Point-to-point motion control for flexible crane systems working in the deep sea

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Abstract
At present, marine resources, especially the deep-sea resources, are becoming more and more important in resource exploitation globally, and hence, widely used deep-sea cranes are playing essential roles. For such systems, the bridge frames and trolleys are set up above the water, while payloads are transported under the water. In this underwater situation, there exist hydrodynamic forces such as complicated disturbances to the crane systems, making the payload vibration and rope flexibility more obvious. For the sake of improving working efficiency, considering the constraints of all the state variables, an anti-vibration trajectory is designed for the trolley motion, which can not only ensure trolley positioning but also suppress the flexible payload's vibrations. Then, the state variables are constrained within preset safety ranges. Finally, numerical simulation results prove the satisfactory performance of the designed method.

Keywords
Underactuated systems, distributed-parameter systems, flexible payload, trajectory planning, vibration control

Introduction
In these years, as powerful equipment, cranes have been widely applied in architectonics, nuclear power construction, petrochemical industry, port construction, industrial environmental protection, and so on. Therefore, crane systems¹–¹² are playing significant roles in modern industry gradually.

According to various practical applications, cranes can be divided into rotary cranes, tower cranes, gantry cranes, overhead cranes, and so on. In the light of the working environment, there are cranes operating on land and under water. Although their specific models and characteristics are different, they all belong to typical underactuated systems with more degrees of freedom (DOFs) than their independent control inputs.¹³–²² Underactuated systems have stronger non-linearities with variables being coupled with each other, making it more difficult to analyze and solve their control problems compared with fully actuated systems. However, in terms of energy saving, structural complexity, and cost reduction, underactuated systems are superior to fully actuated ones. Aside from cranes, translational oscillations with a rotational actuator (TORA)²³–²⁵ systems and underactuated robot²⁶–²⁸ systems also belong to underactuated systems.

An underwater crane system, as the term suggests, refers to a type of crane whose driving part is set above water with its flexible payload being transported in water. At present, the crane system is greatly used in oil exploration, sea resources exploitation, nuclear reactor, and other fields. In view of the flexible payload and high-precision requirements, the crane’s control objectives are to rapidly and precisely transport the payload to the preset position and also effectively restrict the vibration of the payload throughout the entire transportation process, bringing about zero residual vibration at the end of the process. There are two obvious

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problems that need to be addressed during the payload transportation. On one hand, the vibration of the payload under water will be influenced by the amplification of the hydrodynamic force compared with the one on land, which may cause large-amplitude vibration. On the other hand, in practice, cranes need to be manually operated by operators who are experienced and skilled, but minor operational mistakes caused by manual operation are inevitable, thus increasing the possibility of accidents. Consequently, considering the above two problems, it is demanded to devise a control approach to realize the goal of instantly reaching the desired position and effectively suppressing the residual vibration of the flexible payload simultaneously.

The underactuated systems’ control problems have been studied in academia, and many researchers have made remarkable achievements in recent decades. At present, the two most common existing methods are open-loop control strategies and closed-loop control methods. Specifically, open-loop control strategies include trajectory planning, input shaping, and feedforward control. For closed-loop control, there are many representative approaches such as sliding mode control, fuzzy control, model predictive control, and feedback linearization. In different application scenarios, these two kinds of methods have their respective advantages.

In fact, in some applications, crane systems exhibit complex flexibility; for instance, when the overhead crane uses a flexible cable for payload transportation, the cable exhibits the characteristics of a flexible cord. Recently, the control of flexible systems on land has attracted much attention in the control and mechatronics society. For example, in He and Ge, cooperative control laws are put forward to limit the flexible cable’s deflection. Based on the dynamic model, a singular perturbation method is applied in Peng et al. to control the coupling system. Then, in He et al., a hybrid system is obtained and a well-designed controller is proposed to eliminate vibration. Next, in Xie et al., a smoother is proposed to reduce the flexible payload’s vibration by command smoothing, which achieves vibration reduction of multi-mode systems. Also, in Fatehi et al., two effective controllers are designed on the basis of the singular perturbation method to regulate the crane system with large swing.

