Anti-Sway Control System Design for the Container Crane

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Abstract: The sway control problem of the pendulum motion of the container crane hanging on the trolley, which transports containers from the container ship to the truck, is considered in this paper. In the container crane control problem, the main issue is to suppress the residual swing motion of the container at the end of the acceleration, deceleration or the case of that the unexpected disturbance input exists. For this problem, in general, the trolley motion control strategy is introduced and applied to real plants. In this paper, we suggest a new type of swing motion control system for a crane system in which a small auxiliary mass is installed on the spreader. The actuator reacting against the auxiliary mass applies inertial control forces to the spreader of the container crane to reduce the swing motion in the desired manner. In this paper, we consider that the length of the rope varies is we design the anti-sway control system based on LMI(linear matrix inequality) approach. And, it will be shown that the proposed control strategy is useful and it can be easily applicable to the real world. So, in this study, we investigate usefulness of the proposed anti-sway system and evaluate system performance from simulation and experimental studies.

Keywords: swing motion control, container crane, trolley, anti-sway system, LMI approach.

1. INTRODUCTION

The container crane is widely used to transport containers from the container ship to the trucks. But there is a residual swing motion of the crane system at the end of acceleration and deceleration or in the case of that the unexpected disturbance input exists. Figure 1 and 2 show kinds of the crane[1]. In these systems, the trolley motion control technique is very well known strategy to suppress undesirable swing motion[2,3]. But it has some problems such as increase of fatigue and discomfort of the crane drivers who work for a long time. So, we introduce a new solution[4] to suppress swing motion as illustrated in Figure 3, which is installed on the spreader of the crane. A suggested system is consists of a damper mass, a belt or ball-screw to transfer power to the moving mass and a motor to move a damper mass etc. In this system, the actuator reacting against the auxiliary mass applies inertial control forces to the crane system to reduce the undesirable swing motion.

Figure 1. Transfer crane

Figure 2. Container crane

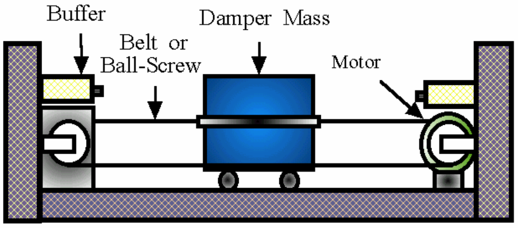


Figure 3. An active anti-sway control system

2. MODELLING AND PROBLEM FORMULATION

Figure 4 shows dynamic model of the container crane as the controlled system.

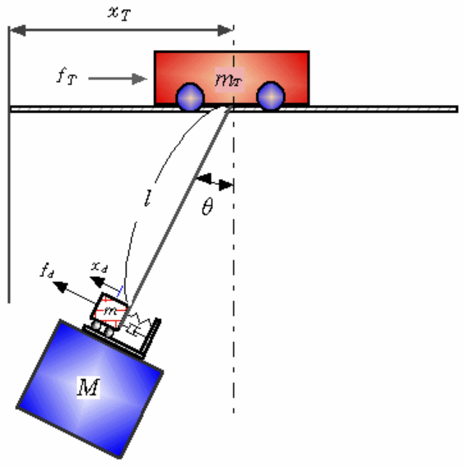


Figure 4. Dynamic model of the controlled system

In this plant, if we suppose that the center of gravity of the spreader is equal to that of the damper mass, then the center x_G, y_G can be written as

$$x_G = l\sin\theta + x_T\,,\quad y_G = -l\cos\theta \tag{1}$$

And, if we denote that K is kinetic energy and V is potential energy, then they are given as following:

$$K = \frac{1}{2}m_T {\dot{x}_T}^2 + \frac{1}{2}(M+m)({\dot{x}_G}^2 + {\dot{y}_G}^2) \tag{2}$$

$$V = -(M+m)gl\cos\theta \tag{3}$$

Here, let $L = K - V$ to calculate dynamic equations of the controlled system using Lagrange's dynamic equations:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}_T}\right) - \frac{\partial L}{\partial x_T} = f_T$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) - \frac{\partial L}{\partial \theta} = T - T_d \tag{4}$$

