THE NONLINEAR DYNAMICS AND ANTI-SWAY TRACKING CONTROL FOR OFFSHORE CONTAINER CRANE ON A MOBILE HARBOR

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Key words: offshore container crane, nonlinear dynamics, ship motions, anti-sway control, position tracking.

ABSTRACT

The offshore container crane (OCC), as a novel maritime container transfer system, can handle container from a large container-ship anchored in open sea to address the congestion and limited water-depth of port. However, for the wave- and wind-induced movements of the ship, the crane’s control system should be redesigned to ensure the load transfer on the sea. In this paper, we derive the nonlinear dynamic equations of OCC system subjected to the ship motions based on dynamic analysis. Then a double-layer sliding manifold is constructed to realize the position tracking and sway control simultaneously, irrespective of ship motions and parameters perturbation. The Lyapunov method is utilized to prove the stability of the proposed control law. Next, virtual prototype of the OCC is established, including the multi-body dynamics model of OCC with flexible rope and the proposed control scheme. Sufficient simulations are provided to illustrate its improved performance versus conventional controller. Experiments are also implemented to evaluate its practical control performance of trajectory following and sway angle suppression.

I. INTRODUCTION

With the integration of the global economy, rapid development has been made in container logistics industry. Meanwhile, the harbor congestion and modern large container-ship unable to dock at shallow waters harbor have become urgent problems (Oscar, 2015; Jin, 2016). Compared with the expansion of harbor scale, mobile harbors have become the most flexible, economical and environmental-friendly solution (Jonghoe, 2012; Baird, 2013). As illustrated in Fig. 1, this concept is to install the container crane on an offshore platform, which is called “offshore container crane” (OCC), to load and unload containers (payloads) for large container-ship anchoring in open sea and transport them to destination harbor (Jang, 2014).

Due to the effects of waves and trolley motion trajectory, the transferred payloads will generate a complex nonlinear dynamic response. Especially, the residual sway of payload decreases operation efficiency sharply, which can lead to serious damages. In fact, the trolley trajectory planning and payload anti-swing of land-based cranes are widely used in transportation and construction industries. There have also been a lot of research work focused on this area in the past two decades. The researchers in crane dynamics and control fields are interested in fast, no residual swing and high efficient anti-swing solutions. Existing work can be divided into open-loop control and closed-loop control. The open-loop control method with representative of input shaping (Garrido, 2008) and optimal control (Terashima, 2007) has a strong dependence on the accuracy of system mathematical model. The closed-loop control methods are usually combined with intelligent control, such as H-2/H-infinity Control (Hilhorst, 2015), fuzzy control (Chang, 2007; Li, 2015; Wu, 2016), neural networks control (Saiedi, 2013), sliding mode control (Almutairi, 2009), etc. Moreover, the dynamics and control strategies of ship-mounted cranes, which are divided into boom crane (Chin, 2001; Skaare, 2006; Sanfilippo, 2016) and container crane (Park, 2012; Le, 2015), also have been studied. Henry et al. (2001) proposed a delayed feedback control law to suppress the oscillations of the load of boom cranes. Cha et al. (2010) and Ham et al. (2015) studied the multi-body dynamics of floating cranes. Ngo et al. (2012) developed a sliding model controller based on Lyapunov method to reduce the sway angle of the con-
tainer crane. Sun et al. (2015) presented a self-adaptive PID controller based on GA to control the floating crane. Ismail et al. (2015) constructed a LQR-based sliding surface to track the crane’s desired trajectory in the presence of waves and winds. Although the dynamics analysis and control of cranes have made great progress, there are still few researches on the OCC. It is necessary to improve the nonlinear dynamic equations of the offshore crane under the coupling influence of ship motions and trolley motions during lifting on the sea. Moreover, the present control approaches generally linearize the nonlinear dynamic model at the equilibrium point or neglect some nonlinear term in the equations to design the controller. It is feasible for land cranes since the external disturbance is little and the system is hardly far away from the equilibrium point. While for the offshore crane, there are permanent external disturbances from the sea environment. The system is easy to be far away from the equilibrium point by the effect of the disturbances, thus the performances of controllers based on the linear control theory will decrease greatly and even cause accidents. Therefore, it is urgent to design a nonlinear controller according to the nonlinear dynamic model of the crane under the ocean environment, without any linear approximation to ensure the control performance under permanent external disturbances.

