E-ISSN: 2664-8784
P-ISSN: 2664-8776
Impact Factor: RJIF 8.26
IJRE 2025; 7(2): 184-193

www.engineeringpaper.net
Received: 27-08-2025
Accepted: 30-09-2025

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Position and anti-swing control of gantry crane based on swing angle compensator

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DOI: https://www.doi.org/10.33545/26648776.2025.v7.i2b.144

Abstract
Position and swing angle control of gantry crane systems (GCS) provides high work efficiency such as loading, unloading and carrying of loads in factories, harbors and power plants, etc. In this paper, we propose a new control method for the position and anti- swing of gantry crane. First, the control problem is solved by designing a position controller and combining a compensator for the swing angle with the position control system. On the other hand, the design of the control system is simplified as the position controller is designed by the PI controller. Also, it is possible to simultaneously achieve both the position tracking for rapid carrying and the anti-swing control to reduce the swing angle of the load. We simulate the proposed algorithm in several cases using MATLAB. The simulation results show that the proposed method can significantly reduce the oscillation of the swing angle without allowing the oscillation of the position control response.

Keywords: Gantry crane, anti-swing, position control, compensator, swing angle

Introduction

In a GCS, which requires rapid load transfer, it is important to reduce the inevitable load swing as small as possible.

A DE optimal PID controller for anti-swing in GCS was proposed in [1]. The DE-differential evolution algorithm was used to tune the PID controller parameters offline. The theoretical complexity and the feasibility of the algorithm of various control methods using model predictive control are compared in [2]. Based on the analysis of the simulation of the 2D motion of a gantry crane, membership functions of input and output variables for fuzzy control were designed in [3]. Based on the Single Input Rule Modules (SIRMs), the problem of anti-swing and position control of gantry cranes is solved in [4]. A hybrid control scheme of input tracking and anti-swing control of gantry crane system based on LQR control is proposed in [5]. The problem of simultaneous control for position and anti-swing of gantry cranes was solved by using a sliding control method in [6]. A nonlinear model predictive control (MPC) for gantry crane was proposed in [7]. A non-linear and linear model of gantry cranes based on Lyapunov equation and Taylor expansion and a state feedback controller design method based on Lyapunov method were proposed in [8]. A hybrid control strategy combining the advantages of both zero-vibration double derivative input shaping control and neuro-adaptive sliding mode control is proposed in [9].

This paper aims to design a simple and reliable control system for gantry cranes, with rapid position tracking and anti-swing performance. To this end, we design a hybrid control system by putting the swing angle compensator into position control system.

Design of the proposed control system

The plane model of the gantry crane system is shown in Fig 1.

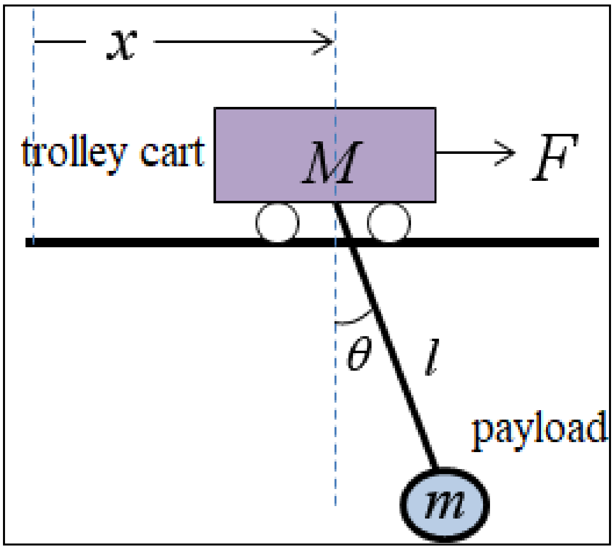

Fig 1: Plane model of GCS.

To simplify the construction of the dynamic model of GCS, the following assumptions are made:

- Trolley cart and loads are only moved on the horizontal plane.
- Uncertainty due to disturbances is ignored.
- The mass of the rope is ignored.
- The friction force trolley cart is ignored.

