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PRACTICAL SWAY MOTION CONTROL FOR DOUBLE PENDULUM-TYPE OVERHEAD CRANE SYSTEM

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Abstract- The sway motion of crane can be successfully suppressed by properly shaping the reference command. Input shaping is a one type of feed-forward shaping method that is based on linear superposition. In this paper, we present the impact of double pendulum type overhead crane (DPTOC) system on the effectiveness of input shaping. An unshaped bang-bang input force is used to determine the characteristic parameters of the system for design and evaluation of the input shaping control techniques. The input shapers with the derivative effects are designed based on the properties of the system. The response DPTOC system to shaped input is experimentally verified in time and frequency domain. The performance of the input shaper is examined in terms of sway angle reduction and time response specification. Experimental results demonstrate the effectiveness of the proposed approach in reducing the sway motion of crane system.

Index terms***:*** **Input shaping, double pendulum, and sway motion.**

I. INTRODUCTION

The sway motion of crane payloads causes safety hazards, time delays and difficulty in positioning. Much of the previous work on crane control has attempted to address this issue using techniques based around a single-pendulum model of a crane. Most of the experienced crane operators can eliminate much of the payload sway by causing an oscillation during deceleration that cancels the oscillation induced during acceleration if a crane behaves like a single pendulum. However, certain types of payloads and riggings result in double pendulum dynamics [1,2]. Under these conditions, the manual method of eliminating sway motion becomes very difficult, even for skilled operators. To overcome this problem, some researchers have suggested a feedback control to suppress the double pendulum dynamics [3,4]. However, it is very challenging due to the difficulty of measuring the payload motion. Since the control of sway or vibration of a kind of flexible structure required the frequencies information [5, 6], a classical and robust controller have been proposed in [7] and [8], respectively, to control the first mode of vibration. However, the results do not directly translated to higher number of vibration mode. As a consequence, the input shaping techniques is more preferable in this study.

This paper focuses on the application of input shaping schemes to reduce a hook and load sway motion of double pendulum type overhead crane. Input shaping is a method of command filtering that allows many oscillatory systems to be moved without inducing residual vibration. Input shaping is implemented by convolving a series of impulses, known as the input shaper, with a desired reference command. This produces a command that will drive the system while limiting residual vibration [9,10]. This process is illustrated in Figure 1 with a smooth initial reference command. Input shaping is attractive because the scheme only estimates the natural frequencies and damping ratios of the dominant modes of vibration. Input shaping relies on the superposition of impulse responses of a second-order system. To design the input shaper, the response of a second-order harmonic oscillator of frequency, ω, and damping ratio, ζ, to a series of n impulses is set equal to zero, or a near-zero value. The equation is then solved to determine the impulse amplitudes, A_i, and time locations, t_i, that would produce such a small value of residual vibration. This low-vibration impulse sequence is then used in the shaping process shown in Figure 1.

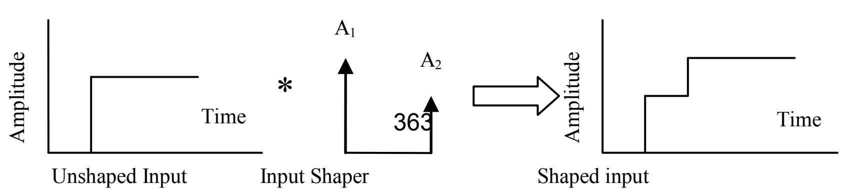

Figure 1. Illustration of input shaping technique.

The earliest form of input shaping was developed by Smith. However, his posicast control method was extremely sensitive to modeling errors [11]. This sensitivity to modeling errors prohibited the input shaper from practical use on many systems. Singer and Seering were the first to develop an input-shaping technique robust enough to be used in most practical applications. To reduce the sensitivity of the input shaper to errors in natural frequency, they set the derivative of the vibration with respect to the natural frequency to zero at the modeling frequency.

However, the rise-time penalty incurred for the added robustness of this shaper. To solve this drawback, Singhose and his co-workers [12] have proposed the extra-insensitive (EI) shaper. In order to increase the robustness of input shapers without adding additional time delays, the requirement of having exactly zero vibration at the natural frequency need to be relaxed. Instead of forcing the vibration to exactly zero value, it is allowed to equal some small nonzero value. As a consequence, the shaper can be more robust without incurring an additional rise-time penalty.

Input shaping was first implemented on a gantry crane at the Savannah River Technology Center [13]. Fixed-duration (FD) shapers were implemented on this crane, in which the shaper duration was held fixed while the robustness to modeling errors was maximized. This process creates a set of shapers for different payload suspension lengths with identical rise times. Constant rise times are desirable from an operator standpoint, as they do not have to adjust for variable deceleration times. In previous research, input shaping schemes has been proposed for sway angle suppression of various types of crane system [14,15,16]. Hong and Hong [17] showed simulation results for point-to-point motions of container cranes using a deflection-limiting input shaping technique and nonlinear vibration stabilization control.

