Vibration Control and Smart Integrated Transportation Strategy of Crane Systems

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Abstract. This paper presents number of vibration control schemes and smart integrated transportation strategies for crane systems. First, robust control problem of payload’s skew rotation in crane systems is discussed. A robust integral sliding mode controller is proposed to realize skew transportation without residual vibration. Robust stability conditions with respect to parametric uncertainties of the closed-loop system are imposed on control gains. Second, transportation of bulk material using an overhead crane system is considered, in which the transferred material can be dropped/discharged while in the air. This paper introduces a new concept, named tossing control methodology, to enhance transportation productivity. A specific type of tossing controller is established, which relies on the phenomenon of linear resonance. It will be shown that the resonance-based tossing control is able to break the time limitation of the minimum-time vibration suppression control. Simulation and experimental results are provided to verify the effectiveness of the proposed control systems. Finally, a diffusion-based obstacle-avoidance path planning method for crane systems is outlined to realize a fully autonomous crane in future.

1. Introduction
Cranes are the most important means of transportation in various places such as factories, shipyards, construction sites, and especially harbors, to transfer heavy and hazardous material point-to-point within the workspace. The 21st century witnessed a revolution of maritime industry, where the containerization played a key role. Every year, billion tons of merchandise and containers are transported by sea and it is generally accepted that more than 90% of global trade belongs to seaborne commerce. In order to transload such an enormous and annually increasing amount of goods, crane systems are getting larger in size and in lifting capacity. They are also being designed to operate at a much higher productivity than before to shorten transloading time. As a result, transportation cost can be significantly reduced. The main focus factor in the pursuit of improving productivity of cranes is the oscillation of the payload around its equilibrium. The vibration phenomenon is a natural characteristic of every crane. The central idea here is to control that vibration as our will to suit for different engineering purposes. At present, the main stream of studies focuses on the vibration suppression control, in which the vibration is seen as an unwanted phenomenon that needs to be eliminated. However, the vibration might be undesirable for some applications but favorable for others (e.g., bulk material transportation described in Section 3). Crane vibration control should therefore also be understood in the sense of vibration excitation control, not just in the direction of...
vibration suppression control. Due to its significance in fostering both theory and application advancements, vibration control of cranes attracted considerable attention from both control theorists and control engineers. The last 50 years have seen a tremendous interest and effort in modeling and control of crane systems. Since the accumulative literature is too vast to list every single detail of all efforts made on the topic, only swing suppression control of the overhead crane system will be subsequently reviewed. It can be classified into two main classes: feedback and feedforward controls. These types can be combined to obtain a two degrees of freedom control system which could give a better control performance.

An extensive background can be found for the feedback control category [1]. Naturally, vibration suppression control of the overhead crane evolved along with the development of control theory field. New achievements of the linear controls and model predictive controls are promptly employed on crane systems in [2–4] and [5] respectively. Feedback linearization and differential-flatness based methodology [6], which aims at transforming the nonlinear model to a linear one to utilize well-established linear controls, are also widely applied [7, 8]. To deal with parametric uncertainties and unmodeled dynamics, sliding-mode based controls [9–12], adaptive controls [13–15], energy and passivity-based nonlinear controls [16, 17]. Intelligence—fuzzy and neural network—controls are also successfully implemented on the crane systems [18, 19]. To sum up, feedback control techniques showed their excellent properties in handling parametric uncertainties and external disturbances. However, they demand a sophisticated and costly sensor structure which requires regular maintenance. In numerous circumstances, feedforward controls are preferable in industry due to their sensorless nature and ease of implementation. In the feedforward control category, the motion trajectory of actuators is planned such that the payload arrives at the target position without residual vibration. The swing oscillation information will not be used during the transfer process. In order to do so, one can utilize the input shaping technique [20–23], S-curve commands [24–28], optimal controls [29–32], linear/nonlinear programming algorithms [33, 34]. The main disadvantage of the feedforward control is that the robustness to parametric uncertainties and disturbances are difficult to be assured.