The Lagrangian equation for the gantry crane system becomes

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right)-\frac{\partial L}{\partial q_i}=Q_i,\quad i=1,2,\cdots,n \tag{1}$$

$$L=T-P \tag{2}$$

where T is the total kinetic energy, P is the total potential energy, Q is the non-conservative generalized force, and n is the total number of independent generalized coordinates;

$$n=2,\ \begin{bmatrix} q_1 \\ q_2 \end{bmatrix}=\begin{bmatrix} x \\ \theta \end{bmatrix},\ \begin{bmatrix} Q_1 \\ Q_2 \end{bmatrix}=\begin{bmatrix} F \\ 0 \end{bmatrix}$$

The kinematic energy of load is;

$$T_1=\frac{1}{2}mv_1^2$$

For the velocity analysis, we use the cosine law as follows:

$$v_1^2=\dot{x}^2+l^2\dot{\theta}^2+2\dot{x}l\dot{\theta}\cos\theta$$

The kinematic energy of load and trolley cart using the cosine law is given by Equations (3) and (4).

$$T_1=\frac{1}{2}m(\dot{x}^2+l^2\dot{\theta}^2+2\dot{x}l\dot{\theta}\cos\theta) \tag{3}$$

$$T_2=\frac{1}{2}M\dot{x}^2 \tag{4}$$

Therefore

$$L = \frac{1}{2}m(\dot{x}^2 + l^2\dot{\theta}^2 + 2\dot{x}l\dot{\theta}\cos\theta) + \frac{1}{2}M\dot{x}^2 + mgl\cos\theta \tag{5}$$

Finally, the nonlinear model for the GCS is;

$$(m+M)\ddot{x} + ml\ddot{\theta}\cos\theta - ml\dot{\theta}^2\sin\theta = F \tag{6}$$

$$ml^2\ddot{\theta} + ml\ddot{x}\cos\theta + mgl\sin\theta = 0 \tag{7}$$

To introduce the linear system design method, let us linearize the above nonlinear model.
Assuming that the swing angle of the load is small, the approximation to the above nonlinear model is;

$$(m+M)\ddot{x} + ml\ddot{\theta} = F \tag{8}$$

$$ml^2\ddot{\theta} + ml\ddot{x} + mgl\theta = 0 \tag{9}$$

On the other hand, from Equation (8) and Equation (9), the transfer function can be obtained as follows:

$$\begin{bmatrix} X(s) \\ \theta(s) \end{bmatrix} = \begin{bmatrix} \dfrac{s^2 l + g}{s^2[Mls^2 + g(M+m)]} \\ -\dfrac{1}{[Mls^2 + g(M+m)]} \end{bmatrix} F(s) \tag{10}$$

The motor is a DC motor, and the input is approximated by a first-order system with voltage *u(t)* and the output *f(t)* as follows:

$$G_m(s) = \frac{k_m s}{T_0 s + 1} U(s) \tag{11}$$

Thus, combining the two equations (10) and (11), we obtain

$$\begin{bmatrix} X(s) \\ \theta(s) \end{bmatrix} = \begin{bmatrix} \dfrac{k_m(s^2 l + g)}{s^2[Mls^2 + g(M+m)](T_0 s + 1)} \\ -\dfrac{k_m s}{[Mls^2 + g(M+m)](T_0 s + 1)} \end{bmatrix} U(s) \tag{12}$$

if $M \gg m$, The above expression is simplified as follows.

$$\begin{bmatrix} X(s) \\ \theta(s) \end{bmatrix} = \begin{bmatrix} \dfrac{k_m}{Ms(T_0 s + 1)} \\ -\dfrac{k_m s}{[Mls^2 + g(M+m)](T_0 s + 1)} \end{bmatrix} U(s) \tag{13}$$

In the control system design of GCS, first, the position control system is designed and then the control system for anti-swing is designed. The control system is shown in Fig 2.

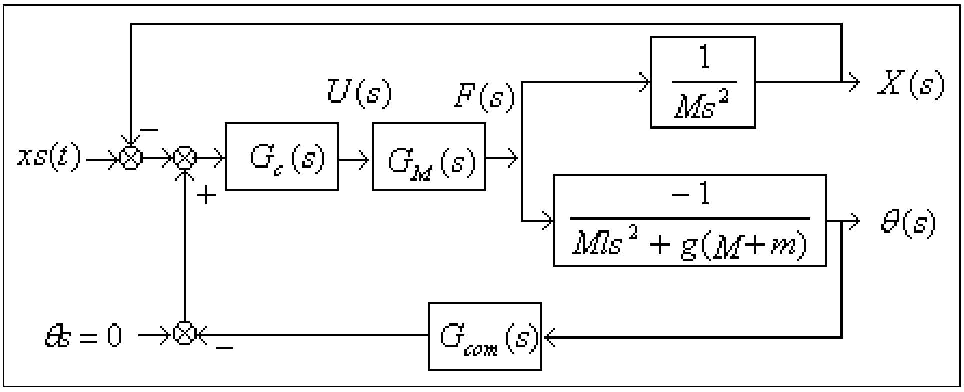

Fig 2: Schematic diagram of overhead crane control system.