This paper presents investigations into the development of input shaping schemes for anti-swaying control of a double- pendulum-type overhead crane (DPTOC) system. An experimental rig of DPTOC system is considered in this work. An unshaped bang-bang force input is used to determine the characteristic parameters of the system for design and evaluation of the input shaping control techniques. The positive zero-sway (PZS), positive zero-sway derivative (PZSD) and positive zero-sway-derivative-derivative (PZSDD) input shapers are then designed based on the properties of the system for anti-sway control. Experimental results of the response of the DPTOC system to the shaped inputs are presented in time and frequency domains. Performances of the shapers are examined in terms of swing angles reduction and time response specifications. Finally, a comparative assessment of the input shaping techniques with different derivative order is presented and discussed.

II. THE DOUBLE PENDULUM-TYPE OVERHEAD CRANE SYSTEM

The DPTOC system with its hook and load considered in this work is shown in Figure 2, where x is the trolley position, m is the trolley mass, and m_1 and m_2 are the hook and load mass respectively. θ_1 is the hook swing angle, θ_2 is the load swing angle, l_1 and l_2 are the cable length of the hook and load, respectively, and F is the trolley drive force. In this simulation, the hook and load can be considered as point masses.

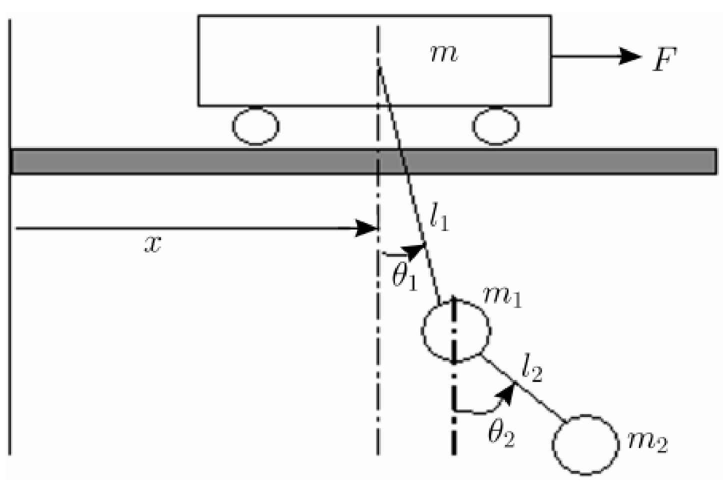

Figure 2. Description of the DPTOC system.

III. EXPERIMENTAL TEST BED

In order to verify the effectiveness of the proposed input shaping techniques, experimental research was conducted on a double pendulum type overhead crane system as shown in Figure 3. The pendulum system used in this study transmits the rotational power of the motor that is generated as the motor rotates through the ball screw and the rotation is changed into the straight line motion through the ball screw. The straight line motion of the ball screw moves the trolley that is connected to it and the pendulum angle that is connected to the trolley is controlled.

Figure 3. Double pendulum type overhead crane system at Control and Instrumentation Lab, UMP.

Since the overhead crane system using two pendulums, it requires two encoder sensors to sense the sway motion at hook and load of the pendulum. The location of the trolley is recognized by the encoder that is connected to the motor. The detail specification of the lab-scale overhead crane system is shown in Table 1. The input shaping schemes is designed and implemented using CEMTool and SIMTool software with the sampling period selected at 1 ms. The encoder sensor's signals from the angle and trolley motion are connected to analogue I/O Port of RG-DSPIO01 with a voltage range of -10V to +10V. The output of the controller is also sending to the analogue I/O Port of RGDSPIO01 using 25P connector.

Table 1: Specifications of double pendulum type overhead crane system

Item		Specifications
Mechanical	W x L x H (mm)	1330 x 200 x 250
	Length of hook	200mm
	Length of load	400mm
	Weight of hook	0.5kg
	Weight of load	0.2kg
	Displacement movement	*900mm*
	Ball screw pitch	12.7mm
Electrical	Motor Output	24V, 60W
	Maximum rotation of motor	*3800 rpm*
	Encoder Pulse	4000 pulses
	Motor input voltage	0-5V

IV. INPUT SHAPING CONTROL SCHEMES

The design objectives of input shaping are to determine the amplitude and time locations of the impulses in order to reduce the detrimental effects of system flexibility. These parameters are obtained from the natural frequencies and damping ratios of the system. This input shaping design has been motivated by the previous study from the Woodruff School of Mechanical Engineering, Georgia Institute of Technology [18,19]. They have study the input shaper design to suppress the multimode vibration from the double pendulum dynamics. The great finding of their research is the important of the second mode when the mass of the hook is significant when compared to the mass of the payload. They used the linearized natural frequencies of the double pendulum dynamics equation from Blevins [20] as a simple platform and verified the input shaper design to the portable crane system. The outcome of their study has been tabulated in Figure 4 which is show the variation of low and high frequencies as a function of the mass ratio and the length of load.

