Anti-sway control of marine cranes under the disturbance of a parallel manipulator

Ayman A. El-Badawy
Mechanical Engineering Department, Al-Azhar University
and Mechatronics Engineering Department
German University in Cairo
Cairo, Egypt
ayman.elbadawy@guc.edu.eg

Mina Maged George Shehata
Mechatronics Engineering Department
German University in Cairo
Cairo, Egypt
mina.m.george@gmail.com

***Abstract*— Cranes are used to move loads from one location to another in minimum time such that the load reaches its destination without swinging. This swinging problem gets aggravated in the presence of external disturbances such as in the case of marine cranes. This paper investigates the modeling of a tower crane model and different types of control schemes for both anti-swaying and input tracking control for the tower crane model under continuous external disturbances. The input shaping technique was applied for a tower crane model under continuous external disturbance. Then a closed loop PID input shaper was implemented instead of the open loop input shaper to compensate for the external disturbances. Also the response of the closed loop PID input shaper was compared to a controller based on inverse dynamics. The environmental disturbances such as sea waves will be simulated by the movement of a Stewart platform.**

Keywords*— *Stewart platform, parallel manipulator, tower crane, gantry crane, input shaping, anti-sway, inverse dynamics

I. INTRODUCTION

Cranes are used to move a load from point to point in minimum time such that the load reaches its destination without swinging. Usually a skillful operator is responsible for this task. During the operation, the load is free to swing in a pendulum- like motion. If the swing exceeds a proper limit, it must be damped or the operation must be stopped until the oscillations die out. It is practically impossible to completely remove payload oscillations in all possible circumstances. External disturbances such as wind can easily initiate an oscillation. After a while, the oscillations can become larger, until the point is reached where the crane operations have to be stopped and the oscillation has to die out, which can take a very long time. Even without external disturbances, oscillations due to the movement of the tower crane are still hard to suppress.

This swinging problem gets aggravated in the presence of external disturbances such as in the case of marine cranes. According to the United States military logistics over shore department, marine cranes operations should be performed in sea state 0 through sea state 3. Conditions above sea state 3 occurs 15% of the times [1][2][3]. Researchers have therefore tried to find different ways to control the movement of cranes in order to prevent and reduce the payload swinging.

Input Shaping is an open loop controller that drives the system to cancel out its own oscillations. Input shaping is implemented in real time by convolving the command signal with an impulse sequence. Neil C. Singer and Warren P. Seering showed that input shaping could be robust to uncertainties in the system parameters using different configurations of input shapers [4]. They demonstrated that the 4- pulses (ZVDD) input shaper is the most robust and most efficient input shaper algorithm of the input shapers algorithms mentioned previously. D. Blackburn, J. Lawrence, J. Danielson, W. Singhose, T. Kamoi and A. Taura also proved later on that the 4- pulses input shaper is better than the radial-motion assisted command shapers for nonlinear tower crane rotational slewing [5]. In their paper, they tried including the radial motion of trolley while rotating the jib of the tower crane in order to improve the efficiency of the input shaper and to reduce the sway angle. Other papers discussed adding PID controllers for input tracking with the input shapers [6][7]. Although all the previously mentioned efforts, the resulting open- loop controller is still sensitive to external disturbances and to parameter variations. Although all the previously mentioned efforts, the resulting open- loop controller is still unresponsive to external disturbances and to parameter variations.

On the other hand, feedback control which is well known to be less sensitive to disturbances and parameter variations is also adopted for controlling the gantry crane system [8]. Khalid L. Sorensen, William Singhose and Stephen Dickerson have presented a feedback PID control in addition to the input shaper to control both the position and the sway angle [9]. They applied their controller on a simple one dimensional gantry crane. Their control algorithms included a closed loop PID input shaper. In order to test their external disturbances rejection module, they applied a single, non- continuous, impulse disturbance to the payload of the gantry crane. Some researchers such as Hanafy M. Omar proposed using delayed feedback controllers for both the position and the anti- swing controls [10]. Also Hanafy M. Omar has proposed a gain scheduling feedback controller [11]. Other researchers compared the open loop input shaper to the inverse dynamics on a tower crane model [12].

