A New Approach to Anti-Sway System Design Problem

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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, as the basic and first step, we apply the H_∞ control approach to anti-sway control system design problem. 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 through simulation and experimental studies.

Key Words : Swing Motion Control, Crane, Auxiliary Mass, Spreader, Anti-Sway System

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. Fig. 1 shows an example of a kind of container crane(Noriaki et al., 2001). In this system, the trolley motion control technique is very well known strategy to suppress undesirable swing motion(Cheng and Li, 1993; Hong et al, 1997, 1999, 2000, 2003; Lee, 2001; Nomura et al., 1997). But it has some problems such as increase of fatigue and discomfort of the crane drivers who work for a long time. We introduced a new solution(Kim, 2002) to suppress swing motion as illustrated in Fig. 2, which is installed on the spreader of the container 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 container to reduce the undesirable swing motion. Especially in the part of construction technology etc, this type of actuator has been introduced.

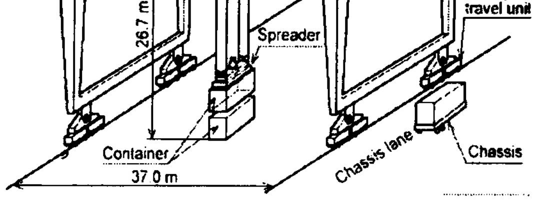

Fig. 1 An example of container crane

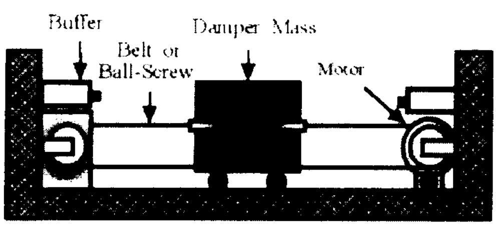

Fig. 2 Schematic diagram of active anti-sway control system

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2. Modeling

Figure 3 shows dynamic model of the container crane as the controlled system:

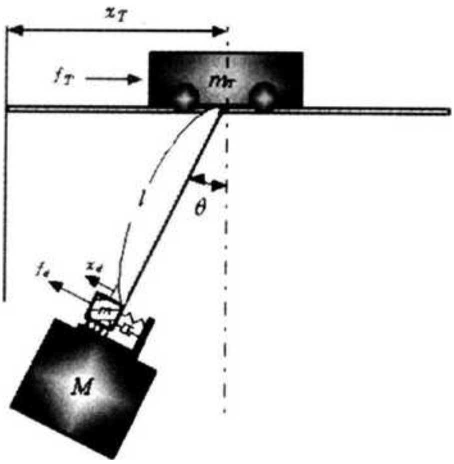


Fig. 3 Dynamic model I of the controlled system

Where,

f_T : driving force for trolley
F_d : horizontal force generated by actuator
l : rope length
m : mass of damper mass
m_T: mass of trolley
M : mass of container
x_G, y_G : gravity center
θ : sway angle

If we suppose that the center of gravity of the spreader is equal to that of the damper mass as shown in Fig. 4, then the center x_G, y_G can be written as

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

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:

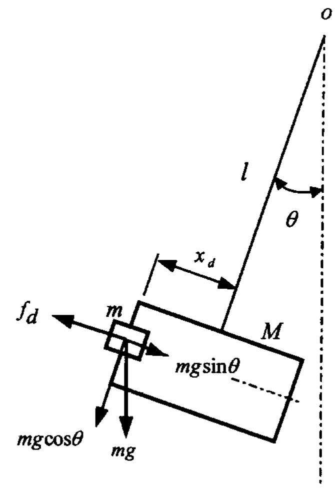


Fig. 4 Dynamic model II of the controlled system

$$\begin{aligned} \frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}_T}\right) - \frac{\partial L}{\partial x_T} &= f_T \\ \frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) - \frac{\partial L}{\partial \theta} &= T - T_d \end{aligned} \tag{4}$$

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 kinetic energy at point 'o' in Fig. 4 is calculated as following:

$$(M+m)l^2\ddot{\theta} = -Mgl\sin\theta - mg(x_d\cos\theta + l\sin\theta) - f_d l + T \tag{5}$$

