Comparative Analysis of Input Shaping Techniques for Sway Control of Nonlinear Crane System

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Abstract – *This paper compares output based input shaping with a conventional command shaping in suppressing sway of a nonlinear crane system. The output-based input shaping filter is designed using the output signal of the target system while conventional input shaping filters are designed using the natural frequency and damping ratio of the system. Zero vibration (ZV), zero vibration derivative (ZVD), zero vibration derivative, derivatives (ZVDD) and zero vibration and triple derivative (ZVDDD) were designed and compared with the output based filter to investigate the performances and robustness of the filters. Level of sway reduction of the payload and time response analysis is used to assess the performance of the shapers. Simulation results showed that the output based filter has a better performance and it is more robust compared to the conventional input shapers and does not require model information of the system.*

Keywords: *Crane, command shaping, sways, Output based filter, Nonlinear*
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I. Introduction

Heavy items such as containers in construction sites and industries are usually loaded and unloaded using Crane system. To achieve an improved rate of production, it is necessary to maintain payload sway to the barest minimum, which will, in turn, require a very fast crane maneuvering technique [1]. Swinging, twisting and bouncing of the load are the most common unwanted crane motions which are generally encountered as a result of an important aspect of the crane maneuvering technique called payload hoisting. These negatively affect the overall efficiency, payload positioning as well as the operation safety [2]. Based on an investigation carried out on flexible structures, sway control strategies have been classified into feedback and feedforward.

The feedback control strategy is employed in estimating the states of the system and to minimize the oscillation effects on its load positioning. While the feedforward strategy is used in modifying the input signal for sway cancellation [3]. A combination of the two control strategies can therefore, offer a vibration-free system performance. The feedforward control helps in reducing the design complexity of the feedback control [4]. The most effective and well-proven feedforward control strategy is the input shaping technique. It is the commonly employed technique for residual vibration control of the flexible structures [5]. The input shaping technique has been developed as in [6]. The vibration reduction can be accomplished by convolving a sequence of impulses of the input signal.

Many input shaping techniques with certain modifications have now been designed and implemented on various flexible systems. Some of these techniques are; the vector diagram approach input shaping [7], negative input shaping [8], finite-state input shaping [9], multi-hump extra intensive input shaping [10], two-mode input shaping [11] and unit magnitude input shaping [12]. These techniques have been implemented on flexible systems such as non-linear tower crane [13] and flexible manipulator system [14].

In addition, various other input shaping techniques are proposed. Some of which are; input shaping using system output speed [15], continuous function input shaping [16] and a comparison between digital filtering and input shaping [17]. Since input shaping control strategy cannot track the position of payload, a feedback control strategy is needed, to have both oscillation and position control. Optimal control for gantry crane payload sway minimization [18], time-optimal control for an overhead crane with visual feedback control [19], a practical optimal controller for vibration control of under actuated gantry crane system [20] are some of the various optimal feedback control employed on flexible structures. Also, numerous intelligent control strategies have been employed for position and sway control of flexible structure. Some of which are; Genetic Algorithm (GA)-tuned PID with input shaping for vibration and tracking control of flexible structures [21], GA optimization of command shaper for vibration control [22], multi-objective GA optimization for vibration control [23], hybrid fuzzy logic with GA optimization [24], PID-type fuzzy logic tuning with PSO optimization for vibration control [25] and also a fuzzy PID control based on hybrid optimization [26].

Moreover, robust control algorithms have also been designed and implemented on various flexible structures. As for example; a robust control scheme for vibrations and position control of an overhead crane system [27], a wave-based robust control for vibration control of crane system [28], robust control of a nonlinear system with uncertainty [29] and hybrid robust control for sway control of the gantry crane system [30].

