Smart Structures and Systems, *Vol. 11, No. 6 (2013) 623-635*
DOI:http://dx.doi.org/10.12989/sss.2013.11.6.623

Active vibration suppression of a 1D piezoelectric bimorph structure using model predictive sliding mode control

Byeongil Kim*[1], Gregory N. Washington[2] and Hwan-Sik Yoon[3]

[1]Powertrain CAE Team, Research & Development Division, Hyundai-Kia Motors 772-1, Jangduk-Dong, Hwaseong-Si, Gyeonggi-Do, Republic of Korea
[2]The Henry Samueli School of Engineering, University of California, Irvine 5200 Engineering Hall, Irvine CA 92697, USA
[3]Department of Mechanical Engineering, The University of Alabama, Box 870276, Tuscaloosa, AL 35487, USA

(Received January 3, 2012, Revised November 12, 2012, Accepted December 1, 2012)

Abstract. This paper investigates application of a control algorithm called model predictive sliding mode control (MPSMC) to active vibration suppression of a cantilevered aluminum beam. MPSMC is a relatively new control algorithm where model predictive control is employed to enhance sliding mode control by enforcing the system to reach the sliding surface in an optimal manner. In previous studies, it was shown that MPSMC can be applied to reduce hysteretic effects of piezoelectric actuators in dynamic displacement tracking applications. In the current study, a cantilevered beam with unknown mass distribution is selected as an experimental test bed in order to verify the robustness of MPSMC in active vibration control applications. Experimental results show that MPSMC can reduce vibration of an aluminum cantilevered beam at least by 29% regardless of modified mass distribution.

Keywords: active vibration control; piezoceramic actuator; 1D piezoelectric bimorph structure; model predictive sliding mode control

1. Introduction

Over the last few decades, a significant amount of research has been devoted to active vibration control of flexible structures. Several smart material based actuators have been utilized as active elements including piezoelectric patches (Yoon and Washington 2008), magnetorheological fluid (Rajamohan *et al*. 2011, Lara-Prieto *et al*. 2010), shape memory alloy (Suzuki and Kagawa 2010), electromagnetic actuator (Cheng and Oh 2009) and piezoelectric composite actuator (Suhariyono *et al*. 2008). Among these, piezoelectric patches (PZT, lead zirconate titanate) are commonly used for thin flexible structures since they are lightweight, compact, relatively inexpensive, having wide bandwidth and generating very high forces. There are a variety of practical applications employing PZT patches for active vibration and noise control. First, some automobile parts are thin-walled and they can be great transmitters of unexpected noise and vibration from many sources. Thus, there have been many efforts to apply PZT patches on automotive parts such as body panels

*Corresponding author, Senior Research Engineer, E-mail: kim.2057@osu.edu

http://www.techno-press.org/?journal=sss&subpage=8 ISSN: 1738-1584 (Print), 1738-1991 (Online)

(Bianchini 2008), powertrain oil pan (Wolff *et al*. 2007) and muffler system (Raju *et al*. 2005). Second, mechanical and aerodynamic vibration and noise from rotorcraft blade (Shevtsov *et al*. 2009), aircraft fin-tip (Rao *et al*. 2008) and aeroelastic flutter (Song and Li 2011) were treated by active control with PZT patches. Third, vibrations from actuator arms of hard disk drives were controlled by piezoelectric shunt circuits (Sun *et al*. 2009) and they could be utilized for detecting or fixing cracked beams (Lim and Soh 2011, Wu and Wang 2010). Also, Flexible structures could be utilized as energy harvesting devices and Makihara (2012) investigated a hybrid-type switching control method in order to mitigate the vibration as well as to improve the energy efficiency of the device. Li *et al*. (2012) developed a visualization technique to identify and attenuate vibration al energy of a 2D plate using constrained damping layer.

In order to make the active control system work as expected, proper control scheme should be applied or designed for each specific system. Piezoelectric shunt circuits have been utilized widely for passive control of vibrating systems. A single piezoelectric shunt circuit can manage only one vibration mode at a time (Kashani *et al*. 2001). Many efforts have been devoted to deal with more than one mode such as multiple piezoelectric shunt circuits (Trindade and Maio 2008), periodic piezoelectric shunt (Beck *et al*. 2011) and synchronized switch damping on voltage (Ji *et al*. 2009). However, these are still passive techniques that pre-tuning is required. Various basic and advanced active control algorithms also have been investigated such as dynamic hysteresis compensator (Nguyen and Choi 2010), positive position feedback (Mahmoodi *et al*. 2010, Fanson and Caughey 1990), acceleration feedback control (Mahmoodi and Ahmadian 2010, Preumont *et al*. 1993), full state feedback control (Yoon and Washington 2008), pole placement control (Sethi and Song 2008) and a hybrid control methodology including a variable structure control and a Lyapunov based controller (Mirzaee *et al*. 2011). Based on these studies, a general consensus has been reached that if an accurate structural model is available, flexible structures can be effectively controlled to attenuate the structural vibration in an active manner. However, many of the control algorithms exhibit performance degradation when the structure model does not reflect the structural dynamics accurately enough. For example, if dynamics of a structure changes over time due to structural modification or change, the control algorithm that was designed for the original structure may not perform as it used to be. To address this issue, robust control algorithms have been developed. For example, H-infinity control has been applied to suppress low-frequency modal vibrations of a thin plate without spillovers of the truncated modes (Xie *et al*. 2004).