Skew orientation control of the payload is crucial in practical cranes when the payload (e.g. a container) cannot be considered a point mass. It is also important for an obstacle avoidance application when crane systems work in a narrow workspace. Despite urgent requirements from the harbor industry and in contrast to the well-established anti-swing controls, there are only a few studies paying attention to design payload anti-skew oscillation controllers. Therefore, we will introduce a robust integral sliding mode control for the skew rotation process of crane systems, which is elaborated in Section 2. This paper will also focus on the development of a tossing controller to further enhance transfer productivity of crane systems in the context of bulk material transportation, in which the transferred material can be dropped/discharged while in the air. By imposing upon such a special property, one can break the time limitation of the minimum-time swing suppression control—the fastest controller in the swing suppression control group. Theoretical developments and results of the tossing controller are given in Section 3. In Section 4, to realize a fully autonomous crane in future, a diffusion-based path planning method is outlined to address the obstacle avoidance task.

2. Skew rotation control of crane systems

Skew orientation control of a payload plays an important role in crane systems. For instance, consider the overhead crane system shown in Fig. 1 where the payload’s center of gravity $G$ has arrived at the correct position in the workspace without any oscillation. However, when the target orientation is taken into account, the transferring process is said to be incomplete since there exists a skewing mismatch angle $\beta$ between the payload’s orientation and its desirable pose. Therefore, it is essential to rotate the payload at a skew angle of $\beta$ along the Z axis in the counter-clockwise direction to accomplish the transferring task. A similar issue is also
encountered in boom crane systems in use at harbors where the skew orientation of the container must be adjusted to comply with that of the truck or vessel [11].

In order to fully adjust the skew orientation of the payload, crane systems utilize a simple device called rotary hook. This device is able to directly control the skew angle of the payload by a servo motor which is mounted inside its housing. As shown in the zooming area of Fig. 1, the payload is directly attached to the motor shaft and the entire suspended load is hung below the trolley through two flexible ropes. By using this kind of direct mechanism, the skewing orientation can be controlled to any desired value (> 360 degrees). However, along with the above advantage, after reaching the desired angle, the payload will continue to oscillate rather than stop at the designated skew angle due to the law between inertia force and rope tensions. In practice, the payload inertia is large and the damping ratio of the system is low hence an artificial oscillation suppression needs to be imposed. Therefore, an automatic controller should be developed to reduce the workload for the crane operator as well as drastically enhance productivity of the system.

2.1. Mathematical modeling and validation

The skew residual oscillation resulting from the rotation of the payload is depicted in Fig. 2, where $\varphi(t)$ denotes the relative angle between the payload and the hook. Moreover, $\theta(t)$ and $\gamma(t)$ represent the absolute angles between the hook and the payload versus $x$ axis respectively. Therefore, two aforementioned control objectives can be translated to $\varphi(t) \to \varphi_d$ and $\theta(t) \to 0$, where $\varphi_d$ is a reference skew angle of the payload. The dynamics of the skew rotation process was established by using the Lagrangian mechanics under a holonomic constraint, which can be
elaborated as follows

\[
\ddot{\theta} = f + bu,
\]

where

\[
f = \frac{1}{I_L + I_H + \frac{mR^4\sin^2\theta}{a^2}} \times \left[ \frac{-mR^2\sin\theta}{a} \left( \frac{R^4\dot{\theta}^2\sin^2\theta}{a^3} + \frac{R^2\dot{\theta}^2\cos\theta}{a} + g \right) - c\dot{\theta} \right]
\]

\[
b = \frac{I_L}{I_L + I_H + \frac{mR^4\sin^2\theta}{a^2}}
\]

\[
u = \ddot{\varphi}_d,
\]

and \(a = (l^2 - 2R^2(1 - \cos\theta))^{1/2}\) is the holonomic constraint due to inelastic rope. In (1), \(I_L\) is the inertia of the payload: \(I_L \in [2.64, 41.89] \text{ kgm}^2\); \(I_H\) is the inertia of the hook: \(I_H \in [1, 1.5] \text{ kgm}^2\); \(m\) is the total mass of the payload and the hook: \(m \in [77.39, 207] \text{ kg}\); \(R\) is the skewing radius: \(R = 0.25 \text{ m}\); \(l\) is the rope length: \(l \in [2.5, 5.5] \text{ m}\); \(c\) is the viscous friction coefficient: \(c \in [0.1, 0.5] \text{ Ns/rad}\); and \(g\) is the gravity acceleration: \(g = 9.81 \text{ m/s}^2\).

