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Adaptive sliding mode control of container cranes

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Abstract: An adaptive sliding mode control scheme for container cranes is investigated in this study. A sliding surface is designed in such a way that the sway motion of the payload is incorporated into the trolley dynamics. Included in the proposed control law is a varying control gain, obtained by an adaptation law, which transitions the system into the sliding mode. The control law guarantees the asymptotic stability of the closed-loop system. To demonstrate the efficiency of the proposed algorithm, experimental results are provided.

1 Introduction

Container cranes (or quay cranes) are used for vessel-to-truck and truck-to-vessel loading and unloading of containers at container terminals. Container cranes consist of a supporting structure called the gantry that can traverse the length of a quay or yard, a trolley and a moving gripper called a spreader. The trolley runs along rails, which are located on the top or sides of the boom, transferring containers to or from the ship. The spreader can be lowered on top of a container, and locks on to the container's four locking points by means of twist-lock mechanisms. The container is then lifted and transferred onto a truck, which takes the container to a storage yard. The crane also unloads containers from the truck and transfers them to the ship. Cranes normally transport single containers; however, some new cranes are capable of loading/unloading up to four 20 ft containers at once.

Since the movement of the trolley sways the container during transport, the main issue in container crane control is quick suppression of the vibrations caused by trolley motions at the trolley's goal position. Additionally, a residual sway, because of crane dynamics and disturbances such as winds, occurs at the end of the trolley's movement. Thus, researchers working in the area of container crane control always have been obliged to deal with sway suppression.

Container oscillation and the obligation to suppress it, in fact, constitute a transportation bottleneck headache for container terminal officials. Although the 'container' crane is a special case of many types of cranes, early crane control results are still relevant to the present concern. When the rope length is constant, it is known that the input (or command) shaping control is very effective [1–10]. In their studies, the reference signal was modified and implemented in real time, where modification depends on the natural frequency of the system. If the considered system has an uncertain parameter, control design based on the input shaping technique may not fulfil the goal. Besides, the input shaping cannot be used for disturbance rejection, and it should be used in conjunction with a feedback control to reject disturbances. Time optimal control is another frequently used control method suitable for crane control [11, 12]. Time optimal control, similarly to input shaping control, requires exact system information. Both optimal control and input shaping techniques are limited by the fact that they are extremely sensitive to variations in nominal parameters, changes in initial conditions and external disturbances. In short, these methods require highly accurate system parameter values to achieve satisfactory system responses.

Linear control methods and algorithms have been applied to crane systems [13–18]. However, these algorithms cannot be implemented to real systems without tuning their control gains. Under modelling uncertainties and measurement errors, gain tuning is not an easy task. Additionally, varying rope length and friction can present challenges to gain tuning. Since friction and disturbance cannot be precisely represented in models, they can have strongly adverse effects on those systems. Many approaches to tackle the uncertainties in crane systems have been proposed, among which are sliding mode control (SMC) [19–29], fuzzy control [30] and adaptive control [31–34]. Also, by considering a crane system as an axially moving string system, researchers have designed boundary control laws to suppress transverse vibrations [35–38]. Moreover, some conditions for asymptotical tracking must be satisfied if the crane system is considered as an under-actuated control system [39, 40].

In the present study, an adaptive SMC algorithm [41] based on a non-linear container crane model [42] is designed. According to the control scheme, the trolley position and the sway angle are incorporated into a sliding surface. A varying control gain, as obtained by an adaptation law, is the key feature in the proposed control law, transitioning the system into the sliding mode. The control gain increases to high gains until the sliding surface reaches the sliding mode, at which time the gain is switched to a low value to avoid chattering. The stability of the proposed method is also answered.

The paper is organised as follows. In Section 2, the system dynamics of a container crane are derived. In Section 3, the SMC law and the adaptation law for control gains are proposed, and the system stability is analysed. In Section 4, simulation and experiment results of the closed-loop system are discussed. Finally, in Section 5, conclusions are drawn.

2 Dynamic model of container crane

Consider the container crane illustrated in Fig. 1. The container (payload) is picked up by the spreader, both being suspended from the trolley by a rope of length l. The masses of the trolley and the payload are m_t and m_p, respectively. A control force f_x is applied to the trolley. In an actual crane, four ropes are used to hoist the spreader (including the payload). However, for simplicity, only one rope is assumed in the present paper. It is also assumed that the motions of both the spreader and the rope occur in the vertical plane, that is, the X–Y-plane (see Fig. 1). Let x be the trolley position along the X-axis, θ be the sway angle and g be the gravitational acceleration.

Considering the motions of the trolley and the payload in the two-dimensional (2D) plane, the kinetic energy T and potential energy U of the entire system are given by

$$\begin{aligned} T &= \frac{1}{2}m_t\dot{x}^2 + \frac{1}{2}m_p[(\dot{l}\cos\theta - l\dot{\theta}\sin\theta)^2 \\ &\quad + (\dot{x} + \dot{l}\sin\theta + l\dot{\theta}\cos\theta)^2] \\ &= \frac{1}{2}(m_t + m_p)\dot{x}^2 + \frac{1}{2}m_p\dot{l}^2 + \frac{1}{2}m_p l^2\dot{\theta}^2 \\ &\quad + m_p\dot{l}\dot{x}\sin\theta + m_p l\dot{x}\dot{\theta}\cos\theta \end{aligned} \tag{1}$$

$$U = m_p g l(1 - \cos\theta) \tag{2}$$

Taking $\boldsymbol{q} = (x, \theta)$ as the generalised coordinates corresponding to the generalised forces $\boldsymbol{f} = (f_x, 0)$, and using Lagrange's equation