In this paper, we propose to apply a recently developed robust control algorithm named Model Predictive Sliding Mode Control (MPSMC) to active vibration suppression of a cantilevered beam (Neelakantan 2005). This new control algorithm combines the optimal control characteristics of model predictive control with the robustness of sliding mode control. It has been shown that MPSMC can be successfully used in displacement control of piezoelectric actuators. However, no study has been conducted yet to apply MPSMC to active vibration control of flexible structures. In this study, a cantilevered aluminum beam is used as a target structure. The aluminum beam has two PZT actuators attached to both sides of the beam, a PZT sensor attached to one side of the beam, and an accelerometer attached at the free end of the beam. For the controller design, the beam was modeled in the state space using experimental data from an impact test. Then, a designed MPSMC controller is applied to the structure to attenuate vibration amplitude of the beam using the PZT sensor and actuators. In order to study the robustness of MPSMC in active vibration control, the same controller is applied to modified structures with added masses. The results show that the MPSMC can effectively suppress the structural vibration even if the structure parameters are changed from their original values.

2. Model predictive sliding mode control

Sliding mode control (SMC) is one of the most popular nonlinear control methods. SMC alters the dynamics of a nonlinear system by applying a high-frequency switching control so that the state trajectory follows a pre-defined discontinuity surface called the sliding surface in the state space (Young *et al*. 1999, Utkin *et al*. 1999). In this way, the motion of the system does not rely on its original dynamics but rather follows the dynamics of the desired sliding surface. The main advantage of SMC lies in its robustness and insensitivity to noise. Once the system reaches the sliding surface, it will have prescribed dynamics associated with desirable performance characteristics and the system response will be insensitive to disturbances or uncertainties in the system parameters. In Discrete-time Sliding Mode Control (DSMC) that is widely used in modern digital control, the control law is usually designed to enforce the system to reach the sliding surface at the very next sampling instant. However, this often results in saturation of the controller or overcompensation of the system. Therefore, it is desirable to develop an algorithm to relax the engagement of sliding mode at the very next time instance.

Model Predictive Control (MPC) is a robust discrete-time control methodology that explicitly uses the system model (Rossiter 2003). Its ability to produce optimal control action at each step to enhance tracking performance has been one of the main attractions of this methodology. By combining the model predictive control and the sliding mode control, a new control scheme called model predictive sliding mode control (MPSMC) was formulated (Neelakantan 2005). The idea is to enforce the system to reach the sliding surface in an optimal manner using the MPC strategy.

Consider the following discrete-time state space equation with a disturbance vector h_k

$$x_{k+1} = Ax_k + Bu_k + h_k \tag{1a}$$

$$y_k = Cx_k \tag{1b}$$

where A, B and C represent the system, input, and output matrices, and x_k and u_k are the system and control vectors, respectively. A new state error vector e_k is introduced by subtracting a desired reference vector r_k from the state vector x_k as in Eq. (2) for reference tracking problems. The system equations can then be rewritten with respect to e_k as Eq. (3(a)).

$$e_k = x_k - r_k \tag{2}$$

$$e_{k+1} = Ae_k + Bu_k + d_k \tag{3a}$$

$$d_k = Ar_k + h_k - r_{k+1} \tag{3b}$$

Note that the additional term d_k encompasses disturbance, nonlinearity, hysteresis, model uncertainties, and effects of the reference input. In the following derivation, it is more convenient to augment the system state by adding the control input u_{k-1} at the precedent instant to the state vector. Thus, the augmented state vector z_k becomes

$$z_k = \begin{bmatrix} e_k \\ u_{k-1} \end{bmatrix} \tag{4}$$

With the new state vector, the state equations can be rewritten as

$$\begin{bmatrix} e_{k+1} \\ u_k \end{bmatrix} = \begin{bmatrix} A & B \\ 0 & I \end{bmatrix}\begin{bmatrix} e_k \\ u_{k-1} \end{bmatrix} + \begin{bmatrix} B \\ I \end{bmatrix}\Delta u_k + \begin{bmatrix} I \\ 0 \end{bmatrix} d_k \tag{5}$$

or simply

$$z_{k+1} = \Gamma z_k + \Phi \Delta u_k + \Theta d_k \tag{6}$$

where $\Gamma = \begin{bmatrix} A & B \\ 0 & I \end{bmatrix}$, $\Phi = \begin{bmatrix} B \\ I \end{bmatrix}$, and $\Theta = \begin{bmatrix} I \\ 0 \end{bmatrix}$. In the new state equation, the difference in the control signal Δu_k defined as $\Delta u_k = u_k - u_{k-1}$, is used as the control input. With the new state vector, the desired sliding surface s_k can be defined as

$$s_k = G_s z_k \tag{7}$$

where G_s determines the dynamic characteristics of the sliding mode. By substituting Eq. (6) into Eq. (7), the sliding mode equation can be obtained. Then, by combining the sliding mode equations at the k+1-th through the k+N-th sampling instants, a prediction equation for the sliding mode can be derived as follows

$$s_{(k+1:k+N)} = \hat{\Gamma} z_k + \hat{\Phi}\Delta u_{(k:k+N-1)} + \hat{\Theta} d_{(k:k+N-1)} \tag{8a}$$

$$s_{(k+1:k+N)} = \begin{bmatrix} s_{k+1} & \cdots & s_{k+N} \end{bmatrix}^T \tag{8b}$$

$$\Delta u_{(k:k+N-1)} = \begin{bmatrix} \Delta u_k & \cdots & \Delta u_{k+N-1} \end{bmatrix}^T \tag{8c}$$

$$d_{(k:k+N-1)} = \begin{bmatrix} d_k & \cdots & d_{k+N-1} \end{bmatrix}^T \tag{8d}$$

and the matrices $\hat{\Gamma}$, $\hat{\Phi}$, and $\hat{\Theta}$ are defined as follows.

