Sliding-Mode Antisway Control of an Offshore Container Crane

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***Abstract*—In this paper, a sliding-mode control for an offshore container crane is discussed. The offshore container crane is used to load/unload containers between a huge container ship (called the "mother ship") and a smaller ship (called the "mobile harbor"), on which the crane is installed. The purpose of the mobile harbor is to load/unload containers in the open sea and transport them to shallower water where they can be offloaded at existing conventional ports, thereby obviating the need for expansive and expensive new facilities. The load/unload control objective is to suppress the pendulum motion (i.e., "sway") of the load in the presence of the wave- and wind-induced movements (heave, roll, and pitch) of the mobile harbor. A new mechanism for lateral sway control, therefore, is proposed as well. A sliding surface is designed in such a way that the longitudinal sway of the load is incorporated with the trolley dynamics. The asymptotic stability of the closed-loop system is guaranteed by a control law derived for the purpose. The proposed new mechanism can suppress lateral sway, which functionality is not possible with conventional cranes. Simulation results are provided.**

***Index Terms*—Antisway control, mobile harbor, offshore container crane, ship motions, sliding-mode control.**

I. Introduction

SINCE the introduction of containers to the world-trade industry, an increasing number of goods have been loaded onto vessels for transport to all over world destinations. When a container ship arrives at a port, it is necessary that containers destined for that port be unloaded and that new containers bound for other ports be loaded, as quickly as possible. The demand on container terminals for maximum efficiency loading/unloading will grow even greater as shipping companies continue to increase the sizes of their vessels [1].

An unfortunate consequence of ever-larger ships, however, is the need for ever-larger ports, not to mention cranes. Fortunately though, open-sea loading/unloading of containers, in which process a relatively small ship (called the "mobile harbor") equipped with a crane loads/unloads containers from a large anchored container ship (called the "mother ship"), is a potential solution as a new trend. Fig. 1 shows the typical arrangement of a mother ship and a mobile harbor.

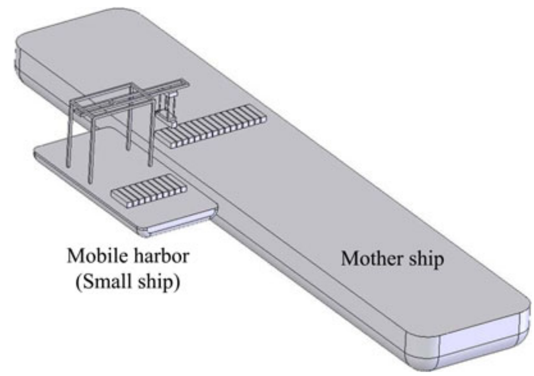

Fig. 1. Offshore container crane at a mobile harbor.

Owing to crane dynamics and disturbances, residual pendulum motion (i.e., sway) occurs at the end of every trolley movement. Payload oscillation and the obligation to suppress it, in fact, have become a bottleneck headache for the transportation and construction industries, even where relatively simple gantry cranes are employed. Researchers in crane control, therefore, have focused on quick, seamless, and efficient sway-suppression solutions. The resultant crane control schemes developed include input shaping controls [2]–[8], optimal controls [9], [10], linear/nonlinear controls [11]–[17], boundary controls from the axially moving system perspective [18], [19], sliding-mode controls [20]–[23], fuzzy controls [24], [25], and adaptive controls [26], [27]. Moreover, dynamics and control algorithms relevant to ship cranes have also been studied [28]–[35]. Although orientation control of a container itself is another important issue (see [36] for a treatment of skew control), in this study, the container is considered as a particle, not a rigid body.

The control methods developed for conventional quay cranes are inapplicable to mobile harbor cranes, owing to the additional, lateral sway motions imparted to the load by the pitching of the mobile harbor. Specifically, sea-excited motions of the mobile harbor, due to large-amplitude sea waves, winds, or other external disturbances, amplify the pendulum (i.e., sway) motion of a hanging load. Such external-disturbance motion must be compensated for by a mobile harbor's container crane; otherwise, offshore loading/unloading operations are not possible.

This study investigates the applicability and benefits of splitting the sway angle of the load into two components:

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longitudinal and lateral. The longitudinal component is suppressed by means of the conventional quay crane control method (i.e., via trolley movement). The lateral component, however, cannot be suppressed in this way. Hence, this paper proposes a new mechanism and a new lateral sway control strategy, namely a sliding-mode control [37]–[44] that combines ship dynamics, trolley dynamics, and sway dynamics. A component of this control method is a sliding surface that accounts for the lateral sway, the longitudinal sway angle, and the position/velocity errors of the trolley. A stability analysis and simulations are performed to prove the asymptotic stability of the closed-loop system.

This paper represents the inaugural work in each of what might be called the subfields (mechanism, equations of motion, and control strategy) of mobile harbor studies. Here, for the first time, the necessity of a new mechanism for the mobile harbor is identified, and treated, from a control point of view. (Indeed, the load's lateral sway caused by the pitching motion of a ship cannot be suppressed with a conventional quay crane.) The new mechanism proposed for lateral sway control is an important contribution, and the designed sliding-mode control combining ship dynamics, trolley dynamics, and sway dynamics is also novel. Here too, the asymptotic stability of the proposed sliding-mode control can be considered to be established.

The paper is organized as follows. In Section II, the system dynamics of an offshore container crane are derived, and a lateral sway control problem is formulated. In Section III, a sliding surface is proposed, and a corresponding sliding-mode control law, as formulated, is introduced; additionally, the asymptotic stability of the closed-loop system is proved. In Section IV, simulation results for open-loop control without ship motions, open-loop control with ship motions, and sliding-mode control with ship motions, are discussed. Finally, conclusions are drawn in Section V.

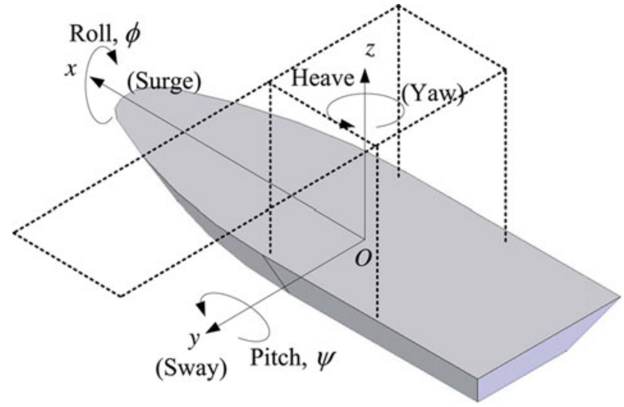


Fig. 2. Considered motions of mobile harbor: heave, roll, and pitch.

