GAIN-SCHEDULING APPROACH TO MASS DAMPER TYPE ANTI-SWAY SYSTEM DESIGN

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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 consider a mass damper type of swing motion control system for a crane 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 is varied in the specified range and we design the anti-sway control system based on gain-scheduling approach. And, we investigate usefulness of the proposed anti-sway system and evaluate system performance from experimental study.

Keywords: mass damper type, swing motion control, container crane, trolley motion control strategy, anti-sway system, gain-scheduling.

1. INTRODUCTION

The container crane is widely used to transport containers from the container ship to the trucks. But there is 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. For these systems, the trolley motion control technique is very well known strategy to suppress undesirable swing motion (Cheng and Li, 1993, 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. So, we introduced a new solution (Kim, 2002) to suppress swing motion as illustrated in Fig. 1, which is installed on the spreader of the crane.

The 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.

And, it is well known that the rope length in the control system design is should be considered. So, in this paper, we focus on the time-varying parameter, for example rope length change. For this, we assume that the rope length as a parameter which can be estimated in real time, is varying and apply the linear parameter varying(LPV) technique(gain-scheduling control) to the control system design problem. In this control system, the controller dynamics are adjusted in real-time according to the time-varying rope length. The experimental result shows that the proposed control strategy is shown to be useful to the case of varying rope length and robust to disturbances.

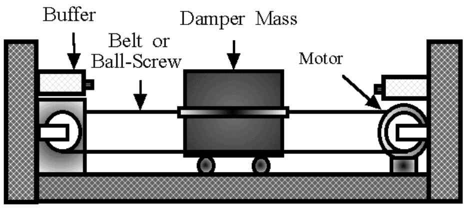


Fig. 1. An active anti-sway control system (Kim, 2002)

2. MODELLING AND PROBLEM FORMULATION

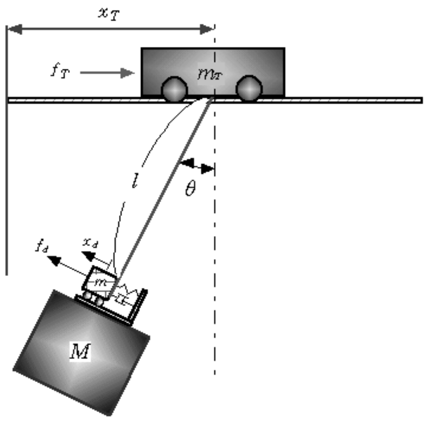
Fig. 2 shows dynamic model of the container crane as the controlled system considered in this paper. In this plant, if we suppose that the center of gravity of the spreader is equal to that of the damper mass, then the center (x_G , y_G) can be written as

$$x_G = l\sin\theta + x_T\,,\ y_G = -l\cos\theta \qquad (1)$$

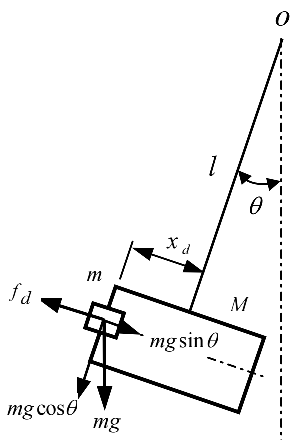
And, if we denote that K is kinetic energy and V is potential energy, then they are given as following:

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

$$\Re = (M+m)gl\cos\theta \qquad (3)$$



(a) schematic diagram of controlled system



(b) geometrical interpretation of controlled system

Fig. 2. Dynamic model of the controlled system

Here, to calculate dynamic equations of the controlled system using Lagrange's dynamic equations:

$$\frac{d}{dt}\left(\frac{\partial \Im}{\partial \dot{\theta}}\right) - \frac{\partial \Im}{\partial \theta} + \frac{\partial \aleph}{\partial \dot{\theta}} + \frac{\partial \Re}{\partial \theta} = \upsilon$$

where υ denotes disturbance and control inputs.

