A New Anti-Sway Control Scheme for Trolley Crane System

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***Abstract* - This paper proposes a new vision-sensorless anti-sway control scheme, where the sway angle of the load is estimated with a simple acceleration sensor, which can be equipped on the trolley. The sway of the suspended load is suppressed by adding the damping factor, which is calculated with the estimated sway angle. While the conventional electric anti-sway systems need the expensive external vision sensor system, the proposed scheme is easily realized on the trolley speed controller with an acceleration sensor. The effectiveness of the proposed scheme is verified by the computer simulation and experiment with the miniature of the trolley crane system.**

I. INTRODUCTION

Trolley crane systems are widely used in industrial applications. In the trolley crane systems, the load suspended from the trolley on the crane is subject to sway caused by improper operator's control of trolley and disturbance on the load (e.g. wind, collision with an object, etc). Thus, concerns about these sway problem have been increased and many researches for anti-sway system have been progressed for a long time after the first appearance of the crane. Generally, anti-sway solutions are classified as follows [1].

- Mechanical or hydro-mechanical anti-sway systems.
- Electrical anti-sway systems.

(a) Predetermined speed reference profile method.

(b) Detecting and control sway angle with vision system.

(c) Vision-sensorless control.

In mechanical or hydro-mechanical anti-sway systems, which increase natural damping, has been used to minimize load sway at all times. While this technique has been generally successful, it is accompanied by both high initial and maintenance cost.

Scheme (a) is to use a predetermined speed reference profile, which has been simulated or recorded to produce minimum sway. For automatic moves where the starting and final positions of the load are known in advance, this approach works well. However, there is no ability to reduce the sway caused by random motion of the trolley induced by improper operator control or the disturbance.

Scheme (b) is to use a sway regulator with sway angle feedback. In this scheme, the sway angle is detected with vision sensor (e.g. camera) and image processing system. There are some strategies to control the sway using the vision system, for example, fuzzy control [2], neural network [3], non-linear control [4] and so on. While this scheme is suitable for reducing or eliminating the sway induced by random trolley motion, it requires a complex, expensive vision system. Also, because such a vision sensor is subject to the weather (a dense fog, heavy rain, direct sunlight, and so on), it is not suitable to the trolley crane in the quayside (e.g. RTGC-Rubber Tyred Gantry Crane, RMGC-Rail Mounted Gantry Crane, RMQC-Rail Mounted Quayside Crane, and so on)

Recently, many researches have been actively carried out for scheme (c). It is to control the sway without the additional vision system, for example, adaptive control using dynamic feedback linearization [5], impedance control with trolley speed feedback [6], linear regulator using sway angle observer [1] and so on. In [1] and [6], the linearized model of the crane is used to design the controller and the observer. That is, the sway angle is estimated with the speed of the trolley. Considering the mechanical structure of the trolley (gear ratio, diameter of the wheel, etc), the speed of the trolley can be calculated with the rotational speed of the trolley motor. But, there are nonlinearity and uncertainty in the mechanical structure, such as the backlash of the gear, the slip of the wheel (in case of the self driven trolley), the droop of the trolley rope (in case of the rope towed trolley), the abraded wheel, and so on. Therefore, the calculated speed of the trolley is not exactly equal to the real speed of the trolley. In the steady state, such nonlinearity and uncertainty can be neglected. But, in case of using the calculated speed to estimate the sway angle, they cause the error on the estimation of the sway angle. For the estimation of the sway angle, it is not proper to use the calculated speed of the trolley.

In this paper, a new vision-sensorless anti-sway control scheme is proposed. It regulates sway angle using the acceleration sensor, which can be equipped on trolley as the cost-effective acceleration sensors are available. The sway angle is estimated with the real acceleration of the trolley instead of the rotational speed of the trolley motor. The sway is suppressed by additional damping factor, which is generated by anti-sway regulator with the estimated sway angle.

II. SYSTEM DYNAMICS

Fig.1 shows the simplified diagram of a trolley crane system with a suspended load. The load, which is suspended from point 0, is assumed to be a rigid body symmetric about its axial axis with mass m and center point G of mass m. The position vector of the point of suspension with respect to the fixed axes coordinate system is

$$r_0 = x\bar{i} + y\bar{j} = x\bar{i}. \tag{1}$$

where $\bar{i}$, $\bar{j}$ are x, y directional unit vectors, respectively.