$$\hat{\Gamma} = \begin{bmatrix} G_s\Gamma \\ G_s\Gamma^2 \\ \vdots \\ G_s\Gamma^N \end{bmatrix}, \quad \hat{\Phi} = \begin{bmatrix} G_s\Phi & 0 & \cdots & 0 \\ G_s\Gamma\Phi & G_s\Phi & \cdots & 0 \\ \vdots & \vdots & \ddots & 0 \\ G_s\Gamma^{N-1}\Phi & G_s\Gamma^{N-2}\Phi & \cdots & G_s\Phi \end{bmatrix}, \quad \hat{\Theta} = \begin{bmatrix} G_s\Theta & 0 & \cdots & 0 \\ G_s\Gamma\Theta & G_s\Theta & \cdots & 0 \\ \vdots & \vdots & \ddots & 0 \\ G_s\Gamma^{N-1}\Theta & G_s\Gamma^{N-2}\Theta & \cdots & G_s\Theta \end{bmatrix} \tag{9a-c}$$

Note that $s_{(k+1:k+N)}$, $\Delta u_{(k:k+N-1)}$, and $d_{(k:k+N-1)}$ are arrays of the sliding mode, control input difference, and disturbance vectors up to N future time instants, respectively. In order to obtain an optimal vector for the control input difference, a cost function J is defined as

$$J = \left\| s_{(:)} \right\|^2 + \lambda \left\| \Delta u_{(:)} \right\|^2 \tag{10}$$

where λ is a weighting factor. Minimization of the cost function leads to the optimal solution for the array of the control difference vectors as follows.

$$\Delta u_{(:)} = -(\hat{\Phi}^T \hat{\Phi} + \lambda I)^{-1} \hat{\Phi}^T [\hat{\Gamma} z_k + \hat{\Theta} d_{(:)}] \tag{11}$$

Finally, the first vector at the k-th time instant is extracted from the array of the control difference vectors in Eq. (11) by multiplying it by a row vector as follows.

$$\Delta u_k = [I \quad 0 \quad \cdots \quad 0] \Delta u_{(:)} \tag{12}$$

Note that in Eq. (11), $d_{(:)}$ represents the future prediction of the disturbance that is generally unknown. A simple way of overcoming this difficulty is to obtain the previous value of the disturbance and assume that the disturbance remains the same for the next N time instants, which is basically an extended zero-order estimate. In this case, $d_{(:)}$ can be replaced by its estimate such that

$$d_{(:)}^{est} = \left[d_{k-1} \quad d_{k-1} \quad \cdots \quad d_{k-1} \right]^T \tag{13}$$

where $d_{k-1} = e_k - Ae_{k-1} - Bu_{k-1}$. A block diagram for model predictive sliding mode control system is shown in Fig. 1

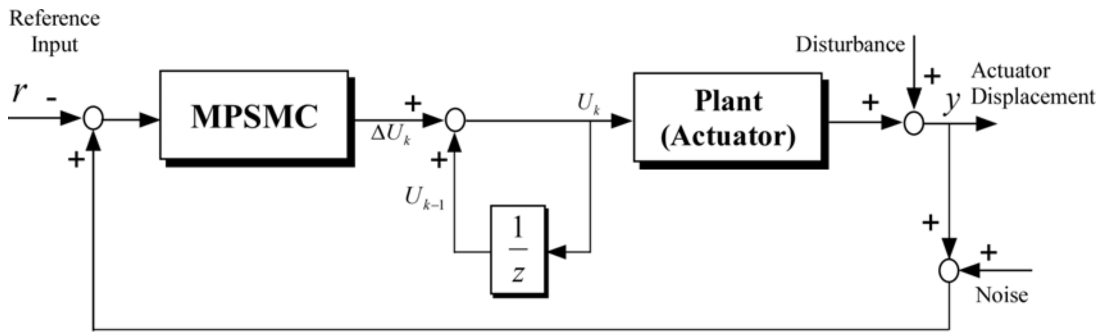


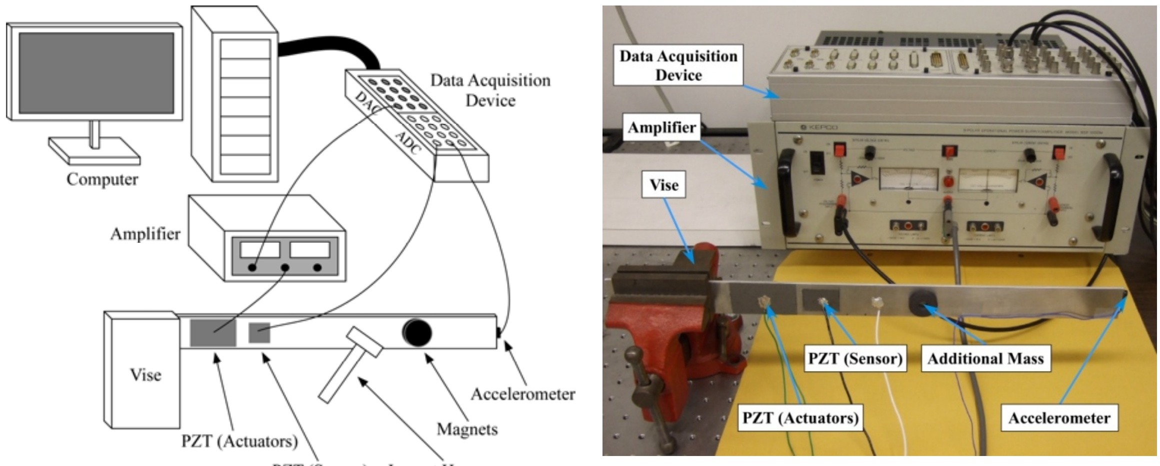
Fig. 1 General structure of model predictive sliding mode control system

This control methodology has been verified with the force phase of a two-stage actuation system (Neelakantan and Washington 2008). In the study, MPSMC exhibited superior performances in making a piezoceramic actuator follow step inputs. However, its performance in dynamic vibration suppression applications has not been studied until now.