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

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

$$T_d = mgx_d\cos\theta + f_d l \qquad (6)$$

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

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
x_d : displacement of the moving-mass

In this paper, we assume that θ is small value and the spreader takes a levelling movement which means that the displacement of the spreader part $x = l\theta$. These facts denote that $\sin\theta \cong \theta,\ \cos\theta \cong 1$, and the equations (5)~(7) can be rewritten as follows:

$$(M+m)l\ddot{x} + C\dot{x} + (M+m)gx = T - T_d \qquad (8)$$

$$T_d = mgx_d + f_d l \qquad (9)$$

$$m\ddot{x}_d = -mg\,\theta + f_d - C_d\dot{x}_d - k_d x_d \qquad (10)$$

3. PARAMETER ESTIMATION

3.1 Parameter estimation of the spreader

To design anti-sway control system, we consider the reduction model of a container crane shown in Fig. 3 and 4. For this system, at first, let us estimate the unknown parameters appeared in equation (8), which denote the dynamics of spreader part. Where, we use the initial response obtained from experiment as shown in Fig. 5. Using equation (8), the free vibration of the spreader part is described as follows:

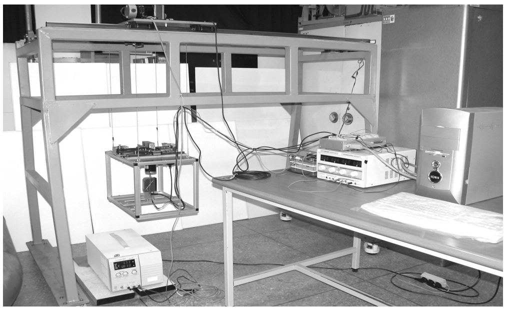


Fig. 3. Reduction model of the anti-sway control system

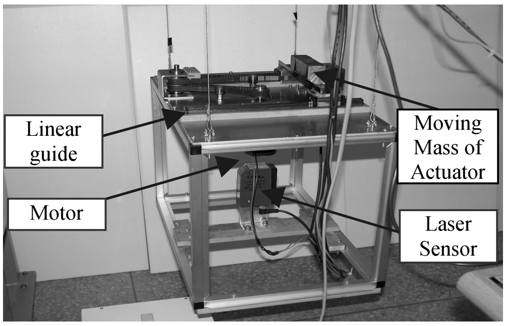


Fig. 4. Spreader part of the anti-sway control system

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

Then the equation (11) can be rewritten by following second order system:

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = 0 \qquad (12)$$

where,

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

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

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

It means that if we calculate the vibration period and damping ratio from the free vibration response as

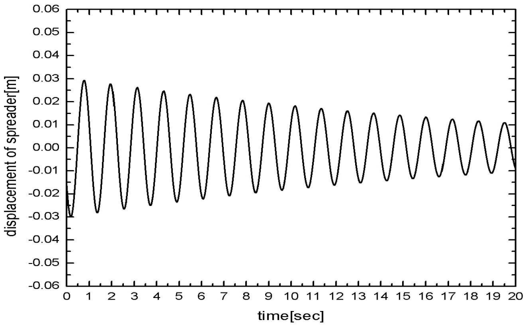


Fig. 5. Free vibration response of the spreader

illustrated in Fig. 5, then the unknown parameter 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 previous section, anti-sway control system is installed on the spreader part as shown in Fig. 4. The actuator part is made up with motor, belt and other apparatus. Then, the dynamical equation is described by the equation (6) and two unknown parameter are calculated as following :

$$C_d = 1.5865, \quad k_d = 0.00095 \qquad (15)$$

So, the step responses of the actuator system obtained from simulation and experiment are shown in Fig. 6.

