Anti-Sway Control Schemes of a Boom Crane Using Command Shaping Techniques

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Graphical abstract
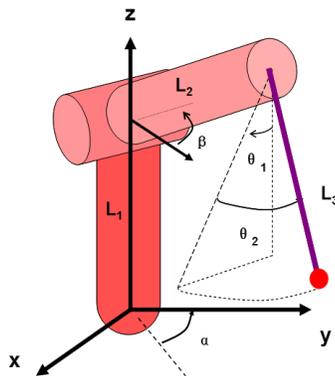
Abstract

This paper presents investigations into the applications and performance of command shaping techniques for control of payload sway of a boom crane based on filtering and the input shaping technique. The mathematical dynamic model describing the motion of the boom crane is developed using the Lagrange-Euler's equation. The dynamic characteristics of the system are studied and analysed using the Matlab Simulink in time and frequency domains. Command shaping techniques based on filtering and the input shaping techniques are then developed and used to control the payload sway of the boom crane. The performance of the control techniques are studied in terms of the level of sway reduction, time response and robustness. Finally, a comparative assessment of the effectiveness of the control schemes for sway control of a boom crane is presented and discussed.

Keywords: Anti-sway control; boom crane; command shaping; simulation

Abstrak

Kertas kerja ini membentangkan kajian ke atas aplikasi dan prestasi teknik pembentukan arahan untuk mengawal goyangan muatan 'boom crane' berdasarkan teknik penapisan dan pembentukan input. Model dinamik matematik yang menerangkan pergerakan 'boom crane' telah diperolehi dengan menggunakan persamaan Lagrange-Euler. Ciri-ciri dinamik sistem dikaji dan dianalisis menggunakan Matlab Simulink dalam domain masa dan frekuensi. Teknik-teknik pembentukan arahan berdasarkan penapisan dan teknik-teknik pembentukan input kemudiannya dibangunkan dan digunakan untuk mengawal goyangan muatan 'boom crane'. Prestasi teknik kawalan dikaji dari segi tahap pengurangan kuasa, masa tindakbalas dan kelasakan. Akhirnya, satu penilaian perbandingan tahap keberkesanan kawalan goyangan ke atas 'boom crane' dibentang dan dibincangkan.

Kata kunci: Kawalan anti-goyangan; 'boom crane'; pembentukan arahan; simulasi

■1.0 INTRODUCTION

Cranes are important machinery that has been intensively used for container handling in harbour and structural object shifting in construction sites. Generally cranes can be categorized into two major groups: gantry cranes and boom cranes. Boom cranes are common industrial structures that are used in building construction, factories, harbours, oil rig/platform and shipyard. Besides, they are widely used to transport heavy loads and hazardous materials in shipyards, factories, nuclear installations, and high building construction. These cranes are usually operated manually where operators use a joystick and an accelerator pedal to control the movements and direction of the cranes. On the shipyard, they are mounted on ships to transfer cargo between ships or on the harbour pavements to transfer cargo between ships and offshore structures.

In general, the movement of cranes has no prescribed path. They are used to move a load from one point to another. Boom cranes experience payload sway problems when commanded to perform fast motions. At very low speeds, the payload's sways are not significant and can be ignored. However, at a higher speed, these sway angles become larger and significant, and cause the payload hard to settle down when unloading. The overall system performance will be affected when significant sways angle of the payload occurs during the movement of a boom crane. This is a very severe problem especially for the applications in the industries that require high productivity and efficiency. Moreover, as most payloads are heavy, payload sway pose a safety hazard to workers and objects in the workspace.

With the size of these cranes becoming larger and higher, and the motion expected to be faster, the process of controlling them to guarantee fast turn-over and to meet safety requirements is a challenging task. The requirement of precise sway control of boom cranes implies that residual sway of the payload should be zero or near zero [1].

Over the years, investigations have been focused on the development of efficient controllers for gantry cranes. However, only a limited number of investigations have been carried out to devise control approaches to reduce the payload sway of boom cranes [2]. Feedback and feed-forward control strategies are two major sway control schemes that can be utilised. Feedback control techniques use measurement and estimations of the system states to attenuate the swaying of the system. Feedback controllers can be designed to be robust to parameter uncertainties. On the other hand, feed-forward techniques for sway attenuation involve developing the control input through consideration of the physical and swaying properties of the system, so that system swaying at dominant response modes is reduced. This method does not require additional sensors or actuators and does not account for changes in the system once the input is developed. For boom cranes, feed-forward and feedback control techniques can be used for sway attenuation and position control respectively. An acceptable system performance without payload sway that accounts for system changes can be achieved by developing a hybrid controller consisting of both control techniques. Thus, a properly designed feed-forward controller is required. Furthermore, the complexity of the required feedback controllers can be reduced.

A number of techniques have been proposed as feed-forward control strategies for sway control of boom cranes. A strategy to achieve a time-optimal slew motion only while minimising the residual sway has also been proposed [3]. In this work, a slew angle acceleration profile is shaped to perform the slew motion and control the sways. Numerical simulation has shown that the strategy provides considerable level of reduction of the payload sway. Lewis *et al.* [4-5] has applied quasi-static notch filters to the operator's input commands to avoid exciting the cable-payload assembly at its natural frequency. The notch location varies with the length of the cable to filter out excitations at the current natural frequency of the cable-payload assembly. Simulation results have shown reductions in both the in-plane and out-of-plane payload pendulations. However, the filtering process imposes delays between the operator input and the actual filtered input to the crane. Anti-sway control for boom cranes based on an optimal control approach has also been proposed [2]. Simulation and experimental results have shown that the proposed optimal trajectories are capable in reducing payload sway of the system.

