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Active Vibration Control of Piezoelectric Composite Plates Using Gain Scheduling Method

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ABSTRACT

This paper presents an active vibration control methodology to suppress the vibration of a square laminated composite plate. The plate undergoes large deflection, where the dynamics of the system is nonlinear. The main layer of the composite plate is made of a graphite/epoxy composite (T300/976) with angle orientations of [-45/45/-45/45], with a total thickness of 1mm, in addition to two piezoelectric layers of type PZT G1195 N, each with a thickness of 0.1 mm. The two piezoelectric layers cover the entire area of the upper and lower surfaces of the laminated composite plate where the upper layer senses vibration and feeds it back to the controller, whereas the lower layer acts as an actuator which receives a voltage signal from the controller. The proposed control method involved using a Proportional-Integral-Derivative (PID) controller with a gain scheduling strategy where the system nonlinearity is addressed by changing the controller parameters momentarily, with optimal tuned parameters selected according to the dynamic characteristics during the vibration of the plate. The numerical model of the laminated composite plate is formulated using the finite element method (FEM) for large deflection vibrating composite plates. The controller performance was tested by simulating the vibration of the plate under the action of the controller for four cases with different vibration modes, and the results showed the effectiveness of the proposed PID controller with gain scheduling strategy.

1. INTRODUCTION

Composite laminated structures have been theoretically and experimentally proven to possess high strength and stiffness, making them highly demanded for aerospace and other various applications. Therefore, many researchers have investigated this fact over the last decades. Also, composite materials have good resistance to corrosion and fatigue characteristics. These materials can change the nature of strength by adjusting the fiber orientation or manipulating the sequence of fiber and matrix materials. Laminated composites have another advantage, which is the ability to design the system and the required mechanical properties before manufacture. Moreover, the mechanical properties of laminated materials depend on the orientation of fibers and the thickness of the lamina [1, 2]. Consequently, the lamina needs to be designed to meet the specific requirements of each particular application, which results in maximizing the benefits from the directional properties of materials. Additionally, it is crucial to accurately analyze and model the laminated structures based on their design sensitivity, fiber orientation, geometry, and dimensions [3, 4].

However, these structures are lightweight, which makes them prone to mechanical vibration, and this issue has been studied extensively in the last two decades. To address this issue, various vibration control methodologies have been developed to suppress vibration in composite structures. In general, vibration control systems are of two types: passive and active. The passive control method involves adding inertial, elastic, and damping components, and does not require an external energy source. Passive vibration control was an early method used to reduce the vibration by attaching resonators to the system at a low cost [5]. The passive control system can be designed by adjusting system characteristics elements where the vibration energy can be absorbed or isolated [6]. To improve the work of the absorber, a nonlinear approach was suggested to dampen vibration energy, utilizing a nonlinear energy sink (NES). However, the passive control method has limitations and is not suitable for advanced applications. Therefore, active control methodologies have been developed recently, which utilize an external energy source to interact with the system vibration energy [7]. The effectiveness of the active control method has become a research focus in recent decades. In the active control system, the system state is measured via a sensor and processed by a control model, which generates a control signal that can be executed by the actuator [8]. This flexibility makes the active control method preferable due to its strong applicability. Piezoelectric material are excellent candidates that can be effectively implemented in the active control systems since it can be used either as sensor or actuators. Due to their responsiveness, high precision, electromechanical efficiency, and compactness, piezoelectric actuators and sensors are commonly utilized for active vibration control [9]. The measured or sent signals are processed through control algorithms designed to suppress undesired vibrations. Piezoelectric materials were used effectively to control the vibrations in plates and cylindrical shells, and more extensive investigations were focused on vibration suppression of cylindrical shells by enhancing the performance of piezoelectric actuators [10]. The use of piezoelectric materials can be classified into two categories: optimal patch location and advanced control algorithms [11]. The optimal patch location approach is suitable when the nature of the application and geometry allows for the placement of piezoelectric actuators at fixed ends, such as in the vibration control of cantilever beams where low-frequency modes are dominant [12]. Numerical dynamics analyses, such as the finite element method, or experimental testing are typically used to determine the optimal placement of sensors and actuators [9, 13]. On the other hand, the advanced control algorithm approach is preferred for structures involving higher frequencies and complex mode shapes. Common control algorithms for linear smart structures include positive position feedback (PPF) and proportional-derivative (PD) control. For nonlinear structures with inherent uncertainties, adaptive control methods are used to deal with high-frequency dynamics [14-16]. The flutter velocity in laminated composite plates can be improved by 1.15 to 1.2 times when using piezoelectric actuators [17], and flutter is successfully suppressed by using PZT actuators at optimal locations [18]. On the other hand, multi-layered actuators have been proven to be more efficient compared to conventional stack actuators in controlling the vibration in aeroelastic applications [17]. Energy dissipation is another form of vibration control method, where the proper arrangement of PZT materials on laminated composite wings has been proven to be an effective approach to reduce vibration and ensuring aeroelastic stability [19]. Nonlinear Model Predictive Control (NMPC) is one of the most powerful strategies that can be effectively utilized to control the vibration in composite plates. The concept of NMPC is based on predicting the future behavior of the system dynamics. Since predicting the behavior of a nonlinear system is difficult, NMPC can utilize a system identification algorithm that runs online during the control process. In our previous works [20-22], NMPC has shown excellent ability to control systems with strong nonlinearity. However, Model Predictive Control (MPC) is computationally expensive and requires fast computers to perform real-time simulations.

