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Model Reference Simple Adaptive Control System using an Ideal-Input Model for Overhead Travelling Cranes with Load Mass Change

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***Abstract*-This paper proposes an ideal-input model reference simple adaptive control scheme (MRSACS) with an adaptive proportional internal derivative (PID) mechanism in a reference model for anti-sway and position control of an overhead travelling crane that uses various pendulum (rope) lengths, after which its performance is verified via various simulations. It was found that the system's parallel PID adaptive mechanism was robust against load change properties, and the simulation results showed that the effectiveness of pendulum length and weight changes were satisfactory for simultaneous position control and anti-sway measures.**

Keywords- Overhead Travelling Crane Model; Model Reference Adaptive Control; Position Control

I. INTRODUCTION

Recently, increasingly advanced crane systems have been introduced into factories for conveying work between locations. Also, hopes have risen that fully automated operation of such cranes can soon be achieved. One aspect of crane operation, anti-sway control, which considers the setting up, turning down, load raising, and load lowering characteristics of slewing cranes such as jib cranes, has emerged as an important crane control study topic [1-8]. Anti-sway control is also an important factor when considering raising and lowering loads for running cranes such as overhead traveling cranes [1-6, 9].

When considering automated control, it is possible to envision the use of applied artificial intelligence (AI) that might perform actions in ways that would apply a skilled operator's know-how to a rule set and knowledge base, providing that a fuzzy logic control method that made use of various membership functions, a fuzzy rule table, etc., was adopted and applied to actual equipment [3, 5, 8].

There are also increasing numbers of recent studies on adopted adaptive and/or sliding mode crane control [1-3, 5]. Meanwhile, overhead crane feedback control has been evaluated from the standpoint of general robustness [9], even though the sliding-mode control system that was evaluated in that study was a robust non-linear control. Robust control [10], which generally services one target, and proportional integral derivative (PID) control [11, 12] are currently the most basic and popular control methods utilized by control engineers.

This paper aims to propose a control method for transporting a hanging load suspended from an overhead traveling crane to a target position in a short time after interference is applied to a travel gear by changing the mass of the hanging load, while simultaneously controlling the immediate and residual sway of the hanging load during transportation. Moreover, model reference adaptive control and robustness imposed by integral control are combined in this study.

II. CONTROLLED OBJECTS

In Fig. 1, a physical model of an overhead travelling crane is shown in which non-linear model equations for the crane and a friction equation for the cart have been introduced. The notation used in the simulation is shown in Table 1.

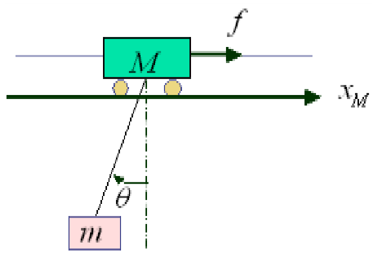

Fig. 1 Overhead travelling crane physical model

TABLE 1 NOTATION USED IN SIMULATION MODEL

x_M	Position of cart [m]
θ	Swing angle [rad]
M	Mass of cart [kg]
m	Mass of pendulum [kg]
g	Gravity acceleration [m/s^2]
f	Tension of belt [N]
f_M	Motor generation force [N]
f_F	Friction force [N]
δ	Small moving distance of cart [m]
ν	Dynamic friction coefficient
$\mu_{\max}$	Maximum Static Friction Coefficient
l_g	Length from hanger center of cart to gravity center of pendulum [m]
J_m	Polar inertia moment of mass and pendulum around hanger center of cart [kg·m^2]
E	Motor input voltage [V]

Energy equation (kinematic energy and potential energy):

$$T = \frac{1}{2}Mx_M^2 + \frac{1}{2}m(\dot{x}_M - l_g\dot{\theta}\cos\theta)^2 + \frac{1}{2}J_m\dot{\theta}^2$$
$$U = mgl_g(1-\cos\theta) \tag{1}$$

Lagrange kinematic equation:

$$\frac{d}{dt}[\frac{\partial T}{\partial \dot{q}_s}] - \frac{\partial T}{\partial q_s} + \frac{\partial U}{\partial q_s} = Q_s \tag{2}$$

where $q_1 = \theta$, $q_2 = x$.