An approach in command shaping techniques known as input shaping has been proposed by Singer and co-workers which are currently receiving considerable attention in vibration control [6]. The input shaping technique has been proven to be a practical and effective method of reducing vibrations especially in flexible systems [7]. The main concept of input shaping is to alter the input to the system in order to reduce unwanted dynamics which contributing to the swaying of the system. Its significance of this approach is that the physical of the system does not need to be altered. Using this method, a response without sway can be achieved, however, with a slight time delay approximately equal to the length of the impulse sequence. With more impulses, the system becomes more robust to flexible mode parameter changes, but this will result in longer delay in the system response.

This paper presents investigations into the applications and performance of the command shaping techniques for control of payload sway of a boom crane based on filtering and the input shaping techniques. Moreover, this paper provides a comparative assessment of the performance of these control schemes. In this work, the dynamic model describing the motion of the boom crane is derived using the Lagrange-Euler's equation. For the command shaping with the filtering technique, Butterworth low pass and band stop filters are considered. On the other hand, with the input shaping technique, input shaper with three impulse sequence known as zero-vibration-derivation (ZVD) is considered. Simulations of the system are performed within the boom crane simulation environment designed using Matlab and Simulink. Initially, to obtain the characteristic parameters of the system, the boom crane is excited with an unshaped bang-bang torque input. In this investigations, slew angle, tangential pendulation (in-plane) and radial sway (out-of-plane) responses of the crane are studied both in time and frequency domains. The filters and input shapers are designed based on the dynamic behaviour of the system and applied to minimise the swaying of the system. Performances of the developed controllers are assessed in terms of level of sway reduction, time response specifications and robustness to errors in the natural frequency. In this case, the robustness of the control schemes is assessed with up to 50% error tolerance in natural frequencies. Simulation results in time and frequency domains of the response of the boom crane to the unshaped input and shaped inputs with the filters and input shaper are presented. Moreover, a comparative assessment of the effectiveness of the controllers in minimising the swaying of the boom crane is discussed. The results of this work will be helpful in designing efficient algorithms for sway control of boom crane systems.

■2.0 THE BOOM CRANE SYSTEM

The boom crane system considered in this work is shown in Figure 1, where XOY represents the base coordinates. The three degrees of freedom crane system consists of a fixed vertical column, a rigid boom link, a hoisting line and a payload. α, β, L1, L2 and L3 represent the slew angle, luff angle, the length of the vertical column, the length of the rigid boom link and the length of the hoisting line respectively. The slew angle is the rotary angle of the hub of the boom crane or slewing pedestal controlled by the operator's slew command whereas the luff angle is the elevation or luffing angle of the boom link. Sway angles are excited as the system operates, namely the tangential pendulation, $\theta 1$ and the radial sway, $\theta 2$. In this particular case, the tangential pendulation and radial sway are in-plane and out-of-plane pendulations respectively. In this study, the payload is regarded as a point mass and the system exhibits the behavior of a pendulum. The length of the boom link, L2 of the system is considered as 1 m, gravitational force, g = 9.8 m/s2, weight of the point mass, m = 0.9 kg and the length of the hoisting line, L3 = 1m.
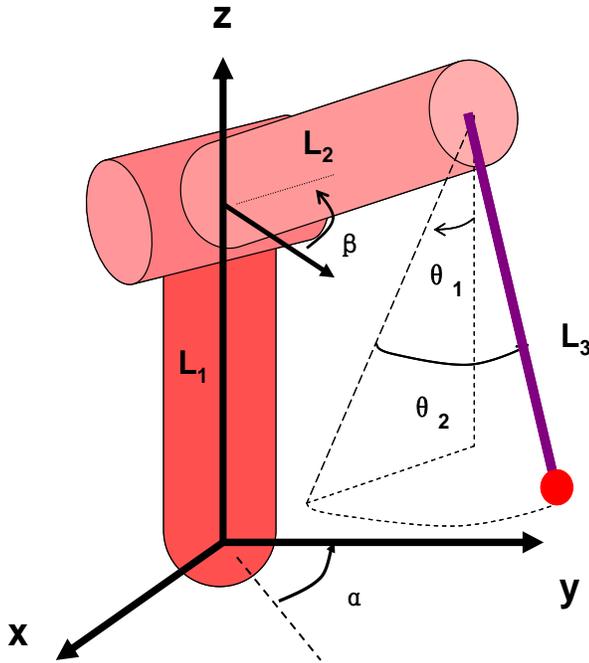
Figure 1 Description of the boom crane system

■3.0 MODELLING OF THE BOOM CRANE

This section provides a three dimensional mathematical modelling of the boom crane system, as a basis of a simulation environment for development and assessment of the command shaping control techniques. In this work, the dynamic equations of motion of the system are derived based on Lagrange-Euler equation. With the payload trajectory defined by the vector sum of the three kinematics links, L1, L2 and L3, each position vector is determined by transforming each link motion from a local coordinate frame to a fixed inertial coordinate frame attached to the base of the crane.