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial q_i} = f_i, i = 1, 2 \tag{3}$$

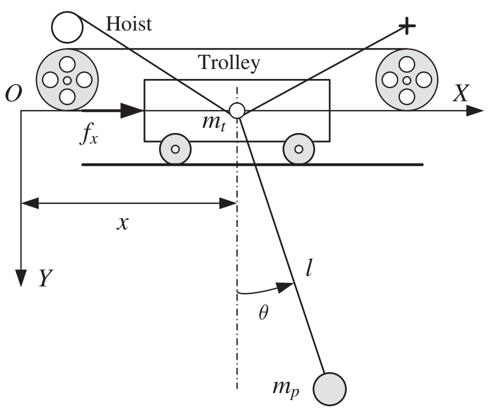

Fig. 1 *Container crane model*

the equations of motion can be obtained as

$$\begin{aligned} &(m_t + m_p)\ddot{x} + m_p\ddot{l}\sin\theta + m_p l\ddot{\theta}\cos\theta + 2m_p\dot{l}\dot{\theta}\cos\theta \\ &\quad - m_p l\dot{\theta}^2\sin\theta = f_x \end{aligned} \tag{4}$$

$$m_p l\ddot{x}\cos\theta + m_p l^2\ddot{\theta} + 2m_p l\dot{l}\dot{\theta} + m_p g l\sin\theta = 0 \tag{5}$$

The container crane equations of (4) and (5) are rewritten as

$$\ddot{x} = h_1(\dot{\theta}, \theta) + g_1(\theta)f_x \tag{6}$$

$$\ddot{\theta} = h_2(\dot{\theta}, \theta) + g_2(\theta)f_x \tag{7}$$

where

$$h_1(\dot{\theta}, \theta) = \frac{m_p g\sin\theta\cos\theta - m_p\ddot{l}\sin\theta + m_p l\dot{\theta}^2\sin\theta}{m_t + m_p\sin^2\theta}$$

$$h_2(\dot{\theta}, \theta) = \frac{-(m_p + m_t)g\sin\theta + m_p\ddot{l}\sin\theta\cos\theta - m_p l\dot{\theta}^2\sin\theta\cos\theta}{m_t l + m_p l\sin^2\theta} - \frac{2\dot{l}\dot{\theta}}{l}$$

$$g_1(\theta) = \frac{1}{m_t + m_p\sin^2\theta}$$

$$g_2(\theta) = -\frac{\cos\theta}{m_t l + m_p l\sin^2\theta}$$

3 Adaptive SMC

3.1 Control law design

In this section, a sliding mode anti-sway control scheme is designed. First, an error vector $\boldsymbol{e} = [e_x \ \ e_\theta]^{\mathrm{T}}$ consisting of the trolley position error and the sway angle error is defined as

$$\boldsymbol{e} = [e_x \quad e_\theta]^{\mathrm{T}} = [x - x_d \quad \theta - \theta_d]^{\mathrm{T}} \tag{8}$$

where x_d and θ_d are the trolley goal position and the desired sway angle (θ_d is assumed to be zero), respectively. Without loss of generality, it can be assumed that the first and second time derivatives of the trolley goal position are bounded. Additionally, it can be assumed that θ is not close to $\pi/2$, and, to avoid zero rope length, l is not equal to zero. Next, the sliding surface s is defined as

$$s = \dot{e}_x + k_1 e_x - k_2\theta \tag{9}$$

where k_1 and k_2 are positive control gains. Note that both the trolley and sway dynamics are incorporated into the sliding surface. Finally, the following adaptive SMC law is proposed

$$f_x = -\frac{h_1(\dot{\theta}, \theta) - \ddot{r} + \hat{k}(t)\mathrm{sat}(s)}{g_1(\theta)} \tag{10}$$

where $\ddot{r} = \ddot{x}_d - k_1\dot{e}_x + k_2\dot{\theta}$, and the gain $\hat{k}(t)$ is defined as follows:

- If $|s| > \varepsilon > 0$ (where ε is a small positive constant), $\hat{k}(t)$ is tuned as

$$\dot{\hat{k}}(t) = \gamma|s| \tag{11}$$

where $\gamma > 0$, and $\hat{k}(0) > 0$.

- If $|s| \leq \varepsilon$, $\hat{k}(t)$ is given by

$$\hat{k}(t) = \lambda|\eta| \tag{12}$$

where $\lambda > 0$ and η is the average of sgn(s) obtained through a low-pass filter [41] as follows

$$\sigma\dot{\eta} + \eta = \text{sgn}(s), \quad \sigma > 0 \tag{13}$$

The saturation function sat(s) is also defined as follows [43]

$$\text{sat}(s) = \begin{cases} s/\varepsilon, & \text{if } |s| \leq \varepsilon \\ sgn(s), & \text{if } |s| > \varepsilon \end{cases} \tag{14}$$

Remark: The objective of SMC is to bring the sliding surface s to zero. However, the adaptive SMC here is to design a control law for practical implementation. Hence, it is impossible to reach $s = 0$, because of sampled data and noisy measurement. Therefore the control law (10) only guarantees that the sliding surface reaches a zero region, which is bounded by a small positive constant ε. Without loss of generality (as far as the stability analysis is concerned), the sliding surface is assumed to be zero when s reaches the zero region.

3.2 Stability analysis

Theorem: Consider the crane systems (6) and (7) with control laws (10)–(14). Then, (i) the control gain $\hat{k}(t)$ in (11)–(14) is upper bound, that is, there exists a positive constant K such that $\hat{k}(t) \leq K, \forall t > 0$; (ii) s and $\dot{s}$ asymptotically converge to zero as $t \to \infty$. As a result, the position error e_x and the sway angle θ also converge asymptotically to zero as $t \to \infty$.