The ship of mobile harbor, on which the container crane is installed, is easy to be disturbed away from the designated position horizontally and vertically. A lot of dynamic positioning systems have been proposed based on the nonlinear control theories to control the horizontal movement of the ship (Do, 2002; Serrano, 2014). On the other hand, the heave compensation system is designed to deal with the vertical shifting of the ship (Kuchler, 2014). However not enough attention has been paid to the rolling motion of the ship. For container transfer operations, the rolling motion of the ship holds great significance for the trolley position tracking and payload anti-sway.

Thus, in this paper, we construct dynamics equations of OCC system, which reveals the influence of ship motion and trolley movement on payload dynamic behavior, considering the trolley motion, heave motion and roll motion of the ship. An anti-sway tracking control strategy of SOSM (second order sliding mode) is presented for sway suppression and trajectory tracking despite the ship motion disturbances. We not only prove the stability of the sliding surface at all layers theoretically, but also implement the simulations and experiments to evaluate its excellent control performance.

II. DYNAMICS DEVELOPMENT AND ANALYSIS

For OCC system, due to the pitch motion of the ship, the lateral swing will be generated. But thanks to the immobilization of trolley at the lateral direction, many mechanical anti-sway devices can be utilized to eliminate lateral sway (Hong, 2009; Wang, 2013). Thus, this paper will not discuss the lateral anti-sway, which means we can ignore the pitch motion.

The three coordinate systems are introduced to derive mathematical dynamic equations of OCC, as shown in Fig. 2. $O\xi\eta\nu$ is the inertial coordinate frame, which defines the direction from port to starboard as positive direction of $X_0$ axis. $O\xi\eta\nu$, denotes the ship coordinate frame affixed to the center of the gravity of the hull. $O_{xs}\eta_{ys}\nu_{ys}$, denotes the trolley coordinate frame attached to the trolley. $M, m$ respectively represents the masses of the trolley and the payload (container). $h$ denotes the height of crane gantry. $x$ and $y$ are the trolley position and the ship heave displacement in the inertial coordinate frame. $l(t)$ is the varying length of rope. $\theta$ is the sway angle of payload in the plane of trolley motion. $f_t$ is the control force applied to the trolley. $\phi$ is roll angle of the ship under sea wave excitation. So ship motion vector is defined as $(y, \phi)$. In inertial coordinate frame, the trolley position $p_{m}$ and payload position $p_{m}$ can be derived as below:

\[
P_{M} = \begin{bmatrix} x \cos \phi - h \sin \phi \\ y + h \cos \phi + x \sin \phi \end{bmatrix}
\]

(1)

\[
P_{m} = \begin{bmatrix} x \cos \phi - h \sin \phi + l \sin \theta \\ y + h \cos \phi + x \sin \phi - l \cos \theta \end{bmatrix}
\]

(2)

Based on (1) and (2), trolley velocity $v_{m}$ and payload velocity $v_{m}$ can be obtained as follows:

\[
v_{M} = \begin{bmatrix} v_{Mx} \\ v_{My} \end{bmatrix}
\]

(3)

\[
v_{m} = \begin{bmatrix} v_{mx} \\ v_{my} \end{bmatrix}
\]

(4)

where

\[
v_{Mx} = \dot{x} \cos \phi - x \dot{\phi} \sin \phi - h \dot{\phi} \cos \phi
\]

\[
v_{My} = \dot{y} + \dot{x} \sin \phi + x \dot{\phi} \cos \phi - h \dot{\phi} \sin \phi
\]

\[
v_{mx} = \dot{x} \cos \phi - x \dot{\phi} \sin \phi - h \dot{\phi} \cos \phi + \dot{l} \sin \theta + l \dot{\theta} \cos \theta
\]

\[
v_{my} = \dot{y} + \dot{x} \sin \phi + x \dot{\phi} \cos \phi - h \dot{\phi} \sin \phi - l \cos \theta + l \dot{\theta} \sin \theta
\]