Moreover, there are several difficulties and challenges associated with the active vibration control method for laminated composite plates, including but not limited to nonlinearities [23] and complex dynamics [24], material heterogeneity [25], and the integration of sensors and actuators without affecting structural integrity [26]. The level of challenges increases with the need for computational effort for real-time control, in addition to the need for reliable control strategies that can respond to different operation conditions and uncertainties. Recently, several advancements are proposed to address these challenges with adaptive control systems, which are capable of adjusting control parameters in real-time to handle changes in system dynamics. However, these systems require complex design and are harder to implement compared to traditional control methods, in addition to having higher computational requirements [21], which is considered a drawback for real-time applications. However, the aforementioned vibration control methods are suitable for systems with linear or weakly nonlinear characteristics. There are more sources of nonlinearity in laminated composite plates, such as material properties, geometric factors, boundary conditions, contact behavior, and loading conditions. The presence of any sources of nonlinearity results in complex behavior in the dynamics of composite structures, making the prediction of future dynamics more complicated and thus, controlling the vibration of nonlinear structures more difficult. However, PID controllers (Proportional-Integral-Derivative) are widely used in several applications due to their simplicity, fast response, and effectiveness. PID controllers can be easily tuned to achieve the control goal by adjusting their proportional, integral, and derivative parameters. Nevertheless, PID controllers show excellent performance with linear systems, but the performance decreases when the system has strong nonlinearity [14]. Nevertheless, the PID controller can be implemented in nonlinear systems by dividing the system response into several nonlinearity regions and approximating each region to a linear system within that region. Hence, a PID controller can be implemented in each nonlinear region independently, and the operation of the system will have a set of PID controllers. The control system switches based on each operating region [27, 28]. This control method is known as the gain scheduling strategy.

In this work, we implement the gain scheduling methodology to suppress the nonlinear vibration of laminated composite plate undergoing large deflections which presents significant challenges due to the inherent nonlinearities. Two piezoelectric materials layers cover the upper and lower surface of the plate where one layer is utilized as sensor and the other acts as an actuator. Utilizing gain scheduling control method allow that the dynamics of the system can be controlled under varying operating conditions. When the system characteristics change, the controller parameters are adjusted accordingly over various scenarios, which allows for the dealing of nonlinearities and uncertainties in the system. This methodology inherits the fast response of the PID controller's nature, in addition to its powerful ability to control systems with strong nonlinearity, making it an excellent control method with more advantages compared to other active control methods.