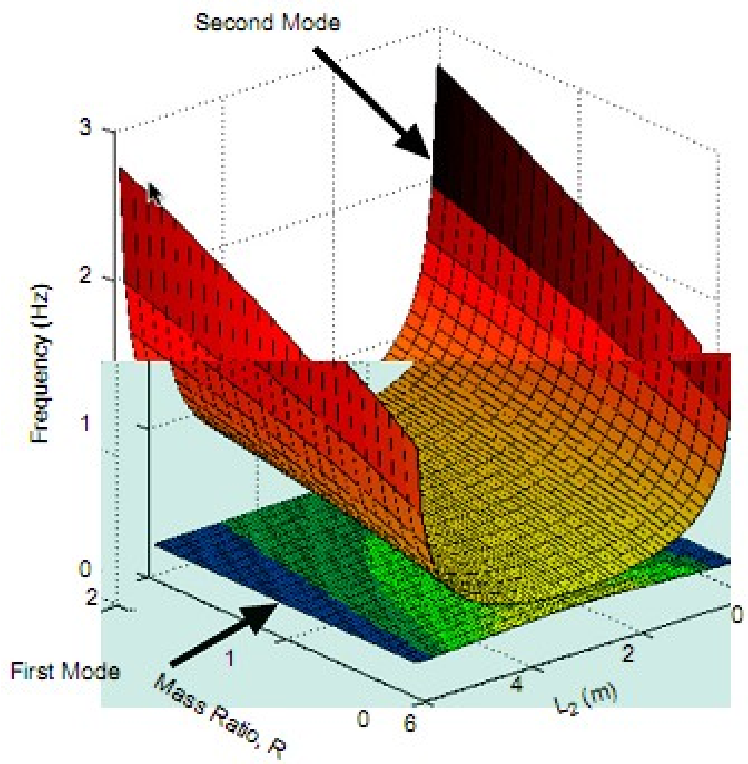

Figure 4: Variation of low and high frequencies [17].

However, their investigations only limited to two mode of sway frequency only. This study can be more practical if three modes of sway frequencies are considered to design the input shapers. Since the ratio of the payload mass to the hook mass is very significant in this study, the higher mode should be taken into account when designing the input shaper. The corresponding design relations for achieving a zero residual of the first three modes sway of a system and to ensure that the shaped command input produces the same rigid body motion as the unshaped command. Generally, a vibratory system of any order can be modelled as a superposition of second order systems each with a transfer function

$$G(s) = \frac{\omega^2}{s^2 + 2\zeta\omega s + \omega^2} \tag{1}$$

where ω is the natural frequency of the vibratory system and ζ is the damping ratio of the system. Thus, the response of the system in time domain can be obtained as

$$y(t) = \frac{A\omega}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega(t-t_0)} \sin\left(\omega\sqrt{1-\zeta^2}\,(t-t_0)\right) \tag{2}$$

where A and t_0 are the amplitude and the time location of the impulse respectively. The response to a sequence of impulses can be obtained by superposition of the impulse responses. Thus, for N impulses, with $\omega_d = \omega\left(\sqrt{1-\zeta^2}\right)$, the impulse response can be expressed as

$$y(t) = M \sin\left(\omega_d t + \beta\right) \tag{3}$$

where

$$M = \sqrt{\left(\sum_{i=1}^{N} B_i \cos\phi_i\right)^2 + \left(\sum_{i=1}^{N} B_i \sin\phi_i\right)^2},$$

$$B_i = \frac{A_i\omega}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega(t-t_0)}, \; \phi_i = \omega_d t_i$$

and A_i and t_i are the amplitudes and time locations of the impulses.

The residual single mode vibration amplitude of the impulse response is obtained at the time of the last impulse, t_N as

$$V = \sqrt{V_1^2 + V_2^2} \tag{4}$$

where

$$V_1 = \sum_{i=1}^{N} \frac{A_i\omega_n}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega_n(t_N-t_i)} \cos(\omega_d t_i);$$

$$V_2 = \sum_{i=1}^{N} \frac{A_i\omega_n}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega_n(t_N-t_i)} \sin(\omega_d t_i)$$

To achieve zero vibration after the last impulse, it is required that both V_1 and V_2 in Equation (4) are independently zero. This is known as the zero residual vibration constraints. In order to ensure that the shaped command input produces the same rigid body motion as the unshaped reference command, it is required that the sum of amplitudes of the impulses is unity. This yields the unity amplitude summation constraint as

$$\sum_{i=1}^{N} A_i = 1 \tag{5}$$

In order to avoid response delay, time optimality constraint is utilised. The first impulse is selected at time $t_1 = 0$ and the last impulse must be at the minimum, i.e. min (t_N). The robustness of the input shaper to errors in natural frequencies of the system can be increased by taking the derivatives of V_1 and V_2 to zero. Setting the derivatives to zero is equivalent to producing small changes in vibration corresponding to the frequency changes. The level of robustness can further be increased by increasing the order of derivatives of V_1 and V_2 and set them to zero. Thus, the robustness constraints can be obtained as

$$\frac{d^i V_1}{d\omega_n{}^i} = 0\,; \quad \frac{d^i V_2}{d\omega_n{}^i} = 0 \tag{6}$$