In this study, we concentrate on the reduction of swing motion through the total process including moving and stop of trolley. Of course, the end states of the loading and unloading process are considered. But, in this study we don't consider dynamic of trolley, because it can be regarded as a kind of disturbance input. Then the linearized dynamic equations of the system are given by

$$(M+m)l^2\ddot{\theta} + C\dot{\theta} + (M+m)gl\sin\theta = T - T_d \tag{5}$$

$$T_d = mgx_d\cos\theta + f_d l \tag{6}$$

$$m\ddot{x}_d = -mg\sin\theta + f_d - C_d\dot{x}_d - k_d x_d \tag{7}$$

where,

M : mass of container
m : mass of damper mass
l : rope length
C : damping constant
T : moment generated by disturbance
T_d : moment generated by actuator
g : acceleration of gravity
f_d : horizontal force generated by actuator
C_d : damping constant of actuator
k_d : stiffness of actuator

In this paper, we assume that θ is small value and the spreader takes a leveling movement. This facts denote that $\sin\theta \cong \theta$ and $\cos\theta \cong 1$. Then, the Eqs. (5)~(7) can be rewritten as follows:

$$(M+m)l^2\ddot{\theta} + C\dot{\theta} + (M+m)gl\theta = T - T_d \tag{8}$$

$$T_d = mgx_d + f_d l \tag{9}$$

$$m\ddot{x}_d = -mg\theta + f_d - C_d\dot{x}_d - k_d x_d \tag{10}$$

3. PARAMETER ESTIMATION

3.1 Parameter estimation of the spreader

At first, let us estimate the unknown parameters appeared in Eq. (8), which denotes the dynamics of spreader part. For example, if we use the initial response which is obtained from experiment as shown in Figure 5. Then, using Eq. (8), free vibration of the spreader is described as follows:

$$\ddot{x} + \frac{C}{(M+m)l^2}\dot{x} + \frac{g}{l}x = 0 \tag{11}$$

Eq. (11) is rewritten by following second order system:

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega^2{}_n x = 0 \tag{12}$$

where,

$$2\zeta\omega_n = \frac{C}{(M+m)l^2},\quad \omega^2{}_n = \frac{g}{l} \tag{13}$$

From these facts, if we consider the vibration period λ and damping ratio ρ in Figure 4, the following relations are obtained.

$$\lambda = 2\pi/(1-\zeta^2)^{1/2}\omega_n,\quad \rho = \exp(-2\pi\zeta/(1-\zeta^2)^{1/2}) \tag{14}$$

It means that if we obtain the vibration period and damping ratio from the free vibration response, the unknown parameter

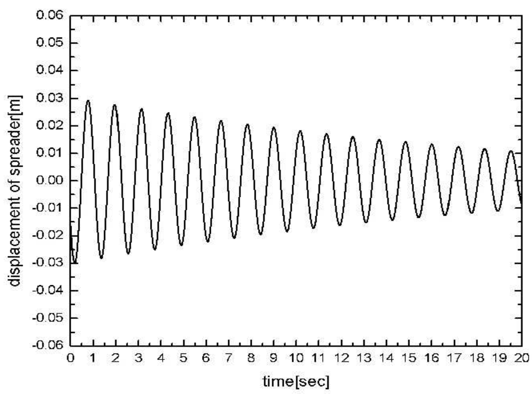


Figure 5. Free vibration of the spreader

is estimated. In the result, an unknown parameter C(damping constant) is calculated as $C = 0.005324$ using some known and defined parameters where the rope length is 0.36[m].

3.2 System representation of the actuator system

As illustrated in the previews section, anti-sway control system is installed on the spreader part. This actuator part is made up with motor, belt and other apparatus. Therefore, it is difficult to derive the system representation exactly. So, in this paper, we derive it using step responses obtained from simulation and experiment. The results are shown in Figure 6. Using this fact, the estimated parameters appeared in Eqs. (7) and (10) are determined as follows:

$$C_d = 1.5865, \quad k_d = 0.00095 \tag{15}$$

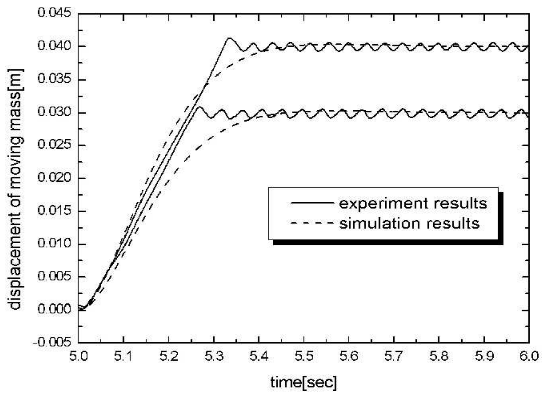


Figure 6. Step responses of actuator system

3.3 Overall system representation

In the result, the state equation for controlled system is given by

$$\begin{aligned} \dot{x}_p &= Ax_p + Bu + Dw \\ y &= Cx_p \end{aligned} \tag{16}$$

where, the states $x_p = [x \quad \dot{x} \quad x_d \quad \dot{x}_d]^T$, $u = v$ (input voltage to motor), $w = T$ and