2. NUMERICAL MODEL

To accurately describe the dynamics of the composite plate, a numerical model is required to determine the displacement field momentarily. In this section, we introduce the numerical model of a composite plate consisting of n-layers, where each lamina can be oriented at an arbitrary angle θ with respect to the x-axis of the coordinate system. Two additional layers are added at the upper and lower surfaces, each consisting of piezoelectric materials. The upper layer is implemented as sensor and the lower layer as an actuator as shown in Figure 1. It has been proven that FEM is a powerful numerical tool used for analyzing the dynamics of laminated composite plates. FEM requires rigorous consideration of mesh quality, boundary conditions, material modeling, computational tools, and user expertise to reduce its limitations and achieve accurate results. Nevertheless, in this work. The finite element method (FEM) is implemented to obtain the first-order shear deformation to determine the displacement field where an isoparametric element is implemented for the nonlinear model due to its consistent interpolation, geometric flexibility, higher-order accuracy, computationally efficient numerical integration, in addition to its versatility and compatibility to different material models [29]. The plate is discretized into number of finite elements, where each element involved displacement vector $\{q\}$, in addition to the potential vector $\{\phi\}$ and the variables field and shape functions for the element geometry are given as:

$$\{q\} = \sum_{i=1}^{NN}[N_i]\{q\}_i,\ \{\phi\} = \sum_{i=1}^{NN}[N_{\phi i}]\{\phi_i\},\ x = \sum_{i=1}^{NN}[N_i]x_i \text{ and } y = \sum_{i=1}^{NN}[N_i]y_i \quad (1)$$

where, $[N_{\phi i}]$ and $[N_i]$ are the shape functions for the i^{th} node, $\{q\}_i$ is the vector of displacements to be calculated for the i^{th} node and NN is the number of nodes per each element. By using Hamilton's principle, the nonlinear finite element discretization can be obtained as follows [30]:

$$\begin{bmatrix} M & 0 \\ 0 & 0 \end{bmatrix}\begin{Bmatrix} \ddot{q} \\ \ddot{\phi} \end{Bmatrix} + \begin{bmatrix} K_q & K_{q\phi} \\ K_{q\phi}^T & K_{\phi\phi} \end{bmatrix}\begin{Bmatrix} q \\ \phi \end{Bmatrix} = 0 \quad (2)$$

where,

$$K_q = [K + \gamma K_c] + \frac{1}{2}[K_{1nl}] + [K_{2nl}] + \frac{1}{2}[K_{3nl}],$$
$$[K_{q\phi}] = [K_1] + [K_{4nl}],$$
$$[K_{\phi q}] = [K_1]^T + \tfrac{1}{2}[K_{5nl}], \text{ and}$$
$$[K_{\phi\phi}] = [K_2].$$

Also, Eq. (2) is usually introduced as uncoupled form as [30]:

$$[M]\{\ddot{q}\} + [K_q]\{q\} + [K_{q\phi}]\{\phi\} = 0, \text{ and}$$
$$[K_{q\phi}]^T\{q\} + [K_{\phi\phi}]\{\phi\} = 0 \quad (3)$$

Which can be simplified further to the standard vibration equation as follows:

$$[M]\{\ddot{q}\} + [\bar{K}]\{q\} = 0 \quad (4)$$

where, $[\bar{K}] = [K_q] - [K_{q\phi}][K_\phi]^{-1}[K_{q\phi}]^T$.

Assuming the composite plate vibrates with the fundamental natural frequency ω, Eq. (4) can be simplified to basic form of to nonlinear generalized eigenvalue problem as follows:

$$[\bar{K}]\{q\} = \omega^2[M]\{q\} \quad (5)$$

where, ω^2 is the eigenvalue, which can be obtained by solving Eq. (5) by using direct iterative method. In the initial step the nonlinear terms $[K_{1nl}]$ to $[K_{5nl}]$ are set to zero and the initial stiffness matrix is obtained, and the resulted modes then used as initial stiffness to the next step. This procedure will continue till the eigenvalues converged to an accepted error ($\epsilon = 10^{-3}$). The corresponding value of ω will be considered nonlinear natural frequency of the laminated composite plate. Also, the proportional damping matrix is $[C] = \alpha[M] + \beta[\bar{K}]$, where: α and β are the proportional parameters.