Proof: First, the boundedness of $\hat{k}(t)$ is proved. The differentiation of the sliding surface (9) yields

$$\dot{s} = \ddot{x} - \ddot{x}_d + k_1\dot{e}_x - k_2\dot{\theta} \tag{15}$$

The substitution of (6) and (10) into (15) yields

$$\dot{s} = -\hat{k}(t)\text{sat}(s) \tag{16}$$

Suppose that $|s| > \varepsilon > 0$. From (11) and (14), the gain $\hat{k}(t)$ will increase. Also, if $s < -\varepsilon$, the derivative of s is greater than zero, which implies that the sliding surface increases. If $s > \varepsilon$, $\dot{s} < 0$, which means that s is decreasing. Therefore (16) implies that the absolute value of the sliding surface $|s|$ decreases when the gain $\hat{k}(t)$ increases. If $|s|$ decreases until $|s| \leq \varepsilon$, the gain $\hat{k}(t)$ is obtained by (12)–(13) which is bounded because of the boundedness of η. Therefore there always exists a positive constant K such that $\hat{k}(t) \leq K$ for all $t \geq 0$ (the boundedness is graphically illustrated in Fig. 2. Also, this behaviour will be verified in experiment; see Figs. 9 and 10).

Second, the convergence of s and $\dot{s}$ is proved. Suppose that $|s| > \varepsilon > 0$. A Lyapunov function candidate

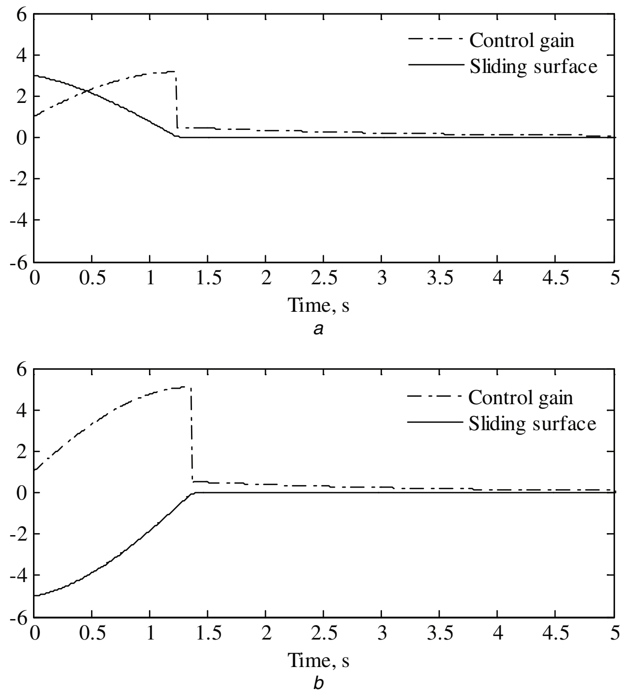

Fig. 2 *Relationship between the control gain $\hat{k}(t)$ and the sliding surface s*

a When $s(0) > 0$
b When $s(0) < 0$

is introduced as

$$V = \frac{1}{2}s^2 + \frac{1}{2\gamma}(\hat{k} - K)^2 \tag{17}$$

where γ and K are positive constants. The differentiation of (17) yields

$$\dot{V} = s\dot{s} + \frac{1}{\gamma}(\hat{k} - K)\dot{\hat{k}} \tag{18}$$

The substitution of (11), (14) and (16) into (18) yields

$$\dot{V} = -K|s| \leq 0 \tag{19}$$

The first derivative is a semi-negative function. The second derivative of the function, that is, $\ddot{V} = -K|\hat{k}(t)\text{sat}(s)|$, is bounded, because $\hat{k}(t)$ is bounded. Application of Barbalat's lemma indicates that $s \to 0$ as $t \to \infty$. Owing to (14) and (16), $\dot{s} \to 0$ as $s \to 0$.

Finally, the stability of the load dynamics is considered. The dynamics in the sliding mode provides that $s = 0$ and $\dot{s} = 0$. Hence, the following holds

$$s = \dot{e}_x + k_1e_x - k_2\theta = 0 \tag{20}$$

and

$$\dot{s} = \ddot{x} - \ddot{x}_d + k_1\dot{e}_x - k_2\dot{\theta} = 0 \tag{21}$$

The first-order differential (20) provides the solution as

$$e_x = e_x(0)e^{-k_1t} + k_2\int_0^t \theta(\tau)e^{k_1(\tau-t)}\,\mathrm{d}\tau \tag{22}$$

Substituting (22) into (21), the acceleration of the trolley is obtained as

$$\ddot{x} = \ddot{x}_d + k_1^2e_x(0)e^{-k_1t} + k_1^2k_2\int_0^t \theta(\tau)e^{k_1(\tau-t)}\mathrm{d}\tau - k_1k_2\theta + k_2\dot{\theta} \tag{23}$$

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The load dynamics (5) with the acceleration of the trolley in (23) and a constant goal position becomes

$$l\ddot{\theta} = -(k_2\cos\theta + 2\dot{l})\dot{\theta} - k_1^2\cos\theta\, e_x(0)e^{-k_1 t} - g\sin\theta - k_1k_2\cos\theta\left(\theta - k_1\int_0^t \theta(\tau)e^{k_1(\tau - t)}\,\mathrm{d}\tau\right) \quad (24)$$