II. Model Description

3D crane system is an industrial machine which is normally used to transport loads from one place to another. This 3D crane commonly found used in construction industries, nuclear plant, warehouse, seaport, heavy machine installations, and etc. In this work, a laboratory scale 3D crane system was used as shown in Fig.1, and Fig. 2 shows the main components of the system hardware such as; cart, rail and a pendulum [31].

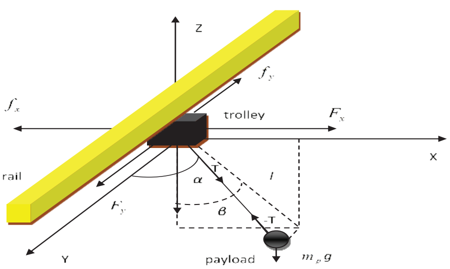

Fig. 1. Schematic diagram and forces

With *XYZ* as the coordinates of the system, α is the angle of lift-line with *Y*-axis and β is the angle between the negative part of *Z*-axis and the projection of the payload cable onto the *XZ* plane. T is a reaction force in the payload cable acting on the trolley, F_x and F_y are the forces driving the rail and trolley respectively. F_z is a force lifting the payload and *fx*, *fy* and *fz* are corresponding frictional forces. These are defined in [31] as;

$$\mu_1 = \frac{m_p}{m_t}, \mu_2 = \frac{m_p}{m_t + m_r}$$

$$u_1 = \frac{F_x}{m_t}, u_2 = \frac{F_y}{m_t + m_r}, u_3 = \frac{F_z}{m_p}$$

$$f_1 = \frac{f_x}{m_t}, f_2 = \frac{f_y}{m_t + m_r}, f_3 = \frac{f_z}{m_p}$$

$$K_1 = u_1 - f_1, K_2 = u_2 - f_2, K_3 = u_3 - f_3$$

In which, m_p, m_t and m_r are the payload mass, trolley mass and moving rail respectively. l is the length of the lift-line. The dynamic equations of motion of the crane can be obtained as in [31].

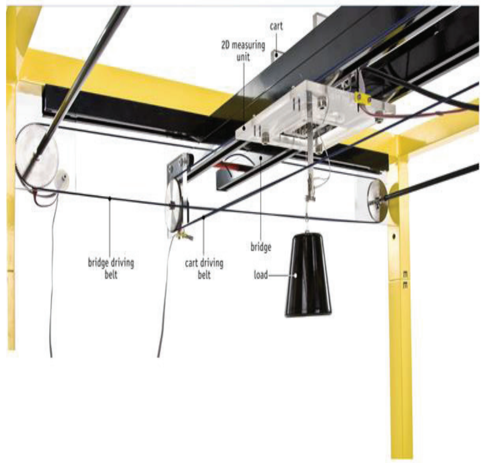

Fig. 2. Laboratory Scale 3D Crane System

$$\ddot{x}_t = K_2 + \mu_2 K_3 \sin\alpha \sin\beta \quad (1)$$

$$\ddot{y}_t = K_1 + \mu_1 K_3 \cos\alpha \quad (2)$$

$$\begin{aligned} \ddot{x}_p &= \ddot{x}_t + (\ddot{l} - l\dot{\alpha}^2 - l\dot{\beta}^2)\sin\alpha\sin\beta + \\ &2l\dot{\alpha}\dot{\beta}\cos\alpha\cos\beta + (2\dot{l}\dot{\alpha} + l\ddot{\alpha})\cos\alpha\sin\beta \\ &+(2\dot{l}\dot{\beta} + l\ddot{\beta})\sin\alpha\cos\beta \end{aligned} \quad (3)$$

$$\ddot{y}_p = \ddot{y}_t + (\ddot{l} - l\dot{\alpha}^2)\cos\alpha - (2\dot{l}\dot{\alpha} + l\ddot{\alpha})\sin\alpha \quad (4)$$

$$\begin{aligned} \ddot{z}_p &= (-\ddot{l} + l\dot{\alpha}^2 + l\dot{\beta}^2)\sin\alpha\cos\beta + \\ &2l\dot{\alpha}\dot{\beta}\cos\alpha\sin\beta - (2\dot{l}\dot{\alpha} + l\ddot{\alpha})\cos\alpha\cos\beta \\ &+(2\dot{l}\dot{\beta} + l\ddot{\beta})\sin\alpha\sin\beta \end{aligned} \quad (5)$$