3. Experimental setup

A schematic of the experimental setup used in this study is shown in Fig. 2(a). An aluminum beam, whose dimension is 40 cm × 3.8 cm × 0.33 cm, was clamped with a steel vise to form a cantilevered beam. On both sides of the aluminum beam, two 6.4 cm × 2.5 cm PZT (PSI-5A-S4-ENH) actuators were bonded at the same location so that they work as a bending actuator. Additionally, one 3.6 cm × 2.5 cm PZT sensor was attached to measure the beam strain for the input to the control system. Also, an accelerometer was attached at the free end of the cantilevered beam as a truth sensor. Finally, permanent magnets were prepared and attached to the beam to simulate the cases where the beam dynamics are modified.

For real-time control, a data acquisition device, dSPACE, was used to receive voltage signal from the PZT sensor and generate control signal calculated by the proposed MPSMC algorithm. The control signal is then amplified to a higher voltage signal and sent to the two PZT actuators. In order to obtain the system model, a system identification technique based on impulse response was utilized. An actual picture of the experimental setup is shown Fig. 2(b).



(a) (b)

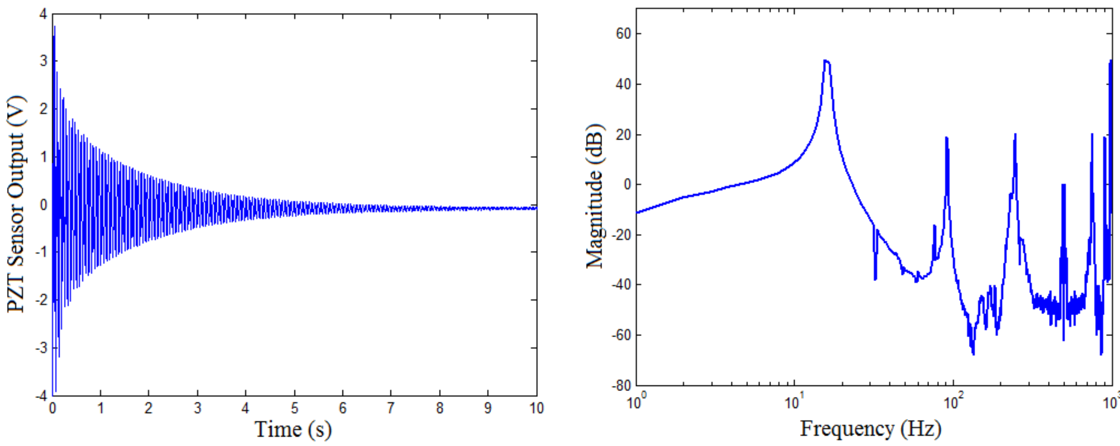
Fig. 2 Active beam experiment, (a) Schematic and (b) picture

4. System modeling

Although an accurate system model is desirable in the design of a model-based controller, it is not a stringent requirement for MPSMC. Due to robustness of the employed MPSMC, a *basic* model of the structure will suffice for the controller design purpose. Then, MPSMC can overcome moderate model inaccuracy or uncertainty and impose desirable dynamics to the controlled system.

For the system modeling, a system identification technique based on the frequency response was used (Kollar 1993). This technique utilizes an estimator called maximum likelihood estimator (MLE) to minimize a weighted cost function with the frequency response data of the system (Marquardt 1963, Björck 1996). Fig. 3 shows the impulse response of the beam in both time domain and frequency domain. In our implementation of the system identification, after an estimated system transfer function was obtained, coefficients of the transfer function were fine tuned so that it matches better with the measured frequency response. The results of the system identification are shown in Fig. 4 where frequency responses of transfer functions with three different approximations are compared with the experimental data. Among the three presented transfer functions, the fifth-order model was chosen to design the controller in this study. This was the result of a compromise between the simplicity of the controller design and accuracy of the transfer function to capture at least two lowest natural frequencies of the structure. The resulting fifth-order transfer function of the aluminum cantilevered beam is as follows.

$$G_{beam}(s) = \frac{100000s^3 + 2.092\times10^7 s^2 + 2.265\times10^{10} s + 2.265\times10^9}{s^5 + 71.22s^4 + 3.437\times10^5 s^3 + 2.102\times10^7 s^2 + 3.425\times10^9 s + 2.039\times10^{11}} \tag{14}$$



(a) (b)

Fig. 3 Measured voltage of the PZT sensor (impulse response) in (a) Time domain and (b) frequency domain

5. Experimental results

With the system model, several experiments were conducted on two different beam configurations: the baseline beam structure and modified beams with added masses. First, using the fifth-order system model in Eq. (14), a controller was designed based on MPSMC and control parameters were tuned to obtain the best result. Selected parameters such as the weighting factor in the cost function λ, the receding horizon or the number of future time instants N, and the sliding surface G_s are as follow.