Non-linear model equation of crane (hook and cart):

$$-m(\ddot{x}_M l_g \cos\theta - \dot{x}_M l_g \dot{\theta}\sin\theta - l_g\ddot{\theta}\cos^2\theta - 2l_g\dot{\theta}^2\cos\theta\sin\theta)$$
$$+ J_m\ddot{\theta} + mgl_g\sin\theta = 0 \tag{3}$$

$$(M+m)\ddot{x}_M - ml_g\ddot{\theta}\cos\theta - ml_g\dot{\theta}^2\sin\theta = f(\mathrm{t}) \tag{4}$$

Where

$$J_m = \frac{4ml_g^2}{3} + ml_g^2 \tag{5}$$

Approximating only trigonometric functions by small variation assumption around an equilibrium point $\theta = 0$,

$$\begin{bmatrix} \dot{x}_M \\ \dot{\theta} \\ \ddot{x}_M \\ \ddot{\theta} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & -(a_1/a_0)\dot{\theta}^2 - (a_3/a_0)\dot{\theta} & 0 & 0 \\ 0 & -(a_2/a_0)\dot{\theta}^2 - (a_4/a_0)\dot{\theta} & 0 & 0 \end{bmatrix} \begin{bmatrix} x_M \\ \theta \\ \dot{x}_M \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ b_1/a_0 \\ b_2/a_0 \end{bmatrix} f(t)$$

$$a_0 = \{(M+m)J_m + Mml_g^2\}, a_1 = ml_g(J_m + ml_g^2), a_2 = m^2 l_g^2,$$
$$a_3 = m^2 l_g^2 g, a_4 = (M+m)ml_g g, b_1 = J_m + ml_g^2, b_2 = ml_g \tag{6}$$

Friction model equation of cart and friction force:

$$\ddot{x}_M(t) = -g\{\mu_{\max}\delta(t) + \nu(\dot{x}_M(\mathrm{t}))\left|\dot{x}_M(t)\right|\}\,\mathrm{sgn}(\dot{x}_M(\mathrm{t})) + \frac{1}{M+m}f(t) \tag{7}$$

$$f(t) = f_M(t) - f_F(\mathrm{t}) \tag{8}$$

Motor model (from experimental data):

$$G_{motor}(s) = \frac{T_{mu}s + 1}{T_{md}s + 1} \tag{9}$$

The Lagrange kinematic Eq. (2) using energy Eq. (1) results in Eqs. (3) and (4). Eq. (6) was obtained from the force balance. Kinematic friction acts on the cart as it moves, while static friction was only considered to be present when cart movement begins. The coefficients of kinematic friction were considered as functions of the cart speed. They were obtained by piecewise linearization from the friction property graph describing in the Mechanical Engineering Hand Book [12]. Eq. (8) was obtained experimentally from the frequency response. The results of parameter identification were T_{md} = 0.01[s] and T_{mu} = 0.001[s]. The precise model for simulation is well known [13] as follows.

$$\begin{bmatrix} \dot{\omega}_m(t) \\ \dot{i}_m(t) \end{bmatrix} = \begin{bmatrix} -\frac{b}{J_m} & \frac{k_t}{J_m} \\ -\frac{k_e}{L_a} & -\frac{R_a}{L_a} \end{bmatrix} \begin{bmatrix} \omega_m(t) \\ i_m(t) \end{bmatrix} + \begin{bmatrix} -\frac{1}{J_m}\tau_l(t) \\ \frac{1}{L_a}v_m(t) \end{bmatrix} \tag{10}$$

where ω_m is motor angular velocity [rad/s], i_m is motor current [A], v_m is motor voltage [V], τ_l is load torque [Nm], J_m is inertia of motor and load [$\mathrm{Nms^2/rad}$], b is viscous resistance, k_t is torque constant [Nm/A], k_e is counter electromotive force constant [Vs/rad], R_a is armature resistance [Ω], and L_a is armature inductance [H].

