An anti-sway positioning control method via load generalized position tracking with disturbance observer

Dan Niu\textsuperscript{1,3}, Yuxuan Zhu\textsuperscript{1,3}, Xisong Chen\textsuperscript{1,3}, Qi Li\textsuperscript{1,3}, Xiaojun Wang\textsuperscript{1,3}, Yanlan Yang\textsuperscript{1,3} and Simin Wang\textsuperscript{2}

Abstract
The bridge crane system is widely used in the industrial production for transporting large loads. Its anti-sway positioning control is quite crucial for enhancing handling efficiency and safety, but it is also difficult due to underactuated dynamics and various disturbances. In this paper, an anti-sway positioning control algorithm for unmanned crane is proposed based on the load generalized position tracking control algorithm (GPTC), which combines with a disturbance observer to effectively reject the lumped disturbances. The test results show that the proposed method can effectively achieve anti-sway and positioning with prominent disturbance suppression improvements.

Keywords
Bridge crane, anti-sway control, disturbance observer, load generalized position tracking

Date received: 11 June 2020; accepted: 8 September 2020

Introduction
In the past few decades, researchers have been making a lot of efforts to explore effective control strategies for underactuated mechanical systems. Underactuated bridge cranes are of great importance mechanical equipment, and are widely used for heavy material transportation in many industrial sites (such as ports, factories, workshops, etc.).\textsuperscript{1–4} They have many advantages, including high transportation efficiency, low energy consumption, and simple mechanical structure. However, the bridge crane systems have only one control input and two degrees of freedom. The underactuated feature greatly increases the system control complexity and it is quite challenging. The key control goal is to drive the trolley from the initial position to the target position quickly and accurately, while the payload swing must be effectively attenuated. Due to the wide application of bridge crane systems, the anti-sway positioning control and interference suppression has captured a lot of well-deserved attentions from the industrial electronics and control community.

Many researchers have done a lot of important work to control underactuated crane systems, which can be roughly divided into two categories: trajectory planning method and feedback control methods (closed loop).\textsuperscript{5} In the former methods, the input shaping technique is widely employed.\textsuperscript{6–10} In Magsoudi et al.,\textsuperscript{7} an improved input shaping method based on particle swarm optimization is proposed for the 3D crane system, and the payload swing is greatly reduced in the control process. In Wu and Xia,\textsuperscript{9} an energy-optimal trajectory planning method is designed for safety and energy saving. Zhang\textsuperscript{9} considered some constraints, including available speed, the allowable swing amplitude and acceleration, and proposed a minimum time trajectory planning method based on the quasi-convex optimization. Sun et al.\textsuperscript{10} designs a motion planning method for the double pendulum crane system. Compared with the trajectory planning methods, closed-loop feedback control methods exhibit better performance in terms of external disturbances and parameter uncertainties.\textsuperscript{11–30} In Sun et al.,\textsuperscript{12} the output feedback control method is designed assuming that the speed signals are not obtained. Ramli et al.\textsuperscript{13} assumes viscous damping and proposes a composite control scheme for bridge cranes with lifting/lowering payloads. In Zhang et al.,\textsuperscript{15–23} adaptive and sliding mode control methods are proposed to tackle external disturbances.

\textsuperscript{1}School of Automation, Southeast University, Nanjing, China
\textsuperscript{2}Jiangyin Zhixing Industrial Control Co., Ltd, Wuxi, China
\textsuperscript{3}Key Laboratory of Measurement and Control of CSE, Ministry of Education

Corresponding author:
Dan Niu, School of Automation, Southeast University, 2# Sipailou, Nanjing 210096, China.
Email: danniu1@163.com

Creative Commons CC BY: This article is distributed under the terms of the Creative Commons Attribution 4.0 License (https://creativecommons.org/licenses/by/4.0/) which permits any use, reproduction and distribution of the work without further permission provided the original work is attributed as specified on the SAGE and Open Access pages (https://us.sagepub.com/en-us/nam/open-access-at-sage).
disturbances or system uncertainties. What is more, several other efficient control methods have also been adopted, including model predictive control,\(^{24}\) optimal control,\(^{25–28}\) intelligent control,\(^{29,30}\) and so on.