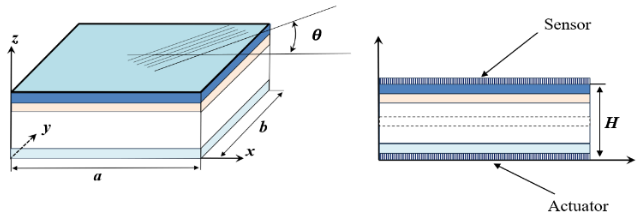

Figure 1. Laminated piezoelectric composite plate

2.1 Active control system

In order to obtain the numerical solution directly, it is necessary to decouple the variables in Eq. (4). Presenting the modal analysis matrix [P] will achieve this goal for the fundamental mode of vibration, where [P] is the matrix of eigenvectors and hence $\{q\} = [P]\{u\}$. The decoupled equations now can be implemented to design the controller. The conventional Proportional-Integral-Derivative (PID) controller is commonly used in industry because it can be designed in simple steps, where only three parameters are required to be tuned to meet design specifications. However, although the PID controller is effective and efficient in linear systems, it has many limitations when it is applied to nonlinear systems. Fortunately, characterizing the system nonlinearities for each input, allows to map the stability limits during the operating of the system. Thus, the PID parameters can be adjusted according to each operating region, where the controller invokes suitable predetermined PID parameters, this process is known as gain scheduling. Therefore, the controller can achieve effective control within a wide range of varying system dynamics during the operation.

The PID method is to present the control vector $u(t)$ as a summation of the proportional, integral and derivative of the vector of the output error *e(t)* as follows:

$$u(t) = K_p e(t) + K_i \int e(t)dt + K_d \frac{de(t)}{dt} \quad (6)$$

Also, the discrete form of PID controller can be written as follows:

$$\Delta u(k) = \left(K_p + \frac{K_p \cdot \Delta t}{\tau_I} + \frac{K_p \cdot \tau_D}{\Delta t}\right) \cdot e_k + \left(-K_p - \frac{2 \cdot K_p \cdot \tau_D}{\Delta t}\right) \cdot e_{k-1} + \frac{K_p \cdot \tau_D}{\Delta t} \cdot e_{k-2} \quad (7)$$

where, $\Delta u(k)$ is the change in control input, e_k is the error between measured output and setpoint, and K_p, τ_I, and τ_D are the controller parameters to be determined by using Ziegler-Nichols closed loop tuning method. The PID controller is shown in Figure 2. In systems with gain scheduling algorithm, the switching mechanism has a significant role in adjusting controller parameters based on the system nonlinearity. In the laminated composite plate, the measured deflection is to be used to identify system nonlinearity during vibration. The vibration in a laminated composite plate is usually considered nonlinear when ($\delta/h > 0.1$) where δ the deflection and (h) is the total thickness [31]. In this study, the system nonlinearity is devided into three regions namely, region 1 where the system is linear ($\delta/h < 0.1$), region 2 where the system has weak nonlinearity ($0.15 > \delta/h > 0.1$), region where the system is strongly nonlinear (δ/h > 0.15). where each region is associated with specific controller gains. As the system transitions between these regions, the switching mechanism updates the controller parameters for optimal performance during different operating conditions.

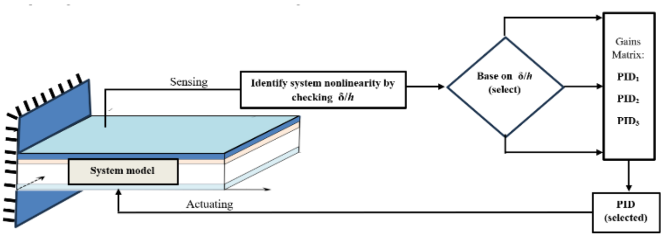

Figure 2. Feedback control loop of PID controller with switching mechanism for gain scheduling

3. NUMERICAL SIMULATION

The proposed PID with gain scheduling controller is tested numerically by developing MATLAB code to simulate the plate response and the controller action. The plate undergoes nonlinear vibration in the first principal modes, where the upper layer senses vibration and feeds it back to the controller. The controller then applies voltage to the lower layer, which acts as an actuator. Thus, the piezoelectric layers cover the entire area of the upper and lower surfaces of the laminated composite plate. Initially, the plate is excited by applying a static deflection to the free tip of the center line. The numerical model tested a square laminated plate with dimensions of 200×200 mm, where the main layer is made of a graphite/epoxy composite (T300/976) with angle orientations of [-45/45/-45/45]. The total thickness of the laminated plate is 1 mm, in addition to two piezoelectric layers of type PZT G1195 N, each with a thickness of 0.1 mm. The material properties of the components for the laminated composite plate are listed in Table 1.