The kinetic energy and potential energy of trolley-payload system are expressed as:
\[ T = \frac{1}{2} M \left( v_{x_t}^2 + v_{y_t}^2 \right) + \frac{1}{2} m \left( v_{x_m}^2 + v_{y_m}^2 \right) \]  
\[ U = M g (y + h \cos \theta + x \sin \phi) - m g l \cos \theta \]
\[ + m g (y + h \cos \phi + x \sin \theta) \]

where, \( g \) denotes the acceleration of gravity. The ship’s kinetic and potential energy are not included, since the ship motions \((y, \phi)\) are treated as external disturbance. \( q = (x, \theta) \) is defined as generalized coordinate and \( f = (f_x, 0) \) denotes generalized force. The Lagrange’s equations are:

\[ \frac{d}{dt} \left( \frac{\partial T}{\partial \dot{q}_i} \right) - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial \dot{q}_i} = f_i \]  

The following nonlinear dynamic equations are obtained:

\[(M + m) \ddot{x} - (M + m) x \ddot{\phi} - (M + m) h \dot{\phi} + (M + m) \dot{y} \sin \phi
- m l \dot{\theta}^2 \sin (\theta - \phi) + m l \dot{\theta} \cos (\theta - \phi) + m l \sin (\theta - \phi)
+ (M + m) g \sin \phi + 2 m l \dot{\theta} \cos (\theta - \phi) = f_x \]  
\[(8)
\]

\[ x \cos (\theta - \phi) + \dot{y} \sin \theta + x \dot{\phi} \sin (\theta - \phi) + l \ddot{\theta} + g \sin \theta
- x \dot{\phi}^2 \cos (\theta - \phi) - h \dot{\phi} \cos (\theta - \phi) + 2 l \dot{\theta}
- h \ddot{\phi} \sin (\theta - \phi) + 2 x \dot{\phi} \sin (\theta - \phi) = 0 \]  
\[(9)

In order to verify the validity of dynamics model derived in Eqs. (8) and (9), an experimental setup is established. The specific parameters of the setup can be found in section V. A driving force \( f_s = f_0 u(t) \) is applied to the trolley. \( u(t) \) is step function once every 10 seconds. The amplitude \( f_0 \) is 10 \( N \). The wave-induced rolling motion of the ship is set as \( \phi(t) = 0.01 \sin(0.8 \text{ time}) \) rad. The Runge-Kutta method is utilized to obtain numerical solution for Eqs. (8) and (9) (Sun, 2017). The numerical results and experimental results, as shown in Figs. 3 and 4, are provided together for comparison. The results show that the numerical results of dynamics equations and the experimental results are basically consistent, which verify the validity of the derived dynamic equations.

III. CONTROL STRATEGY DESIGN AND ANALYSIS

A second order sliding mode (SOSM) position track and anti-sway control strategy is presented based on the nonlinear dynamics equations of OCC derived in the previous section.

Based on (8) and (9), the state space equations of OCC system can be obtained as follows:

\[
\begin{align*}
\dot{x}_1 (t) &= x_2 (t), \\
\dot{x}_2 (t) &= g_1 (X) + b_1 (X) u + d_1 (t), \\
\dot{x}_3 (t) &= x_4 (t), \\
\dot{x}_4 (t) &= g_2 (X) + b_2 (X) u + d_2 (t), \\
y(t) &= [x_1 (t), x_3 (t)]^T
\end{align*}
\]

where \( x_1 = x, x_2 = \dot{x}, x_3 = \theta, x_4 = \dot{\theta}, u = f_x, d_1 (t), d_2 (t) \) are bounded parameters perturbation terms. \( g_1 (X), g_2 (X), b_1 (X), b_2 (X) \) are nonlinear functions as follows:

\[
\begin{align*}
g_1 &= \frac{\zeta_1 \cos (\theta - \phi) - \zeta_1}{M + m - m \cos^2 (\theta - \phi)} \\
h_1 &= \frac{1}{M + m - m \cos^2 (\theta - \phi)} \\
g_2 &= \frac{\zeta_2 \cos (\theta - \phi) - \zeta_2 \cos^2 (\theta - \phi) - \zeta_1 (M + m - m \cos^2 (\theta - \phi))}{l (M + m - m \cos^2 (\theta - \phi))} \\
b_2 &= \frac{\cos (\theta - \phi)}{l (M + m - m \cos^2 (\theta - \phi))}
\end{align*}
\]