Where, x_p, y_p and z_p are positions of payload in *X*, *Y* and *Z* axes respectively. x_t and y_t are positions of the trolley in *X* and *Y* axes respectively. The dots are the derivative of the respective quantities.

The parameters of the system are shown in Table I [31].

TABLE I
System Parameters

Variables	Values
Mass of payload, m_p	1 kg
Mass of trolley, m_f	1.155 kg
Mass of moving rail, m_r	2.2 kg
Cable length, l	0.72 m
Gravitational constant, g	9.8 m/s
Corresponding friction forces, f_x, f_y, f_z	100, 82, 75 V_s/m

III. Control Design

In this section, output based filter, *ZV*, *ZVD*, *ZVDD* and *ZVDDD* were designed. Time delay filters are designed using the natural frequency and damping ratio of the nonlinear crane system, which were obtained using the curve fitting tool box in MATLAB. The output-based filter was designed using the output of the system, which was measured in MATLAB Simulink.

A. Time delay input shaping

Time delay filters are obtained by convolving the reference input with a sequence of impulses. The amplitude and time instants of these impulses are obtained using the natural frequency and damping ratio of the system. An input shaping process is shown in Fig. 3, containing two impulses; ZV. To demonstrate the algorithms, a second order response of a crane system can be considered as under-damped system of the form [31];

$$G(s)=\frac{\omega^2}{s^2+2\zeta\omega s+\omega^2} \tag{6}$$

Where ω is natural frequency and ζ is the damping ratio of the system, it can also be in the time domain as in [31];

$$y(t)=\frac{A\omega}{\sqrt{(1-\zeta^2)}}e^{-\zeta\omega(t-t_0)}\sin\left(\omega(t-t_0)\sqrt{(1-\zeta^2)}\right) \tag{7}$$

Where A and t_0 are respectively the amplitude and time instant of the impulse. Using superposition, it can be obtained as;

$$y(t)=\sum_{i=1}^{n}\left[\frac{A_i\omega_n}{\sqrt{(1-\zeta^2)}}e^{-\zeta\omega_n(t-t_i)}\right]\sin\left(\omega(t-t_i)\sqrt{(1-\zeta^2)}\right) \tag{8}$$

The following trigonometric function can be used to find the amplitude of the residual vibrations as;

$$\sum_{i=1}^{n}B_i\sin(\omega t+\beta_i)=A\sin(\omega t+\varphi) \tag{9}$$

where

$$A=\sqrt{\left(\sum_{i=1}^{n}B_i\cos(\beta_i)\right)^2+\left(\sum_{i=1}^{n}B_i\sin(\beta_i)\right)^2} \tag{10}$$

Comparing the equation (8) and (9), we have;

$$B_i=\frac{A_i\omega_n}{\sqrt{(1-\zeta^2)}}e^{-\zeta\omega_n(t-t_i)} \tag{11}$$

To calculate the residual oscillation amplitude, equation (10) is evaluated at the last impulse; $t=t_n$. Substituting equation (11) into (10) and taking the constant part of the coefficients out of the square roots gives;

$$A=\frac{\omega}{\sqrt{(1-\zeta^2)}}e^{-\zeta\omega t_n}\sqrt{R_1{}^2+R_2{}^2} \tag{12}$$

where

$$R_1=\sum_{i=1}^{n}A_ie^{\zeta\omega t_i}\sin\left(\omega t_i\sqrt{(1-\zeta^2)}\right)$$

$$R_2=\sum_{i=1}^{n}A_ie^{\zeta\omega t_i}\cos\left(\omega t_i\sqrt{(1-\zeta^2)}\right)$$