The kinetic energy produced by the mass m is given as following:

$$m\ddot{x}_d = f_d - mg\sin\theta \tag{6}$$

If we consider the damping and stiffness constant of the system, then the linearized dynamic equations of the system are given by

$$(M+m)l^2\ddot{\theta} + C\dot{\theta} + (M+m)gl\sin\theta = T - T_d \tag{7}$$

$$T_d = mgx_d\cos\theta + f_d l \tag{8}$$

$$m\ddot{x}_d = -mg\sin\theta + f_d - C_d\dot{x}_d - k_d x_d \tag{9}$$

where,

C : damping constant
T : moment generated by disturbance
T_d : moment generated by actuator

g : acceleration of gravity
C_d : damping constant of actuator
k_d : stiffness of actuator

In this paper, we assume that θ is small and the spreader takes a leveling movement. This facts denote that $\sin\theta \cong \theta$, $\cos\theta \cong 1$ and $x = l\theta$. If we consider the rope length is constant, the Eqs. (7) ~ (9) can be rewritten as follows :

$$(M+m)l\ddot{x} + \frac{C}{l}\dot{x} + (M+m)gx = T - mgx_d - f_d l \quad (10)$$

$$m\ddot{x}_d = -\frac{mg}{l}x + f_d - C_d\dot{x}_d - k_d x_d. \quad (11)$$

Parameter estimation of the spreader

At first, let us estimate the unknown parameters appeared in Eq. (10) which denotes the dynamics of spreader part. For this, we use the initial response shown in Fig. 5. Using Eq. (10), free vibration of the spreader is described as Eq. (12).

$$\ddot{x} = \frac{C}{(M+m)l^2}\dot{x} + \frac{g}{l}x = 0. \quad (12)$$

Let us rewrite Eq. (12) as the following second order system :

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = 0 \quad (13)$$

where,

$$2\zeta\omega_n = \frac{C}{(M+m)l^2}, \quad \omega_n^2 = \frac{g}{l}. \quad (14)$$

From these facts, if we consider the vibration period λ and damping ratio ρ in Fig. 5, the following relations are obtained.

$$\lambda = 2\pi/(1-\zeta^2)^{1/2}\omega_n, \quad \rho = \exp(-2\pi\zeta/(1-\zeta^2)^{1/2}) \quad (15)$$

It means that if we obtain the vibration period and damping ratio from the free vibration response, the unknown parameters are estimated. In the result, an unknown parameter, C (damping constant) is calculated as $C = 0.005324$ using some known and defined parameters.

Parameter estimation 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 the simulation and experiment which are shown in Fig. 6. Using this fact, the estimated parameters appeared in Eq. (11) are given as follows :

$$C_d = 1.5865, \quad k_d = 0.00095. \quad (16)$$

Overall system representation

Consequently, the state equation of overall system is given by

$$\begin{aligned}\dot{x}_p &= Ax_p + Bu + Dw \\ y &= Cx_p\end{aligned} \quad (17)$$

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

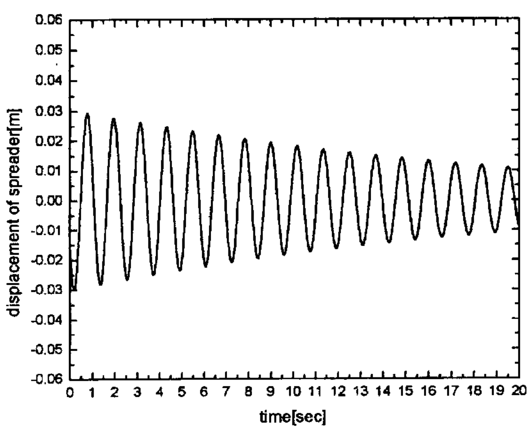


Fig. 5 Free vibration of the spreader

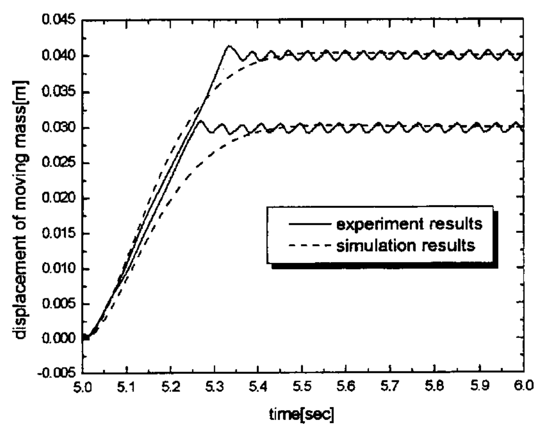


Fig. 6 Step responses of actuator system

$$A=\begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{g}{l} & \frac{C}{(M+m)l^2} & \frac{mg}{(M+m)l} & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{g}{l} & 0 & -\frac{k_d}{m} & -\frac{C_d}{m} \end{bmatrix}, \quad (18)$$

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

$$C=\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix}, \; D=\begin{bmatrix} \frac{mg}{(M+m)l^2} & 0 & 0 & 0 \end{bmatrix}^T.$$

K_m is the motor torque constant which is $K_m=150$.