Then, an inverse compensator of the current feedback PID controller for a motor experimental model Eq. (8) is as follows.

$$Inv_{motor} = \frac{T_{md} + 1}{T_{mu}s + 1} \tag{11}$$

Moreover, an inverse compensator of velocity feedback PID controller for a crane experimental model is as follows.

$$Inv_{crane} = \frac{T_{cd} + 1}{T_{cu}s + 1} \tag{12}$$

III. REFERENCE MODEL

A. Trapezoidal Type Velocity Reference Value

Herein, an equal length leg trapezoidal velocity reference pattern is adopted for the moving control of a cart. These patterns have the advantage of short transportation times because they make it simple to calculate moving time without overshoot and/or undershoot.

Where the move starting time is t_0, acceleration time is from t_0 to t_1, constant speed moving time is from t_1 *to* t_2, and deceleration time is from t_2 to t_3. Then, *a* is inclination and *T* is the time for acceleration and deceleration. The following equation is obtained if the cart running process is described mathematically.

$$x'_M = \begin{cases} 0 & 0 \le t \le t_0 \\ \frac{1}{2}a(t-t_0)^2 & t_0 \le t \le t_1 \\ aT(t-t_1)+\frac{1}{2}aT^2 & t_1 \le t \le t_2 \\ -\frac{1}{2}a(t-t_2)^2 + aT(t-t_1)+\frac{1}{2}aT^2 & t_2 \le t \le t_3 \\ aT(t_3-t_1)=S & t_3 \le t \end{cases} \tag{13}$$

B. Operation Time for Stopping the Swing

The experimental device input used here is motor voltage, which is assumed to be a simple ideal input, such as a trapezoidal shape, if the linear relation between the motor voltage and the cart speed is maintained steadily. Consequently, an ideal model is created by ignoring the motor dynamics and the friction between the cart and the track (the running road of the cart), and by ignoring any disturbance and inference force imposed on the cart by the swing and mass of the hanging load. The reference model configuration is shown in Fig. 2. A hook model of input $\dot{x}'_M$ and output θ', $\dot{\theta}'$ in the hook separate reference model can be derived from nonlinear Eq. (3).

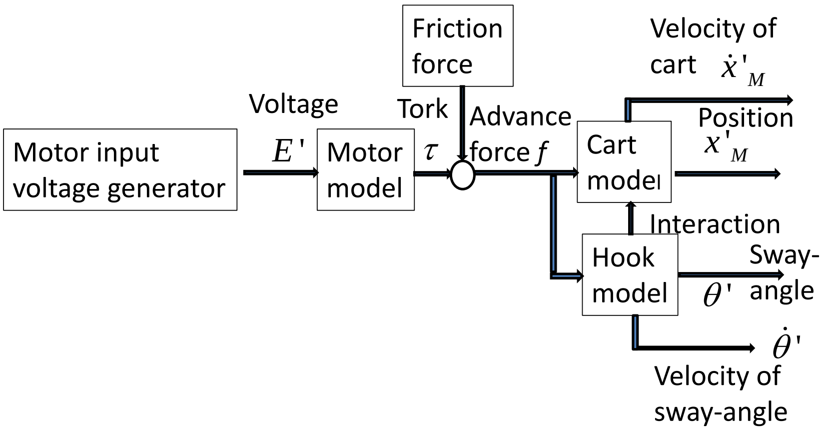

Fig. 2 Reference model configuration

A simulation showing how cart movement changes the hanging load sway angle, as induced from this reference model, was created using MATLAB/Simulink (Mathworks Co. Ltd.), from which the mass of pendulum *m* [kg], the length of pendulum l_g [m], the input rate to motor a_M [V/s], the operation time *T* [s], and the swing angle θ' [rad] shown in Fig. 3, were obtained. The parameter values used in this simulation are shown in Table 2.