The formulated mathematical model is now validated on an experimental apparatus, which is described in Fig. 3. A representative example of the model validation process is illustrated in Fig. 4. It can be observed that the experimental data are in complete agreement with the mathematical model, which verifies the correctness of the established dynamical model.

The actual specification of the rotary hook system in use at the harbor is characterized by wide parametric uncertainties; hence, without a robustness consideration, the controller will likely cause instability to the closed-loop system. Therefore, a robust nonlinear integral sliding mode control (ISMC) is subsequently proposed as a vibration controller for the payload’s skew rotation process to cope with parametric uncertainties in the system parameters.

2.2. Integral sliding mode control design

Denoting the reference trajectory of the payload angle as \(\varphi_d(t)\). Thus, the tracking error term can be expressed by \(e(t) = \varphi(t) - \varphi_d(t)\). The coupled integral sliding function is proposed as
follows

\[ \sigma = \beta_1 \dot{e} + \beta_2 e + \int_0^t (\beta_3 \dot{e}(\tau) + \beta_4 \dot{\theta}(\tau)) \, d\tau + \beta_5 \dot{\theta} + \beta_6, \]

(2)

where \( \beta_1, \ldots, \beta_6 \in \mathbb{R} \). By adding the constant term \( \beta_6 \) into the sliding function, it is expected to completely cancel the reaching phase (i.e. force the system to slide on the sliding surface at the very beginning of time). Therefore, the constraint \( \sigma(t = 0) = 0 \) should be imposed, which implies that

\[ \beta_6 = -\beta_1 \dot{e}(0) - \beta_2 e(0) - \beta_5 \dot{\theta}(0). \]

(3)

By letting \( \dot{\sigma} = -K \text{sgn}(\sigma) \) where \( K \) is a strictly positive real number satisfying \( K > |\beta_1| |\ddot{\phi}_d|_{\text{max}} + \eta \) and \( \eta > 0 \) (\( \eta \) can be arbitrarily small), the sliding mode control law can be obtained as

\[ u_I = -\beta_1^{-1} \left( \beta_2 \dot{e} + \beta_3 e + \beta_4 \dot{\theta} + \beta_5 \dot{\theta} + K \text{sgn}(\sigma) \right). \]

(4)

By choosing a Lyapunov function \( V = \frac{1}{2} \sigma^2 \), it is easy to verify that the control law (4) can render the sliding condition \( \dot{V} = \dot{\sigma} \leq -\eta |\sigma| \). Therefore, the sliding surface \( \sigma = 0 \) becomes an invariant set [35, p. 280]. Note that, with the choice of \( \beta_6 \) as in (3) and the usage of the sliding mode control law (4), the system will be attracted to the invariant set \( \sigma = 0 \) at \( t = 0 \), and thus the reaching phase is actually eliminated.

On the sliding surface \( \sigma = 0 \), the equation \( \dot{\sigma} = 0 \) also holds, hence by defining the state variables as \( x_1 = \theta \), \( x_2 = \dot{\theta} \), \( x_3 = \varphi - \varphi_d \), \( x_4 = \dot{\varphi} - \dot{\varphi}_d \), the closed-loop system under sliding mode is governed as follows

\[
\begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= f - b\beta_1^{-1}(\beta_4 x_1 + \beta_5 x_2 + \beta_3 x_3 + \beta_2 x_4) \\
\dot{x}_3 &= x_4 \\
\dot{x}_4 &= -\beta_1^{-1}(\beta_4 x_1 + \beta_5 x_2 + \beta_3 x_3 + \beta_2 x_4).
\end{align*}
\]