Equation (24) shows that $k_1k_2\cos\theta\left(\theta - k_1\int_0^t\theta(\tau)\,e^{k_1(\tau-t)}\,\mathrm{d}\tau\right)$ is much smaller in magnitude than $g\sin\theta$, since $k_1k_2 \ll g$. Without loss of generality, this term can be omitted from (24), which yields

$$l\ddot{\theta} = -(k_2\cos\theta + 2\dot{l})\dot{\theta} - k_1^2\cos\theta\, e_x(0)\,e^{-k_1 t} - g\sin\theta \quad (25)$$

For proving the stability of the load dynamics (25), the non-negative function V_θ is introduced as

$$V_\theta = \frac{1}{2}l\dot{\theta}^2 + g(1 - \cos\theta) \quad (26)$$

The time derivative of V_θ yields

$$\dot{V}_\theta = l\dot{\theta}\ddot{\theta} + \frac{1}{2}\dot{l}\dot{\theta}^2 + g\sin\theta\dot{\theta} = -(k_2\cos\theta - 1.5\dot{l})\dot{\theta}^2 - k_1^2\dot{\theta}\cos\theta\, e_x(0)\,e^{-k_1 t} \quad (27)$$

A positive control gain k_2 can be chosen to satisfy the condition $k_2\cos\theta > 1.5|\dot{l}|$. Now, suppose that $\dot{\theta}$ does not approach zero asymptotically as $t \to \infty$, and that the function $\dot{V}_\theta$ in (27) will be negative at some finite time t_0, which implies that $\theta, \dot{\theta} \to 0$ asymptotically for all $t > t_0$. Since $s, \dot{s} \to 0$ and $\theta, \dot{\theta} \to 0$, (9) assures that $e_x, \dot{e}_x \to 0$. □

4 Simulation and experimental verification

4.1 System robustness

The robustness of the designed control system is tested through simulations. Equations (6) and (7) are rewritten as follows

$$\ddot{x} = h_1(\dot{\theta}, \theta) + \Delta h_1(\dot{\theta}, \theta) + (g_1(\theta) + \Delta g_1(\theta))f_x \quad (28)$$

$$\ddot{\theta} = h_2(\dot{\theta}, \theta) + \Delta h_2(\dot{\theta}, \theta) + (g_2(\theta) + \Delta g_2(\theta))f_x \quad (29)$$

where $\Delta h_1(\dot{\theta}, \theta), \Delta h_2(\dot{\theta}, \theta), \Delta g_1(\theta)$ and $\Delta g_2(\theta)$ represent modelling uncertainties. In simulations, the used system parameters are $l = 0.3$ m, $m_t = 1.67$ kg and $m_p = 0.73$ kg. The uncertain functions Δh_1, Δh_2, Δg_1 and Δg_2 are set to have ±20% discrepancy from the nominal values. However, the proposed control law (10) was designed by using the nominal parameters. The results are shown in Fig. 3. Although there are some differences in control performances, the proposed control law (10) assures the stability of the system well.

4.2 Control performances with different control gains

The selection of the control gains in (9) must satisfy the stability condition, $k_1k_2 \ll g$ and $k_2\cos\theta > 1.5|\dot{l}|$. The set of (k_1, k_2) is neither empty nor unique. The selection of control gains will affect the control performances as shown in Fig. 4. When k_1 and k_2 are small, the sliding surface s in