The residual oscillation amplitude of a unity magnitude at t=0 can be obtained as in [32];

$$A_\uparrow=\frac{\omega}{\sqrt{(1-\zeta^2)}} \tag{13}$$

Hence, dividing equation (12) by (13), the percentage residual vibration can be obtained as;

$$R=\frac{A}{A_\uparrow}=e^{-\zeta\omega(t_n)}\sqrt{R_1^2+R_2^2} \tag{14}$$

R_1 and R_2 are set to zero to obtain *ZV* after the last impulse, this is *ZV* constraint. The sum of the shaper's amplitudes of the impulse should be unity. Constraints of summation are obtained as;

$$\sum_{i=1}^{n}A_i=1 \tag{15}$$

In addition, setting the time instants of the first impulse at $t_1=0$ using the *ZV* constraints yields the *ZV* parameters as;

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{1+k} & \frac{k}{1+k} \\ 0 & \tau_d \end{bmatrix} \qquad (16)$$

where; $\tau_d = \frac{\pi}{\omega\sqrt{(1-\zeta^2)}}$ and $k = e^{\frac{-\pi\zeta}{\sqrt{(1-\zeta^2)}}}$

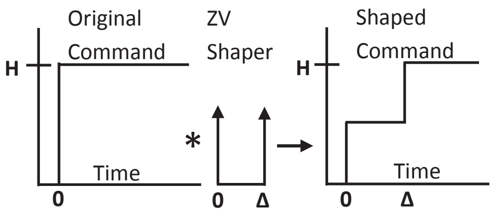

Fig. 3. Input shaping process

However, to increase the robustness to the frequency error, the derivatives of R_1 and R_2 are set to zero. This can be obtained in the form;

$$\frac{\partial^i R_1}{\partial \omega^i} = 0 \text{ and } \frac{\partial^i R_2}{\partial \omega^i} = 0 \qquad (17)$$

Solving the constraint equations of (14), (15) and (17) gives the three impulse *ZVD* shaper's parameters as;

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{(1+k)^2} & \frac{2k}{(1+k)^2} & \frac{k^2}{(1+k)^2} \\ 0 & \tau_d & 2\tau_d \end{bmatrix} \qquad (18)$$

In addition, further differentiation and solving the constraint equations of (13), (14) and (17) gives the four impulse *ZVDD* shaper's parameters as;

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{(1+k)^3} & \frac{3k}{(1+k)^3} & \frac{3k^2}{(1+k)^3} & \frac{k^3}{(1+k)^3} \\ 0 & \tau_d & 2\tau_d & 3\tau_d \end{bmatrix} \qquad (19)$$

Continuously using the same approach *ZVDDD* shaper's parameters can be obtained as [33];

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{(1+k)^4} & \frac{4k}{(1+k)^4} & \frac{6k^2}{(1+k)^4} & \frac{4k^3}{(1+k)^4} & \frac{k^4}{(1+k)^4} \\ 0 & \tau_d & 2\tau_d & 3\tau_d & 4\tau_d \end{bmatrix} \qquad (20)$$

where k and τ_d are the same as in equation (16).

B. Output based input shaping filter

In this technique, a critically damped system is designed as a reference system and it is designed based on the dynamic response of the system. This is used to measure the output of the system using simulation. The filter gains were obtained by writing a program code in MATLAB.

To explain the basic principle of this technique, a second order system shown in equation (21) is considered as in [34].

$$G(s) = \frac{Kw_n^2}{s^2 + 2\xi w_n s + w_n^2} \qquad (21)$$

where k_m, ζ_m and w_n are static gain, damping ratio and natural frequency respectively.