Currently, the anti-sway positioning control of crane system remains an open topic. For one thing, several kinds of existing control methods rely on the precise model of the system and they are sensitive to internal and external disturbances, such as unmodeled dynamics, friction variations, rope length errors, and so on. For another thing, several robust control methods, such as H-infinity control, have been presented to deal with the adverse effects caused by uncertain disturbances. However, their disturbance suppression control is through feedback control and the robustness is obtained at the expense of their nominal control performance.\(^{31}\)

Inspired by the development of feedforward compensation control methods,\(^{32–40}\) a generalized position tracking control method integrated with disturbance observer is proposed in this work to achieve high-performance anti-sway positioning control with significant disturbance rejection improvements. Firstly, the nonlinear crane dynamic equations are linearized near the equilibrium point. Then, an anti-sway positioning control algorithm combing the load generalized position tracking with disturbance observer is put forward. The disturbance observer is utilized to estimate the disturbances and uncertainties to conduct feedforward compensation. The robust stability criterion of the closed-loop system is given. Finally, disturbance suppression performance of the proposed method is demonstrated.

**Problem formulation**

The bridge crane is a complex underactuated system,\(^{41–43}\) in which the number of independent control variables is less than the number of system freedom degrees. In this system, the weight block (payload) is hung on the overhead crane trolley to make an approximate single pendulum motion through the wire rope.\(^{13}\) The schematic diagram of the crane trolley is shown in Figure 1. To facilitate system analysis, the following reasonable assumptions are made: firstly, only the movement of bridge driving and lifting heavy objects is considered. Secondly, the length of wire rope remains unchanged when the bridge is traveling. Finally, lifting heavy objects only moves in a plane and is always in a horizontal state.

In this figure, \(F\) is the resultant force, consisting of the actuating force and bridge friction. The bridge crane with mass \(M\) moves along the track. The payload with mass \(m\) is hung on the bridge and travels by wire rope to make an approximate single pendulum movement.

After the force analysis, the system dynamic equations can be established and the crane system is modeled using the Euler-Lagrange method, which is shown in equation (1):

\[
\begin{bmatrix}
\ddot{\theta} + \ddot{x} \cos \theta + g \sin \theta = 0 \\
(M + m)\ddot{x} + m \ell (\ddot{\theta} \cos \theta - \dot{\theta}^2 \sin \theta) = F
\end{bmatrix}
\]  

(1)

where \(x\) and \(\theta\) denote the crane position and the payload angle with respect to the vertical direction, respectively; \(M\) and \(m\) is the crane mass and the payload mass, respectively; \(\ell\) is the length of wire rope and \(g\) represents the gravity acceleration. The lumped disturbances including parameter uncertainties, unmodeled dynamics, friction variations, and so on.

Crane system (1) can be expressed into the matrix form:

\[
M(q)\ddot{q} + C(q, \dot{q})\dot{q} + K(q) = R
\]  

(2)

where \(q\) and \(R\), respectively, denote state variable matrix and input variable matrix with the following expressions:

\[
\begin{align*}
q^T &= [x \quad \theta] \\
R^T &= [F \quad 0]
\end{align*}
\]  

(3)\quad (4)

The expressions of other variable matrices in equation (2) are as follows:

\[
M(q) = \begin{bmatrix} M + m & ml \cos \theta \\ ml \cos \theta & ml^2 \end{bmatrix}
\]  

(5)

\[
C(q, \dot{q}) = \begin{bmatrix} 0 & -ml \sin \theta \dot{\theta} \\ 0 & 0 \end{bmatrix}
\]  

(6)

\[
K(q) = \begin{bmatrix} 0 \\ mgl \sin \theta \end{bmatrix}
\]  

(7)

It is known that the energy of the bridge crane system contains the kinetic and potential energies\(^{12}\)

\[
E = \frac{1}{2} \dot{q}^T M(q) \dot{q} + mgl(1 - \cos \theta)
\]  

(8)

In this system, achieving anti-swing and positioning means that \(\lim q(t) = [x_o \quad 0]\) and \(\lim E(t) = 0\), where \(x_o\) is the target position of the crane.
Control design

Load generalized position tracking controller

By substituting equation (1) into the time derivative of formula $E$, obtains

$$\dot{E} = \dot{q}^T R = \dot{x} F$$  \hspace{1cm} (9)

