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Fuzzy-tuned PID Anti-swing Control of Automatic Gantry Crane

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Abstract: Anti-swing control is a well-known term in gantry crane control. It is designed to move the payload of gantry crane as fast as possible while the payload swing angle should be kept as small as possible at the final position. A number of studies have proposed anti-swing control using the well-known proportional, integral, derivative (PID) control method. However, PID controllers cannot always effectively control systems with changing parameters. Some studies have also proposed intelligent-based control including fuzzy control. However, the designers often have to face the problem of tuning many parameters during the design to obtain optimum performance. Thus, a lot of effort has to be taken in the design stage. In this paper Fuzzy-tuned PID controller design for anti-swing gantry crane control is presented. The objective is to design a practical anti-swing control which is simple in the design and also robust. The proposed Fuzzy-tuned PID utilizes fuzzy system as PID gain tuners to achieve robust performance to parameters' variations in the gantry crane. A complex dynamic analysis of the system is not needed. PID controller is firstly optimized in MATLAB using a rough model dynamic of the system which is identified by conducting a simple open-loop experiment. Then, the PID gains are used to guide the range of the fuzzy outputs of the Fuzzy-tuned PID controllers. The experimental results show that the proposed anti-swing controller has satisfactory performance. In addition, the proposed method is straightforward in the design.

Keywords: Anti-swing control, gantry crane, fuzzy-tuned PID.

1. INTRODUCTION

Gantry cranes are widely used in industry for transporting heavy loads and hazardous materials in shipyards, factories, nuclear installations, and high building constructions. The crane should move the load as fast as possible without causing any excessive movement at the desired position. However, most of the common gantry crane results in a swing or sway motion when payload is suddenly stopped after a fast motion (Omar, 2003). The swing motion can be reduced but it will be time-consuming i.e. reducing the productivity. Moreover, the gantry crane needs a skilful operator to control it manually, based on the operator's experience, in

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Figures 2, 5, 6 appear in color online: http://jvc.sagepub.com

order to stop the swing immediately at the accurate position. Furthermore, to unload, the operator has to prevent the load stops from swaying. Failure to control the crane may also cause accidents and harm people and surroundings.

Various attempts of anti-swing control for automatic gantry cranes have been proposed. Singhose et al. (1997), Park et al. (2000) and Garrido et al. (2008) adopted input shaping technique, which is an open loop approach. However, these methods could not successfully damp the residual swing angle. Gupta and Bhowal (2004) also presented a simplified open-loop anti-swing technique. They implemented this technique based on velocity control during motion. This is an open-loop approach which is sensitive to parameters' change of the system and disturbances. On the other hand, anti-swing feedback controls which are well known to be less sensitive to parameter variations and disturbances have also been proposed.

Anti-swing control of gantry or overhead cranes has attracted considerable attention due to the underactuation property in the payload swing. Sridokbuap et al. (2007) proposed I-PD+PD control for an overhead crane using characteristic ratio assignment (CRA). However, it is natural that PID control is usually not robust to large parameter variations. It is also assumed that the dynamic model of the system is known. Chang et al. (2005) combined PID and Fuzzy compensation to control trolley position and swing motion ofan overhead crane. However, the effect of parameter variations (i.e. payload cable length) was not considered. Matsuo et al. (2004) used PID+Q based controller for anti-swing control. The study focused on the payload swing suppression, but did not concentrate on error position of the trolley.

Some researches have also applied nonlinear control theory to analyze the properties of the crane system (Fang et al., 2003). These approaches are usually too complex for practical use and involve rigorous mathematical analysis. Hua and Shine (2007) proposed that adaptive coupling control with a nonlinear control scheme incorporating parameter adaptive mechanism be devised to ensure the overall closed-loop system stability.

Furthermore, many researchers have applied fuzzy logic to control overhead cranes because it can mimic human behavior accurately (Renno et al., 2004). Although they do not have an apparent structure of PID controllers, fuzzy logic controllers may be considered as nonlinear PID controllers whose parameters can be determined online based on the error signal and its time derivatives. As the nature of a fuzzy control system requires expert knowledge to tune the parameters, which is often difficult and time-consuming, instead they sometimes adopted fuzzy logic and combined with their proposed techniques. Benhidjeb and Gissinger (1995) discussed the comparison of a fuzzy logic control system and Linear Quadratic Gaussian control (LQG) for an overhead crane. Wahyudi et al. (2007) proposed fuzzy control combined with a practical control approach. Furthermore, Liu and Zhoa (2005) proposed an adaptive sliding mode control method with fuzzy tuning of slope of sliding surface for a 2-dimension overhead crane. In Li et al. (2005), self-adaptive fuzzy PID is presented combined with a feedforward anti-swing scheme for bridge crane control. Only fixed cable length is considered, however. The fuzzy rules are also not concise. In the other work, Trabia et al. (2006) have proposed general anti-swing fuzzy control for crane system. In their system, inverse dynamics is still needed. As far as the authors are concerned, there is no general procedure for a robust and practical anti-swing control of gantry crane found in the literature.

This paper discusses the design of Fuzzy-tuned PID anti-swing control of an automatic gantry crane system. The objective of this work is to design a practical anti-swing control which is simple in the design and also robust. The time-consuming process of system modeling and analysis should also be eliminated. One of the reasons to use Fuzzy is because it is basically a model independent approach. Only information about error and error rates are used. Thus, it is also possible to develop the proposed control method further for self-tuning fuzzy control. At the end, a robust and practical anti-swing control of gantry crane can be achieved. In practice, only a rough model of the system is needed, which is straightforward to identify experimentally. The developed dynamic mathematical model here is basically used for simulation study.

