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DOB-based piezoelectric vibration control for stiffened plate considering accelerometer measurement noise

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Abstract. This paper presents a composite control strategy for the active suppression of vibration due to the unknown disturbances, such as external excitation, harmonic effects and control spillover, as well as high-frequency accelerometer measurement noise in the all-clamped stiffened plate. The proposed composite control action based on the modal approach, consists of two contributions including feedback part and feedforward part. The feedback part is the well-known PID controller, which is widely used to increase the structure damping and improve its dynamic performance close to the resonance frequencies. In order to get better performance for vibration suppression, the weight matrixes is optimized by chaos sequence. Then an improved disturbance observer (IDOB) as the feedforward compensation part is developed to enhance the vibration suppression performance of PID under various disturbances and uncertainties. The proposed IDOB can simultaneously estimate the various disturbances dynamically as well as measurement noise acting on the system and suppress them by feedforward compensation design. A rigorous analysis is also given to show why the IDOB can effectively suppress the unknown disturbances and measurement noise. In order to verify the proposed composite control algorithm (IDOB-PID), the dSPACE real-time simulation platform is used and an experimental platform for the all-clamped stiffened plate active vibration control system is set up. The experimental results demonstrate the effectiveness, practicality and strong anti-disturbances ability of the proposed control strategy.

Keywords: active vibration control; all-clamped stiffened plate; disturbance observer; piezoelectric; high frequency measurement noise; composite control

1. Introduction

Stiffened plate structure, consisting of deep frame, long truss and rivets, is one of the most typical aircraft or automobiles skin (Balamurugan asnd Narayanan 2010). Substantial value vibration of the skin structure is easily occurred, because the aircraft and automobiles are suffered from various external disturbances in the process of moving. This kind of long-term vibration of stiffened plate may lead to fatigue crack and structure damage, even hidden trouble of heavy accidents. Thus, it’s very important to suppress the vibration of stiffened plate structure. Piezoelectric materials structure is one of the most attractive smart structures for structural

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vibration suppression because of the good mechanical-electrical coupling characteristics, frequency response and reliability (Qiu and Ji 2010).

The previous researches of stiffened plate vibration control based on the piezoelectric element can be divided into three categories: passive, semi-active and active control (Beck *et al*. 2011, Boudaoud *et al*. 2009, Qiu *et al*. 2009). These methods are involved in the application of piezoelectric smart structures that utilize distributed piezoelectric actuators and sensors to suppress unwanted vibration in stiffened plate structures. In order to meet the requirement of high vibration performance and due to the complexity of piezoelectric stiffened panel, the researches on active vibration control for this kind of structure are more and more important than before. It is well known that linear control schemes, e.g., the optimal linear quadratic regulator (LQR) strategy (Ang *et al*. 2002), velocity negative feedback controller (Malgaca and Karagulle 2009), and PID scheme (Ma and Ghasemi-Nejhad 2005), which should know the precise model a priori, are already widely applied in the vibration control of stiffened plate due to their relative simple implementation. However, the error inevitably exists between the actual and theoretical stiffened structure models with unavoidable and unmeasured disturbances, as well as external disturbance excitations. So it is very difficult for these linear vibration controllers to obtain sufficient high performance of these stiffened structures. Therefore, we should rationally design the active control algorithm for vibration suppression of these structures to guarantee the robustness of the whole closed system. Recently, with the rapid growth of microprocessors, especially microcontroller and modern control theories, many researches have aimed to develop several kinds of active control methods for the smart structures, not merely stiffened panels. And various active vibration suppression algorithms also have been proposed, e.g., intelligent control (Lin 2005), ADRC scheme (Li *et al*. 2012), sliding-mode control (Hu 2012), LQG method (Montazeri *et al*. 2011), adaptive control (Radecki *et al*. 2010), LMS feed-forward method (Ji *et al*. 2009), decentralized control (Jiang and Li 2010) and robust control (Grossard *et al*. 2011). These algorithms have improved the vibration suppression performance of piezoelectric structures from the different aspects.

However, in real stiffened plate used for aircraft or automobile skin structure, is always confronted with different unknown disturbances. These disturbances may come from internal, e.g., unmodeled error, harmonics and uncontrolled model effects, or external, e.g., external unknown excitation, or high-frequency measurement noise. Conventional feedback-based active methods usually cannot handle directly and fast to reject these disturbances and sensor noise, simultaneously, although some of these control methods can finally suppress them through regulation in a relatively slow way. The sliding-mode controller is the only exception, which shows a good robustness to disturbances. However, it faces an unavoidable application problem called chattering phenomenon (Hu 2012), which leads to degradation of whole system performance when meeting severe disturbances even sensor noise.

To deal with the existing internal and external disturbances, one efficient way is to introduce a feed-forward compensation part into the controller besides conventional feedback part. However, in real stiffened panel applications, it is usually impossible to measure the disturbances directly, so disturbance estimation techniques have to be developed. The disturbances observer (DOB)-based composite control method was originally presented by Ohnishi in 1987 for servo system (Ohnishi *et al*. 1987). Following this direction, many DOB-based control methods have been reported in different applications because of their simplicity and powerful ability to compensate various disturbances, e.g., mechatronics system (Yang *et al*. 2013), hard-disk drive system (Ren *et al*. 2009), robotic systems (Chen 2003), chemical process industry (Yang *et al*. 2011), vibration control of plates (Li *et al*. 2011, Li *et al*. 2012). Among these results, different kinds of DOBs have been developed, including linear DOBs (Yang *et al*. 2011, Li *et al*. 2011, Li *et al*. 2012), neural network DOBs (Ren 2009), nonlinear DOBs (Yang *et al*. 2013, Ren *et al*. 2009, Chen 2003), etc.

The DOB-based composite controller can successfully suppress the vibration of stiffened panel structure, mainly attributing to its remarkable features: (1) the design of DOB usually does not rely on precise disturbance models; (2) DOB has the advantage of handling the unmeasured disturbances including the external and internal ones. However, in stiffened plate vibration systems, the measurement of acceleration is easier than displacement or velocity and to measure and control the amplitude of vibration, the accelerometer is often used as sensor in structure vibration system. But the accelerometer unavoidably imports high-frequency measurement noise. Meanwhile, the problem of phase hysteresis or time delay inevitably exists between the acceleration sensor and piezoelectric actuators when an accelerometer is non-collocated to measure the modes of the stiffened panel. These high-frequency measurement noises, time delay from non-collocated pair of sensor and actuator and unknown disturbances can not only degrade the performance of the whole vibration control system, but also induce instability in the closed-loop system.