In equation (9), the crane system takes $F$ as input, $\dot{x}$ as output, and $E(t)$ as energy storage function. The system (9) is passive, but the system is underactuated and $\dot{E}$ does not include the load swing motion ($\theta(t)$ or $\theta'(t)$). In order to enhance the coupling of the system states, a new energy storage function $E_x$ instead of $E$ is proposed in Sun and Fang.\cite{43} Its derivative is expected to take the following form as:

$$\dot{E}_x = F \cdot \dot{X}_P$$  \hspace{1cm} (10)

where $X_P = x + g(\theta)$ represents the generalized horizontal displacement of the load. $g(\theta) = K \sin \theta$ is set and it is a definite scalar, indicating the generalized displacement caused by the payload swing. Thus, the generalized displacement $X_P$ contains the position information and swing angle information of the load, which can be used to design controller. The basis of $E_x$ can be further expressed as:

$$\dot{E}_x = F \cdot \left[ \dot{x} + g'(\theta) \cdot \dot{\theta} \right]$$  \hspace{1cm} (11)

$$g'(\theta) = \frac{dg(\theta)}{d\theta}$$  \hspace{1cm} (12)

It can be seen from equation (11) that the new energy storage function $E_x$ is still passive, but the system output is changed to the generalized displacement of the load $X_P$. It converts the under-driven system into a “full-drive” system and completes the coupling control design of swing angle.\cite{43}

According to equations (9) and (11), $E_x$ can be denoted as:

$$E_x = E + E_a$$  \hspace{1cm} (13)

where $E$ is presented in equation (8). $E_a$ is the added energy function, satisfying $\dot{E}_a = F \cdot g'(\theta) \cdot \dot{\theta}$.

Substituting $g(\theta) = K \sin \theta$, yields

$$E_x = E + E_a = \frac{1}{2} \dot{q}^T M(q) \dot{q} + mg(1 - \cos \theta)
+ \frac{1}{2} K (M + m \sin^2 \theta) \dot{\theta}^2 + K (M + m) g s(1 - \cos \theta)
= \frac{1}{2} \begin{bmatrix} \dot{x} \\ \dot{\theta} \end{bmatrix} \begin{bmatrix} M + m & ml \cos \theta \\ ml \cos \theta & ml^2 \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{\theta} \end{bmatrix}
+ \frac{1}{2} K (M + m \sin^2 \theta) \dot{\theta}^2
+ (ml + K(M + m)) g s(1 - \cos \theta)
= \frac{1}{2} \begin{bmatrix} \dot{x} \\ \dot{\theta} \end{bmatrix} \begin{bmatrix} M + m & ml \cos \theta \\ ml \cos \theta & ml^2 + KM + Kms^2 \theta \end{bmatrix}
+ (ml + K(M + m)) g s(1 - \cos \theta)
= \frac{1}{2} \dot{q}^T T(q) \dot{q} + (ml + K(M + m)) g s(1 - \cos \theta)$$  \hspace{1cm} (14)

In order to ensure that $E_a$ is a positive scalar function, $K$ needs to satisfy the condition: $K < 0$.

It is easy to prove that $q(q)$ in the first term of $E_x$ is a positive definite matrix, and the second term of $E_x$ is a positive definite scalar function. Thus, $E_x$ is a positive definite scalar function. Next, the Lyapunov energy function $V(t)$ can be constructed:

$$V(t) = E_x(t) + \frac{k_1}{2} \delta^2$$  \hspace{1cm} (15)

where the parameter $k_1$ is a positive real number. $\delta$ is the generalized positioning error of the load, obtained from the generalized position $X_P$ of the driving load and the target position $x_a$:

$$\delta(t) = X_P - x_a = x + g(\theta) - x_a = x - x_a + K \sin \theta$$  \hspace{1cm} (16)

The time derivative of $V(t)$ in equation (15), yields:

$$\dot{V}(t) = \dot{E}_x(t) + k_1 \dot{\delta} \ddot{\delta} = F \dot{X}_P + k_1 \dot{\delta} \ddot{\delta}
= F \frac{d}{dt} (\delta(t) + x_a) + k_1 \dot{\delta}(t) \frac{d}{dt} (\delta(t))
= [F + k_1 \dot{\delta}(t)] \frac{d}{dt} (\delta(t)) V(t) = E_x(t)
+ \frac{k_1}{2} \ddot{\delta}^2 V(t) = E_x(t) + \frac{k_1}{2} \ddot{\delta}^2$$  \hspace{1cm} (17)