Analysing the schematic model in Figure 1, the link L1 coordinate system with respect to the base coordinate can be written as

$$z_1 = L_1 \tag{1}$$

Similarly, the link L_2 coordinate system (x_2, y_2 and z_2) with respect to the base coordinate can be written as

$$x_2 = - L_2 \sin \alpha \cos \beta \tag{2}$$

$$y_2 = L_2 \cos \alpha \cos \beta \tag{3}$$

$$z_2 = L_2 \sin \beta \tag{4}$$

and the link L_3 coordinate system (x_3, y_3 and z_3) with respect to the base coordinate as

$$x_3 = L_3 \sin \theta_2 \sin \alpha - L_3 \sin \theta_1 \cos \theta_2 \cos \alpha \tag{5}$$

$$y_3 = L_3 \sin \theta_2 \sin \alpha - L_3 \sin \theta_1 \cos \theta_2 \cos \alpha \tag{6}$$

$$z_3 = - L_3 \cos \theta_1 \cos \theta_2 \tag{7}$$

Finally, the total of x, y and z can be obtained as

$$x = - L_2 \sin \alpha \cos \beta + L_3 \sin \theta_2 \sin \alpha - L_3 \sin \theta_1 \cos \theta_2 \cos \alpha \tag{8}$$

$$y = L_2 \cos \alpha \cos \beta + L_3 \sin \theta_2 \sin \alpha - L_3 \sin \theta_1 \cos \theta_2 \cos \alpha \tag{9}$$

$$z = L_1 + L_2 \sin \beta - L_3 \cos \theta_1 \cos \theta_2 \tag{10}$$

Considering the payload as a point mass, m the potential energy can be obtained as

$$P = mgh = -mgz \tag{11}$$

where g is the gravitational acceleration and h is the height in z component. On the other hand, the kinetic energy can be obtained as

$$K = \frac{1}{2} m v^2 = \frac{1}{2} m (\dot{x}^2 + \dot{y}^2 + \dot{z}^2) \tag{12}$$

In this work, the Lagrange's Equation, $\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}}\right) - \frac{\partial L}{\partial q} = 0$, where $q = (\theta_1, \theta_2)$ and the Lagrange operator, $L = K - P$ is used to derive the dynamic equations of motion of the system. Due to the complexity of the equations, Matlab Symbolic Toolbox is utilized to derive and verify the equations. Solving the Lagrange's Equation yields the tangential pendulation and radial sway of the boom crane as

$$\ddot{\theta}_1 + \frac{2\dot{L}_3}{L_3}\dot{\theta}_1 + 2\dot{\alpha}\dot{\theta}_2 + (-\dot{\alpha}^2 - \frac{L_2 \sin(\beta)\dot{\beta}^2}{L_3} + \frac{L_2 \cos(\beta)\ddot{\beta}}{L_3} + \frac{g}{L_3})\theta_1 + (\ddot{\alpha} + \frac{2\dot{\alpha}\dot{L}_3}{L_3})\theta_2$$
$$= \frac{L_2 \cos(\beta)\ddot{\alpha}}{L_3} + \frac{2L_2 \sin(\beta)\alpha\beta}{L_3} \tag{13}$$

$$\ddot{\theta}_2 - 2\dot{\alpha}\dot{\theta} + \frac{2\dot{L}_3}{L_3}\dot{\theta}_2 + (\frac{2\dot{\alpha}\dot{L}_3}{L_3} - \ddot{\alpha})\theta_1 + (-\dot{\alpha}^2 - \frac{L_2 \sin(\beta)\dot{\beta}^2}{L_3} + \frac{L_2 \cos(\beta)\ddot{\beta}}{L_3} + \frac{g}{L_3})\theta_2$$
$$= \frac{L_2 \cos(\beta)\dot{\alpha}^2}{L_3} + \frac{L_2 \sin(\beta)\ddot{\beta}}{L_3} + \frac{L_2 \cos(\beta)\beta^2}{L_3} \tag{14}$$

In this work, the unshaped bang-bang input of amplitude ± 0.3 rad/s^2 is used as the input command. Figure 2 shows the single-switch bang-bang acceleration of the joystick and foot pedal used as the input to the system. The inputs are applied at the slew (hub) of the boom crane. The bang-bang input is required to have positive and negative periods to allow the boom crane to, initially, accelerate and then decelerate and eventually, stop at the target position. Three system responses namely slew-angle, tangential pendulation of payload and radial sway of payload of the boom crane are obtained. To investigate the behaviour of the system in the frequency domain, power spectral density (PSD) of the tangential pendulation response is also studied. The results are recorded with a sampling frequency of 1 KHz. In this work, the first three modes of sway/oscillation are considered, as these dominantly characterise the behaviour of the crane system.
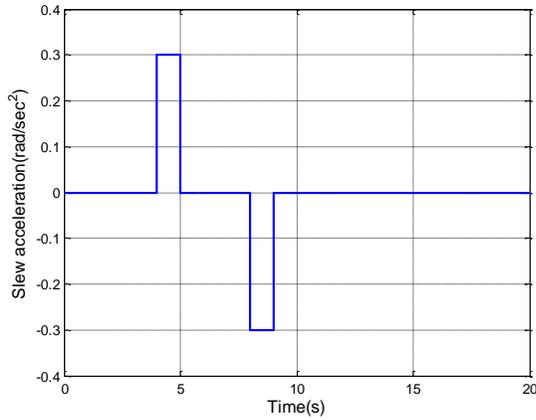
Figure 2 The bang-bang input
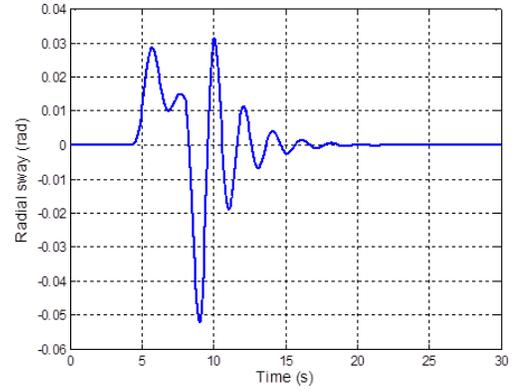
Figure 3 shows the slew-angle response, the radial sway and tangential pendulation of payload of the crane to the unshaped bang-bang input in time and frequency domains. It is noted that the steady-state slew-angle of 1.2 rad for the boom crane system with no overshoot was achieved within the settling time of 8.6 s. The radial sway and tangential pendulation responses reveal that a significant payload sway occurs during the movement of the boom crane. The responses were found to oscillate between -0.05 and 0.03 rad and between -0.05 and 0.05 degrees for the tangential pendulation and radial sway respectively. Moreover, the responses settle down only after 20 s. Natural frequencies of the system were obtained by transforming the time-domain representation of the system responses into the frequency domain using power spectral analysis. Figure 3(d) shows the PSD of the tangential pendulation of the system. The natural frequencies of the boom crane system were obtained as 0.48, 1.37 and 2.35 Hz for the first three modes respectively. These results were considered as the system response to the unshaped input and will be used to evaluate the performance of the command shaping techniques in suppressing the boom crane payload oscillations.

