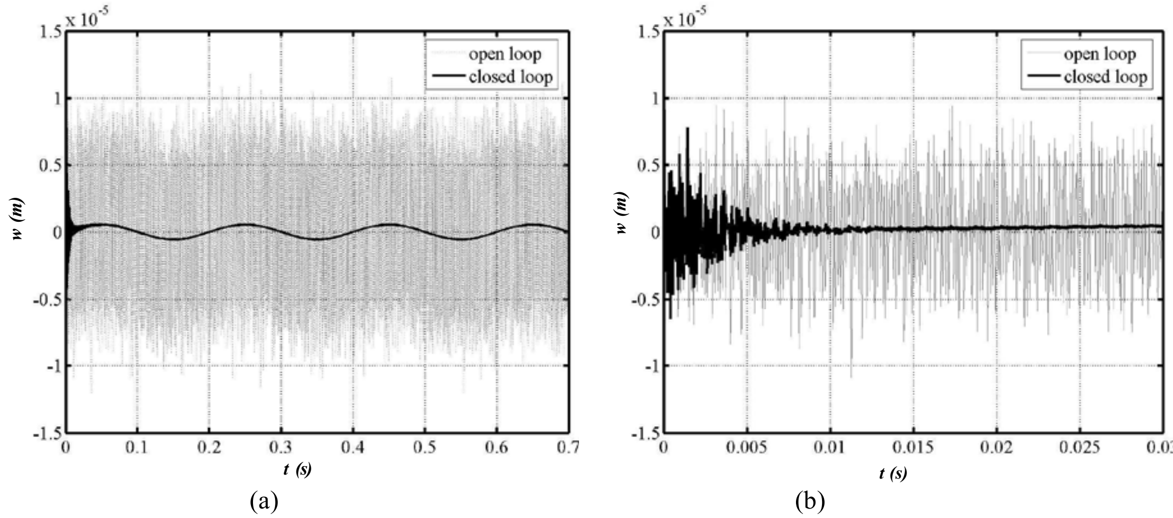


Fig. 7: a) The panel vibration in the location of point force, b) zoom region (f=sin(10pt))

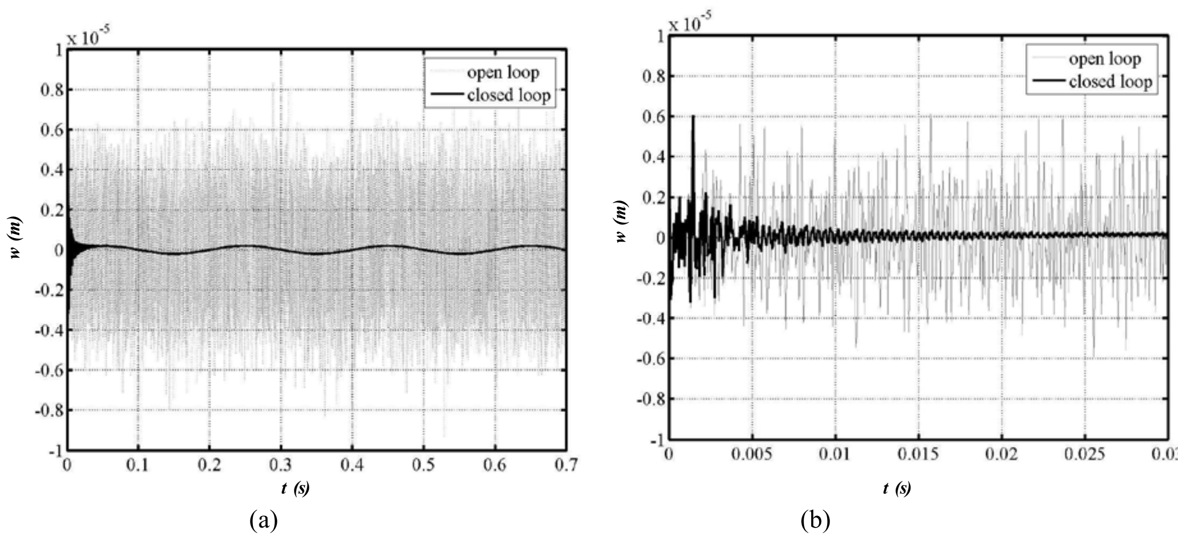


Fig. 8: a) The panel vibration in the location of lump mass, b) zoom region (f=sin(10pt))