(5)

Figure 4. Validation result of the mathematical model.
It is observed that when the integral sliding surface (2) is used, under the sliding mode, the closed-loop system (5) remains the same order with the original system (1). Next, define the state vector as \( x = [x_1 \ x_2 \ x_3 \ x_4]^\top \), linearized model of the nonlinear autonomous system (5) around the equilibrium point \( x = 0 \) can be described as

\[
\dot{x} = Ax,
\]

where

\[
A = \begin{bmatrix}
0 & -\mu_1 & -\mu_2 & -\mu_3 & -\mu_4 \\
-\beta_1^{-1}\mu_4 & 0 & 0 & 0 & 1 \\
-\beta_1^{-1}\mu_5 & -\beta_1^{-1}\mu_5 & -\beta_1^{-1}\mu_5 & -\beta_1^{-1}\mu_2 \\
\end{bmatrix},
\]

therein \( \mu_1 = \omega^2_n + \bar{b}\beta_1^{-1}\mu_4, \mu_2 = c_N + \bar{b}\beta_1^{-1}\mu_5, \mu_3 = \bar{b}\beta_1^{-1}\mu_5, \mu_4 = \bar{b}\beta_1^{-1}\mu_2, \) where \( c_N = c/(I_L + I_H) \), \( \bar{b} = I_L/(I_L + I_H) \), and \( \omega^2_n = (mgR^2)/(l(I_L + I_H)) \). It is easy to check that the linearized closed-loop system is stable regardless of parametric uncertainties in all system parameters if the following conditions are made

\[
\beta_1 > 0, \beta_2 > 0, \beta_3 > 0, \beta_4 > 0, \beta_5 > 0, \beta_2\beta_4 - \beta_3\beta_5 > 0.
\]

Therefore, based on the indirect (or linearization) Lyapunov method [35, p. 55], the nonlinear autonomous system (5) is robustly asymptotically stable at the equilibrium point \( x = 0 \) if the control gains adhere the constraints in (8).

In order to obtain a good control performance while ensuring the robust stability, an optimization routine based on the metaheuristic particle swarm optimization mechanism can be employed, which adopts the robust stability conditions (8) as inequality constraints [11]. The optimized gains of ISMC are

\[
\beta_1 = 1.988, \beta_2 = 10.258, \beta_3 = 0.0058, \beta_4 = 14.136, \beta_5 = 1.058, K = 3.726.
\]

To reduce the chattering in the control signal, the signum function \( \text{sgn}(\sigma) \) in (4) should be replaced by a saturation function \( \text{sat}(\sigma) \) which is defined in (10), therein \( \varepsilon \) is a constant related to the thickness of the boundary layer [35]

\[
\text{sat}(\sigma) = \begin{cases} 
1 & \text{if } \sigma/\varepsilon > 1 \\
-1 & \text{if } \sigma/\varepsilon < -1 \\
\sigma/\varepsilon & \text{if } |\sigma/\varepsilon| < 1.
\end{cases}
\]

2.3. Simulation and experimental results

The designed robust controllers will now be applied on the experimental apparatus in three cases: minimum load case, nominal (or average) load case, and maximum load case. The control structure of ISMC is implemented in the C programming language. The Motion Designer software platform provided by TAMAGAWA SEIKI CO., LTD. is utilized to compose the executing C code. Note that the controller gains of ISMC used in the simulation will be directly employed on the experimental testbed. In the other words, there is not any gains tuning process in our experiments. It is observed that the experimental data of ISMC (see Figs. 5–7) strongly agree with the simulation results in all cases. Moreover, despite the fluctuation in the control input, the sliding functions are kept inside the neighborhood of zero bounded by the designated boundary layer with a thickness of \( \varepsilon \). Due to the effect of inertia, peak of the skew vibration \( \theta(t) \) is smallest in the minimum case and largest in the maximum case, which adheres to the common physical sense.
experimental results are presented to verify the effectiveness of the proposed method.