(a) Slew-angle

(b) Radial sway of the payload
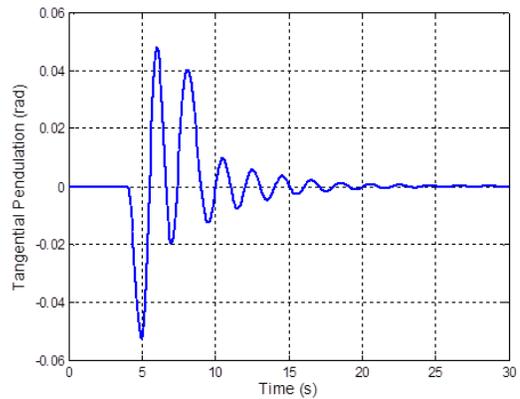
(c) Tangential pendulation of the payload
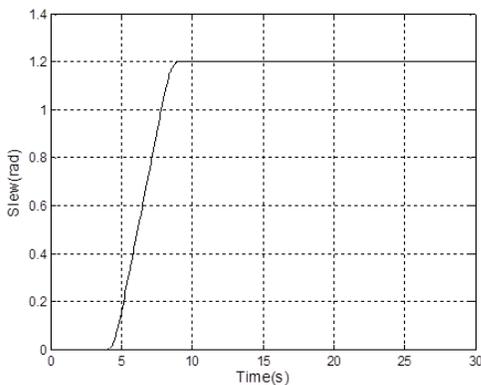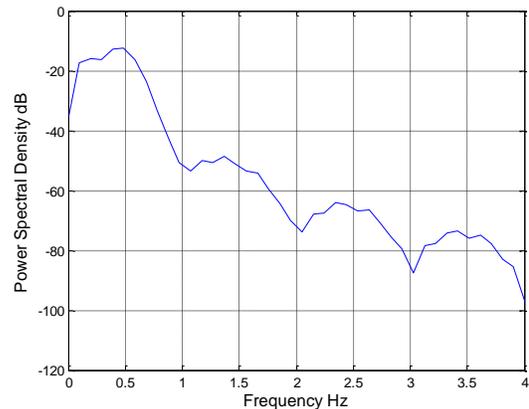
(d) PSD of tangential pendulation of the payload

Figure 3 Response of the boom crane to the unshaped bang-bang input

■4.0 COMMAND SHAPING CONTROL SCHEMES

Command shaping techniques are feed-forward control technique that functions as a notch filter on the commanded maneuver to remove the payload pendulation frequencies. In this work, filtering and input shaping techniques are investigated for control of payload sway of a boom crane. The following section provides a brief description of both feed-forward control techniques.

4.1 Filtering Technique

In this work, command shaping based on feed-forward filtering technique is developed on the basis of extracting the energies around the natural frequencies of the system using filtering techniques. The input signal is shaped to notch out the spectral components at system's resonance frequencies. The filters are thus used for pre-processing the input signal so that no energy is fed into the system at the natural frequencies. In this manner, the sway modes of the system are not excited, leading to a sway-free motion.

The low pass filter (LPF) and band stop filter (BSF) are used to shape the input command. The low pass filter is designed with a cut-off frequency lower than the first natural frequency of the oscillation of the system. As for band stop filter, the filter is designed with its center frequencies at the natural frequencies of the system. The low-pass and band stop filters thus designed are then implemented in cascade to preprocess the input signal. With the reference to the natural frequency obtained from the spectral analysis of the system, the filters are designed to eliminate the energy components in that frequency. There are various filter types such as Butterworth, Chebyshev and Elliptic that can be designed and employed. In these investigations, infinite impulse response (IIR) Butterworth low-pass and band-stop filters are examined.

4.2 Input Shaping Technique

Input shaping technique is a feed-forward control technique that involves convolving a desired command with a sequence of impulses known as input shaper [6]. The shaped command that results from the convolution is then used to drive the system. Design objectives are to determine the amplitude and time locations of the impulses, so that the shaped command reduces the detrimental effects of system flexibility. These parameters are obtained from the natural frequencies and damping ratios of the system. Thus, sway reduction of a boom crane system can be achieved with the input shaping technique. Several techniques have been investigated to obtain an efficient input shaper for a particular system. A brief description and derivation of the control technique is presented in this section.

Generally, a vibratory system of any order can be modeled as a superposition of second order systems each with a transfer function

$$G(s) = \frac{\omega^2}{s^2 + 2\zeta\omega s + \omega^2} \tag{15}$$

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

$$y(t) = \frac{A\omega}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega(t-t_0)} \sin\left(\omega\sqrt{1-\zeta^2}(t-t_0)\right) \tag{16}$$

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

$$y(t) = M \sin(\omega_d t + \beta) \tag{17}$$

where

$$M = \sqrt{\left(\sum_{i=1}^{N} B_i \cos\phi_i\right)^2 + \left(\sum_{i=1}^{N} B_i \sin\phi_i\right)^2} \tag{18}$$

$$B_i = \frac{A_i\omega}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega(t-t_0)} \tag{19}$$

$\phi_i = \omega_d t_i$ and A_i and t_i are the amplitudes and time locations of the impulses.

