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Vibration suppression of cart-pendulum system by combining the input-shaping control and the position-input position-output feedback control †

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Abstract

This study is concerned with the active vibration control of a cart-pendulum system. The input-shaping control alone is not sufficient to suppress vibrations of the cart payload, especially when external disturbance is present. In order to solve this problem, a new control technique consisting of the input-shaping and the position-input position-output feedback controls is proposed. The input-shaping control minimizes vibrations during cart motion and the position-input position-output feedback control takes charge of suppressing residual vibrations after the cart reaches the desired position. The stability of the proposed position-input position-output feedback control was investigated theoretically. The testbed was built to validate the proposed method. It was proved both theoretically and experimentally that the proposed control technique can be successfully used to control vibrations of the pendulum.

Keywords: Pendulum vibration; Input-shaping; Position-input position-output feedback; Vibration experiment

1. Introduction

Vibrations occur when a container is moved by a crane at an industrial site. If the object continues to vibrate, not only it is difficult to transport safely, but it can also cause an accident colliding with other structures. In order to minimize the vibration of the container that occurs during and after cart transportation, the container needs to move so slowly but its working efficiency is greatly decreased. Research on how to move the container quickly without vibrations has been pursued for a long time [1, 2].

Various feedback controllers have been developed to suppress the vibration of the container during crane operation. Marttinen et al. [3] proposed a method to suppress the vibration of the payload during crane operation by applying PID controller and pole-placement control. Kim et al. [4] proposed the controller multivariable state feedback controller with an integrator to suppress the sway. Omar and Nayfeh [5] developed a controller based on gain-scheduling feedback to minimize the swing of the crane. Ngo and Hong [6] carried out the study to suppress the vibration while moving the crane quickly by applying a sliding-mode controller. Zavari et al. [7] applied the H-infinity controller to suppress the vibration of the crane. Hua and Shine [8] proposed adaptive nonlinear control and showed better performance than the PID control or sliding-mode control method. The fuzzy controller [9, 10], fuzzy-tuned PID control [11], and neural network control [12] have been proposed to move the load precisely as well as to eliminate its sway.

In case of feedback control method, there is a limitation to the implementation because the system information should be measured and the controller is complicated. In view of the actual implementation of the method for suppressing the vibration of the crane, it is preferable that the controller has a simple structure and has robustness. Thus, an open-loop control method which is more simple than a closed-loop controller has been proposed.

In the case of the open-loop control method, the system information does not have to be measured in real-time. Based on the theoretical model of the crane, a feedforward method using optimal control has been studied to calculate the optimal path for anti-sway [13-16]. However, since the optimal control method requires an accurate theoretical model, there are limitations to apply to complex real systems.

In order to compensate for this, an input-shaping method with high robustness has been proposed, which can calculate the travel path of a crane using less system information. Smith [17, 18] first introduced the concept of zero vibration (ZV) input-shaping. Singer and Seering [19] developed zero vibration derivative (ZVD) and zero vibration derivative derivative (ZVDD) input-shaping to improve the robustness of ZV input-shaping. Experimental studies on actual implementation of the input-shaping control based on ZV or ZVD have been carried

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out by some researchers [20-22]. Hyde and Seering [23] developed an input-shaping method for multi-mode vibration reduction of structures based on existing ZVD and ZVDD techniques. Murphy and Watanabe [24] have implemented the digital shaping filter based on the input-shaping technique and verified the performance of the input-shaping technique. Tzes and Yurkovich [25] developed an adaptive input-shaping technique using a frequency domain identification scheme to suppress the vibration of a flexible structure.