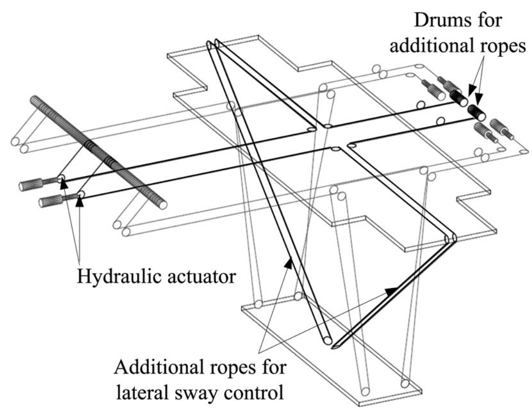


Fig. 3. Proposed new mechanism for lateral sway control of load (patented) [45].

II. Dynamic Model of Offshore Container Crane

Fig. 1 depicts an offshore container crane installed on a mobile harbor that is moored to a mother ship. The relative motions between the mother ship and the mobile harbor due to waves are critical. The mother ship nonetheless is assumed to be stationary in the ocean; thanks to its huge size, its rotational maneuvering is not significantly affected by sea waves. Moreover, the system mooring the mobile harbor to the mother ship will restrain the motions of the former. Therefore, only three mobile harbor motions are considered in this study: heave (up and down), roll, and pitch. Fig. 2 shows the considered three degree of freedom motions of the ship out of three translational motions (surge, sway, and heave) and three rotational motions (roll, pitch, and yaw).

For a container crane operating on the ground, it can be assumed that the trolley motions and the sway motions are coplanar. In this case, the trolley can be used to suppress the sway motion of the container (i.e., the conventional approach). But in the case of mobile harbors, there is an additional lateral sway motion of the container induced by the pitching motion of the ship. And this cannot be suppressed in the conventional way, because the lateral component of the sway angle and the trolley moving direction (i.e., the longitudinal direction) are not on the same plane. If the sway motion of the load is split into two components, longitudinal and lateral, the longitudinal component can be suppressed by manipulating the trolley movements; however, there is no direct way to suppress the lateral component. For mobile harbors then, a new mechanism and a new control method capable of eliminating both components simultaneously are necessary.

Fig. 3 shows this paper's proposed new mechanism for lateral sway control [45]. As can be seen, two additional ropes, with the necessary pulleys and drums, are added to the current mechanism. The drums are needed to compensate for the rope-length differences between the additional ropes and the main hoist ropes when the container is moved up and down. In those additional ropes, hydraulic actuators generate tension, through which lateral-sway-suppressing moment can be imparted.

A. Offshore Crane Dynamics

Fig. 4 depicts the three coordinate frames introduced to develop a mathematical model of the offshore crane: $O_0x_0y_0z_0$ denotes the reference coordinate frame (a stationary frame affixed to the mother ship); $O_sx_sy_sz_s$ is the ship coordinate frame attached to the center of gravity of the mobile harbor;

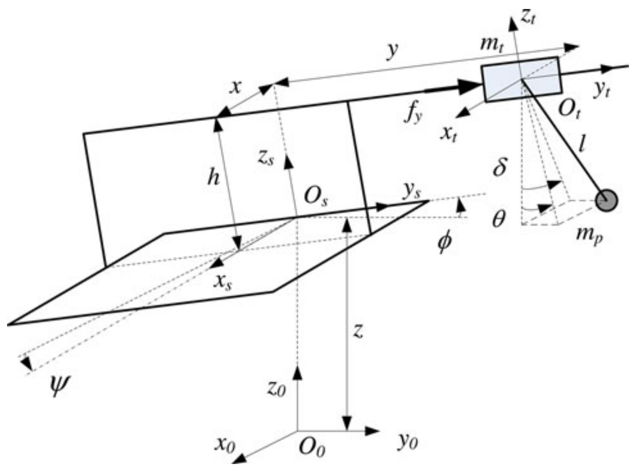


Fig. 4. Introduced coordinate frames: reference (mother ship), ship, and trolley.

$O_t x_t y_t z_t$ is the trolley coordinate frame affixed to the trolley. Let m_t and m_p be the masses of the trolley and the payload (container), respectively. Let h be the crane height. Let x and y represent the position of the gantry and that of the trolley in the ship coordinate frame. Let l denote the rope length, and let θ and δ define the longitudinal and lateral sway angles of the load in the reference coordinate frame. Practically, the sway angles are measured from the trolley: hence, θ and δ can be computed using the transformation matrix between the trolley and reference frames. But, in the interests of brevity, the coordinate transformations between individual coordinate frames will not be discussed in this paper. Finally, let f_y denote the control force applied at the trolley for longitudinal sway control. In order to simplify the modeling complexity, a number of assumptions are made: the individual ropes are a massless rigid rod; the friction in the trolley mechanism is ignored; the load is assumed to be a point mass.