CONCLUSION

An adaptive inverse dynamics control was used to suppress the vibration of a simply supported panel with equipments. The piezoelectric layers as sensors and actuators were attached to the panel. The governing equations of motion were derived using Lagrange-Rayleigh-Ritz method and the system was simulated. The simulation results show the capability of the controller.

REFERENCES

1. Han, J.H. and I. Lee, 1998. Analysis of composite plates with piezoelectric actuators for vibration control using layerwise displacement theory. Journal of Composites, 29: 621-632.
2. Chee, C., L. Tong and G. Steven, 2000. A mixed model for adaptive composite plates with piezoelectric for anisotropic actuation. Jornal of Computers and Structures, 77: 253-268.
3. Amant, Y.S. and L. Cheng, 2000. Simulations and experiments on active vibration control of a plate with integrated piezoceramics. Thin-Walled Structures, 38: 105-123.
4. Balamurugan, V. and S. Narayanan, 2001. Shell nite element for smart piezoelectric composite plate/shell structures and its application to the study of active vibration control. Finite Elements in Analysis and Design, 37: 713-738.
5. Benjeddou, A., J.F. Deu and S. Letombe, 2002. Free vibrations of simply-supported piezoelectric adaptive plates: An exact sandwich formulation. Thin-Walled Structures, 40: 573-593.