In the control loop for *x(t)*, object is

$$G_{xo}(s) = \frac{k_m}{Ms(T_0 s + 1)} \tag{14}$$

Consider the case when the controller is chosen as a proportional controller, $G_c(s) = k_c$.
Thus, the closed-loop transfer function of the control loop with respect to $x(t)$ is

$$\varphi_x(s) = \frac{1}{\frac{MT_0}{k_c k_m} s^2 + \frac{M}{k_c k_m} s + 1} \tag{15}$$

where k_c is the gain.

Also, using $T_x^{\ 2} = \frac{MT_0}{k_c k}$ and $2\zeta_x T_x = \frac{M}{k_c k_m}$, the transfer function of the closed-loop system is expressed as

$$\varphi_x(s) = \frac{1}{T_x^2 s^2 + 2\zeta_x T_x s + 1} \tag{16}$$

Therefore, the gain of the controller can be obtained as follows:

$$k_c = \frac{M}{4\zeta^2 k_m T_0}$$

Consider the case when the controller is chosen as a proportional- derivative controller, $G_c(s) = k_c(1 + T_D s)$.
Thus, the closed-loop transfer function of the control loop is

$$\varphi_x(s) = \frac{k_m k_c (1 + T_D s)}{Ms(T_0 s + 1) + k_m k_c (1 + T_D s)} \tag{17}$$

Here, if the derivative coefficient of the PD controller is chosen as $T_D = T_0$, the transfer function of the closed-loop system is a first-order inertial component with

$$\varphi_x(s) = \frac{1}{1 + \frac{M}{k_m k_c} s} \tag{18}$$

with strong robustness, we can choose k_c arbitrarily.

Next, we design a compensator $G_{com}(s)$.

In the control loop for $\theta(t)$, the object is

$$G_{\theta o}(s)=\frac{-k_c k_m s}{[Mls^2+g(M+m)](T_0 s+1)} \tag{19}$$

Then the closed-loop transfer function of the compensation loop for $\theta(t)$ is

$$\varphi_\theta(s)=\frac{G_{\theta o}(s)}{1+G_{\theta o}(s)G_{com}(s)} \tag{20}$$

Consider the case when the controller is chosen as a proportional controller, $G_c(s)=k_c$.
Thus, the closed-loop transfer function of the control loop is

$$\varphi_\theta(s)=\frac{1}{G_{com}(s)+\dfrac{[Mls^2+g(M+m)](T_0 s+1)}{k_c k_m s}} \tag{21}$$

And if the compensator is a proportional element, i.e., $G_{com}(s)=k_{com}$, the closed-loop transfer function is a third-order system with

$$\varphi_\theta(s)=\frac{k_c k_m s}{k_c k_m k_{com} s+[Mls^2+g(M+m)](T_0 s+1)} \tag{22}$$

Consider the case when the controller is chosen as a proportional controller, $G_c(s)=k_c(1+T_D s)$.
Thus, the closed-loop transfer function of the control loop is

$$\phi_\theta(s)=\frac{1}{G_{com}(s)+\dfrac{[Mls^2+g(M+m)](T_0 s+1)}{k_c k_m s(1+T_D s)}} \tag{23}$$

Here, if choose $T_D=T_0$, the closed-loop transfer function is

$$\phi_\theta(s)=\frac{\dfrac{k_c k_m}{g(M+m)}s}{\dfrac{Ml}{g(M+m)}s^2+\dfrac{k_c k_m}{g(M+m)}G_{com}(s)s+1} \tag{24}$$

Then if $G_{com}(s)=k_{com}$, then

$$\varphi_\theta(s)=\frac{k_\theta s}{T_\theta^2 s^2+2\zeta_\theta T_\theta s+1} \tag{25}$$

Here; $T_\theta=\sqrt{\dfrac{Ml}{g(M+m)}}$, $\xi_\theta=\dfrac{k_c k_m k_{com}}{2g(M+m)T_\theta}$, $k_\theta=\dfrac{k_c k_m}{g(M+m)}$

Thus, the unique natural frequency is determined by the system parameters, and the damping ratio is determined by the gain of the controller and the compensator. Therefore, given the gain of the controller, the gain of the compensator corresponding to the desired value of the damping ratio can be obtained. On the other hand, choosing ξ_φ from the viewpoint of oscillation of the angle of oscillation, we can determine k_{com} satisfying ξ_φ.