In this paper, a composite control approach combining a feed-forward compensation part based on an improved DOB and a feedback regulation part based on PID (IDOB-PID) is developed to improve the vibration suppression performance of the all-clamped piezoelectric stiffened plate with non-collocated accelerometer and piezoelectric actuator. Considering the conventional DOB can only deal with the systems with time delay, disturbances or measurement noise respectively, an improved DOB (IDOB) is designed for the structural vibration control to deal with the time delay, disturbances and high-frequency measurement noise, simultaneously. Moreover, to improve the transient performance of the PID control, the chaotic optimization technique is employed to satisfy the anticipated response of the closed-loop system.

The organization of the paper is as follows. In Section 2, the electromechanical model of the all-clamped piezoelectric stiffened plate with acceleration sensor is derived. The design of an IDOB-PID composite controller with output pre-estimator is presented in detail in Section 3. A rigorous analysis of the internal disturbances, external disturbance excitation, and high-frequency accelerator measurement noise observed and compensated ability for the whole stiffened plate is deduced. In Section 4, an experimental apparatus of piezoelectric smart stiffened plate for active vibration control is built up and experimental results of the all-clamped piezoelectric plate are presented. Finally, conclusions are given in Section 5.

2. Mathematical model of piezoelectric structure

The electromechanical model of the all-clamped stiffened plate structure equipped with two piezoelectric patches and an acceleration sensor is proposed. The surface mounted PZT patches pasted in the middle and top of the structure are used to excite and control the system. In order to avoid the phenomenon of local strain, the accelerometer is often non-collocated to measure the vibration modes of the structure. The actuators and sensor are polarized in the thickness directions which are perpendicular to the plate, as shown in Fig. 1.

It is well known that almost all engineering structures, including beams, plate and shell, are a continuum, but the electromechanical model based on damping-spring-mass system gives a good description of the behavior of vibration embedded piezoelectric elements near the resonant frequencies, as shown in Fig. 2. When using classical assumptions on analytical structural modeling, and finite element modeling, the dynamical behavior equations of a piezoelectric structure can be written as

$$m\ddot{\delta} + c\dot{\delta} + k^e\delta = \sum_i f_i \tag{1}$$

where δ is the displacement vector, m, c and k^e are the mass, damping and stiffness matrices, respectively. And $\sum_i f_i = f_p + f_e$ represents the sum of other forces applied to the equivalent rigid mass, comprising forces applied by piezoelectric elements. Here, f_p is the electrically dependent part of the force applied by piezoelectric elements on the structure, f_e represents the external force applied to the structure. The piezoelectric elements bonded on the host structure considered ensure the electromechanical coupling described by Eqs. (2) and (3).

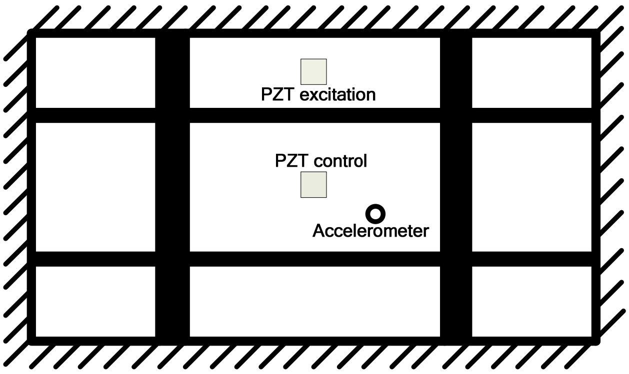


Fig. 1 Schematic diagram of piezoelectric stiffened plate with non-collocated accelerator

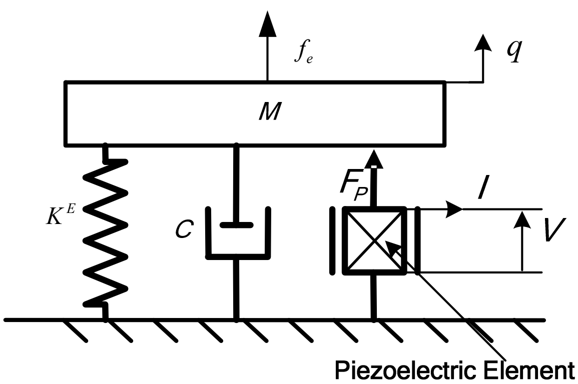


Fig. 2 Diagram of the electromechanical model

$$f_p = -\alpha V \tag{2}$$

$$I = \alpha \dot{z} - C_o \dot{V} \tag{3}$$

where α is the electromechanical coupling coefficient and V is the piezoelectric element voltage vector. I is the outgoing current from piezoelectric elements, C_o is the blocked capacitance of the piezoelectric elements. After modal analysis, the natural frequency and modal function can be obtained.

The physical model Eq. (1) can be described by modal equation after the following change of variables

$$q = \phi^{-1}\delta \tag{4}$$

where ϕ and q are the modal function and modal coordinator of the structure mode, respectively. Ignoring the effect of the force for excitation (f_e), the model function of the piezoelectric structure is shown as follows

$$M\ddot{q} + C\dot{q} + K^E q = -\alpha V \tag{5}$$

$$I = \theta \dot{q} - C_0 \dot{V} \tag{6}$$

where M, C and K^E are the mass, damping and stiffness in modal coordinator, respectively. $\theta = \phi^T \alpha$ is the modal electromechanical coupling coefficient.