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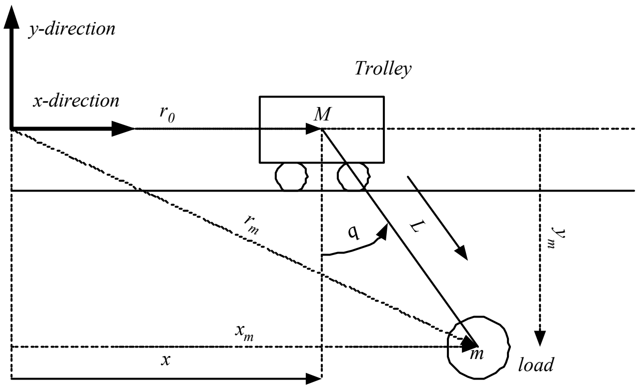


Fig. 1. Trolley crane system.

Using the moving circular coordinate q to indicate the relative position of G with respect to 0, the position vector of G can be written as follows

$$\bar{r}_m = x_m \bar{i} + y_m \bar{j}. \tag{2}$$

where $x_m = x + L\sin q$, $y_m = -L\cos q$, L is the length of suspension rope and q is the sway angle of load.

For simplicity, in this paper, the following assumptions are made:

(a) The trolley and the load can be regarded as point masses.
(b) Friction force which may exists in the trolley can be neglected.
(c) Elongation of the rope due to tension force is neglected.
(d) The trolley and the load move in the x-y plane.

The kinetic energy is represented by

$$\mathcal{T} = \tfrac{1}{2} M\dot{x}^2 + \tfrac{1}{2} m(\dot{x}_m{}^2 + \dot{y}_m{}^2), \tag{3}$$

where M and m are the mass of the trolley and the load, respectively.

The potential energy is also represented by

$$\mathcal{V} = mgy_m = -mgL\cos q, \tag{4}$$

where g is the gravitational acceleration.

The Lagrangian function $\mathcal{L} = \mathcal{T} - \mathcal{V}$ is obtained as:

$$\begin{aligned}\mathcal{L} &= \tfrac{1}{2} M\dot{x}^2 + \tfrac{1}{2} m(\dot{x}^2 + \dot{L}^2 + L^2\dot{q}^2 \\ &\quad + 2\dot{x}\dot{L}\sin q + 2\dot{x}L\dot{q}\cos q) + mgL\cos q.\end{aligned} \tag{5}$$

Now, we will use the general form of Lagrange's equations with a Lagrange multiplier. With Lagrange's equations

$$\frac{d}{dt}\left(\frac{\partial\mathcal{L}}{\partial\dot{q}}\right) - \frac{\partial\mathcal{L}}{\partial q} = Q, \tag{6}$$

The equations of motion associated with the generalized coordinates $q = [x, q, L]^T$ are represented. From the Lagrange's equation (6), we formulate the relation $\frac{d}{dt}\left(\frac{\partial\mathcal{L}}{\partial\dot{q}}\right) - \frac{\partial\mathcal{L}}{\partial q} = 0$, and the following equation can be derived.

$$\ddot{x}\cos q + L\ddot{q} + 2\dot{L}\dot{q} + g\sin q = 0. \tag{7}$$

Next, the relation about the 2-directional forces is also formulated,

$$\frac{d}{dt}\left(\frac{\partial\mathcal{L}}{\partial\dot{x}}\right) - \frac{\partial\mathcal{L}}{\partial x} = Q_x, \quad \frac{d}{dt}\left(\frac{\partial\mathcal{L}}{\partial\dot{L}}\right) - \frac{\partial\mathcal{L}}{\partial L} = Q_l. \tag{8}$$

From (8), the following equations related to the x and L-directions are obtained.

$$\begin{aligned}&(m+M)\ddot{x} + mL(\ddot{q}\cos q - \dot{q}^2\sin q) \\ &+2m\dot{L}\dot{q}\cos q + m\ddot{L}\sin q = Q_x.\end{aligned} \tag{9}$$

$$m\ddot{x}\sin q - mL\dot{q}^2 + m\ddot{L} - mg\cos q = Q_L \tag{10}$$

In summary, the equations of motion associated with the generalized coordinates x, q, L can be written, respectively, as

$$\begin{aligned}x \;:\; Q_x &= (M+m)\ddot{x} + mL(\ddot{q}\cos q - \dot{q}^2\sin q) \\ &\quad + m\ddot{L}\sin q + 2m\dot{L}\dot{q}\cos q.\end{aligned} \tag{17}$$

$$q \;:\; 0 = \ddot{x}\cos q + L\ddot{q} + 2\dot{L}\dot{q} + g\sin q. \tag{18}$$

$$L \;:\; Q_L = m\ddot{L} + m\ddot{x}\sin q - mL\dot{q}^2 - mg\cos q. \tag{19}$$

Assume that the sway angle is sufficiently small and the length of the rope is constant as

$$\sin q \approx q, \quad \cos q \approx 1, \quad \dot{q}^2 \approx 0, \quad \dot{L} = \ddot{L} = 0. \tag{20}$$