Most of the previously mentioned work was done on a one dimensional, one degree of freedom gantry crane. Also most of

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the previously mentioned work did not include continuous external disturbance, but rather a single impulse external disturbance. This work studies the case of a two degrees of freedom tower crane model under continuous external disturbance. To simulate this case, the tower crane model was mounted on top of a Stewart platform. The movement of the Stewart platform will simulate the external disturbances such as the sea waves. Stewart platform is one of the most popular parallel manipulators. It has six degrees of freedom positioning system that consists of two bodies connected by six legs. The Stewart platform was originally proposed and presented in 1965 as a flight simulator by Stewart [13].

This paper presents the development of control algorithms for both the input tracking and anti- swaying of a tower crane model system under continuous external disturbances. The controller consists of a feedback PID control in addition to a closed loop input shaper to control both the position and the sway angle. For the tower crane model, the sway angle is composed of a radial direction swing angle and a tangential direction swing angle. Then a controller based on the inverse dynamics of the tower crane model is put into comparison with the closed loop PID input shaper. Numerical simulations were compared with the experimental setup. The performance of the control schemes is examined in terms of sway angle reduction.

II. DYNAMIC MODELING

To derive the equations of motion of the tower crane, the coordinate system has to be clearly defined. The cartesian coordinate system (xyz) is centered at a reference point that lies in the plane of the jib at the center of the crane tower, with its positive z- axis being along the tower upward axis (Fig. 1).

The x- axis and the y- axis are in the plane of the jib, with the x- axis being along the jib. The xyz coordinate system is attached to the rotating jib. The jib rotates and traces an angle $\gamma(t)$. The trolley moves on the jib with its position $r(t)$ being the distance measured from the origin of the xyz coordinate system. The angle $\gamma(t)$ and the radial distance $r(t)$ are the inputs to the system. They are used to control the system behavior.

A. Modeling of Tower crane

The equations of motion describing the tower crane model is given by [11][12]

$$M\ddot{x} + m\ddot{x} + ml\varphi\dot{\varphi}^2 - ml\ddot{\varphi} = F_x \tag{1}$$

$$\ddot{\varphi} = -\frac{g}{L}\varphi + \frac{1}{L}\ddot{x} \tag{2}$$

$$(J_o + (M+m)x^2)\ddot{\gamma} + mlx(-\theta\dot{\theta}^2 + \ddot{\theta}) = T_\gamma \tag{3}$$

$$\ddot{\theta} = -\frac{g}{L}\theta - \frac{x}{L}\ddot{\gamma} \tag{4}$$

As shown from the equations, the system inputs are F_x and T_γ. Reformulating the equations in terms of motors input voltages, the two motors are modeled as constant gains [11]

$$F_x = K_{mx}V_x \tag{5}$$

$$T_\gamma = K_{m\gamma}V_\gamma \tag{6}$$

where

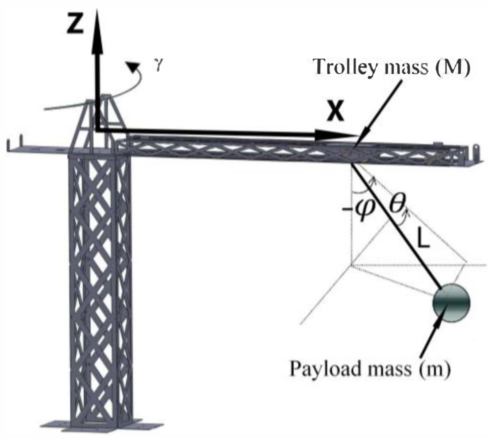

Figure 1. Tower crane model

K_{mx} is the tower crane's trolley motor constant gain, V_x is the input voltage for the tower crane's cart motor, $K_{m\gamma}$ is the tower crane's jib motor constant gain and V_γ is the input voltage for the tower crane's jib motor.