Based on equation (16), the controller is constructed as follows:

$$F = - k_1 \dot{\delta}(t) - k_2 \ddot{\delta}(t), k_2 \in \mathbb{R}^+$$  \hspace{1cm} (18)

$V(t)$ is a positive-definite function. Setting $V(t)$ be the Lyapunov function and substituting equation (18) into equation (17), obtains

$$\dot{V}(t) = - k_2 \ddot{\delta}^2(t) \leq 0$$  \hspace{1cm} (19)

According to Lyapunov’s stability theorem, the closed-loop system is Lyapunov stable.\cite{43,44} Moreover, by LaSalle’s invariance principle,\cite{43,44} it can be deduced that the generalized positioning error $\delta(t)$ of the load will converge to zero, that is, the driving positioning error tends to zero and the swing angle of the load is suppressed and eliminated.

Design of disturbance observer

The payload generalized position tracking algorithm (GPTC) can realize the anti-sway and positioning of the system, however, the disturbance suppression performance is not satisfactory. Here, a disturbance observer is added to enhance the performance of the GPTC feedback controller, which is shown in Figure 2.

In this figure, $G_{ao}(s)$ is the inverse of the nominal model of the system. $Q(s)$ is designed as a lowpass filter with unit steady-state gain. It can obtain:

$$A'(s) = A(s) - D(s) + D(s)$$  \hspace{1cm} (20)
where $\hat{D}(s)$ denotes the disturbance estimations; $D(s)$ represents the external disturbances; $A(s)$ is the feedback control input and $A'(s)$ is the actual control action. When the disturbance estimations $D(s)$ is equal to the external disturbances $D(s)$, $A'(s)$ is equal to the feedback input $A(s)$, which reduces the influence of the external disturbances on the system to a minimum. According to Figure 2, we can draw:

$$X(s) = G_X(s)A(s) + G_X(s)D(s) \quad (21)$$

From Figure 2, $G_{Ms}$, $G_{Md}$ are calculated as:

$$\begin{align*}
G_{Xs}(s) &= \frac{X(s)}{D(s)} = \frac{G_{Xs}(s)G_{Xs}(s)}{G_{Xs}(s) + [G_{Xs}(s) - G_{Xs}(s)]Q(s)} \\
G_{X'}(s) &= \frac{X(s)}{P(s)} = \frac{G_{X'}(s)G_{X'}(s)[1-Q(s)]}{G_{X'}(s) + [G_{X}(s) - G_{X}(s)]Q(s)}
\end{align*} \quad (22)$$

In the bridge crane system, the frequencies of external disturbances are mainly concentrated in the low frequency band. If the frequency band of the low-pass filter $Q(s)$ is $f_q$, the system state can be divided into two cases:

1. when $f \leq f_q$, $Q(s) \approx 1$, substituting into $G_{Xs}(s)$, $G_{X'}(s)$, then:

$$X(s) = G_n(s)A(s) \quad (23)$$

2. when $f > f_q$, $Q(s) \approx 0$, substituting into $G_{Xs}(s)$, $G_{X'}(s)$, obtain:

$$X(s) = G_p(s)A(s) + G_p(s)D(s) \quad (24)$$

From equations (23) and (24), the output expression with disturbance observer does not contain the disturbance term $D(s)$ in the frequency band below $f_q$. It means that the external low frequency disturbances are suppressed and the system will not be affected by external interferences.

In summary, the disturbance observer can realize the function of suppressing disturbances without affecting the control performance of the system. This feature can be utilized to design the disturbance observer and the anti-sway feedback controller separately.

Considering that the time-varying parameters of the actual model of bridge crane will affect the stability of the disturbance observation error system, it is necessary to determine the conditions that can ensure the stability of the disturbance observer. $G_n(s)$ is the nominal model of $G_p(s)$,

$$G_p(s) = G_n(s)(1 + \Delta(s)) \quad (25)$$

where $\Delta(s)$ is a variable representing the transfer function’s uncertain factors. According to the robust stability theorem, the stability of the system with disturbance observer needs to meet the following conditions:

$$\| T(s)\Delta(s) \|_\infty < 1 \quad (26)$$

where $T(s)$ represents the complementary sensitivity function of the disturbance observer. The relationship between $T(s)$ and the sensitivity function $SF(s)$ is: $T(s) + SF(s) = 1$. The sensitivity $SF(s)$ represents the sensitivity of the closed-loop system to the plant parameters varying during the process. It is defined as the ratio of the change rate of the transfer function $G_{Xs}(s)$ to the change rate of the plant transfer function $G_n(s)$. Moreover, the complementary sensitivity function $T(s)$ can be obtained from the noise channel transfer function:

$$T(s) = \frac{G_p(s)Q(s)}{G_n(s) + [G_p(s) - G_n(s)]Q(s)} \quad (27)$$

From equation (26), the sufficient condition that the disturbance observer is stable can be rewritten as:

$$\| \frac{G_p(s)Q(s)}{G_n(s) + [G_p(s) - G_n(s)]Q(s)}\Delta(s)\|_\infty < 1 \quad (28)$$