The positive ZS input shaper, i.e. two-impulse sequence is designed by taking into consideration the zero residual sway constraints, time optimality constraints and unity magnitude constraints. Hence, by setting V_1 and V_2 in Equation (4) to zero, $\sum_{i=1}^{N} A_i = 1$, $t_1 = 0$ to avoid response delay and solving yields a two-impulses sequence with parameters as

$$t_1 = 0, \;\; t_2 = \frac{\pi}{\omega_d},$$

$$A_1 = \frac{1}{1+K}, \; A_2 = \frac{K}{1+K} \tag{7}$$

where

$$K = e^{-\zeta\pi / \sqrt{1-\zeta^2}}, \quad \omega_d = \omega_n \sqrt{1-\zeta^2}$$

(ω_n and ζ representing the natural frequency and damping ratio respectively) and t_j and A_j are the time location and amplitude of impulse j respectively. The positive ZS shaper does not consider the robustness constraints. To increase the robustness of the positive input shaper, the robustness constraints must be considered in solving for the time locations and amplitudes of the impulses sequence. The robustness constraints equations can be obtained by setting the derivatives of V_1 and V_2 in Equation (4) to zero. By solving the zero-residual sway, robustness, unity magnitude and time optimality constraints yield a three-impulse sequence known as the positive ZSD shaper.

$$t_1 = 0,\ t_2 = \frac{\pi}{\omega_d},\ t_3 = \frac{2\pi}{\omega_d}$$

$$A_1 = \frac{1}{1+2K+K^2},\ A_2 = \frac{2K}{1+2K+K^2},\ A_3 = \frac{K^2}{1+2K+K^2} \tag{8}$$

To obtain a positive input shaper with higher level of robustness, another set of constraints equation, i.e. by setting the second derivatives of V_1 and V_2 in Equation (4) to zero must be considered in solving for the amplitudes and time locations of the impulse sequence. Simplifying $d^2V_i / d\omega_n^2$, yields

$$\begin{aligned} \frac{d^2V_1}{d\omega_n{}^2} &= \sum_{i=1}^{N} A_i t_i^2 e^{-\zeta\omega_n (t_N - t_i)} \sin(\omega_d t_i) \\ \frac{d^2V_2}{d\omega_n{}^2} &= \sum_{i=1}^{N} A_i t_i{}^2 e^{-\zeta\omega_n (t_N - t_i)} \cos(\omega_d t_i) \end{aligned} \tag{9}$$

The positive ZSDD input shaper, i.e. four-impulse sequence is obtained by setting Equations (4) and (9) to zero and solving with the other constraint equations. Hence, a four-impulse sequence can be obtained with the parameters as

$$t_1 = 0,\ t_2 = \frac{\pi}{\omega_d},\ t_3 = \frac{2\pi}{\omega_d},\ t_4 = \frac{3\pi}{\omega_d}$$

$$A_1 = \frac{1}{1+3K+3K^2+K^3},\ A_2 = \frac{3K}{1+3K+3K^2+K^3}$$

$$A_3 = \frac{3K^2}{1+3K+3K^2+K^3},\ A_4 = \frac{K^3}{1+3K+3K^2+K^3} \tag{10}$$

where K as is equation (1).

V. IMPLEMENTATION AND RESULTS

In this investigation, input shaping control schemes are implemented and tested within the experimental environment of the DPTOC system and the corresponding results are presented. The bang-bang input force of ±1 N is applied to the trolley of the DPTOC. The bang-bang input is required to have positive and negative period to allow the DPTOC to, initially, accelerate and then decelerate and eventually, stop at the target position. For the sway suppression schemes, positive zero-sway (PZS), positive zero-sway-derivative (PZSD) and positive zero-sway-derivative-derivative (PZSDD) are designed based on the sway frequencies and damping ratios of the DPTOC system. The first three modes of sway of the system are considered, as these dominate the dynamic of the system. The responses of the DPTOC system to the unshaped input were analyzed in time-domain and frequency domain (spectral density). These results were considered as the system response to the unshaped input and will be used to evaluate the performance of the input shaping techniques. The sway frequencies for both hook cable and load cable were obtained as 0.977 Hz, 1.953 Hz and 2.686 Hz for the first three modes of sway.

Figures 5-9 show the response of the trolley position, swing angle of hook and load cable and its power spectral density. Table 2 summarises the levels of sway reduction of the system responses at the first three modes. Higher levels of sway reduction were obtained using positive ZSDD shaper as compared using positive ZSD and ZS. However, with positive ZS shaper, the system response is faster. The corresponding rise time, settling time and overshoot of the trolley position response for positive shapers is depicted in Table 2. By comparing the results presented in Table 2, it is noted that the higher performance in the reduction of sway of the system is achieved using positive ZSDD shaper. This is observed and compared to the positive ZSD and ZS shaper at the first three modes of sway.