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

$$V = \sqrt{V_1^2 + V_2^2} \tag{20}$$

where

$$V_1 = \sum_{i=1}^{N} \frac{A_i\omega_n}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega_n(t_N-t_i)} \cos(\omega_d t_i) \tag{21}$$

$$V_2 = \sum_{i=1}^{N} \frac{A_i\omega_n}{\sqrt{1-\zeta^2}} \exp^{-\zeta\omega_n(t_N-t_i)} \sin(\omega_d t_i) \tag{22}$$

To achieve zero sway after the last impulse, it is required that both V_1 and V_2 in Equation (20) are independently zero. This is known as the zero residual sway 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

$$\sum_{i=1}^{N} A_i = 1 \tag{23}$$

In order to avoid response delay, time optimality constraint is utilized. The first impulse is selected at time $t_1 = 0$ and the last impulse must be at the minimum. 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 sway 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

$$\frac{d^i V_1}{d\omega_n^i} = 0 ; \quad \frac{d^i V_2}{d\omega_n^i} = 0 \tag{24}$$

The ZV 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 (20) 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

$$t_1 = 0, \quad t_2 = \frac{\pi}{\omega_d}, \quad A_1 = \frac{1}{1+K}, \quad A_2 = \frac{K}{1+K} \tag{25}$$

where

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

The ZV 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 (20) to zero. By solving the zero-residual sway, robustness, unity magnitude and time optimality constraints yield a three-impulse sequence known as the ZVD shaper.

Hence, a three-impulse sequence can be obtained with the parameters as

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

$$\left.\begin{aligned} A_1 &= \frac{1}{1+3K+3K^2+K^3} \\ A_2 &= \frac{3K}{1+3K+3K^2+K^3} \\ A_3 &= \frac{3K^2}{1+3K+3K^2+K^3} \;, \\ A_4 &= \frac{K^3}{1+3K+3K^2+K^3} \end{aligned}\right\} \qquad (26)$$

where K as is equation (25).

In order to handle higher sway modes, an impulse sequence for each sway mode can be designed independently. Then, the impulse sequences can be convoluted together to form a sequence of impulses that attenuate sway at higher modes.

◼ 5.0 IMPLEMENTATION AND RESULT

The feed-forward control techniques were designed on the basis of vibration frequencies and damping ratios of the boom crane system. These were obtained from the developed dynamic model of the boom crane as presented in the previous section. For evaluation of robustness, the control techniques were designed based on 50% error tolerance in the sway frequencies. As a consequence, the system oscillation modes were considered at 0.72 Hz, 2.06 Hz and 3.53 Hz under this situation. The filters and input shapers thus designed were used for pre-processing the bang-bang input command. The shaped and filtered torque inputs were then applied to the system in an open-loop configuration as shown in Figure 4 to reduce the sway of the payload of the boom crane. In this process, the shaped and filtered inputs were designed within the Matlab and Simulink environment with a sampling frequency of 1 kHz.

Simulation results of the response of the boom crane to the shaped and filtered inputs are presented in this section in the time and frequency domains. To investigate the performance of both techniques, the results are examined in comparison to the unshaped bang-bang input for a similar input level in each case. Similarly, three system responses are investigated namely the slew-angle, tangential pendulation and radial sway of the payload. Moreover, the PSD of the tangential pendulation response is evaluated to investigate the dynamic behavior of the system in frequency domain. Three criteria are used to evaluate the performances of the control schemes:

(1) Level of sway reduction at the natural frequencies. This is accomplished by comparing the responses to the shaped and filtered inputs with the response to the unshaped input.

(2) The time response specifications. Parameters that are evaluated are settling time and overshoot of the slew angle response. The settling time is calculated on the basis of ⬚2% of the steady-state value. Moreover, the magnitude of oscillation of the system response is observed.

(3) Robustness to parameter uncertainty. To examine the robustness of the techniques, the system performance is assessed with 50% error tolerance in natural frequencies. This is incorporated in the design of the filters and input shapers.

In this work, for a valid performance comparison of the control schemes in suppression of payload oscillations/sways, the parameters of the low-pass and band-stop filters and ZVD that provide a similar settling time of the slew-angle response are chosen.
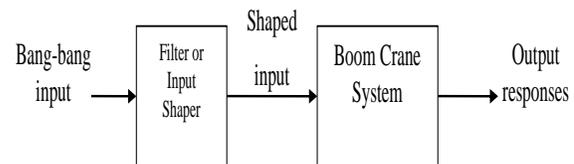
Figure 4 Block diagram of feed-forward control configuration

5.1 Filtered and Shaped Inputs

Using the low-pass filter, the input energy at all frequencies above the cut-off frequency can be attenuated. In this study with several investigations to achieve similar settling time of the slew-angle response, low-pass filters with cut-off frequency at 65% of the first sway mode were designed. Thus, for the boom crane, the cut-off frequencies of the filters were selected at 0.17 Hz and 0.26 Hz for the two cases of exact and 50% erroneous natural frequencies respectively. On the other hand, using the band-stop filter, the input energy at selected (dominant) resonance modes of the system can be attenuated. In this study, band-stop filters with bandwidth of 0.54 Hz were designed for the first three resonance modes. Similarly, the filters were designed with consideration of exact and 50% error in natural frequencies. In both cases, sixth order filters were designed and examined. The filtered inputs with the low-pass and band-stop filters are shown in Figure 5.