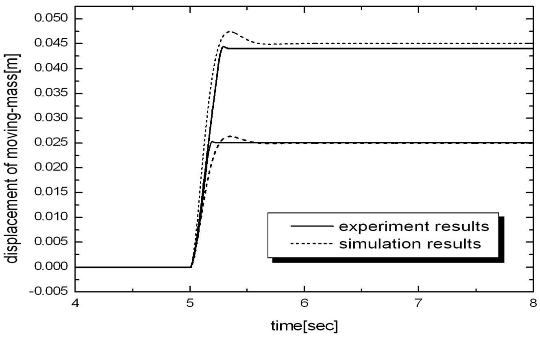


Fig. 6. Step responses of actuator system

3.3 Overall system representation

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

$$\dot{x}_p = Ax_p + Bu + Dw$$
$$y = Cx_p \qquad (16)$$

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

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

$$B = \begin{bmatrix} 0 & -\frac{K_m}{(M+m)} & 0 & \frac{K_m}{m} \end{bmatrix}^T, \qquad (17)$$

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

Where K_m is the motor torque constant and it is given by K_m =150.

4. CONTROLLER DESIGN AND EXPERIMENT

Gain scheduling is a widely used technique for controlling certain classes of nonlinear or linear time-varying systems. Rather than seeking a single robust linear time invariant(LTI) controller for the entire operating range, gain scheduling consists in designing an LTI controller for each operating and in switching controller when the operating conditions change. This section presents systematic tools to design gain-scheduled H_∞ controllers for linear parameter-dependent systems.

The synthesis technique discussed below is applicable to affine parameter-dependent plants with equations, in other words LPV system.

$$P(.,l)\begin{cases} \dot{x} = A(l)x + B_1(l)w + B_2u \\ z = C_1(l)x + D_{11}(l)w + D_{12}u \\ y = C_2x + D_{21}w + D_{22}u \end{cases} \quad (18)$$

where

$$l(t) = (l_1(t),...,l_n(t),\ \underline{l}_i \le l_i(t) \le \bar{l}_i \quad (19)$$

is a time-varying vector of physical parameters and $A(\cdot), B_1(\cdot), C_1(\cdot), D_{11}(\cdot)$ are affine functions of $l(t)$. This is a simple model of systems whose dynamical equations depend on physical coefficients that vary during operation. When these coefficients undergo large variations, it is open impossible to achieve high performance over the entire operating range with a single robust LTI controller. Provided that the parameter values are measured in real time, it is then desirable to use a controller that incorporates such measurements to adjust to the current operating conditions. Such controllers are said being scheduled by the parameter measurements. This control strategy typically achieves higher performance in the face of large variations in operating conditions.

However, in this paper we consider a gain-scheduled control synthesis based on H_∞-control approach.
At first, the parameter dependent plant (18) can be rewritten as following

$$\begin{bmatrix} \dot{x} \\ z \\ y \end{bmatrix} = \begin{bmatrix} A & B_1 & B_2 \\ C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{bmatrix} \begin{bmatrix} x \\ w \\ u \end{bmatrix} \quad (20)$$

$$w := \begin{bmatrix} w_1 \\ w_2 \end{bmatrix},\ z := \begin{bmatrix} z_1 \\ z_2 \end{bmatrix},\ w_1(t) = \Delta(t)z_1(t) \quad (21)$$

where x : states, u : control inputs and y : controlled outputs. w_2 and z_2 are signals to evaluate the control performance, w_1 and z_1 are signals to consider how the time-varying parameter $\Delta(t)$ affects the control system, where $\Delta(t)$ is the time-varying parameter which is assumed to be described by

$$\Delta := \{\mathrm{diag}(\delta_1 I_{q1}, \cdots, \delta_s I_{qs}) : |\delta_i| \le 1\} \subset \mathbf{R}^{q\times q} \quad (22)$$