For comparative assessment, the levels of sway reduction of the hoisting angles of the hook and load cables using positive ZS, ZSD and ZSDD shapers are shown with the bar graphs in Figure 10 and Figure 11, respectively. The result shows that, highest level of sway reduction is achieved in control schemes using the positive ZSDD shaper, followed by the positive ZSD and ZS shaper for all modes of sway, for both of hook and load swing angles. Therefore, it can be concluded that the positive ZSDD shapers provide better performance in sway reduction as compared to the positive ZSD and positive ZS shapers in overall. Comparisons of the specifications of the trolley position response of input shaping control schemes using positive ZSDD, positive ZSD and ZS are summarized in Figure 12 for the rise and settling times. It is noted that settling time of the trolley position response by using the positive ZS shaper is faster than the case using the positive ZSDD shaper. It shows that, in term of settling time, the speed of the system response can be improved by using positive ZS shapers.

Table 2: Level of sway reduction of the hook and load swing angle of the pendulum and specification of trolley position response.

Types of shaper	Swing angle	Attenuation (dB) of sway			Specification of trolley position response		
		Mode 1	Mode 2	Mode 3	Rise time (s)	Settling time (s)	Overshoot (%)
PZS	Hook swing angle, θ_1	09.91	24.38	11.95	1.4114	2.8222	0.0036
	Load swing angle, θ_2	09.30	14.86	13.45			
PZSD	Hook swing angle, θ_1	14.46	28.19	11.95	1.4007	3.8834	0.0257
	Load swing angle, θ_2	15.31	42.01	20.97			
PZSDD	Hook swing angle, θ_1	16.18	37.62	18.38	1.7517	4.3984	1.0986

	Load swing angle, θ_2	15.73	42.86	23.55			

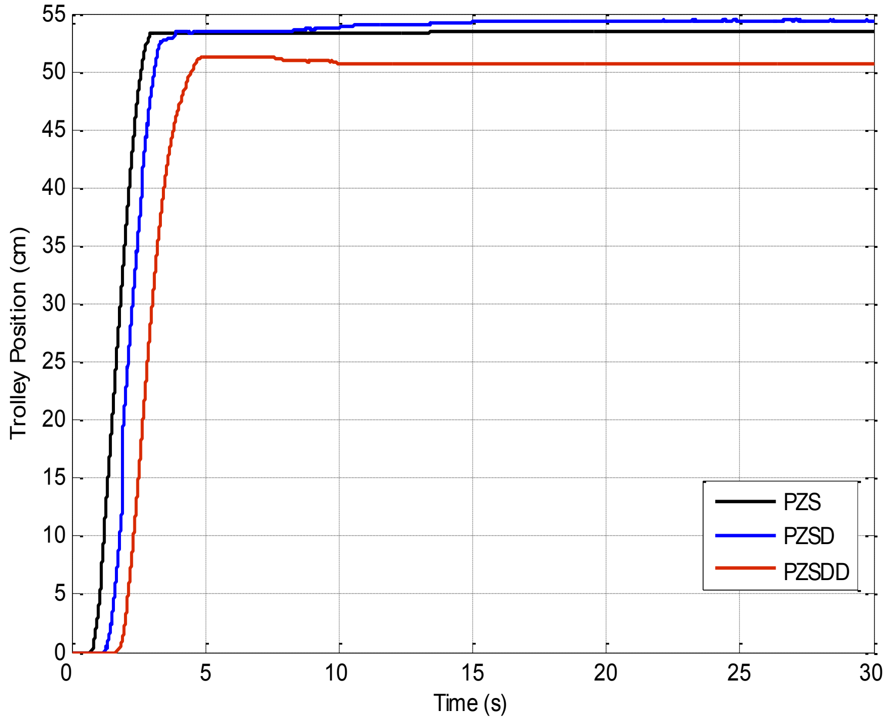

Figure 5. Response of the trolley position.