The proposed Fuzzy-tuned PID control has simple structure of PID control. Instead of having fixed PID gains, the gains are determined directly by means of a Mamdani-fuzzy inference system. Weighting factors are also added to fuzzy output, which is directly related to the PID gains, so that the universe of discourse of the fuzzy system can be normalized. In addition, this weighting factor can also be tuned further by a self-tuning mechanism in order to develop a robust and practical anti-swing control using a model independent approach. Indeed, the proposed Fuzzy-PID method here can be classified as fuzzy self-tuning PID or a class of adaptive fuzzy control (Mudi and Pal, 1999) although the scheme has not been found before in any of the literature.

Experimental study is also conducted to evaluate the effectiveness of the proposed method. Previously, a rough model of the system was identified by open-loop experiment. Then, PID controller was optimized using Simulink Response Optimization in MATLAB according to the identified rough model. Thus, the purpose of PID controller is twofold here. First, it is used as a comparison with the proposed Fuzzy-tuned PID controller. Second, the PID gains are used as a guideline to the fuzzy output in the Fuzzy-tuned PID controllers. The result shows that the proposed Fuzzy-tuned PID anti-swing controllers outperforms the PID anti-swing controller. The proposed controller has the obvious advantage of robustness to parameters' variations for anti-swing control.

2. CRANE DYNAMIC

The gantry crane system is an underactuated system where the number of inputs is less than the number of outputs. When the input signal is given to the actuator, the trolley starts to accelerate whilst causing a swing of payload hanging on a flexible cable. Nonlinear dynamic model of 2-D gantry crane prototype is derived using Lagrange equations. Notice that a 3-D gantry crane can be decoupled into its 2-D one. Figure 1 shows the diagram of gantry crane mechanism where $m_1, m_2, l, x, \theta, B$ and F are payload mass, trolley mass, cable length, horizontal position of trolley, swing angle, damping friction and driving force respectively. The generalized coordinates are trolley position and swing angle. The equation of motion of gantry crane system is obtained. See Table 8 for the complete list of symbols.

$$(m_1 + m_2)\ddot{x} + m_1 l\ddot{\theta}\cos\theta - m_1 l\dot{\theta}^2 \sin\theta + B\dot{x} = F \tag{1}$$

$$m_1 l^2\ddot{\theta} + m_1 l\ddot{x}\cos\theta + m_1 gl\sin\theta = 0. \tag{2}$$

The trolley is driven by a DC motor. The input voltage to the motor is limited by -5 to $+5$ volts. The dynamic of the DC motor circuit is also included.

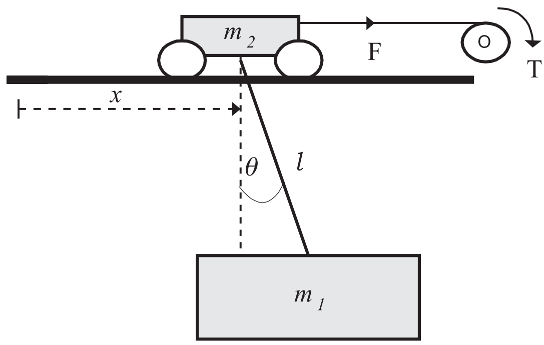

Figure 1. Trolley crane model.

$$V = Ri + L\dot{i} + K_e\dot{\theta}_m \tag{3}$$

$$T_m = K_t i. \tag{4}$$

When L is neglected, equation (3) becomes:

$$V = Ri + K_e\dot{\theta}_m. \tag{5}$$

Applying Newton's second law of motion to the motor shaft, an equation is obtained as:

$$J_m\ddot{\theta}_m = T_m - \frac{T_L}{r}. \tag{6}$$

Since moment inertia of motor, J_m, is very small, then equation (6) can be written as:

$$T_m = \frac{T_L}{r} \tag{7}$$

with V, R, L, i, T_m, T_L, K_t, K_e, θ_m and r are respectively input voltage, resistance, inductance, armature current, motor torque, load torque, torque constant, electric constant, rotor angle position, and gear ratio. In addition, some equations related to rotational horizontal motions are (r_p is radius of pulley):

$$T_L = Fr_p \tag{8}$$

$$\theta_m = \frac{r}{r_p}x. \tag{9}$$

Finally, equations (3–9) are combined with equation (1) resulting in nonlinear mathematical model. It can be written as follows:

$$V = A_1\dot{x} + B_1\ddot{x} + C_1(\ddot{\theta}\cos\theta - \dot{\theta}^2\sin\theta) \tag{10}$$

where:

$$A_1 = \frac{K_e r}{r_p} + \frac{BRr_p}{K_t r} \quad C_1 = \frac{m_1 l R r_p}{K_t r} \quad B_1 = \frac{Rr_p}{K_t r}(m_1 + m_2).$$

In order to make simpler analysis in linear control theory, the nonlinear model can be linearized by assuming small θ during control ($\sin\theta \approx \theta$ and $\cos\theta = 1$). The linearized model of gantry crane dynamic is:

$$V = A_1\dot{x} + B_1\ddot{x} + C_1\ddot{\theta} \tag{11}$$

$$l\ddot{\theta} + \ddot{x} + g\theta = 0. \tag{12}$$

Equations (11) and (12) show the dynamic of this underactuated system in differential equations. In transfer functions, these can be written into two equations to respectively represent input output transfer as:

$$\frac{X(s)}{V(s)} = \frac{ls^2 + g}{s((B_1 l - C_1)s^3 + A_1 ls^2 + B_1 gs + A_1 g)} \tag{13}$$

$$\frac{\theta(s)}{V(s)} = \frac{-s}{(B_1 l - C_1)s^3 + A_1 ls^2 + B_1 gs + A_1 g}. \tag{14}$$

