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Decoupling and Tracking Control for Offshore Crane System Effect by Unknown Roll/Heave Wave Motions Disturbances

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Abstract— **This paper introduces an output feedback control method for an offshore crane system (OCS) with unknown wave disturbance caused by roll-heave motions. The method combines a backstepping controller and a high-gain observer to achieve the desired tracking trajectories of the cart position and the rope length. The dynamic model of the offshore crane system is initially formulated to account for unknown dynamic friction, dynamic coupling, and external disturbances while ensuring dynamic decoupling of the cart position and hoisting dynamics. The backstepping controller is proposed to stabilize the system and achieve trajectory tracking, where the extended high gain observer is used to estimate the dynamic states and external disturbances. The effectiveness of the proposed control approach is verified through simulations for both the system with disturbances generated by roll-heave motions. The simulation results demonstrate that the proposed approach is capable of achieving the desired trajectories, even under conditions of unknown nonlinearities and wave motion disturbances.**

I. INTRODUCTION

Offshore crane systems are critical equipment for the offshore oil and gas industry, as they are used for lifting heavy equipment and supplies onto and off offshore platforms and vessels. These systems are subject to demanding conditions, including high winds, waves, and corrosive saltwater environments [1]. In recent years, there have been significant advances in offshore crane technology, including developing new safety features, remote operation, automation, digitization, and energy efficiency [2]. These advancements have helped improve offshore crane systems' safety, efficiency, and sustainability. One key area of focus for offshore crane systems is safety. Crane operators must comply with strict regulations and standards to ensure the safe operation of cranes in offshore environments. Manufacturers have developed new features such as collision avoidance, load monitoring, and remote monitoring capabilities to improve safety.

The automatic control of offshore cranes has been a subject of significant research effort in recent years, yet it remains a challenging area due to the complex system dynamics and disturbances, resulting in a relatively limited number of published results [2]. Only a few control methods have been proposed to date; see, for example [3]–[9].

An online trajectory planning approach is designed to address boom luffing/rotating and rope hoisting motions in a 3-dimensional offshore boom crane system with superior anti-swing capabilities [6]. The control problem of a 5-DOF offshore crane system operating in a 3D space under persistent ship yaw and roll perturbations were addressed in [8], where an output feedback regulation control approach was proposed to achieve asymptotic stability results through rigorous theoretical analysis [8]. In [7], an optimal learning sliding mode controller is proposed for a dual ship-mounted crane system using a policy iterative algorithm to solve the Hamilton-Jacobi-Bellman equation and a critic neural network to approximate unknown functions. In [9], an adaptive neural-network-based tracking control method for offshore ship-to-ship crane systems that aims to achieve accurate positioning, swing suppression, and disturbance rejection while handling uncertain trajectories resulting from the target ship's movement. In [10], a control method that combines an adaptive controller with an output feedback controller is proposed for a shipboard rotary crane system. The adaptive controller achieves precise boom positioning and overcomes the unfavorable effects of system parameter uncertainties, while the output feedback is designed to avoid using velocity signals. In [11], a nonlinear energy-based controller is designed for dual ship-mounted crane systems to ensure the stability of the desired equilibrium point and enable the precise positioning of the crane boom and suppression of payload swing. In [12], a tracking controller is presented for boom crane and tower crane systems with multiple inputs and outputs (MIMO) that are subject to uncertainty. The goal of the proposed controller is to achieve precise positioning and tracking control while minimizing control efforts, even with saturated inputs.

A. Main contribution

Most controllers for offshore crane systems rely on a combination of multiple techniques or require knowledge of the dynamic model to achieve their control objectives. In this work, we propose a solution for trajectory tracking of the offshore crane system with unknown dynamics due to joint coupling and nonlinearities and subjected to external disturbances from roll-heave wave motion. The proposed control approach comprises three main components. First, we

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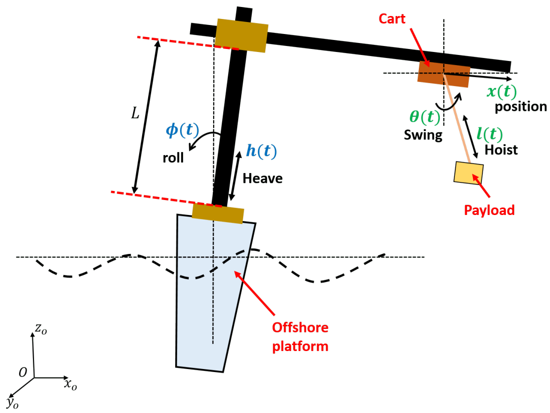


