S-Curve Trajectory Generation for Residual Load Sway Suppression in a Rotary Crane System Using Only Horizontal Boom Motion

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Abstract
Generally, both horizontal and vertical boom motion must be used to suppress the load sway caused by horizontal boom motion of a rotary crane. However, it would be less energy intensive and also safer if a control scheme could be developed that only used horizontal boom motion, i.e., without the need for any vertical boom motion. In addition, if load sway can be suppressed without measuring it, reduction in sensor cost can be achieved. Furthermore, employing simple velocity trajectory patterns such as trapezoidal and S-curve patterns, which are widely used in industrial automation systems, may provide cost effective controller implementation. This study presents a simple S-curve trajectory generation method for rotary crane motion that can suppress residual load sway using only horizontal boom motion without load sway sensing. Numerical simulations and experimental results demonstrate the effectiveness of the proposed method.

Key words: Rotary Crane, Motion Control, Residual Load Sway Suppression, Trajectory Generation, Tracking Control

1. Introduction

Because crane systems use fewer actuators and have a simpler structure than industrial robots, they are widely used at construction sites and in harbors. In particular, rotary cranes do not require large equipment such as traveling rails, and hence they are also used in environments with limited space such as on vehicle trucks. However, one-dimensional horizontal boom motion of a rotary crane typically generates undesirable two-dimensional load sway; therefore, crane operators must be highly skilled in order to control the crane’s motion.

To reduce the burden on human operators, automatic control systems have been widely investigated. Various types of control schemes have been developed for crane control, including optimal control\(^1\),\(^4\), gain scheduling control\(^5\),\(^7\), sliding mode control\(^8\),\(^10\), adaptive control\(^11\), back-stepping control\(^12\), input-shaping control\(^13\), disturbance observer based control\(^14\), and nonlinear control based on the Lyapunov stability theorem\(^15\),\(^17\).

In most existing control schemes, both horizontal and vertical boom motion must be used to suppress load sway. However, it would be less energy intensive and also safer if a control scheme could be developed that used only horizontal boom motion, i.e., with no need for vertical boom motion, which must overcome gravity.

A few studies in the literature present controller designs for load sway suppression that use only horizontal boom motion\(^18\),\(^20\). These methods can be classified into two types: trajectory generation methods for horizontal boom motion that can suppress load sway without requiring load sway information (i.e., open-loop control)\(^18\),\(^20\) and methods using load sway information measured by sensor systems in real time to suppress load sway (i.e., closed-loop control)\(^21\),\(^24\). Because the latter methods require a sensor system, their hardware costs may
be higher. In addition, since cranes are inherently stable system, feedback control seems undesirable, especially when applied to welfare and nursing works. On the other hand, although the controllers designed in Refs. (18)-(20) do not require load sway sensors, the generated trajectories for horizontal boom motion are complicated, and it is difficult to employ simple industrial controllers that accept only trapezoidal or S-curve trajectories. Using a simple trapezoidal or S-curve trajectory would allow us to use cost-efficient industrial controllers.

This study proposes a control scheme that can suppress two-dimensional load sway using only horizontal boom motion. Because the control system should be robust with respect to variation in parameters such as joint friction, first, a controller design on the basis of a disturbance, plant gain, and command voltage from the computer, respectively. Equation (3) assumes that a nonlinear force term including the disturbance, plant gain, and command voltage from the computer, respectively. Equation (3) assumes that $J$ is constant, although the inertia term is varied according to the change of load mass and vertical boom angle $\theta_3$. The effect of this change is included in the disturbance term $d$.

To compensate for the effect of the disturbance, the following disturbance observer (24) is applied to Eq. (3):

$$ u = \frac{J}{K} \left\{ \dot{v} + \frac{\omega}{s + \omega} \left( K u - \dot{h}_3 \right) \right\} \tag{4} $$

A block diagram of this control system is shown in Fig. 2, where $s$, $v$, and $\omega$ represent the differential operator, new virtual control input calculated by the feedback controller, and cut-off angular frequency of the low-pass filter, respectively. The dynamics of the system in the
Because the centrifugal force caused by horizontal boom motion and gravitational force affect the two-dimensional load sway significantly, in order to derive the trajectory by numerical calculation easily, Eqs. (1) and (2) are linearized except for the centrifugal and gravitational force terms by assuming that \( \theta_i \) and \( \dot{\theta}_i \) (i=1,2) are small such that \( \theta_i \theta_j \approx 0 \), \( \theta_i \dot{\theta}_j \approx 0 \) (j=1,2), \( \dot{\theta}_i \dot{\theta}_j \approx 0 \), and \( \theta_i \ddot{\theta}_j \approx 0 \) are satisfied. 