Without considering the other effect, the only one degree of the freedom modal space equation of motion in linear state space is shown in Eq. (7) obtained from Eq. (5) and current Eq. (6),

$$\begin{cases} \dot{x} = Ax + Bu \\ y_p = C_p x \end{cases} \qquad x = \begin{bmatrix} q \\ \dot{q} \end{bmatrix} \tag{7}$$

And the measured acceleration signal by the fixed accelerometer is,

$$y_a = C_a \dot{x} \tag{8}$$

Here, $u = V_a$ is the control signal, $y_p = V_s$ is the output of piezoelectric patches. And, system matrices,

$$A = \begin{bmatrix} 0 & 1 \\ -M^{-1}K^E & -M^{-1}C \end{bmatrix}, B = \begin{bmatrix} 0 \\ M^{-1}\alpha \end{bmatrix}, C_p = \begin{bmatrix} 1 & 0 \end{bmatrix}, C_a = \begin{bmatrix} 0 & 1 \end{bmatrix}.$$

These model parameters can be deduced from following equation

$$\alpha = \lambda C_0,\ K^E = \alpha\lambda \frac{f_0^2}{f_1^2 - f_0^2},\quad M = \frac{K^E}{4\pi^2 f_0^2},\ C = 4\pi\varepsilon M f_1 \tag{9}$$

where ε is the inherent mechanical damping coefficient, λ is proportionality coefficient of voltage to displacement in open circuit ratio. And, f_1 and f_0 are the open and short circuit of the first resonant frequency, whose measurement accuracy have the direct relationship with the epoxy resin layer, respectively. The measurement definitions and the values of Eq. (9) are detailedly shown in (Li 2011).

The panel structure as the research object is composed of aluminum alloy LY12CZ used for Aircraft-ARJ21. And the stiffened plate is one of the typical aircraft structure. We know that the stiffened plate (aluminum alloy LY12CZ) has the features such as lightweight, simple structure and high rigidity and stiffness. And then the vibration modes of the aircraft are far from the truss structure of the body structure. And considering that the dimension of this stiffened plate is rather small, so the vibration mode of middle bay is most severe as well as having the greatest impact on the whole structure. The aim of the structural vibration control is to attenuate the vibration amplitute of this location, further to suppress the vibration of the whole structure. This is the reason why we choose the vibraiton mode of the middle bay and only attenuate the first mode vibration of this stiffened structure.

Moreover, according to the results of the modal testing in (Yuan *et al*. 2012), we can see that the first vibration mode behaves like a monopole radiator, while local mode is notable at high frequencies. Besides, these mode shapes also indicate that central part of the stiffened plate is relatively easier to be excited than other parts. Thus, it becomes the main radiation area where active control is needed.

3. Control strategy

A high vibration suppressing performance of single-mode of the piezoelectric panel must provide good dynamic output stabilization as well as be insensitive to the external and internal disturbances, such as other uncontrolled mode, spillover, unmodeled dynamics, and harmonic effects. Note that a DOB can observe the unknown disturbances of the system, and the closed-loop system with PID usually has a progressive stability. Therefore, to achieve a better vibration suppressing property, a composite controller using DOB-based PID control scheme is designed for the vibration control of the all-clamped piezoelectric stiffened plate. However, the control law developed in this paper is based on the signal measured by an acceleration sensor. As shown in Fig. 1, the system acceleration output and control voltage input are employed by the accelerometer and piezoelectric patch. Hence, there are two very important issues should be considered. First, the accelerometer measurement noise often appears in vibration control system and is also the cause of performance degradation factor for control systems. Second, the system has non-minimum phase zeros due to the non-collocated pair of the acceleration sensor and piezoelectric actuator. And non-minimum phase zeros may cause the phenomenon of phase lag and time delay. The sensor measurement noise and time delay will degrade the vibration suppression performance of the high-lever vibration control systems and will also induce instability. Therefore, it poses challenging problems for controller design due to the non-collocated pair of the acceleration sensor and PZT actuator. In this paper, an improved DOB-based vibration control (IDOBVC) structure is proposed to reject the external excitation, control spillover, harmonics effects and high frequency noise of the sensor signal, simultaneously. In addition, an output predication method is used to compensate for the phase lag and time delay of the feedback vibration control, caused from non-collocated pair of the sensor and the actuator. Also, the problem of control spillover can be significantly reduced by the proposed vibration control scheme. The design procedures of the composite controller are presented as follows.

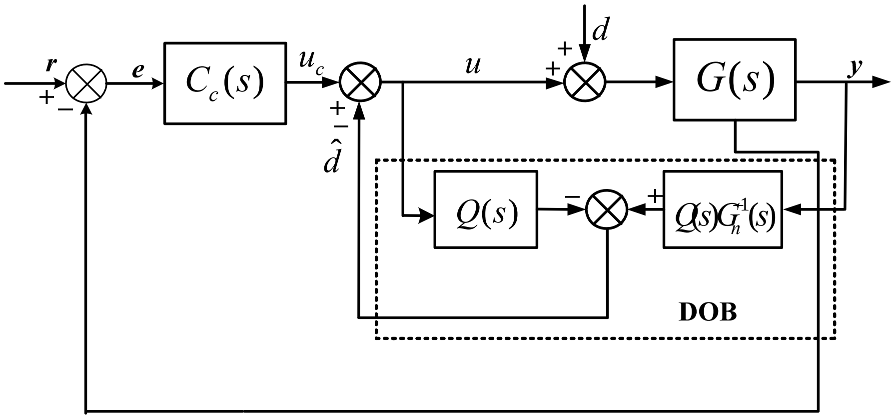


Fig. 3 Bock diagram of the conventional DOB

3.1 IDOBVC design for disturbance and accelerometer measurement noise rejection

A. Brief overview of conventional DOBVC

The block diagram of the standard DOBVC for plate structural is shown in Fig. 3, where $G(s)$ is the second-order real structure mode to be controlled and $G_n(s)$ is the nominal model of $G(s)$; u_c, u and y are the feedback input signal, composite input voltage and output from piezoelectric patches, respectively; d is the equivalent disturbance formed by unmodeled dynamics, unknown dynamics and harmonic effects; $\hat{d}$ is the estimation of d; $Q(s)$-filter is a low-pass filter to restrict the effective bandwidth of the DOB.

It can be seen that the procedure of the DOBVC closes a loop around the controlled model to reject unknown disturbances and lets this loop to approximate the nominal model. Tuning of the loop is accomplished through the adjustment of the parameters and structure of $Q(s)$-filter. Generally, $Q(s)$ is designed to be $Q(s)=1/(\tau s+1)^2$,where the time constant τ is the most meaningful parameter which determines the ability. The process and theory analysis of the DOBVC for the structural vibration rejection has been discussed in (Li *et al*. 2011) in detail.