Now, let z be the heave motion (displacement) of the ship in the reference coordinate frame; let ϕ and ψ be the rolling and pitching angular displacements of the ship, respectively, in the reference coordinate frame. Then, given the ship motions (z, ϕ, ψ), the trolley position p_t and the load position p_l in the reference coordinate frame are obtained as follows:

$$p_t = \begin{bmatrix} x\cos\psi + y\sin\psi\sin\phi + h\sin\psi\cos\phi \\ y\cos\phi - h\sin\phi \\ z - x\sin\psi + y\cos\psi\sin\phi + h\cos\psi\cos\phi \end{bmatrix} \quad (1)$$

$$p_l = \begin{bmatrix} x\cos\psi + y\sin\psi\sin\phi + h\sin\psi\cos\phi - l\cos\theta\sin\delta \\ y\cos\phi - h\sin\phi + l\sin\theta \\ z - x\sin\psi + y\cos\psi\sin\phi + h\cos\psi\cos\phi - l\cos\theta\cos\delta \end{bmatrix} \quad (2)$$

where the crane position x and the height h are considered constants, but for which there may be different values, depending on the actual load/unload processes. By differentiating (1) and (2) in time, the trolley velocity v_t and the load velocity v_l are given as

$$v_t = \begin{bmatrix} v_{tx} & v_{ty} & v_{tz} \end{bmatrix}^T \quad (3)$$

$$v_l = \begin{bmatrix} v_{lx} & v_{ly} & v_{lz} \end{bmatrix}^T \quad (4)$$

where

$$\begin{aligned} v_{tx} &= -x\dot{\psi}\sin\psi + \dot{y}\sin\psi\sin\phi + y\dot{\psi}\cos\psi\sin\phi \\ &\quad + y\dot{\phi}\sin\psi\cos\phi + h\dot{\psi}\cos\psi\cos\phi - h\dot{\phi}\sin\psi\sin\phi \\ v_{ty} &= \dot{y}\cos\phi - y\dot{\phi}\sin\phi - h\dot{\phi}\cos\phi \\ v_{tz} &= \dot{z} - x\dot{\psi}\cos\psi + \dot{y}\cos\psi\sin\phi - y\dot{\psi}\sin\psi\sin\phi \\ &\quad + y\dot{\phi}\cos\psi\cos\phi - h\dot{\psi}\sin\psi\cos\phi - h\dot{\phi}\cos\psi\sin\phi \\ v_{lx} &= -x\dot{\psi}\sin\psi + \dot{y}\sin\psi\sin\phi + y\dot{\psi}\cos\psi\sin\phi \\ &\quad + y\dot{\phi}\sin\psi\cos\phi + h\dot{\psi}\cos\psi\cos\phi - h\dot{\phi}\sin\psi\sin\phi \\ &\quad - \dot{l}\cos\theta\sin\delta + l\dot{\theta}\sin\theta\sin\delta - l\dot{\delta}\cos\theta\cos\delta \\ v_{ly} &= \dot{y}\cos\phi - y\dot{\phi}\sin\phi - h\dot{\phi}\cos\phi + \dot{l}\sin\theta + l\dot{\theta}\cos\theta \\ v_{lz} &= \dot{z} - x\dot{\psi}\cos\psi + \dot{y}\cos\psi\sin\phi - y\dot{\psi}\sin\psi\sin\phi \\ &\quad + y\dot{\phi}\cos\psi\cos\phi - h\dot{\psi}\sin\psi\cos\phi - h\dot{\phi}\cos\psi\sin\phi \\ &\quad - \dot{l}\cos\theta\cos\delta + l\dot{\theta}\sin\theta\cos\delta + l\dot{\delta}\cos\theta\sin\delta. \end{aligned}$$

The kinetic and potential energies of the trolley and load systems are obtained as follows:

$$T = \frac{1}{2}m_t\left(v_{tx}^2 + v_{ty}^2 + v_{tz}^2\right) + \frac{1}{2}m_p\left(v_{lx}^2 + v_{ly}^2 + v_{lz}^2\right) \quad (5)$$

$$\begin{aligned} U &= m_t g\left(z - x\sin\psi + y\cos\psi\sin\phi + h\cos\psi\cos\phi\right) \\ &\quad + m_p g\left(z - x\sin\psi + y\cos\psi\sin\phi + h\cos\psi\cos\phi\right) \\ &\quad - m_p g l\cos\theta\cos\delta \end{aligned} \quad (6)$$

where g is the gravitational acceleration. Note that the kinetic/potential energies of the ship are not included, because in this study, the ship motions (z, ϕ, ψ) are considered as disturbances. Finally, by applying the Lagrange equation, the equations of motion of the trolley and the load are obtained as follows:

$$\begin{aligned} &(m_t + m_p)\ddot{y} + m_p l\ddot{\theta}\left(\sin\phi\sin\theta\cos(\delta - \psi) + \cos\phi\cos\theta\right) \\ &\quad + m_p l\ddot{\delta}\sin\phi\cos\theta\sin(\delta - \psi) + c_1 = f_y \end{aligned} \quad (7)$$

$$m_p\left(\sin\phi\sin\theta\cos(\delta - \psi) + \cos\phi\cos\theta\right)\ddot{y} + m_p l\ddot{\theta} + c_2 = 0 \quad (8)$$

$$m_p l\ddot{y}\sin\phi\cos\theta\sin(\delta - \psi) + m_p l^2\ddot{\delta}\cos^2\theta + c_3 = \tau \quad (9)$$

where

$$\begin{aligned} c_1 &= (m_t + m_p)\left(-x\ddot{\psi}\sin\phi - h\ddot{\phi} - h\dot{\psi}^2\sin\phi\cos\phi\right) \\ &\quad + (m_t + m_p)(g + \ddot{z})\cos\psi\sin\phi \\ &\quad - (m_t + m_p)\left(\dot{\psi}^2\sin^2\phi + \dot{\phi}^2\right)y \\ &\quad + m_p\ddot{l}\cos\phi\sin\theta - m_p\ddot{l}\sin\phi\cos\theta\cos(\delta - \psi) \\ &\quad + 2m_p\dot{l}\dot{\delta}\sin\phi\cos\theta\sin(\delta - \psi) \end{aligned}$$

$$
\begin{aligned}
&+ 2m_p\dot{l}\dot{\theta}\sin\phi\sin\theta\cos(\delta-\psi) + 2m_p\dot{l}\dot{\theta}\cos\phi\cos\theta\\
&+ m_p l\left(\dot{\theta}^2+\dot{\delta}^2\right)\sin\phi\cos\theta\cos(\delta-\psi)\\
&- m_p l\dot{\theta}^2\cos\phi\sin\theta - 2m_p l\dot{\theta}\dot{\delta}\sin\phi\sin\theta\sin(\delta-\psi)
\end{aligned}
$$