Table 1. Material properties for PZT G1195N and T300/976 graphite/epoxy composites [32]

Properties	PZTG1195N	T300/976
Young's Moduli (GPa)		
E_{11}	63.0	150
$E_{11} = E_{33}$	63.0	9
Poisson's ratio		
v_{23}	0.3	0.3
$v_{12} = v_{13}$	0.3	0.3
Shear moduli (GPa)		
G_{23}	24.2	7.1
$G_{12} = G_{13}$	24.2	2.5
Density ρ(kg/m^3)	7600	1600
Piezoelectric constants (m/V) $d_{31} = d_{32}$	254×10^{-12}	
Electrical permittivity (F/m)		
$\varepsilon_{11} = \varepsilon_{22}$	15.3×10^{-9}	
ε_{33}	15.0×10^{-9}	

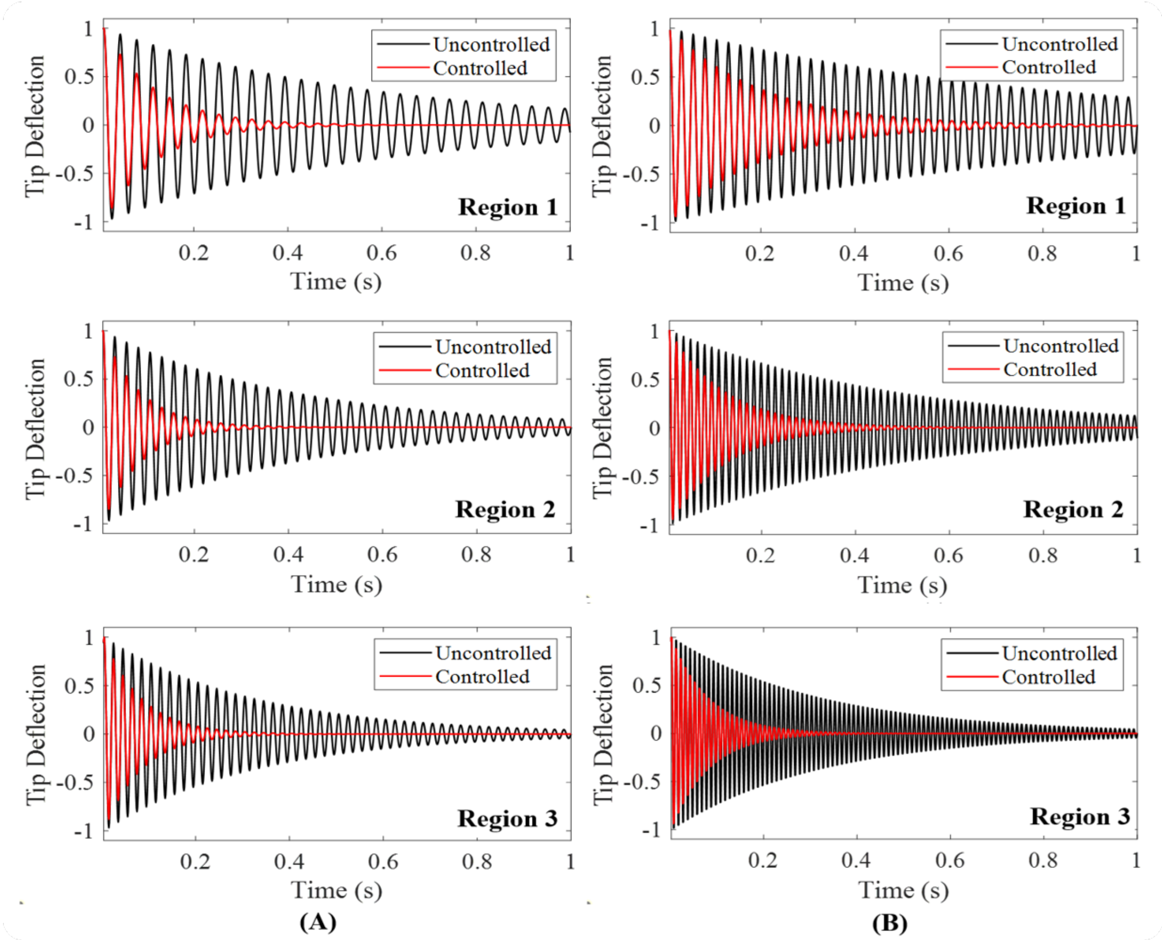

Figure 3. Normalized deflections of tip at the centreline of the plate with different nonlinearity regions (A) First modes (B) Second modes