The time delay of vibration control, caused by non-collocated pair of sensor and actuator, is another factor of degrading the performance of control system. For system with time delay $G(s)=G_n(s)e^{-\theta s}$, considering the inverse model of time delay part is physically unrealizable, an amended DOB structure is proposed in (Yang *et al*. 2011, Zhou *et al*. 2012) through designing $Q(s)=e^{-\theta s}/(\tau s+1)$, as shown in Fig. 4. Here, the time delay part $e^{-\theta s}$ is inserted into the channel of the plant input, and the rational part is moved to the feedback channel of the vibration model output to substitute structure model. Rigorous analysis of disturbance rejection properties of the closed-loop system is given in the presence of both model mismatches, external disturbances and time delay in the (Yang *et al*. 2011). And the simulation results of (Yang *et al*. 2011, Zhou *et al*. 2012) show that the disturbances and time delay effects can be rejected separately by tuning different controller parameters.

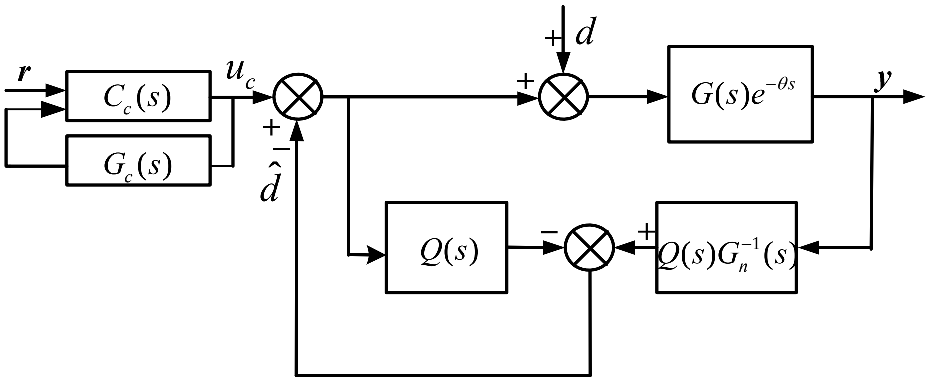


Fig. 4 Block diagram of an amended DOB-based method for time delay system

Although this DOB-based technique can really deal with low-frequency disturbance estimation problem for the first-order process plant, it is no longer suit for the all-clamped stiffened plate vibration control because the vibration mode of the plate is a second-order system with high-frequency measurement noise.

In order to purify the accelerometer signal, a low-pass filter is employed to filter out the high-frequency measurement noise, proposed in (Qiu *et al*. 2009). This scheme was actually improved the performance of the vibration for the flexible cantilever beam. However, the unwanted disturbances, such as, other uncontrolled mode, phenomenon of phase lag, may be significantly increased by introducing the second low-pass filter in the whole closed-loop system. In order to reject the vibration and obtain good positioning accuracy, the DOB-based control scheme was proposed to reject the narrow-band disturbances at two frequencies higher than servo bandwidth for a micro-actuator in (Teoh *et al*. 2008). Not only high-frequency measurement noise, but also all kinds of disturbances are the important factors of affecting real structural vibration control system. Therefore, to design a vibration control method for high vibration rejection performance, high-frequency measurement noise and all kinds of disturbances, such as unmodeled error, harmonic effects, external unknown excitation, etc., should be considered. Neither the traditional vibration control nor the DOBVC-based scheme can deal with the rejection problems of disturbances and high-frequency accelerometer noise at the same time.

B. IDOBVC design for disturbance and accelerometer measurement noise rejection

An improved DOB-based vibration control scheme is proposed based on the adopted DOB-based structure in ref.[28]. The block diagram of the feedforward-feedback IDOBVC for all-clamped piezoelectric stiffened panel is shown in Fig. 5. Similar as the conventional DOBVC structure, $G(s)$ and $G_n(s)$ are the transfer functions of the actual stiffened plate and the nominal plant of the structure, respectively. And the time delay constant θ caused by non-collocated pair of sensor and actuator can be obtained by Lissajou figure method, shown in Section 4.2. Compared with the conventional DOBVC system, another compensator $Q_2(s)$ is added to compensate for the feedback signal of system output to reject the high-frequency measurement noise. (Yang *et al*.

2011) deduced that the effect of the time delay to control system can be eliminated by independently designing the feedback channel part and the proposed IDOB. In order to analyze the performances of disturbance and measurement noise rejection, the transfer functions between the system output and the disturbances, and measurement noise are given as follows, respectively

$$G_{yd}(s)=\frac{1-(Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s))G_n(s)}{G(s)C_c(s)+1+(Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s))(G(s)-G_n(s))}$$

$$G_{y\xi}(s)=\frac{G(s)\left[(Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s))+C_c(s)\right]}{G(s)C_c(s)+1+(Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s))(G(s)-G_n(s))}$$

The influence of disturbances and measurement noise on the output can be eliminated when $G_{yd}(s)=0$ and $G_{y\xi}(s)=0$.

Then, it can be obtained that

$$\begin{cases}1-(Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s))G_n(s)=0\\ G_n(s)\left[(Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s))+C_c(s)\right]=0\end{cases}$$

That is to say,

$$\begin{cases}Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s)=G_n^{-1}(s)\\ Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s)=-C_c(s)\end{cases} \tag{10}$$

It is necessary to simultaneously eliminate the disturbance and measurement noise at different frequency ranges, because the disturbance and measurement noise belong to the low and the high frequency ranges, respectively. So Eq. (10) can be written as

$$\begin{cases}Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s)=G_n^{-1}(s), & \omega\in(0,\omega_d)\\ Q_1(s)G_n^{-1}(s)-C_c(s)Q_2(s)=-C_c(s), & \omega\in(\omega_\xi,\infty)\end{cases} \tag{11}$$