Fig. 1. Schematic representation of the offshore crane system.

formulate the dynamic model of the offshore crane system to ensure decoupling in the dynamics, considering totally unknown dynamics. Second, we design a state feedback control based on the backstepping method to achieve the global stability of the system and desired trajectories. Finally, we introduce an extended high-gain observer (EHGO) to estimate the system's states, unknown dynamic couplings, and external dynamics. The EHGO is combined with the state feedback controller to represent output feedback control.

II. DYNAMIC MODELING AND PROBLEM FORMULATION

This study examines the offshore crane system illustrated in Figure 1. The system consists of a tower crane mounted on an offshore platform, including a stationary vertical tower, a cart that moves horizontally, a wire, and a payload. The system has five degrees of freedom, with three related to the motion of the crane and two related to the movement of the platform due to wave motion. The crane's dynamics are presented in terms of the horizontal motion of the cart in the x-axis (x), the hoist motion of the wire (l), and the swing motion of the payload (θ). In contrast, the dynamic model of the offshore platform due to wave motion is described by the roll-heave movement (ϕ and h) generated by the wave motion.

A. Dynamic Model

The dynamic model of the OCS is formulated by utilizing Lagrangian mechanics [5], [9]. Thus, the dynamic equations that describe the 3-DOF motions of the crane can be expressed as

$$(M_1+M_2)\begin{bmatrix}\ddot{x}\\ \ddot{l}\\ \ddot{\theta}\end{bmatrix}+C\begin{bmatrix}\dot{x}\\ \dot{l}\\ \dot{\theta}\end{bmatrix}+G=\begin{bmatrix}F_x\\ F_l\\ 0\end{bmatrix}-\begin{bmatrix}f_{fx}\\ f_{fl}\\ f_{f\theta}\end{bmatrix}+\begin{bmatrix}d_{wx}\\ d_{wl}\\ d_{w\theta}\end{bmatrix} \quad (1)$$

where M_1 is the linear term of the system's inertial matrix, M_2 is the nonlinear term of the system's inertial matrix, C is the Coriolis–centripetal matrix, and G is the gravity matrix. These matrices are given by

$$M_1=\begin{bmatrix}m_c+m_p & 0 & 0\\ 0 & m_p & 0\\ 0 & 0 & 0\end{bmatrix},\quad G=\begin{bmatrix}0\\ -m_pg\cos(\theta-\phi)\\ m_pgl\sin(\theta-\phi)\end{bmatrix},$$

$$M_2=\begin{bmatrix}0 & m_p\sin(\theta-\phi) & m_pl\cos(\theta-\phi)\\ m_p\sin(\theta-\phi) & 0 & 0\\ m_pl\cos(\theta-\phi) & 0 & m_pl^2\end{bmatrix},$$

$$C=\begin{bmatrix}0 & m_p\cos(\theta-\phi)\dot{\theta} & m_p\cos(\theta-\phi)\dot{l}-m_pl\sin(\theta-\phi)\dot{\theta}\\ 0 & 0 & -m_pl\dot{\theta}\\ 0 & m_pl\dot{\theta} & m_pl\dot{l}\end{bmatrix},$$

The states of the system are x, which represents the horizontal motion of the cart, l, which represents the rope length, and θ, which represents the swing angle of the payload. The cart is driven by a controlled force F_x, and the rope is driven by a controlled force F_l.