Let the reference system designs in a form:

$$M(s) = \frac{k_m w_m^2}{s^2 + 2\xi_m w_m s + w_m^2} \qquad (22)$$

The filter can be designed as:

$$F_o(s) = \frac{k_m w_m^2 s^2 + 2\xi w_n s + w_n^2}{Kw_n^2 s^2 + 2\xi_m w_m s + w_m^2} \qquad (23)$$

Then based on zero-pole cancellation, a product of *G(s)* and *F₀(s)* will give *M(s)*, therefore, adequate static gain, damping ratio and bandwidth can be achieved by choosing k_m, ζ_m, w_n respectively [34], [35].

Thus,

$$F(s) = \frac{s^2 a_2 + a_1 s + a_0}{s^2 + 2\xi_m w_m s + w_m^2} \qquad (24)$$

The aim is to obtain the filter gains (a_0, a_1, a_2) so that zeroes of *F(s)* will cancel the poles of *G(s)*, thus *F(s)* = $F_0(s)$ and poles of *G(s)* are identical [34],[35].

A critically damped system considered as a reference system can be realized as:

$$G_r(s) = \frac{w_c^2}{(s + w_c)^2} \qquad (25)$$

where w_c is the bandwidth of the system and is selected based on the time response of the system. This system has little or zero vibration, by selecting $w_c=2$ and using equation (25), the reference system is as;

$$G_r(s) = \frac{16}{s^4 + 8s^3 + 24s^2 + 32s + 16} \qquad (26)$$

Therefore, the filter gains are calculated in MATLAB using the following relation:

$$\begin{bmatrix} a_1 \\ a_2 \\ a_3 \\ a_4 \end{bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & S_{14} \\ S_{21} & S_{22} & S_{23} & S_{24} \\ S_{31} & S_{32} & S_{33} & S_{34} \\ S_{41} & S_{42} & S_{43} & S_{44} \end{bmatrix} \begin{bmatrix} S_{1r} \\ S_{2r} \\ S_{3r} \\ S_{4r} \end{bmatrix} \tag{27}$$

Hence, the filter gains are obtained as;

$a_1 = 50.4$, $a_2 = 8.1$, $a_3 = 2.5$ and $a_4 = 0.4$

Substituting in equation (24), the filter is obtained as;

$$F(s) = \frac{0.4s^4 + 2.5s^3 + 8.1s^2 + 50.4s}{s^4 + 8s^3 + 24s^2 + 32s + 16} \tag{28}$$

V. Results and Discussions

In this section, the results of the designed input shapers are discussed. To study the dynamic behavior of nonlinear crane system, bang-bang input torque was used as an input to the system, this causes the system to have positive and negative accelerations. The natural frequency and damping ratio of the system were determined using the curve fitting tool box in MATLAB. *ZV, ZVD, ZVDD*, and *ZVDDD* were designed and the parameters are presented in Table II. The performance of the filters are assessed using the level of sway reduction. This was achieved by comparing the mean absolute error (MAE) value of the unshaped and shaped responses, which are 0.1073 rad, 0.0194 rad, 0.0186 rad, 0.018 rad, 0.0168 rad and 0.007rad respectively. The sway and angular velocity are as shown in Fig. 4 and Fig. 5 respectively. In time delay filter, it can be observed that, *ZVDDD* has performed better in sway reduction, but has the slowest response time. However, to increase speed and robustness, the output based filter was designed and compared with the time delay filters as in Fig. 6 and Fig. 7. And it has shown superior performances as compared with time delay filters. Table III shows the time response specifications and sway reduction performances of the two types of filters. The performances of all filters were also compared using the MAE value as in Fig. 8.