Among the conventional input-shaping methods, ZVDD, zero vibration, and triple derivative (ZVDDD) techniques have improved insensitivity more than ZV or ZVD can, but the operating time is increased. To improve this, an extra-insensitive (EI) input shaper with the same operating time as ZVD and better insensitivity has been developed [26, 27]. In addition, to improve the performance of the input-shaping method, a specified-insensitivity (SI) input shaper [28] and modified input-shaping (MIS) [29] technique have also been developed. The input-shaping method that applies the smooth command [30, 31] has been developed to prevent the higher modes from being excited. The robust input-shaping technique has been developed to improve the performance, efficiency, and insensitivity [32].

The input-shaping technique is basically an open-loop control so that it cannot suppress unexpected vibrations alone. Hence, many control techniques have been developed to improve the open-loop based input shaping control. Huey et al. [33] investigated the closed-loop signal shaping for force and sensor disturbance rejection, hard nonlinearity accommodation, and human-in-the-loop control scenarios. Zuo et al. [34] proposed the combination of the input shaping controller and the linear quadratic regulator (LQR). Kapila et al. [35] proposed the combination of the input shaping controller and the full-state feedback controller with multiple input delays to suppress vibrations of flexible structure.

As mentioned above, various control techniques have been developed to enhance the performance of the input-shaping control by adding additional feedback control. The combined control applies to the system until the cart position reaches its destination. However, there is no control after the cart reaches its final position. Hence, residual vibrations caused by unexpected disturbances remain uncontrolled. We propose a sequential control method in which the cart motion is controlled by the input-shaping control until the cart reaches the desired position and the residual vibrations are then suppressed by the position-input position-output (PIPO) feedback controller. The input-shaping control is in action but the PIPO feedback controller is turned off until the cart reaches the desired position. The PIPO feedback controller is turned on but the input-shaping control is turned off after the car reaches the desired position.

The proposed control method uses the fact that the cart motion is generated by a motor, gearbox, motor driver, and the position-tracking controller. Hence, the input-shaping command for the cart motion is in the form of the desired cart

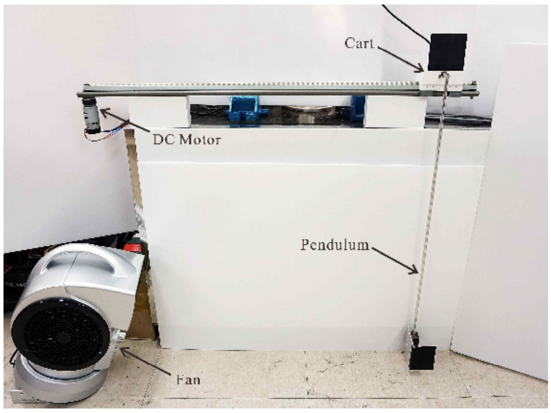

Fig. 1. Experimental setting of the testbed.

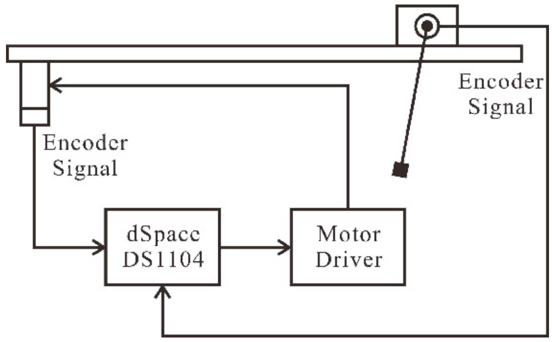

Fig. 2. Schematic diagram for the experiment.

position. Based on the assumption that the position of the pendulum can be measured relative to the cart, the PIPO feedback controller uses the angular position measurement of the pendulum and produces the desired displacement command of the cart. The stability of the PIPO feedback control was investigated theoretically; we found that the stability condition is static, which implies that the stability doesn't depend on frequency. The proposed PIPO feedback control algorithm is simple enough to be easily implemented on a real system.

In order to validate the proposed control method, we constructed a cart-pendulum experimental system. A dynamic model was also derived and validated experimentally. Both theoretical and experimental results show that the proposed control method can effectively suppress vibrations of the container during and after the operation of the cart even though the external disturbance is present.