The dynamic system is affected by the following friction forces: The cart-girder friction f_{rx}, the rope friction f_{rl}, and the swing angle friction force $f_{r\theta}$. According to [9], [13], the friction forces are approximated as

$$f_{fx}=a_x\tanh(\dot{x}/b_x)-c_x|\dot{x}|\dot{x}+d_x\dot{x}, \quad (2)$$

$$f_{fl}=a_l\tanh(\dot{l}/b_l)-c_l|\dot{l}|\dot{l}+d_l\dot{l}, \quad (3)$$

$$f_{r\theta}=d_\theta l\dot{\theta}, \quad (4)$$

where a_x, $a_l, b_x, b_l, c_x, c_l, d_x, d_l$, and d_θ are friction coefficients. The bounded external disturbances d_{wx}, d_{wl}, and $d_{w\theta}$ denote the dynamic effect on the crane system through the roll-heave movement, and they are given as [5]

$$d_{wx}=m_c\left(L\ddot{\phi}+x\dot{\phi}^2\right)+m_p\Big(L\ddot{\phi}-l\cos(\theta)\ddot{\phi}+x\dot{\phi}^2 + L\sin(\theta)\dot{\phi}^2-2\dot{l}\dot{\phi}\cos(\theta)+2l\sin(\theta)\dot{\phi}\dot{\theta}\Big), \quad (5)$$

$$d_{wl}=m_p\sin(\theta)\Big(L\ddot{\phi}-l\cos(\theta)\ddot{\phi}+x\dot{\phi}^2+l\sin(\theta)\dot{\phi}^2 - 2\dot{l}\dot{\phi}\cos(\theta)+2l\sin(\theta)\dot{\phi}\dot{\theta}\Big)+m_p\cos(\theta)\Big(\ddot{h}+x\ddot{\phi} - h\dot{\phi}^2+l\sin(\theta)\ddot{\phi}+l\cos(\theta)\dot{\phi}^2+2\dot{l}\sin(\theta)\dot{\phi}+2\dot{x}\dot{\phi} + 2l\cos(\theta)\dot{\phi}\dot{\theta}\Big), \quad (6)$$

$$d_{w\theta}=m_pl\cos(\theta)\Big(L\ddot{\phi}-l\cos(\theta)\ddot{\phi}+x\dot{\phi}^2+l\sin(\theta)\dot{\phi}^2 - 2\dot{l}\dot{\phi}\cos(\theta)+2l\sin(\theta)\dot{\phi}\dot{\theta}\Big)-m_pl\sin(\theta)\Big(\ddot{h}+x\ddot{\phi} - \dot{h}\dot{\phi}^2+l\sin(\theta)\ddot{\phi}+l\cos(\theta)\dot{\phi}^2+2\dot{l}\sin(\theta)\dot{\phi}+2\dot{x}\dot{\phi} + 2l\cos(\theta)\dot{\phi}\dot{\theta}\Big), \quad (7)$$

where ϕ and h are considered predefined signals.

B. Problem Formulation

For the control design objective, the transformation is applied for the dynamic model (1) as follows: Consider the states of the dynamic system are defined by $x_1=x$, $x_2=\dot{x}$, $\eta_1=l$, $\eta_2=\dot{l}$, $\theta_1=\theta$, and $\theta_2=\dot{\theta}$, then the dynamic model (1) can be rewritten for the cart motion as

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\end{bmatrix} = \begin{bmatrix}0 & 1\\ 0 & 0\end{bmatrix}\begin{bmatrix}x_1\\ x_2\end{bmatrix} + \begin{bmatrix}0\\ 1\end{bmatrix}\Big(\frac{1}{m_c+m_p}F_x + \sigma_x\Big), \quad (8)$$

where $\sigma_x \triangleq \frac{1}{m_c+m_p}(-\Psi_x - f_{fx} + d_{wx})$ and Ψ_x is an unknown bounded nonlinear function that represents the cart motion's dynamic coupling and nonlinearities, defined as

$$\Psi_x \triangleq m_p \sin(\theta-\phi)\ddot{l} + m_p l\cos(\theta-\phi)\ddot{\theta} + 2m_p\cos(\theta-\phi)\dot{\theta}\dot{l} - m_p l\sin(\theta-\phi)\dot{\theta}^2,$$

and for the swing motion as

$$\begin{bmatrix}\dot{\theta}_1\\ \dot{\theta}_2\end{bmatrix} = \begin{bmatrix}0 & 1\\ 0 & 0\end{bmatrix}\begin{bmatrix}\theta_1\\ \theta_2\end{bmatrix} + \begin{bmatrix}0\\ 1\end{bmatrix}\frac{1}{m_p l^2}(-\Psi_\theta - f_{f\theta} + d_{w\theta}) \quad (9)$$

where Ψ_θ is an unknown bounded nonlinear function that represents the swing motion's dynamic coupling and nonlinearities, and it is defined as

$$\Psi_\theta \triangleq m_p l\cos(\theta-\phi)\ddot{x} + m_p l^2\ddot{l} + 2m_p l\dot{\theta}\dot{l} + m_p gl\sin(\theta-\phi).$$