$$\lambda = 8 \tag{15}$$

$$N = 5 \tag{16}$$

$$G_s = [1 \quad 1 \quad 0] \tag{17}$$

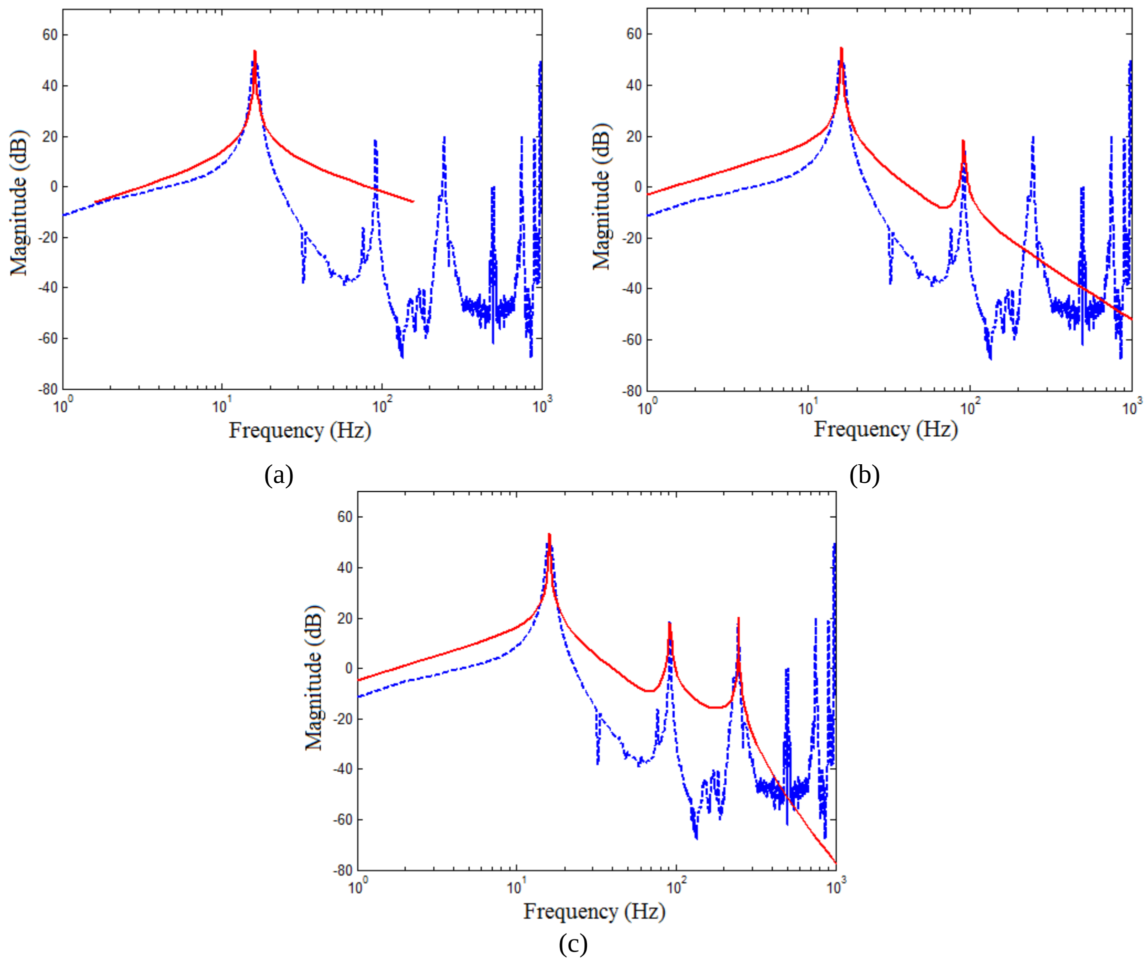


Fig. 4 Comparison of three different beam models in frequency domain. (a) Second order, (b) Fifth order and (c) Seventh order. Key: ----, measured frequency spectra ; ——, predicted frequency spectra

Fig. 5 shows the experimental results for the original baseline structure. Fig. 5(a) shows the PZT sensor output signals in time domain. In the open-loop case, it was seen that vibration exists until 6.3 seconds. However, when MPSMC control was applied, it was observed that the vibration stops at around 1.1 seconds. Fig. 5(b) compares the frequency responses of the open-loop and closed-loop cases. Peak reduction at dominant resonance frequencies can be observed in the plot.

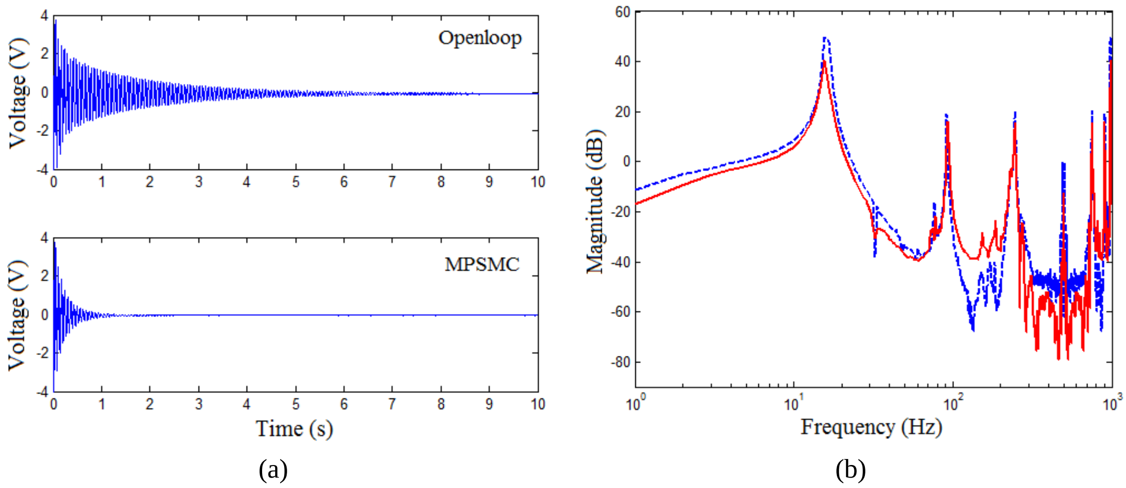


Fig. 5 Impulse response of the baseline structure in (a) Time domain and (b) frequency domain. Key: - - - -, uncontrolled; ——, controlled with MPSMC

Next, in order to test robustness of the controller to plant variation, additional masses (about 25% of the beam mass) were attached to the baseline structure at two different locations as shown in Fig. 6. In either case, the same controller developed for the baseline structure was applied to the modified structure without any change in the control parameters. One can test robustness of such a controller in this manner. Experimental results for the beam with an added mass at location 1 are shown in Fig. 7 and results for added mass at location 2 are presented in Fig. 8.