Using the properties of the system, an input shaper with three-impulse sequence (ZVD) was designed for three sway modes of the system. With the exact natural frequencies of 0.48 Hz, 1.37 Hz and 2.35 Hz, the time locations and amplitudes of the impulses were obtained by solving equation (26). For evaluation of robustness, input shapers with error in natural frequencies were also evaluated. With the 50% error in natural frequencies, the system sways were considered at 0.72 Hz, 2.06 Hz and 3.53 Hz for the three modes of sway. Similarly, the amplitudes and time locations of the input shapers with 50% erroneous natural frequencies for the ZVD input shapers were calculated. For digital implementation of the input shapers, locations of the impulses were selected at the nearest sampling time. The shaped input using ZVD shaper with exact natural frequency is shown in Figure 5.
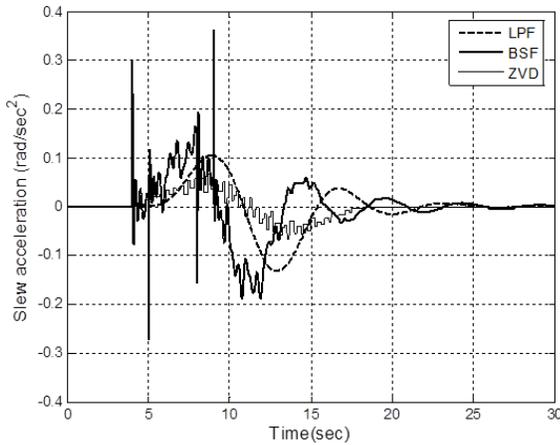
Figure 5 The filtered and shaped inputs

5.2 Exact Natural Frequencies

Figure 6 shows the slew-angle responses of the boom crane with the three control schemes with exact natural frequencies. As designed, all the responses give almost similar settling times of 16 s and achieve a similar level of steady-state slew angle of 1.2 rad. Moreover, the slew-angle responses are achieved with overshoots of 5.9%, 5% and 0% using LPF, BSF and ZVD respectively.

The tangential pendulation and radial sway responses of the payload of the boom crane with the three control schemes are shown in Figures 7 and 8 respectively. It is noted that the magnitude of the in-plane and out-of-plane sways were significantly reduced with the controllers as compared to the unshaped input (Figure 3). With the tangential pendulation response, the responses were found to oscillate between -0.01 to 0.02 rad, -0.013 to 0.023 rad and -0.005 to 0.023 rad using LPF, BSF and ZVD respectively. On the other hand, with the radial sway, the responses were found to oscillate between -0.014 to 0.022 rad, -0.078 to 0.025 rad and -0.004 to 0.007 rad respectively. The results show sway reduction of more than 64% and 52% of the unshaped bang-bang input for the tangential pendulation and radial sway responses respectively. It is clearly shown that ZVD provides the best performance in suppressing sway of the payload.
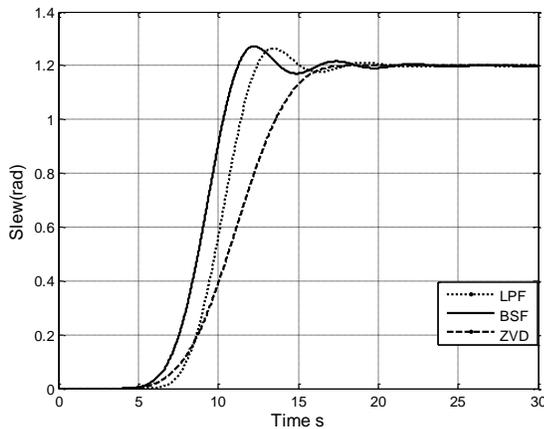
Figure 6 Slew angle responses of the boom crane using LPF, BSF and ZVD with exact frequencies
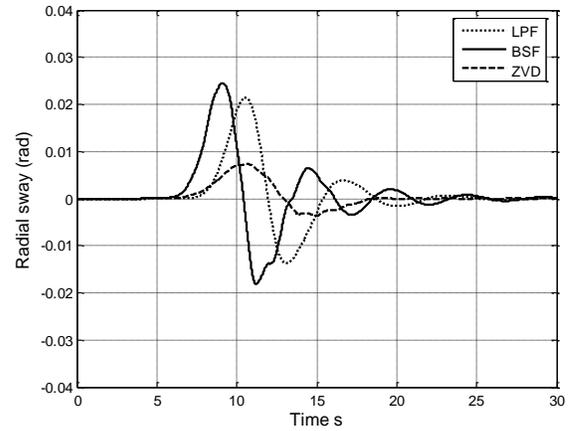
Figure 7 Tangential pendulation responses of the boom crane payload using LPF, BSF and ZVD with exact frequencies
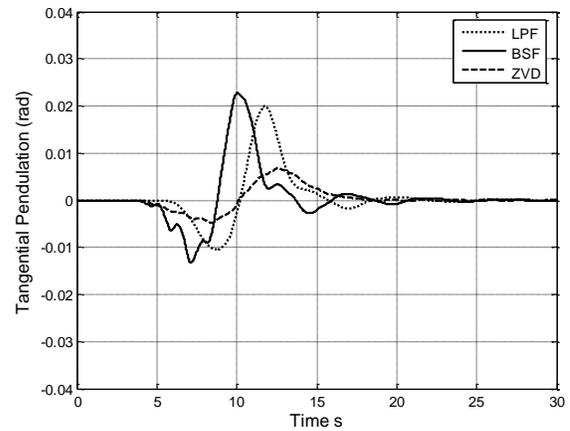
Figure 8 Radial sway responses of the boom crane payload using LPF, BSF and ZVD with exact frequencies