2. Experimental setup

The experimental system consisting of a cart with a pendulum was built as shown in Fig. 1. To drive the cart, a geared DC motor with the encoder (D & J WITH, IG-32PGM + Encoder 02TYPE, 1/5 gear ratio) was used. The geared DC motor was then attached to the end of the rail and a belt pulley was connected to the motor shaft. The belt was attached to the cart so that rotating the motor would move the cart from left to right or vice versa. The pitch of the belt and the pulley is 3 mm. The so-called W rail (Igus, Drylin, W-10-40-1000) was used as a guide rail. A motor driver (Sabertooth, dual 12A) was used to drive the DC motor. The schematic diagram of the experimental setup is shown in Fig. 2. In order to accurately track the desired position of the cart, the PID control was designed and implemented by using the Simulink block. The position command is compared to the encoder signal of the motor and the error signal is fed into the PID controller, which generates a voltage command to the DC motor driver. Hence, the real position of the cart can accurately follow the desired position. The position of the cart is measured by using the encoder. To measure the angular position of the pendulum, an encoder (Autonics, E30S-1000-3-2MJ) fixed to the cart is used. A tip mass was attached to the end of the rod, and a small plate was attached to the tip in order to amplify the external disturbance caused by the wind.

3. Dynamic modeling

Let us derive a dynamic model for the experimental system. Fig. 3 shows a simple model for the cart-pendulum system. l is the length from the pivot of the cart to the center of the tip mass; m_b is the mass of the uniform rod; u is the position of the cart; θ is the angular position of the pendulum; and M represents the tip mass.

It has been found experimentally that the geared motor doesn't allow the coupled motion with the pendulum because the friction inside the gearbox is very large and the position of the cart can be accurately controlled independently of the pendulum. Hence, it is assumed that the input to the dynamic model is the cart position.

The kinetic energy T and potential energy V for this model can be derived as follows.

$$T = \frac{1}{2}\left(M + \frac{m_b}{3}\right)l^2\dot{\theta}^2 + \left(M + \frac{m_b}{2}\right)l\dot{u}\dot{\theta} + \frac{1}{2}\left(M + m_b\right)\dot{u}^2 \quad (1)$$

$$V = \left(M + \frac{m_b}{2}\right)gl(1 - \cos\theta) \quad (2)$$

where g is the gravitational constant. Using Eqs. (1) and (2) and the Lagrange equation, the equation of motion can be derived as

$$M_t\ddot{\theta} + C_t\dot{\theta} + K_t\theta = -U_t\ddot{u} \quad (3)$$

where

$$M_t = \left(M + \frac{m_b}{3}\right)l^2, \quad K_t = \left(M + \frac{m_b}{2}\right)gl, \quad (4a, b)$$

$$U_t = \left(M + \frac{m_b}{2}\right)l \quad (4c)$$

and the viscous damping coefficient, ζ is added. Dividing Eq. (3) by M_t results in

Table 1. Parameters of the experimental system.

Parameters	Numerical values
Mass of the tip weight	0.4692 kg
Length of the rod	0.815 m
Mass of the uniform rod	0.1082 kg
Damping coefficient of pendulum	0.0055

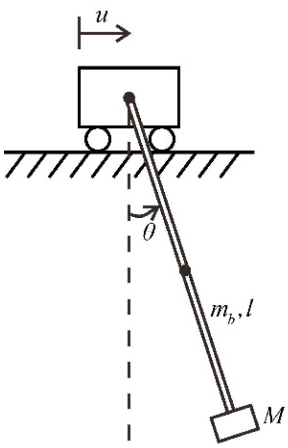

Fig. 3. Cart and pendulum model.

$$\ddot{\theta} + 2\zeta\omega_n\dot{\theta} + \omega_n^2\theta = -g_t\ddot{u} \quad (5)$$

where

$$\zeta = \frac{C_t}{2M_t\omega_n}, \quad \omega_n = \sqrt{\frac{K_t}{M_t}}, \quad g_t = \frac{U_t}{M_t} \quad (6a\text{-}c)$$

in which ζ represents the damping factor and ω_n represents the natural frequency of the pendulum.