but may be undefined exactly. For this uncertain plant with $\Delta(t) \in \mathbf{\Delta}$, we design a controller such that the worst case closed-loop RMS gain from w_2 to z_2 does not exceed some level $\gamma \succ 0$. In this problem, if there is no information about $\Delta(t)$ except $\Delta(t) \in \mathbf{\Delta}$, it is considered just as a robust control problem. But, in this paper, we consider that $\Delta(t)$ is the rope length which can be estimated in real time. Therefore, we can use the controlled outputs as well as the information about the time-varying parameter $\Delta(t)$ to produce the control signals. Then a controller suitable for this conception is given by

$$C(s) := \begin{bmatrix} \dot{x}_c \\ z_c \\ u \end{bmatrix} = \begin{bmatrix} A_c & B_{c1} & B_{c2} \\ C_{c1} & D_{c11} & D_{c12} \\ C_{c2} & D_{c21} & D_{c22} \end{bmatrix} \begin{bmatrix} x_c \\ w_c \\ y \end{bmatrix} \quad (23)$$

$$w_c = \Delta(t)z_c(t)$$

In other words, it is said that the structure of the controller is same as that of the plant as illustrated in Fig. 7.

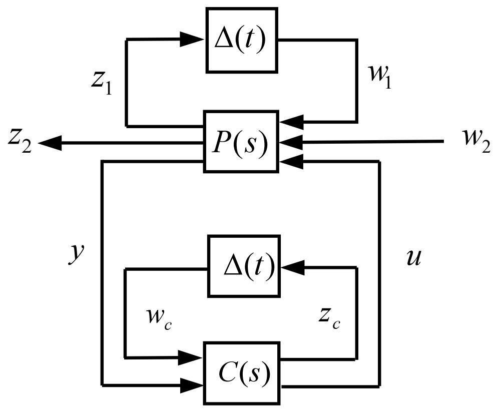


Fig. 7. Gain-scheduling control system

From these facts, a gain-scheduling problem can be described as following.

[Gain-Scheduling Problem] Design a controller $C(s)$ satisfying the following properties :
●the closed-loop system is stable for all admissible parameter trajectories $\Delta(t)$
●the worst-case closed-loop L_2 gain from signal w_2 to z_2 does not exceed some level $\gamma \succ 0$ and this constraint is denoted by

$$\gamma_{rp} := \sup_{\substack{0 \ne w_2 \in L_2 \\ \Delta(t) \in \mathbf{\Delta}}} \frac{\| z_2 \|_{L_2}}{\| w_2 \|_{L_2}} \quad (24)$$

where $\gamma_{rp} \prec \gamma$.

From these, the expression for the control system Fig. 7 is rewritten by $\hat{P}(s)$ the new state space realization and the uncertainty $\hat{\Delta}(t)$:

$$\hat{P}(s) := \begin{bmatrix} \dot{x} \\ z_c \\ z \\ w_c \\ y \end{bmatrix} = \begin{bmatrix} A & 0 & B_1 & 0 & B_2 \\ 0 & 0 & 0 & I_l & 0 \\ C_1 & 0 & D_{11} & 0 & D_{12} \\ 0 & I_l & 0 & 0 & 0 \\ C_2 & 0 & D_{21} & 0 & 0 \end{bmatrix} \begin{bmatrix} x \\ w_c \\ w \\ z_c \\ u \end{bmatrix} \quad (25)$$

$$\hat{\Delta}(t) := \mathrm{diag}(\Delta(t),\ \Delta(t)) \quad (26)$$

It is verified that this synthesis problem can be reduced to the robust control problem.

In this chapter, we evaluate the system performance and show the usefulness of the controlled system by the simulation and experimental studies.

In this paper we consider that the rope length l is varied in the specified range :

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

Considering the fact the ship speed l is an uncertain parameter such that it can be denoted by

$$1/l = l_0 + \alpha_u \Delta, \quad |\Delta| \le 1 \quad (27)$$

where, l_0 is the nominal and α_u is a weighting factor used in controller design process. It is a synthesis technique to follow the affine parameter-dependent representation in equation (18).