$$
\begin{aligned}
c_2 = m_p \Big(&2\dot{l}\dot{\theta} + 2\dot{y}\dot{\psi}\sin\phi\sin\theta\sin(\delta-\psi)\\
&+ 2\dot{y}\dot{\phi}\cos\phi\sin\theta\cos(\delta-\psi) - 2\dot{y}\dot{\phi}\sin\phi\cos\theta\\
&+ 2y\dot{\psi}\dot{\phi}\cos\phi\sin\theta\sin(\delta-\psi) - y\dot{\phi}^2\cos\phi\cos\theta\\
&- y\dot{\psi}^2\sin\phi\sin\theta\cos(\delta-\psi) - y\ddot{\phi}\sin\phi\cos\theta\\
&- y\dot{\phi}^2\sin\phi\sin\theta\cos(\delta-\psi) + h\dot{\phi}^2\sin\phi\cos\theta\\
&+ y\ddot{\psi}\sin\phi\sin\theta\sin(\delta-\psi) - h\ddot{\phi}\cos\phi\cos\theta\\
&+ y\ddot{\phi}\cos\phi\sin\theta\cos(\delta-\psi) - x\dot{\psi}^2\sin\theta\sin(\delta-\psi)\\
&- x\ddot{\psi}\sin\theta\cos(\delta-\psi) + l\dot{\delta}^2\sin\theta\cos\theta\\
&- 2h\dot{\psi}\dot{\phi}\sin\phi\sin\theta\sin(\delta-\psi) - h\dot{\psi}^2\cos\phi\sin\theta\cos(\delta-\psi)\\
&- h\dot{\phi}^2\cos\phi\sin\theta\cos(\delta-\psi) + h\ddot{\psi}\cos\phi\sin\theta\sin(\delta-\psi)\\
&- h\ddot{\phi}\sin\phi\sin\theta\cos(\delta-\psi) + (g+\ddot{z})\sin\theta\cos\delta\Big)
\end{aligned}
$$

$$
\begin{aligned}
c_3 = m_p l\Big(&-2l\dot{\delta}\dot{\theta}\sin\theta\cos\theta + 2\dot{l}\dot{\delta}\cos^2\theta\\
&+ 2\dot{y}\dot{\phi}\cos\theta\cos\phi\sin(\delta-\psi) - 2\dot{y}\dot{\psi}\sin\phi\cos\theta\cos(\delta-\psi)\\
&- y\ddot{\psi}\sin\phi\cos\theta\cos(\delta-\psi) + y\ddot{\phi}\cos\phi\cos\theta\sin(\delta-\psi)\\
&- y\left(\dot{\phi}^2+\dot{\psi}^2\right)\sin\phi\cos\theta\sin(\delta-\psi)\\
&- 2y\dot{\psi}\dot{\phi}\cos\phi\cos\theta\cos(\delta-\psi)\\
&- h\ddot{\psi}\cos\phi\cos\theta\cos(\delta-\psi) - h\ddot{\phi}\sin\phi\cos\theta\sin(\delta-\psi)\\
&- h\left(\dot{\phi}^2+\dot{\psi}^2\right)\cos\phi\cos\theta\sin(\delta-\psi)\\
&+ 2h\dot{\phi}\dot{\psi}\sin\phi\cos\theta\cos(\delta-\psi) - x\ddot{\psi}\cos\theta\sin(\delta-\psi)\\
&+ x\dot{\psi}^2\cos\theta\cos(\delta-\psi) + (g+\ddot{z})\cos\theta\sin\delta\Big).
\end{aligned}
$$

B. Rope Tensions

Fig. 5 shows a side view of the mechanism for lateral sway control. Let a denote the distance from the spreader center to the pulley holding the rope at the spreader side, let b denote the distance from the trolley center to the pulley holding the rope at the trolley side, let f_1 and f_2 denote the rope tensions, and let τ be the control torque to be applied to the spreader for lateral sway suppression. The control torque τ is produced by adjusting the tensions in the additional ropes, f_1 and f_2. For a given lateral sway angle δ, the control torque τ is obtained as follows:

$$\tau = d_1 f_1 - d_2 f_2 \tag{10}$$

where d_1 and d_2 are the distances from the trolley center to the additional ropes, which are obtained as follows:

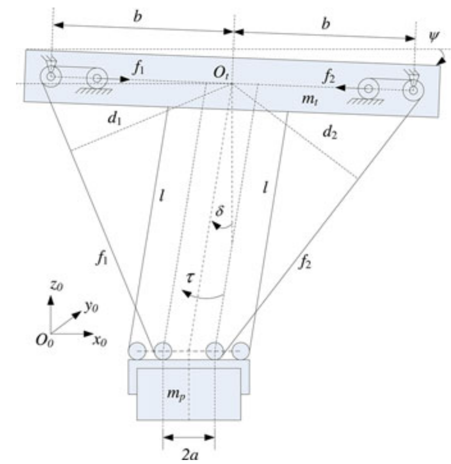


Fig. 5. Torque generation mechanism for lateral sway control.

$$d_1 = bl\cos(\psi-\delta)\Big/\sqrt{(b-a)^2 + l^2 + 2l(b-a)\sin(\psi-\delta)}$$

$$d_2 = bl\cos(\psi-\delta)\Big/\sqrt{(b-a)^2 + l^2 - 2l(b-a)\sin(\psi-\delta)}.$$

One important aspect of this scheme is the impossibility of imparting negative tension to the rope. Therefore, the following constraints need to be satisfied:

$$0 < f_{\min} \le f_i \le f_{\max}, \qquad i = 1, 2 \tag{11}$$

where $f_{\min}$ and $f_{\max}$ are the minimum and maximum tensions, respectively, allowed in the auxiliary ropes. Also, note that a positive value of $f_{\min}$ is imposed to keep the ropes taut and that $f_{\max}$ is smaller than the tension of the main hoisting rope. Let f_0 be the initial tension in each rope. Then, the proposed tension variations in both ropes are

$$
\begin{aligned}
f_1 &= f_0 + \Delta f\\
f_2 &= f_0 - \Delta f
\end{aligned} \tag{12}
$$

where Δf is the control input to the hydraulic actuators shown in Fig. 3.

Substituting (10) into (8), the control torque is rewritten as

$$\tau = (d_1 - d_2) f_0 + (d_1 + d_2)\Delta f. \tag{13}$$

As discussed in the next section, two control inputs (i.e., Δf for lateral sway and f_y for longitudinal sway) will be designed.