The DC motors transfer functions relating the displacement of the trolley with the input voltage V_x and the jib's rotation with the input voltage V_γ[14].

$$\frac{x(s)}{V_x(s)} = \frac{K_{tx}K_{trx}}{s[(L_{indx}s+R_{coilx})(J_{mx}s+B_{mx})+K_{bx}K_{tx}]} \tag{7}$$

$$\frac{\gamma(s)}{V_\gamma(s)} = \frac{K_{t\gamma}K_{tr\gamma}}{s[(L_{ind\gamma}s+R_{coil\gamma})(J_{m\gamma}s+B_{m\gamma})+K_{b\gamma}K_{t\gamma}]} \tag{8}$$

where

K_t is the motor's torque constant gain, K_{tr} is the transmission ratio constant gain relating the motors rotations to the trolley displacement and the jib rotation, L_{ind} is the inductance of the motor, R_{coil} is the resistance of the motor, J_m is the load inertia including both the motor's rotor inertia and the load inertia, B_m is the damping factor and K_b is the back emf constant gain.

The block diagram of the system with no swing control is shown in Fig 2. There is a feedback signal for the positioning module while there is no feedback on the swing angle controller.

In this work, the tower crane model is under continuous disturbance, similar to the case of marine cranes. To model such situation, the tower crane was mounted on top of a Stewart platform. The movement of the Stewart platform simulates the external disturbances (Fig. 3).

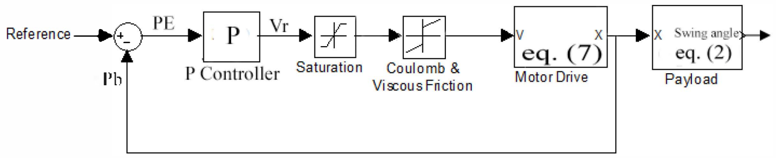

Figure 2. No swing control block diagram

B. Stewart platform

The Stewart platform is shown in Fig. 4, the lower body is called the base and the upper is called the platform [15]. Each of the six legs has one of its end points fixed in the base and the other end point fixed in the platform. These legs are linear actuators that can vary in length and each leg is considered to be composed of two different bodies connected by a prismatic joint.

For a given input disturbance to the tower crane model, the motion of the upper platform of the Stewart platform has to be known in terms of the three translational positions(x_{sp}, y_{sp}, z_{sp}) and the three rotational angles (θ_{xsp}, θ_{ysp}, θ_{zsp}). The objective of the Stewart platform's inverse kinematics is to determine the length of each individual leg to achieve the required position and orientation. It is also important to notice that the Stewart platform does not have independent drive systems for each degree of freedom, but rather achieves motion in all degrees of freedom by a combination of all actuator extensions.

Since the Stewart platform consists of two bodies, it is convenient to define two coordinate systems, $[X_b, Y_b, Z_b]$ for the lower base and $[X_p, Y_p, Z_p]$ for the upper plate (Fig. 4) [16]. The first coordinate system will be the global reference frame that is attached to the centroid of the base since the base is always grounded and do not move. The second coordinate system is attached to the centroid of the moving platform.

A transformation matrix will be used to transform vectors from platform coordinates to base coordinates.

Figure 3. Tower crane mounted on a Stewart Platform

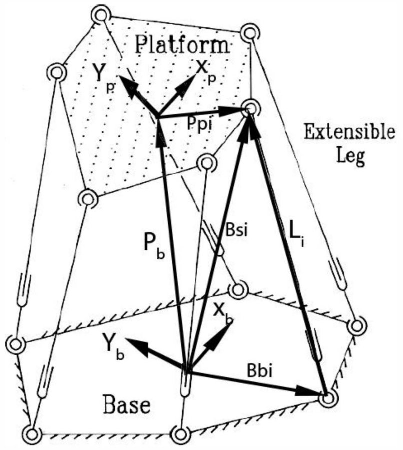

Figure 4. Stewart Platform [15]