\[
\ddot{\theta}_1 = -\frac{g}{l} \theta_1 + \frac{L \sin \theta_3}{l} \dot{\theta}_3^2 \\
\ddot{\theta}_2 = -\frac{g}{l} \theta_2 - \frac{L \sin \theta_3}{l} \dot{\theta}_3
\]

Equations (6) and (7) lead to the following simple dynamical model of rotary crane motion:

\[
\ddot{\theta}_1 = -\omega_n^2 \theta_1 + b \dot{\theta}_1^2 \\
\ddot{\theta}_2 = -\omega_n^2 \theta_2 - b \dot{\theta}_2
\]

where \( \omega_n = \sqrt{g/l} \) and \( b = L \sin \theta_3/l \).
3. Trajectory Generation

This section describes generation of an S-curve velocity trajectory for horizontal boom motion that can suppress load sway on the basis of the crane model in Eqs. (8) and (9). The S-curve velocity trajectory shown in Fig. 3 is widely used in industrial applications. Parameters \( t_1, t_2, t_3, \) and \( \dot{\theta}_4c \) are the acceleration period, constant-velocity period, deceleration period, and constant-velocity, respectively.

The S-curve velocity trajectory can be represented as follows:

\[
\dot{\theta}_4 = \begin{cases} 
\frac{\dot{\theta}_4c}{2} (1 - \cos \omega_1 t) & 0 \leq t < t_1 \\
\dot{\theta}_4c t_1 & t_1 \leq t < t_1 + t_2 \\
\frac{\dot{\theta}_4c}{2} \{1 + \cos \omega_2(t - t_1 - t_2)\} & t_1 + t_2 \leq t \leq t_1 + t_2 + t_3 
\end{cases} \tag{10}
\]

The acceleration trajectory \( \ddot{\theta}_4 \) can be obtained as follows:

\[
\ddot{\theta}_4 = \begin{cases} 
\frac{\omega_1 \dot{\theta}_4c}{2} \sin \omega_1 t & 0 \leq t < t_1 \\
0 & t_1 \leq t < t_1 + t_2 \\
-\frac{\omega_2 \dot{\theta}_4c}{2} \sin \omega_2(t - t_1 - t_2) & t_1 + t_2 \leq t \leq t_1 + t_2 + t_3 
\end{cases} \tag{11}
\]

where \( \omega_1 = \pi/t_1 \), and \( \omega_2 = \pi/t_3 \).

By substituting Eq. (10) into Eq. (8), the load sway angle \( \theta_1 \) for \( 0 \leq t < t_1, \theta_{11} \), is derived as follows:

\[
\theta_{11} = C_{11} \cos \omega_n t + C_{12} \sin \omega_n t + W 
\]

\[
W = \frac{A \left[3 \omega_n^4 - 15 \omega_n^2 \omega_1^2 + 12 \omega_1^4 + (-\omega_2^2 \omega_1^2 + \omega_0^2) \cos 2 \omega_1 t + (16 \omega_n^2 \omega_1^2 - 4 \omega_0^2) \cos \omega_1 t\right]}{2 \omega_n^6 - 10 \omega_n^4 \omega_1^2 + 8 \omega_n^2 \omega_1^4}
\]

where \( A = b \dot{\theta}_4c^2 / 4 \), and \( C_{11} \) and \( C_{12} \) can be determined from the boundary conditions \( \theta_{11}(t) = 0 \) and \( \dot{\theta}_{11}(t) = 0 \) at \( t = 0 \).

The load sway angle \( \theta_1 \) for \( t_1 \leq t < t_1 + t_2, \theta_{12}, \) is determined as follows:

\[
\theta_{12} = C_{13} \cos \omega_n t + C_{14} \sin \omega_n t + \frac{b \dot{\theta}_4c^2}{\omega_n^6} \tag{13}
\]
The time derivative of Eq. (18) is derived as follows:

\[ \dot{\theta}_1 = C_{13} \cos \omega_n t + C_{16} \sin \omega_n t + W_1 \]

where \( C_{13} \) and \( C_{14} \) can be determined from the boundary conditions \( \theta_1(t) = \theta_{11}(t) \) and \( \dot{\theta}_1(t) = \dot{\theta}_{11}(t) \) at \( t = t_1 \).