III. Sliding-Mode Antisway Control Design

Based on the developed crane dynamics discussed in the preceding section, a sliding-mode antisway control scheme is designed. First, an error vector $e = [e_y \quad e_\theta \quad e_\delta]^T$ consisting of the trolley position error and the sway errors is defined as

$$e = [e_y \quad e_\theta \quad e_\delta]^T = [y - y_d \quad \theta - \theta_d \quad \delta - \delta_d]^T \tag{14}$$

where y_d, θ_d, and δ_d are the desired trolley position and the desired sway angles (θ_d and δ_d are assumed to be zero), respectively. Without loss of generality, it can be assumed that the first and second time derivatives of the desired trolley position are bounded. Next, the sliding surface s is defined as

$$s = [s_y \quad s_\delta]^T = \left[\dot{e}_y + k_1 e_y - k_2\theta \quad \dot{\delta} + k_3\delta\right]^T \tag{15}$$

where k_1, k_2, and k_3 are positive control gains. Note that, in the first component s_y, the trolley motion and the longitudinal sway are coupled. In this way, the trolley dynamics can be incorporated with the longitudinal sway dynamics into a sliding surface. Suppose that s and $\dot{s}$ are controlled in such a way that $s, \dot{s} \to 0$ asymptotically as $t \to \infty$. Then, the control law guarantees that $e, \dot{e} \to 0$ asymptotically for all time $t \geq t_1$ with some finite constant t_1. This implies, crucially, that the asymptotic stability of the offshore container crane system can be achieved by proving the asymptotic stability of the sliding surface s.

In order to separate the θ-dynamics (unactuated dynamics) from the actuated dynamics (i.e., y-dynamics), (7) and (8) are combined together, eliminating $\ddot{\theta}$. Note that, by making $s, \dot{s} \to 0$, the θ-dynamics will go to zero as well. Using (9), the crane dynamics can be rewritten in the state space form as

$$\begin{bmatrix} m_t + m_p - m_p\left(\sin\phi\sin\theta\cos(\delta-\psi) + \cos\phi\cos\theta\right)^2 & m_p l\sin\phi\cos\theta\sin(\delta-\psi) \\ m_p l\sin\phi\cos\theta\sin(\delta-\psi) & m_p l^2\cos^2\theta \end{bmatrix}\begin{bmatrix}\ddot{y}\\ \ddot{\delta}\end{bmatrix} + \begin{bmatrix} c_1 - c_2\left(\sin\phi\sin\theta\cos(\delta-\psi) + \cos\phi\cos\theta\right) \\ c_3 \end{bmatrix} = \begin{bmatrix} f_y \\ \tau \end{bmatrix}. \tag{16}$$

The substitution of (13) into (16) yields the state equation in (y, δ)

$$M(q)\,\ddot{q} + W(q, \dot{q}) = u \tag{17}$$

where

$$\begin{aligned}
q &= [y \quad \delta]^T \\
M(q) &= \begin{bmatrix} m_{11} & m_{12} \\ m_{21} & m_{22}\end{bmatrix} \\
W(q,\dot{q}) &= [c_1 - c_2\left(\sin\phi\sin\theta\cos(\delta-\psi) + \cos\phi\cos\theta\right) \\
&\qquad \times c_3 - (d_1 - d_2)\, f_0/(d_1 + d_2)]^T \\
u &= [f_y \quad \Delta f]^T \\
m_{11} &= m_t + m_p - m_p\left(\sin\phi\sin\theta\cos(\delta-\psi) + \cos\phi\cos\theta\right)^2 \\
m_{12} &= m_p l\sin\phi\cos\theta\sin(\delta-\psi) \\
m_{21} &= m_p l\sin\phi\cos\theta\sin(\delta-\psi)/(d_1 + d_2) \\
m_{22} &= m_p l^2\cos^2\theta/(d_1 + d_2).
\end{aligned}$$

Now, to prove the asymptotic stability of the system, a positive definite function is considered

$$V(t) = \frac{1}{2}s^T s. \tag{18}$$

The time derivative of (18) using (14) and (15) becomes

$$\dot{V}(t) = s^T\dot{s} = s^T\left[\ddot{y} - \ddot{y}_d + k_1\dot{e}_y - k_2\dot{\theta} \quad \ddot{\delta} + k_3\dot{\delta}\right]^T. \tag{19}$$

Let $\ddot{r} = \left[\ddot{y}_d - k_1\dot{e}_y + k_2\dot{\theta} - k_3\dot{\delta}\right]^T$. Then, the time derivative of $\dot{V}(t)$ can be rewritten, using (17), as

$$\begin{aligned}\dot{V}(t) &= s^T(\ddot{q} - \ddot{r}) \\ &= s^T\left(M^{-1}(q)\left[-W(q,\dot{q}) + u\right] - \ddot{r}\right).\end{aligned} \tag{20}$$

From (20), the following sliding-mode control law is proposed to assure that $\dot{s} \to 0$:

$$u = M(q)\left[\ddot{r} - \mu\,\mathrm{sgn}(s)\right] + W(q,\dot{q}) \tag{21}$$

where $\mu\,\mathrm{sgn}(s) = [\mu_y\mathrm{sgn}(s_y) \quad \mu_\delta\mathrm{sgn}(s_\delta)]^T$, μ_y and μ_δ are positive constants. Substituting the control law (21) into (20), the time derivative of $\dot{V}(t)$ becomes

$$\dot{V}(t) = -\mu_y|s_y| - \mu_\delta|s_\delta|. \tag{22}$$

Equations (18) and (22) imply that s_y and s_δ are uniformly bounded and monotonically decreasing, and that it also converges asymptotically to zero as $t \to \infty$, with the sliding-mode control [37].

Theorem: Consider the offshore container crane system (7)–(9) with (13). Let $e = [y - y_d \quad \theta \quad \delta]^T$ and $q = [y \quad \delta]^T$. Then, the sliding-mode control law (21) guarantees that $e, \dot{e} \to 0$ asymptotically as $t \to \infty$.