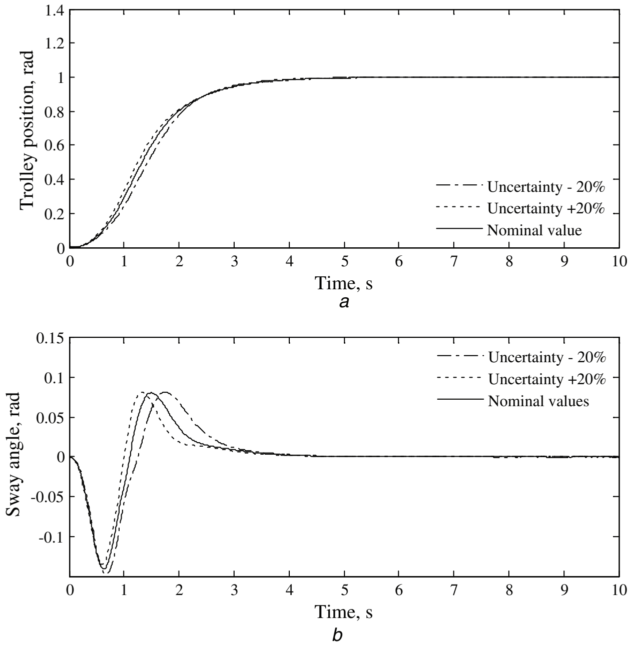

Fig. 3 *Robustness verification*

a Trolley position
b Sway angle

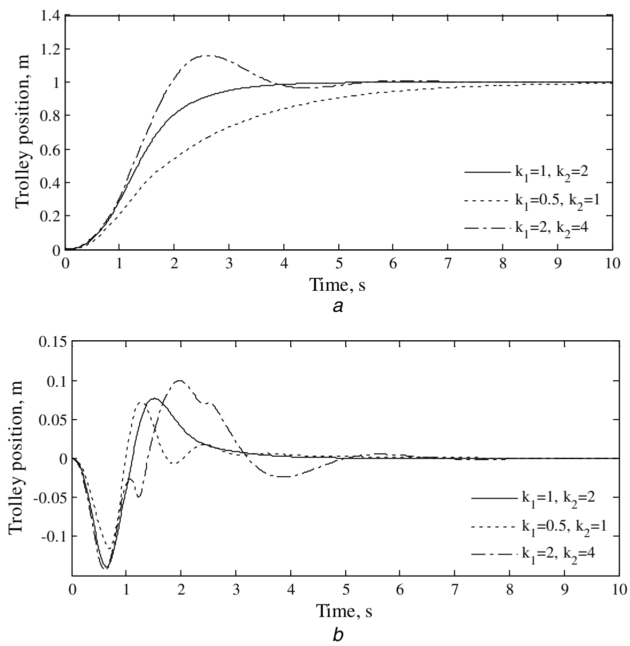

Fig. 4 *Control performances with different control gains*

a Trolley position
b Sway angle

(9) is also small. Therefore the increase of $\hat{k}(t)$ in (11) (to the values that provide fast system responses) may not be fast enough. However, the increase of k_1 and k_2 beyond the stability condition will make the system unstable.

4.3 Experimental results

A 3D crane from Inteco Company (www.inteco.com.pl) was used in experiments. The rope length is set to $l = 0.5$ m and the nominal masses were $m_t = 1.67$ kg and $m_p = 0.73$ kg. The experiment was performed for trolley movement from

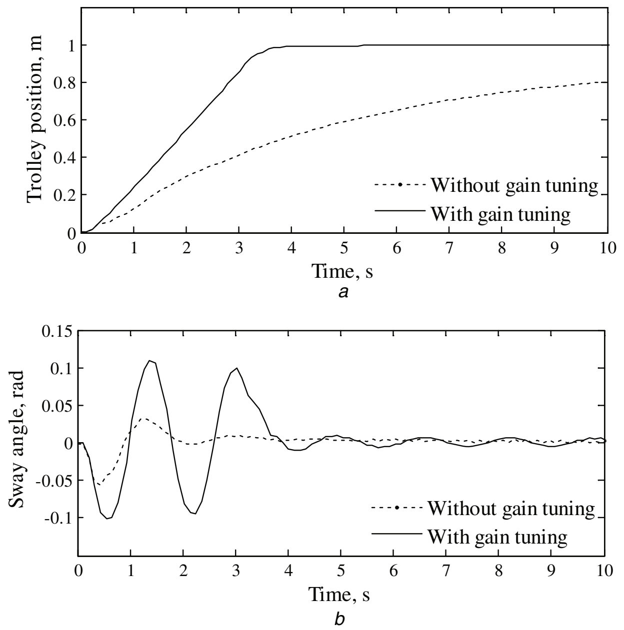

Fig. 5 *Comparison of two linear controls: with and without gain tuning*

a Trolley position
b Sway angle

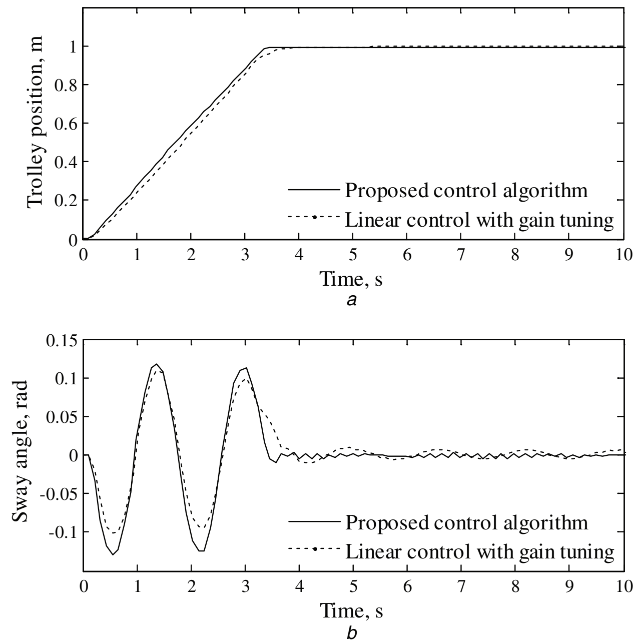

Fig. 6 *Comparison between the proposed control and the linear control laws with gain tuning*

a Trolley position
b Sway angle

0 to 1 m. The control performance of the proposed control law is compared with that of the conventional linear anti-sway control scheme using the conventional linear-quadratic regulator theory. The control input was

$$u_{\text{linear}} = -\frac{1}{k_{\text{m}}}(\boldsymbol{K}_{\textbf{linear}}\boldsymbol{z} + \boldsymbol{K}_{\textbf{g}}\boldsymbol{z}_{\textbf{d}}) \tag{30}$$

where k_{m} is a DC motor gain, $\boldsymbol{z} = [x \ \theta \ \dot{x} \ \dot{\theta}]^{\text{T}}$ and $\boldsymbol{z}_{\textbf{d}} = [x_d \ \ 0]^{\text{T}}$, and $\boldsymbol{K}_{\textbf{linear}}$ and $\boldsymbol{K}_{\textbf{g}}$ are a set of linear control gains obtained by the Riccati equation and given by $\boldsymbol{K}_{\textbf{linear}} = [1.4142 \quad -7.588 \quad 3.5825 \quad -4.0771]$ and $\boldsymbol{K}_{\textbf{g}} = [-1.4142 \quad 0]$. Without gain tuning, the control performance was poor: the trolley did not reach the goal position, even though the sway angle was almost zero, because the linear control did not take the friction into account when obtaining the control gains using the Riccati equation. Given that the set of control gains was applied directly to the practical crane system, the control input did not compensate for the friction. This incurred a steady-state error, as shown in Fig. 5. However, the trolley, after the control gain was tuned several times, reached the goal position with an acceptable sway angle. Therefore when a linear control strategy is applied, the control gains must be tuned for a real crane.