The input to the DC motor driver is the voltage input, which is then converted into the PWM current signal that drives the geared DC motor. As stated earlier, it was found that the geared motor doesn't allow the coupled motion with the pendulum. Hence, the coupled equations of motion between the cart and the pendulum are not necessary since the pendulum motion does not affect the cart motion because of gearbox. As shown in Eq. (5), the acceleration of the cart affects the pendulum motion. If the cart starts from rest, then the pendulum is excited inevitably, which causes vibration.

The parameters used for numerical calculations are listed in Table 1. The damping factor was obtained from free vibration experiments and the logarithmic decrement. Numerical calculations and experiments were conducted to verify the validity of the dynamic model derived in this study. The actual displacement profile of the cart is shown in Fig. 4 and was also used for numerical simulation.

The calculated angular displacement of the pendulum was then compared to the measured angular displacement as shown in Fig. 5, which shows that the theoretical prediction

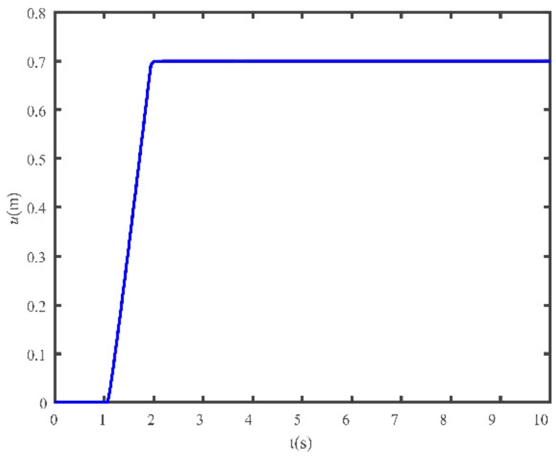

Fig. 4. Displacement of the cart (ramp input).

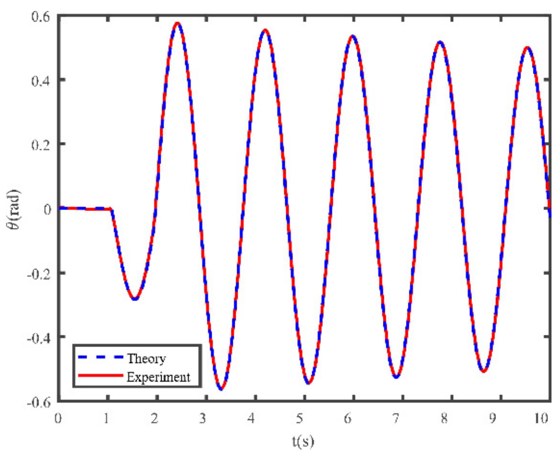

Fig. 5. Angular position of the pendulum for ramp input (theory vs. experiment).

obtained by using the dynamic model given by Eq. (5) is in good agreement with the experimental result, so that the dynamic model is valid.

4. Input-shaping control and pipo feedback control

The input-shaping control is a well-known open-loop control algorithm that can minimize vibration after the cart motion stops.

The concept of the input-shaping control is based on the consecutive impulses that generate the response, resulting in a zero response after a certain period of time. In order to apply the appropriate impulses to an arbitrary input profile, the convolution needs to be carried out, so that the resulting command can generate zero response as shown in Fig. 6. The kinetic energy and potential energy for this model can be derived as follows.