Remark: To eliminate the chattering phenomenon induced by the term $\mathrm{sgn}(s)$ in control law (21), a saturation function $\mathrm{sat}(s,\lambda)$ is defined as follows:

$$\mathrm{sat}(s,\lambda) = \begin{cases} s/\lambda, & \text{if } |s| \leq \lambda \\ \mathrm{sgn}(s,\lambda), & \text{if } |s| > \lambda. \end{cases} \tag{23}$$

Then, (23) can make the control signals smooth in a thin boundary layer of thickness λ neighboring the sliding surface [37].

IV. Simulation Results

Table I lists the parameter values used in the simulation. The trolley was assumed to move to a desired position ($y_d = 10$ m). Fig. 6 shows the simulation results for the sway motions of the load (without control) when the mobile harbor is stationary (i.e., $\phi = 0$ and $\psi = 0$). As was done in a study on conventional quay crane control [8], only the trolley velocity was controlled to follow a trapezoidal velocity profile (in the conventional quay crane setting, this is an open-loop control under a constant rope length). Under such an open control (in our case, full nonlinear dynamics was used except $\phi = 0$ and $\psi = 0$), the trolley reached the goal position in 10 s without any residual sway (either longitudinal or lateral). Fig. 6 results, thus, validate the correctness of the derived equations (7)–(9). Subsequently, the same control strategy was applied to an offshore container crane and the attendant ship motions (i.e., $\phi \neq 0$ and/or $\psi \neq 0$) (see Fig. 7). When the same trolley velocity profile was applied under the condition $\phi = 0.02\sin(1.25t)$ and $\psi = 0.02\sin(1.25t)$, the trolley reached its target position in 10 s as expected, but both longitudinal and lateral sway were evident. Even if $\delta(0)$ were zero, the lateral sway would be excited by the pitching motion of the ship. To indicate the long-term behavior, the simulation time in Fig. 7(b) was extended to 50 s.

TABLE I
PARAMETER VALUES USED IN SIMULATION

Parameters	Values
Rolling motion (angle)	$\phi = 0.02\sin(1.25t)$ rad
Pitching motion (angle)	$\psi = 0.01\sin(1.25t)$ rad
Heaving motion (angle)	$z = 0.02\sin(1.25t)$ m
Crane height	$h = 10$ m
Rope length	$l = 8$ m
Trolley mass	$m_t = 6000$ kg
Load	$m_p = 20000$ kg
Gantry position	$x = 5$ m
Pulleys position	$b = 4$ m, $a = 0.5$ m
Initial tension	$f_0 = 8000$ N
Control gains of the sliding mode control	$k_1 = 0.2, k_2 = 19, k_3 = 1,$ $\mu_y = 23, \mu_\delta = 0.5, \lambda = 0.01$
Control gains of the PD control	$k_{p1} = 5\times10^3, k_{d1} = 21\times10^3, k_a = 5\times10^4,$ $k_{p2} = 10^6, k_{d2} = 10^6$

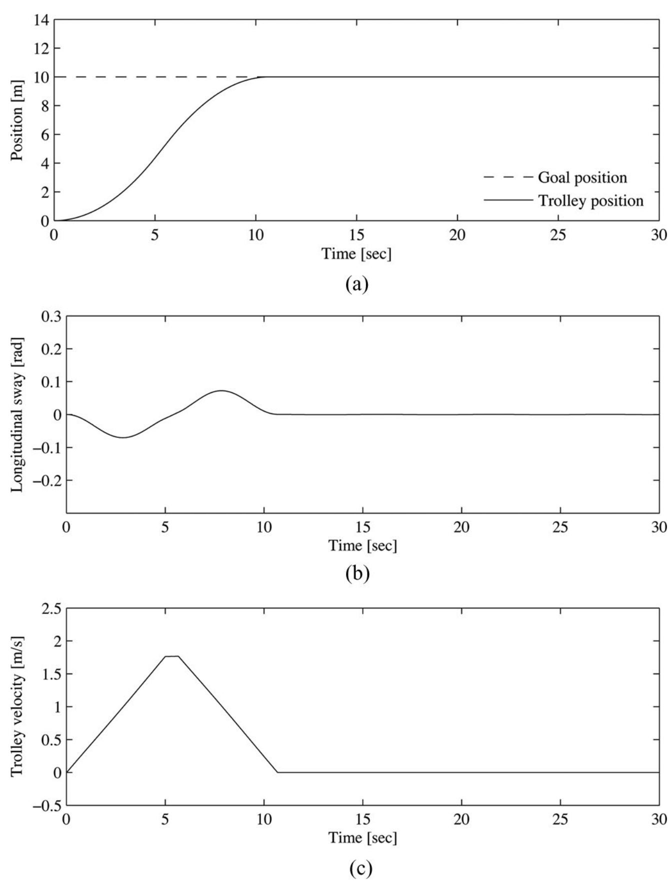


Fig. 6. Trolley movement and sway motions of load when mobile harbor is stationary (i.e., $\phi = 0$ and $\psi = 0$): open loop control. (a) Trolley movement. (b) Longitudinal sway. (c) Trolley velocity (trapezoidal velocity profile was used so that residual sway at end of trolley movement was zero).

Next, the proposed sliding-mode control was examined. First, separate control of rolling and pitching motions was performed, followed by simultaneous control of both roll and pitch. Fig. 8 shows the sliding-mode control in the presence of a pitching motion of the ship, $\psi = 0.02\sin(1.25t)$, and an initial lateral sway angle of $\delta(0) = -0.1$ rad. In this case, the travel time of the trolley was increased to 12.5 s (a 25% increase), but the lateral sway was suppressed before the trolley reached its goal position. Fig. 9 shows the control performance in the presence of only a rolling motion of the ship, $\phi = 0.02\sin(1.25t)$, and an initial sway angle of $\delta(0) = -0.1$ rad. In this case, the lateral sway was perfectly eliminated; however, the longitudinal sway was not (this difference is discussed in the next paragraph). A similar phenomenon was evident in the case where there were both rolling and pitching motions (see Fig. 10).