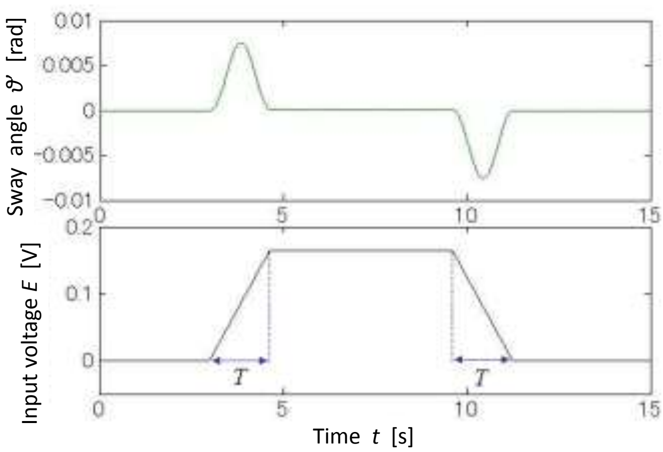

Fig. 3 Sway angle and input voltage

TABLE 2 SIMULATION CONDITIONS

m [kg]	0.02
l_g [m]	0.5
a_M [v/s]	0.1
T [s]	1.63

A. Computation of Operation Time

The results of the computation of operation time T for a swinging stop based on a pendulum length change are shown in Fig. 4. Here, the equation for the computation of operation time T was searched for via the lattice method as an initial value of Eq. (14), and the equation used to evaluate the integral of signal evaluation (ISE) was Eq. (15). The optimal operation time was then used in the reference model.

$$T_g = 2\pi\sqrt{\frac{l_g}{g}} \tag{14}$$

$$ISE = \int_{t_f}^{t_f+t_g} \theta_e(t)^2 dt \tag{15}$$

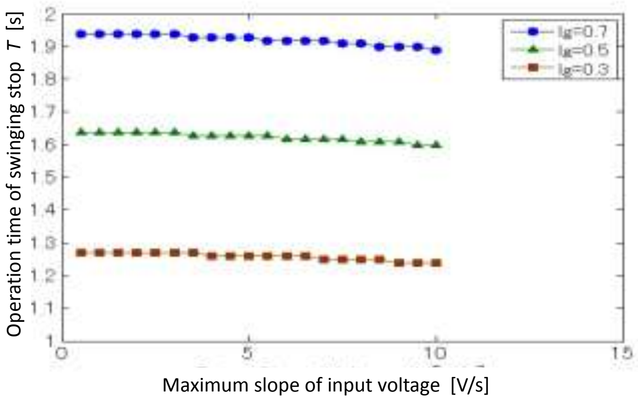

Fig. 4 Operation time for swinging stop on pendulum length change

IV. MODEL REFERENCE SIMPLE ADAPTIVE CONTROL SYSTEM AND PROPERTY OF ROBUSTNESS

First, the Simple Adaptive Control System (SACS) is fully defined in this section. It is necessary to consider the interference power disturbance imparted to the cart by the hanging load sway. This includes the frictional force of the cart and the cart motor dynamics, which are not considered in the reference model of an actual ceiling crane.

A static, linear relationship between the input voltage and the rotational speed of the motor is maintained by introducing the inverse system for the motor dynamics, as shown in Fig. 4.

In addition, as an error function, the difference between the actual ceiling crane output and the reference model output on the position and the speed of the cart is used to absorb the frictional force of the cart as well as the interference power disturbance to the cart imparted by the hanging load sway. At this time, an attempt was made to simplify the adaptive mechanism by using the PID control, even though the error function and the nonlinear I/O function are commonly used in conventional adaptive mechanisms [14].

The results were then adapted to simultaneously obtain cart position control and achieve sway control over the hanging load, which was accomplished by altering the length of the rope in the reference model in order to match the actual rope length change. Since it was assumed that there would be no changes in the mass of the hanging load used in the reference model, the technique used for making the adjustments needed to achieve sway control over the hanging load used the robust controlling effect of the steady-state deviation to an unknown disturbance. This was imposed by the natural robust function of the MRSACS integration (Fig. 5).