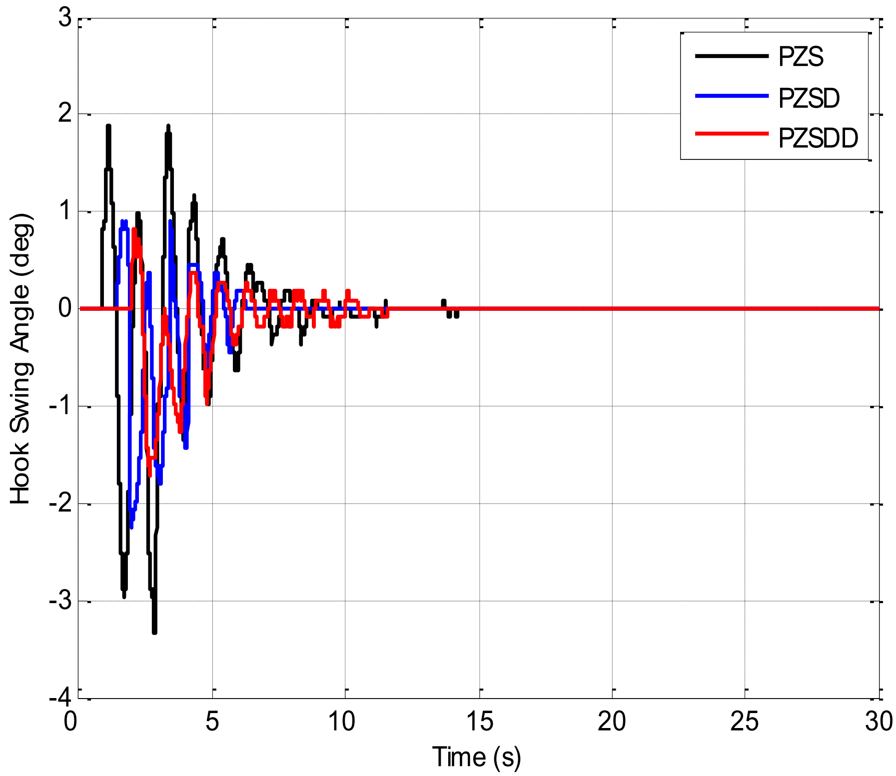

Figure 6. Response of the hook swing angle.

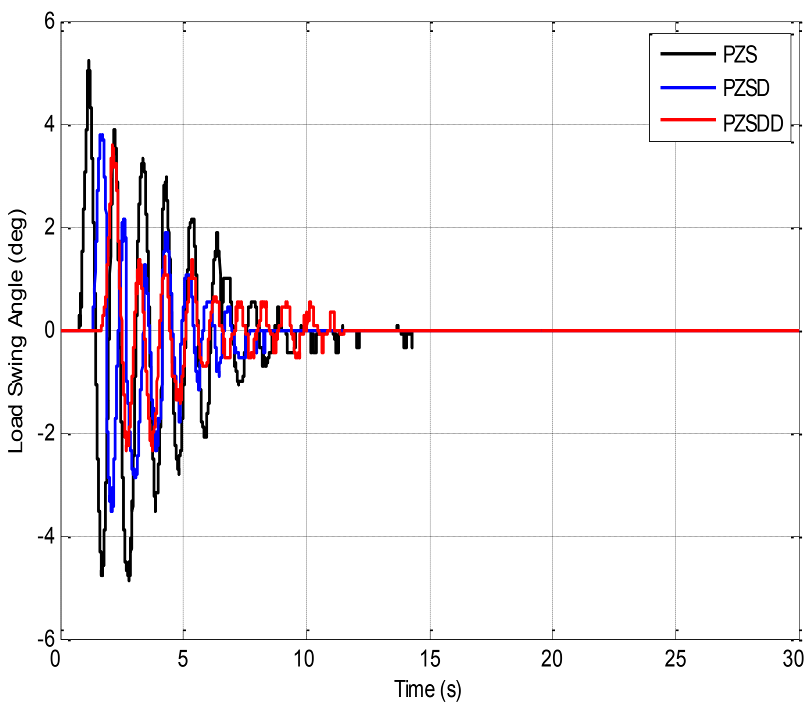

Figure 7. Response of the load swing angle.

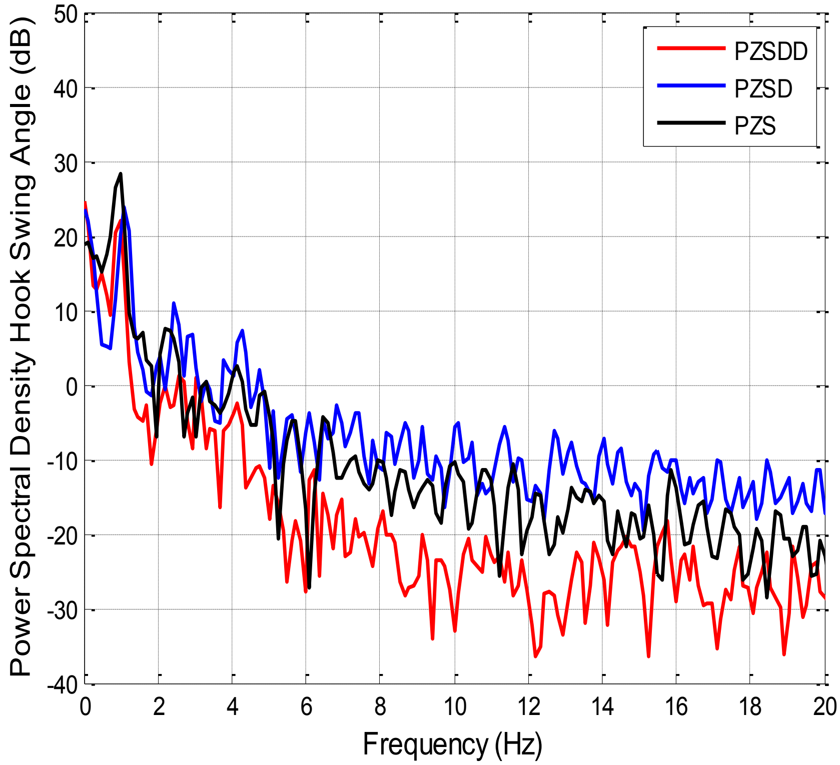

Figure 8. Power spectral density of the hook swing angle.