Similarly, the transformed model of the hoist motion is given as

$$\begin{bmatrix}\dot{\eta}_1\\ \dot{\eta}_2\end{bmatrix} = \begin{bmatrix}0 & 1\\ 0 & 0\end{bmatrix}\begin{bmatrix}\eta_1\\ \eta_2\end{bmatrix} + \begin{bmatrix}0\\ 1\end{bmatrix}\left(\frac{1}{m_p}F_l + \sigma_l\right) \quad (10)$$

where $\sigma_l \triangleq \frac{1}{m_p}(-\Psi_l - f_{fl} + d_{wl})$ and Ψ_l is an unknown bounded nonlinear function that represents the hoist motion's dynamic coupling and nonlinearities defined as

$$\Psi_l \triangleq m_p\sin(\theta-\phi)\ddot{x} - m_p l\dot{\theta}\dot{l} - m_p g\cos(\theta-\phi),$$

Remark 1: The models (8),(9), and (10) have been transformed to allow the offshore crane system's dynamics to be represented as a second-order system with unknown dynamic coupling and nonlinearities. This transformation offers the benefit of being able to design a decoupling controller for the cart position and rope length. Furthermore, as demonstrated in this study, an extended high-gain observer can be used to estimate the dynamic coupling and nonlinearities.

III. Proposed Control Design

This section presents the design process for controlling the cart's position and the rope's length. To achieve this, we will employ the backstepping approach initially, which considers the system's nonlinearities and the available states. However, this approach is not practical for the crane system. Therefore, we will use an extended high-gain observer (EHGO) to estimate all system states, including the unknown nonlinearities and external disturbances.

A. Full-state Feedback Control

The primary goal of the controller is to ensure that the output position x_1 tracks the desired reference trajectory profile x_d, and the output η_1 tracks the desired reference trajectory profile l_d, despite the presence of unknown nonlinearities Ψ_x and Ψ_l, unknown friction forces f_{fx} and f_{fl}, and external disturbances caused by wave motion d_{wx} and d_{wl}. All derivatives of x_d and l_d are piecewise continuous and bounded for all $t \geq 0$.

In state feedback tracking controllers, variables are changed to transform an original system into an error coordinate system with an equilibrium at the origin. A state feedback control is then designed so that the origin of the transferred system is exponentially stable to achieve the desired tracking result.

*1) **Cart's Position Control**:* Toward accomplishing the objective of the control system, we consider the change in variables as $e_{x1} = x_1 - x_d$, $e_{x2} = x_2 - w_x$, and the desired signal w_x is selected as $w_x = -a_1 e_{x1} + \dot{x}_d$, where a_1 is a positive constant to be designed. This choice of change of variables will transform the system (8) into the following error dynamics,

$$\dot{e}_{x1} = -a_1 e_{x1} + e_{x2},$$
$$\dot{e}_{x2} = \frac{1}{m_c+m_p}F_x + \sigma_x - \dot{w}_x. \quad (11)$$

The control objective, now, is to design a state feedback control to stabilize the system (11). Towards this goal, suppose that the control input F_x can be designed to stabilize the origin ($e_{x1} = 0$, $e_{x2} = 0$). For this purpose, consider a smooth and positive definite Lyapunov function candidate as $V_x = \frac{1}{2}e_{x1}^2 + \frac{1}{2}e_{x2}^2$, where the derivative of V_x along the systems (11) can be expressed by

$$\dot{V}_x = e_{x1}(-a_1 e_{x1} + e_{x2}) + e_{x2}\left(\frac{1}{m_c+m_p}F_x + \sigma_x - \dot{w}_x\right). \quad (12)$$

Thus, the control signal u_p can be designed to make $\dot{V}_x$ strictly negative as

$$F_x = (m_c+m_p)(-a_2 e_{x2} - \sigma_x + \dot{w}_x), \quad (13)$$

where a_2 is the positive constant to be designed, σ_x is available by the EHGO, as will be shown later. With the proposed control signal F_x, (11) can be written as

$$\begin{bmatrix}\dot{e}_{x1}\\ \dot{e}_{x2}\end{bmatrix} = \begin{bmatrix}-a_1 & 1\\ 0 & -a_2\end{bmatrix}\begin{bmatrix}e_{x1}\\ e_{x2}\end{bmatrix} \quad (14)$$

where the constants a_1 and a_2 are selected such as the system (14) is exponentially stable and $\dot{V}_x < 0$.