Using equations (18)-(27), the controller is calculated as follows:

$$C(s) := \begin{bmatrix} A_c & B_c \\ C_c & D_c \end{bmatrix} \quad (28)$$

where,

$$A_c = \begin{bmatrix} 1.1525\times10^4 & 11.8598 & 70.0479 & 2.8725\times10^5 \\ -7.0122\times10^5 & -6.2663\times10^3 & -8.1602\times10^4 & -3.7224\times10^8 \\ -2.0081\times10^3 & -31.0959 & -570.6733 & -2.6089\times10^6 \\ -2.5670\times10^4 & -17.8869 & -36.7397 & -9.2362\times10^4 \end{bmatrix}$$

$$B_c = \begin{bmatrix} -0.1236 & 1.2368\times10^6 & -5.7673 \\ 1.1294\times10^{-4} & -11.2885 & -0.0050 \\ -0.0328 & 3.2758\times10^3 & -0.2046 \\ 0.2868 & -2.8696\times10^4 & 12.7302 \end{bmatrix}, \quad (29)$$

$$C_c = [-6.9299 \ \ -0.0601 \ \ -0.8064 \ \ -3.6785\times10^3],$$

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

Using this controller, the simulation and experimental results for the designed reduction model as shown in Fig. 3 and Fig. 4 can be obtained.

At first, let us show the initial responses of the open and closed-loop systems where the lope length is 0.5[m]. Fig. 8 illustrates the initial response of open-loop system(uncontrolled case) and Fig. 9 shows the controlled case. In Fig. 9, we can see that the vibration of spreader is effectively suppressed by moving the damper mass as shown in (b) of Fig. 9.

To check the robustness to the disturbance input, let us show the system responses. Where we assume that the rope length varies as the transition pattern of (a) of Fig. 10 and Fig. 11 respectively. In these conditions, Fig. 11 shows the disturbance response of the uncontrolled case, and Fig. 11 is the controlled case. Especially, in Fig. 11, (b) illustrates the displacement of spreader, where the step type disturbance input to the plant irregularly. Comparing these two cases, we can see that good control performance is obtained in the existence of disturbance.

From the simulation and experimental results, it is clear that the robustness to the disturbance input and good control performance are obtained. Also, the usefulness of the proposed anti-sway system is verified and the possibility that the considered system can be easily applied to the real plants is certified in a sense.

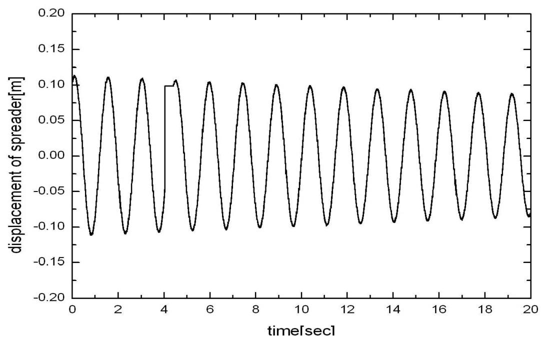
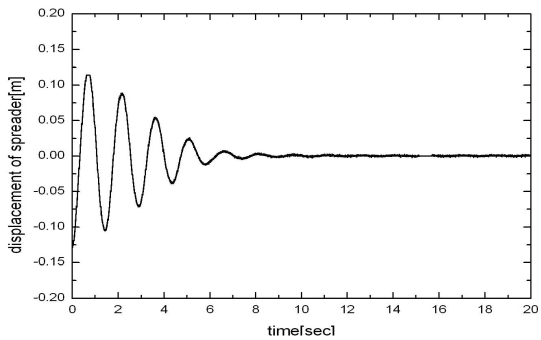
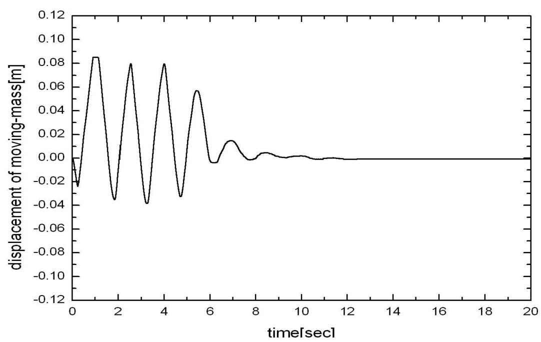