The exploitation about marine resources, especially the deep-sea resources, is gradually becoming an increasingly important resource developing trend around the world. As a sequel, deep-sea crane systems are widely applied to exhibit important research values. However, there are few published works on the topic of underwater transportation control. The aforementioned existing methods are designed for flexible systems operating on land without considering underwater situations, which bring much stronger flexibility and are much more challenging. Until now, the research on underwater flexible crane systems is still open. For such systems, the hydrodynamic force will bring larger deflection in the flexible payload under water than on land, that is, producing greater flexibility, causing the payload to vibrate severely, and making the system exhibit much more complicated nonlinear characteristics during transportation. Furthermore, regarding sensors, it is extremely uncertain whether they function well or not in water. Hence, compared with cranes working above water, it is more difficult to achieve precise and efficient transportation for flexible systems in underwater situations. There are some studies about the flexible payload immersed in fluids. For instance, in Shah et al., the system model is calculated and a boundary control method is designed to limit the vibration of the rod. Then, in He et al., a robust adaptive control method is provided to deal with the vibration of the controlled object in water. In Zhao et al., a boundary control method is proposed without simplifying the dynamic equations to restrict the vibration of the flexible riser. To reduce the vibrations in both two directions of the fuel rod, a relatively indirect but comprehensive control law is developed in Shah and Hong to regulate the movement of the trolley and the bridge. In Shah et al., considering the constraint of the rod’s deflection range and a similitude law, a modified input shaping method is developed to limit the flexible rod’s residual vibrations.

In this paper, an anti-vibration polynomial-based trajectory planning method without simplifying and linearizing the dynamic model is proposed. On one hand, the method can restrict the flexible payload’s vibration. On the other hand, it can improve transportation efficiency. Specifically, we divide the velocity of the driving device (e.g., trolley) into three segments: the velocity of the first and last segments is symmetrical, and the velocity of the middle segment is the value allowed by the equipment. Next, the satisfactory performance of the designed method is verified by simulation results. In conclusion, the main contributions of this paper are summarized as follows: (1) without simplifying or linearizing the dynamic model, an anti-vibration polynomial-based trajectory is designed, which can not only meet the requirements of payload transportation but also improve the efficiency; (2) a function $s(t)$ is constructed to obtain the proposed trajectory, which can eliminate the residual vibration; and (3) the proposed method can solve the problem of transporting the flexible payload smoothly in the deep-sea environment, considering complicated hydrodynamics.

The paper is organized as follows. First, in section “Problem statement,” we provide the system model and then summarize the problems to be solved by analyzing the control objective and for the purpose of improving transportation efficiency. Next, in section “Anti-vibration trajectory planning method,” a new planned trajectory, which is designed by planning the trolley’s velocity, is used for this system, and the effectiveness of the designed trajectory is verified by numerical simulations. Finally, in section “Conclusion,” we present a summary about the major work of this paper.
**Problem statement**

Because of the underactuated characteristic of the deep-sea crane system, the vibration at the end point of the payload cannot be directly controlled. In order to ensure accurate and rapid trolley positioning and to suppress the payload’s vibration by constraining the maximum vibration amplitude within a given range, the coupling relationship among state variables must be fully utilized to plan the control input, that is, the acceleration. Furthermore, considering the practice factors, the state variables should meet some specific constraints.

**System model**

Figure 1 shows a two-dimensional (2D) schematic diagram of the underwater crane system, where in the inertial coordinate iOj, \( x(t) \) represents the displacement of the trolley, \( \gamma_j(r,t) \) represents the lateral deviation of the payload at \( r \) and \( t \), \( m_r \) is the mass of the payload, and \( m_t \) is the mass of the trolley. In addition, \( u(t) \) denotes the control input of the system and \( l, I, E, d \), and \( l \) denote the area moment of inertia, Young’s modulus, diameter, and length of the payload, respectively.