*2) **Hoist Control**:* Toward accomplishing the objective of the control system, we consider the change in variables as $e_{\eta1} = \eta_1 - l_d$, $e_{\eta2} = \eta_2 - w_l$, and the desired signal w_l is selected as $w_l = -b_1 e_{\eta1} + \dot{l}_d$, where b_1 is a positive constant to be designed. This choice of change of variables will transform the system (10) into the following error dynamics,

$$\dot{e}_{\eta1} = -a_1 e_{\eta1} + e_{\eta2},$$
$$\dot{e}_{\eta2} = \frac{1}{m_p}F_l + \sigma_l - \dot{w}_l. \quad (15)$$

The control objective, now, is to design a state feedback control to stabilize the system (15). Towards this goal, suppose that the control input F_l can be designed to stabilize the origin ($e_{\eta1} = 0$, $e_{\eta2} = 0$). For this purpose, consider a smooth and positive definite Lyapunov function candidate as $V_\eta = \frac{1}{2}e_{\eta1}^2 + \frac{1}{2}e_{\eta2}^2$, where the derivative of V_η along the systems (15) can be expressed by

$$\dot{V}_\eta = e_{\eta1}\left(-b_1 e_{\eta1} + e_{\eta2}\right) + e_{\eta2}\left(\frac{1}{m_p}F_l + \sigma_l - \dot{w}_l\right). \quad (16)$$

Thus, the control signal u_p can be designed to make $\dot{V}_\eta$ strictly negative as

$$F_l = m_p\left(-b_2 e_{\eta2} - \sigma_l + \dot{w}_l\right), \quad (17)$$

where b_2 is the positive constant to be designed, σ_l is available by the EHGO, as will be shown later. With the proposed control signal F_l, (15) can be written as

$$\begin{bmatrix}\dot{e}_{\eta1}\\ \dot{e}_{\eta2}\end{bmatrix} = \begin{bmatrix}-b_1 & 1\\ 0 & -b_2\end{bmatrix}\begin{bmatrix}e_{\eta1}\\ e_{\eta2}\end{bmatrix} \quad (18)$$

where the constants b_1 and b_2 are selected such as the system (18) is exponentially stable and $\dot{V}_\eta < 0$.

***Remark** 2:* The control laws presented in equations (13) and (17) are independently designed using the backstepping approach, enabling decoupling in the resulting controller. This allows the designer to choose the gains a_1 and a_2 for the cart position controller independently of the gains b_1 and b_2 for the hoist controller, facilitating the achievement of desired transient performances. Moreover, this approach reduces the complexity of the controller design.

B. Extended High-gain Observer

The previous section presented the controller design for the cart position and the rope length, assuming all dynamic states and knowledge of the nonlinearities are available in the system for the controller. However, this assumption is impractical, and only the outputs are available for measurements. This section presents the design of an extended high-gain observer (EHGO) that uses the measured output to estimate all the dynamic states and the unknown nonlinearities in the system [14]. The extended high-gain observer (EHGO) demonstrates the ability to estimate system states, account for model uncertainties, and handle external disturbances [15]–[17].

For the offshore crane system, the two independent EGHOs will be designed; the first EHGO uses the output $y_1 = x_1$ to estimate the state x_2 and the nonlinear function σ_x. In contrast, the second EHGO uses the output $y_2 = \eta_1$ to estimate the state η_2 and the nonlinear function σ_l. The first EHGO for the cart position dynamic is formulated as

$$\dot{\hat{x}}_1 = \hat{x}_2 + \frac{\gamma_1}{\epsilon_x}(y_1 - \hat{x}_1), \quad (19)$$

$$\dot{\hat{x}}_2 = \frac{1}{m_c + m_p}F_x + \hat{\sigma}_x + \frac{\gamma_2}{\epsilon_x^2}(y_1 - \hat{x}_1), \quad (20)$$