Therefore, the key point of the disturbance observer is to design the low-pass filter $Q(s)$. It is known that H.S. Lee has proposed a low-pass filter:

$$Q(s) = \frac{\sum_{i=0}^{M} a_i s^i}{(\tau s + 1)^N} \quad (29)$$

where $N$ and $M$ represent the order of the denominator and numerator, respectively. Considering the physical implementation and the system’s ability and stability to suppress external interferences, let $N = 1$ and $M = 0$ and the low-pass filter $Q(s)$ is designed as:

$$Q(s) = \frac{1}{(\tau s + 1)^2} \quad (30)$$

The system transfer function can be obtained from the linearized model of the system (1):

$$\begin{align*}
\theta(s) &= -\frac{1}{k s^2 + g} \\
\gamma(s) &= \frac{1}{s^2}
\end{align*} \quad (31)$$

In the bridge crane system, the disturbance estimate generated from the disturbance observer is added to the position input and the system nominal model is taken as $G_n(s) = s^2$. According to the low-pass filter $Q(s)$
and the inverse of the system nominal model \( G_n^{-1}(s) \), the disturbance observer is designed. The system control block diagram is shown in Figure 3.

**Performance comparisons**

In this part, the disturbance suppression responses of the proposed method for the anti-sway and positioning controls are studied under the nominal case and the model mismatch case. Here, the target transport distance of the crane is set to 3.5 m, and the length of the rope is set as 3.5 m.

**Nominal Case without disturbances**

In this case, \( G_p(s) = G_n(s) \) holds and no external disturbances are added. The angle curve and the position curve under the proposed GPTC-DOB method are shown in Figure 4(a) and (b). It can be obtained that the anti-sway and positioning control is effectively achieved and the swing angle at the target position is small. The final position of crane is 3.50 m. The maximum angle is 0.03142 rad and the minimum is –0.03168 rad. The curves of acceleration speed and velocity are shown in Figure 4(c) and (d).

**Nominal case with disturbances**

Next, the disturbance suppression performance of the proposed GPTC-DOB method and the baseline GPTC are verified. In the crane control system, the friction force fluctuates continuously when the crane slides on the track. Therefore, both the step disturbance response and sinusoidal disturbance response are evaluated and presented.

**Step disturbances.** It is assumed that step disturbances occur from the \( t = 5 \) s to \( t = 8 \) s, such as the friction coefficient changes on one part of track. Figure 5(a) and (b) illustrate the angle curves and position curves under the step disturbances. It is clear that smaller swing angle amplitude, especially at the target location, can be obtained under the proposed GPTC-DOB method than the baseline GPTC method. Figure 5(c) and (d) show the curve of the acceleration and velocity. The external disturbances added and the estimations are presented in Figure 6. It can be seen that the proposed method can effectively estimate the external disturbances and achieve small estimation error.

The quantified test results are shown in Table 1. The performance indices include the range of angle variation and the integral of absolute angle error (IAAE).

After the step disturbances are added at \( t = 5 \) s, compared with that without DOB (GPTC method), the range of angle variation is reduced by 11.86% and the integral of absolute angle error (IAAE) can be largely decreased (nearly 50%) under the proposed GPTC-
From Figure 5(a) and Table 1, the proposed GPTC-DOB method has better disturbance rejection performance. In this case, sinusoidal external disturbances ($v = 1.0\text{rad/sec}$) is added from $t = 5\text{s}$. The angle curves and position curves are shown in Figure 7(a) and 7(b). It can be seen that the proposed method possesses much smaller angle fluctuation amplitude than that without DOB in the face of sinusoidal disturbances. The curves of acceleration and velocity are presented in Figure 7(c) and (d). The sinusoidal disturbances and their estimations are depicted in Figure 8. Table 2 gives the quantified test results. From Table 2, the proposed method significantly decreases both the IAAE value and the range of angle variation.