The experimental comparison of the control performances between the adaptive SMC law and the linear control with gain tuning are shown in Fig. 6. In both cases, the trolley reached the goal position in 3.5 s. Neither was there any significant difference in tracking to that position. Significantly although, the proposed control law suppressed the sway motion faster than did the linear control with gain tuning. Moreover, using the adaptive SMC, the residual sway of the load was smaller.

The comparison of the control performances between the proposed control law and the non-linear control is shown in Fig. 7. Here, the non-linear control law was designed by using the feedback linearisation technique [17] and is given

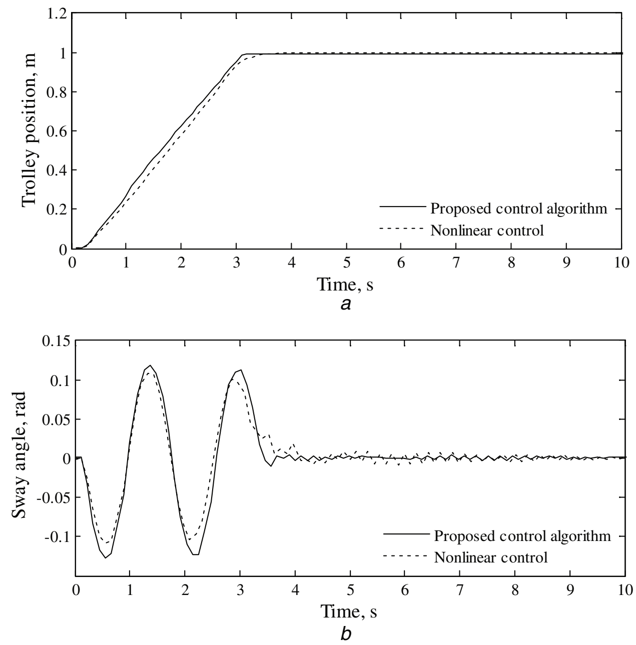

Fig. 7 *Comparison between the proposed control and the non-linear control laws with the feedback linearisation technique*

a Trolley position
b Sway angle

as follows

$$u_{\text{non-linear}} = \frac{g_1(\theta)(v_1 - h_1(\dot{\theta}, \theta)) + g_2(\theta)(v_2 - h_2(\dot{\theta}, \theta))}{g_1^2(\theta) + g_2^2(\theta)} \tag{31}$$

where $v_1 = \ddot{x}_d - k_{x1}(\dot{x} - \dot{x}_d) - k_{x2}(x - x_d)$ and $v_2 = -2k_\theta\dot{\theta} - k_\theta^2\theta$. The results were almost the same as the linear control law case: the sway suppression of the proposed control law

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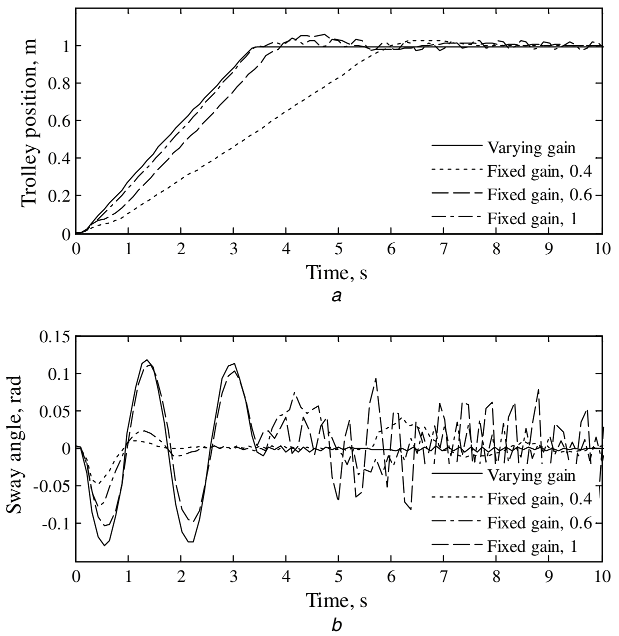

Fig. 8 *Comparison of two sliding mode controls: with varying gain and fixed gain*

a Trolley position
b Sway angle

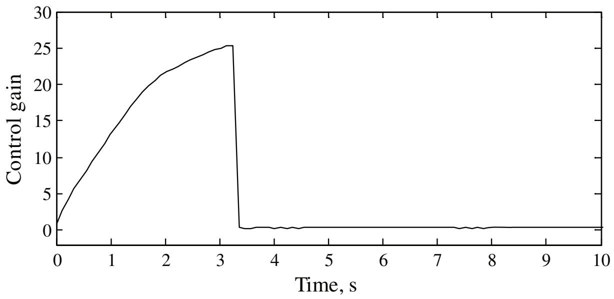

Fig. 9 *Varying control gain*

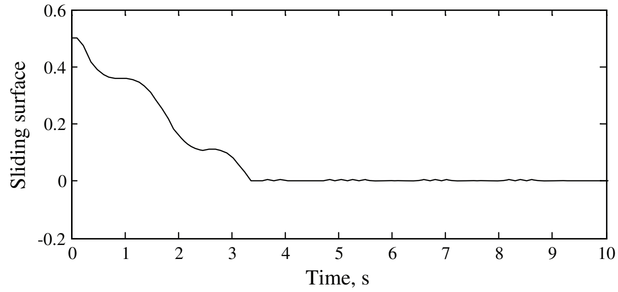

Fig. 10 *Sliding surface*

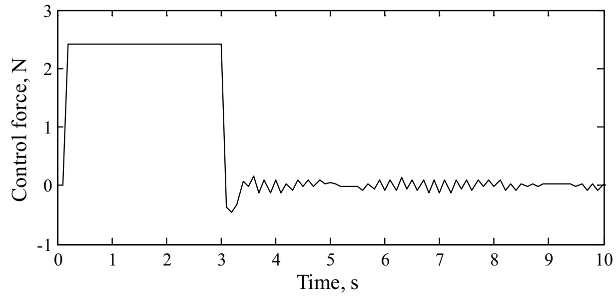

Fig. 11 *Control force*

was superior to that of the non-linear control law. However, the maximum sway angle of the load during the trolley motion in the case of the proposed adaptive SMC was a bit larger than that of the non-linear control. This will not become a problem as long as the sway angle remains within a specified range.