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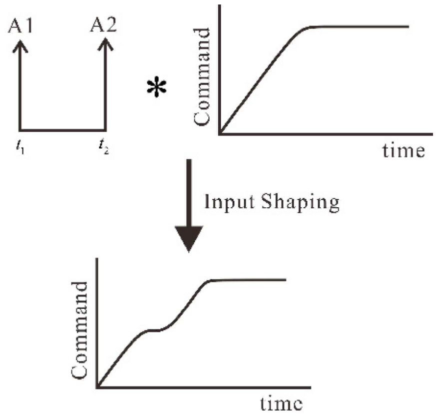

Fig. 6. The process of the ZV input-shaping.

The impulse response of Eq. (5) is obtained as

$$\theta(t) = \frac{g_t \omega_n e^{-\zeta\omega_n t}}{\sqrt{1-\zeta^2}} \sin(\omega_d t + \phi) \tag{7}$$

where

$$\omega_d = \omega_n(1-\zeta^2)\,,\quad \phi = \tan^{-1}\left(\frac{2\zeta\sqrt{1-\zeta^2}}{1-2\zeta^2}\right) \tag{8a,b}$$

The impulse response given by Eq. (7) is different from those obtained by previous researchers, because the acceleration of the cart was taken into consideration. Anyway, it becomes evident that the concept of consecutive impulses still holds for Eq. (5).

In this study, the zero vibration (ZV) input shaper is considered. The formula for the ZV input shaper is

$$u_{is}(t) = \sum_{i=1}^{2} A_i \delta(t_i) \tag{9}$$

where $\delta(t)$ is the Dirac delta function. The amplitude and the time of impulse are calculated by

$$A_1 = \frac{1}{1+K}\,,\quad t_1 = 0\,,\quad A_2 = \frac{K}{1+K}\,,\quad t_2 = \frac{T_d}{2} \tag{10}$$

where $K = e^{-\zeta\pi/\sqrt{1-\zeta^2}}$, $T_d = 2\pi/\omega_d$.

However, the ZV shaper alone is not sufficient when external disturbances, such as wind or structural vibration, excite the pendulum and make the additional vibration controller necessary to suppress residual vibrations. In this study, we propose a PIPO feedback controller to reduce residual vibrations after the cart reaches the desired position. The PIPO feedback controller is given by

$$\ddot{u}+2\zeta_c\omega_n\dot{u}+\omega_n{}^2u=g_c\omega_n{}^2\theta \tag{11}$$

where ζ_c is the damping factor of the PIPO feedback controller and g_c is the control gain. The transfer function of the PIPO feedback controller can be written as

$$\frac{U(s)}{\Theta(s)}=\frac{g_c\omega_n^2}{s^2+2\zeta_c\omega_n s+\omega_n^2} \tag{12}$$

The proposed PIPO feedback control given by Eq. (12) has the same form as the positive position feedback (PPF) control [36] developed to control the smart structure equipped with piezoelectric wafers. However, the PIPO feedback control differs from the PPF control in that the output of the PIPO feedback control is the desired position of the cart, but the output of the PPF control is the actuating force. In addition, the PPF control was designed for the structural equation of motion which has control force instead of acceleration. The proposed PIPO feedback control can be easily implemented by using either an analogue circuit or a digital controller like the PPF control, because the PIPO feedback controller is, in fact, a low-pass filter. To apply the control law given by Eq. (12), the precise position tracking of the cart should be enabled.

Eqs. (5) and (11) lead to the coupled matrix equation:

$$\begin{aligned}&\begin{bmatrix}1 & g_t\\ 0 & 1\end{bmatrix}\begin{Bmatrix}\ddot{\theta}\\ \ddot{u}\end{Bmatrix}+\begin{bmatrix}2\zeta\omega_n & 0\\ 0 & 2\zeta_c\omega_n\end{bmatrix}\begin{Bmatrix}\dot{\theta}\\ \dot{u}\end{Bmatrix}\\ &\quad+\begin{bmatrix}\omega_n^2 & 0\\ -g_c\omega_n^2 & \omega_n^2\end{bmatrix}\begin{Bmatrix}\theta\\ u\end{Bmatrix}=\begin{Bmatrix}0\\ 0\end{Bmatrix}\end{aligned} \tag{13}$$