Then, the equations of motion can be approximated as

$$Q_x = (M+m)\ddot{x} + mL\ddot{q}. \tag{21}$$

$$0 = \ddot{x} + L\ddot{q} + gq. \tag{22}$$

$$Q_L = m\ddot{x}q - mg. \tag{23}$$

Practically, q is very small value. So, (23) can be closely approximated by $Q_L = m\ddot{x}q - mg \approx -mg$. This suggests that the hoist motion is decoupled from the trolley.

III. Control System Design

A. Sway Angle Estimation Using an Acceleration Sensor

Physically, when the trolley speed changes, the load has an oscillation. If the acceleration of the trolley is detected, the sway angle of the load can be estimated. The equation (22) can be rewritten as follows.

$$q(s) = -\frac{1}{L}\frac{1}{s^2 + g/L}\ddot{x}(s). \tag{24}$$

It shows that the acceleration of the trolley is the input of the sway angle of the load. It means that the sway angle can be estimated based on the measured acceleration.

B. Anti-Sway Control Using Sway Angle Estimator

Assuming that the sway angle is sufficiently small, the equation $x_m = x + L\sin q$ is rewritten as follows.

$$q = (x_m - x)/L. \tag{25}$$

After differentiation of (25), the acceleration of sway angle is

$$\ddot{\theta} = (\ddot{x}_m - \ddot{x}) / L. \tag{26}$$

From (22), (25) and (26), The following equation is obtained.

$$\ddot{x}_m + \frac{g}{L} x_m = \frac{g}{L} x. \tag{27}$$

If the trolley is speed-controlled, (27) can be represented as follows.

$$v^* = \frac{L}{g} \ddot{v}_m + v_m. \tag{28}$$

where v^* and v_m represent the speed reference of the trolley and the speed of the suspended load, respectively. The equation (28) is the second order transfer function, which has no damping factor. To suppress the sway of the suspended load, the damping factor can be added to the transfer function. The speed reference v^* can be modified as follows.

$$v^* = v^{**} - K\dot{v}_m, \tag{29}$$

where $K\dot{v}_m$ is the damping factor.

From (28) and (29), the following equation is obtained.

$$\ddot{v}_m + \frac{g}{L} K\dot{v}_m + \frac{g}{L} v_m = \frac{g}{L} v^{**}. \tag{30}$$

From (30), the damping coefficient K can be defined as follows

$$K = 2\zeta / \omega_n, \tag{31}$$

where ζ is the damping ratio, natural frequency $\omega_n = \sqrt{g/L}$.

Fig. 2 shows the step response of the system (30) according to the various damping ratio.

Substituting (26) into (22), the equation is expressed as

$$\dot{v}_m = \ddot{x}_m = -g\theta. \tag{32}$$

Finally, the speed reference of trolley can be modified from (29) and (32).

$$v^* = v^{**} + Kg\theta. \tag{33}$$

Fig. 3 shows the block diagram of the proposed control system.

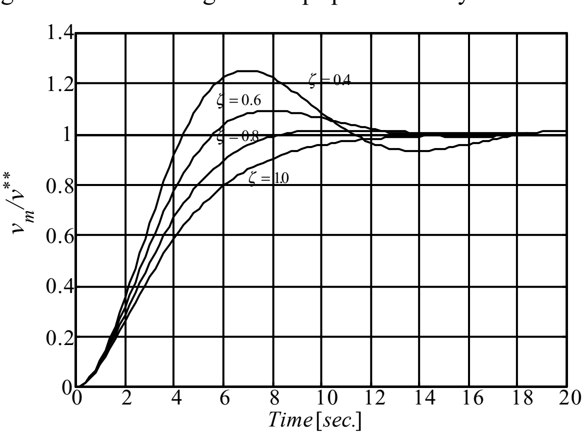


Fig. 2. Step response of the system.

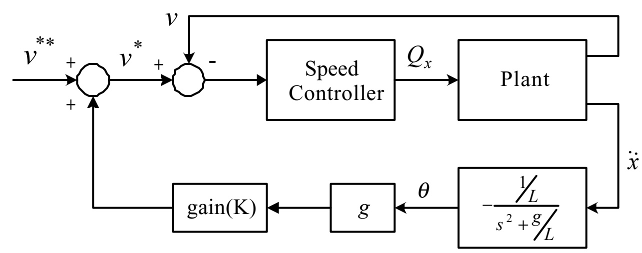


Fig. 3. Control block diagram.