Furthermore, for practicality, if a trolley position is identified experimentally without considering the swing ($l \approx 0$ and payload is attached directly to the cart/trolley), equation (13) is reduced to:

$$\frac{X(s)}{V(s)} = \frac{1/B_1}{s\,(s + A_1/B_1)} \approx \frac{K}{s(s+\tau)} \tag{15}$$

where K and τ are unknown constant parameters which will be identified experimentally. The detail of identification procedure is discussed in the next section. While the swing angle model is derived from equation (12) in which all parameters (l and g) are known:

$$\frac{\theta(s)}{X(s)} = \frac{-s^2}{ls^2 + g}. \tag{16}$$

Finally, the rough model of the gantry crane system with one input and two outputs is obtained by equations (15) and (17) as:

$$\frac{\theta(s)}{V(s)} = \frac{K}{s(s+\tau)}\frac{-s^2}{(ls^2+g)}. \tag{17}$$

Figure 2. Lab-scale gantry crane prototype.

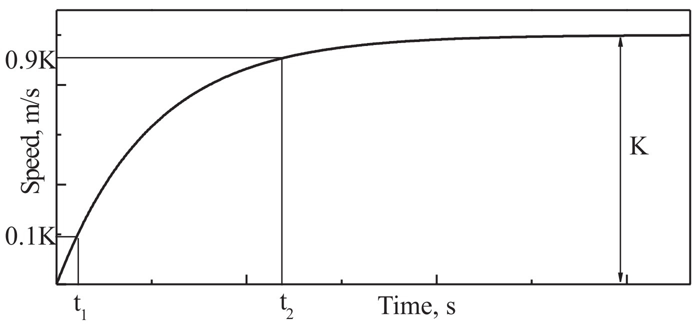

Figure 3. Assumed speed response of trolley crane to a step input.

3. IDENTIFICATION OF UNKNOWN PARAMETERS

To identify the parameters of K and τ, the open loop speed responses of the trolley position (for maximum and minimum payload mass) to step input are recorded. The lab scale gantry crane prototype is shown in Figure 2. In this study, parameter identification is done experimentally following this explanation. The trolley's speed response (V_{cart}) to a unit step input is as shown in Figure 3 and is written as:

$$V_{cart}(s) = \frac{K}{s+\tau}\frac{1}{s} \tag{18}$$

$$V_{cart}(t) = K(1 - e^{\tau t}). \tag{19}$$

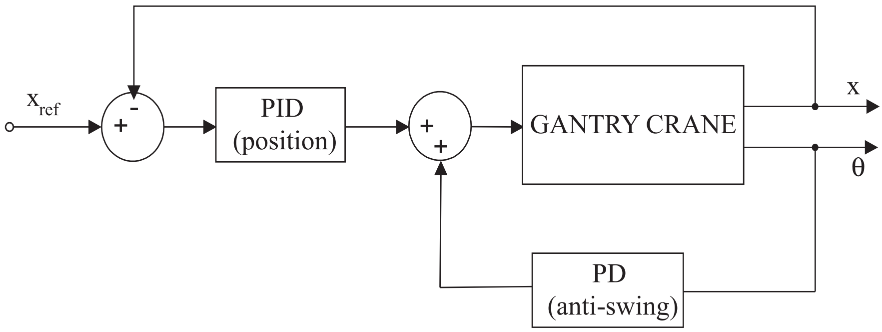

Figure 4. Simplified diagram of anti-swing control.

Table 1. Identified parameter.

K	τ
2.66	11.1

From the response above, K can be measured as steady state speed. Then, the following equations can be derived to solve for unknown constant τ:

$$0.1K = K(1 - e^{-\tau t_1}) \tag{20}$$

$$0.9K = K(1 - e^{-\tau t_2}) \tag{21}$$

$$\tau = \frac{2.2}{t_2 - t_1}. \tag{22}$$

By conducting open-loop experiment to record trolley speed response, the unknown parameters are found for payload mass of 2.5 kg. The values are listed in Table 1.

4. PID ANTI-SWING CONTROL

The main purpose of anti-swing control is to transfer the load as fast as possible without causing any excessive swing at the end position. The automatic gantry crane proposed by the researchers commonly uses two sets of controllers to control both trolley position and swing angle of the payload as shown in Figure 4. PID controller is used for trolley positioning control and PD is used for swing control. The close loop transfer functions can be calculated. The overall close loop characteristic polynomial is found as:

$$K_{cl} = s^5 + a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0 \tag{23}$$

where:

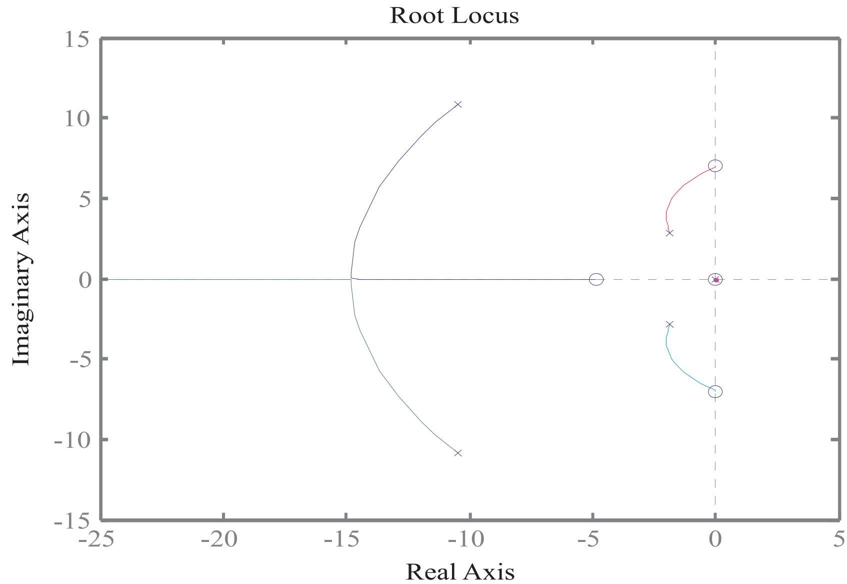

Figure 5. Root locus for minimum cable length ($l = 0.2$ m).