where

\[
\begin{align*}
\zeta_1 &= -(M + m) h \dot{\phi} + (M + m) (g + \dot{y}) \sin \phi - m l \ddot{\theta}^2 \sin (\theta - \phi) \\
&\quad - (M + m) x \ddot{\phi}^2 + m l \sin (\theta - \phi) + 2 m l \dot{\theta} \cos (\theta - \phi) \\
\zeta_2 &= m \left( 2 l \ddot{\theta} + 2 x \dot{\phi} \sin (\theta - \phi) - x \dot{\phi}^2 \cos (\theta - \phi) + x \dot{\phi} \sin (\theta - \phi) \\
&\quad - h \dot{\phi} \cos (\theta - \phi) - h \ddot{\phi} \sin (\theta - \phi) + (g + \dot{y}) \sin \theta \right)
\end{align*}
\]
Assume that there exists positive constants $g_{1M}$, $g_{2M}$, $d_{1M}$ and $d_{2M}$ to ensure $g_{1i}(X) \leq g_{1M}$, $g_{2i}(X) \leq g_{2M}$ and $d_{1i}(t) \leq d_{1M}$, $d_{2i}(t) \leq d_{2M}$.

The entire system state is divided into two sliding surfaces according to the general construction form of sliding surface:

$s = \lambda e + e$.

The target and real-time tracking positions of the trolley are defined as $x_d$ and $x$, respectively. Plan the target swinging angle $\theta_d = 0$ and $\theta$ denotes the real-time swinging angle. Then error vector can be expressed as:

$$e = [e_x \quad e_{\theta}]^T = [x - x_d \quad \theta - \theta_d]^T$$ (11)

The first-layer of sliding surface can be defined as:

$$s_1 = c_1e_x + \dot{e}_x$$
$$s_2 = c_2e_{\theta} + \dot{e}_{\theta}$$ (12)

where, $c_1$, $c_2$ are positive constants

The equivalent control terms $u_{eq1}$ and $u_{eq2}$ on the sliding manifold of each subsystem can be calculated using equivalent control method as follows:

$$u_{eq1} = -\frac{g_{1i}(X) + c_1x_2 + d_1}{b_1}$$ (13)
$$u_{eq2} = -\frac{g_{2i}(X) + c_2x_2 + d_2}{b_2}$$ (14)

As a typical underactuated system, offshore crane system can hardly fulfill the position track of the trolley and loads anti-swing simultaneously with only one control input. For this reason, this paper ensures the synchronous combination control of such two objectives based on secondary sliding surface $S$ by constructing the following second-layer sliding manifold:

$$S = \alpha s_1 + \beta s_2$$ (15)

where, $\alpha$ is a positive constant, and $\beta$ is a variable following the system states.

As for an underactuated system, the controller should ensure both the stability of actuated parts and the self-stability of under-actuated parts. Therefore, to ensure each subsystem is on its own sliding surface, the total system control law must contain the control formula of each subsystem, which can be defined as:

$$u = u_{eq1} + u_{eq2} + u_m = -(\alpha b_1 - \beta b_2)^{-1}[\alpha (c_1x_2 + g_1 + d_1)$$
$$+ \beta (c_2x_4 + g_2 + d_2) + \eta \text{sgn}(S) + kS]$$ (16)

where, $u_m$ is the switching control component when the system is at the reaching phase. $\eta$ and $k$ are positive control gains. $K_1$ is defined as:

$$K_1 = \sup_{t \geq 0} |u_{eq2} - u_{eq1}|$$ (17)

Each parameter of the controller needs to meet the following conditions:

$$\beta = \begin{cases} \beta_0 & s_1s_2 > 0, \beta_0 > 0; \\ -\beta_0 & s_1s_2 \leq 0, \beta_0 > 0; \end{cases}$$ (18)
$$\alpha > \frac{b_1}{b_2} \beta_0, \beta_0 > 0;$$ (19)
$$\eta > |\beta| |b_2| K_1$$ (20)

**Theorem 1**: As for nonlinear system depicted in (10), sliding surfaces with double-layer structures are constructed based on (12) and (15). If controller is expressed in the form of (16) and its parameters satisfy (18) and (20), the second layer sliding surface $S$ is stable, so is the first layer sliding surface $s_1$.