The load sway angle \( \theta_t \) for \( t_1 + t_2 \leq t \leq t_1 + t_2 + t_3 \), \( \theta_{13} \), is determined as follows:

\[ \theta_{13} = C_{15} \cos \omega_n t + C_{16} \sin \omega_n t + W_1 \]  \hspace{1cm} (14)

\[ W_1 = \frac{N}{D} \]

\[ N = A \{3 \omega_n^2 - 15 \omega_n^2 \omega_2^2 + 12 \omega_2^4 + (-\omega_n^2 + \omega_2^2) \sin 2 \omega_2 (t - t_1 - t_2) \]

\[ + (-16 \omega_n^2 \omega_2^2 + 4 \omega_2^4) \cos 2 \omega_2 (t - t_1 - t_2) \} \]

\[ D = 2 \omega_n^6 - 10 \omega_n^4 \omega_2^2 + 8 \omega_n^2 \omega_2^4 \]

where \( C_{15} \) and \( C_{16} \) can be determined from the boundary conditions \( \theta_{11}(t) = \theta_{12}(t) \) and \( \dot{\theta}_{11}(t) = \dot{\theta}_{12}(t) \) at \( t = t_1 + t_2 \).

The time derivative of Eq. (14) is derived as follows:

\[ \dot{\theta}_{13} = -\omega_n C_{15} \sin \omega_n t + \omega_n C_{16} \cos \omega_n t + \dot{W}_1 \]

\[ \dot{W}_1 = A \left\{ (2 \omega_n^2 \omega_2^2 - 2 \omega_2^4) \sin 2 \omega_2 (t - t_1 - t_2) + (16 \omega_n^2 \omega_2^2 - 4 \omega_2^4) \sin \omega_2 (t - t_1 - t_2) \right\} \]

By substituting Eq. (11) into Eq. (9), the load sway angle \( \theta_{21} \) for \( 0 \leq t \leq t_1 \), \( \theta_{21} \), is derived as follows:

\[ \theta_{21} = C_{21} \cos \omega_n t + C_{22} \sin \omega_n t \]

\[ \dot{\theta}_{21} = -\omega_n C_{21} \sin \omega_n t + \omega_n C_{22} \cos \omega_n t \]

\[ \dot{W}_{21} = A \left\{ (2 \omega_n^2 \omega_2^2 - 2 \omega_2^4) \sin 2 \omega_2 (t - t_1 - t_2) + (16 \omega_n^2 \omega_2^2 - 4 \omega_2^4) \sin \omega_2 (t - t_1 - t_2) \right\} \]

where \( B = \beta \omega_n / 2 \), and \( C_{21} \) and \( C_{22} \) can be determined from the boundary conditions \( \theta_{21}(t) = 0 \) and \( \dot{\theta}_{21}(t) = 0 \) at \( t = 0 \).

The load sway angle \( \theta_{22} \) for \( t_1 \leq t \leq t_1 + t_2 \), \( \theta_{22} \), is determined as follows:

\[ \theta_{22} = C_{23} \cos \omega_n t + C_{24} \sin \omega_n t \]

where \( C_{23} \) and \( C_{24} \) can be determined from the boundary conditions \( \theta_{22}(t) = \theta_{11}(t) \) and \( \dot{\theta}_{22}(t) = \dot{\theta}_{11}(t) \) at \( t = t_1 \).

The load sway angle \( \theta_{23} \) for \( t_1 + t_2 \leq t \leq t_1 + t_2 + t_3 \), \( \theta_{23} \), is determined as follows:

\[ \theta_{23} = C_{25} \cos \omega_n t + C_{26} \sin \omega_n t + \frac{B \omega_n^2 \cos \omega_2 (t - t_1 - t_2)}{\omega_n^2 - \omega_2^2} \]

where \( C_{25} \) and \( C_{26} \) can be determined from the boundary conditions \( \theta_{23}(t) = \theta_{22}(t) \) and \( \dot{\theta}_{23}(t) = \dot{\theta}_{22}(t) \) at \( t = t_1 + t_2 \).