**Model mismatch case with external disturbances**

In the anti-sway positioning control system of bridge crane, apart from external disturbances, internal model mismatch is also the key factor affecting control performance. This section will demonstrate lumped disturbance suppression performance.

In general, the length of wire rope has measurement error. Assume the angle channel model is as follows:

$$u(s) = \frac{a(s)}{C_0} \frac{1}{s^2 + g}$$

Comparing (32) with (31), model mismatch exists.

**Step disturbances**. The step external disturbances are also added from the $t = 5\text{s}$ to $t = 8\text{s}$. Figure 9(a) and (b) present the angle curves and position curves of the crane system under the two methods. The curves of acceleration and velocity are shown in Figure 9(c) and (d). Table 3 gives the quantified test results. It can be
obtained from Figure 9(a) and Table 3 that much smaller angel fluctuation amplitude and IAAE value can be achieved by the proposed GPTC-DOB method than baseline GPTC method even in the model mismatch case. The range of angle variation is reduced by 13.06% and IAAE value is largely decreased (nearly 50%).

Sinusoidal external disturbances. In this case, apart from model mismatch, sinusoidal external disturbances are added from $t = 5$ s. Figure 10(a) and (b) present the angle curves and position curves under the two methods. The curves of acceleration and velocity are shown in Figure 10(c) and (d). The quantified test results are given in Table 4. It is clear that the proposed GPTC-DOB method obtains much smaller angel fluctuation amplitude and IAAE value than GPTC method, which

**Table 3.** Quantified test results of GPTC-DOB and GPTC under the step disturbances in the mismatch case.

| Algorithm        | GPTC-DOB | GPTC    |
|------------------|----------|---------|
| Integral of absolute angle error after the disturbances occur (IAAE, rad) | 0.04879  | 0.09386 |
| Range of angle variation after the disturbances occur (rad) | $-0.007775$ to $0.03271$ | $-0.008968$ to $0.03662$ |

**Table 4.** Quantified results for simulation tests of GPTC-DOB and GPTC under the sinusoidal disturbances in the model mismatch case.

| Algorithm        | GPTC-DOB | GPTC    |
|------------------|----------|---------|
| Integral of absolute angle error after the disturbances occur (IAAE, rad) | 0.04879  | 0.09386 |
| Range of angle variation after the disturbances occur (rad) | $-0.007859$ to $0.03271$ | $-0.008968$ to $0.03662$ |

**Sinusoidal external disturbances.** In this case, apart from model mismatch, sinusoidal external disturbances are added from $t = 5$ s. Figure 10(a) and (b) present the angle curves and position curves under the two methods. The curves of acceleration and velocity are shown in Figure 10(c) and (d). The quantified test results are given in Table 4. It is clear that the proposed GPTC-DOB method obtains much smaller angle fluctuation amplitude and IAAE value than GPTC method, which
means that it exhibits better performance in rejecting the lumped disturbances.

**Conclusion**

In the bridge crane system, anti-sway and positioning control is of significant importance for enhancing transport efficiency and safety. However, various disturbances, including external disturbances and internal model mismatches, degrade the control performance. In this paper, the load-based generalized position tracking controller combined with the disturbance observer is proposed, which can estimate disturbances and give feedforward compensation. Test results show that the
The proposed method obtains much smaller angel fluctuation amplitude and IAAE value, and indicate significant disturbance suppression improvements.

Declaration of conflicting interests
The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding
The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported by National Key R&D Program of China (No. 2018YFC1506900), Zhishan Youth Scholar Program of SEU, the Fundamental Research Funds for the Central Universities under Grant 2242020R40032, the Key R&D Program of Jiangsu Province (No. BE2017076, BE2019052), the Key R&D industrialization Program of Suzhou (No. SGC201733, SGC201854).