**Proof**:

The energy function based on Lyapunov theorem is built on the second layer sliding surface $S$. The Lyapunov function candidate is chosen as:

$$V(t) = \frac{1}{2} S^2$$ (21)

The derivative of $V(t)$ in time is calculated as:

$$\dot{V}(t) = S^T \dot{S} = S^T (\alpha \dot{s}_1 + \beta \dot{s}_2)$$

$$= S^T [\alpha (c_1x_2 + g_1 + b_1u + d_1)$$
$$+ \beta (c_2x_4 + g_2 + b_2u + d_2)]$$
$$= -\eta s_1 - \eta \beta s_2 - (\alpha b_1 - \beta b_2)kS^2 \leq 0$$ (22)

According to Lyapunov stability theorem, if $\dot{V} \leq 0$, the system approaches asymptotic stability when $s = 0$ and so the second layer sliding surface $S$ is stable.

Likewise, the Lyapunov energy function is built on the first layer sliding surface $s_1$ and can be denote as $V(t) = \frac{1}{2} S_1^2$, whose time derivative is calculated as:
\[ V_1(t) = s_1^T \dot{s}_1 = s_1^T \left[ c(x_1 + g_1 + b_1u + d_1) \right] = \frac{b_1}{ab_1 + \beta b_2} \left[ b_2 \left( u_{eq2} - u_{eq1} \right) \right] \beta s_1 - \eta s_1 \] (23)

where, \( k^* = b_1k \) and \( ab_1 + \beta b_2 > 0 \) can be guaranteed by (19).

\[ \frac{b_1}{ab_1 + \beta b_2} > 0 \] because of \( b_1 > 0 \). So (23) can be simplified as:

\[ \dot{V}_1 = s_1 \dot{s}_1 = \frac{b_1}{ab_1 + \beta b_2} \left[ b_2 \left( u_{eq2} - u_{eq1} \right) \right] \beta s_1 - \eta s_1 \]

\[ \leq \frac{b_1}{ab_1 + \beta b_2} \left( s_1b_1\beta K_1 - \eta s_1 \right) - k^* \alpha s_1^2 \] (24)

\[ = \frac{b_1}{ab_1 + \beta b_2} \left( s_1b_1\beta K_1 - \eta s_1 \right) - k^* \alpha s_1^2 \]

\[ -k^* \beta s_1 s_2 \text{sgn}(s_1 s_2) \]

when \( \eta > \beta b_1K_1 \) and \( \beta \) satisfies (18), \( \dot{V}_1 < 0 \). It follows that the first layer sliding surface \( s_1 \) is stable as well.

**Theorem 2:** As for nonlinear system described in (10), sliding surfaces with double-layers structures are constructed based on (12) and (15). The controller is in the form of (16). If \( s_1 \) and \( S \) are stable, then \( s_2 \) is stable as well.

**Proof:**

The first-order sliding mode surface \( s_1 \) is stable, indicating that \( s_1 \) has existence and reachability. With arbitrary initial condition \( s_{10} \), there exists a time \( t_1(t_1 \in R^-) \) to ensure \( \lim_{t \to t_1^-} s_1 = 0 \), which means \( s_1 \) can converge to zero within finite time. Similarly, as for the \( S \) with arbitrary initial condition \( S_0 \), there exists a time \( t_2(t_2 \in R^-) \) to ensure \( \lim_{t \to t_2^-} S = 0 \). As a result, the sliding surface \( s_2 \) can be rewritten as:

\[ \lim_{t \to t_2^-} s_2 = \lim_{t \to t_2^-} \frac{1}{\beta} (S - \alpha s_1) = 0 \] (25)

Thus, the first-layer sliding surface \( s_2 \) is stable.

**Remark 1:** the proof of the stability of \( s_2 \) in theorem 2 is conservative, which means \( s_2 \) can converge to zero when any \( t \geq T \). The conclusion is sufficient. But there is no explanation on whether \( s_2 \) can converge to zero when \( t < T \). Actually, in addition to converging to zero with \( s_1 \) or \( S \), \( s_2 \) may converge to zero with \( S = \alpha s_1 \) at a faster convergence rate than \( s_1 \) or \( S \).

**Remark 2:** If the controller can ensure the stability of \( s_1 \) and \( S \), it can control the sliding surface \( s_2 \) as well. It is thus evident that the presented control strategy can fulfill position track of the trolley and anti-swing of the payload simultaneously.

**IV. VIRTUAL PROTOTYPE SIMULATIONS**

Since it is very time-consuming and difficult to build up an OCC and test it under real sea conditions, we utilize virtual prototype technology to testify the designed offshore container crane control system, which can help engineers to modify the mechanical design and improve the controller.