The parameters (K_p, τ_I, and τ_D) of the suggested PID controller with gain scheduling are tuned according to the vibration characteristics of the plate and the deflection index (δ/h) for system nonlinearity. The tuning process includes numerical simulations to obtain the response of the system, where MATLAB is used to simulate the system response and determine the optimal parameters for each region of nonlinearity. The vibration characteristics of the composite plate is divided into regions, each region is identified based on the strength of nonlinearity (deflection and vibration mode), where a set of parameters (K_p, τ_I, and τ_D) are predetermined such that the controller selects the suitable control parameters according to the region of nonlinearity. The MATLAB simulations showed the effectiveness of the gain scheduling method in maintaining optimal control results. The nonlinearity regions are divided into three regions namely: region 1 where the system is linear and ($\delta/h < 0.1$), region 2 where the system has weak nonlinearity ($\delta/h = 0.11$), and the third region where the system is highly nonlinear ($\delta/h > 0.2$). The tuned parameters for each region are obtained as follows; Region 1 ($K_p = 1.98$, $\tau_I = 0.02$, and $\tau_D = 0.00012$), Region 2 ($K_p = 2.15$, $\tau_I = 0.0351$, and $\tau_D = 0.000303$), and Region 3 ($K_p = 2.831$, $\tau_I = 0.0511$, and $\tau_D = 0.000445$).

Figure 3 shows the results from the numerical simulation for the vibration suppression of the laminated composite plate. The PID controller was first tested to suppress the vibration of the plate when it oscillated with its first mode (Figure 3 (A)) and its second mode (Figure 3 (B)). In both tests, at small deflections, the system is linear (region 1), and the PID controller effectively suppresses the vibration in a relatively short time. Similarly, testing the controller in regions 1 and 2 obtained similar results and controller effectiveness; however, the frequency increases as the deflection increases toward the nonlinear region. This is due to the effect of stiffening which occurs when a plate is subjected to large deformation. Hence, as deflection increases, the restoring force tends to become more nonlinear, leading to an increase in the natural frequency of the system [33]. Other tests are conducted to evaluate the performance of the PID controller for a combination of the first three and six modes, as depicted in Figure 4.

The results showed the same trend of the frequency-deflection relationship, and the PID controller was able to suppress the vibration. However, tuning the PID parameters becomes more challenging when higher modes are included. The results showed good vibration control and system stability under different operating conditions, effectively suppressing vibration during the oscillation of the composite plate.