6 cycle of transportation, it will be shown that the proposed resonance-based tossing control is completely stops, and a large swing angle is obtained. For a specific configuration, in one at which velocities of the payload in both horizontal and vertical directions are zero, the trolley completely stops, and a large swing angle is obtained. For a specific configuration, in one evoking component—is introduced to transfer the payload to the target discharging destination, at which velocities of the payload in both horizontal and vertical directions are zero, the trolley completely stops, and a large swing angle is obtained. For a specific configuration, in one 3. Resonance-based tossing control of the overhead crane system
Transportation of bulk material using an overhead crane system possesses a special property, that is, the transferred material can be dropped/discharged right in the air. This paper will show that, under similar transportation requirements, conditions, and constraints, it is possible for the tossing control methodology, which aims at exciting a large swing angle, provides a considerably faster transferring time than the conventional minimum-time swing suppression control, whose objective focuses on suppressing/minimizing the vibration. In order to induce the oscillation with periodically increasing amplitude, the linear resonance phenomenon is employed. Consequently, a resonance-based tossing controller—comprising of an optimal linearization law and a resonance evoking component—is introduced to transfer the payload to the target discharging destination, at which velocities of the payload in both horizontal and vertical directions are zero, the trolley completely stops, and a large swing angle is obtained. For a specific configuration, in one cycle of transportation, it will be shown that the proposed resonance-based tossing control is 6.3 seconds faster than the minimum-time swing suppression control. Both simulation and experimental results are presented to verify the effectiveness of the proposed method.

Figure 5. Experimental result of ISMC in the minimum load case.

Figure 6. Experimental result of ISMC in the nominal load case.
3.1. Introduction to the tossing control and resonance controller design

The fundamental property of the bulk material transport is that the material can be dropped/discharged from the air and they naturally fall down toward their intended destination by gravity. This is completely different to a large block transfer, in which the payload must be eventually laid down on the ground by the crane. Due to the fact that the rope is flexible, the payload will exhibit vibration during and after the transportation [36]. The residual oscillation is usually undesirable when transferring a large block payload, since, for safety, the swinging motion needs to cease before the payload can be safely landed. Therefore, in such circumstances, vibration suppression control must be emphasized. In the context of bulk material transfer, nevertheless, the vibration suppression control is not necessary owning to its distinct transportation property, namely the material can be discharged right in the air. This indeed opens a possibility in further enhancing the transfer productivity by exciting the payload vibration rather than suppressing/minimizing it. Such a type of control is termed tossing control.

Tossing control and vibration suppression control methodologies have a common objective in driving the payload (grab with material inside) above the desired destination, at which the payload will exhibit vibration during and after the transportation [36]. The residual oscillation is usually undesirable when transferring a large block payload, since, for safety, the swinging motion needs to cease before the payload can be safely landed. Therefore, in such circumstances, vibration suppression control must be emphasized. In the context of bulk material transfer, nevertheless, the vibration suppression control is not necessary owning to its distinct transportation property, namely the material can be discharged right in the air. This indeed opens a possibility in further enhancing the transfer productivity by exciting the payload vibration rather than suppressing/minimizing it. Such a type of control is termed tossing control.

Experimental result of ISMC in the maximum load case.

A mathematical description of the overhead crane system in the case of bulk material transportation is illustrated in Fig. 9, in which $x(t)$ (m) and $\theta(t)$ (rad) are the trolley position and the swing angle. The net force driving the trolley is denoted as $F(t)$. In addition, $m$ (kg), $M$ (kg), $l$ (m), and $g$ (= 9.81 m/s$^2$) represents the payload mass, trolley mass, rope length, and gravitational acceleration respectively. From Fig. 9, horizontal and vertical positions of the payload can be computed by

$$x_p(t) = x(t) - l \sin \theta(t)$$
$$y_p(t) = l \cos \theta(t).$$

By differentiating (11) and (12), horizontal and vertical velocities of the payload are