With a constant control gain $\hat{k}$, the control law (10) guarantees the stability of the system. However, the best control performance will not be achieved in this case. If a low value of $\hat{k}(t)$ is used, the trolley takes too much time to reach the goal position. With a large control gain, the time to reach the goal position becomes shorter but the sway motion gets larger. Fig. 8 compares the performance of the adaptive SMC and those of a number of fixed gains, respectively. The adaptive law generates a high control gain as long as the sliding surface does not reach the sliding mode, and switches to a low gain as soon as it achieves ($s = 0$ with $e_x = 0$ and $\theta = 0$); see Figs. 9 and 10. The strategy of using a high gain for fast response and a low gain for reducing chattering is a smart plan for SMC applications. Finally, the control force, which was applied on the trolley, is shown in Fig. 11.

5 Conclusions

In this paper, to achieve fast travelling and minimal chattering of the trolley, an adaptive SMC algorithm for container cranes was developed: if the system does not reach its sliding mode, the adaptive gain is increased. However, once it reaches, the gain gets small to reduce chattering. The derived control law guaranteed the asymptotic stability of the closed-loop system. Moreover, the control performance of the proposed control law was compared with those of a well-tuned LQ control and a feedback linearisation control law. Also, the key advantage of the adaptive SMC in robustness, using a crane model involving 20% uncertainty, was demonstrated.

6 Acknowledgment

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7 References

1 Hong, K.-S., Park, B.J., Lee, M.H.: ‘Two-stage control for container cranes’, *JSME Int. J. C*, 2000, **43**, (2), pp. 273–282
2 Singhose, W., Perter, L., Kenison, M., Krrikku, E.: ‘Effects of hoisting on the input shaping control of gantry cranes’, *Control Eng. Pract.*, 2000, **8**, (10), pp. 1159–1165
3 Hong, K.T., Huh, C.D., Hong, K.-S.: ‘Command shaping control for limiting the transient sway angle of crane systems’, *Int. J. Control Autom. Syst.*, 2003, **1**, (1), pp. 43–53
4 Sorensen, K.L., Singhose, W., Dickerson, S.: ‘A controller enabling precise positioning and sway reduction in bridge and gantry cranes’, *Control Eng. Pract.*, 2007, **15**, (7), pp. 825–837
5 Huey, J.R., Sorensen, K.L., Singhose, W.: ‘Useful applications of closed-loop signal shaping controllers’, *Control Eng. Pract.*, 2008, **16**, (7), pp. 836–846
6 Sorensen, K.L., Singhose, W.: ‘Command-induced vibration analysis using input shaping principles’, *Automatica*, 2008, **44**, (9), pp. 2392–2397

7 Sung, Y.G., Singhose, W.: 'Limited-state commands for systems with two flexible modes', *Mechatronics*, 2009, **19**, (5), pp. 780–787

8 Robertson, M.J., Singhose, W.: 'Robust discrete-time deflection-limiting commands for flexible systems', *IET Control Theory Appl.*, 2009, **3**, (4), pp. 473–480

9 Blackburn, D., Lawrence, J., Danielson, J., Singhose, W., Kamoi, T., Taura, A.: 'Radial-motion assisted command shapers for nonlinear tower crane rotational slewing', *Control Eng. Pract.*, 2010, **18**, (5), pp. 523–531

10 Sung, Y.G., Singhose, W.: 'Robustness analysis of input shaping commands for two-mode flexible systems', *IET Control Theory Appl.*, 2010, **3**, (6), pp. 722–730

11 Al-Garni, A.Z., Moustafa, K.A.F., Javeed Nizami, S.S.A.K.: 'Optimal control of ovehead cranes', *Control Eng. Pract.*, 1995, **3**, (9), pp. 1277–1284

12 Terashima, K., Shen, Y., Yano, K.: 'Modeling and optimal control of a rotary crane using the straight transfer transformation method', *Control Eng. Pract.*, 2007, **15**, (9), pp. 1179–1192

13 Sawodny, O., Aschemann, H., Lahres, S.: 'An automated gantry crane as a large workspace robot', *Control Eng. Pract.*, 2002, **10**, (12), pp. 1323–1338

14 Kim, Y.S., Hong, K.-S., Sul, S.K.: 'Anti-sway control of container cranes: inclinometer, observer, and state feedback', *Int. J. Control Autom. Syst.*, 2004, **2**, (4), pp. 435–449

15 Klosinski, J.: 'Swing-free stop control of the slewing motion of a mobile crane', *Control Eng. Pract.*, 2004, **13**, (4), pp. 451–460

16 Blajer, W., Kolodziejczyk, K.: 'Motion planning and control of gantry cranes in cluttered work environment', *IET Control Theory Appl.*, 2007, **1**, (5), pp. 1370–1379