The denominator equation for the stability check can be derived as

$$\begin{aligned}&s^4+2(\zeta+\zeta_c)\omega_n s^3+(2+g_t g_c+4\zeta\zeta_c)\omega_n{}^2 s^2\\ &\quad+2(\zeta+\zeta_c)\omega_n{}^3 s+\omega_n{}^4=0\end{aligned} \tag{14}$$

The stability condition of the controller can be obtained by applying the Routh-Hurwitz criteria:

$$\text{Stable if } g_c>0 \tag{15}$$

Eq. (15) implies that the closed-loop system is stable if the gain of the controller is positive. In addition, there is no limit on the magnitude of the control gain. This stability condition is different from the stability condition of the PPF control and is very attractive for controlling the cart and the pendulum.

5. Numerical simulation and experiments

The ZV input shaper was applied to the step input motion of the cart, resulting in the command input, as shown in Fig. 7. However, the step-type position control is not possible in a real system, so the resulting cart position appears as shown in Fig. 8. Because of this, both the theoretical and the experimental results show that the pendulum vibration is not suppressed by the input-shaping control alone after the cart stops, as shown in Fig. 9.

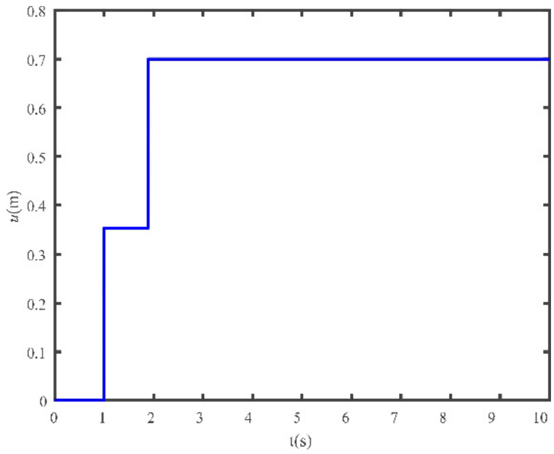

Fig. 7. ZV input-shaping command.

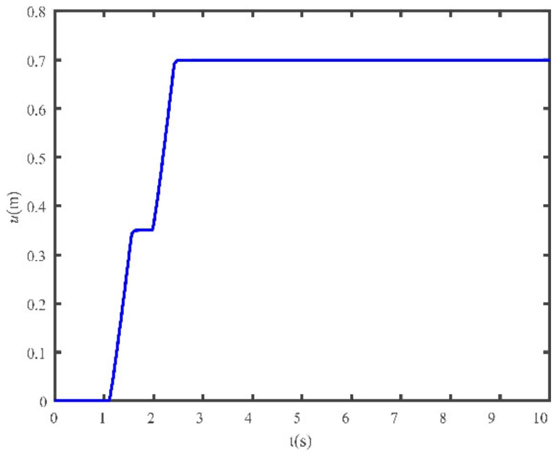

Fig. 8. Actual displacement of the cart by ZV input-shaping command.

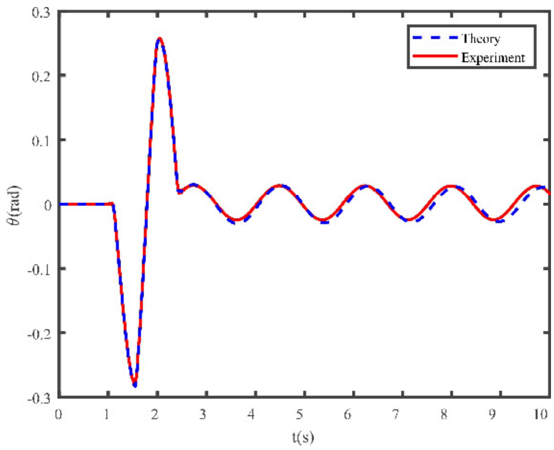

Fig. 9. Angular displacement of the pendulum by ZV input-shaping command (theory vs. experiment).