The time derivative of Eq. (18) is derived as follows:

\[ \dot{\theta}_{23} = -\omega_n C_{25} \sin \omega_n t + \omega_n C_{26} \cos \omega_n t + \frac{B \omega_n^2 \cos \omega_2 (t - t_1 - t_2)}{\omega_n^2 - \omega_2^2} \]

where \( t = t_1 + t_2 + t_3 \). In the above equations, the variables are \( t_1, t_2, t_3, \) and \( \theta_{1c} \). Because there are as equations as there are variables, Eqs. (20)-(23) are intrinsically solvable. Conventional solvers for nonlinear equations such as the Newton-based method can be applied to solve them.

Substituting \( \dot{\theta}_3(T) = 0, \dot{\theta}_4(T) = 0, \) and Eqs. (20)-(23) into Eqs. (8) and (9), \( \dot{\theta}_1(T) = 0 \) and \( \dot{\theta}_2(T) = 0 \) are obtained. Hence, in addition to this, by considering \( \dot{\theta}_{13}(T) = 0 \) and \( \dot{\theta}_{23}(T) = 0 \), the residual load sway can be suppressed.
4. Simulation and Experimental Results

4.1. Experimental System

The experimental system is shown in Fig. 4. A DC servo motor drives the boom and base. The horizontal boom angle $\theta_4$ is measured using a rotary encoder, whose angular measurement resolution is $1.8 \times 10^{-3}$[deg].

In Fig. 5, the load sway angles $\theta_1$ and $\theta_2$ are measured by potentiometers 1 and 2, respectively. Potentiometer 1 is fixed to the boom. Potentiometer 2 is fixed to part 1, which rotates around the rotational axis of potentiometer 1. Part 2 rotates around the rod and the rotational axis of potentiometer 2. Part 3 rotates around the rod, and part 4 rotates around part 3. A linear bearing is installed in the hole of part 4, which provides smooth motion along the rope. In addition, rotary bearings are installed for all rotational parts in Fig. 5. The angular measurement resolution is $6.4 \times 10^{-2}$[deg]. $\theta_1$ and $\theta_2$ can be calculated from the output voltage values of potentiometers 1 and 2. In this study, the sensor system is used only to observe the load sway. The parameters of the rotary crane are shown in Table 1.

<table>
<thead>
<tr>
<th>$J$[kg·m$^2$]</th>
<th>1.02</th>
<th>$K$[N·m/V]</th>
<th>2.06</th>
<th>$L$[m]</th>
<th>0.65</th>
</tr>
</thead>
<tbody>
<tr>
<td>$l$[m]</td>
<td>0.30</td>
<td>$g$[m/s$^2$]</td>
<td>9.81</td>
<td>$\theta_3$[deg]</td>
<td>45.0</td>
</tr>
</tbody>
</table>
4.2. Trajectory Generation Conditions and Results

This section presents the initial conditions and results for trajectory generation on the basis of the proposed method in §3. Because Eqs. (20)-(23) are nonlinear, numerical solutions depend on their initial values. The Levenberg-Marquardt algorithm is employed for solving Eqs. (20)-(23). The initial value for constant-velocity was set as $\dot{\theta}_4 = 0.3$[rad/s]. The initial values for $t_1$ and $t_3$ are the same. They are varied from 0.05[s] to 1.5[s] by 0.01[s] intervals, and Eqs. (20)-(23) are numerically solved for all cases. The destination angle for the boom $\theta_f$ is $\theta_f = 30$[deg], 45[deg], or 60[deg]. The initial value of the constant-velocity period $t_2$ is uniquely determined from the initial values of $t_1$, $t_3$, $\dot{\theta}_4$ and $\theta_f$.

The same results have been obtained for all $t_1 = t_3 = 0.05$-1.5[s] as shown in Table 2. The values of $t_1$, $t_2$, and $t_3$ are the same for all $\theta_f$, because the natural frequency of the load sway $\omega_n$ does not change.

4.3. Tracking Controller Design

This section describes the tracking controller design needed for the boom to achieve the obtained S-curve trajectory in §4.2.