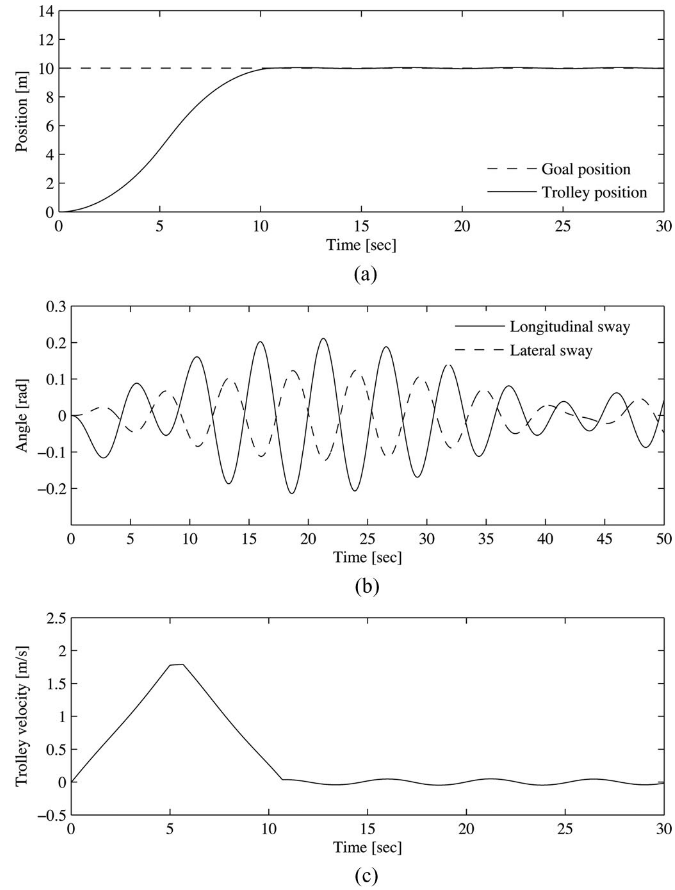
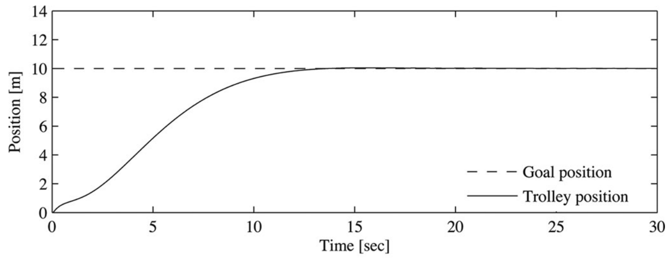


Fig. 7. Trolley movement and sway motions of load when $\phi = 0.02\sin(1.25t)$ and $\psi = 0.02\sin(1.25t)$ (without control, the same velocity profile as in Fig. 6 was applied). (a) Trolley movement. (b) Sway motions. (c) Trolley velocity (due to sway of load after trolley reaches goal position, small movement yet evident).

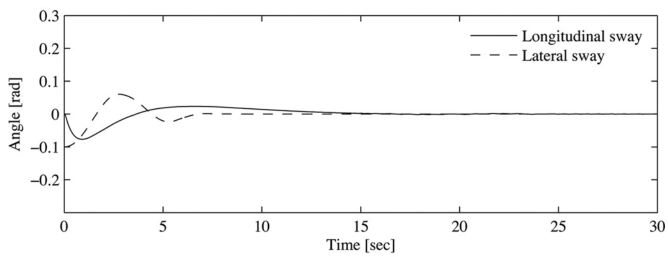
A well-tuned PD control law was also used for verification. The conventional PD control law is given by

$$f_y = -k_{p1}e_y - k_{d1}\dot{e}_y + k_a\theta$$
$$\Delta f = -k_{p2}\delta - k_{d2}\dot{\delta} \quad (24)$$

where $k_{p1}, k_{p2}, k_{d1}, k_{d2}$, and k_a are the proportional and derivative gains associated with the trolley position and the sway angle in suppressing the vibrations of the load. After numerous trials, the parameters of the PD controller are set as shown in Table I. There was no difference between the PD control law and the proposed control law in the presence of a pitching motion of the mobile harbor, $\psi = 0.02\sin(1.25t)$, and an initial lateral sway angle of $\delta(0) = -0.1$ rad as shown in Figs. 8 and 11. However,

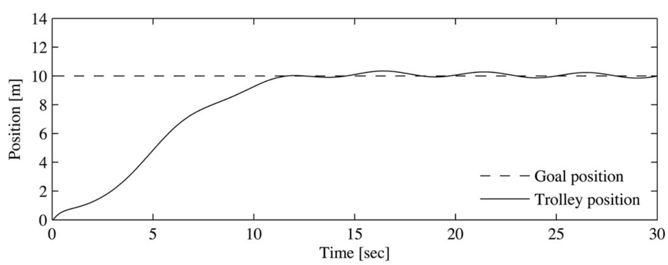


(a)

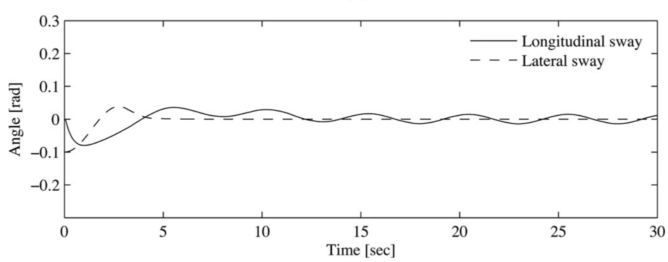


(b)

Fig. 8. Sliding-mode control in the presence only of pitch motion of mobile harbor: $\phi = 0, \psi = 0.02\sin(1.25t)$, and $\delta(0) = -0.1$ rad. (a) Trolley movement. (b) Sway motions.