Simulation

The system parameters used in the simulation are as follows:

$$M = 30,\ m = 4,\ l = 1.45,\ k_m = 72.5,\ T_0 = \frac{1}{2.4}$$

Then Equation (13) becomes

$$\begin{bmatrix} X(s) \\ \theta(s) \end{bmatrix} = \begin{bmatrix} \dfrac{5.8}{s(s+2.4)} \\ -\dfrac{4s}{[s^2+7.775](s+2.4)} \end{bmatrix} U(s)$$

First, the simulation is carried out for the design of controller $G_c(s)$.

To avoid the oscillation of the position control system, let $\zeta = 1$;

$$k_c = \frac{M}{4\zeta^2 k_m T_0} = 0.248$$

The simulation curves for the case of G_c with proportional controller and G_c with proportional-derivative controller and $T_D = T_0$ are shown in Fig 3.

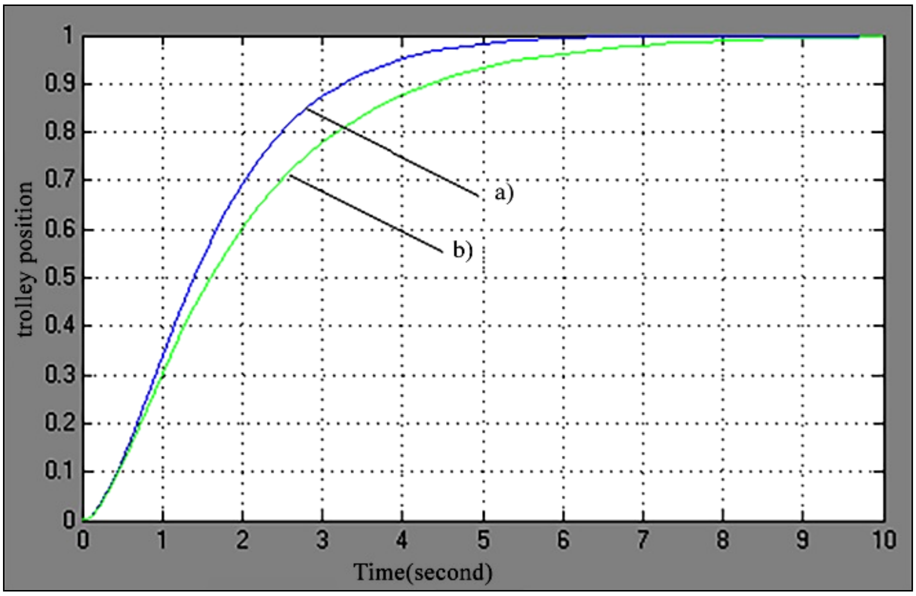

Fig 3: Simulation curve for $k_c = 0.248$, a) controller is proportional controller, b) controller is proportional-derivative controller and $T_D = T_0$.

Then, the simulation curve of the proportional-derivative controller with $T_D = T_0$ and increasing from $k_c = 0.248$ to $k_c = 8$ is shown in Fig 4.