By integrating the velocity trajectory of Eq. (10), the angular trajectory $\theta_4$, which is employed for a desired angular reference $r$ of the boom motion, can be obtained as follows:

$$
\begin{align*}
\theta_4 &= \theta_0 + \frac{\theta_4}{2} - \frac{\theta_4}{2\omega_n^2} \sin \omega_n t \\
&\quad + \frac{\dot{\theta}_4}{2} \sin \omega_2 (t - t_1 - t_2) + \frac{\dot{\theta}_4}{2} \\
&\quad + \frac{\dot{\theta}_4}{2} \sin \omega_2 (t - t_1 - t_2 + t_3) \\
&\quad + \frac{\dot{\theta}_4}{2} \sin \omega_2 (t - t_1 - t_2 + t_3 + t_4)
\end{align*}
$$

(24)

where $\theta_f$ is the final angle, $\theta_0$ is the initial angle, and $\theta_f$ is the final time. They were set to $\theta_0 = 0$[deg] and $\theta_f = 10$[s]. In addition, the first and second time derivatives of $r$, $\dot{r}$ and $\ddot{r}$, are assumed to be available for controller design.

The following Proportional-Derivative (PD) controller for horizontal boom motion was designed.

$$
\begin{align*}
v &= \ddot{r} + K_p e + K_v \dot{e} \\
e &= r - \theta_4, \quad \dot{e} = \dot{r} - \dot{\theta}_4
\end{align*}
$$

(25)

where $K_p$ and $K_v$ are controller gains, and were set to $K_p = 30[1/s^2]$ and $K_v = 50[1/s]$.

Applying Eq. (25) to Eq. (5) yields

$$
\ddot{e} + K_p e + K_v \dot{e} = 0
$$

(26)

Hence, asymptotic stability can be achieved when $K_p > 0$ and $K_v > 0$. 

<table>
<thead>
<tr>
<th>$\theta_f$ [deg]</th>
<th>$\theta_f = 45$</th>
<th>$\theta_f = 30$</th>
<th>$\theta_f = 60$</th>
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<tr>
<td>$\dot{\theta}_4$ [rad/s]</td>
<td>0.286</td>
<td>0.191</td>
<td>0.381</td>
</tr>
<tr>
<td>$t_1$ [s]</td>
<td>1.648</td>
<td>1.648</td>
<td>1.648</td>
</tr>
<tr>
<td>$t_2$ [s]</td>
<td>1.099</td>
<td>1.099</td>
<td>1.099</td>
</tr>
<tr>
<td>$t_3$ [s]</td>
<td>1.648</td>
<td>1.648</td>
<td>1.648</td>
</tr>
</tbody>
</table>
4.4. Simulation Results

Because the trajectory for horizontal boom motion obtained in §4.2 was based on the crane model in Eqs. (8) and (9), which is called a simple model, its effectiveness for not only the simple model but also the original model Eqs. (1)-(3) should be verified.

Simulation results are shown in Fig. 6. The load sway is suppressed in Figs. 6(c) and (d), whereas the boom tracks the reference trajectory in Fig. 6(a). These results confirm the effectiveness of the proposed S-curve trajectory generation for anti-sway control of the load.

Next, the disturbance \(d\) in Eq. (3), which includes not only nonlinear force but also the following friction, was considered:

\[
f = C\dot{\theta}_4 + f_n
\]  

(27)

where \(C\) denotes a viscous friction coefficient, and \(f_n\) denotes static friction or Coulomb friction. The \(f_n\) profile is shown in Fig. 7 and can be represented as follows:

\[
f_n = \begin{cases} 
\text{sgn}(Ku) \cdot \min(\{|Ku|, f_s\}) & (\dot{\theta}_4 = 0) \\
\text{sgn}(\dot{\theta}_4)f_c & (\dot{\theta}_4 \neq 0)
\end{cases}
\]  

(28)

where \(f_s\) and \(f_c\) denote the magnitude of the static and Coulomb friction, respectively. Their numerical values are determined experimentally as \(C = 1.39[N\cdot m/(rad/s)], f_s = 3.71[N\cdot m],\) and \(f_c = 3.58[N\cdot m].\)

To compensate for frictional effects, a control system with the disturbance observer shown in Fig. 2 was designed. The cut-off angular frequency for the disturbance observer was set to \(\omega = 40\ [rad/s].\) All angular velocities are calculated by taking the backward difference between successive positional measurements. Simulation results are shown in Fig. 8. The tracking error shown in Fig. 8(a) has been reduced by using the disturbance observer, and the residual load sway shown in Fig. 8(d) was also reduced. These results confirm the effect of the disturbance observer.
Fig. 7 Friction model

Fig. 8 Effectiveness of disturbance observer (simulation)
4.5. Experimental Results

To demonstrate the effectiveness of the proposed S-curve trajectory, it was compared experimentally with a conventional step trajectory and a cycloid trajectory for the desired angular reference of the horizontal boom motion.