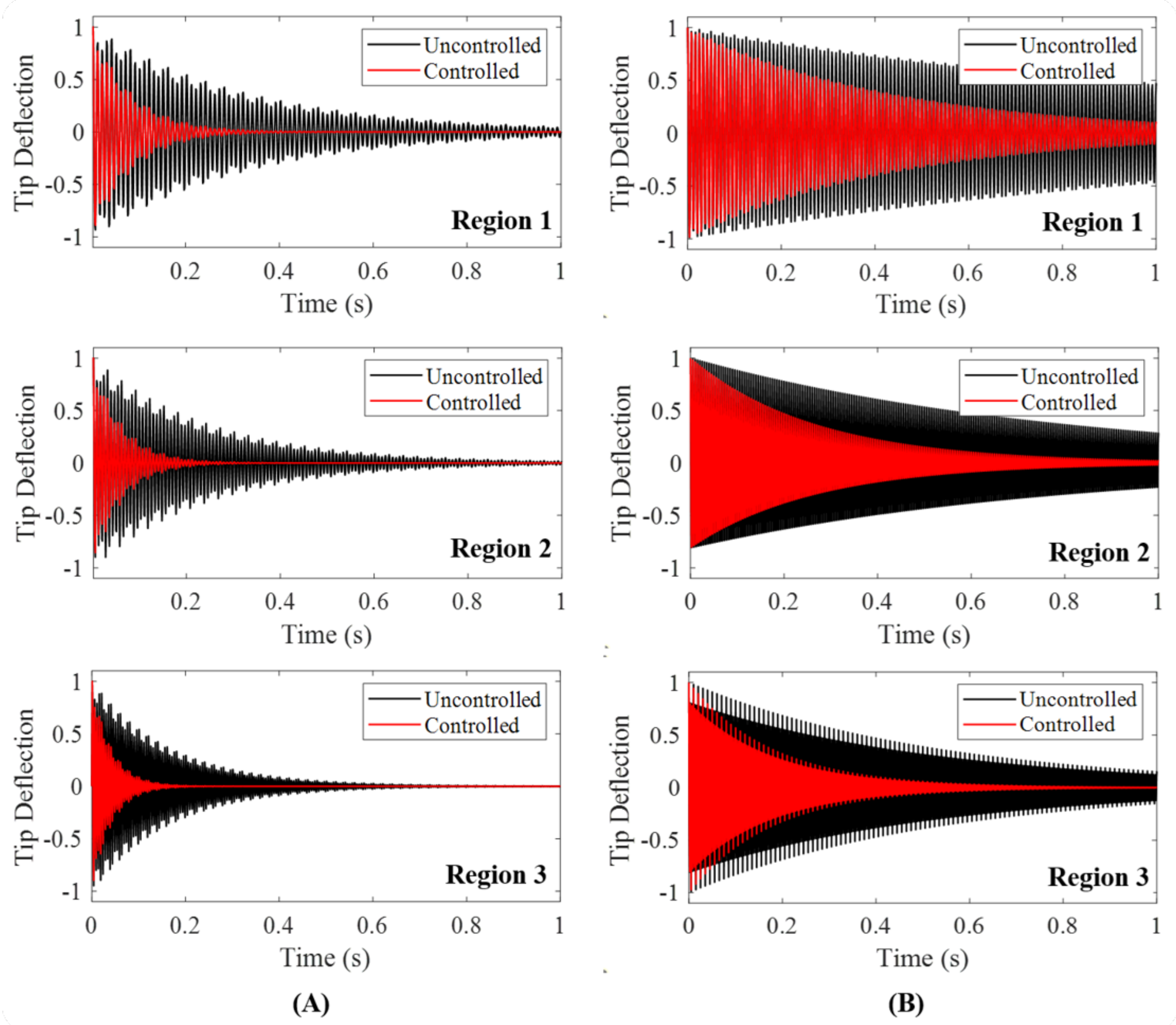

Figure 4. Normalized deflections of tip at the centreline of the plate with different nonlinearity regions (A) Combination of first six modes (B) Combination of first three modes

4. CONCLUSIONS

This paper studies the PID with gain scheduling vibration control of a cantilever piezoelectric composite plate. The composite plate is laminated with sandwiched piezoelectric layers, and the entire set is square with dimensions of 200×200 mm and a thickness of 1 mm. The structure is numerically modeled via the finite element method (FEM) by discretizing the large deflection model of the laminated composite plate system. The numerical model tested a laminated plate where its main layer is made of a graphite/epoxy composite (T300/976) with angle orientations of [-45/45/-45/45] and a thickness of 1 mm, in addition to two piezoelectric layers of type PZT G1195 N, each with a thickness of 0.1 mm. The numerical model is then used along with the proposed controller to suppress the vibration of the plate. The active control method utilizes a Proportional-Integral-Derivative (PID) controller with a gain scheduling strategy, where the system nonlinearity and uncertainties are handled by changing the controller parameters according to the system characteristic. The controller parameters are predetermined using the Ziegler-Nichols closed-loop tuning method. The controller receives a signal from the upper piezoelectric layer, which acts as a sensor and feeds the lower piezoelectric layer with a voltage signal to act as an actuator. The performance of the controller was tested by controlling the vibration in four different cases of plate vibration. In all cases, the controller showed good response ability to suppress the vibration in a relatively short time. The proposed control strategy can be developed further to control the vibration in nonlinear shells and other highly nonlinear applications.

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