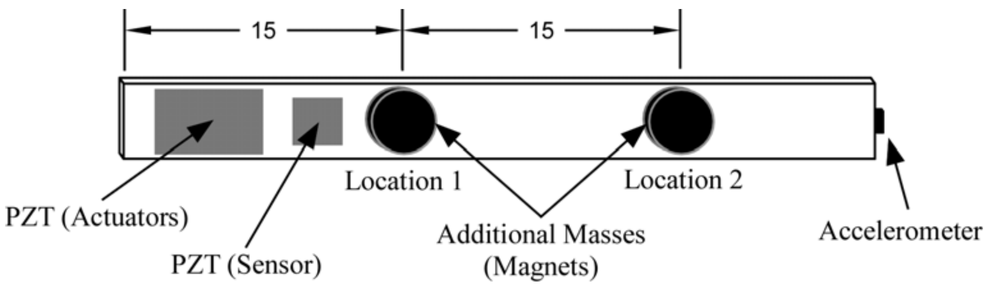


Fig. 6 Location of attached masses

As expected, when masses are attached to the baseline structure, the frequency responses show decrease in resonance frequencies. This effect is observed in both cases. When the MPSMC is applied, the time domain response of the modified structures for an impulse input shows significant reduction in the vibration settling time. Also, amplitudes of the resonance peaks were reduced by the application of the MPSMC controller when compared to the open-loop case regardless of the variation in the plant dynamics. With these results, it is verified that the MPSMC can effectively reduce vibration of the cantilevered beam even if there exist variations in the plant model. Performance of the MPSMC controller compared to the open-loop case is summarized in Table 1 for the three different structural configurations.

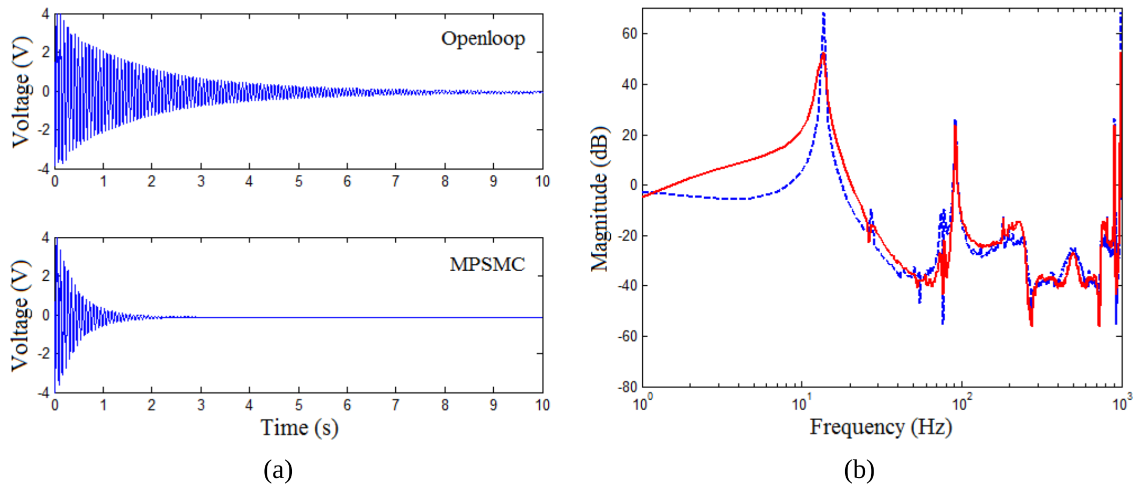


Fig. **7** Impulse response of the modified structure (mass at location 1) in (a) Time domain and (b) Frequency domain. Key: - - - - -, uncontrolled; ———, controlled with MPSMC

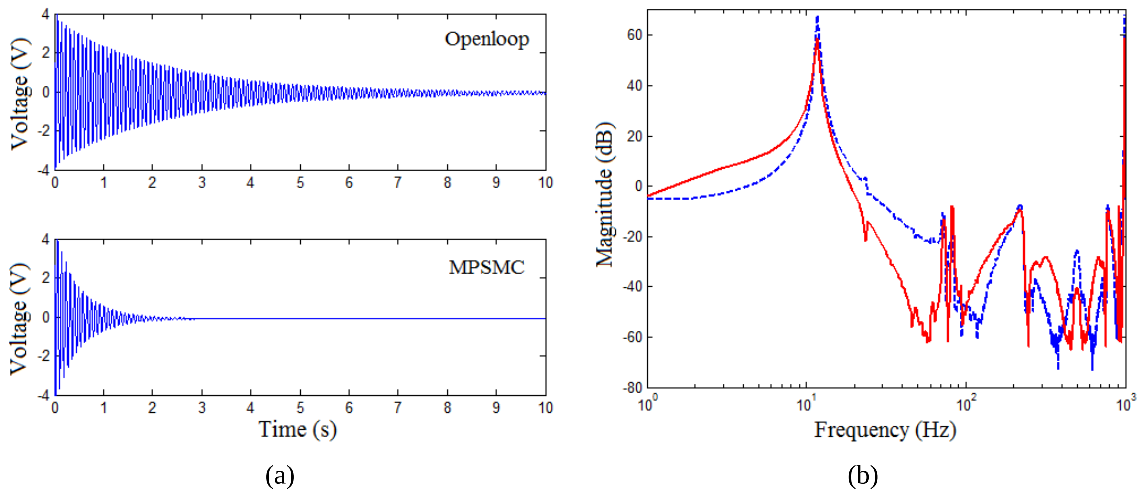


Fig. 8 Impulse response of the modified structure (mass at location 2) in (a) Time domain and (b) Frequency domain. Key: - - - - -, uncontrolled; ———, controlled with MPSMC