Let \( f_w(r,t) \) represent the hydrodynamic force acting on the payload moving in the water. Then, according to Shah et al.,\(^5\) the expression of \( f_w(r, t) \) can be given as follows:

\[
\begin{align*}
  f_w(r, t) &= \frac{\pi}{4} \rho_w C_w(r) d^2 \dot{x}(t) + \frac{1}{2} \rho_d C_d(r) d \dot{v}_r(r, t) \\
  &= \frac{\pi}{2} \rho_w C_w(r) d^2 (\ddot{x}(t) + \dot{\gamma}_j(r, t)) + \frac{1}{2} \rho_d C_d(r) d(\ddot{x}(t) + \dot{\gamma}_j(r, t))(\ddot{x}(t) + \dot{\gamma}_j(r, t))
\end{align*}
\]

In light of Shah et al.,\(^5\) the dynamic model of the system is given as follows:

\[
\begin{align*}
  m_t + m_r \ddot{x}(t) + m \int_0^t \dot{\gamma}_j(r, t) dr &= u(t) \quad (3) \\
  EI \ddot{\gamma}_j^{im}(r, t) + m \ddot{x}(t) + c \dot{\gamma}_j(r, t) &= -m \ddot{x}(t) - f_d(r, t) \quad (4)
\end{align*}
\]

Let \( u(t) = (m_r/4) + (\pi/4) \rho_w C_w(r) d^2 \). When \( r = r_{i0} \), the trolley should arrive at the target point \( p_d \) from the initial position, and the payload vibration should converge to the equilibrium point without obvious oscillation, that is, the velocity, acceleration, and jerk of the trolley, as well as the vibration velocity, vibration acceleration, and vibration jerk of the payload, all should be zero at \( T \) and is given by

\[
\begin{align*}
  x(0) &= 0, \dot{x}(0) = 0, \ddot{x}(0) = 0, \dot{\gamma}_j(0) = 0, \\
  x(T) &= p_d, \dot{x}(T) = 0, \ddot{x}(T) = 0, \gamma_j(r, T) = 0, \gamma_j(r, 0) = 0, \dot{\gamma}_j(r, 0) = 0, \\
  \ddot{\gamma}_j(r, T) &= 0, \gamma_j(r, T) = 0, \gamma_j(r, 0) = 0, \dot{\gamma}_j(r, 0) = 0, \ddot{\gamma}_j(r, T) = 0
\end{align*}
\]

2. Based on hardware restrictions and safety concerns, the state variables, that is, trolley velocity, payload deviation, and so on, should be reasonably limited within a suitable range during the entire transportation process. Therefore, the variables should satisfy the following constraints:

\[
\begin{align*}
  |\dot{x}(t)| &\leq v_{\text{max}}, |\ddot{x}(t)| \leq a_{\text{max}}, \\
  |\dot{\gamma}_j(r, t)| &\leq v_{\gamma_{\text{max}}}, |\ddot{\gamma}_j(r, t)| \leq a_{\gamma_{\text{max}}}
\end{align*}
\]

**Control objective**

Considering the actual operating environment of the deep-sea crane system, the control objective can be described as follows:

1. The time to complete the entire transportation process is defined as \( T \). When \( t = T \), the trolley should arrive at the target point \( p_d \) from the initial position, and the payload vibration should converge to the equilibrium point without obvious oscillation, that is, the velocity, acceleration, and jerk of the trolley, as well as the vibration velocity, vibration acceleration, and vibration jerk of the payload, all should be zero at \( T \) and is given by

\[
\begin{align*}
  x(0) &= 0, \dot{x}(0) = 0, \ddot{x}(0) = 0, \dot{\gamma}_j(0) = 0, \\
  x(T) &= p_d, \dot{x}(T) = 0, \ddot{x}(T) = 0, \gamma_j(r, T) = 0, \gamma_j(r, 0) = 0, \dot{\gamma}_j(r, 0) = 0, \\
  \ddot{\gamma}_j(r, T) &= 0, \gamma_j(r, T) = 0, \gamma_j(r, 0) = 0, \dot{\gamma}_j(r, 0) = 0, \ddot{\gamma}_j(r, T) = 0
\end{align*}
\]

2. Based on hardware restrictions and safety concerns, the state variables, that is, trolley velocity, payload deviation, and so on, should be reasonably limited within a suitable range during the entire transportation process. Therefore, the variables should satisfy the following constraints:

\[
\begin{align*}
  |\dot{x}(t)| &\leq v_{\text{max}}, |\ddot{x}(t)| \leq a_{\text{max}}, \\
  |\dot{\gamma}_j(r, t)| &\leq v_{\gamma_{\text{max}}}, |\ddot{\gamma}_j(r, t)| \leq a_{\gamma_{\text{max}}}
\end{align*}
\]
Table 1. Plant parameters.