ORCID ID
Dan Niu https://orcid.org/0000-0002-0715-7946

References
1. Huang J, Ri S, Fukuda T, et al. A disturbance observer based sliding mode control for a class of underactuated robotic system with mismatched uncertainties. IEEE Trans. Autom. Control 2019; 64(6): 2480–2487.
2. Uchiyama N, Ouyang H and Sano S. Simple rotary crane dynamics modeling and open-loop control for residual load sway suppression by only horizontal boom motion. Mechatronics 2013; 23(8): 1223–1236.
3. LaVD and Nguyen KT. Combination of input shaping and radial spring-damper to reduce tridirectional vibration of crane payload. Mech Syst Signal Process 2019; 116: 310–321.
4. Chang W-J and Lin Y-H. An approach to robust fuzzy control for TORA systems with Takagi-Sugeno fuzzy model subject to multiple constraints. Proc Int Conf Fuzzy Theory Appl (iFUZZY) 2017; 11: 1–6.
5. Wu X. Xu K and He X. Disturbance-observer-based nonlinear control for overhead cranes subject to uncertain disturbances. Mech Syst Signal Process 2020; 139: 18–36.
6. Ramli L, Mohamed Z and Jaafar HI. A neural network-based input shaping for swing suppression of an overhead crane under payload hoisting and mass variations. Mech Syst Signal Process 2018; 107: 484–501.
7. Maghsoudi MJ, Mohamed Z, Sudin S, et al. An improved input shaping design for an efficient sway control of a nonlinear 3D overhead crane with friction. Mech Syst Signal Process 2017; 92: 364–378.
8. Wu Z and Xia X. Optimal motion planning for overhead cranes. IET Control Theory Appl 2014; 8(17): 1833–1842.
9. Zhang X, Fang Y and Sun N. Minimum-time trajectory planning for underactuated overhead crane systems with state and control constraints. IEEE Trans Ind Electron 2014; 61(12): 6915–6925.
10. Sun N, Wu Y, Chen H, et al. An energy-optimal solution for transportation control of cranes with double pendulum dynamics: design and experiments. Mech Syst Signal Process 2018; 102: 87–101.
11. Wu X and He X. Enhanced damping-based anti-swing control method for underactuated overhead cranes. IET Control Theory Appl 2015; 9(12): 1893–1900.
12. Sun N, Fang Y and Zhang X. Energy coupling output feedback control of 4-DOF underactuated cranes with saturated inputs. Automatica 2013; 49(5): 1318–1325.
13. Ramli L, Mohamed Z, Efe MO˘, et al. Efficient swing control of an overhead crane with simultaneous payload hoisting and external disturbances. Mech Syst Signal Process 2020; 135: Article 106326.
14. Miranda-Colorado R and Aguilar LT. A family of anti-swing motion controllers for 2-DOF cranes with load hoisting/lowering. Mech Syst Signal Process 2019; 133: Article 106253.
15. Zhang M, Ma X, Song R, et al. Adaptive proportional–derivative sliding mode control law with improved transient performance for underactuated overhead crane systems. IEEE/CAA J Autom Sin 2018; 5(3): 683–690.
16. Sun N, Fang Y, Chen H, et al. Adaptive nonlinear crane control with load hoisting/lowering and unknown parameters: design and experiments. IEEE/ASME Trans Mechatron 2015; 20(5): 2107–2119.
17. Zhang Z, Wu Y and Huang J. Differential-flatness-based finite-time anti-swing control of underactuated crane systems. Nonlinear Dyn 2017; 87(3): 1749–1761.
18. Ouyang H, Wang J, Zhang G, et al. Novel adaptive hierarchical sliding mode control for trajectory tracking and load sway rejection in double-pendulum overhead cranes. IEEE Access 2019; 7: 10353–10361.
19. Tuan LA, Lee S-G, Ko DH, et al. Combined control with sliding mode and partial feedback linearization for 3D overhead cranes. Int J Robust Nonlinear Control 2014; 24(18): 3372–3386.
20. Chwa D. Sliding-mode-control-based robust finite-time anti-sway tracking control of 3-D overhead cranes. IEEE Trans Ind Electron 2017; 64(8): 6775–6784.
21. Ngo QH and Hong K-S. Adaptive sliding mode control of container cranes. IET Control Theory Appl 2012; 6(5): 662–668.
22. Almutairi NB and Zribi M. Sliding mode control of a three-dimensional overhead crane. J Vib Control 2009; 15(11): 1679–1730.
23. Zhang M, Zhang Y and Cheng X. Finite-time trajectory tracking control for overhead crane systems subject to unknown disturbances. IEEE Access 2019; 7: 55974–55982.
24. Chen H, Fang Y and Sun N. A swing constraint guaranteed MPC algorithm for underactuated overhead cranes. IEEE/ASME Trans Mechatron 2016; 21(5): 2543–2555.
25. Yokoyama K and Takahashi M. Dynamics-based nonlinear acceleration control with energy shaping for a mobile inverted pendulum with a slider mechanism. IEEE Trans Control Syst Technol 2016; 24(1): 40–55.
26. Wu X and He X. Nonlinear energy-based regulation control of three dimensional overhead cranes. IEEE Trans Autom Sci Eng 2017; 14(2): 1297–1308.
27. Singhose W, Porter L, Kenison M, et al. Effects of hoisting on the input shaping control of gantry cranes. Control Eng Pract 2000; 8: 1159–1165.
28. Moustafa KAF, Harib KH and Omar F. Optimum controller design of an overhead crane: Monte Carlo versus pre-filter-based designs. *Trans Inst Meas Control* 2013; 35(2): 219–226.