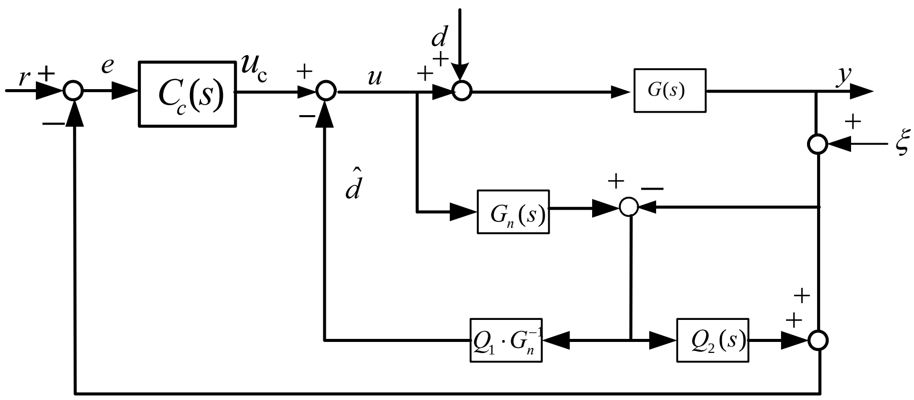


Fig. 5 Improved DOB-based vibration control (IDOBVC) system

where $\omega_d < \omega_\xi$. And we should rationally choose the solutions of $Q_1(s)$ and $Q_2(s)$ to satisfy Eq. (11). It is well known that the stiffened plate is a stable system, so the chosen nominal plant $G_n(s)$ is also stable minimum-phase plant. That is to say the inverse model of $G_n(s)$ is stable. Thus, there are $Q_1(s)G_n^{-1}(s) = G_n^{-1}(s)$ and $Q_2(s) = 0$ in the low frequency range $\omega \in (0, \omega_d)$, while there are $Q_1(s) = 0$ and $Q_2(s) = 1$ in the high frequency range $\omega \in (\omega_d, \infty)$. Consequently, the IDOBVC system not only guarantees the perfect anti-disturbance capability in the low frequency, but also eliminates the influence of the high-frequency measurement noise.

3.2 IDOB-PID composite vibration controller for all-clamped stiffened plate

The method of this work focuses on obtaining well performance of vibration suppression. A composite control scheme combing an improved disturbance observer (IDOB) with the PID, called IDOB-PID, is proposed to treat with the vibration of the all-clamped stiffened plate. Note that the stiffened plate considered here is a second-order system because of the first natural frequency of the piezoelectric structure being studied. Through the Separation Principle, we can design the feed-forward and feedback channel part, respectively. The feed-forward channel consists of IDOB approach shown in Section 3.1. The feedback part is PID controller. Fine effects can be achieved by tuning the parameters of PID and the parameters of the two filters $Q_1(s)$ and $Q_2(s)$ in the proposed IDOB. Here $Q_1(s)$ and $Q_2(s)$ are selected as a second-order-low-pass filter and a second-order-high-pass filter with the stead-state gain of 1 in their relative frequency range, respectively. The structures of the filters are expressed as

$$Q_1(s) = \frac{1}{(\tau_1 s + 1)^2} \tag{12}$$

$$Q_2(s) = \frac{k_o s^2}{s^2 + \varphi\omega_0 s + \omega_o^2} \tag{13}$$

It should be pointed out that the implementation of the proposed IDOB is rather simple, thus the introduction of feed-forward compensation part does not increase much computational complexity. In this paper, the first natural mode frequency of the all-clamped stiffened plate is $\omega_1 = 164.8$ Hz. Therefore, it is reasonable to set the corner frequencies of the second-order low-pass filter and high-pass filter at 180 Hz and 1100 Hz for the experimental research. In additional, there is a phase shift between the original and the filtered acceleration signals. So the time delay from the filters should also be compensated on the experimental research.

The feedback part is dependent on the choice of the parameters of PID. Consider the structure (14), the output of the PID controller in time domain is given by

$$u = K_p e + K_d \frac{de}{dt} + K_i \int_0^t e(\gamma) d\gamma \tag{14}$$

The parameters K_p, K_d, K_i of PID for suppression of vibration depend on the structure behavior. The controller parameters should be appropriately selected to yield the minimum control cost while the pre-specified performance and stability criteria are satisfied. Thus, the transient performance function of the first mode is employed

$$J = \min(w_1 \sum_{k=0} \left| e(k) \right| k / \left| \max(e(k)) \right| + w_2 \sigma) \tag{15}$$

where σ represents overshoot, e is the error between the target output and the actual output of system, w_1, w_2 are the weight coefficients, respectively, which can be chosen flexibly. This function not only reflects the dynamic characteristic but also the steady-state behaviors. Moreover, $e(k)$, $\max(e(k))$ and σ are very easy to obtain.

In order to provide an appropriate control input strength, a chaotic method to auto-regulate the parameters K_p, K_d, K_i is introduced similar as (Li *et al*. 2012). The chaos sequence used here is the well-known logistic equation in this paper.

$$x_{n+1} = \mu x_n (1 - x_n), \quad \chi_n = (\chi_{1,n}, \chi_{2,n}, \chi_{3,n}), \qquad n = 0,1,\cdots,N, \tag{16}$$

where $\chi_{1,n}, \chi_{2,n}, \chi_{3,n}$ represent the gain of K_p, K_d, K_i, respectively. We should choose $\mu = 4$ to satisfy Eq. (16) in chaos state. Moreover, any fixed points should not be selected as initial values. The process of the chaotic optimization used for K_p, K_d, K_i has been detailed in ((Li *et al*. 2012).

4. Experimental verifications

4.1 Introduction of the experimental set-up

To value the performance of the proposed non-collocated acceleration sensor and piezoelectric actuator pair based vibration control algorithm, experimental researches are conducted on an all-clamped stiffened plate (aluminum alloy LY12CZ). Four stiffeners divide this plate into nine rectangular bays, which are evenly divided by these stiffeners. To implement the proposed IDOB-PID composite control, an experimental setup is erected, as shown in Fig. 6. In this research, the clamped stiffened plate between the aluminum blocks that are bonded together tightly with several bolts, and two piezoelectric actuators are bonded to upper surface of the plate for excitation and controlling vibration mode. The plate, representative of real aircraft skin, is 860mm long, 550mm wide and 1mm thick, and the piezoelectric patches are square of size 30mm×30mm. The aluminum alloy plate is broken into nine surfaces by four stringers. The acceleration sensor (Type: PCB-333B32) is fixed on the plate to measure the vibration. The whole vibration control algorithms including the IDOB-PID technique are simulated on Matlab (2006a) and implemented by the dSPACE1103 board with a sample frequency of 10 kHz.