$$a_4 = \tau + \frac{KK_{ds}}{l} + KK_d \quad a_3 = \frac{g}{l} + \frac{KK_{ps}}{l} + KK_p$$

$$a_2 = \frac{g\tau}{l} + KK_i + \frac{KK_d g}{l} \quad a_1 = \frac{KK_p g}{l} \quad a_0 = \frac{KK_i g}{l}.$$

To obtain the PID+PD gains for anti-swing controller, optimization method is used in MATLAB called Simulink Response Optimization Toolbox. It can optimize PID+PD gains from the closed-loop system. The design requirements are expressed in terms of rise time, settling time, overshoot, including the input signal saturation.

Nonetheless, in this optimization, model-following reference control approach is used. The plant is driven so that the trolley motion has to follow model reference. The model reference is taken from 3rd order standard ITAE which is related to desired settling time. This model reference is defined as (Dorf and Bishop, 2007):

$$X_{ref} = \frac{\omega^3}{s^3 + 1.75\omega s^2 + 2.15\omega^2 s + 1.5\omega^3} \tag{24}$$

where ω is selected accordingly corresponding to the desired settling time of trolley positioning. The optimized PID gains are listed in Table 2. During the optimization, the maximum cable length is set ($l_{max} = 0.6$ m). However, the controller must be stable for the range of cable length variations ($l_{min} = 0.2$ m). Figures 5 and 6 show the roots locus of the close loop characteristic equations with the optimized PID gains for minimum and maximum cable length. It is sufficient to justify the stability of the controller.

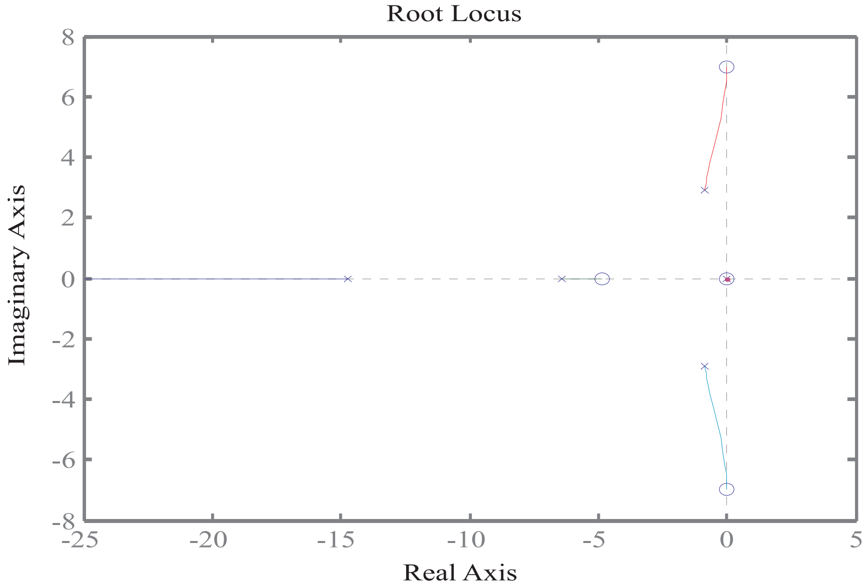

Figure 6. Root locus for maximum cable length ($l = 0.6$ m).

Table 2. Optimized PID gains.

K_p	K_i	K_d	K_{ps}	K_{ds}
2.0	0.7	4.1	16.1	0.2

5. FUZZY-TUNED PID CONTROL DESIGN

The Fuzzy-tuned PID control is then designed. To control the position of the trolley crane, Fuzzy-tuned PID is proposed, whereas anti-swing control is performed by Fuzzy-tuned PD control. The PID gains obtained above are used as a guideline for fuzzy output of the proposed Fuzzy-tuned PID controllers. The controller structure is shown in Figure 7.

Initially, Mamdani fuzzy inference system is designed as fuzzy tuner. It has error and error rate as inputs and the tuned gain as the output. The fuzzy membership for the output is singleton membership function. The fuzzy system for the input has five Gaussian membership functions distributed equivalently over the determined range (universe of discourse).

The range of each fuzzy membership is normalized in order to have simple structure of fuzzy system. The range is set up to from –1 to 1 for fuzzy input and from 0 to 1 for fuzzy output. The fuzzy membership functions for error and error rate are as shown in Figure 8. Figure 9 shows the fuzzy membership functions for fuzzy tuner output. Here, NB, NS, Z, PS and PB are assigned to negative big, negative small, zero, positive small and positive big respectively in the fuzzy rule. The if-then fuzzy rules are simply based on Macvicar-Whelan Matrix (Visioli and Finzi, 1998). This is shown in Table 3. Then, the fuzzy tuner surface can be seen in Figure 10, which is flat near the extremes of the range.