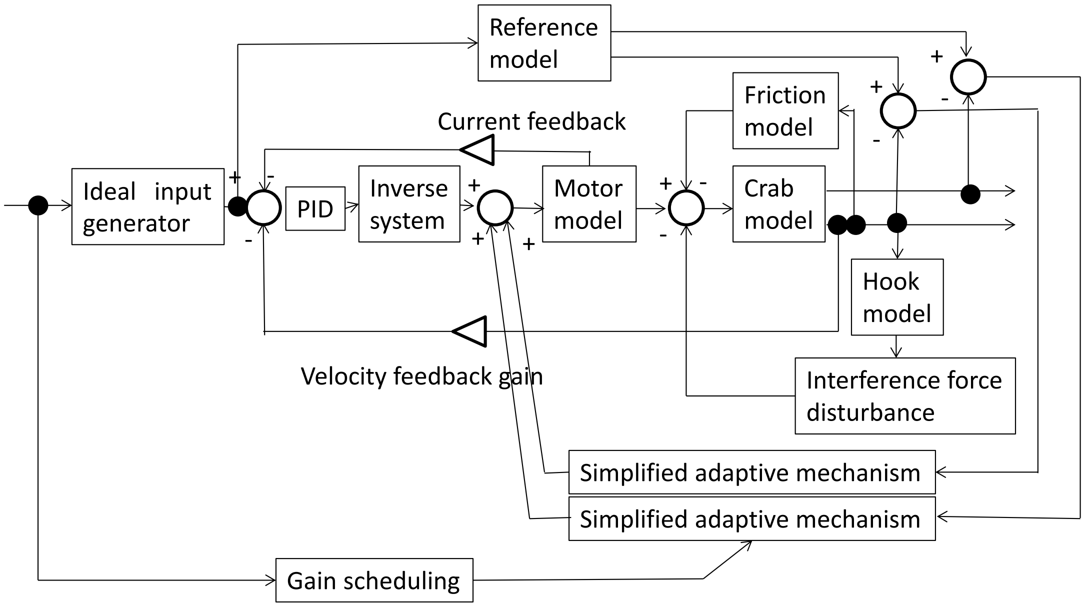

Fig. 5 Configuration of MRSACS for a crane with crab

V. SIMULATION

The simulation used MATLAB/Simulink to examine the controlling effect for load swing imposed by the SACS technique shown in Section 4. Each of the parameter values used by the simulation is explained in Table 3.

TABLE 3 SMALL MODEL SIMULATION CONDITIONS

M [kg]	0.17
m [kg]	0.15–1.05
l_g [m]	0.5
a_M [V/s]	0.12
T [s]	1.64
μ_{max}	0.07

A. Simulation Method

The cart is moved by the input shown in Section 3. It is assumed that l_g = 0.5 [m] in this simulation, and that the length of the rope does not change while the cart is in motion. As an initial condition at the beginning of the simulation, the position of the cart was assumed to be 0 [m], and the sway of the hanging load was completely 0 [rad].

Three seconds after the start of the simulation, the cart begins moving and travels 0.4 [m], which was the distance set for the destination. The position trend of the cart and the sway angle trend of the hanging load were monitored over the time process of the cart movement.

The weight of the hanging load was initially set to m = 0.15 [kg], and the parameters of P, I, and D of the PID control were adjusted. After the PID parameters were adjusted to avoid overshoot, the parameter values were fixed, and the cart position and the sway angle were compared with the cases of m = 0.15, 0.45, 0.75, and 1.05 [kg].

B. Simulation Results

The cart position and hanging load sway trends are shown in Figs. 6 and 7.

From Fig. 6, it can be understood that the cart could move almost all the way to the target position without a position error deviation, even if the mass of the hanging load was changed. Based on previous experience, a safety position control (without overrun) was used so that the speed and the position of the double reference model, as well as the parallel state-action-reward-state-action (SARSA), would work effectively. Since there was change of robustness in the hanging load mass, cart position overrunning was not expected to occur.

Moreover, from Fig. 7, it was thought that the hanging load sway angle was controlled within the practicable range of $m \leq$ 0.45 [kg] for this simulation condition during the periods when cart speed was stable and after the cart had stopped.