6. Mukherjee, A., S.P. Joshi and A. Ganguli, 2002. Active vibration control of piezolaminated stiffened plates. Composite Structures, 55: 435-443.
7. Benjeddou, A. and J.F. Deu, 2002. A two-dimensional closed-form solution for the free-vibrations analysis of piezoelectric sandwich plates. International Journal of Solids and Structures, 39: 1463-1486.
8. Raja, S., P.K. Sinha, G. Prathap and P. Bhattacharya, 2002. Influence of One and Two Dimensional Piezoelectric Actuation on Active Vibration Control of Smart Panels. Aerospace Science and Technology, 6: 209-216.
9. Gao, J.X. and Y.P. Shen, 2003. Active Control of Geometrically Nonlinear Transient Vibration of Composite Plates with Piezoelectric Actuators. Journal of Sound and Vibration, 264: 911-928.
10. Li, Y.Y., L. Cheng and P. Li, 2003. Modeling and vibration control of a plate coupled with piezoelectric material. Composite Structures, 62: 155-162.
11. Ma, K., 2003. Adaptive non-linear control of a clamped rectangular platewith PZT patches. Journal of Sound and Vibration, 264: 835-850.
12. Caruso, G., S. Galeani and L. Menini, 2003. Active vibration control of an elastic plate using multiple piezoelectric sensors and actuators. Simulation Modelling Practice and Theory, 11: 403-419.
13. Wang, S.Y., S.T. Quek and K.K. Ang, 2004. Dynamic stability analysis of finite element modelingof piezoelectric composite plates. International Journal of Solids and Structures, 41: 745-764.
14. Moita, J.M.S., I.F.P. Correia, C.M.M. Soares and C.A.M. Soares, 2004. Active control of adaptive laminated structures with bonded piezoelectric sensors and actuators. Computers and Structures, 82: 1349-1358.
15. Benassi, L. and S.J. Elliott, 2005. Global control of a vibrating plate using a feedback-controlled inertial actuator. Journal of Sound and Vibration, 283: 69-90.
16. Sloss, J.M., J.C. Bruch Jr, I.S. Sadek and S. Adali, 2005. Boundary optimal control of structural vibrations in an annular plate. Journal of the Franklin Institute, 342: 295-309.
17. Chein Lin, J. and M.H. Nien, 2005. Adaptive control of a composite cantilever beam with piezoelectric damping-modal actuators/sensors. Composite Structures, 70: 170-176.
18. Moita, J.M.S., C.M.M. Soares and C.A.M. Soares, 2005. Active control of forced vibrations in adaptive structures using a higher order model. Composite Structures, 71: 349-355.
19. Alessandroni, S., U. Andreaus, F. dell'Isola and M. Porfiri, 2005. A passive electric controller for multimodal vibrations of thin plates. Computers and Structures, 83: 1236-1250.
20. Lin, J., 2005. An Active Vibration Absorber of Smart Panel by Using a Decomposed Parallel Fuzzy Control Structure. Engineering Applications of Artificial Intelligence, 18: 985-998.
21. Moon, S.H. and J.S. Hwang, 2005. Panel Flutter Suppression with an Optimal Controller Based on the Nonlinear Model Using Piezoelectric Materials. Composite Structures, 68: 31-379.
22. Heidary, F. and M.R. Eslami, 2006. Piezo-control of forced vibrations of a thermoelastic composite plate. Composite Structures, 74: 99-105.
23. Robaldo, A., E. Carrera and A. Benjeddou, 2006. A unified formulation for finite element analysis of piezoelectric adaptive plates. Computers and Structures, 84: 1494-1505.
24. Moon, S.H., 2006. Finite Element Analysis and Design of Control System with Feedback Output Using Piezoelectric Sensor/Actuator for Panel Flutter Suppression. Finite Elements in Analysis and Design, 42: 1071 - 1078.
25. Ma, K. and M.N. Ghasemi-Nejhad, 2006. Adaptive Precision Positioning of Smart Composite Panels Subjected to External Disturbances. Mechatronics, 16: 623-630.
26. To, W.S. and T. Chen, 2007. Optimal Control of Random Vibration in Plate and Shell Structures with Distributed Piezoelectric Components. International Journal of Mechanical Sciences, 49: 1389-1398.
27. Zhang, H.Y. and Y.P. Shen, 2007. Vibration suppression of laminated plates with 1-3 piezoelectric fiber-reinforced composite layers equipped with interdigitated electrodes. Composite Structures, 79: 220-228.
28. Chein Lin, J. and M.H. Nien, 2007. Adaptive modeling and shape control of laminated plates using piezoelectric actuators. Journal of Materials Processing Technology, 189: 231-236.
29. Ebrahimi, F. and A. Rastgo, 2008. An analytical study on the free vibration of smart circular thin FGM plate based on classical plate theory. Thin-Walled Structures, 46: 1402- 1408.
30. Qiu, Z., H. Wu and C. Ye, 2009. Acceleration Sensors Based Modal Identification and Active Vibration Control of Flexible Smart Cantilever Plate. Aerospace Science and Technology, 13: 277-290.

31. Kapuria, S., P. Kumari and J.K. Nath, 2009. Analytical Piezoelasticity Solution for Vibration of Piezoelectric Laminated Angle-Ply Circular Cylindrical Panels. Journal of Sound and Vibration, 324: 832-849.
32. Qiu, Z., H. Wu and D. Zhang, 2009. Experimental Researches on Sliding Mode Active Vibration Control of Flexible Piezoelectric Cantilever Plate Integrated Gyroscope. Thin-Walled Structures, 47: 836-846.
33. Tavakolpour, A.R., M. Mailah, I.Z.M. Darus and O. Tokhi, 2010. Self-learning active vibration control of a flexible plate structure with piezoelectric actuator. Simulation Modelling Practice and Theory, 18: 516-532.
34. Thinh, T.I. and L.K. Ngoc, 2010. Static behavior and vibration control of piezoelectric cantilever composite plates and comparison with experiments. Computational Materials Science, 49: S276-S280.
35. Tokhi, O. and S. Veres, 2002. Active Sound and Vibration Control. The Institution of Electrical Engineers, London.
36. Canudas, C., B. Siciliano and G. Bastin, 1996. Theory of Robot Control. Springer Verlag London.