| Parameters | Physical meaning | Unit |
|------------|------------------|------|
| \( l \)    | Payload length   | m    |
| \( d \)    | Payload diameter | m    |
| \( m_p \)  | Payload mass     | kg   |
| \( m_t \)  | Trolley mass     | kg   |
| \( \omega_1 \) | Natural frequency | rad/s |
| \( E \)    | Young’s modulus  | GPa  |
| \( \rho_m \) | Density of the payload | kg/m³ |
| \( \rho_w \) | Density of water  | kg/m³ |
| \( c \)    | Viscous damping coefficient | N·s/m |
| \( C_d \)  | Added-mass coefficient | – |
| \( C_d \)  | Drag coefficient  | – |


where \( v_{max}, a_{max}, f_{max}, \gamma_{max}, v_{\gamma_{max}} \), and \( a_{\gamma_{max}} \) denote the maximum values within the allowed range of the corresponding variables.

**The optimal time under certain conditions**

To improve the efficiency of the system, the value of \( T \) is expected to be as small as possible, on the premise of satisfying the listed conditions. Taking all the constraints into consideration, we can formulate and solve the following problem to obtain the most appropriate time

\[
\begin{align*}
\text{min} & \quad T, \\
\text{s.t.} & \quad |\dot{s}(t)| \leq v_{\max}, |\ddot{s}(t)| \leq a_{\max}, \\
& \quad |\gamma(t)| \leq f_{\max}, |\dot{\gamma}(t)| \leq \gamma_{\max}, \\
& \quad |\ddot{\gamma}(t)| \leq v_{\gamma_{\max}}, |\dot{\ddot{\gamma}}(t)| \leq a_{\gamma_{\max}}
\end{align*}
\]

**Anti-vibration trajectory planning method**

For the purpose of simplifying the calculation, only the deviation value of the end point of the flexible payload is considered. According to the practical situation, if the vibration of the end point of the payload meets the requirements, the vibration of the other points must also be within the allowable range. Hence, this paper only considers the state of the end point, which can equivalently capture the state of the entire payload. At this point, \( r \) can be replaced by \( l \), and then the model coupling relationship given by equation (4) is rewritten as follows

\[
\begin{align*}
\ddot{c}(t) &= -\left(\frac{1}{2}\right)\rho_m C_d(r) \dot{d}(t) + \dot{\gamma}(t)|\dot{s}(t) + \dot{\gamma}(t)| \\
&\quad - m\ddot{\gamma}(t) - m\ddot{x}(t)
\end{align*}
\]

(9)

From equation (9), we construct a signal

\[
s(t) = x(t) + \gamma(t)
\]

(10)

By substituting equation (10) into equation (9), one has that

\[
\ddot{\gamma}(t) = -\frac{1}{c} \left( m\ddot{s}(t) + \left(\frac{1}{2}\right)\rho_m C_d(r)\ddot{d}(t)|\ddot{s}(t) + \dot{\gamma}(t)| \right)
\]

(11)

\[
\dot{s}(t) = \dot{s}(t) + \left(\frac{1}{2}\right)\rho_m C_d(r)|\ddot{s}(t)|\ddot{s}(t)
\]

(12)

After that, all state variables can be represented by the signal \( s(t) \). Then, trajectory planning can be transformed into the problem of designing \( s(t) \). In the subsequence, we can define \( s(t) \) as a polynomial expression.

**Trajectory planning**

If the velocity of the trolley can reach the maximum value within a limited time and last for a while, the running time will be further shortened and the transportation efficiency will be improved compared with the single-peak velocity trajectory, which is a trapezoid velocity trajectory containing three curve segments (increasing velocity, constant velocity, and decreasing velocity). The maximum velocity of the trolley can be obtained based on the limit range of the deflection, that is, it is feasible to plan the trajectory of the trolley’s velocity.