Fig. 8. Initial response(open-loop system), $l = 0.5$[m]

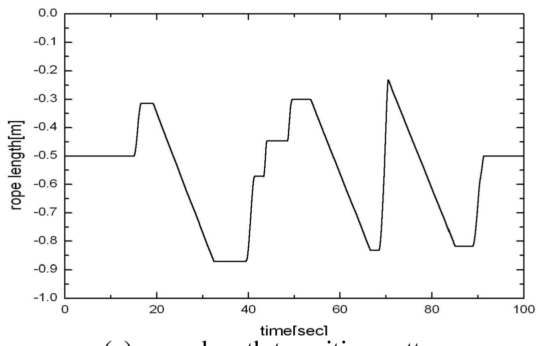


(a) displacement of spreader

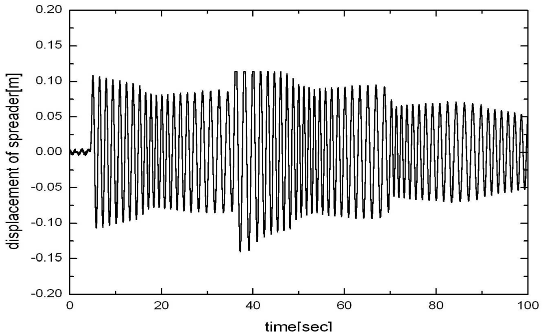


(b) displacement of moving mass

Fig. 9. Initial response(closed-loop system), $l = 0.5$[m]

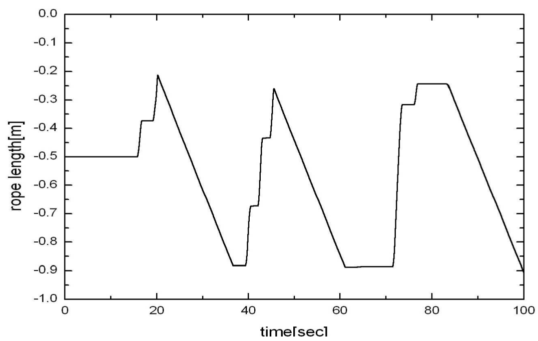


(a) rope length transition pattern

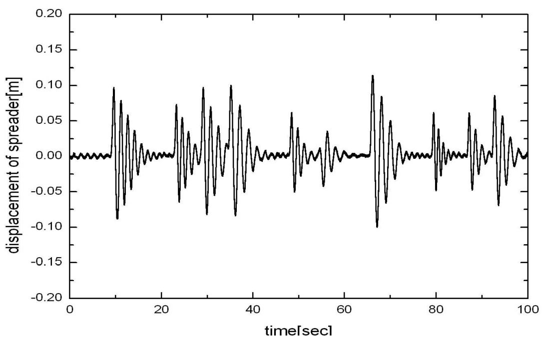


(b) displacement of the spreader

Fig. 10. Disturbance response when the rope length varies(open-loop system)



(a) rope length transition pattern



(b) displacement of moving mass

Fig. 11. Disturbance response when the rope length varies(closed-loop system)

5. CONCLUDING REMARKS

In this study, a new type of swing motion control system for the crane has been considered and the usefulness of the considered system has been verified by 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, in this paper we have considered that the rope length varies and designed a control system to achieve desirable control performance and preserve the system stability based on the gain-scheduling approach.

The advantage of this system is that the system can be easily applied to the real system and desirable anti-sway effect can be obtained.

ACKNOWLEDGEMENT

This work was partially supported by Brain Korea 21 Project in 2004.

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