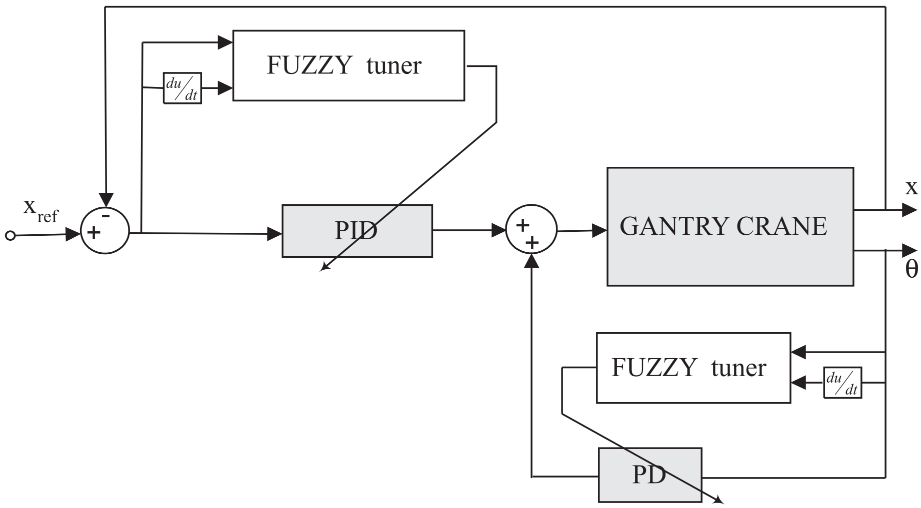

Figure 7. Fuzzy-tuned PID control scheme for anti-swing control.

Table 3. If-then rule for fuzzy tuner.

		ERROR RATE				
		NB	**NS**	**Z**	**PS**	**PB**
	NB	Z	Z	VS	SM	M
	NS	Z	VS	SM	M	BM
ERROR	**Z**	VS	SM	M	BM	VB
	PS	SM	M	BM	VB	B
	PS	M	BM	VB	B	B

The performance of the controller is determined by adding weighting factors on normalized fuzzy output. These weighting/scaling factors are proportional respectively to K_p, K_i, K_d, K_{ps} and K_{ds} which are the proportional, integral and derivative gains for position control and proportional, derivative gains for anti-swing control. The weighting or scaling factors on fuzzy tuner output are selected as αK_x, where x is index for each PID gains. In this paper, α as a multiplier to the PID gains shown in Table 2. The value of α is 1.25 obtained by trial as a compromise. In future work, different values of scaling factor may be proposed for positioning and swing controller. This value can be tuned on-the-fly over the variation of 1 to 2 using self-tuning mechanism providing only information about error and error rate.

Therefore, the purpose of PID controller is twofold. First, it is used as comparison with the proposed Fuzzy-tuned PID controller here. Second, the PID gains are used as a guideline for weighting factors in the output of the proposed Fuzzy-tuned PID controllers to improve the robustness of the original PID controller.

The proposed Fuzzy-tuned PID controller scheme can be seen in Figure 11. This scheme is also applicable for Fuzzy-tuned PD controller with the corresponding weighting factors of αK_{ps} and αK_{ds} respectively. For simplicity, the same structure of fuzzy tuner in terms of

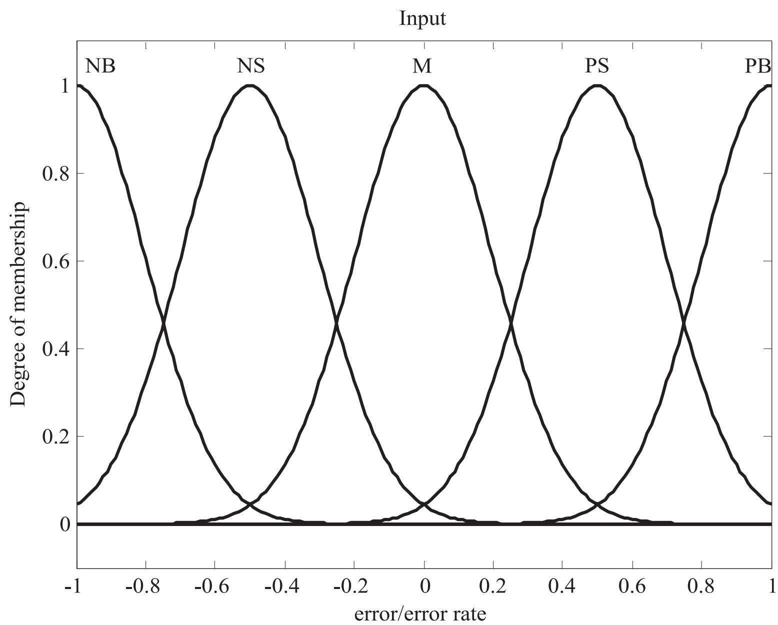

Figure 8. Fuzzy Gaussian membership functions for input.

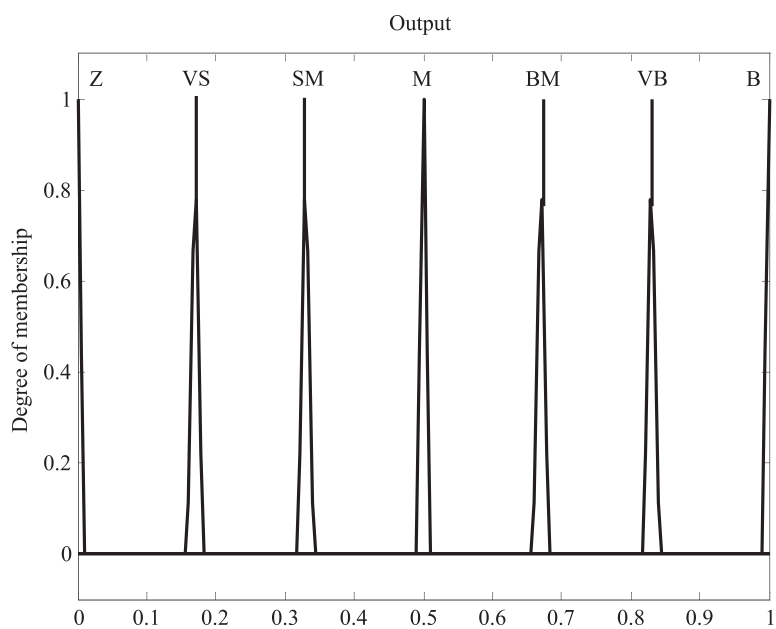

Figure 9. Fuzzy singleton membership functions for output.