3. Controller Design

In this chapter, let us design a controller based on H_∞ control approach to evaluate the system performance and show the usefulness of the controlled system by the experimental study. For this, let us describe the generalized plant as follows:

$$\begin{aligned} \dot{x}_z &= Ax_z + B_1 w + B_2 u \\ z &= C_1 x_z + D_{11} w + D_{12} u \\ y_z &= C_2 x_z + D_{21} w + D_{22} u. \end{aligned} \quad (19)$$

If we use Eq. (17) and (19), the following relations are obtained.

$$\begin{aligned} & B_1=D,\ B_2=B, \\ & C_1=[1\ 0\ 0\ 0],\ C_2=C, \\ & D_{11}=[1],\ D_{12}=[0],\ D_{21}^T=[0\ 0],\ D_{22}^T=[0\ 0] \end{aligned} \quad (20)$$

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 given by following Eq. (21) satisfies the norm condition $\| Z \|_\infty < r (>0)$. Z is the transfer function of the disturbance input to the controlled output. Therefore, the controller is calculated as follows:

$$K(s) := \begin{bmatrix} A_c & B_c \\ C_c & D_c \end{bmatrix} \quad (21)$$

where,

$$A_c=\begin{bmatrix} -2.5234 & -0.5709 & -0.4550 & -2.4035 \\ 0.5100 & -4.3636 & -2.2421 & 1.3109 \\ -4.5748 & 5.0089 & -10.1794 & -3.2709 \\ -7.5437 & -3.4879 & -21.4871 & -43.4999 \end{bmatrix} \quad (22)$$

$$B_c=\begin{bmatrix} 0.8803 & -2.1623 \\ -13.7120 & -1.0277 \\ 12.2821 & -3.1357 \\ 4.8467 & -4.5473 \end{bmatrix}$$

$$C_c=[0.0602\ 0.0658\ 0.1519\ -0642]$$

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

4. Experiment

Using the controller obtained in the previews section, experimental results for the designed reduction model as shown in Fig. 7 are obtained. The parameter values of reduction model are summarized in the table.

Figure 8 shows the initial response of the open-loop system (uncontrolled case) and Fig. 9 illustrates the initial response of closed-loop system (controlled case). Fig. 10 and 11 show the disturbance responses of open-loop and closed-loop system, respectively. Especially, in this control system, the movement of moving-mass is restricted within ±0.1m on the spreader.

However, Fig. 9(b) and Fig. 11(b) show that

Table 1 Parameter values of the reduction model

Parameter	Value
Spreader size:	
length	0.70m
width	0.42m
height	0.42m
weight	0.565kg
Rope lenght	0.36m
Moving mass weight	0.095kg

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

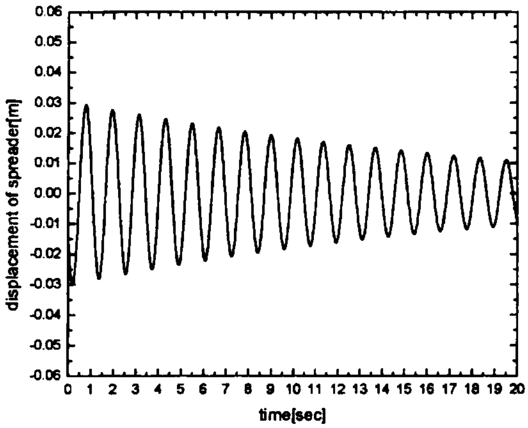


Fig. 8 Initial responses of open-loop system (sway-motion)