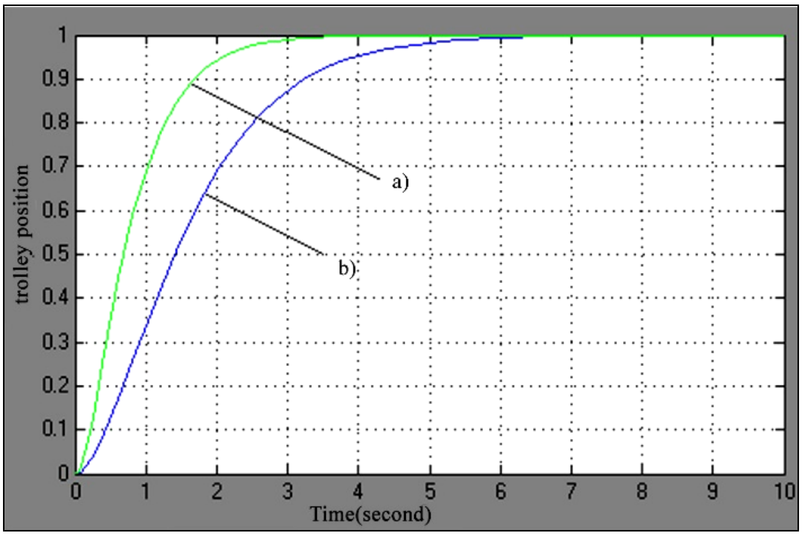

Fig 4: Simulation curve with increasing $k_c = 0.248$ (a) to $k_c = 8$ (b).

Next, the simulation is carried out for the design of compensator $G_{com}(s)$.

In the case of using proportional controller as controller, the simulation curve for $k_c = 0.248$ and $G_{com}(s) = k_{com} = 1$ is shown in Fig 5, the simulation curve for $k_c = 0.248$ and $G_{com}(s) = k_{com} = 5$ is shown in Fig 6.

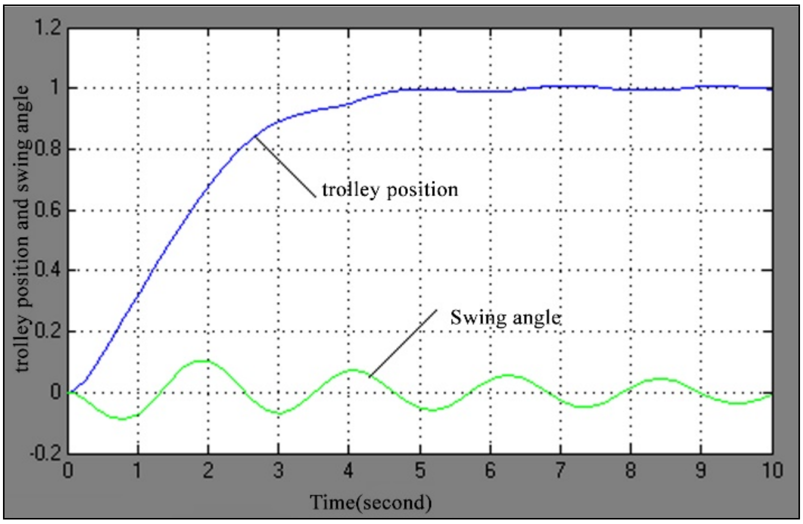

Fig 5: Simulation curves for $k_c = 0.248$, $G_{com}(s) = k_{com} = 1$.

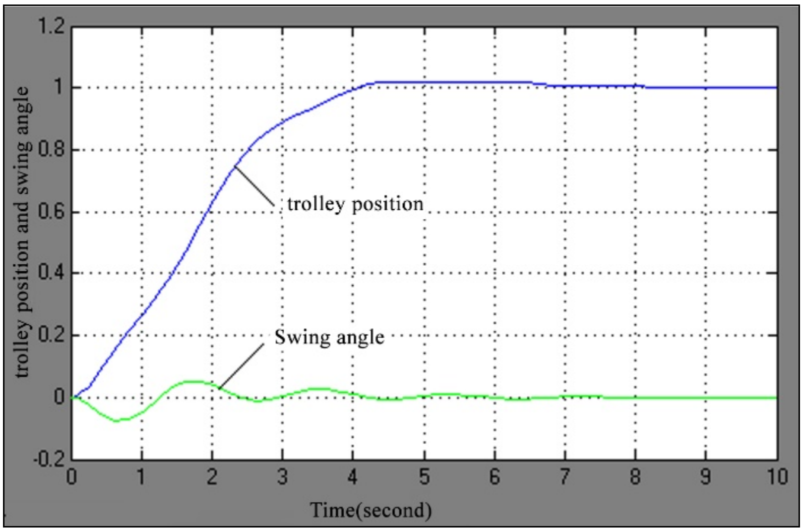

Fig 6: Simulation curves for $k_c = 0.248$, $G_{com}(s) = k_{com} = 5$.

In the case of using proportional controller as controller, the simulated curves for different conditions are shown in Figs 7, 8 and 9.