$$\dot{\hat{\sigma}}_x = \frac{\gamma_3}{\epsilon_x^3}(y_1 - \hat{x}_1), \quad (21)$$

where $\epsilon_x > 0$ is a small parameter to be designed, and γ_1, γ_2, and γ_3, are chosen such that the polynomial $s^3+\gamma_1 s^2+\gamma_2 s+\gamma_3 = 0$ is Hurwitz. The error dynamic using the estimation states is given by $\hat{e}_{x1} = \hat{x}_1 - x_d$, $\hat{e}_{x2} = \hat{x}_2 - \hat{w}_x$, $\hat{w}_x = -a_1\hat{e}_{x1} + \dot{x}_d$.

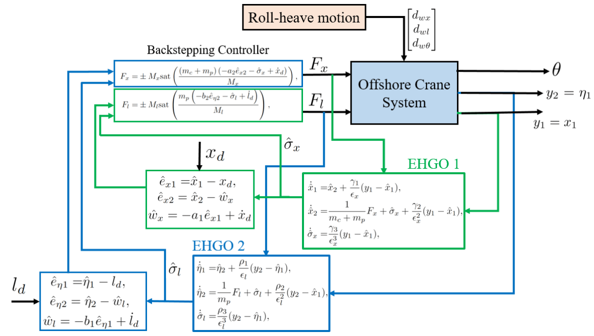


Fig. 2. Block diagram representation of the closed-loop system with the output-feedback controller.

Similarly, the EHGO for the yaw dynamic is formulated as

$$\dot{\hat{\eta}}_1 = \hat{\eta}_2 + \frac{\rho_1}{\epsilon_l}(y_2 - \hat{\eta}_1), \quad (22)$$

$$\dot{\hat{\eta}}_2 = \frac{1}{m_p}F_l + \hat{\sigma}_l + \frac{\rho_2}{\epsilon_l^2}(y_2 - \hat{x}_1), \quad (23)$$

$$\dot{\hat{\sigma}}_l = \frac{\rho_3}{\epsilon_l^3}(y_2 - \hat{\eta}_1), \quad (24)$$

where $\epsilon_l > 0$ is a small parameter to be designed, and ρ_1, ρ_2, and ρ_3, are chosen such that the polynomial $s^3+\rho_1 s^2+\rho_2 s+\rho_3 = 0$ is Hurwitz. The error dynamic using the estimation states is given by $\hat{e}_{\eta1} = \hat{\eta}_1 - l_d$, $\hat{e}_{\eta2} = \hat{\eta}_2 - \hat{w}_l$, $\hat{w}_l = -b_1\hat{e}_{\eta1} + \dot{l}_d$.

C. Output Feedback Control

This section combines the controller (13) with the extended high-gain observer (19)-(21) and the controller (17) with the extended high-gain observer (22)-(24) to solve the trajectory tracking control of the offshore crane system using the output feedback control. Accordingly, the output-feedback control using the estimated states is written by

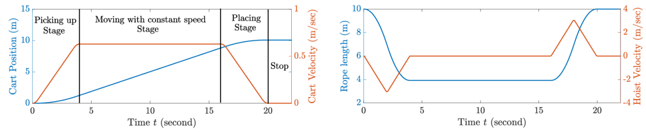
$$F_x = \pm M_x \mathrm{sat}\left(\frac{(m_c+m_p)\left(-a_2\hat{e}_{x2} - \hat{\sigma}_x + \dot{w}_x\right)}{M_x}\right), \quad (25)$$

$$F_l = \pm M_l \mathrm{sat}\left(\frac{m_p\left(-b_2\hat{e}_{\eta2} - \hat{\sigma}_l + \dot{w}_l\right)}{M_l}\right), \quad (26)$$

where $\mathrm{sat}(.)$ is the saturation function which is used to protect the system from peaking in the observer's transient response [18]. The saturation limit M_x and M_l are chosen based on the practical physical constraints of the maximum torque achieved by the motor. The structure of the closed-loop system with the proposed output-feedback controller is illustrated in Figure 2.