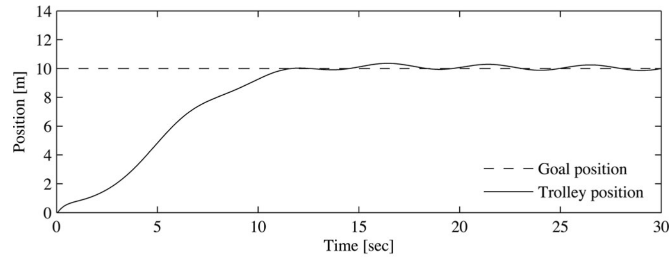


(a)

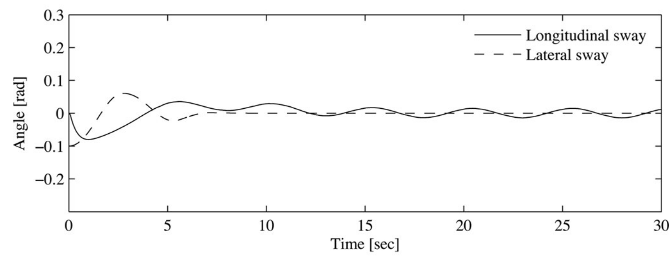


(b)

Fig. 9. Sliding-mode control in presence only of roll motion of mobile harbor: $\phi = 0.02\sin(1.25t)$, $\psi = 0$, and $\delta(0) = -0.1$ rad. (a) Trolley movement. (b) Sway motions.

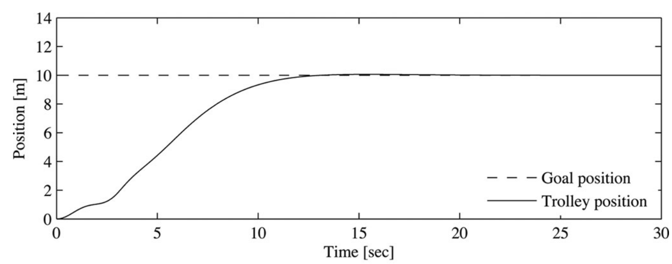


(a)

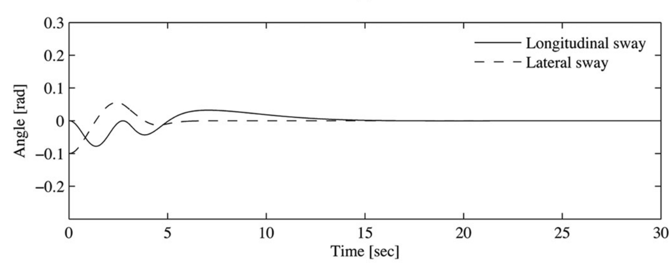


(b)

Fig. 10. Sliding-mode control in the presence of both roll and pitch motions of mobile harbor: $\phi = 0.02\sin(1.25t)$, $\psi = 0.02\sin(1.25t)$, and $\delta(0) = -0.1$ rad. (a) Trolley movement. (b) Sway motions.



(a)



(b)

Fig. 11. PD control in the presence of only pitch motion of mobile harbor: $\phi = 0$, $\psi = 0.02\sin(1.25t)$, and $\delta(0) = -0.1$ rad. (a) Trolley movement. (b) Sway motions.

the PD control law could not guarantee the trolley position as well as the longitudinal sway angle in the presence of both roll and pitch motions of the mobile harbor while the lateral sway was suppressed immediately as shown in Fig. 12.

As shown in Figs. 8–10, the lateral sway can be suppressed perfectly but the longitudinal sway cannot. This is due to the difference between the two sway control mechanisms. For lateral sway control, a force from the hydraulic actuators is directly applied (as a moment) to the spreader through the additional ropes, and therefore, the lateral sway angle resulting from the ship motions can be suppressed. However, in the longitudinal direction, the control force f_y is applied to the trolley, and the trolley motions are used to suppress the longitudinal sway angle. The residual sway that is evident even after the trolley reaches the target position is due to the persistent rolling motion of the mobile harbor. The sliding-mode controller, in attempting to compensate for the occurrence of $\theta(t) \neq 0$, generates small trolley motions and maintains some residual longitudinal sway angles. A solution to this problem is to develop a mooring system that minimizes the relative rolling motion of the mobile harbor from the mother ship while allowing some level of relative pitching motion between them.

In summation, a ship's pitching motion can be compensated for by directly applying a torque to the spreader, which is an

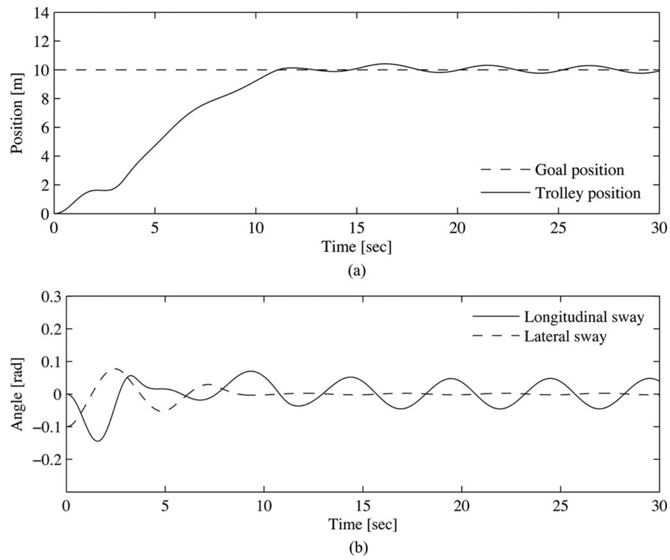


Fig. 12. PD control in the presence of both roll and pitch motions of mobile harbor: $\phi = 0.02\sin(1.25t)$, $\psi = 0.02\sin(1.25t)$, and $\delta(0) = -0.1$ rad. (a) Trolley movement. (b) Sway motions.

important aspect of the new mechanism introduced in this paper; however, rolling motion cannot be compensated for.

V. Conclusion

A mobile harbor having an open-sea container loading/unloading capability was introduced. The sway control was revealed as the key technology to be tackled. The splitting of the sway angle into two components (longitudinal and lateral) was discussed. For lateral sway control, an application of direct torque to the spreader was exploited, whereas for longitudinal sway control, the conventional trolley system was used. A new mechanism for lateral sway control was proposed. By combining the longitudinal dynamics and the trolley motions, a new sliding surface was designed. The proposed sliding-mode control demonstrated a perfect cancellation of the lateral sway with the proposed new lateral sway control mechanism. However, for perfect suppression of the longitudinal sway angle, a new mother ship/mobile harbor mooring system that restricts the rolling motion of the mobile harbor is required. Another possible solution is a hypothetical new feedforward control strategy that can compensate for the rolling motion of the mobile harbor after the trolley reaches its goal position.

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