The piezoelectric acceleration sensor's signal and the PZT signal are amplified by signal conditioning instrument (Type: B&K-2693-A-OS4) to the voltage range of -5v to +5v, and converted into digital data through an A/D (analog to digital) channel in dSPACE1103. The output of controller and excitation are sent to the amplifiers for the PZT patches through a D/A (digital to analog) channel in dSPACE1103. The piezoelectric actuators are driven by power amplifiers (HEA-200D), which changes the low voltage signal in the range -5v to +5v into the high voltage signal in the range -200v to +200v. The first natural frequency is 164.8 Hz, which is estimated experimentally from the frequency response of the plate by sweeping the excitation frequency from 30 to 500 Hz.

Parameters of the model presented in the previous section can be identified from the measurements on the experimental set-up. The measurement definitions and the values of Eq. (8) in Section 2 are detailed in (Li *et al*. 2011) and summarized in Table 1. The measurement definitions and the values of Eq. (9) are detailedly shown in Table 1. And in order to evaluate the parameters of Eq. (9) and Table 1, a laser displacement meter (LK-080) is used to measure the resulting response, and the signals from the laser displacement are sampled through an A/D converter and then transmitted to a DSPACE DS1103 board.

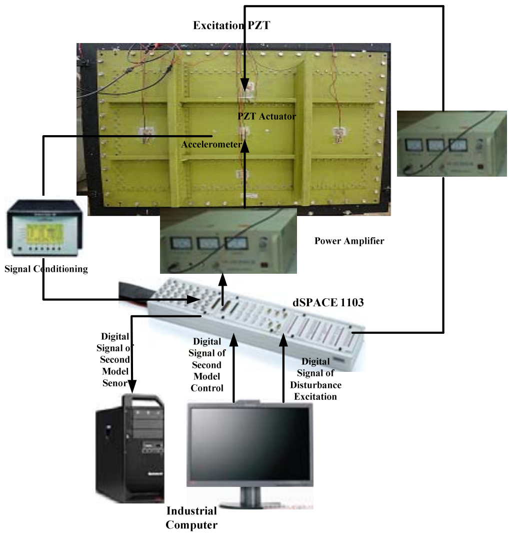


Fig. 6 Photograph of the experimental set-up

Table 1 Values of measurement and model parameters

Short circuit of 1th resonance frequency f_0	164.84 Hz
Open circuit of 1th resonance frequency f_1	164.86 Hz
Open circuit of 1th inherent mechanical damping coefficient ε	0.0074
Proportionality coefficient of voltage to displacement in open circuit ratio λ	16432 V/m
Blocked capacitance of piezoelectric C_0	78.56 nF
Force factor α	0.0012
Stiffness of equivalent piezoelectric elements in short circuit K^E	21215 N/m
The equivalent rigid mass M	20.8 g
Inherent structural damping coefficient C	0.2628 N/(m·s)

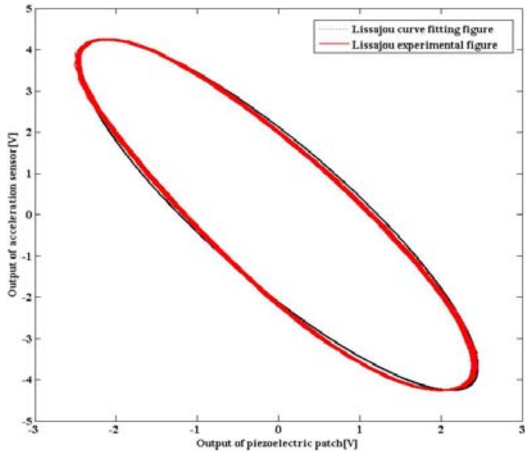


Fig. 7 Lissajou figure between the signals of acceleration sensor and PZT patch: Measured signals Vs Curve fitting

4.2 Time delay parameter identification

Since the accelerometer and the PZT actuator are spatially non-collocated, the phase lag between the sensor and actuator causes instability problems. Thus, according the IDOB-PID controller in Section 3, time delay parameter should be estimated before phase compensation. Phase error identification method is presented that uses persistent excitation. In this paper, based on the (Qiu *et al*. 2009, Li *et al*. 2012), phase hysteresis estimation for the first mode is carried out by persistent excitation at the resonant frequency at 164.8 Hz. The measured signals by PZT patch and the acceleration sensor respectively are used as the X- and the Y-coordinate, then one can plot the Lissajou figure with decayed damping behavior response, as shown in Fig. 7. From Fig. 7, it is easy to known that the non-collocated placement will lead to a decayed phase hysteresis loop for the first mode. And the decayed plot-fitting Lissajou figure can be smoothened by the following Eq. (17).

$$\begin{cases} \mathrm{x} = 2.8\sin(2\pi\omega t) \\ y = 4.2\sin(2\pi\omega t + \varphi) \end{cases} \tag{17}$$

where the first natural frequency is $\omega = 164.8$ Hz, $\varphi = 2.83\,rad$. By employed Eq. (17) as the X- and Y-coordinate, another decayed curve-fitting Lissajou figure can be obtained as shown in Fig. 7. Comparing the solid and dotted lines shown in Fig. 7, it can be seen that they are matched very well. Thus, the phase lag angle of the first mode can be obtained from Eq. (17), and it is $\varphi = 2.83$ *rad*. According to the above-identified results, the phase lag of the non-collocated placement of the accelerometer and PZT actuator for the first mode vibration controlling is $\varphi = 2.83\,rad$. So the time delay of the controlling mode is,

$$\Delta t = \frac{\varphi}{2\pi}\frac{1000}{\omega} = 2.73ms$$

According to the proposed IDOB-PID composite method, when the tuning time is selected as $\Delta t = 2.73\,ms$ in the feedback channel, the phase lag due to the non-collocated placement of the acceleration sensor and the PZT actuator can be compensated for the first natural mode vibration suppression.