IV. Simulations

To verify the performance of the proposed method, simulations have been carried out with Simulink. The parameters used in the simulation are as shown in Table I. The target crane in the simulation is a kind of RMQC (Rail Mounted Quayside Crane) in the post pana-max class. Generally, the structure of the trolley divides RMQC into 3 classes: machinery trolley, self-driven trolley, and rope-towed trolley. In case of rope-towed trolley, the trolley mass M includes only sheaves, operator's cabin, and trolley base without motors and rope drums. It is 1/3 of machinery trolley. The load mass m includes load, spreader, and head block. The rope is assumed to be constant. Its length is assumed to be maximum value.

The simulation consists of two parts. One is the estimation of the sway angle, and the other is the anti-sway control with the estimated sway angle. Real and estimated sway angle of the suspended load are represented in Fig. 4 There is little difference between real angle and estimated angle of the suspended load. Fig. 5(a) shows the movement of the trolley without anti-sway control. It is known that the sway still remains even though the trolley is at rest. Fig. 5(b) shows the result of the anti-sway control. After normal operation (acceleration, constant speed traversing, and deceleration), anti-sway control begins on 16 sec. It is shown that the sway angle is effectively suppressed in 3 sec because of the damping term.

TABLE I
SIMULATION PARAMETERS

Trolley mass M	26 *ton*
Load mass m	67.3 *ton*
Rated power of the trolley motor	110 *kW*
Rated speed of the trolley motor	1150 r/min
Rated speed of the trolley	180 *m/min*
Rated acceleration time	6 *sec*
Length of the hoist rope L	40 *m*
Gravitational acceleration g	9.8 m/s^2

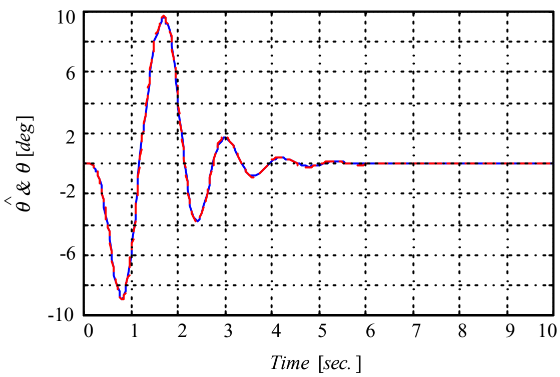
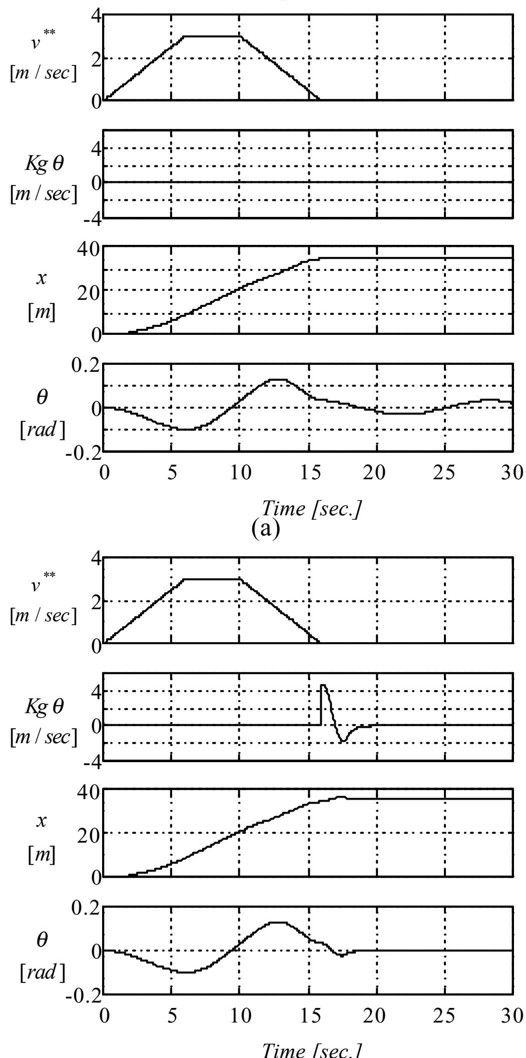


Fig. 4. Real sway angle (solid) and estimated sway angle (dotted) of the suspended load.