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{g}{l} & -\frac{C}{(M+m)l^2} & -\frac{mg}{(M+m)l} & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{g}{l} & 0 & -\frac{k_d}{m} & -\frac{C_d}{m} \end{bmatrix}, \tag{17}$$

$$B = \begin{bmatrix} 0 & -\frac{K_m}{(M+m)} & 0 & \frac{K_m}{m} \end{bmatrix}^T,$$

$$C = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix},$$

$$D = \begin{bmatrix} \frac{mg}{(M+m)l^2} & 0 & 0 & 0 \end{bmatrix}^T$$

Where the estimated motor torgue constant is given by K_m=150.

4. CONTROLLER DESIGN AND SIMULATION

In this chapter, we evaluate the system performance and show the usefulness of the controlled system by the simulation and experimental(to be presented in the session) study. Let us design a controller based on the robust control approach. For this, let us describe the generalized plant as follows:

$$\begin{aligned} \dot{x}_z &= Ax_z + B_1 w + B_2 u \\ z &= C_1 x_z + D_{11} w + D_{12} u \\ y_z &= C_2 x_z + D_{21} w + D_{22} u \end{aligned} \tag{18}$$

where, new state $x_z = x_p$. And if we use Eqs. (16) and (18), then the following relations are obtained.

$$\begin{aligned} & B_1 = D, \quad B_2 = B, \\ & C_1 = [1 \quad 0 \quad 0 \quad 0], \quad C_2 = C, \\ & D_{11} = [1], \quad D_{12} = [0], \quad D_{21}^T = [0 \quad 0], \quad D_{22}^T = [0 \quad 0] \end{aligned} \tag{19}$$

Based on some assumptions and conditions, we have designed a controller which is robust to disturbance input to the plant. In other word, the designed controller satisfies the norm condition $\|Z\|_\infty < \gamma(>0)$. Where, as well known, Z is the transfer function of the disturbance input to the controlled output.

Especially, in this paper we consider that the rope length l is varied in the specified range. Then the Eq. (18) is represented as following.

$$\begin{aligned} \dot{x}_z &= A(\delta)x_z + B_1(\delta)w + B_2 u \\ z &= C_1 x_z + D_{11} w + D_{12} u \\ y_z &= C_2 x_z + D_{21} w + D_{22} u \end{aligned} \tag{20}$$

where, $\delta = l$: rope length[m] and it is assumed that the length is varied as follows:

$$0.25[m] \le l \le 0.75[m]$$

Then, the dynamic matrices $A(\delta)$ and $B_1(\delta)$ are uncertain and assumed to belong to the convex polytopic sets defined as

$$A(\delta) = \sum_{i=1}^{N} \delta_i A_i, \; B(\delta) = \sum_{i=1}^{N} \delta_i B_i, \; \sum_{i=1}^{N} \delta_i = 1, \; \delta_i \ge 0 \tag{21}$$

where A_i and B_i are constant matrices denoting the extreme points of the uncertain sets.

[Lemma 1][5] For the uncertainties and upper bound $\gamma > 0$ of the disturbance input, the control system of Eq. (15) is robustly stable, if there exists a positive definite matrix P such that the inequality:

$$\begin{bmatrix} PA(\delta) + A^T(\delta)P & PB_1(\delta) & C_1^T \\ B_1^T(\delta)P & -\gamma I & 0 \\ C_1 & 0 & -\gamma I \end{bmatrix} < 0 \tag{22}$$

holds.

Based on this lemma, the controller is calculated as follows:

$$K(s) := \begin{bmatrix} A_c & B_c \\ C_c & D_c \end{bmatrix} \tag{23}$$

where,

$$A_c = \begin{bmatrix} -8.8544 & -1.3779 & -0.3746 & -0.2245 \\ 11.9172 & -11.2673 & 17.9408 & -26.4625 \\ -1.4202 & 1.8062 & -5.8168 & 1.8712 \\ -1.0796 & 7.4617 & 35.1117 & -10.4310 \end{bmatrix} \tag{24}$$

$$B_c = \begin{bmatrix} -0.9165 & 8.8098 \\ 0.3266 & -0.5598 \\ -5.3783 & -1.0668 \\ 29.2411 & 0.0909 \end{bmatrix}$$

$$C_c = [-0.0891 \;\; -0.0754 \;\; -0.1705 \;\; 0.2185]$$

$$D_c = [0 \;\; 0].$$

Using this controller, the simulation results for the designed reduction model as shown in Figure 7 are obtained.