**1. Multi-Body Dynamics Mechanical Model of OCC System**

Considering the suspended rope as a flexible body, a rigid-flexible coupling multi-body dynamics model of OCC is constructed in ADAMS environment. The development process of OCC’s virtual prototype is shown in Fig. 5.

Detailed steps are shown as follows: a 3D model of OCC system based on the actual size and shape is built using SOLIDWORKS and imported into ADAMS environment. Then, the parameters of the components such as mass, material property, moment of inertia, etc. must be defined. Next, by utilizing constraints, these parts are connected to each other. For example, the trolley is connected with the boom using translational joint. The crane gantry is mounted on the ship using fixed joint. The flexible cables are generated by Machinery/Cable module. The contact forces are added between winch and winded ropes (Dong, 2015). The ship is induced to roll and heave based on the wave disturbance function. The developed multi-body dynamics model of OCC system is shown in Fig. 6.
Y. Sun et al.: Dynamics and Control for Offshore Crane

3. Co-Simulation Results

Co-simulation results of the virtual prototype are collected to illustrate the performance of the proposed controller. In order to demonstrate increased performance of the proposed controller versus conventional controller, the simulation results of the conventional PID controller are also provided.

According to the real OCC, the parameters of the system are chosen as: \( h = 48 \text{ m}, M = 2.0 \times 10^4 \text{ kg}, l = 15 \text{ m}, x_d = 36 \text{ m}, \phi(t) = 0.007 \sin(1.25 t) \text{ rad (Sea State 3)}, \phi(t) = 0.0165 \sin(0.924 t) \text{ rad (Sea State 4)} \) and \( \phi(t) = 0.0286 \sin(0.714 t) \text{ rad (Sea State 5)} \).

(1) Co-simulation results of PID controller

In order to control the position and sway simultaneously, the double-PID controllers are selected including position PID and sway PID. The controller parameters are sufficiently tuned to obtain the best performance, which yields the following values:

\[
P_1 = 2500, \quad P_2 = 100, \quad P_3 = 18000 \quad \text{for position PID}.
\]

\[
P_4 = 500, \quad P_5 = 20, \quad P_6 = 700 \quad \text{for sway PID}.
\]
Fig. 12. The tracking and anti-sway results of PID controller with $f_{\text{roll}} = 1.25$, $\delta_{\text{roll}} = 0.007$ (sea state 3).

Fig. 13. The tracking and anti-sway results of PID controller with $f_{\text{roll}} = 0.924$, $\delta_{\text{roll}} = 0.0165$ (sea state 4).

Fig. 14. The tracking and anti-sway results of PID controller with $f_{\text{roll}} = 0.714$, $\delta_{\text{roll}} = 0.0286$ (sea state 5).

Fig. 15. The tracking and anti-sway results of the proposed controller with $f_{\text{roll}} = 1.25$, $\delta_{\text{roll}} = 0.007$ (sea state 3).

Fig. 16. The tracking and anti-sway results of proposed controller with $f_{\text{roll}} = 0.924$, $\delta_{\text{roll}} = 0.0165$ (sea state 4).

Fig. 17. The tracking and anti-sway results of proposed controller with $f_{\text{roll}} = 0.714$, $\delta_{\text{roll}} = 0.0286$ (sea state 5).

According to the Figs. 12-17, the relevant data statistics are listed in the Tables 1 and 2 to evaluate the control performance. The co-simulation results demonstrate that conventional PID controller’s settling time is much longer than the required time and the residual sway angle also can’t meet the requirement of control goal. For the proposed SOSM controller, overshoot $= 0.74\%-0.87\%$, settling time $= 10.9\text{-}12.1$ second and steady state error is $0.02\text{-}0.04$ m upon sea state 3 & 4, which can satisfy the control goal perfectly. But due to the ship moves sharply on the sea state 5, the trolley can’t achieve accurate positioning with the steady state error of $0.11$ m.

In summary, for the novel OCC system, because of the complexity of marine environment, the conventional PID control method no longer meet the control goals. The proposed anti-sway tracking controller has a good performance and strong robustness. It can track the desired position quickly and accurately with rather small residual sway angle upon sea state 3 and 4.
V. EXPERIMENTAL RESULTS

After sufficient simulation tests, much effort has been put to perform experiments to further evaluate the performance of the proposed control strategy.