17 Park, H., Chwa, D., Hong, K.-S.: 'A feedback linearization control of container cranes: varying rope length', *Int. J. Control Autom. Syst.*, 2007, **5**, (4), pp. 379–387

18 Messineo, S., Celani, F., Egeland, O.: 'Crane feedback control in offshore moonpool operations', *Control Eng. Pract.*, 2008, **16**, (3), pp. 356–364

19 Bartolini, G., Pisano, A., Usai, E.: 'Second-order sliding-mode control of container cranes', *Automatica*, 2002, **38**, (10), pp. 1783–1790

20 Bartolini, G., Pisano, A., Usai, E.: 'Output feedback control of container cranes: a comparative analysis', *Asian J. Control*, 2003, **5**, (4), pp. 578–593

21 Lee, H.H.: 'A new approach for the anti-swing control of overhead cranes with high-speed load hoisting', *Int. J. Control*, 2003, **76**, pp. 1493–1499

22 Lee, H.H.: 'A new design approach for the anti-swing trajectory control of overhead cranes with high-speed hoisting', *Int. J. Control*, 2004, **77**, (10), pp. 931–940

23 Lee, H.H.: 'A new motion-planning scheme for overhead cranes with high-speed hoisting', *Trans. ASME, J. Dyn. Syst. Meas. Control*, 2004, **126**, (2), pp. 359–364

24 Lee, H.H.: 'Motion planning for three-dimensional overhead cranes with high-speed load hoisting', *Int. J. Control*, 2005, **78**, (12), pp. 875–886

25 Orbisaglia, M., Orlando, G., Longhi, S.: 'A comparative analysis of sliding mode controllers for overhead cranes'. Proc. 16th Mediterranean Conf. Control and Automation, Ajaccio, France, June 2008, pp. 670–675

26 Xi, Z., Hesketh, T.: 'Discrete time integral sliding mode control for overhead crane with uncertainties', *IET Control Theory Appl.*, 2010, **4**, (10), pp. 2071–2081

27 Ma, B., Fang, Y., Zhang, Y.: 'Switching-based emergency braking control for an overhead crane system', *IET Control Theory Appl.*, 2010, **4**, (9), pp. 1739–1747

28 Ngo, Q.H., Hong, K.-S.: 'Sliding mode anti-sway control of an offshore container crane', *IEEE-ASME Trans. Mechatronics*, 2011, **17**, (2), pp. 201–209

29 Almutairi, N.B., Zribi, M.: 'Sliding mode control of a three-dimensional overhead crane', *J. Vib. Control*, 2009, **15**, (11), pp. 1679–1730

30 Chang, C.Y., Chiang, K.H.: 'Fuzzy projection control law and its application to the overhead crane', *Mechatronics*, 2008, **18**, (10), pp. 607–615

31 Liu, D., Yi, J., Zhao, D., Wang, W.: 'Adaptive sliding mode fuzzy control for a two-dimensional overhead crane', *Mechatronics*, 2005, **15**, (5), pp. 505–522

32 Hua, Y.J., Shine, Y.K.: 'Adaptive coupling control for overhead crane systems', *Mechatronics*, 2007, **17**, (2–3), pp. 143–152

33 Mizumoto, I., Chen, T., Ohdaira, S., Kumon, M., Iwai, Z.: 'Adaptive output feedback control of general MIMO systems using multirate sampling and its application to a cart-crane system', *Automatica*, 2007, **43**, (12), pp. 2077–2085

34 Messineo, S., Serrani, A.: 'Offshore crane control based on adaptive external models', *Automatica*, 2009, **45**, (11), pp. 2546–2556

35 Hong, K.-S., Bentsman, J.: 'Direct adaptive control of parabolic systems: algorithm synthesis, and convergence and stability analysis', *IEEE Trans. Autom. Control*, 1994, **39**, (10), pp. 2018–2033

36 d'Andrea-Novel, B., Coron, J.M.: 'Exponential stabilization of an overhead crane with flexible cable via a back-stepping approach', *Automatica*, 2000, **36**, (4), pp. 587–593

37 Kim, C.S., Hong, K.-S.: 'Boundary control of container cranes from perspective of controlling an axially moving string system', *Int. J. Control Autom. Syst.*, 2009, **7**, (3), pp. 437–445

38 Ngo, Q.H., Hong, K.-S., Jung, I.H.: 'Adaptive control of an axially moving system', *J. Mech. Sci. Technol.*, 2009, **23**, (11), pp. 3071–3078

39 Hwang, C.L., Wu, H.M., Shih, C.L.: 'Fuzzy sliding-mode under-actuated control for autonomous dynamic balance of an electrical bicycle', *IEEE Trans. Control Syst. Technol.*, 2009, **15**, (3), pp. 658–670

40 Hwang, C.L., Wu, H.M., Shih, C.L.: 'An autonomous dynamic balance of an electrical bicycle in motion using variable structure under-actuated control', *Asian J. Control*, 2011, **13**, (2), pp. 240–254

41 Plestan, F., Shtessel, Y., Bregeault, V., Poznyak, A.: 'New methodologies for adaptive sliding mode control', *Int. J. Control*, 2010, **83**, (9), pp. 1907–1919

42 Lee, H.H.: 'Modeling and control of a three-dimensional overhead crane', *Trans. ASME J. Dyn. Syst. Meas. Control*, 1998, **120**, (4), pp. 471–476

43 Hong, K.-S.: 'Asymptotic behavior analysis of a coupled time-varying system: application to adaptive systems', *IEEE Trans. Autom. Control*, 1997, **42**, (12), pp. 1693–1697

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