In Table 1, the benefit of applying MPSMC to the beam structure is clearly identified in time domain. For the baseline structure, the 2% settling time has been reduced from 6.3 seconds to 1.1 seconds with the application of the controller. The same controller could reduce the settling time of the modified structures similarly to the baseline structure with slightly less reduction. Since the controller was designed for the baseline structure, performance deterioration for the modified structures was expected and it was experimentally verified. Nonetheless, reduction of the settling time in all three cases was still significant, which implies robustness of the MPSMC to system parameter variations. In frequency domain, average amplitudes of first six dominant peaks were compared before and after the application of the controller. For the baseline structure, it was observed that the controller could reduce the average peak amplitude by 49% compared to the uncontrolled system. However, when the same controller was applied to the first modified system, the percent reduction of the average peak amplitude was 55%, which was greater than the baseline case. This rather unusual result can be explained by the high initial peak amplitudes of the modified system. Because the peak amplitudes of the uncontrolled modified systems were high, relatively small reduction in the absolute peak amplitudes shows up high when the percent reduction is compared. Reduction in the average peak amplitude for the second modified system was calculated to be 29%, which is the lowest in all three cases.

Table 1 Performance improvements with MPSMC in time domain and frequency domain responses

Structure type	Control type	2% settling time (sec)	Average peak amplitude (dB)
Baseline	Uncontrolled	6.3	25.37
	Controlled	1.1 (83% reduction)	12.94 (49% reduction)
Mass at location 1	Uncontrolled	8.4	16.77
	Controlled	1.9 (78% reduction)	7.63 (55% reduction)
Mass at location 2	Uncontrolled	9.7	13.62
	Controlled	2.3 (76% reduction)	9.73 (29% reduction)

6. Conclusions

In this paper, active vibration control of an aluminum cantilevered beam using a new control algorithm was presented. First, an aluminum beam with two PZT actuators and a sensor was prepared and the system model was obtained with frequency response. For the derived structural model, a controller was developed based on model predictive sliding mode control (MPSMC). When the developed controller was applied to the original baseline structure, it exhibited 83% reduction in settling time as well as 49% reduction of the peak amplitude in the frequency domain. The same controller was also applied to two different structure configurations modified with added masses at different locations. The results showed similar reductions in both settling time and peak amplitude in the frequency response, thus verifying robustness of the MPSMC based controller.

Although this study was focused on the application of MPSMC to a cantilevered beam, it is deemed that MPSMC can be applied equally well to any flexible structures. To investigate this, our future efforts will be devoted to application of MPSMC to 2D plate structures. It is expected that once the structure is modeled in discrete state space, the same design procedure outlined in this study can be applied. In addition, automation of control parameter tuning will also be investigated.

This will expedite application of MPSMC to a wide range of structural control problems with reduced tuning efforts.

Acknowledgements

The authors would like to acknowledge and thank the National Science Foundation for support of this research.

References

Beck, B.S., Cunefare, K.A., Ruzzene, M. and Collet, M. (2011), “Experimental analysis of a cantilever beam with a shunted piezoelectric periodic array”, *J. Intell. Mater. Syst. Struct.*, **22**(11), 1177-1187.

Bianchini, E. (2008), “Active vibration control of automotive like panels”, *Proceedings of the 2008 SAE BRASIL Noise and Vibration Conference*, SAE 2008-36-0576, Florianopolis, March.

Björck, A. (1996), *Numerical methods for least squares problems*, SIAM, Philadelphia, PA.

Cheng, T.H. and Oh, I.K. (2009), “Coil-based electromagnetic damper and actuator for vibration suppression of cantilever beams”, *J. Intell. Mater. Syst. Struct.*, **20**(18), 2237-2247.

Fanson, J.L. and Caughey, T.K. (1990), “Positive position feedback control for large space structures”, *AIAA J.*, **28**, 717-724.

Ji, H., Qiu, J., Badel, A. and Zhu, K. (2009), “Semi-active vibration control of a composite beam using an adaptive SSDV approach”, *J. Intell. Mater. Syst. Struct.*, **20**(4), 401-412.

Kashani, R., Mazdeh, A. and Orzechowski, J. (2001), “Shunt piezo damping of a radiating panel”, *Proceedings of the 2001 Noise and Vibration Conference*, SAE 2001-01-1576, Traverse City, May.

Kollar, I. (1993), “On frequency-domain identification of linear systems”, *IEEE T. Instrum. Meas.*, **42**, 2-6.

Lara-Prieto, V., Parkin, R., Jackson, M., Silberschmidt, V. and Kesy, Z. (2010), “Vibration characteristics of MR cantilever sandwich beams: experimental study”, *Smart Mater. Struct.*, **19**(1), 015005.

Li, K., Li, S. and Zhao, D. (2012), “Identification and suppression of vibrational energy in stiffened plates with cutouts based on visualization techniques", *Struct.Eng. Mech.*, **43**(3).

Lim, Y.Y. and Soh, C.K. (2011), “Fatigue life estimation of a 1D aluminum beam under mode-I loading using the electromechanical impedance technique”, *Smart Mater. Struct.*, **20**(12), 125001.

Mahmoodi, S.N. and Ahmadian, M. (2010), “Modified acceleration feedback for active vibration control of aerospace structures”, *Smart Mater. Struct.*, **19**(6), 065015.