IV. Results and Discussion

To assess the efficacy of the proposed control approach, numerical simulations were conducted using assumed dynamic parameters for the offshore crane system based on [19] as $m_c = 6 \times 10^3$kg, $m_p = 20 \times 10^3$kg, $L = 15$m, and $g = 9.81$. The friction force coefficients are chosen as $a_x = 4.4$, $a_l = 2.2$, $b_x = b_l = 0.01$, $c_x = -0.5$,



(a) (b)

Fig. 3. (a) The desired trajectories for cart position and velocity; and (b) the desired trajectories for rope length and hoist velocity.

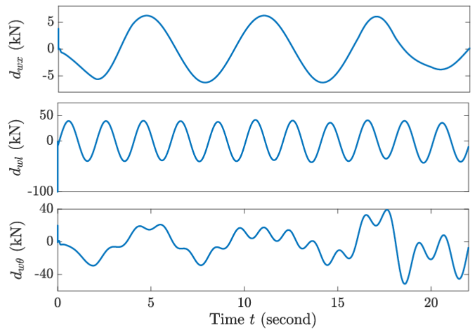


Fig. 4. The generated disturbances d_{wx}, d_{wl}, and $d_{w\theta}$ due to the roll-heave motion in (27).

$c_l = -0.25$, $d_x = d_l = d_{th} = 0.1$. The desired trajectories for the cart motion and hoist motion are considered as shown in Figure 3(a) and (b), respectively. The proposed desired trajectories are selected to accomplish the primary stages of container transfer, which include picking up the container, moving it at a constant speed, and placing it.

For the cart position controller and observer, the parameters are chosen as $a_1 = 50$, $a_2 = 20$, $\epsilon_x = 0.001$, $\gamma_1 = 0.5$, $\gamma_2 = 1$, and $\gamma_3 = 0.02$. For the hoist controller and observer, the parameters are chosen as $b_1 = 75$, $b_2 = 30$, $\epsilon_l = 0.01$, $\rho_1 = 2$, $\rho_2 = 0.3$, and $\rho_3 = 0.01$. The initial states are considered as $x_1(0) = 0$, $x_2(0) = 0$, $\eta_1(0) = 10$ and $\eta_2(0) = 0$. The saturation gains are selected as $M_x = [-5 \times 10^4,\ 5 \times 10^5]$ and $M_l = [-5 \times 10^5,\ 1 \times 10^5]$.

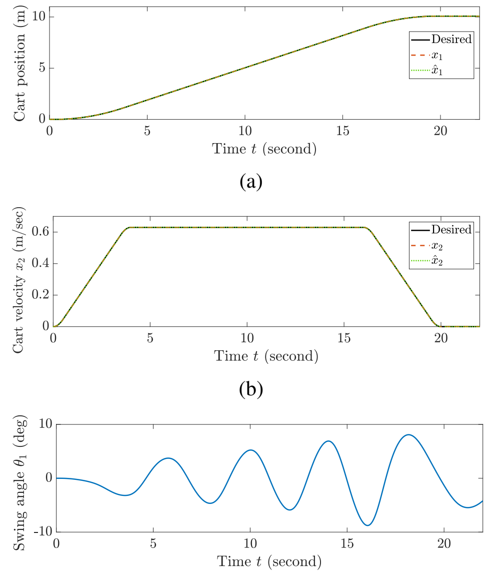
To demonstrate the robustness of the proposed control approach, we consider the roll-heave motion as

$$\phi(t) = 0.02 \sin(t), \quad h(t) = 0.2 \cos(\pi t + 1.3), \tag{27}$$

this generates the disturbances d_{wx}, d_{wl}, and $d_{w\theta}$ as given in Figure 4.

Figure 5 and Figure 6 shows a comparison of the desired trajectory, actual position, and estimated position for the cart's position (x_1), velocity (x_2), rope length (η_1), and hoist velocity (η_2). The swing angle behavior θ is depicted in Figure 5(c), which reveals that the angle is bounded within ± 8 degrees. The proposed controller achieves the desired trajectories with minimal error, as evident from the figures. Furthermore, the estimation of the nonlinear functions σ_x and σ_l with disturbances using the EHGO is illustrated in Figure 7.