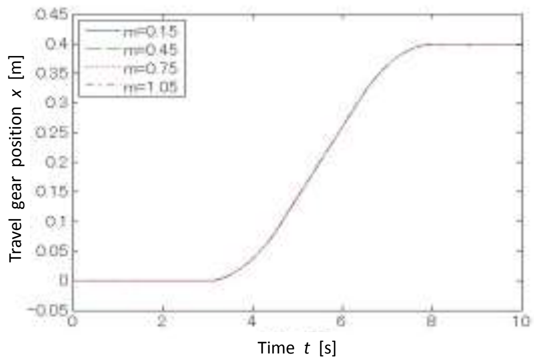

Fig. 6 Travel gear position for load mass change in the model

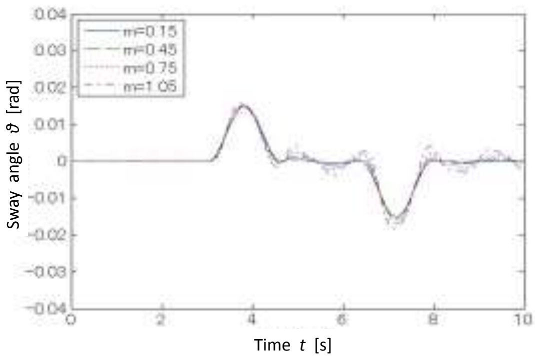

Fig. 7 Sway angle for load mass change in the model

The following two major merits were confirmed in this and another simulation:

1) The MRSACS proposed here was able to adapt to various conditions, such as simultaneously achieving cart position and hanging load sway control, by varying the length of the pendulum among the double reference models in order to reflect changes to the pendulum length in the ceiling travel crane model.

2) Hanging load sway control could be achieved for changes in the hanging load mass that were not included in the reference model by applying the controlling effect of the steady-state deviation to the disturbance using the robustness action integrated into the simple adaptive mechanism.

Three types of PID control robustness will now be discussed. A-type robustness of PI control is introduced in Section 12 of Ref. 10, while N-type robustness of I (with gain KI) control with a LEad Inverse COmpensator (collect capital letters and in the Japanese style: KILEICO) is proven in the literature [11, 13]. The stability of SImple RObust COntroller with INverter (collect capital letters with h and in the Japanese style: ShIRO COIN)-type robustness is also proven in the literature [14]. The results of this last case, which involves tuning, may be the ShIRO COIN-type robustness or some fourth type of robustness.

VI. VERIFICATION

A. Experiment Devices

A verification experiment was performed to show the effectiveness of this method. The running cart type inverted pendulum produced by the servo techno company, which chiefly produces and sells servo motors and their drivers, was improved as a downsized device simulating an overhead traveling crane after the pendulum was turned downward. The Q8 real-time controller board of the PID Co., which chiefly imports the control experiment devices and sells them, was built into a personal computer, and system control software was constructed with Simulink Real-Time (Mathworks Co. Ltd.) under multiple operating systems (OSs). More specifically, Windows was used as the general OS, while Wincon was used as the realtime OS. Fig. 8 shows the control system configuration, while the major parameters of the experimental devices are shown in Table 3.

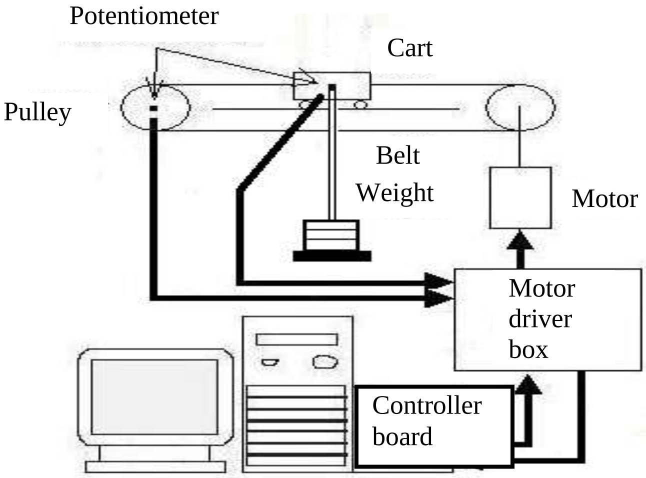

Fig. 8 Control system configuration

The voltage output from the controller board shown in Fig. 8 is provided to the motor in order to rotate the belt and pulley. The potentiometer that measures the position of the cart was installed at the center of the pulley, while the potentiometer that measures the sway angle was installed at the center of the cart. The signals of these two potentiometers were sent to the controller board through the motor driver box. The controller board then sent the motor control signal to the motor driver box.