To realize the control objective, the velocity is divided into three segments, and the acceleration can be guaranteed to be continuous. The first segment is designed as a polynomial from \( t = 0 \) to \( t = t_1 \), the third segment is also a polynomial from \( t = t_2 \) to \( t = T \), and the second segment is a constant \( v_{\max} \) from \( t = t_1 \) to \( t = t_2 \). The derivative of \( s(t) \) with respect to \( t \) is expressed as follows

\[
\dot{s}(t) = \ddot{s}(t) + \dot{\gamma}(t)
\]

First, to obtain the velocity of the first segment, \( \ddot{s}(t) \) is designed as follows

\[
\ddot{s}(t) = p_0 + \sum_{i=1}^{6} p_i \eta^i
\]

(13)

where \( \eta = t/t_1, 0 \leq t \leq t_1 \).

The partial equality constraints of \( \ddot{s}(t) \) can be derived based on the constraints of \( x(t) \) and \( \gamma(t) \), which can be expressed as

\[
\ddot{s}(0) = 0, \ddot{s}(0) = 0, \ddot{s}(0) = 0, \\
\ddot{s}(t_1) = v_{\max}, \ddot{s}(t_1) = 0, \ddot{s}(t_1) = 0
\]

(14)

By substituting equation (14) into equation (13), the following equations and coefficient relationships can be obtained

\[
\begin{align*}
p_0 &= p_1 = p_2 = 0 \\
p_3 &= -p + 10 \\
p_4 &= 3p - 15 \\
p_5 &= -3p + 6 \\
p_6 &= p
\end{align*}
\]

(15)

After that, we let
\[ \dot{x}(t) = g_3 \eta^3 + g_4 \eta^4 + g_5 \eta^5 + g_6 \eta^6, \]
\[ \dot{\eta}(t) = h_3 \eta^3 + h_4 \eta^4 + h_5 \eta^5 + h_6 \eta^6 \]  

(16)

Then, the constraints of \( \dot{x}(t) \) and \( \dot{\eta}(t) \) are written as

\[ \dot{x}(0) = 0, \dot{\eta}(0) = 0, \dot{x}(t) = 0, \dot{\eta}(t) = 0, \]  

(17)

By substituting equation (17) into equation (16), the coefficient relationships can be rewritten as follows

\[ \begin{align*}
g_3 &= -g + 10v_m \\
g_4 &= 3g - 15v_m \\
g_5 &= -3g + 6v_m \\
g_6 &= g \\
h_3 &= -h \\
h_4 &= 3h \\
h_5 &= -3h \\
h_6 &= h
\end{align*} \]  

(18)

(19)

Actually, if the initial values of any two of the three parameters are given, we can then obtain the value of the last parameter. Specifically, from equations (10), (13), (15), (16), (18), and (19), we can derive that \( g + h = p \); that is to say, if the two parameter values are given, we can obtain the remaining one. In addition, according to the physical significance of \( s(t) \), it is the sum of the trolley displacement and the payload endpoint displacement, indicating that \( \dot{x}(t) \geq 0, \dot{s}(t) \geq 0, \) and \( s(t) \leq x(t) \). Then by trial and error, we can adjust the parameters and determine suitable values.

So far, the first velocity expression containing an unknown parameter \( t_1 \) is obtained. Nevertheless, it is necessary to acquire the optimal \( t_1 \) to obtain the first trajectory. It is clear that the transportation efficiency will be improved by reducing \( t_1 \) on the premise of ensuring tracking performances.

To do so, we use the dichotomy to solve the problem in equation (8) and obtain the optimal time. The pseudocode is given in Algorithm 1, where \( T_s \) and \( T_b \in \mathbb{R}^+ \), respectively, represent the minimum and maximum values of \( T \); \( \mu \) denotes the allowable error, which is used as the ending condition of the program and is set artificially as needed; \( v_{\text{max}}, a_{\text{max}}, f_{\text{max}}, \gamma_{\text{max}}, \gamma_{\text{y-max}}, \) and \( a_{\gamma_{\text{max}}} \) are the upper limits of the constraints in equation (7); and \( T^* \) is the optimal value of \( T \).