$$\dot{x}_p(t) = \dot{x}(t) - l \dot{\theta}(t) \cos \theta(t)$$
$$\dot{y}_p(t) = -l \dot{\theta}(t) \sin \theta(t).$$
The dynamics of the overhead crane system in this chapter is similar to that formulated in Section 4.2. By differentiating (6.1) and (6.2), horizontal and vertical velocities of the payload are 

\[ \dot{x} = g \tan \theta - \frac{k \dot{\theta}}{\cos \theta} - \frac{a}{\cos \theta} \sin (\omega_0 t) - \frac{b}{\cos \theta} \cos (\omega_0 t), \]

\[ \dot{y} = \theta. \]

The net force driving the trolley is denoted as 

\[ F = (M + m) \ddot{x} - ml \ddot{\theta} \cos \theta + ml \dot{\theta}^2 \sin \theta \]

\[ l \ddot{\theta} + g \sin \theta = \ddot{x} \cos \theta. \]

The control input of the system is chosen to be \( \ddot{x}(t) \) and only (16) is employed in the subsequent tossing control design. Note that the nonlinear differential equation (16) will be directly used to design the tossing controller without linearizing around the equilibrium point. The reason is that the small angle approximation is not longer valid in the tossing control methodology, as a large swing angle is supposed to be generated.

In order to realize the tossing control requirements, the following terminal conditions must be fulfilled

\[ \dot{x}(t_f) = \dot{x}_p(t_f) = \dot{y}_p(t_f) = 0, x_p(t_f) = \bar{x}_p, \theta(t_f) < 0. \]  

In (17), \( t_f \) is the terminal time and \( \bar{x}_p \) represents the horizontal desired destination of the payload (see Fig. 9). The terminal conditions (17) of the tossing control can easily be transformed to

\[ \dot{x}(t_f) = \dot{\theta}(t_f) = 0, x(t_f) - l \sin \theta(t_f) = \bar{x}_p, \theta(t_f) < 0. \]

Due to the physical limitation of the trolley actuator, the following maximum velocity and maximum acceleration constraints must be strictly complied with

\[ |\dot{x}(t)| \leq v_{\text{max}} \text{ and } |\ddot{x}(t)| \leq a_{\text{max}}, \]

where \( v_{\text{max}} > 0 \) and \( a_{\text{max}} > 0 \). By using the information of the servo motor’s maximum velocity and maximum torque, as well as the structure of the transmission system, the values of \( v_{\text{max}} \) and \( a_{\text{max}} \) can be easily estimated.

The following exact linearization resonance control law is proposed

\[ \ddot{x}(t) = g \left( \tan \theta - \frac{k \dot{\theta}}{\cos \theta} - \frac{a}{\cos \theta} \sin (\omega_0 t) - \frac{b}{\cos \theta} \cos (\omega_0 t) \right), \]

\[ \ddot{\theta}(t) = R(t) - L(t), \]

\[ R(t) = \frac{F(t)}{l}, \]

\[ L(t) = \frac{F(t)}{ml \dot{\theta}^2 \sin \theta}. \]
where $L(t)$ is the linearization law to exactly linearize the nonlinear system (16) and $R(t)$ is the control input for generating the resonance. It is therefore chosen that $\omega_0 = \sqrt{k g / l}$ to invoke resonance. Furthermore, the constant $k$ in (20) is designed to minimize the magnitude of the linearization law $L(t)$. It can be computed by

$$
k = \frac{1}{\cos \theta^* + \theta^* \sin \theta^*},$$

(21)

where $\theta^*$ is determined by

$$
\sin (\bar{\theta} + \theta^*) - \theta^* \cos (\bar{\theta} + \theta^*) = \tilde{\theta}.
$$

(22)

Note that, the parameters $\bar{\theta}, \theta^*$ in (21)–(22), $a, b$ in (20), and the terminal time $t_f$ are unknown, which can be determined by solving (24).