Mahmoodi, S.N., Ahmadian, M. and Inman, D.J. (2010), “Adaptive modified positive position feedback for active vibration control of structures”, *J. Intell. Mater. Syst. Struct.*, **21**(6), 571-580.

Makihara, K. (2012), “Energy-efficiency enhancement and displacement-offset elimination for hybrid vibration control”, *Smart Struct. Syst.*, **10**(3), 193-207.

Marquardt, D. (1963), “An algorithm for least-squares estimation of nonlinear parameters”, *SIAM J. Appl. Math.*, **11**, 431-441.

Mirzaee, E., Eghtesad, M. and Fazelzadeh, S.A. (2011), “Trajectory tracking and active vibration suppression of a smart Single-Link flexible arm using a composite control design”, *Smart Struct. Syst.*, **7**(2), 103-116.

Neelakantan, V.A. (2005), *Modeling, design, testing and control of a two-stage actuation mechanism using piezoelectric actuators for automotive applications*, PhD Dissertation, Mechanical Engineering Department, The Ohio State University, Columbus, Ohio

Neelakantan, V. and Washington, G.N. (2008), “Model predictive control of a two stage actuation system using piezoelectric actuators for controllable industrial and automotive brakes and clutches”, *J. Intell.*

Mater. Syst. Struct., **19**, 845-857.
Nguyen, P.B. and Choi, S.B. (2010), “Open-loop position tracking control of a piezoceramic flexible beam using a dynamic hysteresis compensator”, *Smart Mater. Struct.*, **19**(12), 125008.
Preumont, A., Loix, N., Malaise, D. and Lecrenier, O. (1993), “Active damping of optical test benches with acceleration feedback”, *Machine Vib.*, **2**, 119-124.
Rajamohan, V., Sedaghati, R. and Rakheja, S. (2011), “Optimal vibration control of beams with total and partial MR-fluid treatments”, *Smart Mater. Struct.*, **20**(11), 115016.
Raju, B.B., Bianchini, E., Arata, J. and Roylance, M. (2005), “Improved performance of a baffle-less automotive muffler using piezoelectric materials”, *Proceedings of the SAE 2005 Noise and Vibration Conference and Exhibition*, SAE 2005-01-2353, Traverse City, May.
Rao, A.K., Natesan, K., Bhat, S.M. and Ganguli, R. (2008), “Experimental demonstration of H-inf control based active vibration suppression in composite fin-tip of aircraft using optimally placed piezoelectric patch actuators”, *J. Intell.Mater. Syst. Struct.*,**19**(6), 651-669.
Rossiter, J.A. (2003), *Model-Based Predictive Control: A Practical Approach*, CRC Press, London, UK.
Sethi, V. and Song, G. (2008), “Multimodal vibration control of a flexible structure using piezoceramic sensor and actuator”, *J. Intell. Mater. Syst. Struct.*, **19**(5), 573-582.
Shevtsov, S., Soloviev, A., Acopyan, V. and Samochenko, I. (2009), “Helicopter rotor blade vibration control on the basis of active/passive piezoelectric damping approach”, *Proceedings of the PHYSCON 2009*, Catania, September.
Song, Z.G. and Li, F.M. (2011), “Active aeroelastic flutter analysis and vibration control of supersonic beams using the piezoelectric actuator/sensor pairs”, *Smart Mater. Struct.*, **20**(5), 055013.
Suhariyono, A., Goo, N.S. and Park, H.C. (2008), “Use of lightweight piezo-composite actuators to suppress the free vibration of an aluminum beam”, *J. Intell. Mater. Syst. Struct.*, **19**(1), 101-112.
Sun, H., Yang, Z., Li, K., Li, B., Xie, J., Wu, D. and Zhang, L. (2009), “Vibration suppression of a hard disk driver actuator arm using piezoelectric shunt damping with a topology-optimized PZT transducer”, *Smart Mater. Struct.*, **18**(6), 065010.
Suzuki, Y. and Kagawa, Y. (2010), “Active vibration control of a flexible cantilever beam using shape memory alloy actuators", *Smart Mater. Struct.*, **19**(8), 085014.
Trindade, M.A. and Maio, C.E.B. (2008), “Multimodal passive vibration control of sandwich beams with shunted shear piezoelectric materials”, *Smart Mater. Struct.*, **17**(5), 055015.
Utkin, V., Guldner, J. and Shi, J. (1999), *Sliding Mode Control in Electromechanical Systems*, Taylor & Francis, Philadelphia, PA.
Wolff, K., Lahey, H.P., Nussmann, C., Nehl, J., Wimmel, R., Siebald, H., Fehren, H., Redaelli, M. and Naake, A. (2007), “Active noise cancellation at powertrain oil pan”, *Proceedings of the SAE 2007 Noise and Vibration Conference and Exhibition*, SAE 2007-01-2422, St. Charles, May.
Wu, N. and Wang, Q. (2010), “Repair of vibrating delaminated beam structures using piezoelectric patches”, *Smart Mater. Struct.*, **19**(3), 035027.
Xie, S.L., Zhang, X.N., Zhang, J.H. and Yu, L. (2004), “H-inf robust vibration control of a thin plate covered with a controllable constrained damping layer”, *J. Vib. Control*, **10**, 115-134.
Yoon, H.S. and Washington, G. (2008), “Active vibration confinement of flexible structures using piezoceramic patch actuators”, *J. Intell. Mater. Syst. Struct.*, **19**(2), 145-155.
Young, K.D., Utkin, V.I. and Ozguner, U. (1999), “A control engineer’s guide to sliding mode control”, *IEEE T. Contr. Syst. T.*, **7**, 328-342.

CC