4.3 Experimental results

For easy comparison, the spectrum of the normalized sensor signal is expressed in decibel as follows,

$$\text{The decibel value} = 20\log_{10}(\Gamma(y/y_R)) \tag{18}$$

Here Γ is the symbol of Fourier Transformation and y_R is the reference output of acceleration sensor for normalization. The value of y_R is set to 1v in the following section so that 0dB is equivalent to 1v and -20dB is equivalent 0.1v. To demonstrate the efficiency of the proposed IDOB-PID control method, experiments on the all-clomped stiffened plate have been performed. Three control methods, i.e., PID control, conventional DOB-PID method and the proposed IDOB-PID strategy, are applied to the vibration plate. And the system parameters can be obtained by the identification method in the above section. To have fair comparisons, firstly, the excitation conditions have the same voltage values; secondly, piezoelectric actuator and acceleration sensor are always on the same position. The parameters K_p, K_d, K_i are auto-adjusted by the chaotic optimization method. And we should choose the very small constants as the iteration initial values, where the differences between these numbers should be as small as possible, i.e., 0.00001, 0.00010 and 0.00011. Then, mapping these initial values to $(0, \max(K_p, K_i, K_d))$ and Eq. (15) is chosen as the novel performance function, and the excellent parameters $K_p^* = 483.7$, $K_i^* = 1.954\times 10^5$, $K_d^* = 46.69$ can be obtained. Finally, the second-order filters are chosen in the design of the improved disturbance observer, and the time constant $\tau = 0.001$ is 10 times of the sample time for low-pass filter, meanwhile $\omega_o = 1100$ Hz is chosen for the high-pass filter.

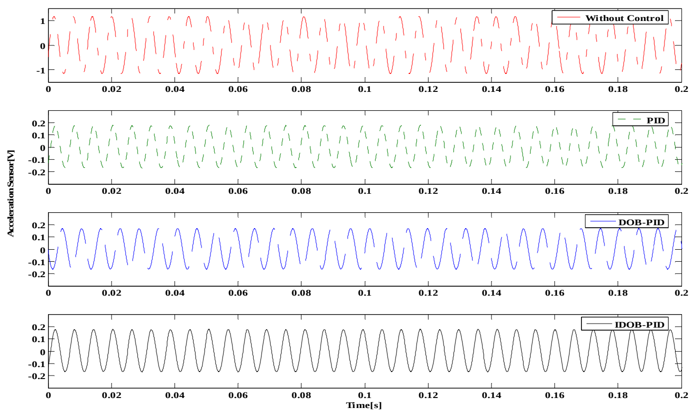
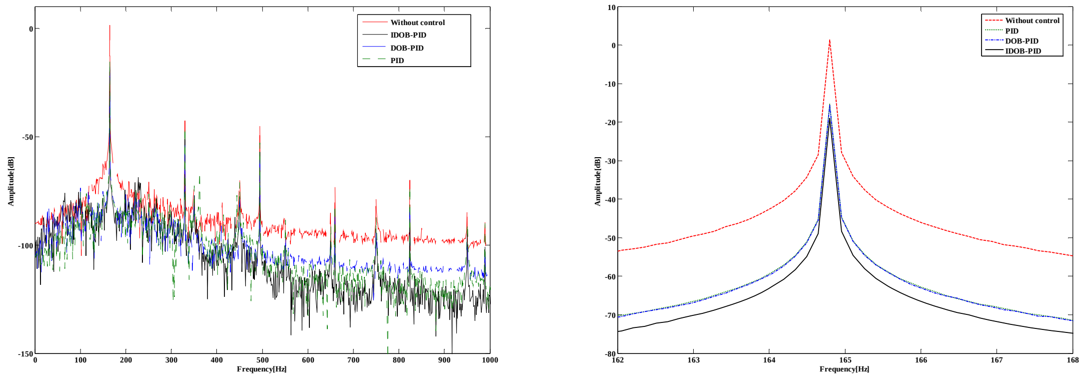


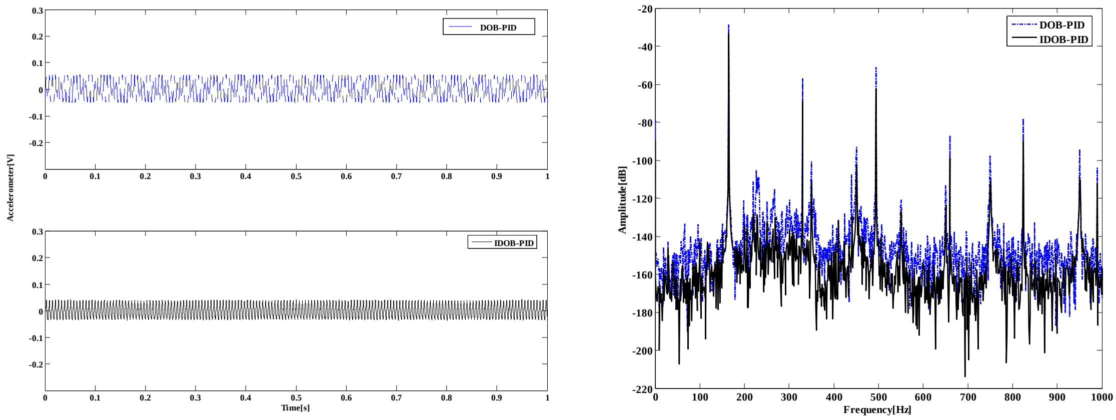
Fig. 8 Performance comparisons of the time-domain responses under the PID, conventional DOB-PID and the proposed IDOB-PID algorithms



(a) (b)

Fig. 9 Experimental results of the controlled and without controlled modes. (a) Frequency-domain responses and (b) Local curves of the frequency responses