Figure 8 and Figure 9 show the frequency responses of the open-loop and closed-loop systems, respectively.

And the impulse response of the open-loop and closed-loop systems are illustrated in the Figure 10 to Figure 12. In each figure, (a) is the response of the open-loop system and (b) denotes the closed-loop system.

In the simulation study, as mentioned before, it is considered that the rope length is changing in the specified range. To preserve the system stability for the uncertainty and disturbance, we designed the controller based on LMI approach and the robustness and system performance are verified from the simulation result.

From these simple simulation results, it is clear that the usefulness of the proposed anti-sway system is verified and the possibility that the proposed system can be easily applied to the real plants is certified in a sense.

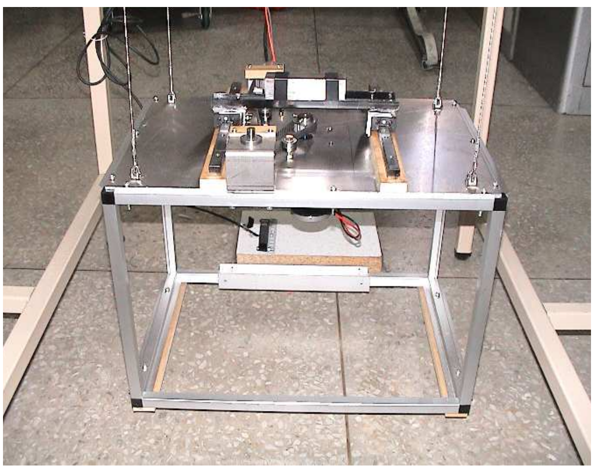


Figure 7. Reduction model of the anti-sway control system

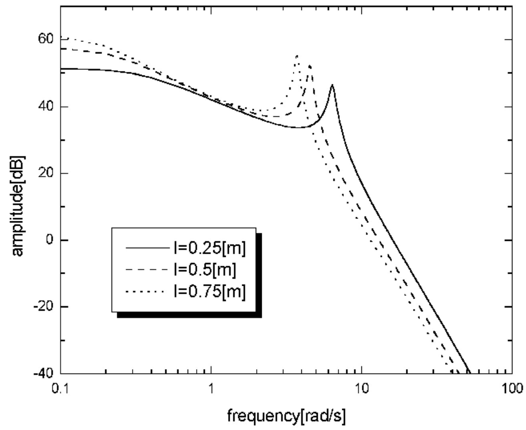


Figure 8. Frequency responses of open-loop systems

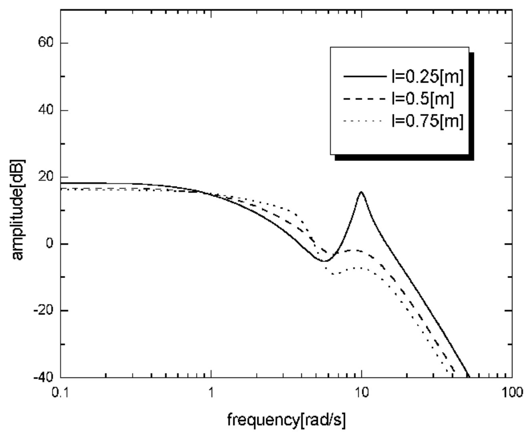
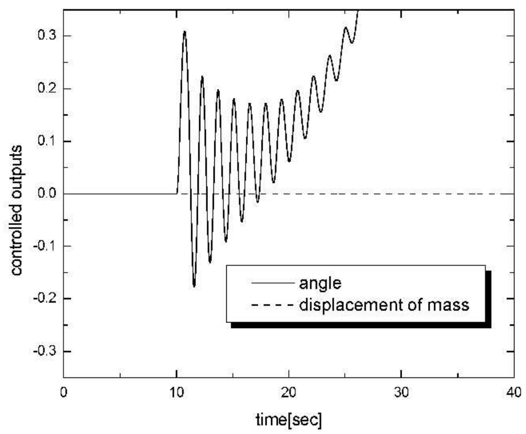
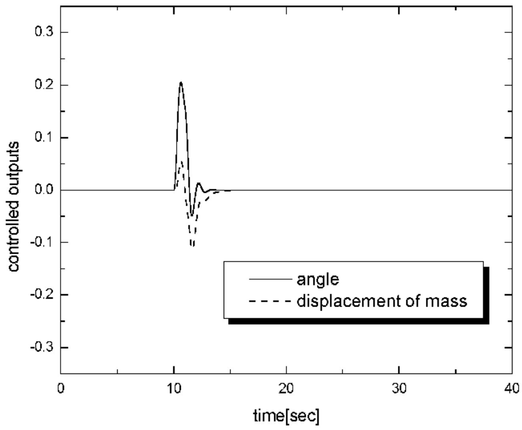