The transformation Matrix=

$$R = R_{zyx} = \mathrm{R}_{\mathrm{p}}^{\mathrm{b}} = \mathrm{R}_{\mathrm{z}}(\theta_{\mathrm{zsp}}) * \mathrm{R}_{\mathrm{y}}(\theta_{\mathrm{ysp}}) * \mathrm{R}_{\mathrm{x}}(\theta_{\mathrm{xsp}}) \quad (9)$$

where

$\mathrm{R}_{\mathrm{z}}(\theta_{\mathrm{zsp}}), \mathrm{R}_{\mathrm{y}}(\theta_{\mathrm{ysp}}), \mathrm{R}_{\mathrm{x}}(\theta_{\mathrm{xsp}})$ are the rotational matrices around the z-axis, y-axis and z-axis respectively.

$$\mathrm{P}_{\mathrm{b}} = [\mathrm{x}, \mathrm{y}, \mathrm{z} + \mathrm{offset}] \quad (10)$$

$$\mathrm{Bsi}_{\mathrm{b}} = \mathrm{P}_{\mathrm{b}} + (\mathrm{R}_{\mathrm{p}}^{\mathrm{b}} * \mathrm{Ppi}_{\mathrm{p}})_{\mathrm{b}} \quad (11)$$

$$\mathrm{Bsi}_{\mathrm{b}} = \mathrm{Bbi}_{\mathrm{b}} + \mathrm{Li}_{\mathrm{b}} \quad (12)$$

$$\mathrm{Li}_{\mathrm{b}} = \mathrm{Bsi}_{\mathrm{b}} - \mathrm{Bbi}_{\mathrm{b}} \quad (13)$$

$$\mathrm{Li}_{\mathrm{b}} = \mathrm{P}_{\mathrm{b}} + (\mathrm{R}_{\mathrm{p}}^{\mathrm{b}} * \mathrm{Ppi}_{\mathrm{p}})_{\mathrm{b}} - \mathrm{Bbi}_{\mathrm{b}} \quad (14)$$

The required length of each leg is equal to the magnitude of the vector Li_{b}, where i=1,…,6

In order to determine the effect of the movement of the Stewart platform on the payload of the tower crane, it is required to define a transformation matrix between the Stewart platform's upper plate and the trolley position.

Defining $\vec{r}_{T/P}$ which is the position of the tower crane's trolley in the x- direction, y- direction and z- direction respectively with respect to the coordinate system $[X_p, Y_p, Z_p]$. The new position of the trolley after the movement of the Stewart platform is given by

$$\vec{r}_{T/P2} = \begin{bmatrix} \mathrm{R}_{\mathrm{zyx}} & \mathrm{St}_{\mathrm{tr}} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \vec{r}_{T/P} \\ 1 \end{bmatrix} = [\, \mathrm{x}_2, \mathrm{y}_2, \mathrm{z}_2, 1] \quad (15)$$

$$\mathrm{St}_{\mathrm{tr}} = [\mathrm{x}_{\mathrm{sp}}, \mathrm{y}_{\mathrm{sp}}, \mathrm{z}_{\mathrm{sp}}] \quad (16)$$

where,

$\mathrm{St}_{\mathrm{tr}}$ is a vector containing the translational motions of the Stewart platform.

Numerically integrating (1) to (4), the radial swing angle is calculated by substituting the variable x by x_2 in (1) and (2), and the tangential swing is calculated by substituting γ by $(\gamma + \theta_{\mathrm{zsp}})$ in (3) and (4).

Figs. (5) and (6) show the simulations of the system when given an input of 25 cm displacement for the trolley and 20 degrees for the rotation of the jib under continuous external disturbance. The external disturbance was caused by the movement of the Stewart platform. The Stewart platform was actuated to rotate sinusoidally around both the x- axis and the z- axis with an amplitude of 4 degrees and 8 degrees respectively and with a frequency of 3 Hz. The value of the oscillations is higher compared to the previous simulation with no external disturbance. The value of the oscillations reached 0.59 radians in the radial direction and a value of 0.4 radians in the tangential direction.