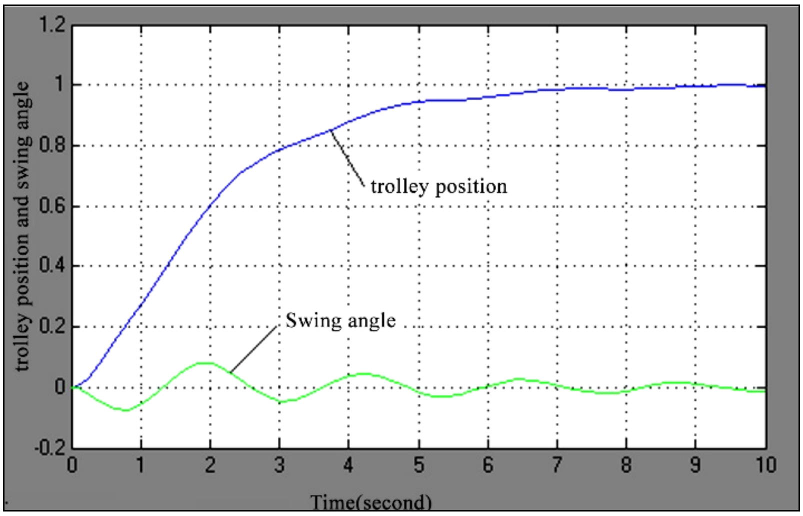

Fig 7: Simulation curves for $k_c = 0.248$, $T_D = T_0$, $G_{com}(s) = k_{com} = 1$.

For $T_D = T_0$ and $G_{com}(s) = k_{com}$, we have $\xi_\varphi = 1$ in the sense that the oscillation of the swing angle is cancelled $T_\theta = 0.36$, $k_{com} = 1.97$, $k_\theta = 0.056$

The simulation results of this case are shown in Fig 10.

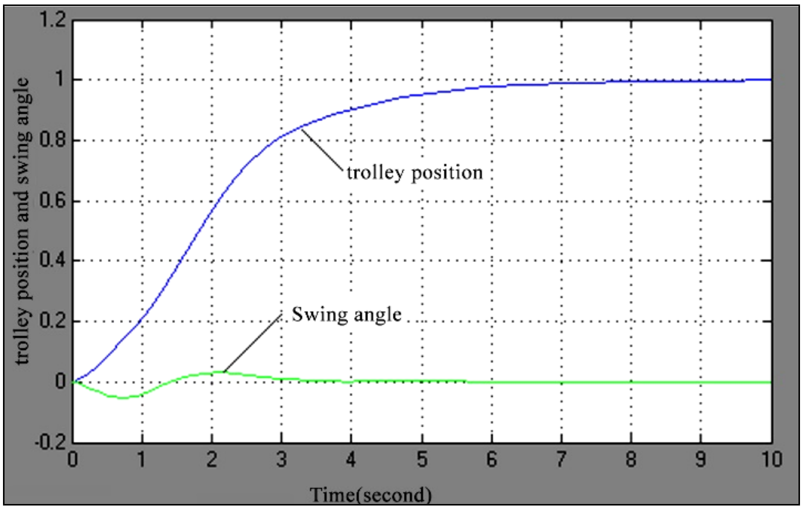

Fig 8: Simulation curves for $k_c = 0.248$, $T_D = T_0$, $G_{com}(s) = k_{com} = 5$.

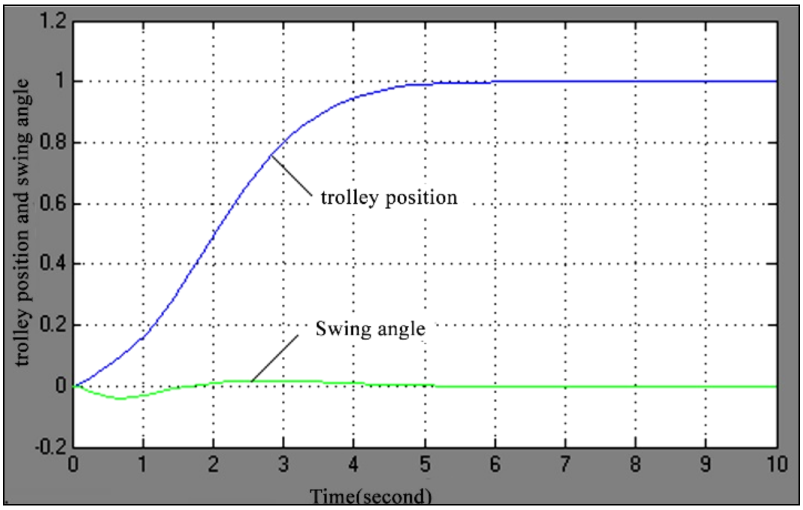

Fig 9: Simulation curves for $k_c = 0.248$, $T_D = T_0$, $G_{com}(s) = k_{com} = 10$.