TABLE 4 EXPERIMENTAL CONDITIONS

Rail Length L [m]	0.4
Cart Mass M [kg]	0.17
Pendulum Length l_g [m]	0.3, 0.5, 0.7
Weight Mass m [kg]	0–1.05

The distance that the cart could be moved was 0.4 [m] at maximum, as shown in Table 4. The pendulum length factor was prepared for three values (0.3, 0.5, and 0.7 [m]).

B. Experiment 1

The result obtained when a correct reference model was used in the simulation is shown in the section above. Next, changes in the cart position and/or hanging load sway angle were examined by an experiment that simulates an improper system operation by using an incorrect reference model in order to examine the model's robustness to the error margin of the reference model. Accordingly, it was assumed that l_g = 0.5 [m] in an actual experimental device and that l_g = 0.5 [m] and l_g = 0.3 [m] in the reference model. They were then compared. As a note, in the initial conditions of the experiment as well as the Section 4 simulation, it is important to ensure that the position of the cart is 0 [m], and that the hanging load sway is in the state of almost 0 [rad] at the starting point. The cart position is as shown in Fig. 9 and the hanging load sway angle is as shown in Fig. 10.

From Fig. 7, it was confirmed that even if an incorrect reference model was used, the cart could reach the target position without overshoot or undershoot. However, from Fig. 10, it was understood that if the wrong reference model was used, the hanging load sway angle could not be controlled at the same velocity both during the movement period and after the cart stops.

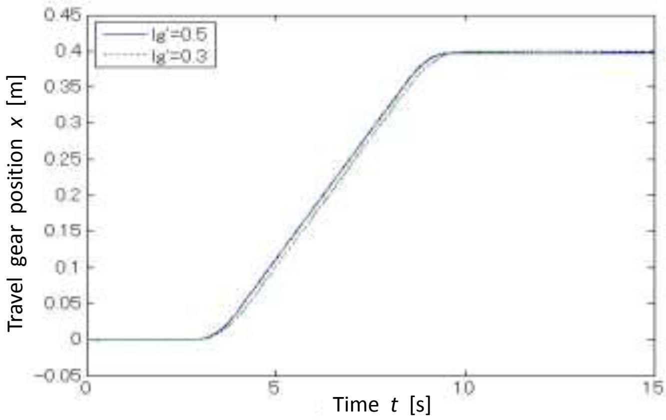

Fig. 9 Difference of position for reference model change

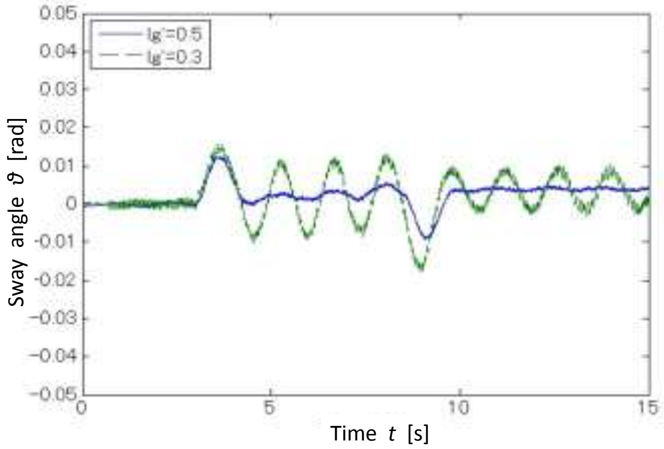

Fig. 10 Difference of sway angle for reference model change

VII. DISCUSSION AND CONSIDERATIONS

A notable feature of this technique is that it does not require sway angle information. This is a major merit in control techniques because the sway angle of actual cranes cannot be measured easily. Additionally, as shown in the previous section, another significant feature of the control system is its ability to adapt to changes in the length of the pendulum and its robustness in response to changes in the weights of the hanging load. The former depends on the reference model, and the latter depends on the integral term of the simple adaptive mechanism.