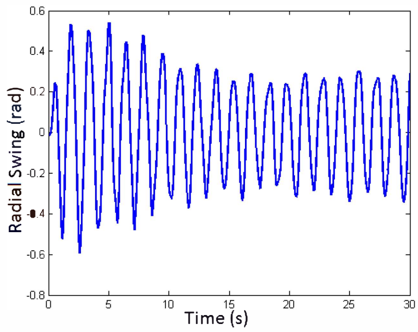

Figure 5. Simulation response of the radial swing angle with no swing control under continuous external disturbance

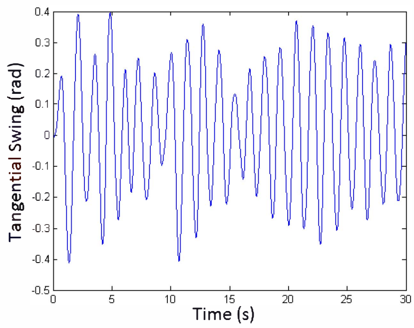

Figure 6. Simulation response of the tangential swing angle with no swing control under continuous external disturbance

III. CONTROL SCHEMES

A. Open loop input shaper

The input shaping is an open loop control method that is going to be used to guarantee free vibration of cargo of the tower crane. It is a successful approach to suppressing oscillations by generating a reference command that drives the system to cancel out its own oscillations. Input shaping is implemented by convolving a sequence of impulses, known as input shaper, with a reference signal [4][17]. The shaped command is then used to drive the system.

The amplitudes and time locations of the impulses comprising the input shaper are determined by solving a set of constraint equations that attempt to limit the unwanted system dynamics. In this work, the ZVDD input shaper was used because it is more effective than the other input shaper techniques and it is more robust for the system parameters variations up to $\pm 40\%$. The ZVDD input shaper is defined by four impulses, where their sum must be equal to a unity magnitude. The four impulses and their corresponding time are defined by

$$t_1 = 0, t_2 = 0.5T_D, t_3 = T_D, t_4 = 1.5T_D \qquad (17)$$

$$A_1 = \frac{1}{D}, A_2 = \frac{3K}{D}, A_3 = \frac{3K^2}{D}, A_4 = \frac{K^3}{D}$$

where

$$D = 1 + 3K + 3K^2 + K^3, K = e^{\frac{-\xi\pi}{\sqrt{1-\xi^2}}}$$

$$T_D = \frac{2\pi}{\omega_d} = \frac{2\pi}{\omega_o\sqrt{1-\xi^2}}, \omega_o = \sqrt{\frac{g}{L}}$$

The block diagram of the open loop input shaper controller is shown in Fig. 7. The open loop input shaper does not incorporate a feedback for the swing angles, which makes the controller unresponsive to external disturbances. The open loop input shaper was tested in the presence of external disturbances from the Stewart platform. The Stewart platform was actuated to rotate sinusoidally around both the x- axis and the z- axis with an amplitude of 4 degrees and 8 degrees respectively and with a frequency of 3 Hz. Figs. (8) and (9) show the simulations of the payload with the open loop input shaper when given the same inputs of 25cm and 20 degrees but under continuous external disturbance from the Stewart platform. It is clearly shown that the controller was no longer able to suppress the oscillations. The value of the oscillations reached 0.32 radians in the radial direction and a value of 0.35 radians in the tangential direction.

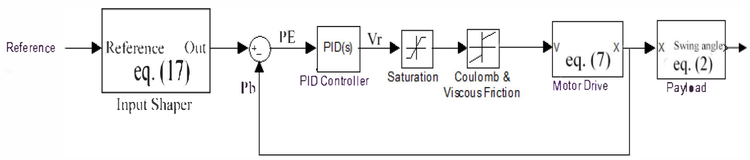

Figure 7. Open loop input shaper block diagram

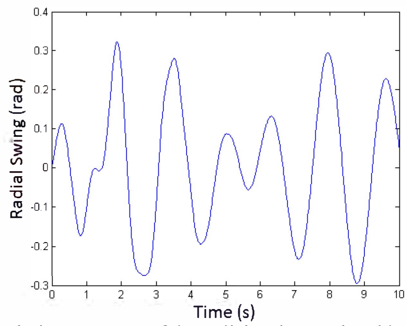

Figure 8. Simulation response of the radial swing angle with open loop input shaping under continuous external disturbance.