Based on equation (7), \( t_1 \) can be solved by Algorithm 1. Thus, the velocity trajectory of the first segment is planned. The velocity of the trolley in the second segment can be derived directly and is constant, that is, \( \dot{x}(t) = \text{v}_{\text{max}}(t_2 \leq t \leq t_2) \), where \( t_2 \) is an unknown parameter, which can be obtained by the third segment. In this paper, the terminal velocity and the first segment are symmetrical, the time relationship \( T - t_2 = t_1 \) is satisfied, and the displacement relationship \( P_d = v_{\text{max}}(t_2 - t_1) = 2x(t_1) \) is also satisfied. Since the first expression is known, the final velocity expression can be obtained after mathematical arrangements, and \( t_2 \) and \( T^* \) can also be calculated. Then, the entire trajectory planning process is completed.

### Numerical simulation results

To obtain the deflection values of the flexible payload based on the trolley movement, the finite difference method (FDM) is applied to solve the partial differential equation (PDE) in equation (4). For the simulations, the parameters are set as in Table 2, which are from Shah et al.\(^50\)

Figure 2 indicates the simulation results of the planned anti-vibration trajectory with \( p_d = 1.2 \text{ m}, v_{\text{max}} = 0.35 \text{ m/s}, \) \( a_{\text{max}} = 10, \) and \( g = 3 \). It indicates that the proposed trajectory can guarantee the continuity of the trolley acceleration.

For the purpose of verifying the effectiveness of the planned trajectory, the paper compares the trajectory planning method with the existing input shaping method in Shah et al.\(^50\) The corresponding simulation results are shown in Figure 3. Compared with the input shaping method, what we can see from Figure 3 is that the transportation time of the anti-vibration trajectory planning method is shorter. Therefore, in terms of

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### Algorithm 1: Algorithm for solving optimal time problem in equation (8)

| Input | \( T_s, T_b, v_{\text{max}}, a_{\text{max}}, f_{\text{max}}, \gamma_{\text{max}}, \gamma_{\text{y-max}}, a_{\gamma_{\text{max}}} \) |
|---|---|
| Output | \( T^* \) |
| 1: while \( |T_s - T_b| > \mu \) do |
| 2: \( \text{let } T = (T_s + T_b)/2 \) |
| 3: if the inequality constraints in equation (7) are all satisfied then |
| 4: \( \text{let } T_s = T \) |
| 5: else |
| 6: \( \text{let } T_r = T \) |
| 7: end if |
| 8: end while |
| 9: \( \text{let } T^* = T_r \) |

### Table 2. System parameters.

| Parameters | Value | Unit |
|---|---|---|
| \( l \) | 1.0 | m |
| \( d \) | 0.008 | m |
| \( m_t \) | 0.0369 | kg |
| \( m_r \) | 3.801 | kg |
| \( a_{\text{r}} \) | 9.52 | rad/s |
| \( E \) | 38.9 | GPa |
| \( P_d \) | 1204.5 | kg/m³ |
| \( \rho_t \) | 1000.0 | kg/m³ |
| \( c \) | 0.6 | N/s/m |
| \( C_0 \) | 0.93 | |
| \( C_2 \) | 1.28 | |
transportation efficiency, the proposed trajectory planning method is better than the input shaping method.

Besides, for the input shaping method, it is noticed from Figure 3 that the trolley acceleration, which is the system’s control input, will suddenly change at some point, that is, the driving force will change suddenly. By contrast, this phenomenon is avoided by the proposed method, which is smoother. Moreover, compared with existing methods (e.g. input shaping method), the proposed method ensures that the accelerations are continuous and also protects the actuators in practice.

Figure 4 is a three-dimensional (3D) diagram of the anti-vibration trajectory planning method.

Conclusion

In this paper, with no linearization to the dynamic model, a new anti-vibration polynomial-based trajectory planning method is proposed, which considers the constraints of all state variables while achieving the control objective. The design of the anti-vibration trajectory is to divide the velocity into three segments and make the trolley move at the maximum velocity for longer time, which improves the transportation efficiency greatly. In the end, simulation results verify the performance of the proposed method. In the near future, we will implement hardware experiments for further verification, and other types of trajectories will also be considered apart from polynomials.

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