(a)

(b)

Fig. 5. Normal operation of the trolley crane.
(a) Without anti-sway control
(b) With anti-sway control

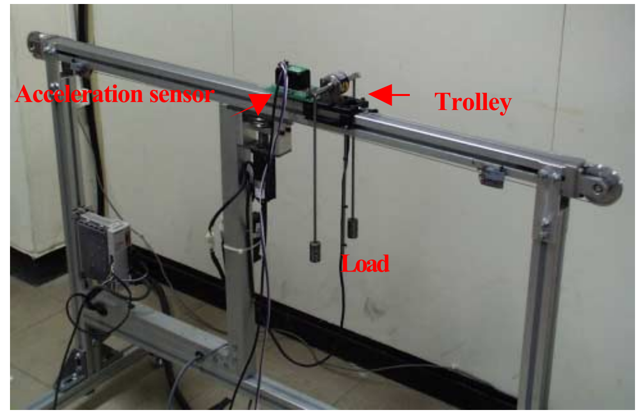


Fig. 6. Experimental setup.

V. EXPERIMENTAL RESULTS

To confirm the effectiveness of the proposed scheme, some experiments have been performed with a miniature of a trolley crane system. It has AC servomotor, inverter, acceleration sensor, sway angle sensor, and control board. The sway angle sensor is the rotary type absolute encoder. It is used to compare the real sway angle with estimated sway angle. Fig. 6 shows the experimental system. The parameters of the experimental system are represented in Table II. The specifications of the acceleration sensor are shown in Table III.

To verify the performance of the proposed anti-sway control scheme using the acceleration sensor, the result of anti-sway control with the real angle is shown in Fig.7. The result of the proposed scheme is represented in Fig. 8. It can be seen from Fig. 7 and 8 that there is little difference between anti-sway control with real sway angle and with estimated sway angle.

TABLE II
EXPERIMENTAL PARAMETERS

Trolley mass M	1 kg
Load mass m	0.8 kg
Rated power of the trolley motor	200 W
Length of the hoist rope L	0.305 m
Gravitational acceleration g	9.8 m/s^2

TABLE III.
SPECIFICATIONS OF THE ACCELERATION SENSOR.

Model	Jewell LCA-100
Acceleration range	±0.5G
Natural frequency	60Hz
Nonlinearity	0.05% of full range
Repeatability	0.005G

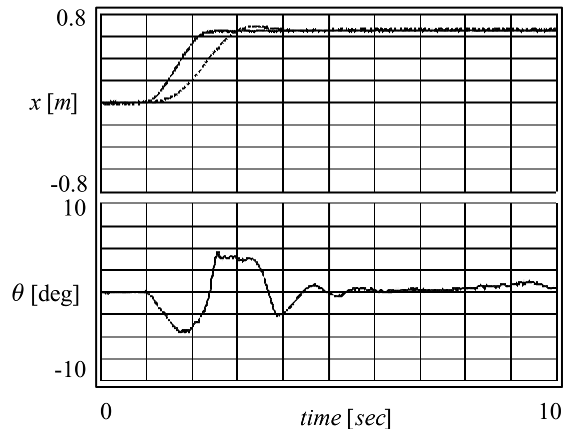


Fig. 7. Anti-sway control with real angle.
(From top to bottom, position reference (solid), real position (dotted) and real sway angle of suspended load)

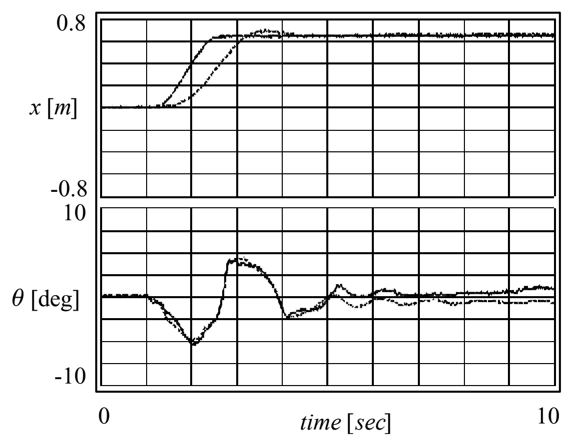


Fig. 8. Anti-sway control with estimated angle.
(From top to bottom, position reference (solid), real position (dotted), real sway angle (solid) and estimated sway angle (dotted))

VI. CONCLUSIONS

In this paper, a new scheme is proposed to control the sway of a suspended load for the trolley crane system. The proposed scheme controls the sway using acceleration sensor instead of the external vision sensor system. The feasibility of it has been verified with the experiment. It has been compared with the conventional anti-sway control method, which uses the vision sensor system. In the experimental results, there is little difference between method using the external vision sensor system and method using acceleration sensor. It comes to reduce the cost of trolley crane system without degrading the performance

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