The step trajectory is represented as follows:

\[ r = \theta_f \quad 0 \leq t \leq t_f \]  \hspace{1cm} (29)

where \( \theta_f = 45[\text{deg}] \) and \( t_f = 10[\text{s}] \).

The cycloid trajectory is represented as follows\(^{24}\):

\[
 r = \begin{cases} 
 (\theta_f - \theta_0) \left( \frac{t}{t_s} - \frac{1}{2\pi} \sin \left( 2\pi \frac{t}{t_s} \right) \right) + \theta_0 & 0 \leq t < t_s \\
 \theta_f & t_s \leq t \leq t_f 
\end{cases} 
\]  \hspace{1cm} (30)

where \( \theta_f \) is the final angle, \( \theta_0 \) is the initial angle, \( t_s \) is the settling time, and \( t_f \) is the final time. They were set to \( \theta_0 = 0[\text{deg}], \theta_f = 45[\text{deg}], t_s = 5[\text{s}], \) and \( t_f = 10[\text{s}] \).

The horizontal boom angle, command voltage, and load sway angles for the step trajectory are shown in Fig. 9. Significant residual load sway is confirmed.

The horizontal boom angle, command voltage, and load sway angles for the cycloid trajectory are shown in Fig. 10. In Fig. 10(c), the residual load sway is slightly different from the simulation result because of the effect of friction between the rope and the top of the boom. Friction gradually reduced the load’s oscillation in the plane of vertical boom motion.

The horizontal boom angle, command voltage, and load sway angles for the proposed S-curve trajectory are shown in Fig. 11. In Fig. 11(c), the simulation result differs from that shown in Fig. 6(c) because of the effect of nonlinearity in the crane model in Eqs. (1) and (2). On the other hand, the experimental result in Fig. 11(c) also differs from the simulation result because of the effect of friction between the rope and the top of the boom.
Fig. 10  Cycloid curve trajectory result ($\theta_f$=45[deg])

Fig. 11  Proposed s-curve trajectory result ($\theta_f$=45[deg])
Fig. 12 Proposed s-curve trajectory result ($\theta_3 = 30[\text{deg}]$)

(a) Horizontal boom angle $\theta_4$
(b) Command voltage $u$
(c) Load sway angle $\theta_1$
(d) Load sway angle $\theta_2$

Fig. 13 Proposed s-curve trajectory result ($\theta_3 = 60[\text{deg}]$)

(a) Horizontal boom angle $\theta_4$
(b) Command voltage $u$
(c) Load sway angle $\theta_1$
(d) Load sway angle $\theta_2$
Fig. 14 Proposed s-curve trajectory result ($\theta_f=30[\text{deg}]$)

Fig. 15 Proposed s-curve trajectory result ($\theta_f=60[\text{deg}]$)
The performance of load sway suppression in Figs. 10(c) and (d) is much better than that in Figs. 9(c) and (d), because the cycloid curve trajectory provides zero acceleration at the initial and terminal points. The proposed S-curve trajectory provides better performance, especially that in Fig. 11(d) relative to that in Fig. 10(d), although the trajectory for $\theta_4$ in Fig. 11(a) approaches $\theta_f$ faster than that in Fig. 10(a).

To demonstrate the effectiveness of the proposed method for different vertical boom angles, simulations and experiments by setting $\theta_3 = 30[^\circ]$ and $\theta_3 = 60[^\circ]$ using the parameters in Table 2 were conducted. The results are shown in Figs. 12 and 13. Almost the same results were obtained in these setups as were obtained with $\theta_3 = 45[^\circ]$.

Simulation and experimental results for $\theta_f = 30[^\circ]$ and $\theta_f = 60[^\circ]$ with the parameters in Table 2 are shown in Figs. 14 and 15, which confirm the effectiveness of the proposed method.

5. Conclusion

Our objective was to suppress two-dimensional residual load sway using only horizontal boom motion. To achieve this, first, a simple model for rotary crane dynamics using a disturbance observer was considered. Next, an S-curve trajectory for horizontal boom motion, which can suppress residual load sway on the basis of the simple model, was numerically generated. The proposed trajectory provides better control performance than a conventional step or cycloid trajectory. Because energy-efficient control is necessary owing to global environmental and energy resource problem, the energy efficiency of the proposed method relative to that of the conventional method using vertical boom motion will be verified in future work.

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