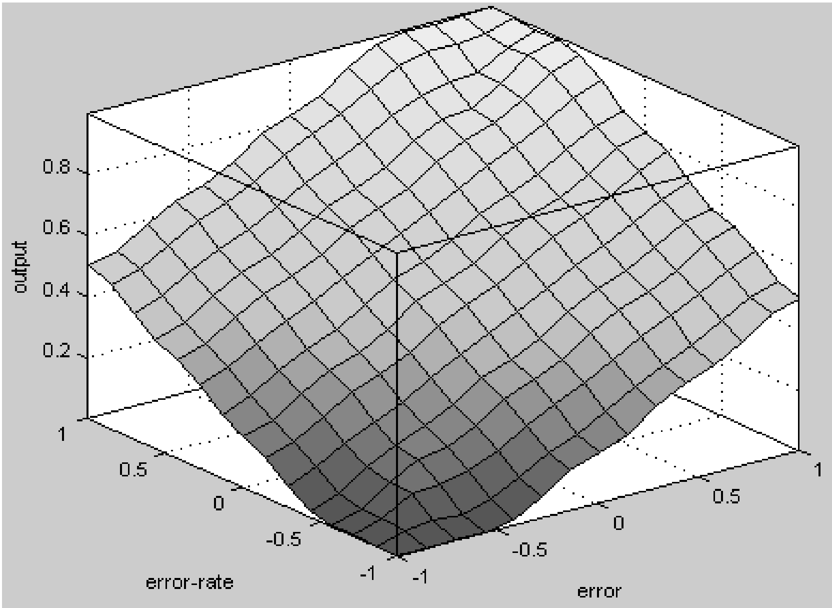

Figure 10. Fuzzy tuner surface.

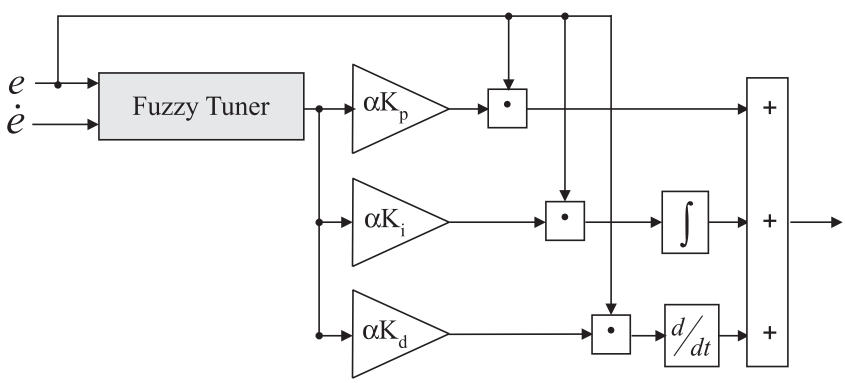

Figure 11. Fuzzy-tuned PID controller scheme. (a) Trolley displacement.(b) Swing angle.

membership functions, fuzzy rule, and inference system is also used for Fuzzy-tuned PD controller.

6. EXPERIMENTAL RESULTS

The proposed control system is expected to improve the robustness of the original anti-swing PID controller to parameter variations (e.g. hoisting cable length, payload mass) which

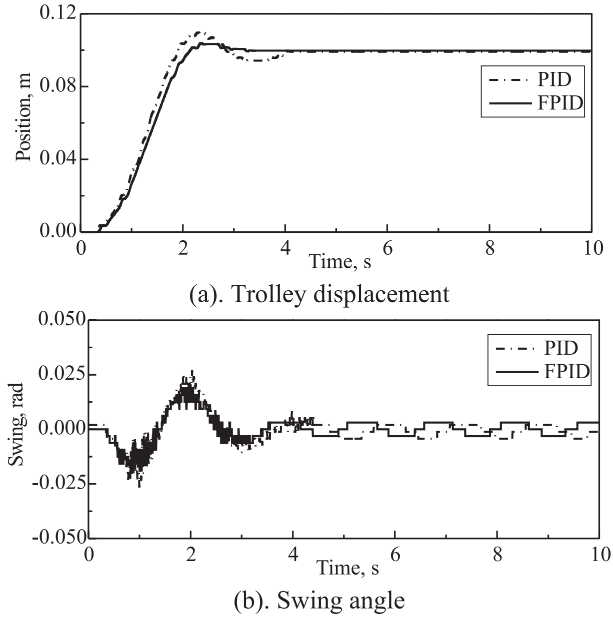

Figure 12. Responses for 0.1 m displacement; $l = 0.6$ m, $m_1 = 2.5$ kg. (a) Trolley displacement. (b) Swing angle.

represents the real conditions. The proposed Fuzzy-tuned PID controller is evaluated through experiment. The experiment is a lab-scale prototype gantry crane, the one with the horizontal track about 0.8 m. The nominal values of cable length and payload mass used in the control design are $l = 0.6$ m and $m_1 = 2.5$ kg.

Figures 12 and 13 show the responses of the system with nominal values of cable length and payload mass (same as in PID optimization) for short and long displacement obtained in experiment. The proposed Fuzzy-PID controller has a smoother response compared with the original PID one. This happens for both cases, long and short displacement. It also has smaller overshoot and settling time.