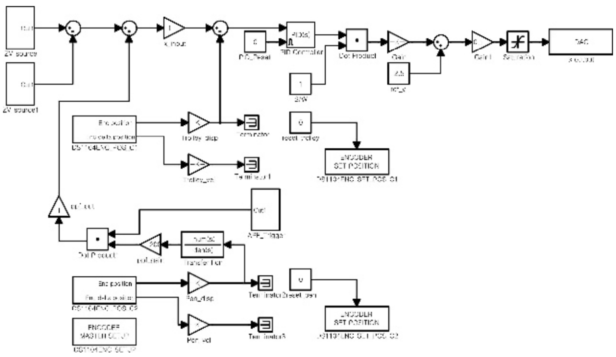

Fig. 10. Simulink block diagram for input-shaping control and PIPO feedback control.

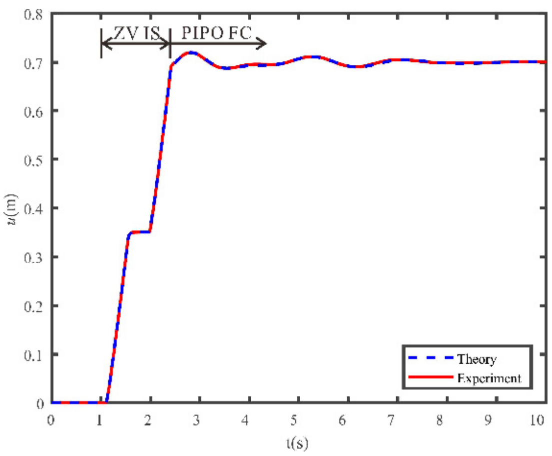

Fig. 11. Cart displacement by ZV input-shaping control and PIPO feedback control.

The proposed PIPO feedback controller was then combined with the ZV input-shaping control, as shown in Fig. 10, which was applied after the cart reaches the final position. The Simulink block diagram consists of the ZV input-shaping control, the PIPO feedback control, and the PID control for the motor control. Fig. 11 shows the displacement of the cart when the PIPO control is applied after the cart reaches the desired position.

Compared to the position obtained by the ZV control, the cart continues to move until the pendulum vibration is suppressed.

Experimental results, Fig. 13, are in good agreement with the theoretical results shown in Fig. 12. Therefore, it is proved both theoretically and experimentally that the PIPO feedback control proposed in this study is effective in suppressing residual vibrations. Its effectiveness becomes more evident when an external disturbance is present.

In the experiment, the fan blows wind to the pendulum, causing large-amplitude residual vibrations. The input-shaping control alone is not sufficient, as shown in Fig. 14. However, Fig. 15 also shows that vibrations caused by wind are successfully suppressed by the proposed PIPO feedback control. The corresponding cart position is shown in Fig. 15, which shows

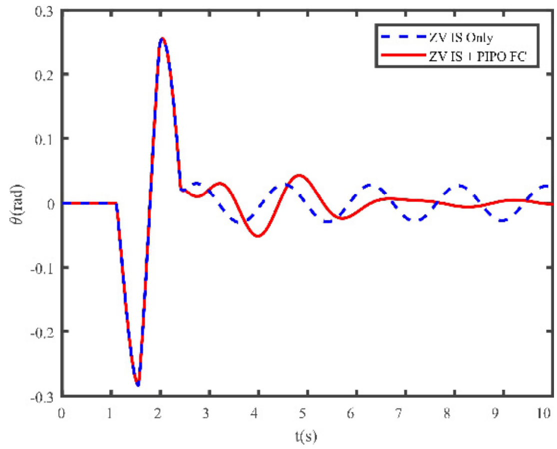

Fig. 12. Theoretical angular displacement with or without PIPO feedback control.