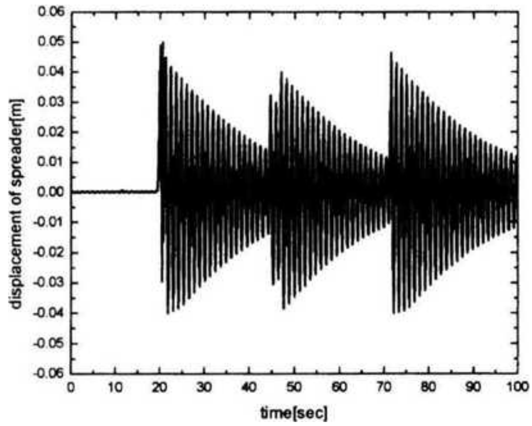
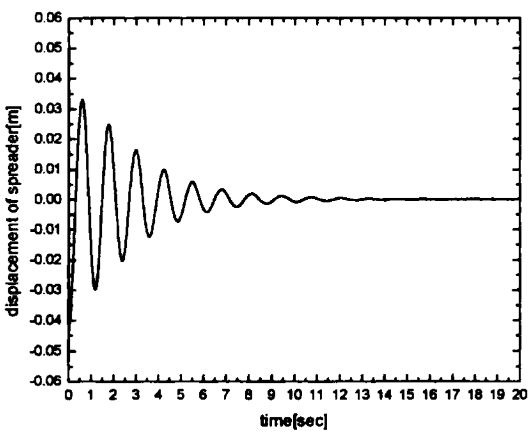
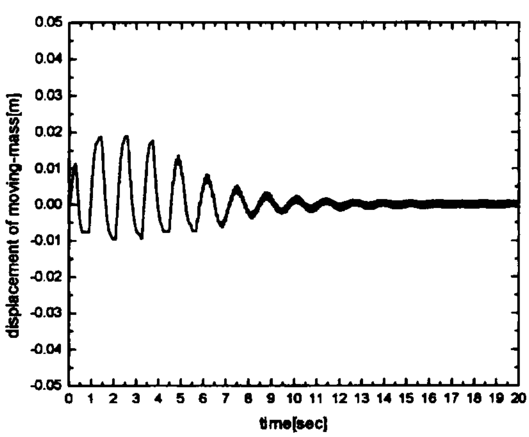


Fig. 10 Disturbance response of open-loop system (sway-motion)

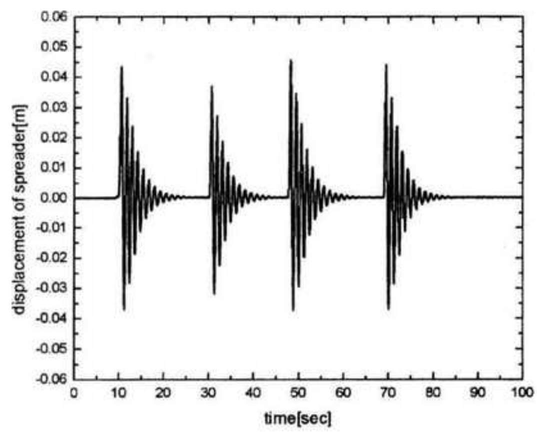


(a) Sway-motion

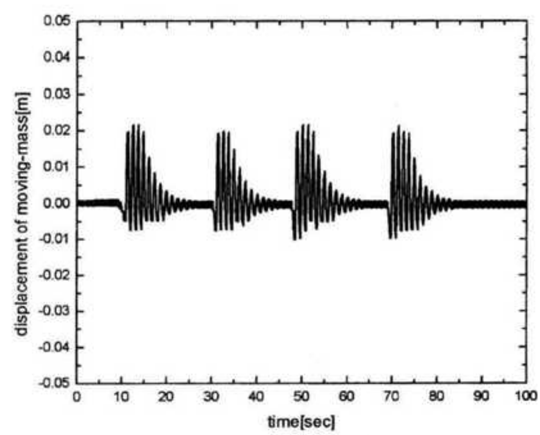


(b) Displacement of moving mass

Fig. 9 Initial response of closed-loop system (sway-motion and displacement of moving mass)



(a) Sway-motion



(b) Displacement of moving mass

Fig. 11 Disturbance response of closed-loop system (sway-motion and displacement of moving mass) the moving-mass is activated in the permissible range. From these simple experimental 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 world is certified in a sense.

5. Concluding Remarks

We have suggested a new type for swing motion control system for a crane spreader system and verified the usefulness of proposed system by the 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. 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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