Figure 9 shows the PSD of the tangential pendulation of the boom crane payload using the control techniques. It is noted that the payload oscillation/sway at the resonance modes have significantly been reduced as compared to the unshaped bang-bang input. In this case, lower magnitudes of the PSD were achieved. Table 1 summarizes the PSD magnitudes of tangential pendulation response at the resonance modes using the control schemes as compared to the response with the unshaped bang-bang input.
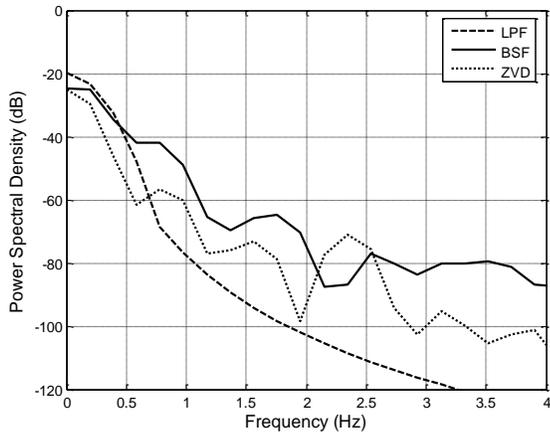
Figure 9 PSD of the tangential pendulation responses of the boom crane payload using LPF, BSF and ZVD with exact frequencies.

Table 1 Level of sway reduction of the tangential pendulation using LPF, BSF and ZVD as compared to the bang-bang input

Frequency	Control techniques	Attenuation (dB) of sway of tangential pendulation		
		Mode 1	**Mode 2**	**Mode 3**
Exact	LPF	9.78	5.38	4.4
	BSF	9.78	3.1	2.53
	ZVD	12.73	3.7	0.64
Error	LPF	9.78	4.26	3.01
	BSF	6.02	3.1	1.24
	ZVD	11.08	3.7	4.0

5.3 Robustness

To examine the robustness of the shaper, the shaper with 50% error in natural frequencies were designed and implemented to the boom crane system. Figure 10 shows the slew-angle responses of the boom crane with the three control schemes with erroneous natural frequencies. It is noted that, a similar level of steady-state slew angle of 1.2 rad is achieved with the control schemes. However, the settling times of the slew-angle response are achieved as 11 s, 12 s and 14 s using LPF, BSF and ZVD respectively, which are faster as compared to the case with exact frequencies. Moreover, the slew-angle responses are achieved with overshoots of 2.5%, 2.4% and 0% respectively. In this case, a better time response is achieved as a shorter impulse sequence and lower cut-off frequencies are used.

Figure 10 Slew angle responses of the boom crane using LPF, BSF and ZVD with erroneous frequencies

The tangential pendulation and radial sway responses of the payload of the boom crane using the control schemes with erroneous frequencies are shown in Figures 11 and 12 respectively. It is noted that the magnitude of the in-plane and out-of-plane sways were considerable reduced with the controllers as compared to the unshaped input. With the tangential pendulation response, the sway reductions of 61%, 40% and 83% were achieved using LPF, BSF and ZVD respectively. On the other hand, with the radial sway, the sway reductions of 48%, 11% and 83% were achieved respectively. However, the results are less than the case without error.
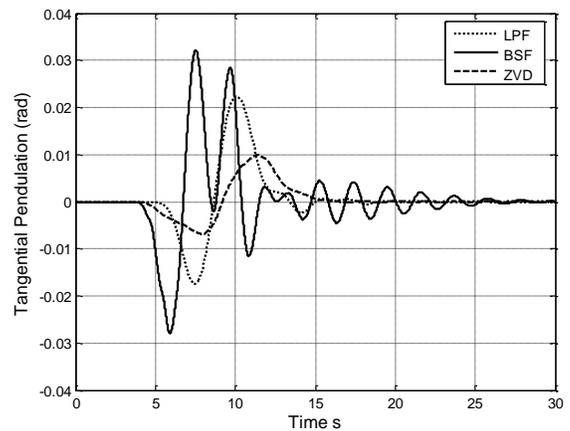
Figure 11 Tangential pendulation responses of the boom crane payload using LPF, BSF and ZVD with erroneous frequencies
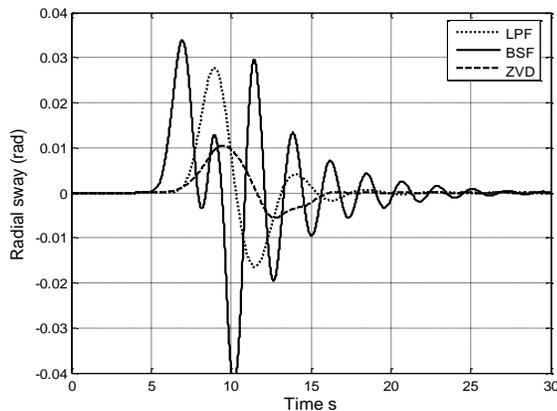
Figure 12 Radial sway responses of the boom crane payload using LPF, BSF and ZVD with erroneous frequencies

Figure 13 shows the PSD of the tangential pendulation of the boom crane payload using the control techniques with erroneous frequencies. As in the time domain, the payload oscillation/sway at the resonance modes have considerable been reduced as compared to the unshaped bang-bang input. Table 1 summarizes the PSD magnitudes of tangential pendulation response at the resonance modes using the control schemes as compared to the response with the unshaped bang-bang input.
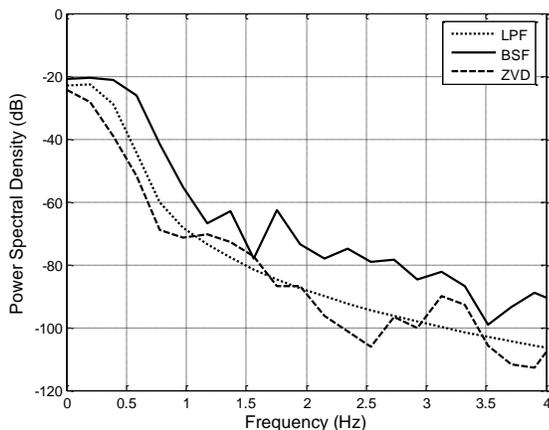
Figure 13 PSD of the tangential pendulation responses of the boom crane payload using LPF, BSF and ZVD with erroneous frequencies