Figure 9. Frequency responses of closed-loop systems

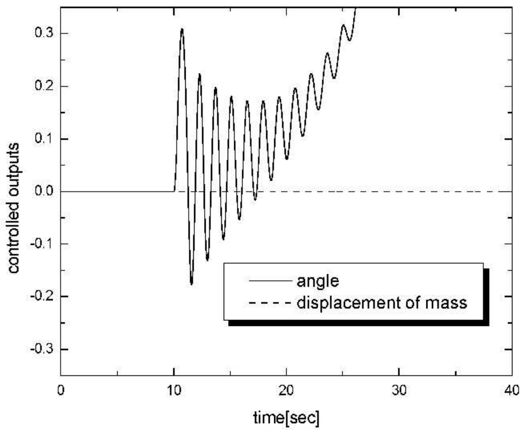


(a) open-loop system

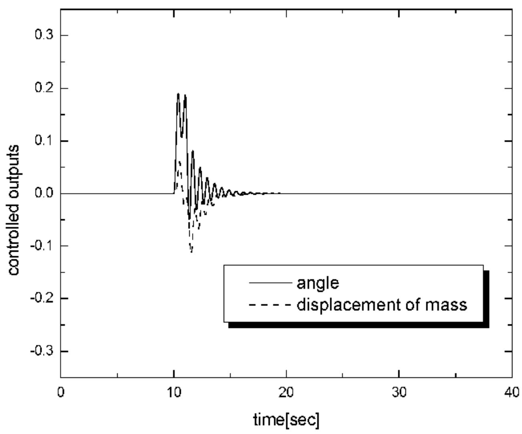


(b) closed-loop system

Figure 10. Impulse responses when the rope length is 0.25[m]

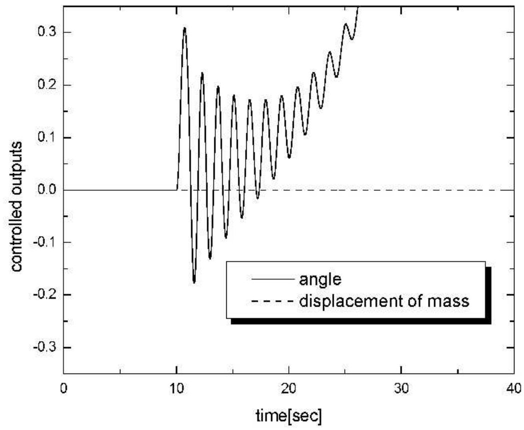


(a) open-loop system

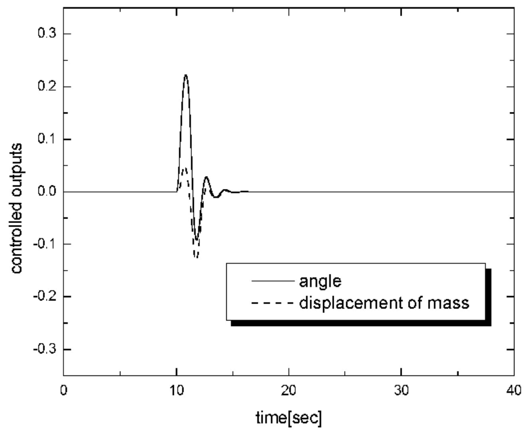


(b) closed-loop system

Figure 11. Impulse responses when the rope length is 0.5[m]



(a) open-loop system



(b) closed-loop system

Figure 12. Impulse responses when the rope length is 0.75[m]

5. CONCLUDING REMARKS

We have suggested a new type of swing motion control system of the crane and verified the usefulness of proposed system by the simulation and experimental studies. This control system can restrain the undesirable swing motion which causes many problems such as increase of fatigue and discomfort of the crane drivers who work for a long time. So, it is verified that the undesirable swing motion can be suppressed efficiently through the reaction of moving the damper mass on the spreader. Especially, the advantage of this system is that the system can be easily applied to the real system and anti-sway effect can be obtained effectively.

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