Since it is hard to build an offshore container crane attached in a mobile harbor with disturbance from currents and waves, an experimental setup is established as illustrated in Fig. 18.

The test bed includes crane structure, 6-DOF motion platform, angular and displacement sensors (encoders), driving system and controller crate shown in Fig. 19. The 6-DOF platform can imitate the roll motion of the ship. The upper computer utilizes Matlab/Simulink RTWT (Real Time Windows Target) as real-time control platform. The DMC-1842 motion control card is chosen to programme the upper computer’s output signal, whose functional block diagram is plotted in Fig. 20.

The experimental system parameters are set to be:

\[ h = 2.5 \, \text{m}, \quad M = 16 \, \text{kg}, \quad m = 4 \, \text{kg}, \quad l = 0.7 \, \text{m}, \quad x_d = 0.5 \, \text{m}. \]

For the experiments, the parameters and gains of SOSM controller are carefully determined as follows:

---

### Table 1. The comparison of the trolley position response.

<table>
<thead>
<tr>
<th>external disturbance</th>
<th>Control method</th>
<th>Settling time (s)</th>
<th>Overshoot (%)</th>
<th>steady-state error (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sea State 3</td>
<td>none</td>
<td>/</td>
<td>/</td>
<td>0.20</td>
</tr>
<tr>
<td></td>
<td>PID</td>
<td>23.6</td>
<td>4.92</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td>SOSM</td>
<td>10.9</td>
<td>0.74</td>
<td>0.02</td>
</tr>
<tr>
<td>Sea State 4</td>
<td>none</td>
<td>/</td>
<td>/</td>
<td>0.40</td>
</tr>
<tr>
<td></td>
<td>PID</td>
<td>24.5</td>
<td>5.23</td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>SOSM</td>
<td>11.3</td>
<td>0.81</td>
<td>0.04</td>
</tr>
<tr>
<td>Sea State 5</td>
<td>none</td>
<td>/</td>
<td>/</td>
<td>1.21</td>
</tr>
<tr>
<td></td>
<td>PID</td>
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<td>5.60</td>
<td>0.38</td>
</tr>
<tr>
<td></td>
<td>SOSM</td>
<td>12.1</td>
<td>0.87</td>
<td>0.11</td>
</tr>
</tbody>
</table>

### Table 2. The comparison of the payload sway angle response.

<table>
<thead>
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<th>Control method</th>
<th>Settling time (s)</th>
<th>residual sway angle (rad)</th>
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Fig. 18. Structure of the experimental test bed.

Fig. 19. Experimental system.
To fully investigate the performance of the proposed track position & anti-sway control strategy, two sets of experiments are implemented to evaluate the controller’s tracking performance and the sway suppression capacity, respectively.

**Experiment 1: Proposed Controller without Ship Motion**

The experimental results for the proposed control algorithm are provided in Figs. 21 and 22 in experiment of the ship is stationary (φ = 0). we can find that the sway angle is eliminated remarkably within 2 seconds before the trolley reach its desired position.

**Experiment 2: Proposed Controller with the Ship Roll Motion**

The roll motion of the ship used for the experiment verifications is depicted in the Fig. 23. Figs. 24-26 illustrate the control performance of the designed control strategy with the existent motion of ship. Fig. 24 shows that the trolley can track the goal position quickly and the position error stays within an acceptable motion region [-0.02 m, 0.02 m]. It can be found in Fig. 25 that the payload
maintains small residual sway angle less than 0.02 rad, which can satisfy the control objective remarkably.

VI. CONCLUSION

In this paper, we have addressed the problem of payload sway suppression and trolley position track for OCC system with disturbances of ship motions and parameters perturbation. By utilizing Euler-Lagrange equations, the mathematical model of OCC system is derived comprehensively and a SOSC control strategy is presented to achieve the control goal. A virtual prototype of OCC is established and extensive simulation results show a good cancellation of the track error and residual sway upon sea state 3 and 4, but not so satisfying on sea state 5. Experimental results are provided to examine its practical control performance. The designed method is also applicable to other types of cranes with convenient modification (including gantry cranes and tower cranes) and can be used for reference to control other underactuated mechatronic systems as well.

ACKNOWLEDGEMENTS

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REFERENCES