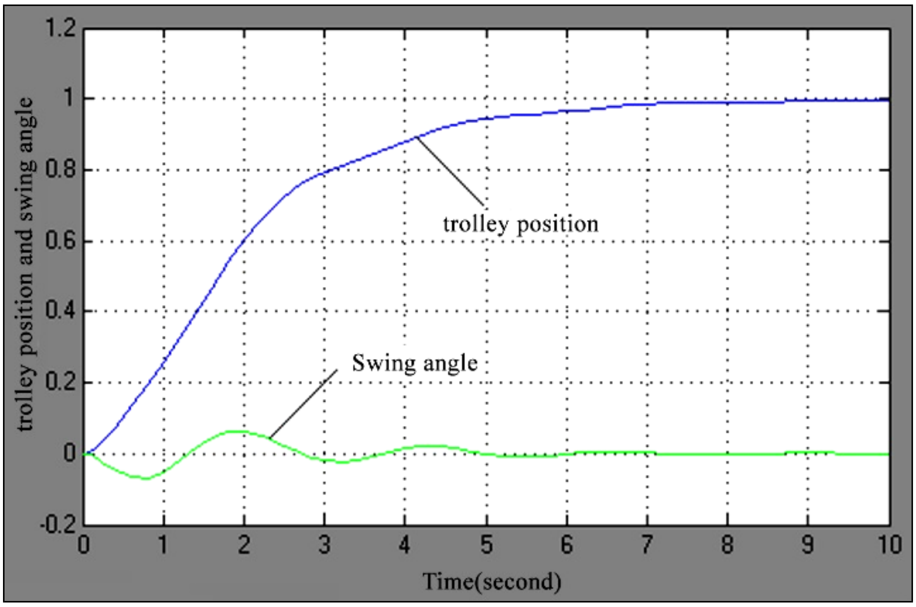

Fig 10: Simulation curves for $T_\theta = 0.36$, $k_{com} = 1.97$, $k_\theta = 0.056$.

The simulation conditions for the above different cases are shown in Table 1.

Table 1: Various simulation conditions.

Controller type G_c	**Controller type** G_{com}	T_D	K_c	K_{com}
P		-	0.248	
PD		T_0	0.248	
PD		T_0	8	
P	P	-	0.248	1
P	P	-	0.248	5
PD	P	T_0	0.248	1
PD	P	T_0	0.248	5
PD	P	T_0	0.248	10

Results

The simulation of the controller G_c design shows the following results.
First, the controller of the position control system can be used as a proportional controller or a proportional-derivative controller to eliminate the oscillations in the response.

Second, it can be seen that the controller with $T_D = T_0$ is a proportional-derivative controller, which can achieve better performance when the controller is a proportional controller by increasing the controller gain. Thus, in this case, the controller gain can be chosen arbitrarily considering the working characteristics of GCS.

The simulation of the controller G_{com} design shows the following results.
First, when the controller is proportional, the compensator transfer coefficient should be increased to avoid the oscillation of the swing angle of the load, but oscillations occur in the position control system response.
Second, to avoid the oscillation of the swing angle of the load, it is desirable to choose the position controller as a proportional-derivative controller.
Third, in the case of using proportional-derivative controller, the gain of the compensator can be increased to reduce the oscillation of the swing angle while eliminating the oscillation of the response characteristic in the position control system.

Conclusion

In this paper, the problem of position and swing angle control of gantry crane is considered, and the following conclusions are drawn from design and simulation. First, the problem of gantry crane control has been solved by designing a controller for position control and combining it with a compensator for the swing angle. Second, the control system has been designed to allow for less oscillation of the swing angle and no oscillation of the position control response. Third, the controller is designed as a proportional-derivative element, which makes the design of the controller and the compensator easier. Fourth, it was found that the gain of the compensator has been achieved to reduce the oscillation of the swing angle. For the application in a wider range of the proposed control system, a deeper evaluation of the stability of the system should be made if the weight of the load carried by the gantry crane increases.

Acknowledgments
We are grateful to our colleagues for their help to make good results and thank the reviewers and editors who helped to improve the quality of the paper.

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