TABLE II
SHAPERS' PARAMETERS

Shaper	ZV	ZVD	ZVDD	ZVDDD
A1 (rad)	0.5091	0.2592	0.1320	0.0672
A2 (rad)	0.4909	0.4998	0.3817	0.2591
A3 (rad)	-	0.2409	0.3680	0.3747
A4 (rad)	-	-	0.1183	0.2409
A5 (rad)	-	-	-	0.0581
t_1 (sec)	0	0	0	0
t_2 (sec)	0.6875	0.6875	0.6875	0.6875
t3 (sec)	-	1.375	1.375	1.375
t4 (sec)	-	-	2.0625	2.0625
t5 (sec)	-	-	-	2.750

TABLE III
SHAPERS' PARAMETERS

Shaper	Max. Overshoot (rad)	Settling Time (sec)	Sway reduction (%)
ZV	-0.1042	1.732	81.22
ZVD	-0.0861	2.325	83.06
ZVDD	-0.0620	3.068	83.26
ZVDDD	-0.0615	4.202	84.54
OBF	0.005	0.825	93.57

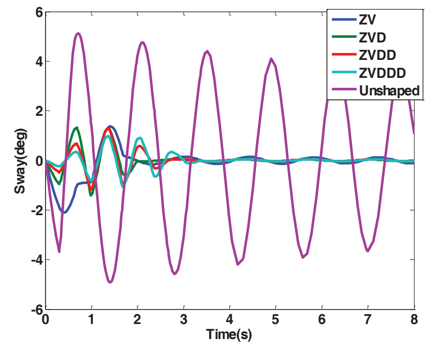

Fig. 4. Comparing the sway reduction of time delay filters

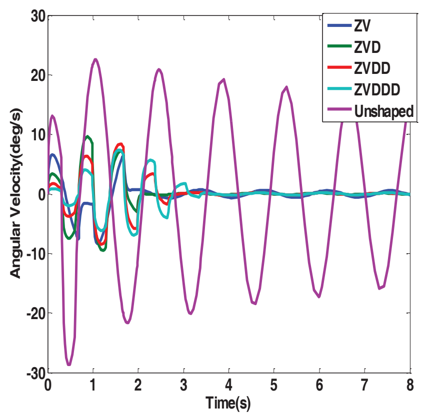

Fig. 5. Comparing the performance of time delay filters

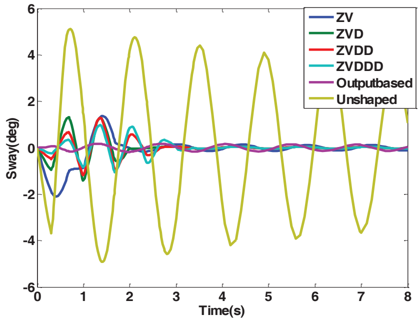

Fig. 6. Comparing the sway reduction of the two types of the filters

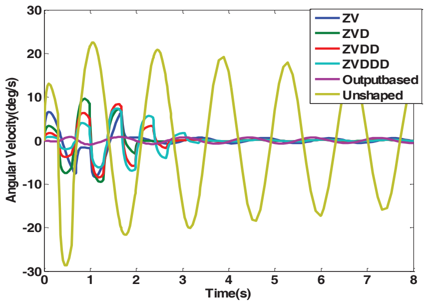

Fig. 7. Comparing the performance of time delay and output based filters

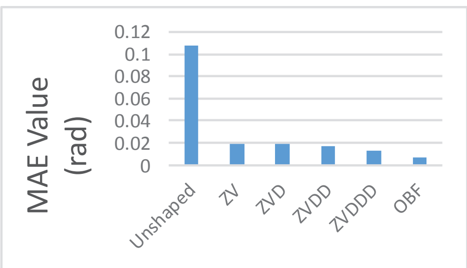

Fig. 8. Comparing the mean absolute error values of all filters

VI. Conclusion

Output based filter and conventional input shapers have been presented and their performances were compared for sway control of nonlinear crane system. Time response analysis and mean absolute error were used as the performance indexes. The simulation results were studied and analyzed. In time delay filters, the more the derivatives the better the sway reduction but the more the delay in the system. However, output based filter shows superior performances, faster and more robust than the time delay filters.

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