The robustness to cable length and payload mass variations are also shown in Figures 14–16. It is clear that the proposed control method has advantages over PID controller for anti-swing gantry crane control. It is more robust compared to a PID controller where the lesser swing angle oscillations are found. The numerical value of the overall comparison between the proposed Fuzzy-tuned PID and the PID anti-swing controller is provided in Tables 4–7.

Furthermore, an external disturbance to the swinging load is also introduced to see the robustness to external disturbance. A moderate punch to the payload in the middle of the motion is given as an external disturbance. Figure 17 shows the response comparison for the condition without and with disturbance. It can be seen that the anti-swing control can handle the external disturbance well as the swing goes to zero at the end of the motion without deteriorating the trolley positioning response.

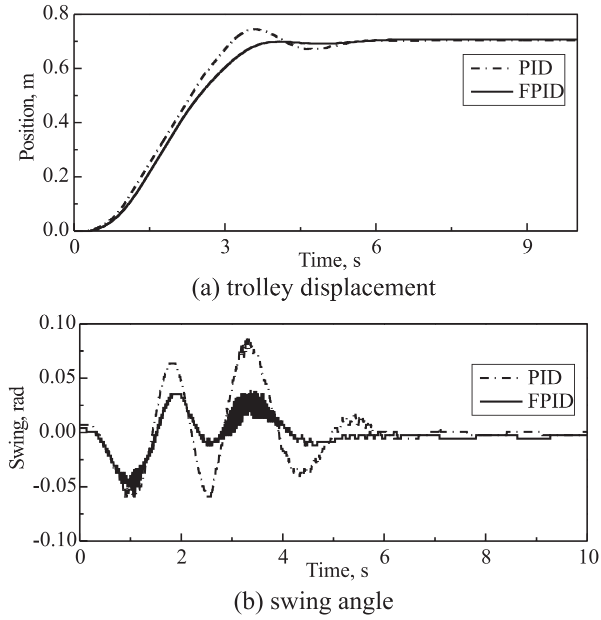

Figure 13. Responses for 0.7 m displacement; $l = 0.6$ m, $m_1 = 2.5$ kg. (a) Trolley displacement. (b) Swing angle.

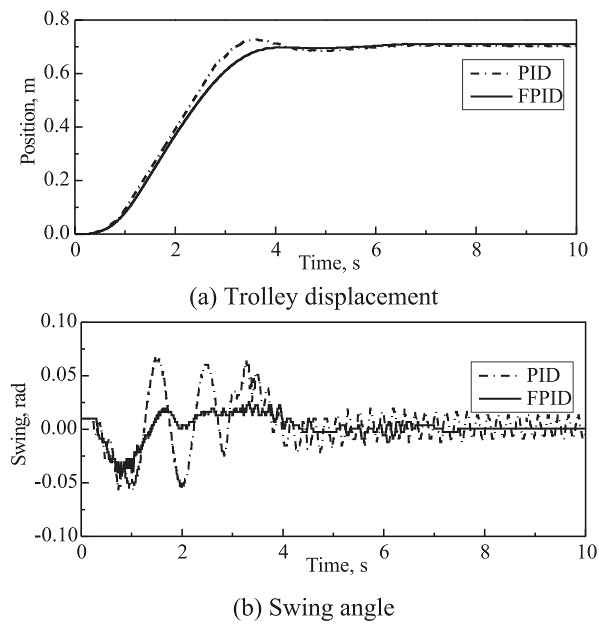

Figure 14. Responses for 0.7 m displacement; $l = 0.2$ m, $m_1 = 0.25$ kg. (a) Trolley displacement. (b) Swing angle.

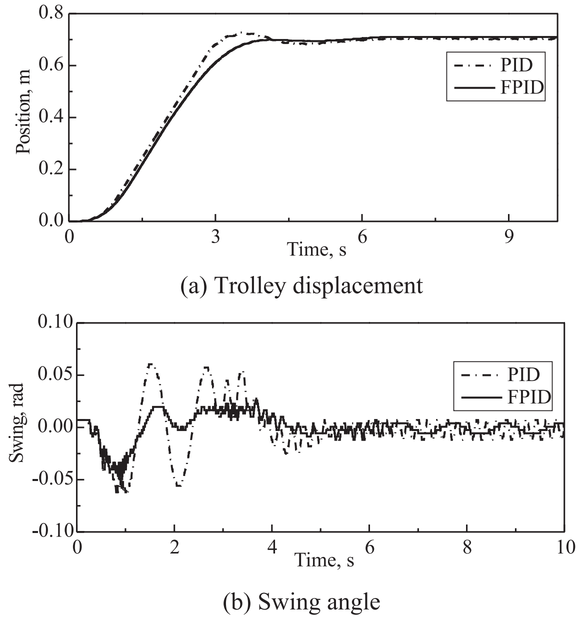

Figure 15. Responses for 0.7 m displacement; $l = 0.2$ m, $m_1 = 2.5$ kg. (a) Trolley displacement. (b) Swing angle.