Figure 8(a)-(b) depicts the tracking error of the cart position and rope length using controllers Γ_x and Γ_l with and without $\hat{\sigma}_x$ and $\hat{\sigma}_l$. The figures show that accurate estimation of the unknown nonlinearities reduces tracking error and enhances the controller's robustness. Finally, the input force

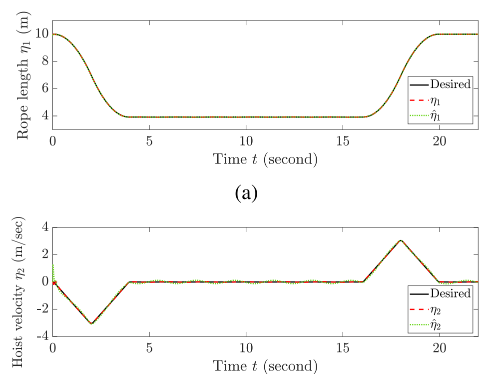


(a)

(b)

(c)

Fig. 5. (a) Tracking performance of the actual and estimation of cart position; (b) tracking performance of the actual and estimation of cart velocity; and (c) the response of swing angle.



(a)

(b)

Fig. 6. (a) Tracking performance of the actual and estimation of rope length; and (b) tracking performance of the actuation and estimation of the hoist velocity.

F_x and F_l, which represent the output of the controller under disturbance effect, are illustrated in Figure 8(c). From the simulation results of the offshore system with and without wave motion disturbances, it can be noticed the ability of the EHGO to estimate all nonlinearities and external disturbances of the system, which increases the robustness of the control system to achieve the desired trajectories and reduces the tracking error.

V. CONCLUSION

This study proposes a novel approach that combines the backstepping controller and extended high gain observer (EHGO) to control the offshore crane system with unknown dynamics and subject to external disturbances caused by the roll-heave motion. The backstepping controller is designed with appropriate gains to ensure exponential stability and desired transient performances, while the EHGO is used

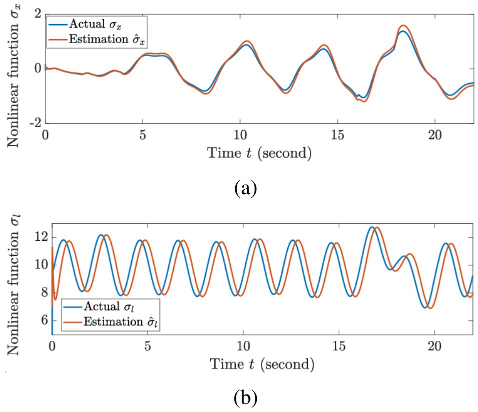


Fig. 7. (a) The comparison between the actual and the estimated of the nonlinear function σ_x; and (b) the comparison between the actual and the estimated of the nonlinear function σ_l.

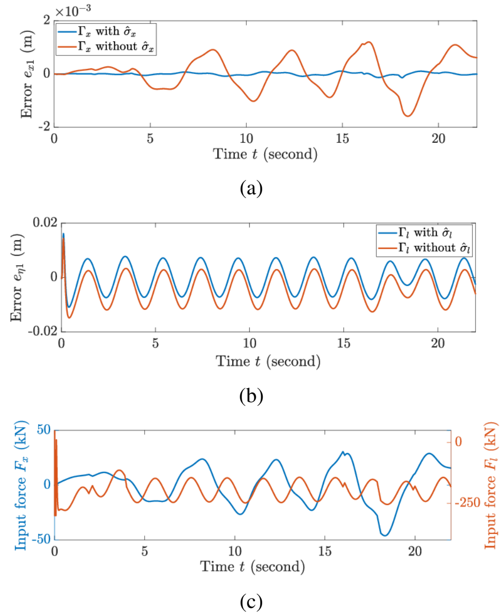


Fig. 8. (a) Tracking error e_{x1} with and without $\hat{\sigma}_x$ in control (25); (b) tracking error $e_{\eta 1}$ with and without $\hat{\sigma}_l$ in control (26); and (c) input force F_x and F_l.

to estimate the system's states and unknown dynamics, thereby enhancing the controller's robustness and accuracy. To evaluate the proposed controller's performance, numerical simulations are conducted for the offshore crane system without and with disturbances caused by wave motion. The simulation results demonstrate that the proposed controller can achieve the desired cart position and rope length trajectories under unknown external disturbances with minimum tracking error while satisfying the control system's decoupling design. Additionally, the EHGO effectively and quickly estimated the unknown dynamics of the offshore crane system and the disturbances caused by the roll-heave motion, highlighting the proposed approach's effectiveness in improving the system's robustness.

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