By using the method of undetermined coefficients, a solution of the system (16) under the action of the control input (20) can be expressed by

$$
\dot{\theta}(t) = \frac{b}{2 \omega_0} \sin (\omega_0 t) - \frac{a}{2l} t \sin (\omega_0 t) - \frac{b}{2l} t \cos (\omega_0 t),
$$

(23)

By exploiting (18), to solve for five unknowns ($a, b, \tilde{\theta}, \theta^*, t_f$) in the resonance control law (20), we have the following set of five nonlinear equations

$$
\begin{align*}
-\frac{a}{2l \omega_0} \sin (\omega_0 t_f) + \frac{t_f}{2l \omega_0} [a \cos (\omega_0 t_f) - b \sin (\omega_0 t_f)] &= -\bar{\theta} \\
\frac{b}{\omega_0} \sin (\omega_0 t_f) + t_f [a \sin (\omega_0 t_f) + b \cos (\omega_0 t_f)] &= 0 \\
\int_0^{t_f} \ddot{x}(t) dt &= 0 \\
l \sin \tilde{\theta} - \int_0^{t_f} \ddot{x}(t) dt &= \bar{x}_p \\
\sin (\tilde{\theta} + \theta^*) - \theta^* \cos (\tilde{\theta} + \theta^*) &= \tilde{\theta}.
\end{align*}
$$

(24)

The MATLAB routine `fsolve` can be utilized to solve (24). Note that, in order to strictly comply with the maximum velocity and maximum acceleration constraints, namely $|\dot{x}(t)| \leq v_{\text{max}}$ and $|\ddot{x}(t)| \leq a_{\text{max}}$, the initial guess of the terminal time $t_f$ can be gradually increased until those inequalities are met.

3.2. Simulation result: comparison with minimum-time swing suppression controller

A real gantry crane in use at the harbor is utilized in the simulation study. The maximum velocity and maximum acceleration are $v_{\text{max}} = 1.5 \text{ m/s}$ and $a_{\text{max}} = 0.6 \text{ m/s}^2$. The rope length is $l = 20 \text{ m}$. The desired transfer distance of the payload is $\bar{x}_p = 10 \text{ m}$. By using those quantities, control parameters of the resonance-based tossing control and the minimum-time suppression control [37] can be computed. Accordingly, it takes 17.72 seconds for the proposed resonance tossing control to complete one cycle of transportation, whereas it requires 24.02 seconds in the case of minimum-time swing suppression control. Therefore, under similar transportation requirements, conditions, and constraints, the resonance-based tossing controller is 6.3 seconds faster than the minimum-time swing suppression controller. Simulation results of
Figure 10. Comparative simulation result between the proposed resonance-based tossing control and the minimum-time swing suppression control.

two controllers are shown in Fig. 10. It can be seen that, at the material dropping/discharging time (represented by the solid blue line and the dash red line for the resonance-based tossing control and the minimum-time vibration control respectively), the payload reaches the target destination with zero horizontal and vertical velocities, whereas the trolley completely stops. Therefore, requirements of the transferring phase are fulfilled by both controllers. Moreover, at the end of the returning phase, in both control schemes, the vibration is completely suppressed. To sum up, two controllers satisfy the necessary conditions of the bulk material transport. The main difference between the proposed resonance-based tossing control and the minimum-time vibration control is explained as follows. At the dropping time, the swing angle in the case of tossing controller is $\dot{\theta} = -0.167$ rad (or $-9.57$ degrees), whereas it is zero degree when the minimum-time swing suppression control is employed (see $\theta$ graph in Fig. 10). This characteristic leads to the fact that, at the discharging point, the payload must not be exactly under the trolley in the tossing control case, which is completely in contrast to the minimum-time swing suppression control (see $x$ and $x_p$ graphs of Fig. 10). For this reason, the tossing controller can take full advantage of the long rope length to save the transferring time since the trolley is not required to travel a full distance as it does in the case of vibration control (see subfigure of $x$).

The above discussions explain the result that the proposed resonance-based tossing control is 6.3 seconds faster than the minimum-time vibration control. To sum up, it is proven that, in the case of bulk material transportation, one could further enhance the transfer productivity by using the tossing control methodology instead of the conventional minimum-time swing suppression control.