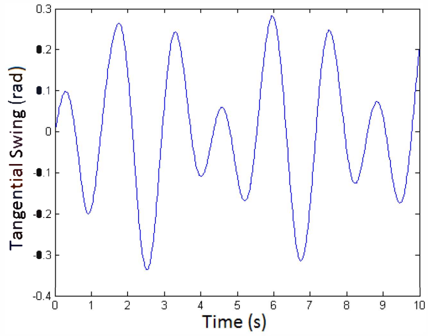

Figure 9. Simulation response of the tangential swing angle with open loop input shaping under continuous external disturbance.

B. Closed loop PID input shaper

In order for the input shaper controller to be responsive to external disturbances, a closed loop PID input shaper controller was implemented. The closed loop PID input shaper block diagram is shown in Fig. 10. The control module was designed to eliminate the cable sway caused by external disturbances acting on the payload due to the movement of the Stewart platform. The closed loop input shaper was tested in the presence of external disturbances from the Stewart platform. The Stewart platform was actuated to rotate sinusoidally around both the x- axis and the z- axis with an amplitude of 4 degrees and 8 degrees respectively and with a frequency of 3 Hz. Figs. (11) and (12) show the experimental response of the payload with the open loop input shaper when given the same inputs of 25cm and 20 degrees but under continuous external disturbance from the Stewart platform. The value of the oscillations reached 0.18 radians in the radial direction and a value of 0.14 radians in the tangential direction.

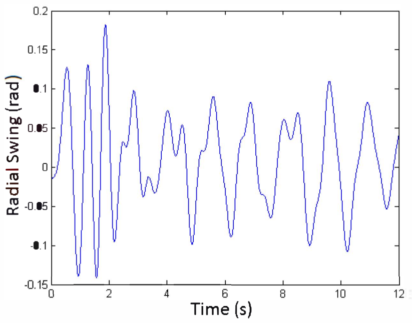

Figure 11. Radial swing with closed loop input shaping under continuous external disturbance

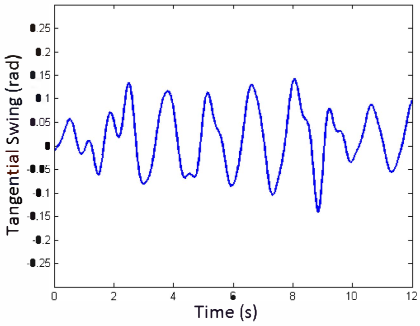

Figure 12. Tangential swing with closed loop input shaping under continuous external disturbance

Although the closed loop PID input shaper showed good results in reducing the oscillations, better results could be achieved if the system dynamics are taken into consideration. The next section studies and compares the results of the closed loop input shaper with the inverse dynamics controller. The inverse dynamics controller takes into consideration all the system parameters, including the parameters that were neglected in the input shaping controller which are the mass of the payload, the mass of the trolley and the inertia of the jib.

C. Inverse Dynamics

Any dynamics model could be rewritten in the matrix form

$$M_i(q)\ddot{q} + C_i(q,\dot{q})\dot{q} + G_i(q) = F_i \tag{18}$$

where

$M(q)$ is the mass matrix, $C(q,\dot{q})$ denotes a damping and centrifugal matrix, $G(q)$ is a matrix of gravity and spring forces and F indicates a matrix of control forces. The subscript, i, is used to differentiate between the system of the trolley displacement and the system of the jib rotation.