5.4 Comparative Performance Assessment

A comparison of the results presented in Figures 7, 8 and 9 and Table 1 reveals that the highest performance in the reduction of sway of the boom crane is achieved with the ZVD (input shaping technique). This is observed and compared to the filters at the first three modes of sway. The performance of ZVD is evidenced in the magnitude of sway with the tangential pendulation and radial sway responses. For the tangential response, a sway reduction of 89% as compared to the bang-bang input is achieved using ZVD whereas LPF and BSF provide sway reductions of 70% and 64% respectively. Similarly, for the radial sway, a sway reduction of 87%, 57% and 49% are achieved using ZVD, LPF and BSF respectively. This is further evidenced in Figure 14 which demonstrated the level of sway reduction achieved using the control schemes at

the resonance modes. It is noted that a better performance in sway reduction of the system is achieved with the low-pass filtered input as compared to the band-stop filtered input. This is mainly due to the higher level of input energy reduction achieved with the low-pass filter, especially at the second and third modes. However, the band-stop filtered input gives higher reduction at the first mode of sway. As expected, system responses were slower with the shaped and filtered inputs as compared to the system response to the unshaped input. Comparisons of specifications of slew-angle responses in Figure 6 show that ZVD is able to maintain the level of overshoot as the unshaped input. However, higher overshoots as compared to the unshaped input were achieved with LPF and BSF.
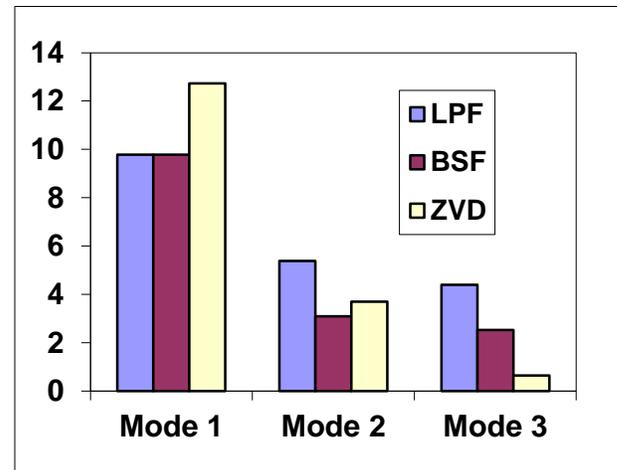
Figure 14 Level of sway reduction with exact natural frequencies using LPF, BSF and ZVD

A comparison of the results presented in Figures 11, 12 and 13 and Table 2 reveals that the highest robustness to uncertainty in natural frequencies is achieved with ZVD. It is noted that the ZVD can successfully handle errors in the natural frequency. In this case, only a slight different in sway reduction in both cases, with exact and erroneous frequency was noted. This is further revealed in Figure 15 that demonstrates the level of reduction of sway of the tangential pendulation response at the resonance modes with the control schemes. The input shaping technique is more robust, as significant reduction was achieved at the first mode of vibration, which is the most dominant mode. The band-stop filtered input did not handle the error as only small amount of reduction of the system vibration was achieved. On the other hand, using the low-pass filter, a significant amount of attenuation of the system sway was achieved at the second and third resonance modes. Moreover, the sway reduction achieved with low-pass filtered inputs was higher than that with the shaped input at these resonance modes.

■6.0 CONCLUSION

Investigations into sway reduction control of a boom crane using command shaping techniques with filtering and input shaping techniques have been presented. The dynamic model of the boom crane utilizing the Euler-Lagrange formulation has been considered. The system response to the unshaped bang-bang torque input has been used to investigate the dynamic behavior of the boom crane and to determine the parameters of the system for evaluation of the control strategies. Significant

M. H. I. Ishak, Z. Mohamed & R. Mamat / Jurnal Teknologi (Sciences & Engineering) 67:5 (2014), 49–58

reduction in the system vibrations has been achieved with these control strategies. Performances of the techniques have been evaluated in terms of level of sway reduction, time response specifications and robustness. For the boom crane and the specifications used in designing the input shapers and filters, the input shaping technique has been demonstrated to provide the best performance in sway reduction, especially in terms of robustness to errors. The low-pass filtered input has also been shown to perform better than the band-stop filtered input.
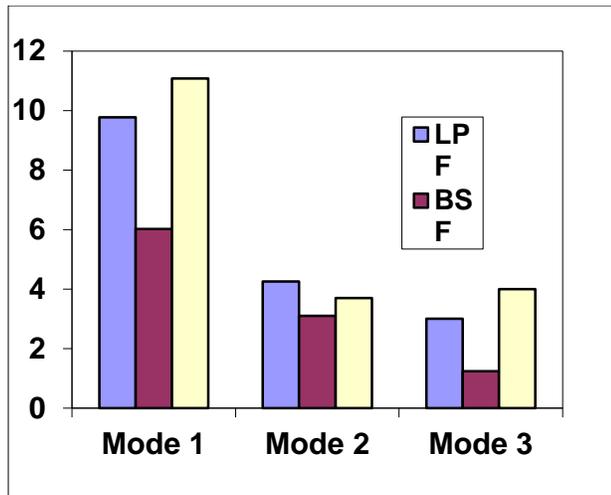
Figure 15 Level of sway reduction with erroneous natural frequencies using LPF, BSF and ZVD

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