In order to clearly illustrate the feasibility of the above-mentioned tossing controller, the images captured from an actual footage of a real-size boom crane system, which utilized a similar tossing control methodology proposed in this paper, are shown in Fig. 11.
4. Towards fully automated crane systems

A future crane system must be fully autonomous. In order to realize such a requirement, one need to integrate three main components, namely anti-swing control, anti-skew control, and path planning for obstacle avoidance. The first part (i.e. anti-swing control) was intensively done in the literature. The second part (i.e. anti-skew control) was discussed in Section 2. The final one (i.e. path planning for obstacle avoidance) will be introduced in the subsequent presentation.

One method can be utilized in the obstacle avoidance for crane systems is based on the diffusion process. Assuming that the motion of a real crane can be reduced to a problem of finding a path in known environments, the following physical analogy can be employed for path-planning purposes. A goal point \( G \) of the crane’s collision-free motion path is considered to be the location of a virtual source like a scent or a perfume. While the concentration at point \( G \) is kept constant, the substance diffuses steadily into the surrounding space.

The diffusion process is modeled by an unsteady or dynamic diffusion equation as described by Fick’s second law

\[
\frac{\partial C}{\partial t} = D \left( \frac{\partial^2 C}{\partial x^2} + \frac{\partial^2 C}{\partial y^2} + \frac{\partial^2 C}{\partial z^2} \right),
\]

where \( D \) is the diffusion coefficient. By applying the standard finite difference methods, the following time and space discretized model of (25) can be obtained for a grid point or node

\[
C_{t+1,i,j,k} = r \left( C_{t,i+1,j,k} + C_{t,i,j+1,k} + C_{t,i,j,k+1} + C_{t,i,j,k+1} + C_{t,i,j,k-1} + C_{t,i,j,k-1} + C_{t,i,j,k-1} \right) + (1 - 6r)C_{t,i,j,k}.\]
The boundary conditions of (26) are given by

\[ C_{t,i,j,k} = \begin{cases} 
1 & \text{(Goal point)} \\
0 & \text{(Wall, Obstacles)} 
\end{cases} \]  
(27)

The initial conditions of (26) are

\[ C_{0,i,j,k} = 0 \quad \text{(except for Goal)}, \]  
(28)

where \( r = D\Delta t/(\Delta h)^2 \), \( i \in x, j \in y, k \in z \). The concentration at time \( t \) and for the space \( (i, j, k) \) is \( C_{t,i,j,k} \). The sampling time and the grid size are \( \Delta t \) and \( \Delta h \) respectively. It is chosen that \( \Delta h = 0.05 \) m. Furthermore, the stable region of the coefficient \( r \) is \( 0 < r \leq 1/6 \).

The outline of the diffusion-based path planning strategy can be explained as follows. First, a crane’s (transfer object’s) collision-free path between a start and a mission-dependent goal point is generated by off-line simulation of a diffusion process, and by evaluating the gradient of the computed concentration distribution functions. Second, for both position and swinging-suppression control, command inputs (reference trajectory given at every time-step) for the path obtained in potential fields must be given in order to implement a feedback control system. For the command inputs obtained, a time-varying controller using a fixed-pole approach is applied to obtain the position control and swinging-suppression control.

5. Conclusions

This paper introduced an integral sliding mode controller for the payload’s skew orientation process in crane systems to cope with parametric uncertainties. By using the indirect Lyapunov method, algebraic inequality constraints for the ISMC gains are formulated to ensure the robust stability of the closed-loop system under the sliding mode and in a context where the reaching phase is completely eliminated. Another crucial conclusion of the paper emphasizes on the result that there are possibilities to break the time limitation of the minimum-time swing suppression controller—the fastest member of the swing suppression control family—by using the tossing control methodology. In addition, a diffusion-based obstacle avoidance method for crane systems is also introduced to realize a fully autonomous crane in future.

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