The nonlinear control input signal is given by [12][18]

$$U_i = -\bar{M}_i(q)(K_{di}\dot{q}_{i1} + (B_{ai} + K_{di})\dot{q}_{i2} + K_{pi}(q_{i1} - q_{i1}^d) + K_{pi}q_{i2} + C_{ai}) \tag{19}$$

The block diagram of the above mentioned equation is shown in Fig. 13. In order to compare all the control schemes, the inverse dynamics controller was tested in the simulation environment under the same condition as before by giving it an input of 25 cm displacement for the trolley and 20 degrees for the rotation of the jib. Figs. (14) and (15) show the response of the payload using the inverse dynamics algorithm. The figures show small oscillations values in both the radial and tangential

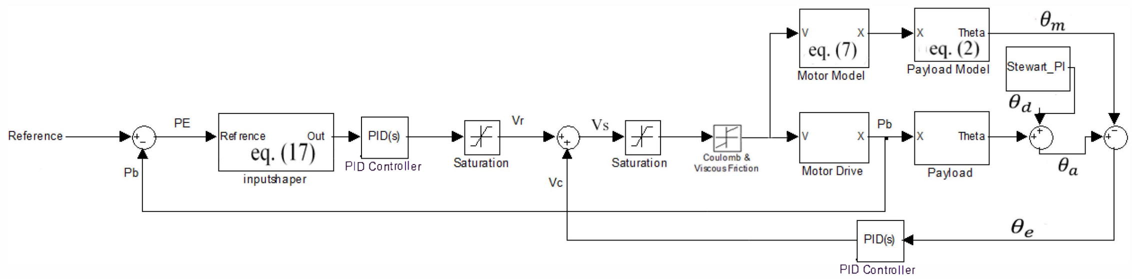

Figure 10. Closed loop PID input shaper

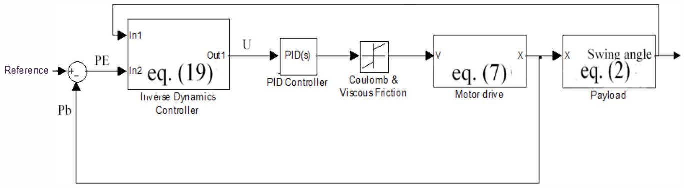

Figure 13. Radial swing with input shaping

directions compared to the closed loop input shaper simulations. The value of the oscillations reached 0.12 radians in the radial direction and a value of 0.13 radians in the tangential direction.

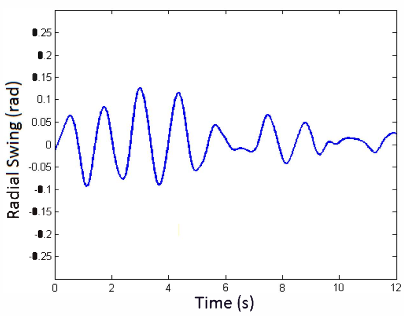

Figure 14. Simulation response of the radial swing angle with inverse dynamics control under continuous external disturbance

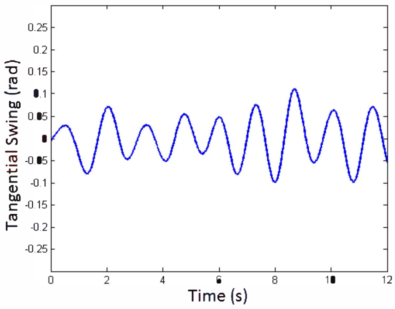

Figure 15. Simulation response of the tangential swing angle with inverse dynamics control under continuous external disturbance

IV. CONCLUSION

The implementation and results of both the closed loop PID input shaper and inverse dynamics have been presented for a marine crane model. The control schemes have been compared in terms of sway angle reduction. The open loop input shaper has been effective in suppressing the oscillations, but it failed in suppressing the oscillations when there are any external disturbances. The closed loop PID input shaper was effective in reducing the oscillations in the presence of external disturbances. An inverse dynamics controller was developed while taking into consideration all of the system dynamics, including the parameters that were neglected in the input shaping controller which are the mass of the payload, the mass of the trolley and the inertia of the jib. The inverse dynamics controller showed better results and a higher oscillation reduction than the closed loop PID input shaper.

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