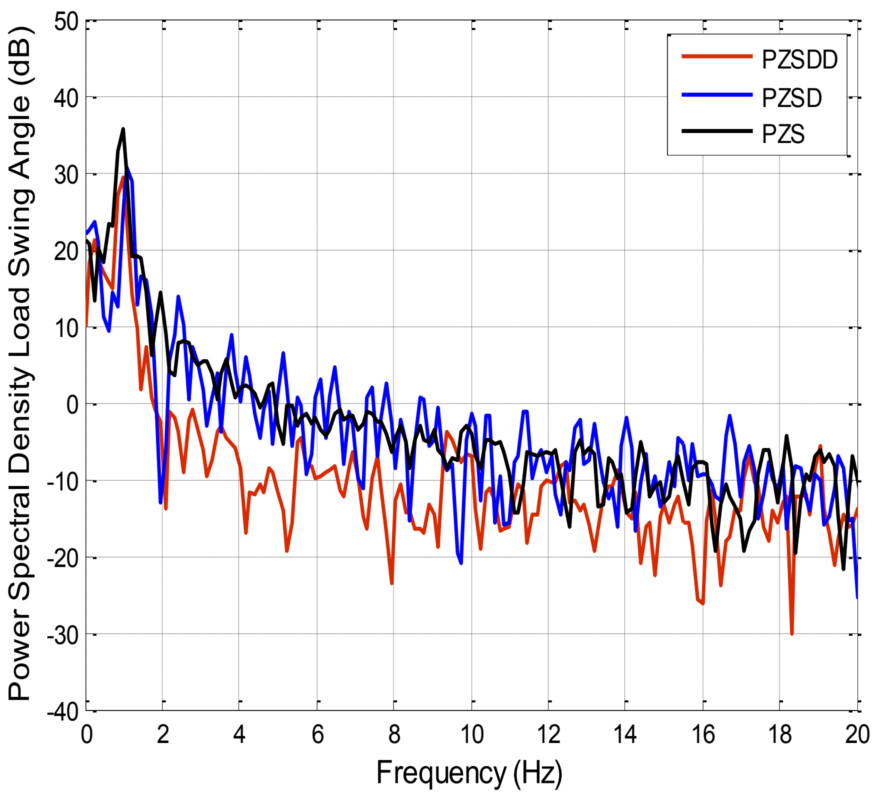

Figure 9. Power spectral density of the load swing angle.

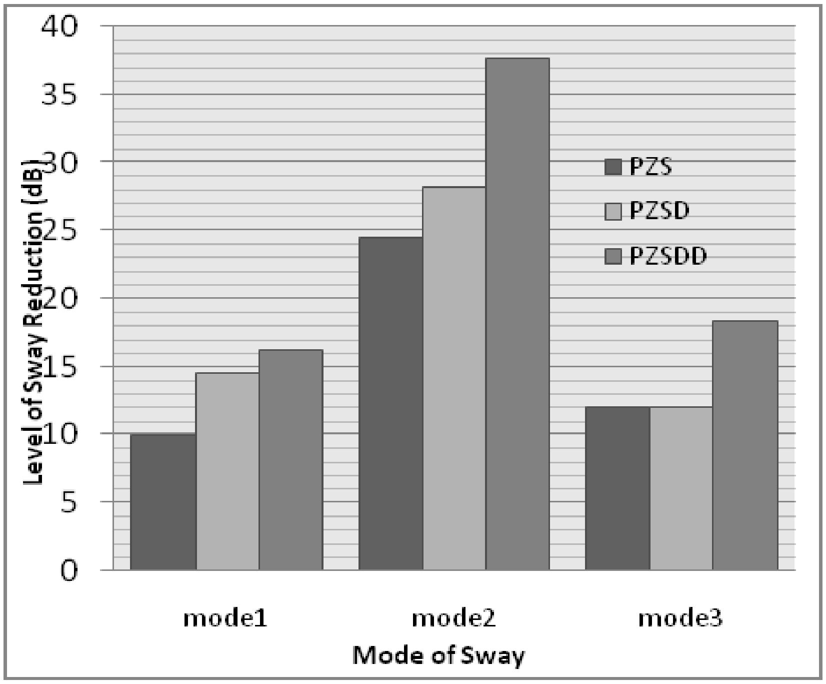

Figure 10. Level of sway reduction for hook swing angle.