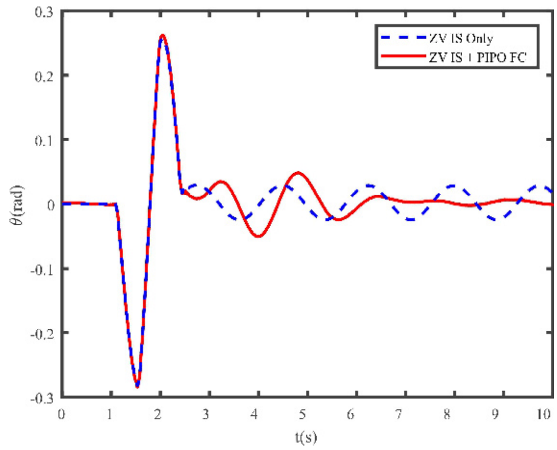

Fig. 13. Experimental angular displacement with or without PIPO feedback control.

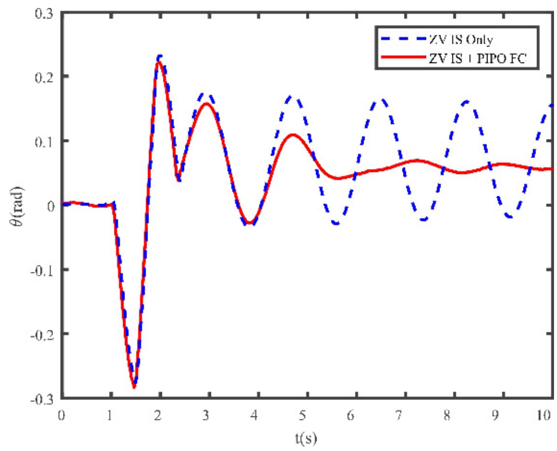

Fig. 14. Measured angular displacement with or without PIPO feedback control subject to wind disturbance.

the cart movement to suppress residual vibrations after the cart reaches the final position.

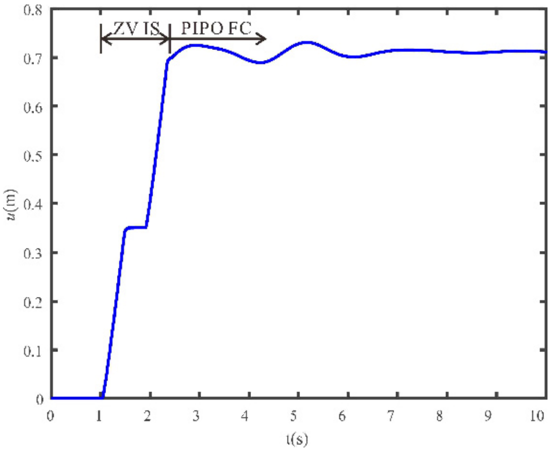

Fig. 15. Cart displacement by PIPO feedback control subject to wind disturbance.

6. Conclusions

In this study, an experiment was carried out to suppress the residual vibration of the pendulum caused by the movement of the cart and external disturbance. The ZV input-shaping control alone cannot suppress residual vibrations caused by external disturbances, so the PIPO feedback control is proposed in this study. The stability analysis shows that the proposed PIPO feedback control is stable if the control gain is positive and the stability condition is static.

The ZV input-shaping control combined with the PIPO feedback control was applied to the experimental cart-pendulum system. Both theoretical and experimental results show that the residual vibrations can be suppressed very quickly when the proposed control is used. The proposed PIPO control becomes more effective when vibrations are occurred by an unexpected disturbance. The experimental results showed that the ZV control alone cannot suppress the residual vibrations caused by an external disturbance, but the PIPO control can suppress them successfully.

Acknowledgments

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