29. Sun Z, Bi Y, Zhao X, et al. Type-2 fuzzy sliding mode anti-swing controller design and optimization for overhead crane. *IEEE Access* 2018; 6: 51931–51938.

30. Chen Y, Niu D and Li Q. An anti-sway positioning algorithm of unmanned crane based on ANFIS. *CCC*, 2020, accepted.

31. Li S, Yang J, Chen WH, et al. Disturbance observer-based control: methods and applications. Boca Raton: CRC Press, 2014. FL33487-2742.

32. Li S, Sun H, Yang J, et al. Continuous finite-time output regulation for disturbed systems under mismatching condition. *IEEE Trans Autom Control* 2015; 60(1): 277–282.

33. Cai HB, Li P, Su CL, et al. Double-layered nonlinear model predictive control based on Hammerstein-Wiener model with disturbance rejection. *Meas Control* 2018; 51(5–6): 192–201.

34. Fang HQ, Yuan XJ and Liu P. Active-disturbance-rejection-control and fractional-order-proportional-integral-derivative hybrid control for hydroturbine speed governor system. *Meas Control* 2018; 51(7–8): 260–275.

35. Sun N, Fu Y, Yang T, et al. Nonlinear motion control of complicated dual rotary crane systems without velocity feedback: design, analysis, and hardware experiments. *IEEE Trans Autom Sci Eng* 2020; 17(2): 1017–1029.

36. Li S, Li J and Tang Y. Model-based model predictive control for a direct-driven permanent magnet synchronous generator with internal and external disturbances. *Trans Inst Meas Control* 2019. DOI: 10.1177/0142331219878574.

37. Zhao Z, Li C, Yang J, et al. Output feedback continuous terminal sliding mode guidance law for missile-target interception with autopilot dynamics. *Aerospace Sci Technol* 2019; 86: 256–267.

38. Yang T, Sun N, Chen H, et al. Neural network-based adaptive anti-swing control of an underactuated ship-mounted crane with roll motions and input dead zones. *IEEE Trans Neural Netw Learn Syst* 2020; 31(3): 901–914.

39. Zhao Z, Yang J, Li S, et al. Composite nonlinear bilateral control for teleoperation systems with external disturbances. *IEEE/CAA J Autom Sin* 2019; 6(5): 1220–1229.

40. Li S, Zhang K, Li J, et al. On the rejection of internal and external disturbances in a wind energy conversion system with direct-driven PMSG. *ISA Trans* 2016; 61(1): 95–103.

41. Zhu Y, Niu D, Li Q, et al. Anti-shake positioning algorithm of bridge crane based on phase plane analysis. *J Eng* 2019; 22: 8370–8373.

42. Sun N, Wu Y, Fang Y, et al. Nonlinear antiswing control for crane systems with double-pendulum swing effects and uncertain parameters: design and experiments. *IEEE Trans Autom Sci Eng* 2018; 15(3): 1413–1422.

43. Sun N and Fang Y. New energy analytical results for the regulation of underactuated overhead cranes: an end-effector motion-based approach. *IEEE Trans Ind Electron* 2012; 59(12): 4723–4734.

44. Khalil HK. *Nonlinear systems*. 3rd ed. Englewood Cliffs, NJ: Prentice-Hall, 2002.

45. Horowitz I. Survey of quantitative feedback theory (QFT). *Int J Control* 1991; 53(2): 255–291.

46. Lee HS. Robust digital tracking controllers for high-speed/high-accuracy positioning systems. Ph.D. Dissertation, Mech. Eng. Dep, Univ. California, Berkeley, 1994.