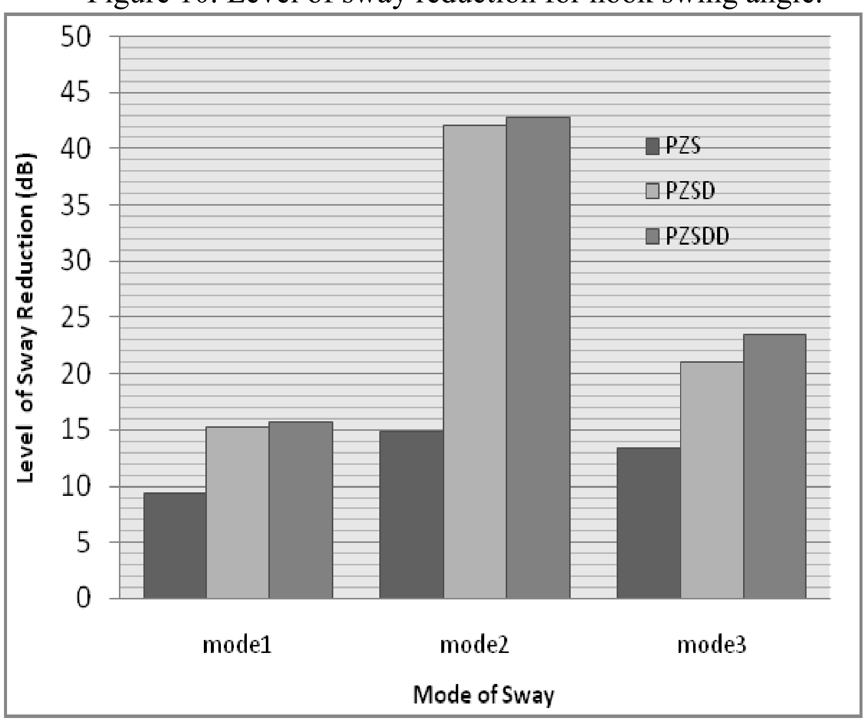

Figure 11. Level of sway reduction for load swing angle.

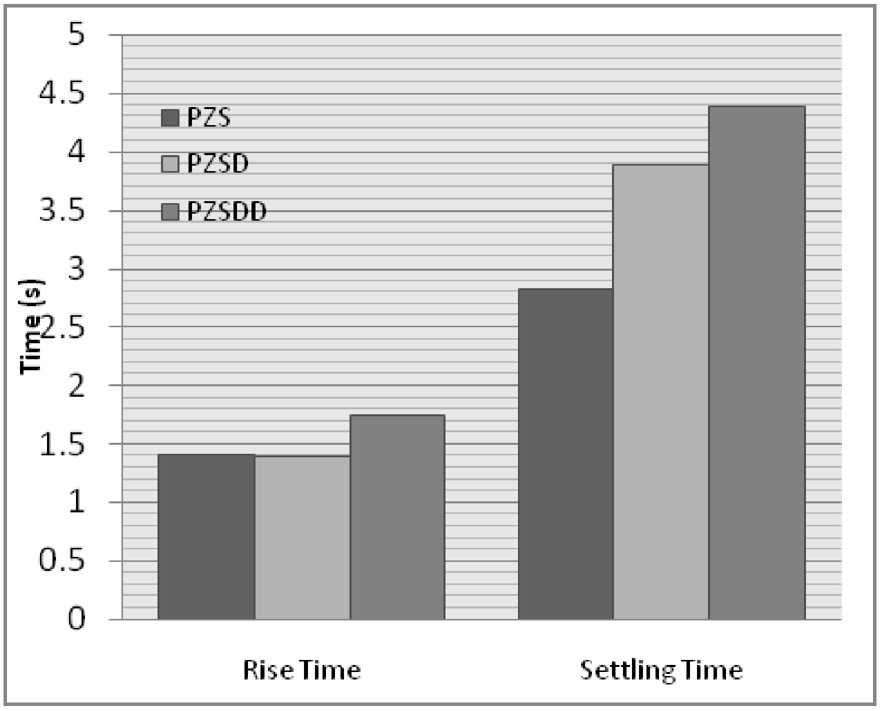

Figure 12. Rise and settling times of trolley position with PZS, PZSD and PZSDD shaper.

VI. CONCLUSIONS

The development of input shaping control schemes for anti-sway of a DPTOC system has been presented. The performances of the control schemes have been evaluated in terms of level of sway reduction and time response specifications. A comparison of the results has demonstrated that the positive ZSDD shapers provide higher level of sway reduction as compared to the cases using positive ZSD and ZS shapers. By using the positive ZS shapers, the speed of the response is slightly improved in term of settling time at the expenses of decrease in the level of sway reduction. It is concluded that the experiment results on DPTOC system has demonstrated the effectiveness and practicality of the proposed approach.

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