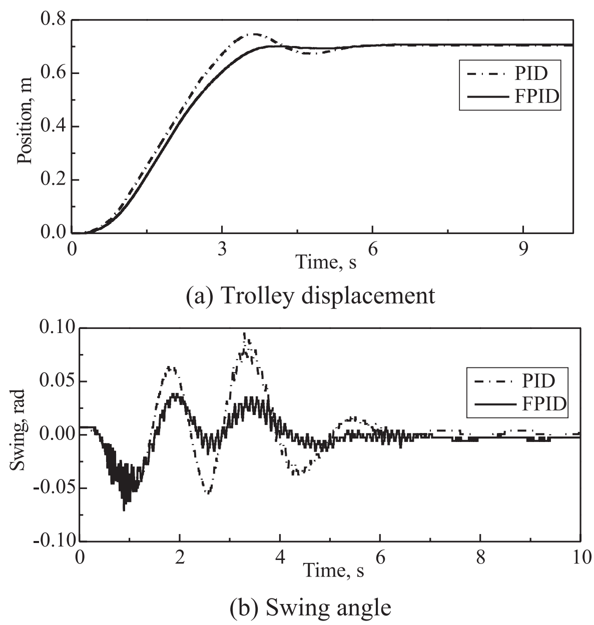

Figure 16. Responses for 0.7 m displacement; $l = 0.6$ m, $m_1 = 0.25$ kg. (a) Trolley displacement. (b) Swing angle.

Table 4. Positioning performance for 0.1 m displacement.

l, m	m_1, kg	PID			FPID		
		OS, %	Ts, s (2%)	Error, m	OS, %	Ts, s (2%)	Error, m
0.2	0.25	6.1	4.6	–	0.7	2.3	0
0.6		8.7	5.8	0.001	1.8	3.2	0.002
0.2	2.5	7.1	9.9	–	0.9	2.2	0
0.6		9.7	4.0	0.001	3.8	2.7	0

Table 5. Anti-swing performance for 0.1 m displacement.

l, m	m_1, kg	PID		FPID	
		Max., rad	Ts, s (0.01rad)	Max., rad	Ts, s (0.01rad)
0.2	0.25	0.025	> 10	0.022	1.9
0.6		0.035	6.6	0.022	2.4
0.2	2.5	0.028	> 10	0.028	2.2
0.6		0.028	2.4	0.019	2.3

Table 6. Positioning performance for 0.7 m displacement.

l, m	m_1, kg	PID			FPID		
		OS, %	Ts, s (2%)	Error, m	OS, %	Ts, s (2%)	Error, m
0.2	0.25	3.9	5.1	–	1.5	3.6	0.010
0.6		6.6	5.2	0.004	1.4	3.6	0.008
0.2	2.5	4.0	5.2	–	1.5	3.7	0.010
0.6		6.4	5.2	0.003	1.0	3.6	0.007

Table 7. Anti-swing performance for 0.7 m displacement.

l, m	m_1, kg	PID		FPID	
		Max., rad	Ts, s (0.02rad)	Max., rad	Ts, s (0.01rad)
0.2	0.25	0.063	9.9	0.050	6.2
0.6		0.095	5.9	0.072	6.4
0.2	2.5	0.063	9.9	0.046	6.1
0.6		0.088	5.8	0.056	4.0

7. CONCLUSIONS

The proposed Fuzzy-tuned PID controller has an advantage of robustness compared with original PID control. With its simple design process and configuration, it looks suitable as a practical control method where the controller is basically designed in a model independent approach.

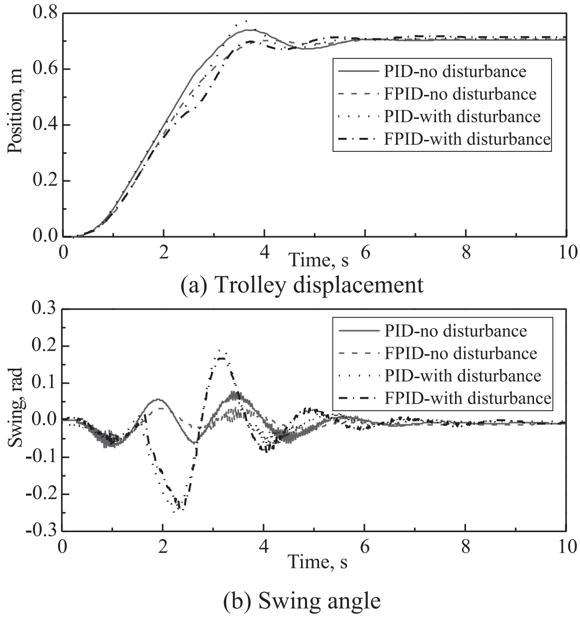

Figure 17. Responses with and without disturbance ($l = 0.6$ m, $m_1 = 1.0$ kg).

Table 8. List of symbols.

Symbol	Definition
m_1	Payload mass
m_2	Trolley mass
l	Cable length
x	Horizontal position of trolley
θ	Swing angle
B	Damping friction
F	Driving force
V	Input voltage
R	Resistance of DC motor
L	Inductance of DC motor
i	Armature current
T_m	Motor torque
T_L	Load torque
K_t	Torque constant
K_e	Electric constant
θ_m	Rotor angle position
r	Gear ratio
r_p	Radius of pulley
K	Unknown constant in numerator

Table 8. List of symbols. *(Continued)*

Symbol	Definition
τ	Unknown constant in denominator
V_{cart}	Trolley speed
ω	Natural frequency
K_p	Proportional gain in position control
K_i	Integral gain in position control
K_d	Derivative gain in position control
K_{ps}	Proportional gain in swing control
K_{ds}	Derivative gain in swing control
α	Scaling/weighting factor in fuzzy-tuned PID

The stability of the proposed Fuzzy-PID controller is guaranteed via stability of the designed PID controller because the outputs of the proposed fuzzy are limited by the PID gains obtained by optimization as a part of the controller design.

Lastly, the scaling factor α is fixed for both position and swing controllers. The value $\alpha = 1.25$ is somehow selected by trial considering all those possible maneuvers above (under variations of displacement, payload mass and cable length). Instead of a fixed value, α can also be tuned on-the-fly using fuzzy self-tuning mechanism, which is a model independent approach. This will be done in the future work towards designing a practical anti-swing control for automatic gantry crane with high robustness.

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