For the sake of comparisons, PID, conventional DOB-PID and proposed IDOB-PID algorithms all achieve relatively good performances of vibration suppressing in Figs. 8 and 9. And the peak values of vibration below 1000 Hz are shown in Table 2. In order to further investigate the improved damping of the resonant modes obtained by using the feedback controller without feed-forward compensation, conventional DOB-based composite controller and IDOB-based method proposed in this paper, the same frequency responses are shown in Figs. 9 and 10(b) and some time-domain responses at accelerometer output are obtained for the experimental system, as shown in Figs. 8 and 10. These figures show the output of the acceleration sensor described in volts. In order to analyze, the figures also show the corresponding time-domain and frequency-domain responses for the uncontrolled plate. The corresponding frequency and time-domain responses for the case in which the plate is excited by piezoelectric actuator at the first resonant frequency are shown in Fig. 8. The results shown in Fig. 9(b) illustrate that PID, conventional DOB-PID and IDOB-PID schemes with the same parameters all can reach to -17dB of the single resonant mode, i.e., all the three control approaches can reduce the vibration to 17dB, 17.36dB and 20.37dB, respectively. But from Fig. 9, the closed-loop of PID vibration control system becomes unstable, because of the control spillover for the uncontrolled mode. If we want a better performance, the stability of whole system will collapse. But the conventional DOB-PID and IDOB-PID methods possess the robust stability and good performance. Fig. 9(a) clearly shows that a peak value is introduced at the second resonant frequency 362.6Hz called control spillover, when only the PID controller is used for the plate vibration control. And the spillover phenomenon has been well compensated due to introducing the disturbance observer, as the solid line shown in Fig. 9. When frequency of second resonant frequency of 362.6 Hz and the other lower frequency is excited, the DOB estimates and compensates it, immediately. The results shown in Fig. 9(a) illustrate that many other modes, the harmonic frequencies and measurement noise come out when the stiffened plate is excited at the frequency 164.8 Hz, because of the complexity of the structure. Fig. 9(a) also indicates that the response amplitude values of 329.7 Hz, 494.5 Hz and 659.3 Hz, i.e., the double-frequency, triple-frequency and quadruple-frequency of the natural frequency, are reduced by 3.3dB(7.4dB, 8.2dB), 9.1dB(13.5dB, 14.3dB) and 7.2dB(10.1dB, 10.2dB), respectively, using PID(DOB-PID, IDOB-PID) methods. The frequencies of 444.4 Hz and 649.6 Hz maybe easy to provoked when the all-clamped piezoelectric stiffened plate is excited by the first resonant disturbance. Obviously, their response values have been suppressed very well by the DOB-based controller. And from Fig. 9(a), the closed-loop of vibrating system with only PID controller will become uncontrolled. But the DOB-based vibration control system always possesses the robust stability and good performances.



(a) (b)

Fig. 10 Experimental results of the controlled and uncontrolled modes with better performances. (a) Frequency-domain responses and (b) Time-domain responses

To show the superiority of the proposed IDOB-PID vibration control scheme, tests have also been performed to evaluate the stability and transient performance which always exist in the closed-loop control system including the active vibration suppressing system. The controller parameters are the same as before, except that the gains of the power amplifier is a little bigger than the same output voltages of the two vibration control schemes, i.e., conventional DOB-PID and proposed IDOB-PID methods. And the similar results with that case can be obtained. As can be seen from Fig. 10 and Table 3, the amplitude of the proposed IDOB-PID method is deduced by 32.5dB, also measured by acceleration sensor, which is much better than that under the conventional DOB-PID method (about 27.4dB). Compared with conventional DOB-PID method, it is obviously shown that the proposed IDOB-PID has the better performances of vibration suppression, not only against the low-frequency disturbances, i.e., external excitation, harmonic effect, spillover effect and so on, but also against the high-frequency acceleration sensor noise, from Fig. 10 and Table 3. Table 3 indicates that the frequencies of 227.1 Hz and 439.6 Hz maybe easy to provoke when the plate is suppressed by the DOB-PID method with bigger gain of the power amplifier. Obviously, Table 3 also shows that the response amplitude values of the other frequencies below 1k Hz are all smaller with the proposed IDOB-PID method. The various harmonic interferences caused by the nature frequencies, the effects of some unknown peaks, and acceleration sensor noise can be compensated well via the proposed improved DOB-based controller. It is the reason that the IDOB-PID has the better performance with robust stability.

Table 2 Performance analysis of control algorithms

	Without-control (dB)	Only PID (dB)	Conventional DOB-PID (dB)	Proposed IDOB-PID (dB)
164.8 Hz	-1.347	-15.63	-16.02	-19.03
329.7 Hz	-42.64	-45.95	-50.03	-50.88
362.6 Hz	-89.29	-66.48	-100.1	-101.3
444.4 Hz	-87.69	-77.15	-100.9	-101.5
494.5 Hz	-43.74	-52.8	-57.21	-58.05
649.6 Hz	-85.16	-91.36	-99.43	-97.91
659.3 Hz	-73.37	-81.5	-83.46	-83.54
749.7 Hz	-76.98	-87.27	-92.83	-92.41
824.2 Hz	-70.18	-73	-82.72	-83.56
949.9 Hz	-82.5	-90.22	-94.62	-96.38
989 Hz	-90.16	-90.26	-100.9	-101.6

Table3 Better Performance indexes of DOB-PID and IDOB-PID

	Conventional DOB-PID (dB)	Proposed IDOB-PID (dB)
164.8 Hz	-27.4	-32.5
329.7 Hz	-57.11	-69.35
494.5 Hz	-51.21	-62.37
649.6 Hz	-113.4	-123.6
659.3 Hz	-86.8	-98.69
749.7 Hz	-97.7	-111.1
824.2 Hz	-77.97	-89.94
949.9 Hz	-93.08	-109.3
989 Hz	-104.1	-112.2

5. Conclusions

In piezoelectric distributed structure vibration system, the low-frequency unknown disturbances including external excitation, harmonic effects and control spillover, and high-frequency measurement noise, have undesirable influences on the active vibration control system. To break out the limitation of the traditional active vibration control scheme, a composite PID control strategy based on an improved DOB (IDOB-PID) technique has been introduced for the piezoelectric all-clamped stiffened plate with non-collocated placement of the PZT actuator and the acceleration sensor. Three kinds of control techniques have been introduced to enhance the vibration suppressing property of the active control systems. Firstly, for the auto-choice of appropriate gains in PID vibration control of all-clamped stiffened piezoelectric plate, the chaotic optimization based on the logistic sequence is invested through the transient performance function to achieve the good dynamic performance. Secondly, in order to eliminate the phase hysteresis from the non-collocated placement of the PZT actuator and acceleration sensor, the delay part obtained by Lissajou figure method, is introduced to the original disturbance observer. Thirdly, the IDOB-based method has been proposed to handle the unknown disturbances consisting of external excitation and harmonic effect, spillover effect caused by active vibration controller. Meanwhile the IDOB-PID vibration controller can do well in eliminating the influence of high-frequency measurement noise of acceleration sensor. Compared with PID and classical DOB-PID methods, experimental results have demonstrated the all-clamped stiffened plate under the active control of the proposed IDOB-PID scheme possesses the best performance of vibration suppressing.

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