In addition, it is possible to adapt the model to various system configurations because there is no need to derive a complex nonlinear function I/O, as is normally required for previous adaptive mechanisms. Instead, by adopting a simple adaptive mechanism based on PID, the proposed system can be expected to achieve significant developments.

From the above simulations, it can be concluded that the mass range that can be used to control hanging load sway could be expanded if the PID parameters could be scheduled in the gain for the targeted value change, and if those parameters could be decided automatically. Furthermore, if additional robustness improvements could be achieved, as expected, this proposal technique may also be applicable to the sway control of persons and loads in passenger and freight vehicles other than overhead travel cranes.

From the results of the above experiments, it can be said that the robustness of the position control is high, but the sway suppression control is low for the model error of the reference model. We believe that the reason for the former is the effect of the integral control that is installed in parallel with the adaptive mechanisms for velocity and position, and that the latter is because none of the adaptive mechanism information, including integral and sway angle information, is used for sway suppression control. Therefore, it appears that the reference model error could not be utilized for anti-sway control, even though it can be used for cart position control.

VIII. CONCLUSION

In this paper, an important reference model was proposed that controls speed instruction and operation time in a way that minimizes hanging load sway angle and manages a trapezoidal-type speed pattern for a ceiling crane model without changing the pendulum length while the cart is moving.

The model reference simple adaptive control system (MRSACS) and property of robustness (PR) technique was then demonstrated by following the ceiling crane models output to the reference model output in order to observe its ability to absorb the interference disturbance to the cart imposed by the cart frictional force and the hanging load sway angle.

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Makoto Katoh was born in Kobe, Japan, on 1953. He earned Bachelor of Engineering, Master of Engineering, and Doctor of Engineering degrees from Osaka University, Japan in 1976, 1978, and 1981, respectively. His major fields of study were control engineering, intelligent systems, and mechatronics.

He worked at Osaka University as an assistant professor from 1981 to 1984, Mitsubishi Heavy Industries Co. Ltd. (MHI) as an engineer from 1984 to 1996, Toin Yokohama University as an associate professor from 1996 to 2000, and Osaka Institute of Technology as a professor from 2000 to present. His major books are Basic Theory of Systems Control (Tokyo, Corona, 2014), Robust Control Theory and Applications (INTECK, 2011), and Standard Automatic Control (Tokyo, Zikkyo, 1999). His recently published English articles are "Trial of Discrete Values Control by a Tracked Model for Wing Sail Eco-ship" (JMMA, 2013), "Safe Dynamics and Control of a Rotor Vehicle on a Track" (JMMA, 2013), "A Plan for Fuzzy Passage Decision of Elevators using Equivalent Person Number" (Int. Journal of Automation and Control Engineering: IJACE, 2012), and "No Overshoot Step Response of an Integral Controller with a Simple Robust Lead Inverse Compensator to Improve Safety of Vehicles" (IJACE, 2016). His current research interests are Table Based Systems with zero, hard servo and process control of non-minimum phase systems, although previous research interests were general control theory and applications, and intelligent systems.

Prof. Katoh is a member of IEEE, SICE, and JSME. He received the Encouragement Prize from SICE in 1980, the Factory Award from the MHI Nagasaki Factory in 1986, and the Headquarters Award from MHI in 1992.

Atsushi Fujiwara was born in Japan on 1983. He earned the Bachelor of Engineering and Master of Engineering degrees from Osaka Institute of Technology, Japan in 2006 and 2008, respectively. His major fields of study were stabilized control of a moving type invert pendulum and anti-sway control of an overhead crane.

He joined the Shiga Office of Daifuku Co. Ltd. in 2008.